Principles of Heating Ventilating and Air Conditioning

8th Edition

Based on the 2017 *ASHRAE Handbook—Fundamentals*

Ronald H. Howell

ASHRAE®

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PREFACE

Principles of Heating, Ventilating, and Air Conditioning, a textbook based on the 2017 *ASHRAE Handbook—Fundamentals*, should provide an attractive text for air-conditioning courses at engineering colleges and technical institutes. The text has been developed to give broad and current coverage of the heating, ventilation, and air-conditioning field when combined with the 2017 *ASHRAE Handbook—Fundamentals*.

The book should prove most suitable as a textbook and subsequent reference book for (a) undergraduate engineering courses in the general field of HVAC, (b) similar courses at technical institutes, (c) continuing education and refresher short courses for engineers, and (d) adult education courses for nonengineers. It contains more material than can normally be covered in a one-semester course. However, several different single-semester or shorter courses can be easily planned by merely eliminating the chapters and/or parts that are least applicable to the objectives of the particular course. This text will also readily aid in self-instruction of the 2017 *ASHRAE Handbook—Fundamentals* by engineers wishing to develop their competence in the HVAC&R field.

Although numerous references are made to the other ASHRAE Handbook volumes, sufficient material has been included from these to make this text complete enough for various courses in the HVAC&R field. The material covered for various audiences in regular university courses, technical institute courses, and short courses can and will vary greatly. This textbook needed to be complete to satisfy all of these anticipated uses and needs. Toward this end, the following major sections are included:

Part I	General Concepts, Chapters 1–10
Part II	Air-Conditioning Systems, Chapters 11–16
Part III	HVAC&R Equipment, Chapters 17–20

Although the 2017 *ASHRAE Handbook—Fundamentals* is published in an SI edition, which uses international units, and an inch-pound (I-P) edition, this single version of *Principles of Heating, Ventilating, and Air Conditioning* is designed to serve the I-P edition with some SI interspersed throughout.

There are several significant changes in this edition. Chapter 4 has new values for climatic design information. Chapter 7 has been extensively revised with new design data. These changes make *Principles* compatible with the 2017 *ASHRAE Handbook—Fundamentals*. In addition, the chapters on system design and equipment have been significantly revised to reflect recent changes and concepts in contemporary heating and air-conditioning system practices. Also, the *Solutions Manual* has been extensively edited.

A particular point of confusion must be pointed out. Because this book was developed to be used with the ASHRAE Handbook's *Fundamentals* volume, a number of tables and figures have been reproduced in the original form, complete with references to material elsewhere in *Fundamentals* (not in this book). Thus, if the subheading in the table or figure indicates that it is a *Fundamentals* table or figure, then all references to other locations, equations, tables, etc., refer to those in *Fundamentals*, not in *Principles*.

Dr. Harry Sauer, Jr., one of the co-authors of this textbook, passed away in June 2008. Likewise, William J. Coad was also a co-author of this textbook and passed away in August 2014. Both Dr. Sauer and Mr. Coad made significant contributions to the book.

September 2017 Ronald H. Howell

Chapter 1

BACKGROUND

This chapter provides a brief background on the heating, ventilating, air-conditioning, and refrigeration (HVAC&R) field and industry, including the early history and some significant developments. An introduction to a few basic concepts is included along with suggestions for further reading.

1.1 Introduction

On the National Academy of Engineering's list of engineering achievements "that had the greatest impact on the quality of life in the 20th century," *air conditioning and refrigeration* came in tenth, indicating the great significance of this field in the world. With many people in the United States spending nearly 90% of their time indoors, it is hardly surprising that providing a comfortable and healthy indoor environment is a major factor in life today. In fact, over $33 billion of air-conditioning equipment was sold in the US during the year 2010 alone.

Air-conditioning systems usually provide year-round control of several air *conditions*, namely, temperature, humidity, cleanliness, and air motion. These systems may also be referred to as *environmental control systems*, although today they are usually called heating, ventilating, and air-conditioning (HVAC) systems.

The primary function of an HVAC system is either (1) the generation and maintenance of comfort for occupants in a conditioned space; or (2) the supplying of a set of environmental conditions (high temperature and high humidity, low temperature and high humidity, etc.) for a process or product within a space. Human comfort design conditions are quite different from conditions required in textile mills or for grain storage and vary with factors such as time of year and the activity and clothing levels of the occupants.

If improperly sized equipment or the wrong type of equipment is used, the desired environmental conditions usually will not be met. Furthermore, improperly selected and/or sized equipment normally requires excess power and/or energy and may have a higher initial cost. The design of an HVAC system includes calculation of the maximum heating and cooling loads for the spaces to be served, selection of the type of system to be used, calculation of piping and/or duct sizes, selection of the type and size of equipment (heat exchangers, boilers, chillers, fans, etc.), and a layout of the system, with cost, indoor air quality, and energy conservation being considered along the way. Some criteria to be considered are

- Temperature, humidity, and space pressure requirements
- Capacity requirements
- Equipment redundancy
- Spatial requirements
- First cost
- Operating cost
- Maintenance cost
- Reliability
- Flexibility
- Life-cycle cost analysis

The following details should be considered to properly design an air-conditioning system:

1. The location, elevation, and orientation of the structure so that the effects of the weather (wind, sun, and precipitation) on the building heating and cooling loads can be anticipated.
2. The building size (wall area, roof area, glass area, floor area, and so forth).
3. The building shape (L-shaped, A-shaped, rectangular, etc.), which influences equipment location, type of heating and cooling system used, and duct or piping locations.
4. The space use characteristics. Will there be different users (office, bank, school, dance studios, etc.) of the space from year to year? Will there be different concurrent requirements from the tenants? Will there be night setback of the temperature controller or intermittent use of the building's facilities?
5. The type of material (wood, masonry, metal, and so forth) used in the construction of the building. What is the expected quality of the construction?
6. The type of fenestration (light transmitting partition) used, its location in the building, and how it might be shaded. Is glass heat absorbing, reflective, colored, etc.?
7. The types of doors (sliding, swinging, revolving) and windows (sealed, wood or metal frames, etc.) used. What is their expected use? This will affect the amount of infiltration air.
8. The expected occupancy for the space and the time schedule of this occupancy.
9. Type and location of lighting. Types of appliances and electrical machinery in the space and their expected use.
10. Location of electric, gas, and water services. These services should be integrated with the locations of the heating and air-conditioning duct, piping, and equipment.

11. Ventilation requirements for the structure. Does it require 100% outdoor air, a given number of CFM per person, or a given number of CFM per square foot of floor area?
12. Local and/or national codes relating to ventilation, gas, and/or electric piping.
13. Outside design temperatures and wind velocities for the location.
14. The environmental conditions that are maintained. Will fluctuations of these conditions with load be detrimental to the purpose served by the structure?
15. The heating and cooling loads (also consider the moisture load, air contaminants, and noise).
16. The type of heating and cooling system to be used in the structure. Is it forced air, circulated water, or direct expansion? Will it be a multizone, single zone, reheat, variable air volume, or another type of system? What method of control will be used? Will a dedicated outdoor air system be considered?
17. The heating and cooling equipment size that will maintain the inside design conditions for the selected outside design condition. Electric heat or fossil fuel? Mechanical vapor compression or absorption chiller?
18. The advantages and disadvantages of oversizing and undersizing the equipment as applied to the structure. Survey any economic tradeoffs to be made. Should a different type of unit be installed in order to reduce operating costs? Should a more sophisticated control system be used to give more exact control of humidity and temperature or should an on-off cycle be used? Fuel economy as related to design will become an even more important factor in system selection and operation.
19. What is the estimated annual energy usage?

In general, no absolute rules dictate correct selections or specifications for each of the above items, so only engineering estimates or educated guesses can be made. However, estimates must be based on sound fundamental principles and concepts. This book presents a basic philosophy of environmental control as well as the basic concepts of design. These ideas relate directly to the *ASHRAE Handbook* series: 2014 *Refrigeration*, 2015 *HVAC Applications*, 2016 *HVAC Systems and Equipment*, and most directly to 2017 *Fundamentals*.

1.2 Historical Notes

Knowing something of the past helps in understanding current design criteria and trends. As in other fields of technology, the accomplishments and failures of the past affect current and future design concepts. The following paragraphs consist mainly of edited excerpts from *ASHRAE Journal* articles: “A History of Heating” by John W. James, “The History of Refrigeration” by Willis R. Woolrich, and “Milestones in Air Conditioning” by Walter A. Grant, with additional information obtained from ASHRAE’s Historical Committee. These excerpts provide a synopsis of the history of environmental control.

Obviously, the earliest form of heating was the open fire. The addition of a chimney to carry away combustion byproducts was the first important step in the evolution of heating systems. By the time of the Romans, there was sufficient knowledge of ventilation to allow for the installation of ventilating and panel heating in baths. Leonardo da Vinci had invented a ventilating fan by the end of the 15th century. Robert Boyle’s law was established in 1659; John Dalton’s in 1800. In 1775, Dr. William Cullen made ice by pumping a vacuum in a vessel of water. A few years later, Benjamin Franklin wrote his treatise on Pennsylvania fireplaces, detailing their construction, installation, and operation.

Although warming and ventilating techniques had greatly improved by the 19th century, manufacturers were unable to exploit these techniques because

- Data available on such subjects as transmission coefficients, air and water friction in pipes, and brine and ammonia properties were sparse and unreliable.
- Neither set design conditions nor reliable psychrometric charts existed.
- A definitive rational theory that would permit performance calculation and prediction of results had not yet been developed.
- Little was known about physical, thermodynamic, and fluid dynamic properties of air, water, brines, and refrigerants.
- No authoritative information existed on heat transmission involving combustion, conduction, convection, radiation, evaporation, and condensation.
- No credible performance information for manufactured equipment was available.

Thanks to Thomas Edison, the first electric power plant opened in New York in 1882, making it possible for the first time to have an inexpensive source of energy for residential and commercial buildings.

1.1.1 Furnaces

By 1894, the year the American Society of Heating and Ventilating Engineers (ASH&VE) was born, central heating was fairly well developed. The basic heat sources were warm air furnaces and boilers. The combustion chambers of the first warm air furnaces were made of cast iron. Circulation in a gravity warm air furnace system is caused by the difference in air density in the many parts of the system. As the force of combustion is small, the system was designed to allow air to circulate freely. The addition of fans (circa 1899) to furnace systems provided a mechanical means of air circulation. Other additions to the modern furnace include cooling systems, humidification apparatuses, air distributors, and air filters. Another important step for the modern heating industry was the conversion of furnaces from coal to oil and gas, and from manual to automatic firing.

1.1.2 Steam Systems

James Watt developed the first steam heating system in 1770. However, the first real breakthrough in design did not occur until the early 1900s when circulation problems in these systems were improved with the introduction of a fluid-operated thermostatic trap.

From 1900 to 1925, two-pipe steam systems with thermostatic traps at the outlet of each radiator and at drip points in the piping gained wide acceptance. In smaller buildings, gravity systems were commonly installed to remove condensate. For larger systems, boiler return traps and condensate pumps with receivers were used. By 1926, the vacuum return line system was perfected for installation in large and moderate-sized buildings.

Hot water heating systems were developed in parallel with steam systems. As mentioned before, the first hot water heating system was the gravity system. In 1927, the circulator, which forced water through the system, was added to two-pipe heating systems. A few years later, a diverting tee was added to the one-pipe system, allowing for forced circulation.

During the 1930s, radiators and convectors were commonly concealed by enclosures, shields, and cabinets. In 1944, the baseboard radiator was developed. Baseboard heating improved comfort conditions as it reduced floor-to-ceiling temperature stratification.

Unit heaters and unit ventilators are two other forms of convection heating developed in the 1920s. Unit heaters were available in suspended and floor types and were classified according to the heating medium used (e.g., steam, hot water, electricity, gas, oil, or coal combustion). In addition to the heating element and fan, unit ventilators were often equipped with an air filter. Many designs provided air recirculation and were equipped with a separate outdoor air connection.

Panel heating, another form of heat distribution, was developed in the 1920s. In panel heating, a fluid such as hot water, steam, air, or electricity, circulates through distribution units embedded in the building components.

1.1.3 Early Refrigeration

Early forms of refrigeration included the use of snow, pond and lake ice, chemical mixture cooling to form freezing baths, and the manufacture of ice by evaporative and radiation cooling of water on clear nights.

By the 18th century, certain mixtures were known to lower temperatures. One such mixture, calcium chloride and snow, was introduced for commercial use. This particular mixture made possible a temperature down to –27°F (–33°C). In Great Britain, machines using chemical mixtures to produce low temperatures were introduced. However, by the time these machines were ready for commercial exploitation, mechanical ice-making processes had been perfected to such an extent that chemical mixture freezing was rendered obsolete except for such batch processes as ice cream making.

1.1.4 Mechanical and Chemical Refrigeration

In 1748, in Scotland, Dr. William Cullen and Joseph Black lectured on the latent heat of fusion and evaporation and “fixed air” (later identified as carbon dioxide). These discoveries served as the foundation on which modern refrigeration is based.

In 1851, Dr. John Gorrie, was granted US Patent No. 8080 for a refrigeration machine that produced ice and refrigerated air with compressed air in an open cycle. Also in 1851, Ferdinand Carre designed the first ammonia absorption unit.

In 1853, Professor Alexander Twining of New Haven, Connecticut, produced 1600 lb of ice a day with a doubleacting vacuum and compression pump that used sulfuric ether as the refrigerant.

Daniel L. Holden improved the Carre machine by designing and building reciprocating compressors. These compressors were applied to ice making, brewing, and meat packing. In 1872, David Boyle developed an ammonia compression machine that produced ice.

Until 1880, mechanical refrigeration was primarily used to make ice and preserve meat and fish. Notable exceptions were the use of these machines in the United States, Europe, and Australia for beer making, oil dewaxing, and wine cooling. At this time, comfort air cooling was obtained by ice or by chilling machines that used either lake or manufactured ice.

1.1.5 History of ASHRAE

The American Society of Heating and Ventilating Engineers (ASHVE) was formed in New York City in 1894 to conduct research, develop standards, hold technical meetings, and publish technical articles in journals and handbooks. Its scope was limited to the fields of heating and ventilating for commercial and industrial applications, with secondary emphasis on residential heating. Years later the Society’s name was changed to the American Society of Heating and Air-Conditioning Engineers (ASHAE) to recognize the increasing importance of air conditioning.

In 1904, the American Society of Refrigerating Engineers (ASRE) was organized and headquartered at the American Society of Mechanical Engineers (ASME). The new Society had 70 charter members and was the only engineering group in the world that confined its activities to refrigeration, which at that time consisted mainly of ammonia systems.

In 1905, ASME established 288,000 Btu in 24 hrs as the commercial ton of refrigeration (within the United States). In the same year, the New York Stock Exchange was cooled by refrigeration. In 1906, Stuart W. Cramer coined the term “air conditioning.”

The First International Congress on Refrigeration was organized in Paris in 1908 and a delegation of 26 was sent from the United States. Most of the participants were members of ASRE.

ASHAE and ASRE merged in 1959, creating the American Society of Heating, Refrigerating and Air-Conditioning Engineers.

Figure 1-1 depicts ASHRAE's history. ASHRAE celebrated its Centennial Year during society year 1994-1995. In commemoration of the centennial, two books on the history of ASHRAE and of the HVAC industry were published, *Proclaiming the Truth* and *Heat and Cold: Mastering the Great Indoors*.

1.1.6 Willis H. Carrier

Willis H. Carrier (1876-1950) has often been referred to as the "Father of Air Conditioning." His analytical and practical accomplishments contributed greatly to the development of the refrigeration industry.

Carrier graduated from Cornell University in 1901 and was employed by the Buffalo Forge Company. He realized that satisfactory refrigeration could not be installed due to the inaccurate data that were available. By 1902, he developed formulas to optimize forced-draft boiler fans, conducted tests and developed multirating performance tables on indirect pipe coil heaters, and set up the first research laboratory in the heating and ventilating industry.

In 1902, Carrier was asked to solve the problem faced by the lithographic industry of poor color register caused by weather changes. Carrier's solution was to design, test, and install at the Sackett-Wilhelms Lithographing Company of Brooklyn a scientifically engineered, year-round air-conditioning system that provided heating, cooling, humidifying, and dehumidifying.

By 1904, Carrier had adapted atomizing nozzles and developed eliminators for air washers to control dew-point temperature by heating or cooling a system's recirculated water. Soon after this development, over 200 industries were using year-round air conditioning.

At the 1911 ASME meeting, Carrier presented his paper, "Rational Psychrometric Formulae," which related dry-bulb, wet-bulb, and dew-point temperatures of air, as well as its sensible, latent, and total heat load, and set forth the theory of adiabatic saturation. The formulas and psychrometric chart presented in this paper became the basis for all fundamental calculations used by the air-conditioning industry.

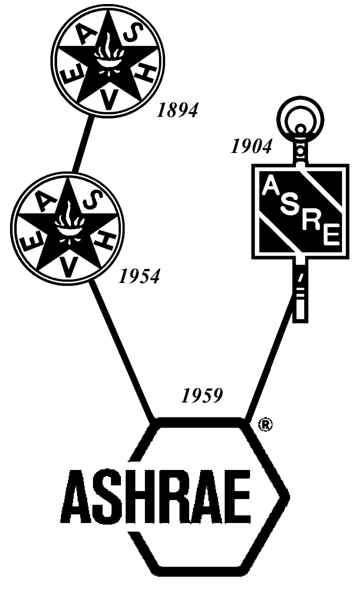

Fig. 1-1 Background of ASHRAE

By 1922, Carrier's centrifugal refrigeration machine, together with the development of nonhazardous, low-pressure refrigerants, made water chilling for large and medium-size commercial and industrial applications both economical and practical. A conduit induction system for multiroom buildings, was invented in 1937 by Carrier and his associate, Carlyle Ashley.

1.1.7 Comfort Cooling

Although comfort air-cooling systems had been built as of the 1890s, no real progress was made in mechanical air cooling until after the turn of the century. At that time, several scientifically designed air-conditioning plants were installed in buildings. One such installation included a theater in Cologne, Germany. In 1902, Alfred Wolff designed a 400-ton system for the New York Stock Exchange. Installed in 1902, this system was in operation for 20 years. The Boston Floating Hospital, in 1908, was the first hospital to be equipped with modern air conditioning. Mechanical air cooling was installed in a Texas church in 1914. In 1922, Grauman's Metropolitan Theater, the first air-conditioned movie theater, opened in Los Angeles. The first office building designed with and built for comfort air-conditioning specifications was the Milam Building, in San Antonio, Texas, which was completed in 1928. Also in 1928, the Chamber of the House of Representatives became air conditioned. The Senate became air conditioned the following year and in 1930, the White House and the Executive Office Building were air-conditioned.

The system of air bypass control, invented in 1924 by L. Logan Lewis, solved the difficult problem of humidity control under varying load. By the end of the 1920s, the first room air conditioner was introduced by Frigidaire. Other important inventions of the 1920s include lightweight extended surface coils and the first unit heater and cold diffuser.

Thomas Midgley, Jr. developed the halocarbon refrigerants in 1930. These refrigerants were found to be safe and economical for the small reciprocating compressors used in commercial and residential markets. Manufacturers were soon producing mass market room air conditioners that used Refrigerant 12.

Fluorinated refrigerants were also applied to centrifugal compression, which required only half the number of impellers for the same head as chlorinated hydrocarbons. Space and materials were saved when pressure-formed extended-surface tubes in shell-and-tube exchangers were introduced by Walter Jones. This invention was an important advance for centrifugal and reciprocating equipment.

Other achievements of the 1930s included

- The first residential lithium bromide absorption machine was introduced in 1931 by Servel.

- In 1931, Carrier marketed steam ejector cooling units for railroad passenger cars.
- As of the mid-1930s, General Electric introduced the heat pump; the electrostatic air cleaner was put out by Westinghouse; Charles Neeson of Airtemp invented the high-speed radial compressor; and W.B. Connor discovered that odors could be removed by using activated carbon.

With the end of World War II, air-conditioning technology advanced rapidly. Among the advances were air-source heat pumps, large lithium-bromide water chillers, automobile air conditioners, rooftop heating and cooling units, small, outdoor-installed ammonia absorption chillers, air purifiers, a vapor cycle aircraft cabin cooling unit, and a large-capacity Lysholm rotary compressor.

Improvements on and expansions of products that already existed include

- Dual-duct central systems for office buildings
- Change from open to hermetic compressors from the smallest reciprocating units to large-capacity centrifugals
- Resurgence of electric heating in all kinds of applications
- Use of heat pumps to reclaim heat in large buildings
- Application of electrostatic cleaners to residences
- Self-contained variable volume air terminals for multiple interior rooms
- Increasing use of total energy systems for large buildings and clusters of buildings
- Larger sizes of centrifugals, now over 5000 tons in a single unit
- Central heating and cooling plants for shopping centers, colleges, and apartment and office building complexes

In the late 1940s and into the early 1950s, development work continued on unitary heat pumps for residential and small commercial installations. These factory-engineered and assembled units, like conventional domestic boilers, could be easily and cheaply installed in the home or small commercial businesses by engineers. In 1952, heat pumps were placed on the market for mass consumption. Early heat pumps lacked the durability needed to withstand winter temperatures. Low winter temperatures placed severe stress on the components of these heat pumps (compressors, outdoor fans, reversing valves, and control hardware). Improvements in the design of heat pumps has continued, resulting in more-reliable compressors and lubricating systems, improved reversing valves, and refined control systems.

In the 1950s came the rooftop unit for commercial buildings. Multizone packaged rooftop units were popular during the 1960s; however, most were very energy inefficient and lost favor during the 1970s. Beginning with the oil embargo of 1973, the air-conditioning field could no longer conduct "business as usual," with concern mainly for the initial cost of the building and its conditioning equipment. The use of crude rules of thumb, which significantly oversized equipment and wasted energy, was largely replaced with reliance upon more scientifically sound, and often computer-assisted, design, sizing, and selection procedures. Variable air volume (VAV) designs rapidly became the most popular type of HVAC system for offices, hospitals, and some school buildings. Although energy-efficient, VAV systems proved to have their own set of problems related to indoor air quality (IAQ), sick building syndrome (SBS), and building related illness (BRI). Solutions to these problems are only now being realized.

In 1987, the United Nations Montreal Protocol for protecting the earth's ozone layer was signed, establishing the phaseout schedule for the production of chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) refrigerants. Contemporary buildings and their air-conditioning equipment must now provide improved indoor air quality as well as comfort, while consuming less energy and using alternative refrigerants.

1.3 Building Energy Use

Energy is generally used in buildings to perform functions of heating, lighting, mechanical drives, cooling, and special applications. A typical breakdown of the relative energy use in a commercial building is given as Figure 1-2.

Energy is available in limited forms, such as electricity, fossil fuels, and solar energy, and these energy forms must be converted within a building to serve the end use of the various functions. A degradation of energy is associated with any conversion process. In energy conservation efforts, two avenues of approach were taken: (1) reducing the amount of use and/or (2) reducing conversion losses. For example, the furnace that heats a building produces unusable and toxic flue gas that must be vented to the outside and in this process some of the energy is lost. Table 1-1 presents typical values for building heat losses and gains at design conditions for a mid-America climate. Actual values will vary significantly with climate and building construction.

The projected total U.S. energy consumption by end-user sector: transportation, industrial, commercial, and residential is shown in Figure 1-3. The per capita energy consumption for the U.S. and the world is shown in Figure 1-4, showing that in 2007 the U.S. consumption was the same as in 1965. This has

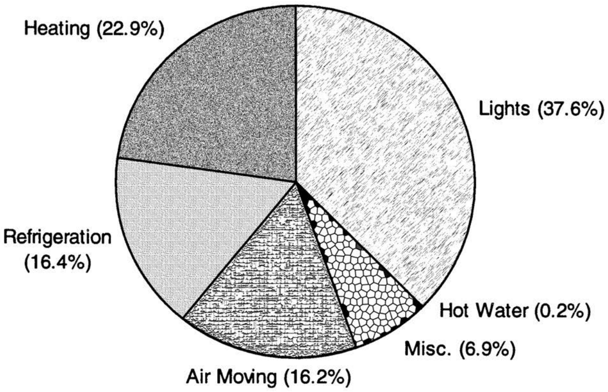

Fig. 1-2 Energy Use in a Commercial Building

been achieved through application of energy conservation principles as well as increased energy costs and changes in the economy.

The efficient use of energy in buildings can be achieved by implementing (1) optimum energy designs, (2) well-developed energy use policies, and (3) dedicated management backed up by a properly trained and motivated operating staff. Optimum energy conservation is attained when the least amount of energy is used to achieve a desired result. If this is not fully realizable, the next best method is to move excess energy from where it is not wanted to where it can be used or stored for future use, which generally results in a minimum expenditure of new energy. A system should be designed so that it cannot heat and cool the same locations simultaneously.

ASHRAE Standard 90.1-2013, "Energy Standard for Buildings Except Low-Rise Residential Buildings," and the 100-2015 series standards, "Energy Conservation in Existing Buildings," provide minimum guidelines for energy conservation design and operation. They incorporate these types of energy standards: (1) prescriptive, which specifies the materials and methods for design and construction of buildings; (2) system performance, which sets requirements for each component, system, or subsystem within a building; and (3) building energy, which considers the performance of the building as a whole. In this last type, a design goal is set for the annual energy requirements of the entire building on basis such as Btu/ft^2 per year (GJ/m^2 per year). Any combination of materials, systems, and operating procedures can be applied, as long as design energy usage does not exceed the building's annual energy budget goal. "Standard 90.1-2013 User's Manual" is extremely helpful in understanding and applying the requirements of ASHRAE Standard 90.1

Table 1-1 Typical Building Design Heat Losses or Gains

	Air Conditioning		Heating	
Building Type	**ft^2/ton**	**m^2/kW**	**Btu/h·ft^3**	**W/m^3**
Apartment	450	12	4.5	45
Bank	250	7	3.0	30
Department Store	250	7	1.0	10
Dormitories	450	12	4.5	45
House	700	18	3.0	30
Medical Center	300	8	4.5	45
Night Club	250	7	3.0	30
Office Interior	350	9	3.0	30
Exterior	275	7	3.0	30
Post Office	250	7	3.0	30
Restaurant	250	7	3.0	30
Schools	275	7	3.0	30
Shopping Center	250	7	3.0	30

This approach allows greater flexibility while promoting the goals of energy efficiency. It also allows and encourages the use of innovative techniques and the development of new methods for saving energy. Means for its implementation are still being developed. They are different for new and for existing buildings; in both cases, an accurate data base is required as well as an accurate, verifiable means of measuring consumption.

As energy prices have risen, more sophisticated schemes for reducing energy consumption have been conceived. Included in such schemes are cogeneration, energy management systems (EMS), direct digital control (DDS), daylighting, closed water-loop heat pumps, variable air volume (VAV) systems, variable frequency drives, thermal storage, dessicant dehumidication, and heat recovery in commercial and institutional buildings and industrial plants.

As detailed in a 1992 Department of Energy Report, "Commercial Buildings Consumption and Expenditures, 1989," more than seventy percent of the commercial-industrial-institutional (C-I-I) buildings recently built in the United States made use of energy conservation measures for heating and cooling.

The type of building and its use strongly affects the energy use as shown in Table 1-2.

Heating and air-conditioning systems that are simple in design and of the proper size for a given building generally have relatively low maintenance and operating costs. For optimum results, as much inherent thermal control as is economically possible should be built into the basic structure. Such

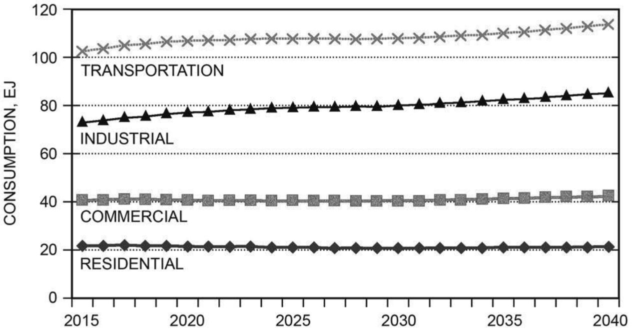

Fig. 1-3 Projected total U.S. Energy Consumption by End-User Sector

(EIA 2016)

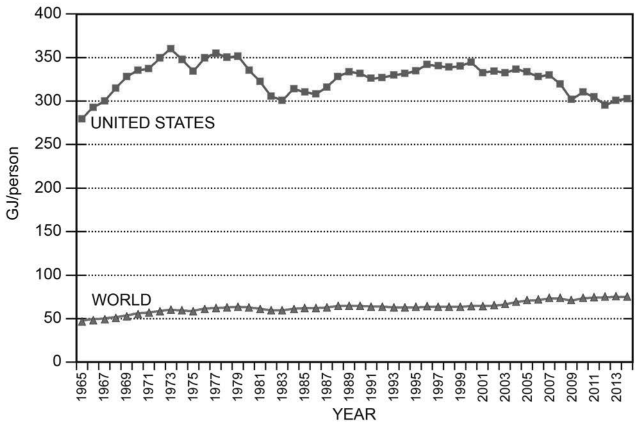

Fig. 1-4 U.S Per Capita Energy Consumption

(BP 2015)

control might include materials with high thermal properties, insulation, and multiple or special glazing and shading devices. The relationship between the shape, orientation, and air-conditioning requirement of a building should also be considered. Since the exterior load may vary from 30 to 60% of the total air-conditioning load when the fenestration (light transmitting) area ranges from 25 to 75% of the floor area, it may be desirable to minimize the perimeter area. For example, a rectangular building with a 4-to-1 aspect ratio requires substantially more refrigeration than a square building with the same floor area.

When a structure is characterized by several exposures and multipurpose use, especially with wide load swings and noncoincident energy use in certain areas, multiunit or unitary systems may be considered for such areas, but not necessarily for the entire building. The benefits of transferring heat absorbed by cooling from one area to other areas, processes, or services that require heat may enhance the selection of such systems.

Buildings in the US consume significant quantities of energy each year. According to the US Department of Energy (DOE), buildings account for 36% of all the energy used in the US, and 66% of all the electricity used. Beyond economics, energy use in the buildings sector has significant implications for our environment. Emissions related to building energy use account for 35% of carbon dioxide emissions, 47% of sulfur dioxide emissions, and 22% of nitrogen oxide emissions.

1.4 Conceptualizing an HVAC System

An important tool for the HVAC design engineer is the ability to develop a quick overview or “concept” of the magnitude of the project at hand. Toward this goal, the industry has developed a number of “rules of thumb,” some more accurate than others. As handy as they might be, these approximations must be treated as just that—approximations. Don’t use them as “rules of dumb.”

Tables 1-1 and 1-2 are examples of such rules-of-thumb, providing data for a quick estimate of heating and cooling equipment sizes and of building energy use, requiring knowledge only of the size and intended use of the building. Other rules-of-thumb include using a face velocity of 500 fpm in determining the face area for a cooling coil, the use of 400 cfm/ton for estimating the required cooling airflow rate, the use of 2.5 gpm/ton for determining the water flow rate through the cooling coil and chiller unit, using 1.2 cfm/sq ft of gross floor area for estimating the required conditioned airflow rate for comfort cooling, and the estimation of 0.6 kW/ton as the power requirement for air conditioning. Table 1-3 provides very approximate data related to the cost of HVAC equipment and systems.

Table 1-2 Annual Energy Use Per Unit Floor Area

Building Type	Annual Energy Use kWh/ft^2
Assembly	18.7
Education	25.5
Food Sales	51.5
Health Care	64.0
Lodging	38.8
Mercantile	24.8
Office	30.5
Warehouse	16.9
Vacant	6.9
All Buildings	26.7

Table 1.4 provides approximate energy costs for commercial consumers in the United States for 2015. Keep in mind that these energy costs are very volatile at this time.

Table 1.5 gives approximate total building costs for offices and medical offices averaged for twenty U.S. locations in 2007.

The material presented in this book will enable the reader to validate appropriate rules as well as to improve upon these approximations for the final design.

1.5 Sustainability and Green Buildings

The following discussion concerning sustainable design and green buildings has been extracted from Chapters 34 and 35 of the 2017 *ASHRAE Handbook—Fundamentals*.

Pollution, toxic waste creation, waste disposal, global climate change, ozone depletion, deforestation, and resource depletion are recognized as results of uncontrolled technological and population growth. Without mitigation, current trends will adversely affect the ability of the earth’s ecosystem to regenerate and remain viable for future generations.

The built environment contributes significantly to these effects, accounting for one-sixth of the world’s fresh water use, one-quarter of its wood harvest, and two-fifths of its material and energy flows. Air quality, transportation patterns, and watersheds are also affected. The resources required to serve this sector are considerable and many of them are diminishing.

Table 1-3 Capital Cost Estimating Factors

Cooling Systems
- $1675/installed ton of cooling

Heating Systems
- $2.92/cfm of installed heating

Fans/Ducting/Coils/Dampers/Filters
- $7.84/cfm all-system

Table 1-4 Approximate Energy Costs to Commercial Consumers (2015)

Electricity ($/kWh)	0.090
Natural Gas ($/therm)	0.84
LPG ($/gal)	2.95
No. 2 Fuel Oil ($/gas)	3.46

Table 1-5 Approximate Total Building Costs ($/sq. ft.)
(Adapted from RSMeans Costs Comparisons 2007)

	2–4-Story Office Building	5–10-Story Office Building	11–20-Story Office Building	Medical Office Building
High	194	181	167	219
Average	149	130	121	169
Low	117	110	98	132

Recognition of how the building industry affects the environment is changing the approach to design, construction, operation, maintenance, reuse, and demolition of buildings and focusing on environmental and long-term economic consequences. Although this sustainable design ethic—*sustainability*—covers things beyond the HVAC industry alone, efficient use of energy resources is certainly a key element of any sustainable design and is very much under the control of the HVAC designer.

Research over the years has shown that new commercial construction can reduce annual energy consumption by about 50% using integrated design procedures and energy conservation techniques. In the past few years several programs promoting energy efficiency in building design and operation have been developed. One of these is Energy Star Label (www.energystar.gov) and another one, which is becoming well known, is Leadership in Energy and Environmental Design (LEED) (www.usgbc.org/leed).

In 1999 the Environmental Protection Agency of the US government introduced the Energy Star Label for buildings. This is a set of performance standards that compare a building's adjusted energy use to that of similar buildings nationwide. The buildings that perform in the top 25%, while conforming to standards for temperature, humidity, illumination, outdoor air requirements, and air cleanliness, earn the Energy Star Label.

LEED is a voluntary points-based national standard for developing a high-performance building using an integrated design process. LEED evaluates "greenness" in five categories: sustainable sites, water efficiency, energy and atmosphere, materials and resources, and the indoor air environmental quality.

In the energy and atmosphere category, building systems commissioning and minimum energy usages are necessary requirements. The latter requires meeting the requirements *ANSI/ASHRAE/IESNA Standard 90.1-2013, Energy Standard for Buildings Except Low-Rise Residential Buildings*, or the local energy code, whichever is more stringent.

Basically LEED defines what makes a building "green" while the Energy Star Label is concerned only with energy performance. Both of these programs require adherence to ASHRAE standards. Chapter 35 of the 2017 *ASHRAE Handbook—Fundamentals* provides guidance in achieving sustainable designs.

The basic approach to energy-efficient design is reducing loads (power), improving transport systems, and providing efficient components and "intelligent" controls. Important design concepts include understanding the relationship between energy and power, maintaining simplicity, using self-imposed budgets, and applying energy-smart design practices.

Just as an engineer must work to a cost budget with most designs, self-imposed power budgets can be similarly helpful in achieving energy-efficient design. For example, the following are possible goals for mid-rise to high-rise office buildings in a typical midwestern or northeastern temperature climate:

• Installed lighting (overall)	0.8 W/ft^2
• Space sensible cooling	15 Btu/h·ft^2
• Space heating load	10 Btu/h·ft^2
• Electric power (overall)	3 W/ft^2
• Thermal power (overall)	20 Btu/h·ft^2
• Hydronic system head	70 ft of water
• Water chiller (water-cooled)	0.5 kW/ton
• Chilled-water system auxiliaries	0.15 kW/ton
• Unitary air-conditioning systems	1.0 kW/ton
• Annual electric energy	15 kWh/ft^2·yr
• Annual thermal energy	5 Btu/ft^2·yr·°F·day

These goals, however, may not be realistic for all projects.

As the building and systems are designed, all decisions become interactive as the result of each subsystem's power or energy performance being continually compared to the "budget."

Energy efficiency should be considered at the beginning of building design because energy-efficient features are most easily and effectively incorporated at that time. Active participation of all members of the design team (including owner, architect, engineer, and often the contractor) should be sought early. Consider building attributes such as building function, form, orientation, window/wall ratio, and HVAC system types early because each has major energy implications.

1.6 Problems

1.1 Estimate whether ice will form on a clear night when ambient air temperature is 45°F (7.2°C), if the water is placed in a shallow pan in a sheltered location where the convective heat transfer coefficient is 0.5 Btu/h·ft^2·°F [2.8 W/(m^2·K)].

1.2 Obtain a sketch or drawing of Gorrie's refrigeration machine and describe its operation.

1.3 Plot the history of the annual energy use per square foot of floor space for nonresidential buildings and predict the values for the years 2014 and 2015.

1.4 Estimate the size of cooling and heating equipment that is needed for a new bank building in middle America that is 140 ft by 220 ft by 12 ft high (42.7 m by 67 m by 3.7 m high). [Answer: 123 tons cooling, 11,109,000 Btu/h heating]

1.5 Estimate the size of heating and cooling equipment that will be needed for a residence in middle America that is 28 ft by 78 ft by 8 ft high (8.5 m by 23.8 m by 2.4 m high).

1.6 Estimate the initial cost of the complete HVAC system (heating, cooling, and air moving) for an office building, 40 ft by 150 ft by 10 ft high (12.2 m by 45.7 m by 3.1 m high).

1.7 Estimate the annual operating cost for the building in Problem 1.6 if it is all-electric. [Answer: $14,640]

1.8 Conceptualize, as completely as possible, using information only from Sections 1.3, 1.4, and 1.5, the building of Project 8 Two-Story Building, Appendix B, Systems Design Problems.

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Chapter 2

THERMODYNAMICS AND PSYCHROMETRICS

This chapter reviews the principles of thermodynamics, evaluates thermodynamic properties, and applies thermodynamics and psychrometrics to air-conditioning and refrigeration processes and systems. Greater detail on thermodynamics, particularly relating to the Second Law and irreversibility, is found in Chapter 2, 2017 *ASHRAE Handbook—Fundamentals*. Details on psychrometric properties can be found in Chapter 1 of the 2017 *ASHRAE Handbook—Fundamentals*.

2.1 Fundamental Concepts and Principles

2.1.1 Thermodynamics

Thermodynamics is the science devoted to the study of energy, its transformations, and its relation to status of matter. Since every engineering operation involves an interaction between energy and materials, the principles of thermodynamics can be found in all engineering activities.

Thermodynamics may be considered the description of the behavior of matter in equilibrium and its changes from one equilibrium state to another. The important concepts of thermodynamics are energy and entropy; the two major principles of thermodynamics are called the first and second laws of thermodynamics. The first law of thermodynamics deals with energy. The idea of energy represents the attempt to find an invariant in the physical universe, something that remains constant in the midst of change. The second law of thermodynamics explains the concept of entropy; e.g., every naturally occurring transformation of energy is accompanied somewhere by a loss in the availability of energy for future performance of work.

The German physicist, Rudolf Clausius (1822–1888), devised the concept of entropy to quantitatively describe the loss of available energy in all naturally occurring transformations. Although the natural tendency is for heat to flow from a hot to a colder body with which it is in contact, corresponding to an increase in entropy, it is possible to make heat flow from a colder body to a hot body, as is done every day in a refrigerator. However, this does not take place naturally or without effort exerted somewhere.

According to the fundamental principles of thermodynamics, the energy of the world stays constant and the entropy of the world increases without limit. If the essence of the first principle in everyday life is that one cannot get something for nothing, the second principle emphasizes that every time one does get something, the opportunity to get that something in the future is reduced by a measurable amount, until ultimately, there will be no more "getting." This "heat death," envisioned by Clausius, will be a time when the universe reaches a level temperature; and though the total amount of energy will be the same as ever, there will be no means of making it available, as entropy will have reached its maximum value.

Like all sciences, the basis of thermodynamics is experimental observation. Findings from these experimental observations have been formalized into basic laws. In the sections that follow, these laws and their related thermodynamic properties will be presented and applied to various examples. These examples should give the student an understanding of the basic concepts and an ability to apply these fundamentals to thermodynamic problems. It is not necessary to memorize numerous equations, for problems are best solved by applying the definitions and laws of thermodynamics.

Thermodynamic reasoning is always from the general law to the specific case; that is, the reasoning is deductive rather than inductive. To illustrate the elements of thermodynamic reasoning, the analytical processes may be divided into two steps:

1. The idealization or substitution of an analytical model for a real system. This step is taken in all engineering sciences. Therefore, skill in making idealizations is an essential part of the engineering art.
2. The second step, unique to thermodynamics, is the deductive reasoning from the first and second laws of thermodynamics.

These steps involve (a) an energy balance, (b) a suitable properties relation, and (c) accounting for entropy changes.

2.1.2 System and Surroundings

Most applications of thermodynamics require the definition of a system and its surroundings. A system can be an object, any quantity of matter, or any region of space selected for study and set apart (mentally) from everything else, which then becomes the surroundings. The systems of interest in thermodynamics are finite, and the point of view taken is macroscopic rather than microscopic. No account is taken of the detailed structure of matter, and only the coarse characteristics of the system, such as its temperature and pressure, are regarded as thermodynamic coordinates.

Everything external to the system is the surroundings, and the system is separated from the surroundings by the system boundaries. These boundaries may be either movable or fixed; either real or imaginary.

2.1.3 Properties and State

A property of a system is any observable characteristic of the system. The more common thermodynamic properties are temperature, pressure, specific volume or density, internal energy, enthalpy, and entropy.

The state of a system is its condition or configuration described in sufficient detail so that one state may be distinguished from all other states. A listing of a sufficient number of independent properties constitutes a complete definition of the state of a system.

The state may be identified or described by observable, macroscopic properties such as temperature, pressure, and density. Each property of a substance in a given state has only one value; this property always has the same value for a given state, regardless of how the substance arrived at that state. In fact, a property can be defined as any quantity that depends on the state of the system and is independent of the path (i.e., the prior history) by which the system arrived at that given state. Conversely, the state is specified or described by its properties.

The state of a macroscopic system is the condition of the system as characterized by the values of its properties. This chapter directs attention to **equilibrium states**, with equilibrium used in its generally accepted context—the equality of forces, or the state of balance. In future discussion, the term *state* refers to an equilibrium state unless otherwise noted. The concept of equilibrium is important, as it is only in an equilibrium state that thermodynamic properties have meaning. A system is in thermodynamic equilibrium if it is incapable of finite, spontaneous change to another state without a finite change in the state of the surroundings.

Included in the many types of equilibria are thermal, mechanical, and chemical. A system in thermal equilibrium is at the same temperature as the surroundings and the temperature is the same throughout. A system in mechanical equilibrium has no part accelerating ($\Sigma F = 0$) and the pressure within the system is the same as in the surroundings. A system in chemical equilibrium does not tend to undergo a chemical reaction; the matter in the system is said to be inert.

Any property of a thermodynamic system has a fixed value in a given equilibrium state, regardless of how the system arrives at that state. Therefore, the change that occurs in the value of a property when a system is altered from one equilibrium state to another is always the same. This is true regardless of the method used to bring about a change between the two end states. The converse of this statement is equally true. If a measured quantity always has the same value between two given states, that quantity is a measure of the change in a property. This latter assertion is useful in connection with the conservation of energy principle introduced in the next section.

The uniqueness of a property value for a given state can be described mathematically in the following manner. The integral of an exact differential dY is given by

$$\int_1^2 dY = Y_2 - Y_1 = \Delta Y$$

Thus the value of the integral depends solely on the initial and final states. Likewise, the change in the value of a property depends only on the end states. Hence the differential change dY in a property Y is an exact differential. Throughout this text, the infinitesimal variation of a property will be identified by the differential symbol d preceding the property symbol. For example, the infinitesimal change in the pressure p of a system is given by dp. The finite change in a property is denoted by the symbol Δ (capital delta), for example, Δp. The change in a property value ΔY always represents the final value minus the initial value. This convention must be kept in mind.

The symbol δ is used instead of the usual differential operator d as a reminder that some quantities depend on the process and are not a property of the system. δQ represents only a small quantity of heat, not a differential. δm represents only a small quantity of matter.

The same qualifications for δ hold in the case of thermodynamic work. As there is no exact differential dW, small quantities of W similar in magnitude to differentials are expressed as δW.

2.1.4 Processes and Cycles

A process is a change in state which can be defined as *any change in the properties of a system*. A process is described in part by the series of states passed through by the system. Often, but not always, some interaction between the system and surroundings occurs during a process; the specification of this interaction completes the description of the process.

Describing a process typically involves specifying the initial and final equilibrium states, the path (if identifiable), and the interactions which take place across the boundaries of the system during the process. The following terms define special processes:

isobaric or **constant pressure**—process wherein the pressure does not change;

isothermal—process that occurs at constant temperature;

isometric—process with constant volume;

adiabatic—process in which no heat is transferred to or from the system;

isentropic—process with no change in entropy.

A **cycle** is a process, or more frequently, a series of processes wherein the initial and final states of the system are identical. Therefore, at the conclusion of a cycle, all the properties have the same value they had at the beginning.

2.1.5 Reversibility

All naturally occurring changes or processes are irreversible. Like a clock, they tend to run down and cannot rewind themselves without other changes in the surroundings occurring. Familiar examples are the transfer of heat with a finite temperature difference, the mixing of two gases, a waterfall, and a chemical reaction. *All of the above changes can be reversed, however.* Heat can be transferred from a region of

low temperature to one of higher temperature; gas can be separated into its components; water can be forced to flow uphill. The important point is that these things can be done *only at the expense of some other system*, which itself becomes run down.

A process is reversible if its direction can be reversed at any stage by an infinitesimal change in external conditions. If a connected series of equilibrium states is considered, each representing only an infinitesimal displacement from the adjacent one, but with the overall result a finite change, then a reversible process exists.

All actual processes can be made to approach a reversible process by a suitable choice of conditions; but like the absolute zero of temperature, the strictly reversible process is only a concept that aids in the analysis of problems. The approach of actual processes to this ideal limit can be made almost as close as is desired. However, the closeness of approach is generally limited by economic factors rather than physical ones. The truly reversible process would require an infinite time for completion. The sole reason for the concept of the reversible process is to establish a standard for the comparison of actual processes. The reversible process is one that gives the maximum accomplishment, i.e., yields the greatest amount of work or requires the least amount of work to bring about a given change. It gives the maximum efficiency toward which to strive, but which will never be equalled. The reversible process is the standard for judging the efficiency of an actual process.

Since the reversible process represents a succession of equilibrium states, each only a differential step from its neighbor, the reversible process can be represented as a continuous line on a state diagram (p-v, T-s, etc.). The irreversible process cannot be so represented. The terminal states and general direction of change can be noted, but the complete path of change is an indeterminate, irreversible process and cannot be drawn as a line on a thermodynamic diagram.

Irreversibilities always lower the efficiencies of processes. Their effect is identical to that of friction, which is one cause of irreversibility. Conversely, no process more efficient than a reversible process can be imagined. The reversible process represents a standard of perfection that cannot be exceeded because

1. It places an upper limit on the work that may be obtained for a given work-producing process;
2. It places a lower limit on the work input for a given work-requiring process.

2.1.6 Conservation of Mass

From relativistic considerations, mass m and energy E are related by the well-known equation:

$$E = mc^2$$

where c = velocity of light.

This equation shows that the mass of a system does change when its energy changes. However, for other than nuclear reactions, the change is quite small and even the most accurate chemical balance cannot detect the change in mass. Thus, conservation of mass and conservation of energy are treated as separate laws in basic thermodynamics.

The mass rate of flow of a fluid passing through a cross-sectional area A is

$$m = AV/v \tag{2-1}$$

where V is the average velocity of the fluid in a direction normal to the plane of the area A, and v is the specific volume of the fluid. For steady flow with fluid entering a system at a section 1 and leaving at a section 2,

$$m_1 = m_2 = A_1V_1/v_1 = A_2V_2/v_2 \tag{2-1a}$$

This is the continuity equation of steady flow. It can readily be extended to any number of system inlets and outlets and is used in nearly all energy analyses.

2.2 Properties of a Substance

2.2.1 Specific Volume and Density

The specific volume of a substance v is the volume per unit mass. The density of a substance ρ is the mass per unit volume, and is therefore the reciprocal of the specific volume. Specific volume and density are intensive properties in that they are independent of the size of the system.

2.2.2 Pressure

When dealing with liquids and gases, we ordinarily speak of *pressure*; in solids we speak of *stresses*. The pressure in a fluid at rest at a given point is the same in all directions. Pressure is defined as the normal component of force per unit area.

Absolute pressure is the quantity of interest in most thermodynamic investigations. Most pressure and vacuum gages, however, read the difference between absolute pressure and the atmospheric pressure existing at the gage (Figure 2-1).

2.2.3 Temperature

Because temperature is difficult to define, equality of temperature is defined instead. Two bodies have **equality of temperature** if no change in any observable property occurs when they are in thermal communication.

The **zeroth law of thermodynamics** states that when two bodies have equality of temperature with a third body, they in turn have equality of temperature with each other. Since this fact is not derivable from other laws, and since in the logical presentation of thermodynamics it precedes the first and second laws of thermodynamics, it has been called the zeroth law of thermodynamics. This law is the basis of temperature measurement. Every time a body has equality of temperature with a thermometer, it is said that the body has the temperature read on the thermometer. The problem remains, however, of relating temperatures that might be read on different thermometers, or that are obtained when different temperature-measuring devices are used, such as thermocouples and resistance thermometers. The need for a standard scale for temperature measurements is apparent.

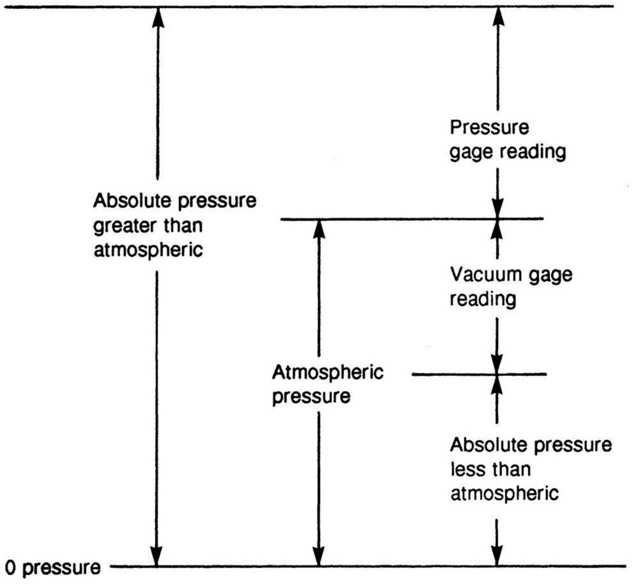


Fig. 2-1 Terms Used in Pressure Measurement

Fahrenheit and **Celsius** are two commonly used temperature measuring scales. The Celsius scale was formerly called the Centigrade scale.

In this text, the abbreviations °F and °C denote the Fahrenheit and Celsius scales, respectively. The symbols *t* and *T* are both used in the literature for temperature on all temperature scales. Unfortunately, little uniformity exists with nomenclature in engineering.

The absolute scale related to the Celsius scale is referred to as the Kelvin scale and is designated K. For SI units, the degree sign is not used with the Kelvin scale. The relation between the SI temperature scales is

$$\text{K} = {}^\circ\text{C} + 273.15$$

The absolute scale related to the Fahrenheit scale is referred to as the Rankine scale and is designated °R. The relation between these scales is

$$^\circ\text{R} = {}^\circ\text{F} + 459.67$$

2.2.4 Internal Energy

Internal energy refers to the energy possessed by a material due to the motion and/or position of the molecules. This form of energy may be divided into two parts: (1) kinetic internal energy, which is due to the velocity of the molecules; and (2) potential internal energy, which is due to the attractive forces existing between molecules. Changes in the average velocity of molecules are indicated by temperature changes of the system; variations in relative distance between molecules are denoted by changes in phase of the system.

The symbol U designates the internal energy of a given mass of a substance. Following the convention used with other extensive properties, the symbol u designates the internal energy per unit mass. As in the case of specific volume, u can represent specific internal energy.

2.2.5 Enthalpy

In analyzing specific types of processes, certain combinations of thermodynamic properties, which are therefore also properties of the substance undergoing the change of state, are frequently encountered. One such combination is $U + pV$. It is convenient to define a new extensive property, called *enthalpy*:

$$H = U + pV$$

or, per unit mass

$$h = u + pv \tag{2-2}$$

As in the case of internal energy, specific enthalpy can be referred to as h, and total enthalpy H. However, both may be called enthalpy, since the context makes it clear which is meant.

2.2.6 Entropy

Entropy S is a measure of the molecular disorder or of the probability of a given state. The more disordered a system, the greater is its entropy; conversely, an orderly or unmixed configuration is one of low entropy.

By applying the theory of probability to molecular systems, Boltzmann showed a simple relationship between the entropy of a given system of molecules and the probability of its occurrence. This relationship is given as

$$S = k \ln W$$

where k is the Boltzmann constant and W is the thermodynamic probability.

Since entropy is the property used in quantifying the Second Law of Thermodynamics, additional discussion from a classical thermodynamic viewpoint will be presented when the Second Law is discussed.

2.2.7 Specific Heats

The constant-volume specific heat and the constant-pressure specific heat are useful functions for thermodynamic calculations—particularly for gases.

The constant-volume specific heat c_v is defined by the relation

$$c_v = (\partial u / \partial T)_v \tag{2-3}$$

The constant-pressure specific heat c_p is defined by the relation

$$c_p = (\partial h / \partial T)_p \tag{2-4}$$

Note that each of these quantities is defined in terms of properties. Thus, the constant-volume and constant-pressure specific heats are thermodynamic properties of a substance.

2.2.8 Dimensions and Units

The fundamental and primitive concepts which underlie all physical measurements and all properties are time, length, mass, absolute temperature, electric current, and amount of substance. Arbitrary scales of measurement must

be established for these primary dimensions, with each scale divided into specific units of size. The internationally accepted base units for the six quantities are as follows:

length	metre (m)
mass	kilogram (kg)
time	second (s)
electric current	ampere (A)
thermodynamic temperature	kelvin (K)
amount of substance	mole (mol)

Each of these has a precise definition according to international agreement. They form the basis for the SI from the French document, *Le Système International d'Unités* (*SI*), or International System of Units.

The mass of a system is often given by stating the number of moles it contains. A mole is the mass of a chemical species equal numerically to its molecular mass. Thus, a kilogram mole of oxygen (O_2) contains 32 kilograms. In addition, the number of molecules in a kilogram mole is the same for all substances. This is also true for a gram mole, and in this case the number of molecules is Avogadro's number, equal to 6.0225×10^{23} molecules.

Many derived units are important in thermodynamics. Examples are force, pressure, and density. Force is determined through Newton's second law of motion, $F = ma$, and has the basic unit $(\text{kg·m})/\text{s}^2$. The SI unit for this composite set is the newton (N). Pressure is defined as force per unit area (N/m^2), called the pascal (Pa); and density is mass per unit volume (kg/m^3).

The US customary engineering system of units also recognizes the second as the basic unit of time, and the ampere as the unit of current. However, absolute temperature is measured in degrees Rankine (°R). The foot (ft) is the usual unit of length and the pound mass (lb_m) is the unit of mass. The molar unit is the pound mole. ASHRAE calls this system the inch-pound (I-P) unit system.

The unit of force, the pound force (lb_f), is defined without reference to Newton's second law, so this law must be written to include a dimensional proportionality constant:

$$F = ma/g_c$$

where g_c is the proportionality constant. In the I-P system, the proportionality constant is

$$g_c = 32.174\ (\text{lb}_m/\text{lb}_f)(\text{ft/s}^2)$$

The unit of density is lb_m/ft^3, and the unit of pressure is lb_f/ft^2 or lb_f/in^2, often written psi. Pressure gages usually measure pressure relative to atmospheric pressure. The term **absolute pressure** is often used to distinguish thermodynamic (actual) pressure (psia) from gage (relative) pressure (psig).

In SI units, the proportionality constant g_c in Newton's law is unity or

$$g_c = 1\ (\text{kg/N})(\text{m/s}^2)$$

In this book, all equations that derive from Newton's law carry the constant g_c.

2.2.9 Pure Substance

A pure substance is one that has a homogeneous and invariable chemical composition. It may exist in more than one phase, but the chemical composition is the same in all phases. Thus, liquid water, a mixture of liquid water and water vapor (steam), or a mixture of ice and liquid water are all pure substances, for every phase has the same chemical composition. On the other hand, a mixture of liquid air and gaseous air is not a pure substance, since the composition of the liquid phase is different from that of the vapor phase.

Sometimes a mixture of gases is considered a pure substance as long as there is no change of phase. Strictly speaking, this is not true. A mixture of gases, such as air, exhibits some of the characteristics of a pure substance as long as there is no change of phase.

Consider as a system that water is contained in the piston-cylinder arrangement of Figure 2-2. Suppose that the piston maintains a pressure of 14.7 lb_f/in. (101.3 kPa) in the cylinder containing H_2O, and that the initial temperature is 59°F (15°C). As heat is transferred to the water, the temperature increases appreciably, the specific volume increases slightly, and the pressure remains constant. When the temperature reaches 212°F (100°C), additional heat transfer results in a change of phase. That is, some of the liquid becomes vapor, and during this process both the temperature and pressure remain constant, while the specific volume increases considerably. When the last drop of liquid has vaporized, further heat transfer results in an increase in both temperature and specific volume of the vapor.

Saturation temperature designates the temperature at which vaporization takes place at a given pressure; this pressure is called the *saturation pressure* for the given temperature. Thus for water at 212°F (100°C), the saturation pressure is 14.7 $\text{lb}_f/\text{in.}^2$ (101.3 kPa), and for water at 14.7 $\text{lb}_f/\text{in.}^2$ (101.3 kPa), the saturation temperature is 212°F (100°C).

If a substance exists as liquid at the saturation temperature and pressure, it is called saturated liquid. If the temperature of the liquid is lower than the saturation temperature for the existing pressure, it is called a subcooled liquid (implying that the temperature is lower than the saturation temperature for the given pressure) or a compressed liquid (implying that the pressure is greater than the saturation pressure for the given temperature).

When a substance exists as part liquid and part vapor at the saturation temperature, its quality is defined as the ratio of the mass of vapor to the total mass. The quality may be considered as an intensive property, and it has the symbol x. Quality has meaning only when the substance is in a saturated state, i.e., at saturation pressure and temperature.

If a substance exists as vapor at the saturation temperature, it is called **saturated vapor**. (Sometimes the term dry saturated vapor is used to emphasize that the quality is 100%.) When the vapor is at a temperature greater than the saturation tempera-

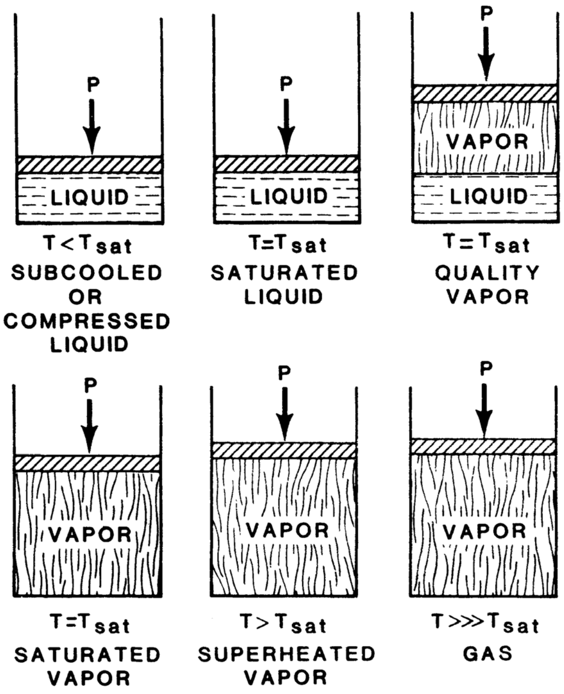


Fig. 2-2 Thermodynamic Fluid States

ture, it is said to exist as superheated vapor. The pressure and temperature of superheated vapor are independent properties because the temperature may increase while the pressure remains constant. Actually, gases are highly superheated vapors.

The entire range of phases is summarized by Figure 2-3, which shows how the solid, liquid, and vapor phases may exist together in equilibrium. Along the sublimation line, the solid and vapor phases are in equilibrium, along the fusion line, the solid and liquid phases are in equilibrium, and along the vaporization line, the liquid and vapor phases are in equilibrium. The only point at which all three phases may exist in equilibrium is the triple point. The vaporization line ends at the critical point because there is no distinct change from the liquid phase to the vapor phase above the critical point.

Consider a solid in state A, Figure 2-3. When the temperature is increased while the pressure (which is less than the triple point pressure) is constant, the substance passes directly from the solid to the vapor phase. Along the constant pressure line EF, the substance first passes from the solid to the liquid phase at one temperature, and then from the liquid to the vapor phase at a higher temperature. Constant-pressure line CD passes through the triple point, and it is only at the triple point that the three phases may exist together in equilibrium. At a pressure above the critical pressure, such as GH, there is no sharp distinction between the liquid and vapor phases.

One important reason for introducing the concept of a pure substance is that the state of a simple compressible pure substance is defined by two independent properties. This means, for example, if the specific volume and temperature of superheated steam are specified, the state of the steam is determined.

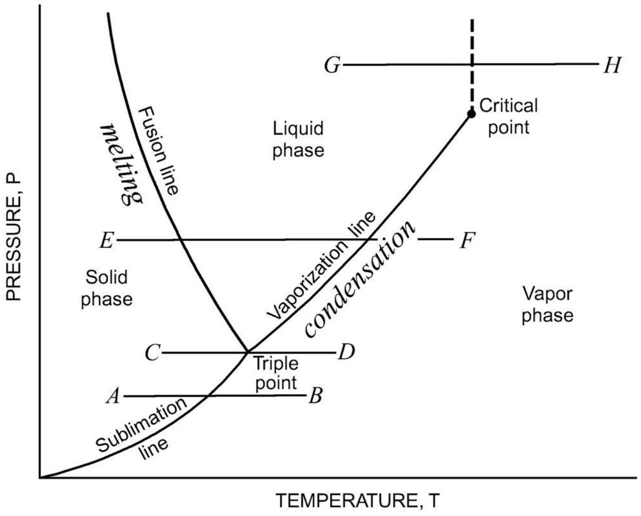


Fig. 2-3 The Pure Substance

To understand the significance of the term **independent property**, consider the saturated-liquid and saturated-vapor states of a pure substance. These two states have the same pressure and the same temperature, but are definitely not the same state. Therefore, in a saturation state, pressure and temperature are not independent properties. Two independent properties such as pressure and specific volume, or pressure and quality, are required to specify a saturation state of a pure substance.

Thus, a mixture of gases, such as air, has the same characteristics as a pure substance, as long as only one phase is present. The state of air, which is a mixture of gases of definite composition, is determined by specifying two properties as long as it remains in the gaseous phase, and in this regard, air can be treated as a pure substance.

2.2.10 Tables and Graphs of Thermodynamic Properties

Tables of thermodynamic properties of many substances are available, and they all generally have the same form. This section refers to the tables for H_2O and R-134a, as well as their respective Mollier diagrams, the *h-s* chart for steam, and the *p-h* diagram for R-134a.

Table 3 in Chapter 1 of the 2017 *ASHRAE Handbook—Fundamentals* gives thermodynamic properties of water at saturation and is reproduced in part as Table 2-1 on the following pages. In Table 2-1, the first two columns after the temperature give the corresponding saturation pressure in pounds force per square inch and in inches of mercury. The next three columns give specific volume in cubic feet per pound mass. The first of these gives the specific volume of the saturated solid (v_i) or liquid (v_f); the third column gives the specific volume of saturated vapor v_g. The difference between these two values, $v_g - v_i$ or $v_g - v_f$, represents the increase in specific volume when the state changes from saturated solid or liquid to saturated vapor, and is designated v_{ig} or v_{fg}.

Table 2-1 Thermodynamic Properties of Water

(Table 3, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °F t	Absolute Pressure		Specific Volume, ft^3/lb$_w$			Specific Enthalpy, Btu/lb$_w$			Specific Entropy, Btu/lb$_w$·°F			Temp., °F t
	p, psi	p, in. Hg	Sat. Solid v_i	Evap. v_{ig}	Sat. Vapor v_g	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
–80	0.000116	0.000236	0.01732	1953234	1953234	—193.50	1219.19	1025.69	–0.4067	3.2112	2.8045	–80
–79	0.000125	0.000254	0.01732	1814052	1814052	–193.11	1219.24	1026.13	–0.4056	3.2029	2.7972	–79
–78	0.000135	0.000275	0.01732	1685445	1685445	–192.71	1219.28	1026.57	–0.4046	3.1946	2.7900	–78
–77	0.000145	0.000296	0.01732	1566663	1566663	–192.31	1219.33	1027.02	–0.4036	3.1964	2.7828	–77
–76	0.000157	0.000319	0.01732	1456752	1456752	–191.92	1219.38	1027.46	–0.4025	3.1782	2.7757	–76
–75	0.000169	0.000344	0.01733	1355059	1355059	–191.52	1219.42	1027.90	–0.4015	3.1701	2.7685	–75
–74	0.000182	0.000371	0.01733	1260977	1260977	–191.12	1219.47	1028.34	–0.4005	3.1619	2.7615	–74
–73	0.000196	0.000399	0.01733	1173848	1173848	–190.72	1219.51	1028.79	–0.3994	3.1539	2.7544	–73
–72	0.000211	0.000430	0.01733	1093149	1093149	–190.32	1219.55	1029.23	–0.3984	3.1459	2.7475	–72
–71	0.000227	0.000463	0.01733	1018381	1018381	–189.92	1219.59	1029.67	–0.3974	3.1379	2.7405	–71
–70	0.000245	0.000498	0.01733	949067	949067	–189.52	1219.63	1030.11	–0.3963	3.1299	2.7336	–70
–69	0.000263	0.000536	0.01733	884803	884803	–189.11	1219.67	1030.55	–0.3953	3.1220	2.7267	–69
–68	0.000283	0.000576	0.01733	825187	825187	–188.71	1219.71	1031.00	–0.3943	3.1141	2.7199	–68
–67	0.000304	0.000619	0.01734	769864	769864	–188.30	1219.74	1031.44	–0.3932	3.1063	2.7131	–67
–66	0.000326	0.000664	0.01734	718508	718508	–187.90	1219.78	1031.88	–0.3922	3.0985	2.7063	–66
–65	0.000350	0.000714	0.01734	670800	670800	–187.49	1219.82	1032.32	–0.3912	3.0907	2.6996	–65
–64	0.000376	0.000766	0.01734	626503	626503	–187.08	1219.85	1032.77	–0.3901	3.0830	2.6929	–64
–63	0.000404	0.000822	0.01734	585316	585316	–186.67	1219.88	1033.21	–0.3891	3.0753	2.6862	–63
–62	0.000433	0.000882	0.01734	548041	547041	–186.26	1219.91	1033.65	–0.3881	3.0677	2.6730	–62
–61	0.000464	0.000945	0.01734	511446	511446	–185.85	1219.95	1034.09	–0.3870	3.0601	2.6730	–61
–60	0.000498	0.001013	0.01734	478317	478317	–185.44	1219.98	1034.54	–0.3860	3.0525	2.6665	–60
–59	0.000533	0.001086	0.01735	447495	447495	–185.03	1220.01	1034.98	–0.3850	3.0449	2.6600	–59
–58	0.000571	0.001163	0.01735	418803	418803	–184.61	1220.03	1035.42	–0.3839	3.0374	2.6535	–58
–57	0.000612	0.001246	0.01735	392068	392068	–184.20	1220.06	1035.86	–0.3829	3.0299	2.6470	–57
–56	0.000655	0.001333	0.01735	367172	367172	–183.78	1220.09	1036.30	–0.3819	3.0225	2.6406	–56
–55	0.000701	0.001427	0.01735	343970	343970	–183.37	1220.11	1036.75	–0.3808	3.0151	2.6342	–55
–54	0.000750	0.001526	0.01735	322336	322336	–182.95	1220.14	1037.19	–0.3798	3.0077	2.6279	–54
–53	0.000802	0.001632	0.01735	302157	302157	–182.53	1220.16	1037.63	–0.3788	3.0004	2.6216	–53
–52	0.000857	0.001745	0.01735	283335	283335	–182.11	1220.18	1038.07	–0.3778	2.9931	2.6153	–52
–51	0.000916	0.001865	0.01736	265773	265773	–181.69	1220.21	1038.52	–0.3767	2.9858	2.6091	–51
–50	0.000979	0.001992	0.01736	249381	249381	–181.27	1220.23	1038.96	–0.3757	2.9786	2.6029	–50
–49	0.001045	0.002128	0.01736	234067	234067	–180.85	1220.25	1039.40	–0.3747	2.9714	2.5967	–49
–48	0.001116	0.002272	0.01736	219766	219766	–180.42	1220.26	1039.84	–0.3736	2.9642	2.5906	–48
–47	0.001191	0.002425	0.01736	206398	206398	–180.00	1220.28	1040.28	–0.3726	2.9570	2.5844	–47
–46	0.001271	0.002587	0.01736	193909	193909	–179.57	1220.30	1040.73	–0.3716	2.9499	2.5784	–46
–45	0.001355	0.002760	0.01736	182231	182231	–179.14	1220.31	1041.17	–0.3705	2.9429	2.5723	–45
–44	0.001445	0.002943	0.01736	171304	171304	–178.72	1220.33	1041.61	–0.3695	2.9358	2.5663	–44
–43	0.001541	0.003137	0.01737	161084	161084	–178.29	1220.34	1042.05	–0.3685	2.9288	2.5603	–43
–42	0.001642	0.003343	0.01737	151518	151518	–177.86	1220.36	1042.50	–0.3675	2.9218	2.5544	–42
–41	0.001749	0.003562	0.01737	142566	142566	–177.43	1220.37	1042.94	–0.3664	2.9149	2.5485	–41
–40	0.001863	0.003793	0.01737	134176	134176	–177.00	1220.38	1043.38	–0.3654	2.9080	2.5426	–40
–39	0.001984	0.004039	0.01737	126322	126322	–176.57	1220.39	1043.82	–0.3644	2.9011	2.5367	–39
–38	0.002111	0.004299	0.01737	118959	118959	–176.13	1220.40	1044.27	–0.3633	2.8942	2.5309	–38
–37	0.002247	0.004574	0.01737	112058	112058	–175.70	1220.40	1044.71	–0.3623	2.8874	2.5251	–37
–36	0.002390	0.004866	0.01738	105592	105592	–175.26	1220.41	1045.15	–0.3613	2.8806	2.5193	–36
–35	0.002542	0.005175	0.01738	99522	99522	–174.83	1220.42	1045.59	–0.3603	2.8738	2.5136	–35
–34	0.002702	0.005502	0.01738	93828	93828	–174.39	1220.42	1046.03	–0.3592	2.8671	2.5078	–34
–33	0.002872	0.005848	0.01738	88489	88489	–173.95	1220.43	1046.48	–0.3582	2.8604	2.5022	–33
–32	0.003052	0.006213	0.01738	83474	83474	–173.51	1220.43	1046.92	–0.3572	2.8537	2.4965	–32
–31	0.003242	0.006600	0.01738	78763	78763	–173.07	1220.43	1047.36	–0.3561	2.8470	2.4909	–31
–30	0.003443	0.007009	0.01738	74341	74341	–172.63	1220.43	1047.80	–0.3551	2.8404	2.4853	–30
–29	0.003655	0.007441	0.01738	70187	70187	–172.19	1220.43	1048.25	–0.3541	2.8338	2.4797	–29
–28	0.003879	0.007898	0.01739	66282	66282	–171.74	1220.43	1048.69	–0.3531	2.8272	2.4742	–28
–27	0.004116	0.008380	0.01739	62613	62613	–171.30	1220.43	1049.13	–0.3520	2.8207	2.4687	–27
–26	0.004366	0.008890	0.01739	59161	59161	–170.86	1220.43	1049.57	–0.3510	2.8142	2.4632	–26
–25	0.004630	0.009428	0.01739	55915	55915	–170.41	1220.42	1050.01	–0.3500	2.8077	2.4577	–25
–24	0.004909	0.009995	0.01739	52861	52861	–169.96	1220.42	1050.46	–0.3489	2.8013	2.4523	–24
–23	0.005203	0.010594	0.01739	49986	49986	–169.51	1220.41	1050.90	–0.3479	2.7948	2.4469	–23
–22	0.005514	0.011226	0.01739	47281	47281	–169.07	1220.41	1051.34	–0.3469	2.7884	2.4415	–22
–21	0.005841	0.011892	0.01740	44733	44733	–168.62	1220.40	1051.78	–0.3459	2.7820	2.4362	–21
–20	0.006186	0.012595	0.01740	42333	42333	–168.16	1220.39	1052.22	–0.3448	2.7757	2.4309	–20
–19	0.006550	0.013336	0.01740	40073	40073	–167.71	1220.38	1052.67	–0.3438	2.7694	2.4256	–19
–18	0.006933	0.014117	0.01740	37943	37943	–167.26	1220.37	1053.11	–0.3428	2.7631	2.4203	–18
–17	0.007337	0.014939	0.01740	35934	35934	–166.81	1220.36	1053.55	–0.3418	2.7568	2.4151	–17
–16	0.007763	0.015806	0.01740	34041	34041	–166.35	1220.34	1053.99	–0.3407	2.7506	2.4098	–16
–15	0.008211	0.016718	0.01740	32256	32256	–165.90	1220.33	1054.43	–0.3397	2.7444	2.4046	–15
–14	0.008683	0.017678	0.01741	30572	30572	–165.44	1220.31	1054.87	–0.3387	2.7382	2.3995	–14

Table 2-1 Thermodynamic Properties of Water (*Continued*)

(Table 3, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °F	Absolute Pressure		Specific Volume, ft^3/lb$_w$			Specific Enthalpy, Btu/lb$_w$			Specific Entropy, Btu/lb$_w$·°F			Temp., °F
t	p, psi	p, in. Hg	Sat. Solid/Liq. v_i/v_f	Evap. v_{ig}	Sat. Vapor v_g	Sat. Solid/Liq. h_i/h_f	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid/Liq. s_i/s_f	Evap. s_{ig}	Sat. Vapor s_g	t
–13	0.009179	0.018689	0.01741	28983	28983	–164.98	1220.30	1055.32	–0.3377	2.7320	2.3943	–13
–12	0.009702	0.019753	0.01741	27483	27483	–164.52	1220.28	1055.76	–0.3366	2.7259	2.3892	–12
–11	0.010252	0.020873	0.01741	26067	26067	–164.06	1220.26	1056.20	–0.3356	2.7197	2.3841	–11
–10	0.010830	0.022050	0.01741	24730	24730	–163.60	1220.24	1056.64	–0.3346	2.7136	2.3791	–10
–9	0.011438	0.023288	0.01741	23467	23467	–163.14	1220.22	1057.08	–0.3335	2.7076	2.3740	–9
–8	0.012077	0.024590	0.01741	22274	22274	–162.68	1220.20	1057.53	–0.3325	2.7015	2.3690	–8
–7	0.012749	0.025958	0.01742	21147	21147	–162.21	1220.18	1057.97	–0.3315	2.6955	2.3640	–7
–6	0.013456	0.027396	0.01742	20081	20081	–161.75	1220.16	1058.41	–0.3305	2.6895	2.3591	–6
–5	0.014197	0.028906	0.01742	19074	19074	–161.28	1220.13	1058.85	–0.3294	2.6836	2.3541	–5
–4	0.014977	0.030493	0.01742	18121	18121	–160.82	1220.11	1059.29	–0.3284	2.6776	2.3492	–4
–3	0.015795	0.032159	0.01742	17220	17220	–160.35	1220.08	1059.73	–0.3274	2.6717	2.3443	–3
–2	0.016654	0.033908	0.01742	16367	16367	–159.88	1220.05	1060.17	–0.3264	2.6658	2.3394	–2
–1	0.017556	0.035744	0.01742	15561	15561	–159.41	1220.02	1060.62	–0.3253	2.6599	2.3346	–1
0	0.018502	0.037671	0.01743	14797	14797	–158.94	1220.00	1061.06	–0.3243	2.6541	2.3298	0
1	0.019495	0.039693	0.01743	14073	14073	–158.47	1219.96	1061.50	–0.3233	2.6482	2.3249	1
2	0.020537	0.041813	0.01743	13388	13388	–157.99	1219.93	1061.94	–0.3223	2.6424	2.3202	2
3	0.021629	0.044037	0.01743	12740	12740	–157.52	1219.90	1062.38	–0.3212	2.6367	2.3154	3
4	0.022774	0.046369	0.01743	12125	12125	–157.05	1219.87	1062.82	–0.3202	2.6309	2.3107	4
5	0.023975	0.048813	0.01743	11543	11543	–156.57	1219.83	1063.26	–0.3192	2.6252	2.3060	5
6	0.025233	0.051375	0.01743	10991	10991	–156.09	1219.80	1063.70	–0.3182	2.6194	2.3013	6
7	0.026552	0.054059	0.01744	10468	10468	–155.62	1219.76	1064.14	–0.3171	2.6138	2.2966	7
8	0.027933	0.056872	0.01744	9971	9971	–155.14	1219.72	1064.58	–0.3161	2.6081	2.2920	8
9	0.029379	0.059817	0.01744	9500	9500	–154.66	1219.68	1065.03	–0.3151	2.6024	2.2873	9
10	0.030894	0.062901	0.01744	9054	9054	–154.18	1219.64	1065.47	–0.3141	2.5968	2.2827	10
11	0.032480	0.066131	0.01744	8630	8630	–153.70	1219.60	1065.91	–0.3130	2.5912	2.2782	11
12	0.034140	0.069511	0.01744	8228	8228	–153.21	1219.56	1066.35	–0.3120	2.5856	2.2736	12
13	0.035878	0.073047	0.01745	7846	7846	–152.73	1219.52	1066.79	–0.3110	2.5801	2.2691	13
14	0.037696	0.076748	0.01745	7483	7483	–152.24	1219.47	1067.23	–0.3100	2.5745	2.2645	14
15	0.039597	0.080621	0.01745	7139	7139	–151.76	1219.43	1067.67	–0.3089	2.5690	2.2600	15
16	0.041586	0.084671	0.01745	6811	6811	–151.27	1219.38	1068.11	–0.3079	2.5635	2.2556	16
17	0.043666	0.088905	0.01745	6501	6501	–150.78	1219.33	1068.55	–0.3069	2.5580	2.2511	17
18	0.045841	0.093332	0.01745	6205	6205	–150.30	1219.28	1068.99	–0.3059	2.5526	2.2467	18
19	0.048113	0.097960	0.01745	5924	5924	–149.81	1219.23	1069.43	–0.3049	2.5471	2.2423	19
20	0.050489	0.102796	0.01746	5657	5657	–149.32	1219.18	1069.87	–0.3038	2.5417	2.2379	20
21	0.052970	0.107849	0.01746	5404	5404	–148.82	1219.13	1070.31	–0.3028	2.5363	2.2335	21
22	0.055563	0.113128	0.01746	5162	5162	–148.33	1219.08	1070.75	–0.3018	2.5309	2.2292	22
23	0.058271	0.118641	0.01746	4932	4932	–147.84	1219.02	1071.19	–0.3008	2.5256	2.2248	23
24	0.061099	0.124398	0.01746	4714	4714	–147.34	1218.97	1071.63	–0.2997	2.5203	2.2205	24
25	0.064051	0.130408	0.01746	4506	4506	–146.85	1218.91	1072.07	–0.2987	2.5149	2.2162	25
26	0.067133	0.136684	0.01747	4308	4308	–146.35	1218.85	1072.50	–0.2977	2.5096	2.2119	26
27	0.070349	0.143233	0.01747	4119	4119	–145.85	1218.80	1072.94	–0.2967	2.5044	2.2077	27
28	0.073706	0.150066	0.01747	3940	3940	–145.35	1218.74	1073.38	–0.2956	2.4991	2.2035	28
29	0.077207	0.157195	0.01747	3769	3769	–144.85	1218.68	1073.82	–0.2946	2.4939	2.1992	29
30	0.080860	0.164632	0.01747	3606	3606	–144.35	1218.61	1074.26	–0.2936	2.4886	2.1951	30
31	0.084669	0.172387	0.01747	3450	3450	–143.85	1218.55	1074.70	–0.2926	2.4834	2.1909	31
32	0.088640	0.180474	0.01747	3302	3302	–143.35	1218.49	1075.14	–0.2915	2.4783	2.1867	32
32*	0.08865	0.18049	0.01602	3302.07	3302.09	–0.02	1075.15	1075.14	0.0000	2.1867	2.1867	32
33	0.09229	0.18791	0.01602	3178.15	3178.16	0.99	1074.59	1075.58	0.0020	2.1811	2.1832	33
34	0.09607	0.19559	0.01602	3059.47	3059.49	2.00	1074.02	1076.01	0.0041	2.1756	2.1796	34
35	0.09998	0.20355	0.01602	2945.66	2945.68	3.00	1073.45	1076.45	0.0061	2.1700	2.1761	35
36	0.10403	0.21180	0.01602	2836.60	2836.61	4.01	1072.88	1076.89	0.0081	2.1645	2.1726	36
37	0.10822	0.22035	0.01602	2732.13	2732.15	5.02	1072.32	1077.33	0.0102	2.1590	2.1692	37
38	0.11257	0.22919	0.01602	2631.88	2631.89	6.02	1071.75	1077.77	0.0122	2.1535	2.1657	38
39	0.11707	0.23835	0.01602	2535.86	2535.88	7.03	1071.18	1078.21	0.0142	2.1481	2.1623	39
40	0.12172	0.24783	0.01602	2443.67	2443.69	8.03	1070.62	1078.65	0.0162	2.1426	2.1589	40
41	0.12654	0.25765	0.01602	2355.22	2355.24	9.04	1070.05	1079.09	0.0182	2.1372	2.1554	41
42	0.13153	0.26780	0.01602	2270.42	2270.43	10.04	1069.48	1079.52	0.0202	2.1318	2.1521	42
43	0.13669	0.27831	0.01602	2189.02	2189.04	11.04	1068.92	1079.96	0.0222	2.1265	2.1487	43
44	0.14203	0.28918	0.01602	2110.92	2110.94	12.05	1068.35	1080.40	0.0242	2.1211	2.1454	44
45	0.14755	0.30042	0.01602	2035.91	2035.92	13.05	1067.79	1080.84	0.0262	2.1158	2.1420	45
46	0.15326	0.31205	0.01602	1963.85	1963.87	14.05	1067.22	1081.28	0.0282	2.1105	2.1387	46
47	0.15917	0.32407	0.01602	1894.71	1894.73	15.06	1066.66	1081.71	0.0302	2.1052	2.1354	47
48	0.16527	0.33650	0.01602	1828.28	1828.30	16.06	1066.09	1082.15	0.0321	2.1000	2.1321	48
49	0.17158	0.34935	0.01602	1764.44	1764.46	17.06	1065.53	1082.59	0.0341	2.0947	2.1288	49
50	0.17811	0.36263	0.01602	1703.18	1703.20	18.06	1064.96	1083.03	0.0361	2.0895	2.1256	50
51	0.18484	0.37635	0.01602	1644.25	1644.26	19.06	1064.40	1083.46	0.0381	2.0843	2.1224	51
52	0.19181	0.39053	0.01603	1587.64	1587.65	20.07	1063.83	1083.90	0.0400	2.0791	2.1191	52

*Extrapolated to represent metastable equilibrium with undercooled liquid.

Table 2-1 Thermodynamic Properties of Water (*Continued*)

(Table 3, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °F	Absolute Pressure		Specific Volume, ft^3/lb$_w$			Specific Enthalpy, Btu/lb$_w$			Specific Entropy, Btu/lb$_w$·°F			Temp., °F
t	p, psi	p, in. Hg	Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	t
53	0.19900	0.40516	0.01603	1533.22	1533.24	21.07	1063.27	1084.34	0.0420	2.0740	2.1159	53
54	0.20643	0.42029	0.01603	1480.89	1480.91	22.07	1062.71	1084.77	0.0439	2.0689	2.1128	54
55	0.21410	0.43591	0.01603	1430.61	1430.62	23.07	1062.14	1085.21	0.0459	2.0637	2.1096	55
56	0.22202	0.45204	0.01603	1382.19	1382.21	24.07	1061.58	1085.65	0.0478	2.0586	2.1064	56
57	0.23020	0.46869	0.01603	1335.65	1335.67	25.07	1061.01	1086.08	0.0497	2.0536	2.1033	57
58	0.23864	0.48588	0.01603	1290.85	1290.87	26.07	1060.45	1086.52	0.0517	2.0485	2.1002	58
59	0.24735	0.50362	0.01603	1247.76	1247.78	27.07	1059.89	1086.96	0.0536	2.0435	2.0971	59
60	0.25635	0.52192	0.01604	1206.30	1206.32	28.07	1059.32	1087.39	0.0555	2.0385	2.0940	60
61	0.26562	0.54081	0.01604	1166.38	1166.40	29.07	1058.76	1087.83	0.0575	2.0334	2.0909	61
62	0.27519	0.56029	0.01604	1127.93	1127.95	30.07	1058.19	1088.27	0.0594	2.0285	2.0878	62
63	0.28506	0.58039	0.01604	1090.94	1090.96	31.07	1057.63	1088.70	0.0613	2.0235	2.0848	63
64	0.29524	0.60112	0.01604	1055.32	1055.33	32.07	1057.07	1089.14	0.0632	2.0186	2.0818	64
65	0.30574	0.62249	0.01604	1020.98	1021.00	33.07	1056.50	1089.57	0.0651	2.0136	2.0787	65
66	0.31656	0.64452	0.01604	987.95	987.97	34.07	1055.94	1090.01	0.0670	2.0087	2.0758	66
67	0.32772	0.66724	0.01605	956.11	956.12	35.07	1055.37	1090.44	0.0689	2.0039	2.0728	67
68	0.33921	0.69065	0.01605	925.44	925.45	36.07	1054.81	1090.88	0.0708	1.9990	2.0698	68
69	0.35107	0.71478	0.01605	895.86	895.87	37.07	1054.24	1091.31	0.0727	1.9941	2.0668	69
70	0.36328	0.73964	0.01605	867.34	867.36	38.07	1053.68	1091.75	0.0746	1.9893	2.0639	70
71	0.37586	0.76526	0.01605	839.87	839.88	39.07	1053.11	1092.18	0.0765	1.9845	2.0610	71
72	0.38882	0.79164	0.01606	813.37	813.39	40.07	1052.55	1092.61	0.0783	1.9797	2.0580	72
73	0.40217	0.81883	0.01606	787.85	787.87	41.07	1051.98	1093.05	0.0802	1.9749	2.0552	73
74	0.41592	0.84682	0.01606	763.19	763.21	42.06	1051.42	1093.48	0.0821	1.9702	2.0523	74
75	0.43008	0.87564	0.01606	739.42	739.44	43.06	1050.85	1093.92	0.0840	1.9654	2.0494	75
76	0.44465	0.90532	0.01606	716.51	726.53	44.06	1050.29	1094.35	0.0858	1.9607	2.0465	76
77	0.45966	0.93587	0.01607	694.38	699.80	45.06	1049.72	1094.78	0.0877	1.9560	2.0437	77
78	0.47510	0.96732	0.01607	673.05	673.06	46.06	1049.16	1095.22	0.0896	1.9513	2.0409	78
79	0.49100	0.99968	0.01607	652.44	652.46	47.06	1048.59	1095.65	0.0914	1.9466	2.0380	79
80	0.50736	1.03298	0.01607	632.54	632.56	48.06	1048.03	1096.08	0.0933	1.9420	2.0352	80
81	0.52419	1.06725	0.01608	613.35	613.37	49.06	1047.46	1096.51	0.0951	1.9373	2.0324	81
82	0.54150	1.10250	0.01608	594.82	594.84	50.05	1046.89	1096.95	0.0970	1.9327	2.0297	82
83	0.55931	1.13877	0.01608	576.90	576.92	51.05	1046.33	1097.38	0.0988	1.9281	2.0269	83
84	0.57763	1.17606	0.01608	559.63	559.65	52.05	1045.76	1097.81	0.1006	1.9235	2.0242	84
85	0.59647	1.21442	0.01609	542.93	542.94	53.05	1045.19	1098.24	0.1025	1.9189	2.0214	85
86	0.61584	1.25385	0.01609	526.80	526.81	54.05	1044.63	1098.67	0.1043	1.9144	2.0187	86
87	0.63575	1.29440	0.01609	511.21	511.22	55.05	1044.06	1099.11	0.1061	1.9098	2.0160	87
88	0.65622	1.33608	0.01609	496.14	496.15	56.05	1043.49	1099.54	0.1080	1.9053	2.0133	88
89	0.67726	1.37892	0.01610	481.60	481.61	57.04	1042.92	1099.97	0.1098	1.9008	2.0106	89
90	0.69889	1.42295	0.01610	467.52	467.53	58.04	1042.36	1100.40	0.1116	1.8963	2.0079	90
91	0.72111	1.46820	0.01610	453.91	453.93	59.04	1041.79	1100.83	0.1134	1.8918	2.0053	91
92	0.74394	1.51468	0.01611	440.76	440.78	60.04	1041.22	1101.26	0.1152	1.8874	2.0026	92
93	0.76740	1.56244	0.01611	428.04	428.06	61.04	1040.65	1101.69	0.1170	1.8829	2.0000	93
94	0.79150	1.61151	0.01611	415.74	415.76	62.04	1040.08	1102.12	0.1188	1.8785	1.9973	94
95	0.81625	1.66189	0.01612	403.84	403.86	63.03	1039.51	1102.55	0.1206	1.8741	1.9947	95
96	0.84166	1.71364	0.01612	392.33	392.34	64.03	1038.95	1102.98	0.1224	1.8697	1.9921	96
97	0.86776	1.76678	0.01612	381.20	381.21	65.03	1038.38	1103.41	0.1242	1.8653	1.9895	97
98	0.89456	1.82134	0.01612	370.42	370.44	66.03	1037.81	1103.84	0.1260	1.8610	1.9870	98
99	0.92207	1.87736	0.01613	359.99	360.01	67.03	1037.24	1104.26	0.1278	1.8566	1.9844	99
100	0.95031	1.93485	0.01613	349.91	349.92	68.03	1036.67	1104.69	0.1296	1.8523	1.9819	100
101	0.97930	1.99387	0.01613	340.14	340.15	69.03	1036.10	1105.12	0.1314	1.8479	1.9793	101
102	1.00904	2.05443	0.01614	330.69	330.71	70.02	1035.53	1105.55	0.1332	1.8436	1.9768	102
103	1.03956	2.11667	0.01614	321.53	321.55	71.02	1034.95	1105.98	0.1349	1.8393	1.9743	103
104	1.07088	2.18034	0.01614	312.67	312.69	72.02	1034.38	1106.40	0.1367	1.8351	1.9718	104
105	1.10301	2.24575	0.01615	304.08	304.10	73.02	1033.81	1106.83	0.1385	1.8308	1.9693	105
106	1.13597	2.31285	0.01615	295.76	295.77	74.02	1033.24	1107.26	0.1402	1.8266	1.9668	106
107	1.16977	2.38168	0.01616	287.71	287.73	75.01	1032.67	1107.68	0.1420	1.8223	1.9643	107
108	1.20444	2.45226	0.01616	279.91	279.92	76.01	1032.10	1108.11	0.1438	1.8181	1.9619	108
109	1.23999	2.52464	0.01616	272.34	272.36	77.01	1031.52	1108.54	0.1455	1.8139	1.9594	109
110	1.27644	2.59885	0.01617	265.02	265.03	78.01	1030.95	1108.96	0.1473	1.8097	1.9570	110
111	1.31381	2.67494	0.01617	257.91	257.93	79.01	1030.38	1109.39	0.1490	1.8055	1.9546	111
112	1.35212	2.75293	0.01617	251.02	251.04	80.01	1029.80	1109.81	0.1508	1.8014	1.9521	112
113	1.39138	2.83288	0.01618	244.36	244.38	81.01	1029.23	1110.24	0.1525	1.7972	1.9497	113
114	1.43162	2.91481	0.01618	237.89	237.90	82.00	1028.66	1110.66	0.1543	1.7931	1.9474	114
115	1.47286	2.99878	0.01619	231.62	231.63	83.00	1028.08	1111.09	0.1560	1.7890	1.9450	115
116	1.51512	3.08481	0.01619	225.53	225.55	84.00	1027.51	1111.51	0.1577	1.7849	1.9426	116
117	1.55842	3.17296	0.01619	219.63	219.65	85.00	1026.93	1111.93	0.1595	1.7808	1.9402	117
118	1.60277	3.26327	0.01620	213.91	213.93	86.00	1026.36	1112.36	0.1612	1.7767	1.9379	118
119	1.64820	3.35577	0.01620	208.36	208.37	87.00	1025.78	1112.78	0.1629	1.7726	1.9356	119
120	1.69474	3.45052	0.01620	202.98	202.99	88.00	1025.20	1113.20	0.1647	1.7686	1.9332	120

Table 2-1 Thermodynamic Properties of Water (*Continued*)

(Table 3, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °F	Absolute Pressure		Specific Volume, ft^3/lb$_w$			Specific Enthalpy, Btu/lb$_w$			Specific Entropy, Btu/lb$_w$·°F			Temp., °F
t	p, psi	p, in. Hg	Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	t
121	1.74240	3.54755	0.01621	197.76	197.76	89.00	1024.63	1113.62	0.1664	1.7645	1.9309	121
122	1.79117	3.64691	0.01621	192.69	192.69	90.00	1024.05	1114.05	0.1681	1.7605	1.9286	122
123	1.84117	3.74863	0.01622	187.78	187.78	90.99	1023.47	1114.47	0.1698	1.7565	1.9263	123
124	1.89233	3.85282	0.01622	182.98	182.99	91.99	1022.90	1114.89	0.1715	1.7525	1.9240	124
125	1.94470	3.95945	0.01623	178.34	178.36	92.99	1022.32	1115.31	0.1732	1.7485	1.9217	125
126	1.99831	4.06860	0.01623	173.85	173.86	93.99	1021.74	1115.73	0.1749	1.7445	1.9195	126
127	2.05318	4.18032	0.01623	169.47	169.49	94.99	1021.16	1116.15	0.1766	1.7406	1.9172	127
128	2.10934	4.29465	0.01624	165.23	165.25	95.99	1020.58	1116.57	0.1783	1.7366	1.9150	128
129	2.16680	4.41165	0.01624	161.11	161.12	96.99	1020.00	1116.99	0.1800	1.7327	1.9127	129
130	2.22560	4.53136	0.01625	157.11	157.12	97.99	1019.42	1117.41	0.1817	1.7288	1.9105	130
131	2.28576	4.65384	0.01625	153.22	153.23	98.99	1018.84	1117.83	0.1834	1.7249	1.9083	131
132	2.34730	4.77914	0.01626	149.44	149.46	99.99	1018.26	1118.25	0.1851	1.7210	1.9061	132
133	2.41025	4.90730	0.01626	145.77	145.78	100.99	1017.68	1118.67	0.1868	1.7171	1.9039	133
134	2.47463	5.03839	0.01627	142.21	142.23	101.99	1017.10	1119.08	0.1885	1.7132	1.9017	134
135	2.54048	5.17246	0.01627	138.74	138.76	102.99	1016.52	1119.50	0.1902	1.7093	1.8995	135
136	2.60782	5.30956	0.01627	135.37	135.39	103.98	1015.93	1119.92	0.1919	1.7055	1.8974	136
137	2.67667	5.44975	0.01628	132.10	132.12	104.98	1015.35	1120.34	0.1935	1.7017	1.8952	137
138	2.74707	5.59308	0.01628	128.92	128.94	105.98	1014.77	1120.75	0.1952	1.6978	1.8930	138
139	2.81903	5.73961	0.01629	125.83	125.85	106.98	1014.18	1121.17	0.1969	1.6940	1.8909	139
140	2.89260	5.88939	0.01629	122.82	122.84	107.98	1013.60	1121.58	0.1985	1.6902	1.8888	140
141	2.96780	6.04250	0.01630	119.90	119.92	108.98	1013.01	1122.00	0.2002	1.6864	1.8867	141
142	3.04465	6.19897	0.01630	117.05	117.07	109.98	1012.43	1122.41	0.2019	1.6827	1.8845	142
143	3.12320	6.35888	0.01631	114.29	114.31	110.98	1011.84	1122.83	0.2035	1.6789	1.8824	143
144	3.20345	6.52229	0.01631	111.60	111.62	111.98	1011.26	1123.24	0.2052	1.6752	1.8803	144
145	3.28546	6.68926	0.01632	108.99	109.00	112.98	1010.67	1123.66	0.2068	1.6714	1.8783	145
146	3.36924	6.85984	0.01632	106.44	106.45	113.98	1010.09	1124.07	0.2085	1.6677	1.8762	146
147	3.45483	7.03410	0.01633	103.96	103.98	114.98	1009.50	1124.48	0.2101	1.6640	1.8741	147
148	3.54226	7.21211	0.01633	101.55	101.57	115.98	1008.91	1124.89	0.2118	1.6603	1.8721	148
149	3.63156	7.39393	0.01634	99.21	99.22	116.98	1008.32	1125.31	0.2134	1.6566	1.8700	149
150	3.72277	7.57962	0.01634	96.93	96.94	117.98	1007.73	1125.72	0.2151	1.6529	1.8680	150
151	3.81591	7.76925	0.01635	94.70	94.72	118.99	1007.14	1126.13	0.2167	1.6492	1.8659	151
152	3.91101	7.96289	0.01635	92.54	92.56	119.99	1006.55	1126.54	0.2184	1.6455	1.8639	152
153	4.00812	8.16061	0.01636	90.44	90.46	120.99	1005.96	1126.95	0.2200	1.6419	1.8619	153
154	4.10727	8.36247	0.01636	88.39	88.41	121.99	1005.37	1127.36	0.2216	1.6383	1.8599	154
155	4.20848	8.56854	0.01637	86.40	86.41	122.99	1004.78	1127.77	0.2233	1.6346	1.8579	155
156	4.31180	8.77890	0.01637	84.45	84.47	123.99	1004.19	1128.18	0.2249	1.6310	1.8559	156
157	4.41725	8.99360	0.01638	82.56	82.58	124.99	1003.60	1128.59	0.2265	1.6274	1.8539	157
158	4.52488	9.21274	0.01638	80.72	80.73	125.99	1003.00	1128.99	0.2281	1.6238	1.8519	158
159	4.63472	9.43637	0.01639	78.92	78.94	126.99	1002.41	1129.40	0.2297	1.6202	1.8500	159
160	4.7468	9.6646	0.01639	77.175	77.192	127.99	1001.82	1129.81	0.2314	1.6167	1.8480	160
161	4.8612	9.8974	0.01640	75.471	75.488	128.99	1001.22	1130.22	0.2330	1.6131	1.8461	161
162	4.9778	10.1350	0.01640	73.812	73.829	130.00	1000.63	1130.62	0.2346	1.6095	1.8441	162
163	5.0969	10.3774	0.01641	72.196	72.213	131.00	1000.03	1131.03	0.2362	1.6060	1.8422	163
164	5.2183	10.6246	0.01642	70.619	70.636	132.00	999.43	1131.43	0.2378	1.6025	1.8403	164
165	5.3422	10.8768	0.01642	69.084	69.101	133.00	998.84	1131.84	0.2394	1.5989	1.8383	165
166	5.4685	11.1340	0.01643	67.587	67.604	134.00	998.24	1132.24	0.2410	1.5954	1.8364	166
167	5.5974	11.3963	0.01643	66.130	66.146	135.00	997.64	1132.64	0.2426	1.5919	1.8345	167
168	5.7287	11.6638	0.01644	64.707	64.723	136.01	997.04	1133.05	0.2442	1.5884	1.8326	168
169	5.8627	11.9366	0.01644	63.320	63.336	137.01	996.44	1133.45	0.2458	1.5850	1.8308	169
170	5.9993	12.2148	0.01645	61.969	61.986	138.01	995.84	1133.85	0.2474	1.5815	1.8289	170
171	6.1386	12.4983	0.01646	60.649	60.666	139.01	995.24	1134.25	0.2490	1.5780	1.8270	171
172	6.2806	12.7874	0.01646	59.363	59.380	140.01	994.64	1134.66	0.2506	1.5746	1.8251	172
173	6.4253	13.0821	0.01647	58.112	58.128	141.02	994.04	1135.06	0.2521	1.5711	1.8233	173
174	6.5729	13.3825	0.01647	56.887	56.904	142.02	993.44	1135.46	0.2537	1.5677	1.8214	174
175	6.7232	13.6886	0.01648	55.694	55.711	143.02	992.83	1135.86	0.2553	1.5643	1.8196	175
176	6.8765	14.0006	0.01648	54.532	54.549	144.02	992.23	1136.26	0.2569	1.5609	1.8178	176
177	7.0327	14.3186	0.01649	53.397	53.414	145.03	991.63	1136.65	0.2585	1.5575	1.8159	177
178	7.1918	14.6426	0.01650	52.290	52.307	146.03	991.02	1137.05	0.2600	1.5541	1.8141	178
179	7.3539	14.9727	0.01650	51.210	51.226	147.03	990.42	1137.45	0.2616	1.5507	1.8123	179
180	7.5191	15.3091	0.01651	50.155	50.171	148.04	989.81	1137.85	0.2632	1.5473	1.8105	180
181	7.6874	15.6518	0.01651	49.126	49.143	149.04	989.20	1138.24	0.2647	1.5440	1.8087	181
182	7.8589	16.0008	0.01652	48.122	48.138	150.04	988.60	1138.64	0.2663	1.5406	1.8069	182
183	8.0335	16.3564	0.01653	47.142	47.158	151.05	987.99	1139.03	0.2679	1.5373	1.8051	183
184	8.2114	16.7185	0.01653	46.185	46.202	152.05	987.38	1139.43	0.2694	1.5339	1.8034	184
185	8.3926	17.0874	0.01654	45.251	45.267	153.05	986.77	1139.82	0.2710	1.5306	1.8016	185
186	8.5770	17.4630	0.01654	44.339	44.356	154.06	986.16	1140.22	0.2725	1.5273	1.7998	186
187	8.7649	17.8455	0.01655	43.448	43.465	155.06	985.55	1140.61	0.2741	1.5240	1.7981	187
188	8.9562	18.2350	0.01656	42.579	42.595	156.07	984.94	1141.00	0.2756	1.5207	1.7963	188
189	9.1510	18.6316	0.01656	41.730	41.746	157.07	984.32	1141.39	0.2772	1.5174	1.7946	189

Table 2-1 Thermodynamic Properties of Water (*Continued*)

(Table 3, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °F *t*	Absolute Pressure		Specific Volume, ft^3/lb_w			Specific Enthalpy, Btu/lb_w			Specific Entropy, $\text{Btu/lb}_w\cdot°\text{F}$			Temp., °F *t*
	p, psi	*p*, in. Hg	Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
190	9.3493	19.0353	0.01657	40.901	40.918	158.07	983.71	1141.78	0.2787	1.5141	1.7929	190
191	9.5512	19.4464	0.01658	40.092	40.108	159.08	983.10	1142.18	0.2803	1.5109	1.7911	191
192	9.7567	19.8648	0.01658	39.301	39.317	160.08	982.48	1142.57	0.2818	1.5076	1.7894	192
193	9.9659	20.2907	0.01659	38.528	38.544	161.09	981.87	1142.95	0.2834	1.5043	1.7877	193
194	10.1788	20.7242	0.01659	37.774	37.790	162.09	981.25	1143.34	0.2849	1.5011	1.7860	194
195	10.3955	21.1653	0.01660	37.035	37.052	163.10	980.63	1143.73	0.2864	1.4979	1.7843	195
196	10.6160	21.6143	0.01661	36.314	36.331	164.10	980.02	1144.12	0.2880	1.4946	1.7826	196
197	10.8404	22.0712	0.01661	35.611	35.628	165.11	979.40	1144.51	0.2895	1.4914	1.7809	197
198	11.0687	22.5361	0.01662	34.923	34.940	166.11	978.78	1144.89	0.2910	1.4882	1.7792	198
199	11.3010	23.0091	0.01663	34.251	34.268	167.12	978.16	1145.28	0.2926	1.4850	1.7776	199
200	11.5374	23.4904	0.01663	33.594	33.610	168.13	977.54	1145.66	0.2941	1.4818	1.7759	200
201	11.7779	23.9800	0.01664	32.951	32.968	169.13	976.92	1146.05	0.2956	1.4786	1.7742	201
202	12.0225	24.4780	0.01665	32.324	32.340	170.14	976.29	1146.43	0.2971	1.4755	1.7726	202
203	12.2713	24.9847	0.01665	31.710	31.726	171.14	975.67	1146.81	0.2986	1.4723	1.7709	203
204	12.5244	25.5000	0.01666	31.110	31.127	172.15	975.05	1147.20	0.3002	1.4691	1.7693	204
205	12.7819	26.0241	0.01667	30.523	30.540	173.16	974.42	1147.58	0.3017	1.4660	1.7677	205
206	13.0436	26.5571	0.01667	29.949	29.965	174.16	973.80	1147.96	0.3032	1.4628	1.7660	206
207	13.3099	27.0991	0.01668	29.388	29.404	175.17	973.17	1148.34	0.3047	1.4597	1.7644	207
208	13.5806	27.6503	0.01669	28.839	28.856	176.18	972.54	1148.72	0.3062	1.4566	1.7628	208
209	13.8558	28.2108	0.01669	28.303	28.319	177.18	971.92	1149.10	0.3077	1.4535	1.7612	209
210	14.1357	28.7806	0.01670	27.778	27.795	178.19	971.29	1149.48	0.3092	1.4503	1.7596	210
212	14.7096	29.9489	0.01671	26.763	26.780	180.20	970.03	1150.23	0.3122	1.4442	1.7564	212
214	15.3025	31.1563	0.01673	25.790	25.807	182.22	968.76	1150.98	0.3152	1.4380	1.7532	214
216	15.9152	32.4036	0.01674	24.861	24.878	184.24	967.50	1151.73	0.3182	1.4319	1.7501	216
218	16.5479	33.6919	0.01676	23.970	23.987	186.25	966.23	1152.48	0.3212	1.4258	1.7469	218
220	17.2013	35.0218	0.01677	23.118	23.134	188.27	964.95	1153.22	0.3241	1.4197	1.7438	220
222	17.8759	36.3956	0.01679	22.299	22.316	190.29	963.67	1153.96	0.3271	1.4136	1.7407	222
224	18.5721	37.8131	0.01680	21.516	21.533	192.31	962.39	1154.70	0.3301	1.4076	1.7377	224
226	19.2905	39.2758	0.01682	20.765	20.782	194.33	961.11	1155.43	0.3330	1.4016	1.7347	226
228	20.0316	40.7848	0.01683	20.045	20.062	196.35	959.82	1156.16	0.3359	1.3957	1.7316	228
230	20.7961	42.3412	0.01684	19.355	19.372	198.37	958.52	1156.89	0.3389	1.3898	1.7287	230
232	21.5843	43.9461	0.01686	18.692	18.709	200.39	957.22	1157.62	0.3418	1.3839	1.7257	232
234	22.3970	45.6006	0.01688	18.056	18.073	202.41	955.92	1158.34	0.3447	1.3780	1.7227	234
236	23.2345	47.3060	0.01689	17.446	17.463	204.44	954.62	1159.06	0.3476	1.3722	1.7198	236
238	24.0977	49.0633	0.01691	16.860	16.877	206.46	953.31	1159.77	0.3505	1.3664	1.7169	238
240	24.9869	50.8738	0.01692	16.298	16.314	208.49	952.00	1160.48	0.3534	1.3606	1.7140	240
242	25.9028	52.7386	0.01694	15.757	15.774	210.51	950.68	1161.19	0.3563	1.3548	1.7111	242
244	26.8461	54.6591	0.01695	15.238	15.255	212.54	949.35	1161.90	0.3592	1.3491	1.7083	244
246	27.8172	56.6364	0.01697	14.739	14.756	214.57	948.03	1162.60	0.3621	1.3434	1.7055	246
248	28.8169	58.6717	0.01698	14.259	14.276	216.60	946.70	1163.29	0.3649	1.3377	1.7026	248
250	29.8457	60.7664	0.01700	13.798	13.815	218.63	945.36	1163.99	0.3678	1.3321	1.6998	250
252	30.9043	62.9218	0.01702	13.355	13.372	220.66	944.02	1164.68	0.3706	1.3264	1.6971	252
254	31.9934	65.1391	0.01703	12.928	12.945	222.69	942.68	1165.37	0.3735	1.3208	1.6943	254
256	33.1135	67.4197	0.01705	12.526	12.147	226.73	939.99	1166.72	0.3764	1.3153	1.6916	256
258	34.2653	69.7649	0.01707	12.123	12.140	226.76	939.97	1166.73	0.3792	1.3097	1.6889	258
260	35.4496	72.1760	0.01708	11.742	11.759	228.79	938.61	1167.40	0.3820	1.3042	1.6862	260
262	36.6669	74.6545	0.01710	11.376	11.393	230.83	937.25	1168.08	0.3848	1.2987	1.6835	262
264	37.9180	77.2017	0.01712	11.024	11.041	232.87	935.88	1168.74	0.3876	1.2932	1.6808	264
266	39.2035	79.8190	0.01714	10.684	10.701	234.90	934.50	1169.41	0.3904	1.2877	1.6781	266
268	40.5241	82.5078	0.01715	10.357	10.374	236.94	933.12	1170.07	0.3932	1.2823	1.6755	268
270	41.8806	85.2697	0.01717	10.042	10.059	238.98	931.74	1170.72	0.3960	1.2769	1.6729	270
272	43.2736	88.1059	0.01719	9.737	9.755	241.03	930.35	1171.38	0.3988	1.2715	1.6703	272
274	44.7040	91.0181	0.01721	9.445	9.462	243.07	928.95	1172.02	0.4016	1.2661	1.6677	274
276	46.1723	94.0076	0.01722	9.162	9.179	245.11	927.55	1172.67	0.4044	1.2608	1.6651	276
278	47.6794	97.0761	0.01724	8.890	8.907	247.16	926.15	1173.31	0.4071	1.2554	1.6626	278
280	49.2260	100.2250	0.01726	8.627	8.644	249.20	924.74	1173.94	0.4099	1.2501	1.6600	280
282	50.8128	103.4558	0.01728	8.373	8.390	251.25	923.32	1174.57	0.4127	1.2448	1.6575	282
284	52.4406	106.7701	0.01730	8.128	8.146	253.30	921.90	1175.20	0.4154	1.2396	1.6550	284
286	54.1103	110.1695	0.01731	7.892	7.910	255.35	920.47	1175.82	0.4182	1.2343	1.6525	286
288	55.8225	113.6556	0.01733	7.664	7.681	257.40	919.03	1176.44	0.4209	1.2291	1.6500	288
290	57.5780	117.2299	0.01735	7.444	7.461	259.45	917.59	1177.05	0.4236	1.2239	1.6476	290
292	59.3777	120.8941	0.01737	7.231	7.248	261.51	916.15	1177.66	0.4264	1.2187	1.6451	292
294	61.2224	124.6498	0.01739	7.026	7.043	263.56	914.69	1178.26	0.4291	1.2136	1.6427	294
296	63.1128	128.4987	0.01741	6.827	6.844	265.62	913.24	1178.86	0.4318	1.2084	1.6402	296
298	65.0498	132.4425	0.01743	6.635	6.652	267.68	911.77	1179.45	0.4345	1.2033	1.6378	298
300	67.0341	136.4827	0.01745	6.450	6.467	269.74	910.30	1180.04	0.4372	1.1982	1.6354	300

The specific volume of a substance having a given quality can be found by using the definition of quality. **Quality** is defined as the ratio of the mass of vapor to total mass of liquid plus vapor when a substance is in a saturation state. Consider a mass of 1 kg having a quality x. The specific volume is the sum of the volume of the liquid and the volume of the vapor. The volume of the liquid is $(1 - x)v_f$, and the volume of the vapor is xv_g. Therefore, the specific volume v is

$$v = xv_g + (1 - x)v_f \tag{2-5}$$

Since $v_f + v_{fg} = v_g$, Equation 2-5 can also be written in the following form:

$$v = v_f + xv_{fg} \tag{2-6}$$

The same procedure is followed for determining the enthalpy and the entropy for quality conditions:

$$h = xh_g + (1 - x)h_f \tag{2-7}$$

$$s = xs_g + (1 - x)s_f \tag{2-8}$$

Internal energy can then be obtained from the definition of enthalpy as $u = h - pv$.

If the substance is a compressed or subcooled liquid, the thermodynamic properties of specific volume, enthalpy, internal energy, and entropy are strongly temperature dependent (rather than pressure dependent). If compressed liquid tables are unavailable, they may be approximated by the corresponding values for saturated liquid (v_f, h_f, u_f, and s_f) at the existing temperature.

In the superheat region, thermodynamic properties must be obtained from superheat tables or a plot of the thermodynamic properties, called a Mollier diagram (Figure 2-4).

The thermodynamic and transport properties of the refrigerants used in vapor compression systems are found in similar tables typified by Table 2-2, which is a section of the R-134a property tables from Chapter 30 of the 2017 *ASHRAE Handbook—Fundamentals*. However, for these refrigerants the common Mollier plot is the *p-h* diagram as illustrated in Figure 2-5.

For fluids used in absorption refrigeration systems, the thermodynamic properties are commonly found on a different type of plot—the enthalpy-concentration diagram, as illustrated in Figure 2-6 for aqua-ammonia and in Figure 2-7 for lithium-bromide/water.

2.2.11 Property Equations for Ideal Gases

An ideal gas is defined as a gas at sufficiently low density so that intermolecular forces are negligible. As a result, an ideal gas has the equation of state

$$pv = RT \tag{2-9}$$

For an ideal gas, the internal energy is a function of temperature only, which means that regardless of the pressure, an ideal gas at a given temperature has a certain definite specific internal energy u.

The relation between the internal energy u and the temperature can be established by using the definition of constant-volume specific heat given by

$$c_v = (\partial u / \partial T)_v$$

Since the internal energy of an ideal gas is not a function of volume, an ideal gas can be written as

$$c_v = du/dT$$

$$du = c_v\, dt \tag{2-10}$$

This equation is always valid for an ideal gas regardless of the kind of process considered.

From the definition of enthalpy and the equation of state of an ideal gas, it follows that

$$h = u + pv = u + RT$$

Since R is a constant and u is a function of temperature only, the enthalpy h of an ideal gas is also a function of temperature only.

The relation between enthalpy and temperature is found from the constant pressure specific heat as defined by

$$c_p = (\partial h / \partial T)_p$$

Since the enthalpy of an ideal gas is a function of the temperature only, and is independent of the pressure, it follows that

$$c_p = dh/dT$$

$$dh = c_p\, dT \tag{2-11}$$

This equation is always valid for an ideal gas regardless of the kind of process considered.

Entropy, however, remains a function of both temperature and pressure, and is given by the equation

$$ds = c_p(dT/T) - R(dp/p) \tag{2-12}$$

where c_p is frequently treated as being constant.

The ratio of heat capacities is often denoted by

$$k = c_p/c_v \tag{2-13}$$

and is a useful quantity in calculations for ideal gases. Ideal gas values for some common gases are listed in Table 2-3.

No real gas exactly satisfies these equations over any finite range of temperature and pressure. However, all real gases approach ideal behavior at low pressures, and in the limit as $p \to 0$ do in fact meet the above requirements.

Thus, in solving problems, ideal behavior is assumed in two cases. First, at very low pressures, ideal gas behavior can be assumed with good accuracy, regardless of the temperature. Second, at temperatures that are double the critical temperature or above (the critical temperature of nitrogen is 126 K), ideal gas behavior can be assumed with good accuracy to pressures of at least 1000 lb_f/in^2 (7000 kPa). In the superheated vapor region, when the temperature is less than twice the critical temperature and the pressure is above a very

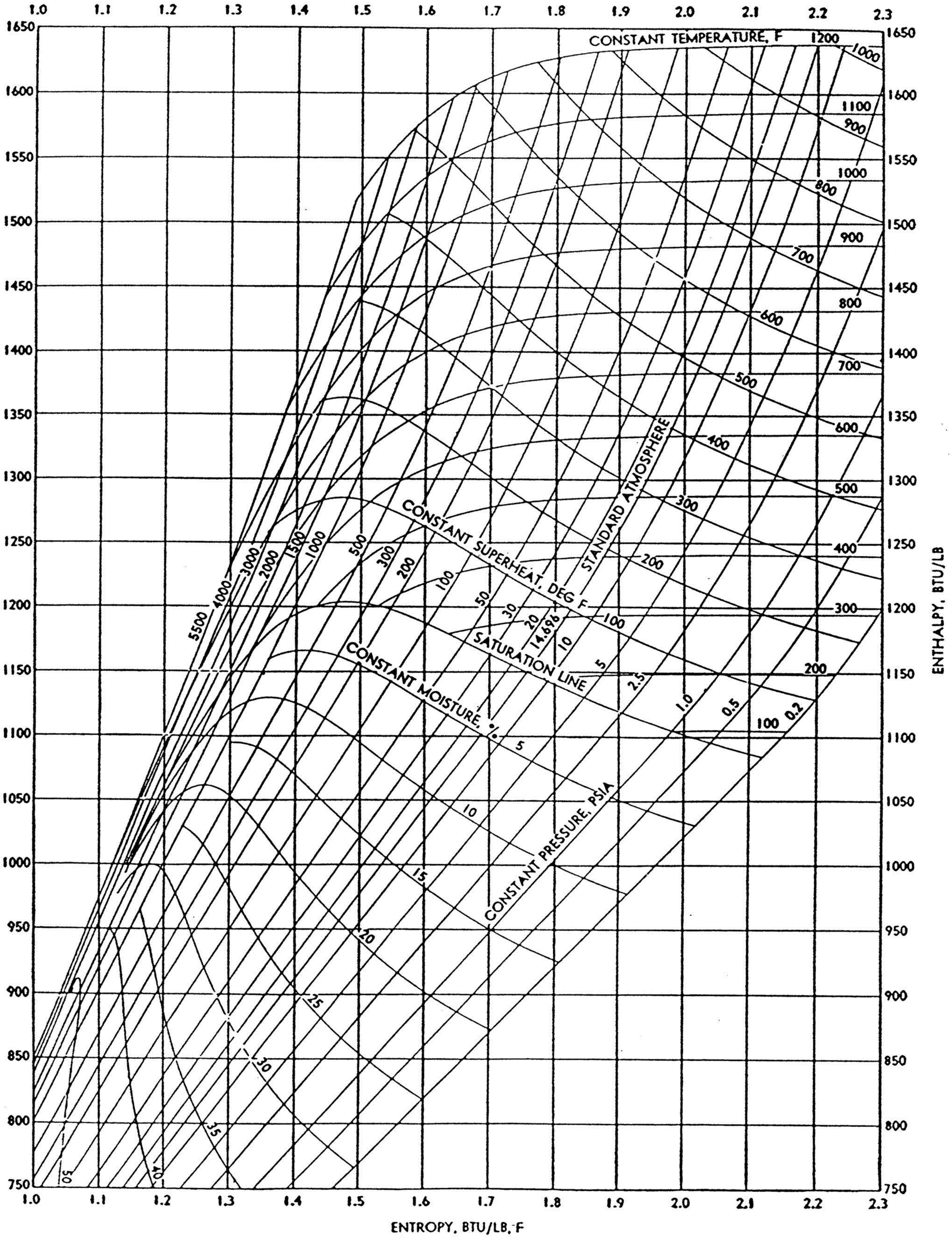


Fig. 2-4 Mollier (h,s) Diagram for Steam

Table 2-2 Refrigerant 134a Properties of Saturated Liquid and Saturated Vapor

(Table Refrigerant 134a, Chapter 30, 2017 ASHRAE Handbook—Fundamentals)

Temp.,* °F	Pres-sure, psia	Density, lb/ft^3	Volume, ft^3/lb	Enthalpy, Btu/lb		Entropy, Btu/lb·°F		Specific Heat c_p, Btu/lb·°F		c_p/c_v	Vel. of Sound, ft/s		Viscosity, lb$_m$/ft·h		Thermal Cond., Btu/h·ft·°F		Surface Tension, dyne/cm	Temp.,* °F
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor		
–153.94a	0.057	99.33	568.59	–32.992	80.362	–0.09154	0.27923	0.2829	0.1399	1.1637	3674.	416.0	5.262	0.0156	0.0840	0.00178	28.07	–153.94
–150.00	0.072	98.97	452.12	–31.878	80.907	–0.08791	0.27629	0.2830	0.1411	1.1623	3638.	418.3	4.790	0.0159	0.0832	0.00188	27.69	–150.00
–140.00	0.129	98.05	260.63	–29.046	82.304	–0.07891	0.26941	0.2834	0.1443	1.1589	3545.	424.2	3.880	0.0164	0.0813	0.00214	26.74	–140.00
–130.00	0.221	97.13	156.50	–26.208	83.725	–0.07017	0.26329	0.2842	0.1475	1.1559	3452.	429.9	3.238	0.0170	0.0794	0.00240	25.79	–130.00
–120.00	0.365	96.20	97.481	–23.360	85.168	–0.06166	0.25784	0.2853	0.1508	1.1532	3360.	435.5	2.762	0.0176	0.0775	0.00265	24.85	–120.00
–110.00	0.583	95.27	62.763	–20.500	86.629	–0.05337	0.25300	0.2866	0.1540	1.1509	3269.	440.8	2.396	0.0182	0.0757	0.00291	23.92	–110.00
–100.00	0.903	94.33	41.637	–17.626	88.107	–0.04527	0.24871	0.2881	0.1573	1.1490	3178.	446.0	2.105	0.0187	0.0739	0.00317	22.99	–100.00
–90.00	1.359	93.38	28.381	–14.736	89.599	–0.03734	0.24490	0.2898	0.1607	1.1475	3087.	450.9	1.869	0.0193	0.0722	0.00343	22.07	–90.00
–80.00	1.993	92.42	19.825	–11.829	91.103	–0.02959	0.24152	0.2916	0.1641	1.1465	2998.	455.6	1.673	0.0199	0.0705	0.00369	21.16	–80.00
–75.00	2.392	91.94	16.711	–10.368	91.858	–0.02577	0.23998	0.2925	0.1658	1.1462	2954.	457.8	1.587	0.0201	0.0696	0.00382	20.71	–75.00
–70.00	2.854	91.46	14.161	–8.903	92.614	–0.02198	0.23854	0.2935	0.1676	1.1460	2909.	460.0	1.509	0.0204	0.0688	0.00395	20.26	–70.00
–65.00	3.389	90.97	12.060	–7.432	93.372	–0.01824	0.23718	0.2945	0.1694	1.1459	2866.	462.1	1.436	0.0207	0.0680	0.00408	19.81	–65.00
–60.00	4.002	90.49	10.321	–5.957	94.131	–0.01452	0.23590	0.2955	0.1713	1.1460	2822.	464.1	1.369	0.0210	0.0671	0.00420	19.36	–60.00
–55.00	4.703	90.00	8.8733	–4.476	94.890	–0.01085	0.23470	0.2965	0.1731	1.1462	2778.	466.0	1.306	0.0212	0.0663	0.00433	18.92	–55.00
–50.00	5.501	89.50	7.6621	–2.989	95.650	–0.00720	0.23358	0.2976	0.1751	1.1466	2735.	467.8	1.248	0.0215	0.0655	0.00446	18.47	–50.00
–45.00	6.406	89.00	6.6438	–1.498	96.409	–0.00358	0.23252	0.2987	0.1770	1.1471	2691.	469.6	1.193	0.0218	0.0647	0.00460	18.03	–45.00
–40.00	7.427	88.50	5.7839	0.000	97.167	0.00000	0.23153	0.2999	0.1790	1.1478	2648.	471.2	1.142	0.0221	0.0639	0.00473	17.60	–40.00
–35.00	8.576	88.00	5.0544	1.503	97.924	0.00356	0.23060	0.3010	0.1811	1.1486	2605.	472.8	1.095	0.0223	0.0632	0.00486	17.16	–35.00
–30.00	9.862	87.49	4.4330	3.013	98.679	0.00708	0.22973	0.3022	0.1832	1.1496	2563.	474.2	1.050	0.0226	0.0624	0.00499	16.73	–30.00
–25.00	11.299	86.98	3.9014	4.529	99.433	0.01058	0.22892	0.3035	0.1853	1.1508	2520.	475.6	1.007	0.0229	0.0616	0.00512	16.30	–25.00
–20.00	12.898	86.47	3.4449	6.051	100.184	0.01406	0.22816	0.3047	0.1875	1.1521	2477.	476.8	0.968	0.0231	0.0608	0.00525	15.87	–20.00
–15.00	14.671	85.95	3.0514	7.580	100.932	0.01751	0.22744	0.3060	0.1898	1.1537	2435.	477.9	0.930	0.0234	0.0601	0.00538	15.44	–15.00
–14.93b	14.696	85.94	3.0465	7.600	100.942	0.01755	0.22743	0.3061	0.1898	1.1537	2434.	477.9	0.929	0.0234	0.0601	0.00538	15.44	–14.93
–10.00	16.632	85.43	2.7109	9.115	101.677	0.02093	0.22678	0.3074	0.1921	1.1554	2393.	478.9	0.894	0.0237	0.0593	0.00552	15.02	–10.00
–5.00	18.794	84.90	2.4154	10.657	102.419	0.02433	0.22615	0.3088	0.1945	1.1573	2350.	479.8	0.860	0.0240	0.0586	0.00565	14.60	–5.00
0.00	21.171	84.37	2.1579	12.207	103.156	0.02771	0.22557	0.3102	0.1969	1.1595	2308.	480.5	0.828	0.0242	0.0578	0.00578	14.18	0.00
5.00	23.777	83.83	1.9330	13.764	103.889	0.03107	0.22502	0.3117	0.1995	1.1619	2266.	481.1	0.798	0.0245	0.0571	0.00592	13.76	5.00
10.00	26.628	83.29	1.7357	15.328	104.617	0.03440	0.22451	0.3132	0.2021	1.1645	2224.	481.6	0.769	0.0248	0.0564	0.00605	13.35	10.00
15.00	29.739	82.74	1.5623	16.901	105.339	0.03772	0.22403	0.3147	0.2047	1.1674	2182.	482.0	0.741	0.0250	0.0556	0.00619	12.94	15.00
20.00	33.124	82.19	1.4094	18.481	106.056	0.04101	0.22359	0.3164	0.2075	1.1705	2140.	482.2	0.715	0.0253	0.0549	0.00632	12.53	20.00
25.00	36.800	81.63	1.2742	20.070	106.767	0.04429	0.22317	0.3181	0.2103	1.1740	2098.	482.2	0.689	0.0256	0.0542	0.00646	12.12	25.00
30.00	40.784	81.06	1.1543	21.667	107.471	0.04755	0.22278	0.3198	0.2132	1.1777	2056.	482.2	0.665	0.0258	0.0535	0.00660	11.72	30.00
35.00	45.092	80.49	1.0478	23.274	108.167	0.05079	0.22241	0.3216	0.2163	1.1818	2014.	481.9	0.642	0.0261	0.0528	0.00674	11.32	35.00
40.00	49.741	79.90	0.9528	24.890	108.856	0.05402	0.22207	0.3235	0.2194	1.1862	1973.	481.5	0.620	0.0264	0.0521	0.00688	10.92	40.00
45.00	54.749	79.32	0.8680	26.515	109.537	0.05724	0.22174	0.3255	0.2226	1.1910	1931.	481.0	0.598	0.0267	0.0514	0.00703	10.53	45.00
50.00	60.134	78.72	0.7920	28.150	110.209	0.06044	0.22144	0.3275	0.2260	1.1961	1889.	480.3	0.578	0.0270	0.0507	0.00717	10.14	50.00
55.00	65.913	78.11	0.7238	29.796	110.871	0.06362	0.22115	0.3297	0.2294	1.2018	1847.	479.4	0.558	0.0273	0.0500	0.00732	9.75	55.00
60.00	72.105	77.50	0.6625	31.452	111.524	0.06680	0.22088	0.3319	0.2331	1.2079	1805.	478.3	0.539	0.0275	0.0493	0.00747	9.36	60.00
65.00	78.729	76.87	0.6072	33.120	112.165	0.06996	0.22062	0.3343	0.2368	1.2145	1763.	477.0	0.520	0.0278	0.0486	0.00762	8.98	65.00
70.00	85.805	76.24	0.5572	34.799	112.796	0.07311	0.22037	0.3368	0.2408	1.2217	1721.	475.6	0.503	0.0281	0.0479	0.00777	8.60	70.00
75.00	93.351	75.59	0.5120	36.491	113.414	0.07626	0.22013	0.3394	0.2449	1.2296	1679.	474.0	0.485	0.0284	0.0472	0.00793	8.23	75.00
80.00	101.39	74.94	0.4710	38.195	114.019	0.07939	0.21989	0.3422	0.2492	1.2382	1636.	472.2	0.469	0.0287	0.0465	0.00809	7.86	80.00
85.00	109.93	74.27	0.4338	39.913	114.610	0.08252	0.21966	0.3451	0.2537	1.2475	1594.	470.1	0.453	0.0291	0.0458	0.00825	7.49	85.00
90.00	119.01	73.58	0.3999	41.645	115.186	0.08565	0.21944	0.3482	0.2585	1.2578	1551.	467.9	0.437	0.0294	0.0451	0.00842	7.13	90.00
95.00	128.65	72.88	0.3690	43.392	115.746	0.08877	0.21921	0.3515	0.2636	1.2690	1509.	465.4	0.422	0.0297	0.0444	0.00860	6.77	95.00
100.00	138.85	72.17	0.3407	45.155	116.289	0.09188	0.21898	0.3551	0.2690	1.2813	1466.	462.7	0.407	0.0301	0.0437	0.00878	6.41	100.00
105.00	149.65	71.44	0.3148	46.934	116.813	0.09500	0.21875	0.3589	0.2747	1.2950	1423.	459.8	0.393	0.0304	0.0431	0.00897	6.06	105.00
110.00	161.07	70.69	0.2911	48.731	117.317	0.09811	0.21851	0.3630	0.2809	1.3101	1380.	456.7	0.378	0.0308	0.0424	0.00916	5.71	110.00
115.00	173.14	69.93	0.2693	50.546	117.799	0.10123	0.21826	0.3675	0.2875	1.3268	1337.	453.2	0.365	0.0312	0.0417	0.00936	5.36	115.00
120.00	185.86	69.14	0.2493	52.382	118.258	0.10435	0.21800	0.3723	0.2948	1.3456	1294.	449.6	0.351	0.0316	0.0410	0.00958	5.03	120.00
125.00	199.28	68.32	0.2308	54.239	118.690	0.10748	0.21772	0.3775	0.3026	1.3666	1250.	445.6	0.338	0.0320	0.0403	0.00981	4.69	125.00
130.00	213.41	67.49	0.2137	56.119	119.095	0.11062	0.21742	0.3833	0.3112	1.3903	1206.	441.4	0.325	0.0324	0.0396	0.01005	4.36	130.00
135.00	228.28	66.62	0.1980	58.023	119.468	0.11376	0.21709	0.3897	0.3208	1.4173	1162.	436.8	0.313	0.0329	0.0389	0.01031	4.04	135.00
140.00	243.92	65.73	0.1833	59.954	119.807	0.11692	0.21673	0.3968	0.3315	1.4481	1117.	432.0	0.301	0.0334	0.0382	0.01058	3.72	140.00
145.00	260.36	64.80	0.1697	61.915	120.108	0.12010	0.21634	0.4048	0.3435	1.4837	1072.	426.8	0.288	0.0339	0.0375	0.01089	3.40	145.00
150.00	277.61	63.83	0.1571	63.908	120.366	0.12330	0.21591	0.4138	0.3571	1.5250	1027.	421.2	0.276	0.0344	0.0368	0.01122	3.09	150.00
155.00	295.73	62.82	0.1453	65.936	120.576	0.12653	0.21542	0.4242	0.3729	1.5738	980.	415.3	0.264	0.0350	0.0361	0.01158	2.79	155.00
160.00	314.73	61.76	0.1343	68.005	120.731	0.12979	0.21488	0.4362	0.3914	1.6318	934.	409.1	0.253	0.0357	0.0354	0.01199	2.50	160.00
165.00	334.65	60.65	0.1239	70.118	120.823	0.13309	0.21426	0.4504	0.4133	1.7022	886.	402.4	0.241	0.0364	0.0346	0.01245	2.21	165.00
170.00	355.53	59.47	0.1142	72.283	120.842	0.13644	0.21356	0.4675	0.4400	1.7889	837.	395.3	0.229	0.0372	0.0339	0.01297	1.93	170.00
175.00	377.41	58.21	0.1051	74.509	120.773	0.13985	0.21274	0.4887	0.4733	1.8984	786.	387.7	0.218	0.0381	0.0332	0.01358	1.66	175.00
180.00	400.34	56.86	0.0964	76.807	120.598	0.14334	0.21180	0.5156	0.5159	2.0405	734.	379.6	0.206	0.0391	0.0325	0.01430	1.39	180.00
185.00	424.36	55.38	0.0881	79.193	120.294	0.14693	0.21069	0.5512	0.5729	2.2321	680.	371.0	0.194	0.0403	0.0318	0.01516	1.14	185.00
190.00	449.52	53.76	0.0801	81.692	119.822	0.15066	0.20935	0.6012	0.6532	2.5041	624.	361.8	0.182	0.0417	0.0311	0.01623	0.90	190.00
195.00	475.91	51.91	0.0724	84.343	119.123	0.15459	0.20771	0.6768	0.7751	2.9192	565.	352.0	0.169	0.0435	0.0304	0.01760	0.67	195.00
200.00	503.59	49.76	0.0647	87.214	118.097	0.15880	0.20562	0.8062	0.9835	3.6309	502.	341.3	0.155	0.0457	0.0300	0.01949	0.45	200.00
205.00	532.68	47.08	0.0567	90.454	116.526	0.16353	0.20275	1.083	1.425	5.136	436.	329.4	0.140	0.0489	0.0300	0.02240	0.26	205.00
210.00	563.35	43.20	0.0477	94.530	113.746	0.16945	0.19814	2.113	3.008	10.512	363.	315.5	0.120	0.0543	0.0316	0.02848	0.09	210.00
213.91c	588.75	31.96	0.0313	103.894	103.894	0.18320	0.18320	•		•	0.	0.0	–	–	•	•	0.00	213.91

*Temperatures are on the ITS-90 scale a = triple point b = normal boiling point c = critical point

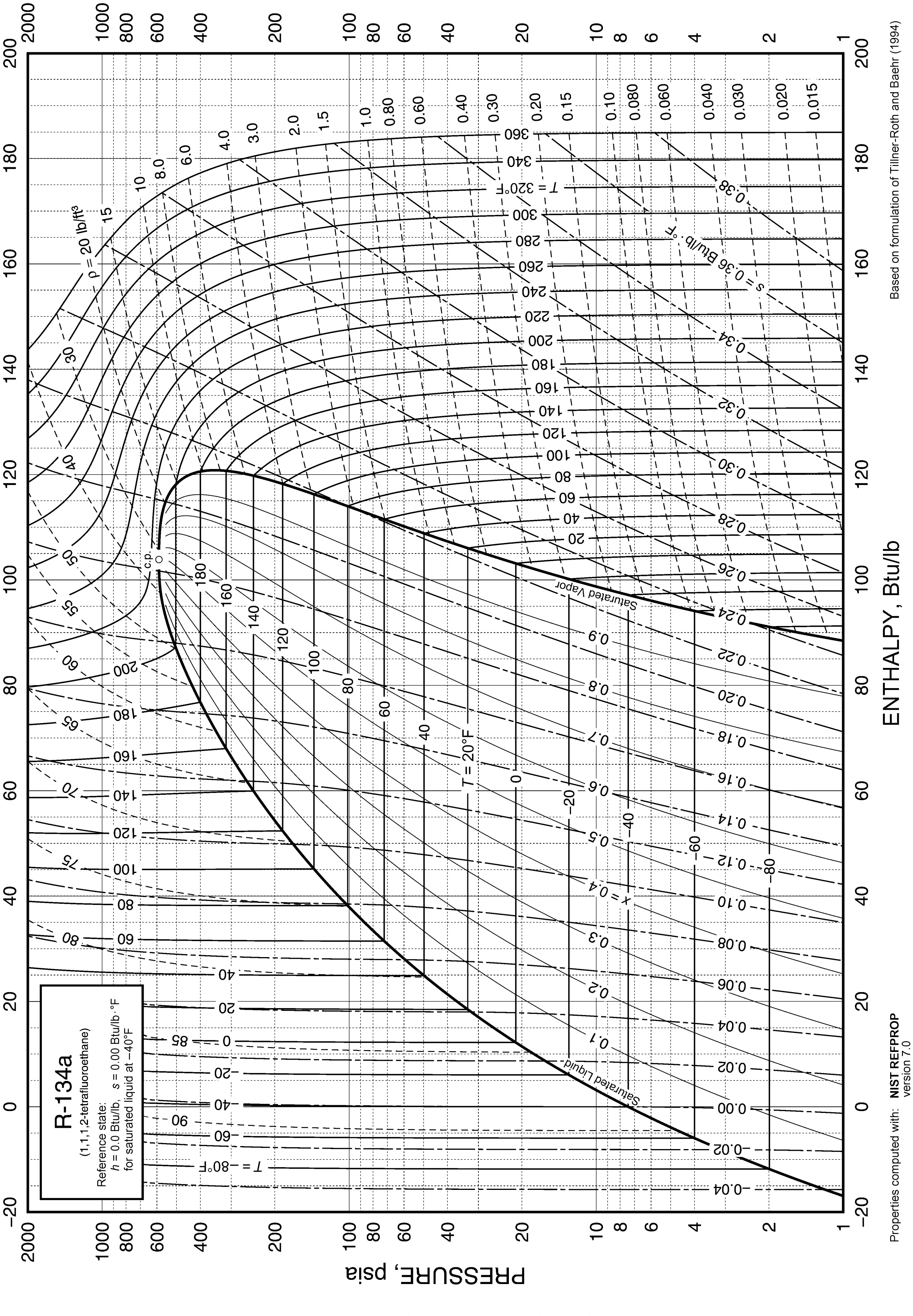


Fig. 2-5 Pressure-Enthalpy Diagram for Refrigerant 134a

(Figure 9, Chapter 30, 2017 ASHRAE Handbook—Fundamentals)

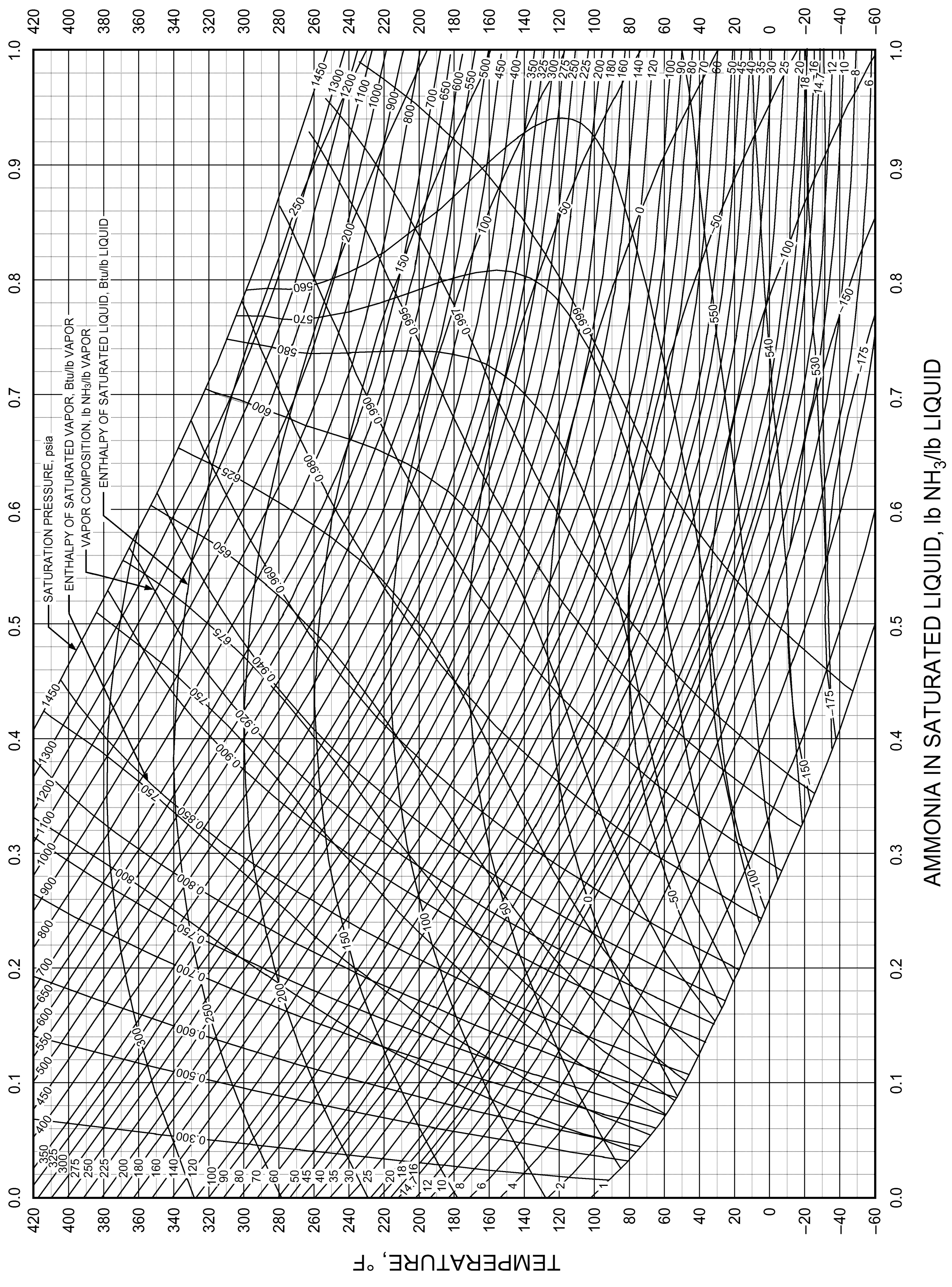
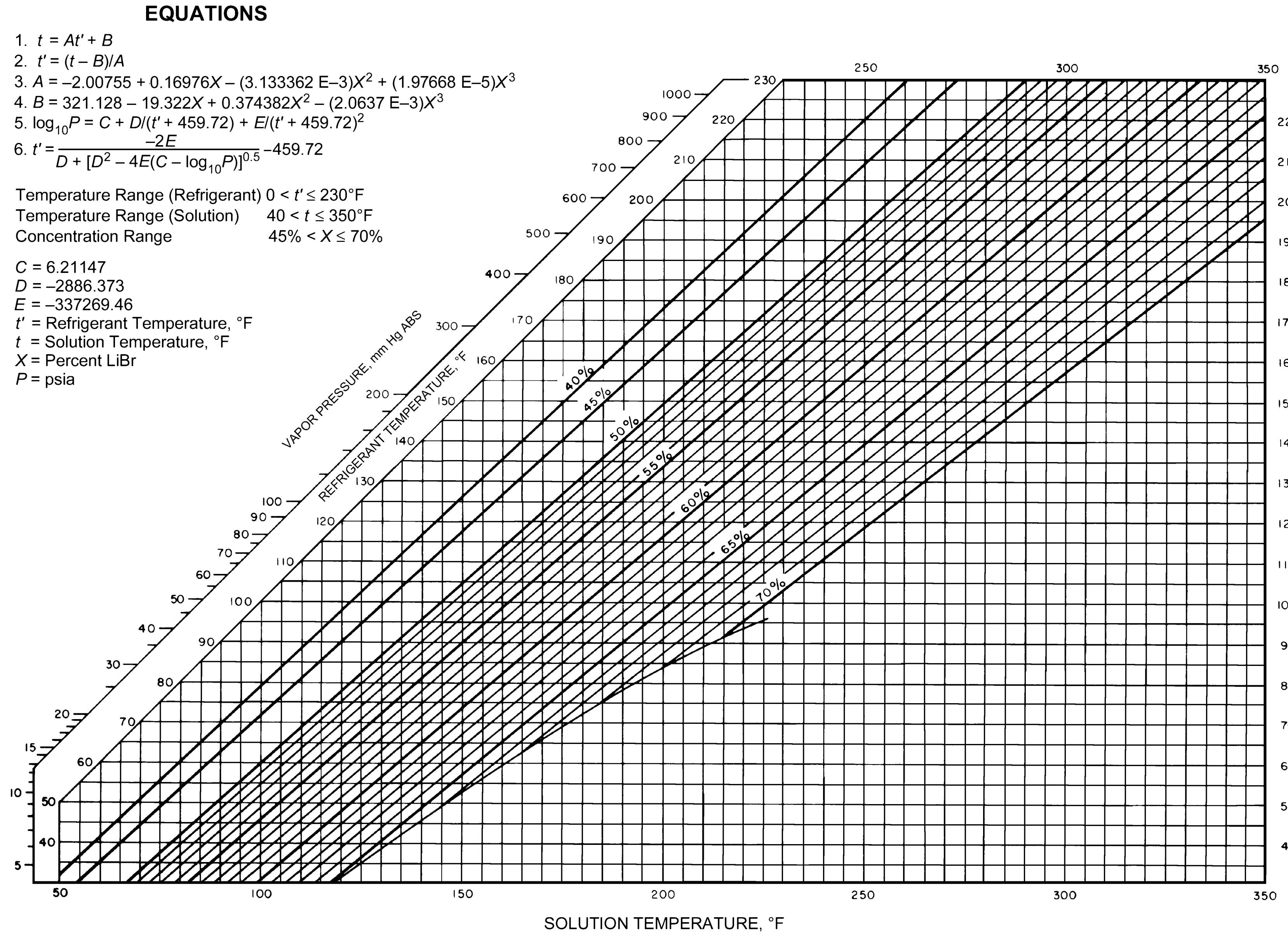


Fig. 2-6 Enthalpy-Concentration Diagram for Aqua-Ammonia

(Figure 35, Chapter 30, 2017 ASHRAE Handbook—Fundamentals)



EQUATIONS

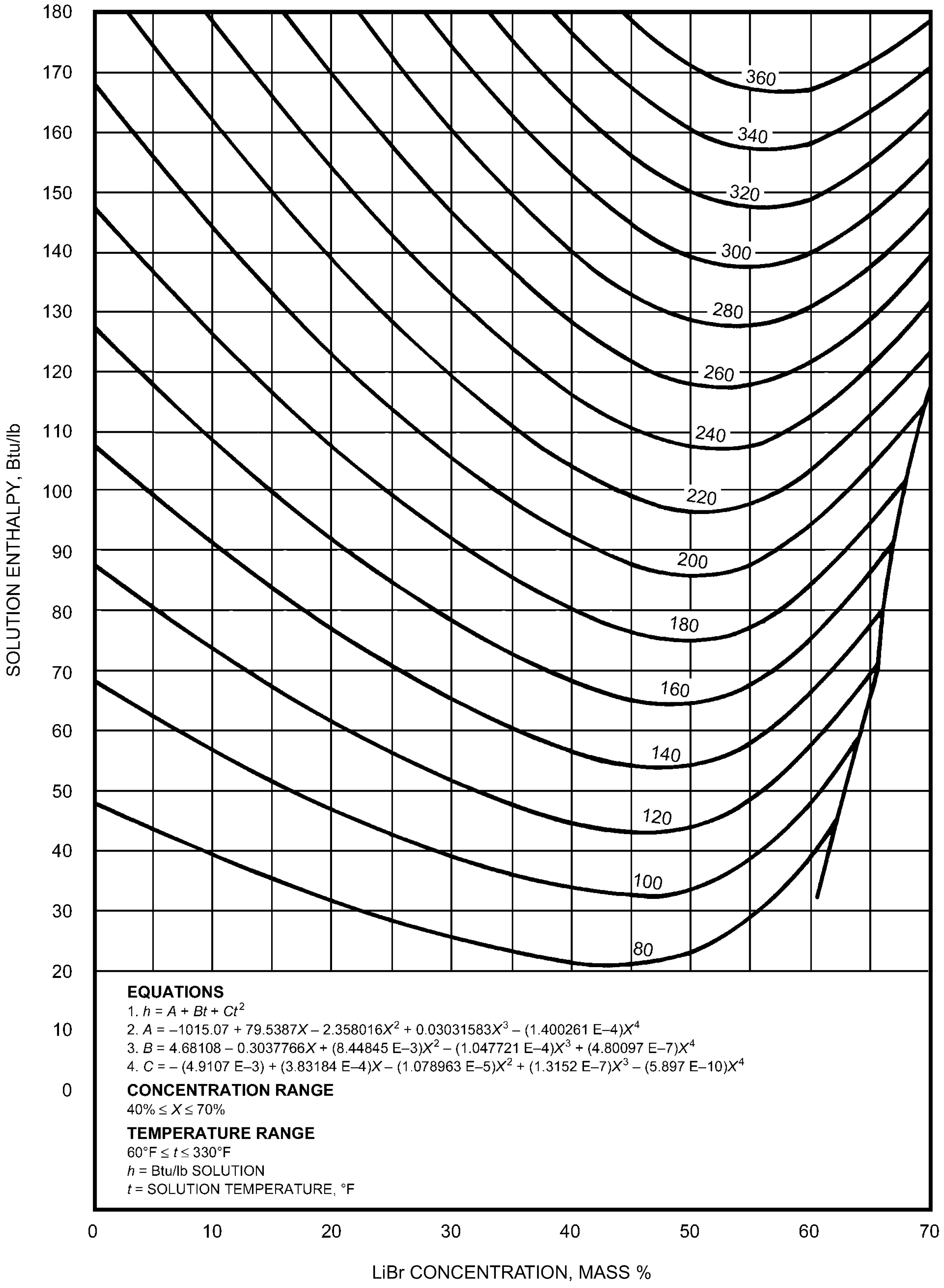
1. $t = At' + B$
2. $t' = (t - B)/A$
3. $A = -2.00755 + 0.16976X - (3.133362\text{ E}{-3})X^2 + (1.97668\text{ E}{-5})X^3$
4. $B = 321.128 - 19.322X + 0.374382X^2 - (2.0637\text{ E}{-3})X^3$
5. $\log_{10}P = C + D/(t' + 459.72) + E/(t' + 459.72)^2$
6. $t' = \dfrac{-2E}{D + [D^2 - 4E(C - \log_{10}P)]^{0.5}} - 459.72$

Temperature Range (Refrigerant) $0 < t' \le 230$°F
Temperature Range (Solution) $40 < t \le 350$°F
Concentration Range $45\% < X \le 70\%$

$C = 6.21147$
$D = -2886.373$
$E = -337269.46$
t' = Refrigerant Temperature, °F
t = Solution Temperature, °F
X = Percent LiBr
P = psia

Fig. 2-7a Equilibrium Chart for Aqueous Lithium Bromide Solutions

(Figure 36, Chapter 30, 2017 ASHRAE Handbook—Fundamentals)



EQUATIONS

1. $h = A + Bt + Ct^2$
2. $A = -1015.07 + 79.5387X - 2.358016X^2 + 0.03031583X^3 - (1.400261\ \text{E}{-4})X^4$
3. $B = 4.68108 - 0.3037766X + (8.44845\ \text{E}{-3})X^2 - (1.047721\ \text{E}{-4})X^3 + (4.80097\ \text{E}{-7})X^4$
4. $C = -(4.9107\ \text{E}{-3}) + (3.83184\ \text{E}{-4})X - (1.078963\ \text{E}{-5})X^2 + (1.3152\ \text{E}{-7})X^3 - (5.897\ \text{E}{-10})X^4$

CONCENTRATION RANGE

$40\% \leq X \leq 70\%$

TEMPERATURE RANGE

$60°\text{F} \leq t \leq 330°\text{F}$

h = Btu/lb SOLUTION

t = SOLUTION TEMPERATURE, °F

Fig. 2-7b Enthalpy-Concentration Diagram for Water/Lithium Bromide Solutions

(Figure 35, Chapter 30, 2017 ASHRAE Handbook—Fundamentals)

Table 2-3 Properties of Gases

Gas	Chemical Formula	Relative Molecular Mass	R, ft·lb$_f$/lb$_m$°R	c_p, Btu/lb$_m$°R	c_p, kJ/kg·K	c_v, Btu/lb$_m$°R	c_v, kJ/kg·K	k
Air	—	28.97	53.34	0.240	1.0	0.171	0.716	1.400
Argon	Ar	39.94	38.66	0.125	0.523	0.075	0.316	1.667
Carbon Dioxide	CO_2	44.01	35.10	0.203	0.85	0.158	0.661	1.285
Carbon Monoxide	CO	28.01	55.16	0.249	1.04	0.178	0.715	1.399
Helium	He	4.003	386.0	1.25	5.23	0.753	3.153	1.667
Hydrogen	H_2	2.016	766.4	3.43	14.36	2.44	10.22	1.404
Methane	CH_4	16.04	96.35	0.532	2.23	0.403	1.69	1.32
Nitrogen	N_2	28.016	55.15	0.248	1.04	0.177	0.741	1.400
Oxygen	O_2	32.000	48.28	0.219	0.917	0.157	0.657	1.395
Steam	H_2O	18.016	85.76	0.445	1.863	0.335	1.402	1.329

low value (e.g., atmospheric pressure), the deviation from ideal gas behavior may be considerable. In this region, tables of thermodynamic properties or charts for a particular substance should be used.

2.2.12 Mixtures

A large number of thermodynamic problems involve mixtures of different pure substances. A pure substance is a substance which is homogeneous and unchanging in chemical composition. Homogeneous mixtures of gases that do not react with each other are therefore pure substances, and the properties of such mixtures can be determined, correlated, and either tabulated or fitted by equations just like the properties of any other pure substance. This work has been done for common mixtures such as air and certain combustion products, but, as an unlimited number of mixtures is possible, properties of all of them cannot be determined experimentally and tabulated. Thus, it is important to be able to calculate the properties of any mixture from the properties of its constituents. Such calculations are discussed in this section, first for gas mixtures and then for gas-vapor mixtures.

Since individual gases can often be approximated as ideal gases, the study of mixtures of ideal gases and their properties is of considerable importance. Each constituent gas in a mixture has its own pressure called the **partial pressure** of the particular gas. The Gibbs-Dalton law states that in a mixture of ideal gases, the pressure of the mixture is equal to the sum of the partial pressures of the individual constituent gases. In equation form

$$p_m = p_a + p_b + p_c \tag{2-14}$$

$$p_a = p_m\,(n_a/n_m),\ p_b = p_m\,(n_b/n_m),\ p_c = (n_c/n_m)$$

where p_m is the total pressure of the mixture of gases a, b, and c, and p_a, p_b, and p_c are the partial pressures. In a mixture of ideal gases, the partial pressure of each constituent equals the pressure that constituent would exert if it existed alone at the temperature and volume of the mixture.

Generally, in gas mixtures, each constituent gas behaves as though the other gases were not present; each gas occupies the total volume of the mixture at the temperature of the mixture and the partial pressure of the gas. Thus, if V_m is the volume of the mixture, then the volume of each component is also V_m, or

$$V_m = V_a = V_b = V_c \tag{2-15}$$

However, the volume of a mixture of ideal gases equals the sum of the volumes of its constituents if each existed alone at the temperature and pressure of the mixture. This property is known as **Amagat's law**, **Leduc's law**, or the **law of additive volumes**. Like Dalton's law, it is strictly true only for ideal gases, but holds approximately for real-gas mixtures, even those in some ranges of pressure and temperature where $pv = RT$ is inaccurate. When the temperature of a real-gas mixture is well above the critical temperatures of all its constituents, the additive volume law is usually more accurate than the additive pressure law.

For ideal-gas mixtures, volumetric analyses are frequently used. The volume fraction is defined as

$$\text{Volume fraction of A} = \frac{V_a(p_m, T_m)}{V_m}$$

$$= \frac{\text{Volume of A existing alone at } p_m, T_m}{\text{Volume of mixture at } p_m, T_m}$$

Note that in a gas mixture, each constituent occupies the total volume; thus volume fraction is not defined as the ratio of a constituent volume to the mixture volume because this ratio is always unity.

Avogadro's law goes on to state, *equal volumes of ideal gases held under exactly the same temperature and pressure have equal numbers of molecules*. If T_m is the temperature of the mixture,

$$T_m = T_a = T_b = T_c \tag{2-16}$$

for the temperature relationship.

The analysis of a gas mixture based on mass is called a **gravimetric analysis**. It is based on the fact that the mass of a mixture equals the sum of the masses of its constituents:

$$m_m = m_a + m_b + m_c \tag{2-17}$$

where the subscript m refers to the mixture and the subscripts a, b, and c refer to individual constituents of the mixture. The ratio m_a/m_m is called the mass fraction of constituent a.

The total number of moles in a mixture is defined as the sum of the number of moles of its constituents:

$$n_m = n_a + n_b + n_c \tag{2-18}$$

The mole fraction x is defined as n/n_m, and

$$M_m = x_a M_a + x_b M_b + x_c M_c \tag{2-19}$$

where M_m is called the apparent (or average) molecular weight of the mixture. The second part of the Gibbs-Dalton law can be taken as a basic definition:

$$U_m = U_a + U_b + U_c \tag{2-20}$$

$$H_m = H_a + H_b + H_c \tag{2-21}$$

$$S_m = S_a + S_b + S_c \tag{2-22}$$

Remember that the constituent entropies here must be evaluated at the temperature and volume of the mixture or at the mixture temperature and the constituent partial pressures. The entropy of any constituent at the volume and temperature of the mixture (and hence at its partial pressure) is greater than its entropy when existing at the pressure and temperature of the mixture (and hence at its partial volume).

Consider the constituents as perfect gases:

$$R_m = (m_a R_a + m_b R_b + m_c R_c)/m_m \tag{2-23}$$

$$c_{vm} = (m_a c_{va} + m_b c_{vb} + m_c c_{vc})/m_m \tag{2-24}$$

$$c_{pm} = (m_a c_{pa} + m_b c_{pb} + m_c c_{pc})/m_m \tag{2-25}$$

2.2.13 Psychrometrics: Moist Air Properties

Consider a simplification of the problem involving a mixture of ideal gases that is in contact with a solid or liquid phase of one of the components. The most familiar example is a mixture of air and water vapor in contact with liquid water or ice, such as the problems encountered in air conditioning or drying. This, and a number of similar problems can be analyzed simply and with considerable accuracy if the following assumptions are made:

1. The solid or liquid phase contains no dissolved gases.
2. The gaseous phase can be treated as a mixture of ideal gases.
3. When the mixture and the condensed phase are at a given pressure and temperature, the equilibrium between the condensed phase and its vapor is not influenced by the presence of the other component. This means that when equilibrium is achieved, the partial pressure of the vapor equals the saturation pressure corresponding to the temperature of the mixture.

If the vapor is at the saturation pressure and temperature, the mixture is referred to as a **saturated mixture**. For an air-water vapor mixture, the term **saturated air** is used.

Psychrometrics is the science involving thermodynamic properties of moist air and the effect of atmospheric moisture on materials and human comfort. As it applies in this text, the definition is broadened to include the method of controlling the thermal properties of moist air.

When moist air is considered to be a mixture of independent, perfect gases, dry air, and water vapor, each is assumed to obey the perfect gas equation of state:

$$\text{Dry air: } p_a V = n_a RT$$

$$\text{Water vapor: } p_w V = n_w RT$$

where

p_a = partial pressure of dry air
p_w = partial pressure of water vapor
V = total mixture volume
n_a = number of moles of dry air
n_w = number of moles of water vapor
R = universal gas constant (8.31441 J/g-mol·K or 1545.32 ft·lb_f/ lb·mol·°R)
T = absolute temperature

The mixture also obeys the perfect gas equation:

$$pV = nRT \text{ or } (p_a + p_w)V = (n_a + n_w)RT$$

Dry-bulb temperature t is the temperature of air as registered by an ordinary thermometer.

Thermodynamic wet-bulb temperature t^* is the temperature at which water (liquid or solid), by evaporating into moist air at a given dry-bulb temperature t and humidity ratio W, can bring the air to saturation adiabatically at the same temperature t^*, while the pressure p is maintained constant. Figure 2-8 may be used as a schematic representation of the adiabatic saturation process, where the leaving air is saturated and at a temperature equal to that of the injected water. A device used in place of the adiabatic saturator is the psychrometer.

The psychrometer consists of two thermometers or other temperature-sensing elements, one of which has a wetted cotton wick covering the bulb (Figure 2-9). When the wet bulb is placed in an airstream, water may evaporate from the wick. The equilibrium temperature the water eventually reaches is called the **wet-bulb temperature**. This process is not one of adiabatic saturation which defines the thermodynamic wet-bulb tempera-

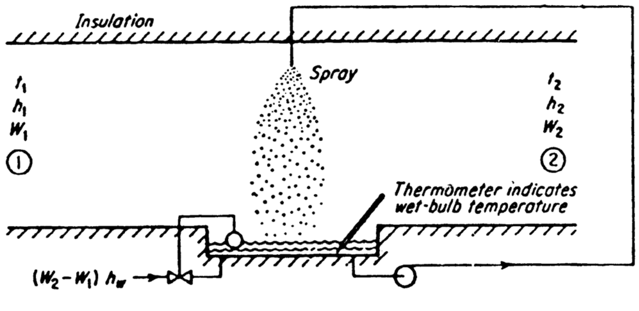


Fig. 2-8 Adiabatic Saturator

ture, but is one of simultaneous heat and mass transfer from the wet-bulb thermometer. Fortunately, the corrections applied to wet-bulb thermometer readings to obtain the thermodynamic wet-bulb temperature are usually small.

Humidity ratio W of a given moist air sample is defined as the ratio of the mass of water vapor to the mass of dry air contained in the sample:

$$\begin{aligned} W &= m_w/m_a \\ W &= 0.62198 p_w/(p - p_w) \end{aligned} \qquad (2\text{-}26)$$

$$W = \frac{(2501 - 2.381t^*)W_s^* - (t - t^*)}{2501 + 1.805t - 4.186t^*} \qquad (2\text{-}27a)$$

where t and t^* are in °C.

In inch-pound units

$$W = \frac{(1093 - 0.556t^*)W_s^* - 0.240(t - t^*)}{1093 + 0.444t - t^*} \qquad (2\text{-}27b)$$

where t and t^* are in °F.

The term W_s^* indicates the humidity ratio if saturated at the wet bulb temperature.

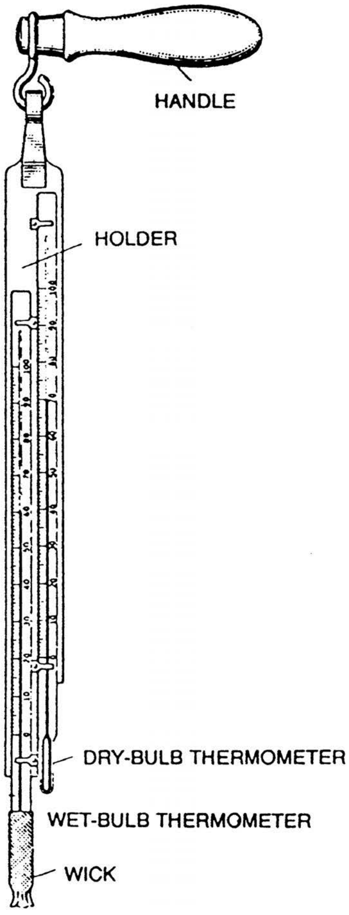


Fig. 2-9 Sling Psychrometer

Degree of saturation μ is the ratio of the actual humidity ratio W to the humidity ratio W_s of saturated air at the same temperature and pressure.

$$\mu = \left.\frac{W}{W_s}\right|_{t,p} \qquad (2\text{-}28)$$

Relative humidity ϕ is the ratio of the mole fraction of water vapor x_w in a given moist air sample to the mole fraction x_{ws} in an air sample which is saturated at the same temperature and pressure:

$$\begin{aligned} \phi &= \left.\frac{x_w}{x_{ws}}\right|_{t,p} \\ \phi &= \left.\frac{p_w}{p_{ws}}\right|_{t,p} \end{aligned} \qquad (2\text{-}29)$$

The term p_{ws} is the saturation pressure of water vapor at the given temperature t.

Dew-point temperature t_d is the temperature of moist air which is saturated at the same pressure p and has the same humidity ratio W as that of the given sample of moist air. It corresponds to the saturation temperature (Column 1) of Table 2-1 for the vapor pressure found in Column 2. As an alternate to using the table, equations have been formulated for the relationship.

For 0°C to 70°C

$$t_d = -35.957 - 1.8726a + 1.1689a^2 \qquad (2\text{-}30a)$$

and for –60°C to 0°C

$$t_d = -60.45 + 7.0322a + 0.3700a^2 \qquad (2\text{-}30b)$$

with t_d in °C and $a = \ln p_w$, with p_w in pascals.

For the temperature range of 32°F to 150°F

$$t_d = 79.047 + 30.5790a + 1.8893a^2 \qquad (2\text{-}31a)$$

and for temperatures below 32°F

$$t_d = 71.98 + 24.873a + 0.8927a^2 \qquad (2\text{-}31b)$$

where t_d is the dew-point temperature in °F and $a = \ln p_w$, with p_w the water vapor partial pressure (in. Hg).

The **volume** v of a moist air mixture is expressed in terms of a unit mass of dry air, with the relation $p = p_a + p_w$, or

$$v = R_a T/(p - p_w) \qquad (2\text{-}32)$$

The **enthalpy** of a mixture of perfect gases is equal to the sum of the individual partial enthalpies of the components. The enthalpy of moist air is then

$$h = h_a + Wh_g$$

where h_a is the specific enthalpy for dry air and h_g is the specific enthalpy for water vapor at the temperature of the mixture. Approximately

Table 2-4 Thermodynamic Properties of Moist Air at Standard Atmospheric Pressure, 14.696 psia

(Table 2, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °F	Humidity Ratio	Specific Volume, ft^3/lb_{da}			Specific Enthalpy, Btu/lb_{da}			Specific Entropy, $Btu/lb_{da} \cdot °F$		Temp., °F
t	W_s, lb_w/lb_{da}	v_{da}	v_{as}	v_s	h_{da}	h_{as}	h_s	s_{da}	s_s	t
–80	0.0000049	9.553	0.000	9.553	–19.221	0.005	–19.215	–0.04594	–0.04592	–80
–79	0.0000053	9.579	0.000	9.579	–18.980	0.005	–18.975	–0.04531	–0.04529	–79
–78	0.0000057	9.604	0.000	9.604	–18.740	0.006	–18.734	–0.04468	–0.04466	–78
–77	0.0000062	9.629	0.000	9.629	–18.500	0.007	–18.493	–0.04405	–0.04403	–77
–76	0.0000067	9.655	0.000	9.655	–18.259	0.007	–18.252	–0.04342	–0.04340	–76
–75	0.0000072	9.680	0.000	9.680	–18.019	0.007	–18.011	–0.04279	–0.04277	–75
–74	0.0000078	9.705	0.000	9.705	–17.778	0.008	–17.770	–0.04217	–0.04215	–74
–73	0.0000084	9.731	0.000	9.731	–17.538	0.009	–17.529	–0.04155	–0.04152	–73
–72	0.0000090	9.756	0.000	9.756	–17.298	0.010	–17.288	–0.04093	–0.04090	–72
–71	0.0000097	9.781	0.000	9.782	–17.057	0.010	–17.047	–0.04031	–0.04028	–71
–70	0.0000104	9.807	0.000	9.807	–16.806	0.011	–16.817	–0.03969	–0.03966	–70
–69	0.0000112	9.832	0.000	9.832	–16.577	0.012	–16.565	–0.03907	–0.03904	–69
–68	0.0000120	9.857	0.000	9.858	–16.336	0.013	–16.324	–0.03846	–0.03843	–68
–67	0.0000129	9.883	0.000	9.883	–16.096	0.013	–16.083	–0.03785	–0.03781	–67
–66	0.0000139	9.908	0.000	9.908	–15.856	0.015	–15.841	–0.03724	–0.03720	–66
–65	0.0000149	9.933	0.000	9.934	–15.616	0.015	–15.600	–0.03663	–0.03659	–65
–64	0.0000160	9.959	0.000	9.959	–15.375	0.017	–15.359	–0.03602	–0.03597	–64
–63	0.0000172	9.984	0.000	9.984	–15.117	0.018	–15.135	–0.03541	–0.03536	–63
–62	0.0000184	10.009	0.000	10.010	–14.895	0.019	–14.876	–0.03481	–0.03476	–62
–61	0.0000198	10.035	0.000	10.035	–14.654	0.021	–14.634	–0.03420	–0.03415	–61
–60	0.0000212	10.060	0.000	10.060	–14.414	0.022	–14.392	–0.03360	–0.03354	–60
–59	0.0000227	10.085	0.000	10.086	–14.174	0.024	–14.150	–0.03300	–0.03294	–59
–58	0.0000243	10.111	0.000	10.111	–13.933	0.025	–13.908	–0.03240	–0.03233	–58
–57	0.0000260	10.136	0.000	10.137	–13.693	0.027	–13.666	–0.03180	–0.03173	–57
–56	0.0000279	10.161	0.000	10.162	–13.453	0.029	–13.424	–0.03121	–0.03113	–56
–55	0.0000298	10.187	0.000	10.187	–13.213	0.031	–13.182	–0.03061	–0.03053	–55
–54	0.0000319	10.212	0.001	10.213	–12.972	0.033	–12.939	–0.03002	–0.02993	–54
–53	0.0000341	10.237	0.001	10.238	–12.732	0.035	–12.697	–0.02943	–0.02934	–53
–52	0.0000365	10.263	0.001	10.263	–12.492	0.038	–12.454	–0.02884	–0.02874	–52
–51	0.0000390	10.288	0.001	10.289	–12.251	0.041	–12.211	–0.02825	–0.02814	–51
–50	0.0000416	10.313	0.001	10.314	–12.011	0.043	–11.968	–0.02766	–0.02755	–50
–49	0.0000445	10.339	0.001	10.340	–11.771	0.046	–11.725	–0.02708	–0.02696	–49
–48	0.0000475	10.364	0.001	10.365	–11.531	0.050	–11.481	–0.02649	–0.02636	–48
–47	0.0000507	10.389	0.001	10.390	–11.290	0.053	–11.237	–0.02591	–0.02577	–47
–46	0.0000541	10.415	0.001	10.416	–11.050	0.056	–10.994	–0.02533	–0.02518	–46
–45	0.0000577	10.440	0.001	10.441	–10.810	0.060	–10.750	–0.02475	–0.02459	–45
–44	0.0000615	10.465	0.001	10.466	–10.570	0.064	–10.505	–0.02417	–0.02400	–44
–43	0.0000656	10.491	0.001	10.492	–10.329	0.068	–10.261	–0.02359	–0.02342	–43
–42	0.0000699	10.516	0.001	10.517	–10.089	0.073	–10.016	–0.02302	–0.02283	–42
–41	0.0000744	10.541	0.001	10.543	–9.849	0.078	–9.771	–0.02244	–0.02224	–41
–40	0.0000793	10.567	0.001	10.568	–9.609	0.083	–9.526	–0.02187	–0.02166	–40
–39	0.0000844	10.592	0.001	10.593	–9.368	0.088	–9.280	–0.02130	–0.02107	–39
–38	0.0000898	10.617	0.002	10.619	–9.128	0.094	–9.034	–0.02073	–0.02049	–38
–37	0.0000956	10.643	0.002	10.644	–8.888	0.100	–8.788	–0.02016	–0.01991	–37
–36	0.0001017	10.668	0.002	10.670	–8.648	0.106	–8.541	–0.01959	–0.01932	–36
–35	0.0001081	10.693	0.002	10.695	–8.407	0.113	–8.294	–0.01902	–0.01874	–35
–34	0.0001150	10.719	0.002	10.721	–8.167	0.120	–8.047	–0.01846	–0.01816	–34
–33	0.0001222	10.744	0.002	10.746	–7.927	0.128	–7.799	–0.01790	–0.01758	–33
–32	0.0001298	10.769	0.002	10.772	–7.687	0.136	–7.551	–0.01733	–0.01699	–32
–31	0.0001379	10.795	0.002	10.797	–7.447	0.145	–7.302	–0.01677	–0.01641	–31
–30	0.0001465	10.820	0.003	10.822	–7.206	0.154	–7.053	–0.01621	–0.01583	–30
–29	0.0001555	10.845	0.003	10.848	–6.966	0.163	–6.803	–0.01565	–0.01525	–29
–28	0.0001650	10.871	0.003	10.873	–6.726	0.173	–6.553	–0.01510	–0.01467	–28
–27	0.0001751	10.896	0.003	10.899	–6.486	0.184	–6.302	–0.01454	–0.01409	–27
–26	0.0001858	10.921	0.003	10.924	–6.245	0.195	–6.051	–0.01399	–0.01351	–26
–25	0.0001970	10.947	0.003	10.950	–6.005	0.207	–5.798	–0.01343	–0.01293	–25
–24	0.0002088	10.972	0.004	10.976	–5.765	0.220	–5.545	–0.01288	–0.01235	–24
–23	0.0002214	10.997	0.004	11.001	–5.525	0.233	–5.292	–0.01233	–0.01176	–23
–22	0.0002346	11.022	0.004	11.027	–5.284	0.247	–5.038	–0.01178	–0.01118	–22
–21	0.0002485	11.048	0.004	11.052	–5.044	0.261	–4.783	–0.01123	–0.01060	–21
–20	0.0002632	11.073	0.005	11.078	–4.804	0.277	–4.527	–0.01069	–0.01002	–20
–19	0.0002786	11.098	0.005	11.103	–4.564	0.293	–4.271	–0.01014	–0.00943	–19
–18	0.0002950	11.124	0.005	11.129	–4.324	0.311	–4.013	–0.00960	–0.00885	–18
–17	0.0003121	11.149	0.006	11.155	–4.084	0.329	–3.754	–0.00905	–0.00826	–17
–16	0.0003303	11.174	0.006	11.180	–3.843	0.348	–3.495	–0.00851	–0.00768	–16
–15	0.0003493	11.200	0.006	11.206	–3.603	0.368	–3.235	–0.00797	–0.00709	–15
–14	0.0003694	11.225	0.007	11.232	–3.363	0.390	–2.973	–0.00743	–0.00650	–14
–13	0.0003905	11.250	0.007	11.257	–3.123	0.412	–2.710	–0.00689	–0.00591	–13
–12	0.0004128	11.276	0.007	11.283	–2.882	0.436	–2.447	–0.00635	–0.00532	–12
–11	0.0004362	11.301	0.008	11.309	–2.642	0.460	–2.182	–0.00582	–0.00473	–11

Table 2-4 Thermodynamic Properties of Moist Air at Standard Sea Level Pressure, 14.696 psi (29.921 in. Hg) *(Continued)*

(Table 2, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °F	Humidity Ratio, lb_w/lb_{da}	Specific Volume, ft^3/lb_{da}			Specific Enthalpy, Btu/lb_{da}			Specific Entropy, $Btu/lb_{da}\cdot°F$			Condensed Water			Temp., °F
											Specific Enthalpy, Btu/lb_w	Specific Entropy, $Btu/lb_w\cdot°F$	Vapor Pressure, in. Hg	
t	W_s	v_{da}	v_{as}	v_s	h_{da}	h_{as}	h_s	s_{da}	s_{as}	s_s	h_w	s_w	p_s	t
–10	0.0004608	11.326	0.008	11.335	–2.402	0.487	–1.915	–0.00528	0.00115	–0.00414	–163.55	–0.3346	0.022050	–10
–9	0.0004867	11.351	0.009	11.360	–2.162	0.514	–1.647	–0.00475	0.00121	–0.00354	–163.09	–0.3335	0.023289	–9
–8	0.0005139	11.377	0.009	11.386	–1.922	0.543	–1.378	–0.00422	0.00127	–0.00294	–162.63	–0.3325	0.024591	–8
–7	0.0005425	11.402	0.010	11.412	–1.681	0.574	–1.108	–0.00369	0.00134	–0.00234	–162.17	–0.3315	0.025959	–7
–6	0.0005726	11.427	0.010	11.438	–1.441	0.606	–0.835	–0.00316	0.00141	–0.00174	–161.70	–0.3305	0.027397	–6
–5	0.0006041	11.453	0.011	11.464	–1.201	0.640	–0.561	–0.00263	0.00149	–0.00114	–161.23	–0.3294	0.028907	–5
–4	0.0006373	11.478	0.012	11.490	–0.961	0.675	–0.286	–0.00210	0.00157	–0.00053	–160.77	–0.3284	0.030494	–4
–3	0.0006722	11.503	0.012	11.516	–0.721	0.712	–0.008	–0.00157	0.00165	0.00008	–160.30	–0.3274	0.032160	–3
–2	0.0007088	11.529	0.013	11.542	–0.480	0.751	0.271	–0.00105	0.00174	0.00069	–159.83	–0.3264	0.033909	–2
–1	0.0007472	11.554	0.014	11.568	–0.240	0.792	0.552	–0.00052	0.00183	0.00130	–159.36	–0.3253	0.035744	–1
0	0.0007875	11.579	0.015	11.594	0.0	0.835	0.835	0.00000	0.00192	0.00192	–158.89	–0.3243	0.037671	0
1	0.0008298	11.604	0.015	11.620	0.240	0.880	1.121	0.00052	0.00202	0.00254	–158.42	–0.3233	0.039694	1
2	0.0008742	11.630	0.016	11.646	0.480	0.928	1.408	0.00104	0.00212	0.00317	–157.95	–0.3223	0.041814	2
3	0.0009207	11.655	0.017	11.672	0.721	0.978	1.699	0.00156	0.00223	0.00380	–157.47	–0.3212	0.044037	3
4	0.0009695	11.680	0.018	11.699	0.961	1.030	1.991	0.00208	0.00235	0.00443	–157.00	–0.3202	0.046370	4
5	0.0010207	11.706	0.019	11.725	1.201	1.085	2.286	0.00260	0.00247	0.00506	–156.52	–0.3192	0.048814	5
6	0.0010743	11.731	0.020	11.751	1.441	1.143	2.584	0.00311	0.00259	0.00570	–156.05	–0.3182	0.051375	6
7	0.0011306	11.756	0.021	11.778	1.681	1.203	2.884	0.00363	0.00635	0.00272	–155.57	–0.3171	0.054060	7
8	0.0011895	11.782	0.022	11.804	1.922	1.266	3.188	0.00414	0.00286	0.00700	–155.09	–0.3161	0.056872	8
9	0.0012512	11.807	0.024	11.831	2.162	1.332	3.494	0.00466	0.00300	0.00766	–154.61	–0.3151	0.059819	9
10	0.0013158	11.832	0.025	11.857	2.402	1.402	3.804	0.00517	0.00315	0.00832	–154.13	–0.3141	0.062901	10
11	0.0013835	11.857	0.026	11.884	2.642	1.474	4.117	0.00568	0.00330	0.00898	–153.65	–0.3130	0.066131	11
12	0.0014544	11.883	0.028	11.910	2.882	1.550	4.433	0.00619	0.00347	0.00966	–153.17	–0.3120	0.069511	12
13	0.0015286	11.908	0.029	11.937	3.123	1.630	4.753	0.00670	0.00364	0.01033	–152.68	–0.3110	0.073049	13
14	0.0016062	11.933	0.031	11.964	3.363	1.714	5.077	0.00721	0.00381	0.01102	–152.20	–0.3100	0.076751	14
15	0.0016874	11.959	0.032	11.991	3.603	1.801	5.404	0.00771	0.00400	0.01171	–151.71	–0.3089	0.080623	15
16	0.0017724	11.984	0.034	12.018	3.843	1.892	5.736	0.00822	0.00419	0.01241	–151.22	–0.3079	0.084673	16
17	0.0018613	12.009	0.036	12.045	4.084	1.988	6.072	0.00872	0.00439	0.01312	–150.74	–0.3069	0.088907	17
18	0.0019543	12.035	0.038	12.072	4.324	2.088	6.412	0.00923	0.00460	0.01383	–150.25	–0.3059	0.093334	18
19	0.0020515	12.060	0.040	12.099	4.564	2.193	6.757	0.00973	0.00482	0.01455	–149.76	–0.3049	0.097962	19
20	0.0021531	12.085	0.042	12.127	4.804	2.303	7.107	0.01023	0.00505	0.01528	–149.27	–0.3038	0.102798	20
21	0.0022592	12.110	0.044	12.154	5.044	2.417	7.462	0.01073	0.00529	0.01602	–148.78	–0.3028	0.107849	21
22	0.0023703	12.136	0.046	12.182	5.285	2.537	7.822	0.01123	0.00554	0.01677	–148.28	–0.3018	0.113130	22
23	0.0024863	12.161	0.048	12.209	5.525	2.662	8.187	0.01173	0.00580	0.01753	–147.79	–0.3008	0.118645	23
24	0.0026073	12.186	0.051	12.237	5.765	2.793	8.558	0.01223	0.00607	0.01830	–147.30	–0.2997	0.124396	24
25	0.0027339	12.212	0.054	12.265	6.005	2.930	8.935	0.01272	0.00636	0.01908	–146.80	–0.2987	0.130413	25
26	0.0028660	12.237	0.056	12.293	6.246	3.073	9.318	0.01322	0.00665	0.01987	–146.30	–0.2977	0.136684	26
27	0.0030039	12.262	0.059	12.321	6.486	3.222	9.708	0.01371	0.00696	0.02067	–145.81	–0.2967	0.143233	27
28	0.0031480	12.287	0.062	12.349	6.726	3.378	10.104	0.01420	0.00728	0.02148	–145.31	–0.2956	0.150066	28
29	0.0032984	12.313	0.065	12.378	6.966	3.541	10.507	0.01470	0.00761	0.02231	–144.81	–0.2946	0.157198	29
30	0.0034552	12.338	0.068	12.406	7.206	3.711	10.917	0.01519	0.00796	0.02315	–144.31	–0.2936	0.164631	30
31	0.0036190	12.363	0.072	12.435	7.447	3.888	11.335	0.01568	0.00832	0.02400	–143.80	–0.2926	0.172390	31
32	0.0037895	12.389	0.075	12.464	7.687	4.073	11.760	0.01617	0.00870	0.02487	–143.30	–0.2915	0.180479	32
32*	0.003790	12.389	0.075	12.464	7.687	4.073	11.760	0.01617	0.00870	0.02487	0.02	0.0000	0.18050	32
33	0.003947	12.414	0.079	12.492	7.927	4.243	12.170	0.01665	0.00905	0.02570	1.03	0.0020	0.18791	33
34	0.004109	12.439	0.082	12.521	8.167	4.420	12.587	0.01714	0.00940	0.02655	2.04	0.0041	0.19559	34
35	0.004277	12.464	0.085	12.550	8.408	4.603	13.010	0.01763	0.00977	0.02740	3.05	0.0061	0.20356	35
36	0.004452	12.490	0.089	12.579	8.648	4.793	13.441	0.01811	0.01016	0.02827	4.05	0.0081	0.21181	36
37	0.004633	12.515	0.093	12.608	8.888	4.990	13.878	0.01860	0.01055	0.02915	5.06	0.0102	0.22035	37
38	0.004820	12.540	0.097	12.637	9.128	5.194	14.322	0.01908	0.01096	0.03004	6.06	0.0122	0.22920	38
39	0.005014	12.566	0.101	12.667	9.369	5.405	14.773	0.01956	0.01139	0.03095	7.07	0.0142	0.23835	39
40	0.005216	12.591	0.105	12.696	9.609	5.624	15.233	0.02004	0.01183	0.03187	8.07	0.0162	0.24784	40
41	0.005424	12.616	0.110	12.726	9.849	5.851	15.700	0.02052	0.01228	0.03281	9.08	0.0182	0.25765	41
42	0.005640	12.641	0.114	12.756	10.089	6.086	16.175	0.02100	0.01275	0.03375	10.08	0.0202	0.26781	42
43	0.005863	12.667	0.119	12.786	10.330	6.330	16.660	0.02148	0.01324	0.03472	11.09	0.0222	0.27831	43
44	0.006094	12.692	0.124	12.816	10.570	6.582	17.152	0.02196	0.01374	0.03570	12.09	0.0242	0.28918	44
45	0.006334	12.717	0.129	12.846	10.810	6.843	17.653	0.02244	0.01426	0.03669	13.09	0.0262	0.30042	45
46	0.006581	12.743	0.134	12.877	11.050	7.114	18.164	0.02291	0.01479	0.03770	14.10	0.0282	0.31206	46
47	0.006838	12.768	0.140	12.908	11.291	7.394	18.685	0.02339	0.01534	0.03873	15.10	0.0302	0.32408	47
48	0.007103	12.793	0.146	12.939	11.531	7.684	19.215	0.02386	0.01592	0.03978	16.10	0.0321	0.33651	48
49	0.007378	12.818	0.152	12.970	11.771	7.984	19.756	0.02433	0.01651	0.04084	17.10	0.0341	0.34937	49
50	0.007661	12.844	0.158	13.001	12.012	8.295	20.306	0.02480	0.01712	0.04192	18.11	0.0361	0.36264	50
51	0.007955	12.869	0.164	13.033	12.252	8.616	20.868	0.02528	0.01775	0.04302	19.11	0.0381	0.37636	51
52	0.008259	12.894	0.171	13.065	12.492	8.949	21.441	0.02575	0.01840	0.04415	20.11	0.0400	0.39054	52
53	0.008573	12.920	0.178	13.097	12.732	9.293	22.025	0.02622	0.01907	0.04529	21.11	0.0420	0.40518	53
54	0.008897	12.945	0.185	13.129	12.973	9.648	22.621	0.02668	0.01976	0.04645	22.11	0.0439	0.42030	54
55	0.009233	12.970	0.192	13.162	13.213	10.016	23.229	0.02715	0.02048	0.04763	23.11	0.0459	0.43592	55
56	0.009580	12.995	0.200	13.195	13.453	10.397	23.850	0.02762	0.02122	0.04884	24.11	0.0478	0.45205	56
57	0.009938	13.021	0.207	13.228	13.694	10.790	24.484	0.02808	0.02198	0.05006	25.11	0.0497	0.46870	57
58	0.010309	13.046	0.216	13.262	13.934	11.197	25.131	0.02855	0.02277	0.05132	26.11	0.0517	0.48589	58
59	0.010692	13.071	0.224	13.295	14.174	11.618	25.792	0.02901	0.02358	0.05259	27.11	0.0536	0.50363	59

*Extrapolated to represent metastable equilibrium with undercooled liquid.

Table 2-4 Thermodynamic Properties of Moist Air at Standard Sea Level Pressure, 14.696 psi (29.921 in. Hg) *(Continued)*

(Table 2, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °F	Humidity Ratio, lb$_w$/lb$_{da}$	Specific Volume, ft^3/lb$_{da}$			Specific Enthalpy, Btu/lb$_{da}$			Specific Entropy, Btu/lb$_{da}$·°F			Condensed Water			Temp., °F
											Specific Enthalpy, Btu/lb$_w$	Specific Entropy, Btu/lb$_w$·°F	Vapor Pressure, in. Hg	
t	W_s	v_{da}	v_{as}	v_s	h_{da}	h_{as}	h_s	s_{da}	s_{as}	s_s	h_w	s_w	p_s	t
60	0.011087	13.096	0.233	13.329	14.415	12.052	26.467	0.02947	0.02442	0.05389	28.11	0.0555	0.52193	60
61	0.011496	13.122	0.242	13.364	14.655	12.502	27.157	0.02994	0.02528	0.05522	29.12	0.0575	0.54082	61
62	0.011919	13.147	0.251	13.398	14.895	12.966	27.862	0.03040	0.02617	0.05657	30.11	0.0594	0.56032	62
63	0.012355	13.172	0.261	13.433	15.135	13.446	28.582	0.03086	0.02709	0.05795	31.11	0.0613	0.58041	63
64	0.012805	13.198	0.271	13.468	15.376	13.942	29.318	0.03132	0.02804	0.05936	32.11	0.0632	0.60113	64
65	0.013270	13.223	0.281	13.504	15.616	14.454	30.071	0.03178	0.02902	0.06080	33.11	0.0651	0.62252	65
66	0.013750	13.248	0.292	13.540	15.856	14.983	30.840	0.03223	0.03003	0.06226	34.11	0.0670	0.64454	66
67	0.014246	13.273	0.303	13.577	16.097	15.530	31.626	0.03269	0.03107	0.06376	35.11	0.0689	0.66725	67
68	0.014758	13.299	0.315	13.613	16.337	16.094	32.431	0.03315	0.03214	0.06529	36.11	0.0708	0.69065	68
69	0.015286	13.324	0.326	13.650	16.577	16.677	33.254	0.03360	0.03325	0.06685	37.11	0.0727	0.71479	69
70	0.015832	13.349	0.339	13.688	16.818	17.279	34.097	0.03406	0.03438	0.06844	38.11	0.0746	0.73966	70
71	0.016395	13.375	0.351	13.726	17.058	17.901	34.959	0.03451	0.03556	0.07007	39.11	0.0765	0.76567	71
72	0.016976	13.400	0.365	13.764	17.299	18.543	35.841	0.03496	0.03677	0.07173	40.11	0.0783	0.79167	72
73	0.017575	13.425	0.378	13.803	17.539	19.204	36.743	0.03541	0.03801	0.07343	41.11	0.0802	0.81882	73
74	0.018194	13.450	0.392	13.843	17.779	19.889	37.668	0.03586	0.03930	0.07516	42.11	0.0821	0.84684	74
75	0.018833	13.476	0.407	13.882	18.020	20.595	38.615	0.03631	0.04062	0.07694	43.11	0.0840	0.87567	75
76	0.019491	13.501	0.422	13.923	18.260	21.323	39.583	0.03676	0.04199	0.07875	44.10	0.0858	0.90533	76
77	0.020170	13.526	0.437	13.963	18.500	22.075	40.576	0.03721	0.04339	0.08060	45.10	0.0877	0.93589	77
78	0.020871	13.551	0.453	14.005	18.741	22.851	41.592	0.03766	0.04484	0.08250	46.10	0.0896	0.96733	78
79	0.021594	13.577	0.470	14.046	18.981	23.652	42.633	0.03811	0.04633	0.08444	47.10	0.0914	0.99970	79
80	0.022340	13.602	0.487	14.089	19.222	24.479	43.701	0.03855	0.04787	0.08642	48.10	0.0933	1.03302	80
81	0.023109	13.627	0.505	14.132	19.462	25.332	44.794	0.03900	0.04945	0.08844	49.10	0.0951	1.06728	81
82	0.023902	13.653	0.523	14.175	19.702	26.211	45.913	0.03944	0.05108	0.09052	50.10	0.0970	1.10252	82
83	0.024720	13.678	0.542	14.220	19.943	27.120	47.062	0.03988	0.05276	0.09264	51.09	0.0988	1.13882	83
84	0.025563	13.703	0.561	14.264	20.183	28.055	48.238	0.04033	0.05448	0.09481	52.09	0.1006	1.17608	84
85	0.026433	13.728	0.581	14.310	20.424	29.021	49.445	0.04077	0.05626	0.09703	53.09	0.1025	1.21445	85
86	0.027329	13.754	0.602	14.356	20.664	30.017	50.681	0.04121	0.05809	0.09930	54.09	0.1043	1.25388	86
87	0.028254	13.779	0.624	14.403	20.905	31.045	51.949	0.04165	0.05998	0.10163	55.09	0.1061	1.29443	87
88	0.029208	13.804	0.646	14.450	21.145	32.105	53.250	0.04209	0.06192	0.10401	56.09	0.1080	1.33613	88
89	0.030189	13.829	0.669	14.498	21.385	33.197	54.582	0.04253	0.06392	0.10645	57.09	0.1098	1.37893	89
90	0.031203	13.855	0.692	14.547	21.626	34.325	55.951	0.04297	0.06598	0.10895	58.08	0.1116	1.42298	90
91	0.032247	13.880	0.717	14.597	21.866	35.489	57.355	0.06810	0.04340	0.11150	59.08	0.1134	1.46824	91
92	0.033323	13.905	0.742	14.647	22.107	36.687	58.794	0.04384	0.07028	0.11412	60.08	0.1152	1.51471	92
93	0.034433	13.930	0.768	14.699	22.347	37.924	60.271	0.04427	0.07253	0.11680	61.08	0.1170	1.56248	93
94	0.035577	13.956	0.795	14.751	22.588	39.199	61.787	0.04471	0.07484	0.11955	62.08	0.1188	1.61154	94
95	0.036757	13.981	0.823	14.804	22.828	40.515	63.343	0.04514	0.07722	0.12237	63.08	0.1206	1.66196	95
96	0.037972	14.006	0.852	14.858	23.069	41.871	64.940	0.04558	0.07968	0.12525	64.07	0.1224	1.71372	96
97	0.039225	14.032	0.881	14.913	23.309	43.269	66.578	0.04601	0.08220	0.12821	65.07	0.1242	1.76685	97
98	0.040516	14.057	0.912	14.969	23.550	44.711	68.260	0.04644	0.08480	0.13124	66.07	0.1260	1.82141	98
99	0.041848	14.082	0.944	15.026	23.790	46.198	69.988	0.04687	0.08747	0.13434	67.07	0.1278	1.87745	99
100	0.043219	14.107	0.976	15.084	24.031	47.730	71.761	0.04730	0.09022	0.13752	68.07	0.1296	1.93492	100
101	0.044634	14.133	1.010	15.143	24.271	49.312	73.583	0.04773	0.09306	0.14079	69.07	0.1314	1.99396	101
102	0.046090	14.158	1.045	15.203	24.512	50.940	75.452	0.04816	0.09597	0.14413	70.06	0.1332	2.05447	102
103	0.047592	14.183	1.081	15.264	24.752	52.621	77.373	0.04859	0.09897	0.14756	71.06	0.1349	2.11661	103
104	0.049140	14.208	1.118	15.326	24.993	54.354	79.346	0.04901	0.10206	0.15108	72.06	0.1367	2.18037	104
105	0.050737	14.234	1.156	15.390	25.233	56.142	81.375	0.04944	0.10525	0.15469	73.06	0.1385	2.24581	105
106	0.052383	14.259	1.196	15.455	25.474	57.986	83.460	0.04987	0.10852	0.15839	74.06	0.1402	2.31297	106
107	0.054077	14.284	1.236	15.521	25.714	59.884	85.599	0.05029	0.11189	0.16218	75.06	0.1420	2.38173	107
108	0.055826	14.309	1.279	15.588	25.955	61.844	87.799	0.05071	0.11537	0.16608	76.05	0.1438	2.45232	108
109	0.057628	14.335	1.322	15.657	26.195	63.866	90.061	0.05114	0.11894	0.17008	77.05	0.1455	2.52473	109
110	0.059486	14.360	1.367	15.727	26.436	65.950	92.386	0.05156	0.12262	0.17418	78.05	0.1473	2.59891	110
111	0.061401	14.385	1.414	15.799	26.677	68.099	94.776	0.05198	0.12641	0.17839	79.05	0.1490	2.67500	111
112	0.063378	14.411	1.462	15.872	26.917	70.319	97.237	0.05240	0.13032	0.18272	80.05	0.1508	2.75310	112
113	0.065411	14.436	1.511	15.947	27.158	72.603	99.760	0.05282	0.13434	0.18716	81.05	0.1525	2.83291	113
114	0.067512	14.461	1.562	16.023	27.398	74.964	102.362	0.05324	0.13847	0.19172	82.04	0.1543	2.91491	114
115	0.069676	14.486	1.615	16.101	27.639	77.396	105.035	0.05366	0.14274	0.19640	83.04	0.1560	2.99883	115
116	0.071908	14.512	1.670	16.181	27.879	79.906	107.786	0.05408	0.14713	0.20121	84.04	0.1577	3.08488	116
117	0.074211	14.537	1.726	16.263	28.120	82.497	110.617	0.05450	0.15165	0.20615	85.04	0.1595	3.17305	117
118	0.076586	14.562	1.784	16.346	28.361	85.169	113.530	0.05492	0.15631	0.21122	86.04	0.1612	3.26335	118
119	0.079036	14.587	1.844	16.432	28.601	87.927	116.528	0.05533	0.16111	0.21644	87.04	0.1629	3.35586	119
120	0.081560	14.613	1.906	16.519	28.842	90.770	119.612	0.05575	0.16605	0.22180	88.04	0.1647	3.45052	120
121	0.084169	14.638	1.971	16.609	29.083	93.709	122.792	0.05616	0.17115	0.22731	89.04	0.1664	3.54764	121
122	0.086860	14.663	2.037	16.700	29.323	96.742	126.065	0.05658	0.17640	0.23298	90.03	0.1681	3.64704	122
123	0.089633	14.688	2.106	16.794	29.564	99.868	129.432	0.05699	0.18181	0.23880	91.03	0.1698	3.74871	123
124	0.092500	14.714	2.176	16.890	29.805	103.102	132.907	0.05740	0.18739	0.24480	92.03	0.1715	3.85298	124
125	0.095456	14.739	2.250	16.989	30.045	106.437	136.482	0.05781	0.19314	0.25096	93.03	0.1732	3.95961	125
126	0.098504	14.764	2.325	17.090	30.286	109.877	140.163	0.05823	0.19907	0.25729	94.03	0.1749	4.06863	126
127	0.101657	14.789	2.404	17.193	30.527	113.438	143.965	0.05864	0.20519	0.26382	95.03	0.1766	4.18046	127
128	0.104910	14.815	2.485	17.299	30.767	117.111	147.878	0.05905	0.21149	0.27054	96.03	0.1783	4.29477	128
129	0.108270	14.840	2.569	17.409	31.008	120.908	151.916	0.05946	0.21810	0.27745	97.03	0.1800	4.41181	129

Table 2-4 Thermodynamic Properties of Moist Air at Standard Sea Level Pressure, 14.696 psi (29.921 in. Hg) *(Continued)*

(Table 2, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °F	Humidity Ratio, lb_w/lb_{da}	Specific Volume, ft^3/lb_{da}			Specific Enthalpy, Btu/lb_{da}			Specific Entropy, $Btu/lb_{da}\cdot°F$			Condensed Water: Specific Enthalpy, Btu/lb_w	Condensed Water: Specific Entropy, $Btu/lb_w\cdot°F$	Condensed Water: Vapor Pressure, in. Hg	Temp., °F
t	W_s	v_{da}	v_{as}	v_s	h_{da}	h_{as}	h_s	s_{da}	s_{as}	s_s	h_w	s_w	p_s	t
130	0.111738	14.865	2.655	17.520	31.249	124.828	156.076	0.05986	0.22470	0.28457	98.03	0.1817	4.53148	130
131	0.115322	14.891	2.745	17.635	31.489	128.880	160.370	0.06027	0.23162	0.29190	99.02	0.1834	4.65397	131
132	0.119023	14.916	2.837	17.753	31.730	133.066	164.796	0.06068	0.23876	0.29944	100.02	0.1851	4.77919	132
133	0.122855	14.941	2.934	17.875	31.971	137.403	169.374	0.06109	0.24615	0.30723	101.02	0.1868	4.90755	133
134	0.126804	14.966	3.033	17.999	32.212	141.873	174.084	0.06149	0.25375	0.31524	102.02	0.1885	5.03844	134
135	0.130895	14.992	3.136	18.127	32.452	146.504	178.957	0.06190	0.26161	0.32351	103.02	0.1902	5.17258	135
136	0.135124	15.017	3.242	18.259	32.693	151.294	183.987	0.06230	0.26973	0.33203	104.02	0.1919	5.30973	136
137	0.139494	15.042	3.352	18.394	32.934	156.245	189.179	0.06271	0.27811	0.34082	105.02	0.1935	5.44985	137
138	0.144019	15.067	3.467	18.534	33.175	161.374	194.548	0.06311	0.28707	0.35018	106.02	0.1952	5.59324	138
139	0.148696	15.093	3.585	18.678	33.415	166.677	200.092	0.06351	0.29602	0.35954	107.02	0.1969	5.73970	139
140	0.153538	15.118	3.708	18.825	33.656	172.168	205.824	0.06391	0.30498	0.36890	108.02	0.1985	5.88945	140
141	0.158643	15.143	3.835	18.978	33.897	177.857	211.754	0.06431	0.31456	0.37887	109.02	0.2002	6.04256	141
142	0.163748	15.168	3.967	19.135	34.138	183.754	217.892	0.06471	0.32446	0.38918	110.02	0.2019	6.19918	142
143	0.169122	15.194	4.103	19.297	34.379	189.855	224.233	0.06511	0.33470	0.39981	111.02	0.2035	6.35898	143
144	0.174694	15.219	4.245	19.464	34.620	196.183	230.802	0.06551	0.34530	0.41081	112.02	0.2052	6.52241	144
145	0.180467	15.244	4.392	19.637	34.860	202.740	237.600	0.06591	0.35626	0.42218	113.02	0.2068	6.68932	145
146	0.186460	15.269	4.545	19.815	35.101	209.550	244.651	0.06631	0.36764	0.43395	114.02	0.2085	6.86009	146
147	0.192668	15.295	4.704	19.999	35.342	216.607	251.949	0.06671	0.37941	0.44611	115.02	0.2101	7.03435	147
148	0.199110	15.320	4.869	20.189	35.583	223.932	259.514	0.06710	0.39160	0.45871	116.02	0.2118	7.21239	148
149	0.205792	15.345	5.040	20.385	35.824	231.533	267.356	0.06750	0.40424	0.47174	117.02	0.2134	7.39413	149
150	0.212730	15.370	5.218	20.589	36.064	239.426	275.490	0.06790	0.41735	0.48524	118.02	0.2151	7.57977	150
151	0.219945	15.396	5.404	20.799	36.305	247.638	283.943	0.06829	0.43096	0.49925	119.02	0.2167	7.76958	151
152	0.227429	15.421	5.596	21.017	36.546	256.158	292.705	0.06868	0.44507	0.51375	120.02	0.2184	7.96306	152
153	0.235218	15.446	5.797	21.243	36.787	265.028	301.816	0.06908	0.45973	0.52881	121.02	0.2200	8.16087	153
154	0.243309	15.471	6.005	21.477	37.028	274.245	311.273	0.06947	0.47494	0.54441	122.02	0.2216	8.36256	154
155	0.251738	15.497	6.223	21.720	37.269	283.849	321.118	0.06986	0.49077	0.56064	123.02	0.2233	8.56871	155
156	0.260512	15.522	6.450	21.972	37.510	293.849	331.359	0.07025	0.50723	0.57749	124.02	0.2249	8.77915	156
157	0.269644	15.547	6.686	22.233	37.751	304.261	342.012	0.07065	0.52434	0.59499	125.02	0.2265	8.99378	157
158	0.279166	15.572	6.933	22.505	37.992	315.120	353.112	0.07104	0.54217	0.61320	126.02	0.2281	9.21297	158
159	0.289101	15.598	7.190	22.788	38.233	326.452	364.685	0.07143	0.56074	0.63216	127.02	0.2297	9.43677	159
160	0.29945	15.623	7.459	23.082	38.474	338.263	376.737	0.07181	0.58007	0.65188	128.02	0.2314	9.6648	160
161	0.31027	15.648	7.740	23.388	38.715	350.610	389.325	0.07220	0.60025	0.67245	129.02	0.2330	9.8978	161
162	0.32156	15.673	8.034	23.707	38.956	363.501	402.457	0.07259	0.62128	0.69388	130.03	0.2346	10.1353	162
163	0.33336	15.699	8.341	24.040	39.197	376.979	416.175	0.07298	0.64325	0.71623	131.03	0.2362	10.3776	163
164	0.34572	15.724	8.664	24.388	39.438	391.095	430.533	0.07337	0.66622	0.73959	132.03	0.2378	10.6250	164
165	0.35865	15.749	9.001	24.750	39.679	405.865	445.544	0.07375	0.69022	0.76397	133.03	0.2394	10.8771	165
166	0.37220	15.774	9.355	25.129	39.920	421.352	461.271	0.07414	0.71535	0.78949	134.03	0.2410	11.1343	166
167	0.38639	15.800	9.726	25.526	40.161	437.578	477.739	0.07452	0.74165	0.81617	135.03	0.2426	11.3965	167
168	0.40131	15.825	10.117	25.942	40.402	454.630	495.032	0.07491	0.76925	0.84415	136.03	0.2442	11.6641	168
169	0.41698	15.850	10.527	26.377	40.643	472.554	513.197	0.07529	0.79821	0.87350	137.04	0.2458	11.9370	169
170	0.43343	15.875	10.959	26.834	40.884	491.372	532.256	0.07567	0.82858	0.90425	138.04	0.2474	12.2149	170
171	0.45079	15.901	11.414	27.315	41.125	511.231	552.356	0.07606	0.86058	0.93664	139.04	0.2490	12.4988	171
172	0.46905	15.926	11.894	27.820	41.366	532.138	573.504	0.07644	0.89423	0.97067	140.04	0.2506	12.7880	172
173	0.48829	15.951	12.400	28.352	41.607	554.160	595.767	0.07682	0.92962	1.00644	141.04	0.2521	13.0823	173
174	0.50867	15.976	12.937	28.913	41.848	577.489	619.337	0.07720	0.96707	1.04427	142.04	0.2537	13.3831	174
175	0.53019	16.002	13.504	29.505	42.089	602.139	644.229	0.07758	1.00657	1.08416	143.05	0.2553	13.6894	175
176	0.55294	16.027	14.103	30.130	42.331	628.197	670.528	0.07796	1.04828	1.12624	144.05	0.2569	14.0010	176
177	0.57710	16.052	14.741	30.793	42.572	655.876	698.448	0.07834	1.09253	1.17087	145.05	0.2585	14.3191	177
178	0.60274	16.078	15.418	31.496	42.813	685.260	728.073	0.07872	1.13943	1.21815	146.05	0.2600	14.6430	178
179	0.63002	16.103	16.139	32.242	43.054	716.524	759.579	0.07910	1.18927	1.26837	147.06	0.2616	14.9731	179
180	0.65911	16.128	16.909	33.037	43.295	749.871	793.166	0.07947	1.24236	1.32183	148.06	0.2632	15.3097	180
181	0.69012	16.153	17.730	33.883	43.536	785.426	828.962	0.07985	1.29888	1.37873	149.06	0.2647	15.6522	181
182	0.72331	16.178	18.609	34.787	43.778	823.487	867.265	0.08023	1.35932	1.43954	150.06	0.2663	16.0014	182
183	0.75885	16.204	19.551	35.755	44.019	864.259	908.278	0.08060	1.42396	1.50457	151.07	0.2679	16.3569	183
184	0.79703	16.229	20.564	36.793	44.260	908.061	952.321	0.08098	1.49332	1.57430	152.07	0.2694	16.7190	184
185	0.83817	16.254	21.656	37.910	44.501	955.261	999.763	0.08135	1.56797	1.64932	153.07	0.2710	17.0880	185
186	0.88251	16.280	22.834	39.113	44.742	1006.149	1050.892	0.08172	1.64834	1.73006	154.08	0.2725	17.4634	186
187	0.93057	16.305	24.111	40.416	44.984	1061.314	1106.298	0.08210	1.73534	1.81744	155.08	0.2741	17.8462	187
188	0.98272	16.330	25.498	41.828	45.225	1121.174	1166.399	0.08247	1.82963	1.91210	156.08	0.2756	18.2357	188
189	1.03951	16.355	27.010	43.365	45.466	1186.382	1231.848	0.08284	1.93221	2.01505	157.09	0.2772	18.6323	189
190	1.10154	16.381	28.661	45.042	45.707	1257.614	1303.321	0.08321	2.04412	2.12733	158.09	0.2787	19.0358	190
191	1.16965	16.406	30.476	46.882	45.949	1335.834	1381.783	0.08359	2.16684	2.25043	159.09	0.2803	19.4468	191
192	1.24471	16.431	32.477	48.908	46.190	1422.047	1468.238	0.08396	2.30193	2.38589	160.10	0.2818	19.8652	192
193	1.32788	16.456	34.695	51.151	46.431	1517.581	1564.013	0.08433	2.45144	2.53576	161.10	0.2834	20.2913	193
194	1.42029	16.481	37.161	53.642	46.673	1623.758	1670.430	0.08470	2.61738	2.70208	162.11	0.2849	20.7244	194
195	1.52396	16.507	39.928	56.435	46.914	1742.879	1789.793	0.08506	2.80332	2.88838	163.11	0.2864	21.1661	195
196	1.64070	16.532	43.046	59.578	47.155	1877.032	1924.188	0.08543	3.01244	3.09787	164.12	0.2880	21.6152	196
197	1.77299	16.557	46.580	63.137	47.397	2029.069	2076.466	0.08580	3.24914	3.33494	165.12	0.2895	22.0714	197
198	1.92472	16.583	50.636	67.218	47.638	2203.464	2251.102	0.08617	3.52030	3.60647	166.13	0.2910	22.5367	198
199	2.09975	16.608	55.316	71.923	47.879	2404.668	2452.547	0.08653	3.83275	3.91929	167.13	0.2926	23.0092	199
200	2.30454	16.633	60.793	77.426	48.121	2640.084	2688.205	0.08690	4.19787	4.28477	168.13	0.2941	23.4906	200

$$h_a = t \text{ (kJ/kg)} \tag{2-33}$$

$$h_g = 2501 + 1.805t \text{ (kJ/kg)} \tag{2-34}$$

where t is the dry-bulb temperature, °C. The moist air enthalpy then becomes

$$h = t + W(2501 + 1.805t) \text{ (kJ/kg dry air)} \tag{2-35}$$

In conventional (I-P) units

$$h_a = 0.240t \text{ (Btu/lb)} \tag{2-36}$$

$$h_g = 1061 + 0.444t \text{ (Btu/lb)} \tag{2-37}$$

$$h = 0.240t + W(1061 + 0.444t) \text{ (Btu/lb)} \tag{2-38}$$

where t is the dry-bulb temperature, °F.

Table 2-4 is a tabulation of the thermodynamic properties of moist air at sea level standard atmospheric pressure. Table 2-5 shows variation of atmospheric pressure with altitude.

2.2.14 Psychrometric Chart

The ASHRAE psychrometric chart may be used to solve numerous process problems with moist air. Processes performed with air can be plotted on the chart for quick visualization, as well as for determining changes in significant properties such as temperature, humidity ratio, and enthalpy for the process. ASHRAE Psychrometric Chart No. 1 in I-P units is shown in Figure 2-10. Some basic air-conditioning processes are shown in Figure 2-11.

Sensible heating only (C) or *sensible cooling* (G) shows a change in dry-bulb temperature with no change in humidity ratio. For either sensible heat change process, the temperature changes but not the moisture content of the air.

Humidifying only (A) or *dehumidifying* only (E) shows a change in humidity ratio with no change in dry-bulb temperature. For these latent heat processes, the moisture content of the air is changed but not the temperature.

Cooling and dehumidifying (F) result in a reduction of both the dry-bulb temperature and the humidity ratio. Cooling coils generally perform this type of process.

Heating and humidifying (B) result in an increase of both the dry-bulb temperature and the humidity ratio.

Chemical dehumidifying (D) is a process in which moisture from the air is absorbed or adsorbed by a hygroscopic material. Generally, the process essentially occurs at constant enthalpy.

Evaporative cooling only (*H*) is an adiabatic heat transfer process in which the wet-bulb temperature of the air remains constant but the dry-bulb temperature drops as the humidity rises.

Adiabatic mixing of air at one condition with air at some other condition is represented on the psychrometric chart by a straight line drawn between the points representing the two air conditions (Figure 2-12).

Table 2-5 Standard Atmospheric Data with Altitude

(Table 1, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Altitude, ft	Temperature, °F	Pressure	
		in. Hg	psia
–1000	62.6	31.02	15.236
–500	60.8	30.47	14.966
0	59.0	29.921	14.696
500	57.2	29.38	14.430
1000	55.4	28.86	14.175
2000	51.9	27.82	13.664
3000	48.3	26.82	13.173
4000	44.7	25.82	12.682
5000	41.2	24.90	12.230
6000	37.6	23.98	11.778
7000	34.0	23.09	11.341
8000	30.5	22.22	10.914
9000	26.9	21.39	10.506
10,000	23.4	20.58	10.108
15,000	5.5	16.89	8.296
20,000	–12.3	13.76	6.758
30,000	–47.8	8.90	4.371
40,000	–69.7	5.56	2.731
50,000	–69.7	3.44	1.690
60,000	–69.7	2.14	1.051

Source: Adapted from NASA (1976).

2.3 Forms of Energy

2.3.1 Energy

Energy is the capacity for producing an effect. Thermodynamics is founded on the *law of conservation of energy*, which says that energy can neither be created nor destroyed. Heat and work are transitory forms of energy, losing their identity as soon as they are absorbed by the body or region to which they are delivered. Work and heat are not possessed by a system and, therefore, are not properties. Thus, if there is a net transfer of energy across the boundary from a system (as heat and/or work), from where did this energy come? The only answer is that it must have come from a store of energy in the given system. These stored forms of energy may be assumed to reside within the bodies or regions with which they are associated. In thermodynamics, accent is placed on the changes of stored energy rather than on absolute quantities.

2.3.2 Stored Forms of Energy

Energy may be stored in such forms as thermal (internal), mechanical, electrical, chemical, and atomic (nuclear).

Internal Energy, *U*. Internal (thermal) energy is the energy possessed by matter due to the motion and/or position of its molecules. This energy is comprised of two components: (1) kinetic internal energy—due to the velocity of the molecules and manifested by temperature; and (2) potential internal energy—due to the attractive forces existing between molecules and manifested by the phase of the system.

Potential Energy, P.E. Potential energy is the energy possessed by the system due to the elevation or position of the system. This potential energy is equivalent to the work

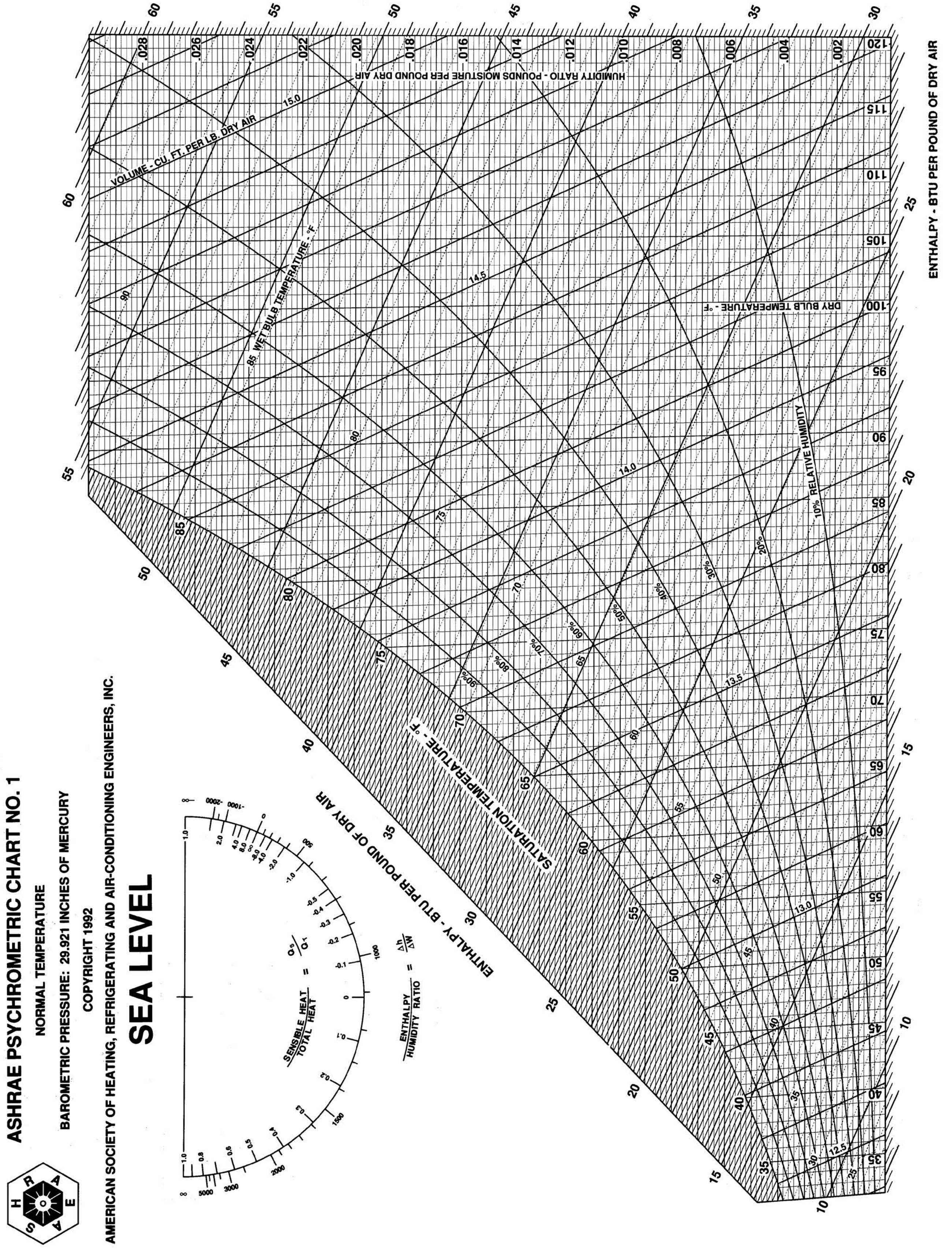


Fig. 2-10 ASHRAE Psychrometric Chart No. 1

required to lift the system from the arbitrary datum (0 elevation) to its elevation z in the absence of friction.

$$F = \frac{ma}{g_c} = \frac{mg}{g_c} \tag{2-39}$$

$$\text{P.E.} = W = \int_0^x F\,dx = \int_0^z m\frac{g}{g_c}dx = m\frac{g}{g_c}z$$

Kinetic Energy, K.E. Kinetic energy is the energy possessed by the system as a result of the velocity of the system. It is equal to the work that could be done in bringing to rest a system that is in motion, with a velocity V, in the absence of gravity.

$$F = \frac{ma}{g_c} = \frac{m}{g_c}\frac{dV}{dt} \tag{2-40}$$

$$\text{K.E.} = W = \int_0^x F\,dx = -\int_V^0 \frac{m}{g_c}\frac{dV}{dt}dx$$

$$= -\int_V^0 \frac{m}{g_c}V\,dV = \frac{mV^2}{2g_c}$$

Chemical Energy, E_c. Chemical energy is possessed by the system because of the arrangement of the atoms composing the molecules. Reactions that liberate energy are termed *exothermic* and those that absorb energy are termed *endothermic*.

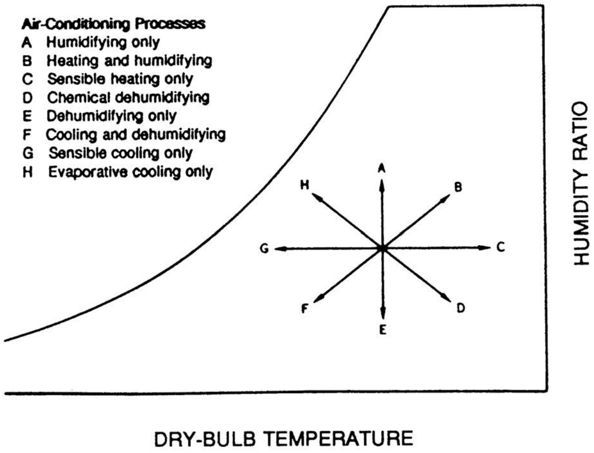


Fig. 2-11 Psychrometric Representations of Basic Air-Conditioning Process

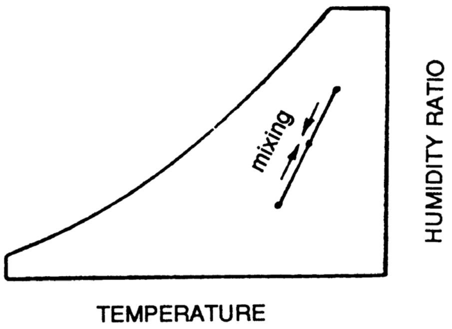


Fig. 2-12 Adiabatic Mixing

Nuclear (Atomic) Energy, E_a. Nuclear energy is possessed by the system due to the cohesive forces holding the protons and neutrons together as the nucleus of the atom.

Stored energy is concerned with

- Molecules of the system (internal energy)
- The system as a unit (kinetic and potential energy)
- Arrangement of the atoms (chemical)
- Nucleus of the atom (nuclear)

Molecular stored energy is associated with the relative position and velocity of the molecules; the total effect is called internal energy. Kinetic energy and potential energy are both forms of mechanical energy, and they can be converted readily and completely into work. Chemical, electrical, and atomic energy would be included in any accounting of stored energy; however, engineering thermodynamics frequently confines itself to systems not undergoing changes in these three forms of energy.

2.3.3 Transient Forms of Energy

Heat, Q. Heat is the mechanism by which energy is transferred across the boundary between systems by reason of the difference in temperature of the two systems, and always in the direction of the lower temperature. Being transitory, heat is not a property. It is redundant to speak of heat as being transferred, for the term *heat* signifies energy in transit. Nevertheless, in keeping with common usage, this text refers to heat as being transferred.

Although a body or system cannot contain heat, it is useful, when discussing many processes, to speak of heat received or heat rejected, so that the direction of heat transfer relative to the system is obvious. This should not be construed as treating heat as a substance.

Heat *transferred* to a system is considered to be positive, and *heat transferred* from a system, negative. Thus, positive heat represents energy transferred to a system, and negative heat represents energy transferred from a system. A process in which there is no heat transfer ($Q = 0$) is called an **adiabatic process**.

Work. Work is the mechanism by which energy is transferred across the boundary between systems by reason of the difference in pressure (or force of any kind) of the two systems, and is in the direction of the lower pressure.

If the total effect produced in the system can be reduced to the raising of a weight, then nothing but work has crossed the boundary. Work, like heat, is not possessed by the system, but occurs only as energy being transferred.

Work is, by definition, the energy resulting from a force having moved through a distance. If the force varies with distance x, work may be expressed as $\delta W = F\,dx$.

In thermodynamics, work is often done by a force distributed over an area, i.e., by pressure p acting through volume V, as in the case of fluid pressure exerted on a piston. In this event,

$$\delta W = p\,dV$$

where p is an external pressure exerted on the system.

Mechanical or shaft work W is the energy delivered or absorbed by a mechanism, such as a turbine, air compressor, or internal combustion engine. Shaft work can always be evaluated from the basic relation for work.

Power is the rate of doing work.

Work done *by* a system is considered positive; work done *on* a system is considered negative. The symbol W designates the work done by a system.

Work may be done on or by a system in a variety of ways. In addition to mechanical work and flow work (the types most frequently encountered in thermodynamics), work may be done due to surface tension, the flow of electricity, magnetic fields, and in other ways.

For nonflow processes, the form of mechanical work most frequently encountered is that done at the moving boundary of a system, such as the work done in moving the piston in a cylinder. It may be expressed in equation form for reversible processes as $W = \int pdV$. Generally, for the nonflow process, work can be expressed as

$$W = \int p\,dv\ldots \tag{2-41}$$

where the dots indicate other ways in which work can be done by or on the system.

The following section shows the derivation of a useful expression for the work of a frictionless steady-flow process. The derivation procedure is

1. Make a free-body diagram of an element of fluid.
2. Evaluate the external forces on the free body.
3. Relate the sum of the external forces to the mass and acceleration of the free body.
4. Solve the resulting relation for the force by which work is done on the fluid.
5. Apply the definition of work as $\int Fds$. A free-body diagram is illustrated in Figure 2-13.

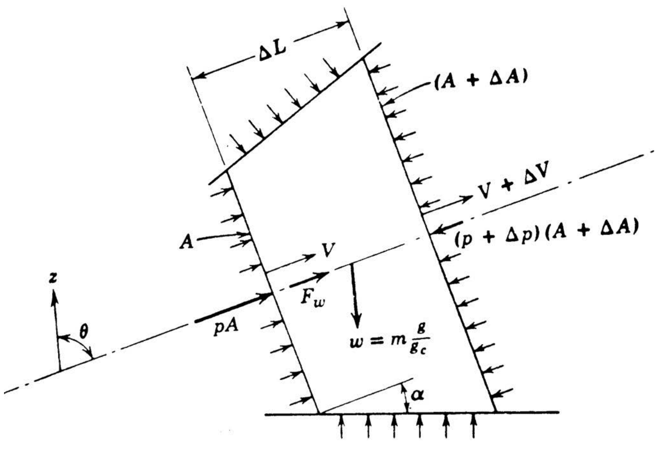


Fig. 2-13 Element of Fluid in Frictionless Steady Flow

Applying Newton's second law of motion, the sum of the external forces on the fluid element must equal ma/g_c. The mass of the element is $\rho(A + \Delta A/2)\Delta L$, and the acceleration is approximately $\Delta V/\Delta\tau$. Thus

$$\Sigma F = \frac{ma}{g_c} = \frac{\rho}{g_c}\left(A + \frac{\Delta A}{2}\right)\Delta L\frac{\Delta V}{\Delta\tau}$$

The sum or resultant of the forces is

$$\begin{aligned}\Sigma F &= pA - (p + \Delta p)(A + \Delta A) - m\frac{g}{g_c}\cos\theta \\ &\quad + \left(p + \frac{\Delta p}{2}\right)\Delta A + F_W \\ &= -A\,\Delta p - \frac{\Delta p \Delta A}{3} - m\frac{g}{g_c}\cos\theta + F_w\end{aligned}$$

$$\text{Work}_{\text{in}} = F_w \Delta L = (\text{Volume})\Delta p + \frac{V\Delta V}{g_C} + \frac{g}{g_c}\Delta z$$

and per unit mass

$$\text{Work}_{\text{in}} = v\,\Delta p + \frac{V\Delta V}{g_C} + \frac{g}{g_c}\Delta Z$$

Now, if ΔL is made to approach dL, then the other differences also approach differentials, and the work (per unit mass) done on the fluid in the distance dL is

$$\delta\text{Work}_{\text{in}} = v\,dp + \frac{VdV}{g_c} + \frac{g}{g_c}dz \tag{2-42}$$

or, for flow between sections a finite distance apart

$$\text{Work}_{\text{in}} = \int v\,dp + \Delta\left(\frac{V^2}{2g_c}\right) + \frac{g}{g_c}\Delta z$$

This important relation shows the mechanical work done on a unit mass of fluid in a frictionless steady-flow process.

In addition to commonly encountering work done at a moving boundary in an **open system**, flow work must be considered. **Flow work** consists of the energy carried into or transmitted across the system boundary because of the work done by the fluid just outside the system on the adjacent fluid entering the system to force or push it into the system. Flow work also occurs as fluid leaves the system, this time by fluid in the system on the fluid just leaving the system. As an analogy, consider two people as particles of fluid, one in the doorway and one just outside. Flow work would be done by the person outside if he or she shoved the person in the doorway into the room (system).

$$\begin{aligned}\text{Flow work} &= \int F dx = \int p dAx, \qquad p = c \\ &= p\int_0^v dV = pv\end{aligned} \tag{2-43}$$

where flow work is per unit mass and v is the specific volume, or the volume displaced per unit mass.

This analysis shows that work must be done in causing fluid to flow into or out of a system. This work is called flow work. Such terms as flow energy and displacement energy are sometimes used.

2.4 First Law of Thermodynamics

The first law of thermodynamics is often called the *law of conservation of energy*. From the first law, it can be concluded that for any system, open or closed, there is an energy balance as

Net amount of energy added to system	=	Net increase in stored energy of system

or

Energy in – Energy out =

Increase in energy in system (2-44)

With both open and closed systems, energy can be added to or taken from the system by means of heat and work. In an open system, there is an additional mechanism for increasing or decreasing the stored energy of the system. When mass enters a system, the stored energy of the system is increased by the stored energy of the entering mass. The stored energy of a system is decreased whenever mass leaves the system because the mass leaving the system takes some stored energy with it. To distinguish this transfer of stored energy of mass crossing the system boundary from heat and work, consider

	Stored energy of mass entering system	–	Stored energy of mass leaving system
+	Net energy added to system heat and work	=	Net increase in stored energy of system

The net exchange of energy between the system and its surroundings must be balanced by the change in energy of the system. Exchange of energy in transition includes either work or heat. However, what is meant by the energy of the system and the energy associated with any matter entering or leaving the system must be described further.

The energy E of the system is a property of the system and consists of all of the various forms in which energy is characteristic. These forms include potential energy (due to position), kinetic energy (due to any motion), electrical energy (due to charge), and so forth. Because work and heat are energy in transition and are not characteristic of the system, they are not included here.

All the energy of a system—exclusive of kinetic and potential energy—is called *internal energy*. The symbol for internal energy per unit mass is u. The symbol for internal energy contained in a mass of m pounds or kilograms is U. Each unit of mass flowing into or out of the system carries with it the energy characteristic of that unit of mass. This energy includes the internal energy u plus the kinetic and potential energy.

Work is always done on or by a system where fluid flows across the system boundary. Therefore, work in an energy balance for an open system is usually separated into two parts: (1) the work required to push a fluid into or out of the system; and (2) all other forms of work.

The work flow per unit mass crossing the boundary of a system is pv. If the pressure, specific volume, or both vary as a fluid flows across a system boundary, the flow work is calculated by integrating $\int pv\delta m$, where δm is an infinitesimal mass crossing the boundary. The symbol δm is used instead of dm because the amount of mass crossing the boundary is not a property. The mass within the system is a property, so the infinitesimal change in mass within the system is properly represented by dm.

Since the work term in an energy balance for an open system is usually separated into two parts, (1) flow work and (2) all other forms of work, the term work W without modifiers stands for all other forms of work except flow work, and the complete, two-word name is always used when referring to flow work.

An equation representing the first law can be written using the symbols defined for the general system of Figure 2-14. Referring to Figure 2-14, let δm_1 be the mass entering the system and δm_2 be the mass leaving. The first law in differential or incremental form becomes

$$[\delta m(e+pv)]_{\text{in}} - [\delta m(e+pv)]_{\text{out}} + \delta Q - \delta W = dE \qquad (2\text{-}45)$$

or

$$\delta m_1\left(u_1 + p_1v_1 + \frac{V_1^2}{2g_c} + z_1\frac{g}{g_c}\right)$$

$$-\delta m_2\left(u_2 + p_2v_2 + \frac{V_2^2}{2g_c} + z_2\frac{g}{g_c}\right) + \delta Q - \delta W = dE$$

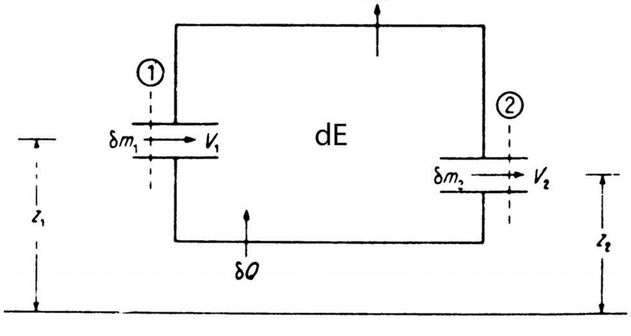


Fig. 2-14 Energy Flows in a General Thermodynamic System

where δQ and δW are the increments of work and heat, and dE is the differential change in the energy of the system. Because E or U (or e or u) are properties of the system, they are treated like any other property such as temperature, pressure, density, or viscosity.

The combination of properties $u + pv$ is also a property, which has been defined as enthalpy. The symbol H stands for the total enthalpy associated with a mass m; h stands for specific enthalpy, or enthalpy per unit mass. In equation form

$$h = u + pv$$

In terms of enthalpy, the first law equation becomes

$$\begin{aligned} &\delta m_1\left(h_1 + \frac{V_1^2}{2g_c} + \frac{g}{g_c}z_1\right) \\ &- \delta m_2\left(h_2 + \frac{V_2^2}{2g_c} + \frac{g}{g_c}z_2\right) + \delta Q - \delta W = dE \end{aligned} \tag{2-46}$$

or in integrated form,

$$\begin{aligned} &\int_0^{m_1} \delta m_1\left(h_1 + \frac{V_1^2}{2g_c} + \frac{g}{g_c}z_1\right) \\ &- \int_0^{m_2} \delta m_2\left(h_2 + \frac{V_2^2}{2g_c} + \frac{g}{g_c}z_2\right) \\ &+ \delta Q - \delta W = E_{\text{final}} - E_{\text{initial}} \end{aligned} \tag{2-47}$$

or, if divided by the time interval $\Delta\tau$,

$$\frac{\delta m_1}{\Delta\tau}\left(h_1 + \frac{V_1^2}{2g_c} + \frac{g}{g_c}z_1\right) - \frac{\delta m_2}{\Delta\tau}\left(h_2 + \frac{V_2^2}{2g_c} + \frac{g}{g_c}z_2\right)$$

$$+\frac{\delta Q}{\Delta\tau} - \frac{\delta W}{\Delta\tau} = \frac{dE}{\Delta\tau}$$

$$\Delta\tau \to 0, \quad \frac{\delta Q}{\Delta\tau} \to \dot{Q}, \quad \frac{\delta W}{\Delta\tau} \to W$$

$$\frac{\delta m_1}{\Delta\tau} \to \dot{m}_1, \quad \frac{\delta m_2}{\Delta\tau} \to \dot{m}_2, \quad \frac{dE}{\Delta\tau} \to \frac{dE}{d\tau}$$

$$\begin{aligned} &\dot{m}_1\left(h_1 + \frac{V_1^2}{2g_c} + \frac{g}{g_c}z_1\right) - \dot{m}_2\left(h_2 + \frac{V_2^2}{2g_c} + \frac{g}{g_c}z_2\right) \\ &+ \dot{Q} - \dot{W} = \frac{dE}{d\tau} \end{aligned} \tag{2-48}$$

where $\dot{Q}$ is heat flow and $\dot{W}$ is the work rate or power.

The most general case in which the prior integration of the first law is possible has the following conditions:

1. The properties of the fluids crossing the boundary remain constant at each point on the boundary.
2. The flow rate at each section where mass crosses the boundary is constant. (The flow rate cannot change as long as all properties, including velocity, remain constant at each point.)
3. All interactions with the surroundings occur at a steady rate.

Integration then yields

$$\begin{aligned} &\sum m_{\text{in}}\left(u + pv + \frac{V_1^2}{2g_c} + \frac{g}{g_c}z\right)_{\text{in}} \\ &- \sum m_{\text{out}}\left(u + pv + \frac{V_2^2}{2g_c} + \frac{g}{g_c}z\right)_{\text{out}} + Q - W \\ &= \left[m_f\left(u + \frac{V^2}{2g_c} + \frac{gz}{g_c}\right) - m_i\left(u + \frac{V^2}{2g_c} + \frac{gz}{g_c}\right)_i\right]_{\text{system}} \end{aligned} \tag{2-49}$$

A special case in engineering applications is the **steady-flow process**. In steady flow, all quantities associated with the system do not vary with time. Consequently,

$$\begin{aligned} &\sum \dot{m}\left(h + \frac{V^2}{2g_c} + \frac{g}{g_c}z\right)_{\text{in}} \\ &- \sum \dot{m}\left(h + \frac{V^2}{2g_c} + \frac{g}{g_c}z\right)_{\text{out}} + \dot{Q} - \dot{W} = 0 \end{aligned} \tag{2-50}$$

A second common application is the **closed-stationary system**, for which the first law equation reduces to

$$Q - W = [m(u_f - u_i)]_{\text{system}} \tag{2-51}$$

Example 2-1 Nitrogen having a mass of 0.85 kg expands in an irreversible manner doing 572 J of work. The temperature of the nitrogen drops from 81°C to 34°C. Determine the heat transfer.

Solution:

First law for stationary closed system

$$Q - W = m(u_f - u_i)$$

For gases

$$U_f - U_i = mc_v(t_f - t_i)$$

$$Q - 572/1000 = 0.85(0.741)(34 - 81)$$

$$Q = -29.03 \text{ kJ } (-27.5 \text{ Btu})$$

Example 2-2 A tank having a volume of 40 ft^3 is initially evacuated. This tank is connected to an air line. The air in this line has an internal energy of 80 Btu/lb, a pressure of 100 psia, and a specific volume of 32 ft^3/lb. The valve is opened, the tank fills with air until the pressure is 100 psia, and then the valve is closed. The process takes place adiabatically, and kinetic and potential energies are negligible. Determine the final internal energy of the air in the tank in Btu/lb.

Solution:

First law

$$m_{in}(u + pv)_{in} = m_f u_f$$

Conservation of mass

$$m_f = m_{in}$$

$$(u + pv)_{in} = u_f = 80 + (100)(144)(32)/778$$

$$= 673 \text{ Btu/lb } (1565 \text{ kJ/kg})$$

2.5 Second Law of Thermodynamics

2.5.1 Second Law from Classical Thermodynamics

As a generality, the second law deals with the fact that many processes proceed only in one direction, and not in the opposite direction. Everyday examples include the fact that a cup of hot coffee cools to its surroundings, but the cooler surroundings never heat up the warmer cup of coffee; or the fact that the chemical energy in gasoline is used as a car is driven up a hill, but coasting down the hill does not restore the gasoline. The second law of thermodynamics is a formalized statement of such observations.

A system that undergoes a series of processes and returns to its initial state is said to go through a cycle. For the closed system undergoing a cycle, from the first law of thermodynamics,

$$\oint \delta Q = \oint \delta W$$

The symbol $\oint$ stands for the cyclical integral of the increment of heat or work. Any heat supplied to a cycling system must be balanced by an equivalent amount of work done by the system. Conversely, any work done on the cycling system gives off an equivalent amount of heat.

Many examples exist of work being completely converted into heat. However, a cycling system that completely converts heat into work has never been observed, although such complete conversion would not be a violation of the first law. The fact that heat cannot be completely converted into work is the basis for the second law of thermodynamics. The justification for the second law is based on empirical observations.

The second law has been stated in different ways; one is the *Kelvin-Planck statement* of the second law which states, *It is impossible for any cycling device to exchange heat with only a single reservoir and produce work.*

In this context, reservoir refers to a body whose temperature remains constant regardless of how much heat is added to or taken from it. In other words, the Kelvin-Planck statement says that heat cannot be continuously and completely converted into work; a fraction of the heat must be rejected to another reservoir at a lower temperature. The second law thus restricts the first law in relation to the way energy is transferred. Work can be continuously and completely converted into heat, but not vice versa.

If the Kelvin-Planck statement were not true and heat could be completely converted to work, heat might be obtained from a low-temperature source, converted into work, and the work converted back into heat in a region of higher temperature. The net result of this series of events would be the flow of heat from a low-temperature region to a high-temperature region with no other effect. This phenomenon has never been observed and is contrary to all experience.

The Clausius statement of the second law is, *No process is possible whose sole result is the removal of heat from a reservoir at one temperature and the absorption of an equal quantity of heat by a reservoir at a higher temperature*.

Two major consequences related to the Kelvin-Planck and Clausius statements of the second law are the limiting values for thermal efficiency of power systems operating on heat and for the coefficients of performance for heat pumps. In accordance with the Kelvin-Planck concept, the maximum possible thermal efficiency (work output/heat input) of a heat engine operating between temperature reservoirs at T_H and T_L is

$$\eta_{\text{thermal, max}} = [1 - T_L/T_H]\ 100$$

The maximum possible coefficient of performance for cooling (cooling effect/work input) is

$$\text{COP}_{c,\max} = T_L/(T_H - T_L)$$

and the maximum possible coefficient of performance for heating (heating effect/work input) is

$$\text{COP}_{h,\max} = T_H/(T_H - T_L)$$

2.5.2 Second Law from Statistical Thermodynamics

To help understand the significance of the second law of thermodynamics, consider the molecular nature of matter. Although a sample of a gas may be at rest, its molecules are not. Rather, they are in a state of continuous, random motion, with an average speed of the same order of magnitude as the speed of sound waves in the gas. For air, this is about 1100 ft/s (335 m/s) at room temperature. Some of the molecules move more rapidly than this and some more slowly.

Due to collisions with one another and with the walls of the containing vessel, the velocity of any one molecule is continually being changed in magnitude and direction. The number of molecules traveling in a given direction with a given speed, however, remains constant. If the gas as a whole is at rest, the molecular velocities are distributed randomly in any direction. From the standpoint of conservation of energy (the first law), molecules of a gas, flying about in all directions and with a wide range of speeds, could get together in a cooperative effort and simultaneously acquire a common velocity component in the same direction, although it is unexpected. Nevertheless, these phenomena have been observed and are described as fluctuations.

Hence, processes whose sole result is the flow of heat from a heat reservoir and the performance of an equivalent amount

of work, do not occur with sufficient frequency or with objects of sufficient size to make them useful.

Thus, an accurate statement of the second law should replace the term *impossible* with *improbable*. Therefore, the second law is a statement of the improbability of the spontaneous passage of the system from a highly probable state (random or disordered) to one of lower probability.

2.5.3 Physical Meaning of Entropy

Entropy is a measure of the random mix or of the probability of a given state. The more completely shuffled any system is, the greater is its entropy; conversely, an orderly or unmixed configuration is one of low entropy. Thus, when a substance reaches a state in which all randomness disappears, it then has zero entropy.

In 1851, Rankine analytically demonstrated that the ratio of the heat exchanged in a reversible process to the temperature of the interaction defined a thermodynamic function that was not consumed in a reversible cycle. The following year, Clausius independently derived the same result, but identified the function as a property of a system, and designated it as the entropy.

The concept of energy serves as a measure of the quantity of heat, but entropy serves as a measure of its quality. Clausius also concluded that although the energy of the world is constant, the entropy increases to a maximum, due to the irreversible nature of real processes. This extreme represents a condition when there are no potential differences in the universe.

In an irreversible process, the entropy of the universe is irretrievably increased, but no energy is lost. From the engineer's point of view, however, the opportunity to convert internal energy to mechanical energy is lost. An example is the mixing of hot and cold water from two reservoirs into a single reservoir. The internal energy of the system is the same before as after mixing, but at the end of the process, no heat can be withdrawn from the single reservoir to operate a cyclic machine. An engine could have been operated between the original hot and cold reservoirs, but once the reservoirs have come to the same temperature, this opportunity is lost.

Entropy is not conserved, except in reversible processes. When a beaker of hot water is mixed with a beaker of cold water, the heat lost by the hot water equals the heat gained by the cold water and energy is conserved. On the other hand, while the entropy of the hot water decreases, the entropy of the cold water increases a greater amount, and the total entropy of the system is greater at the end of the process than it is at the beginning. Where did this additional entropy come from? It was created in the process. Furthermore, once entropy has been created, it can never be destroyed. *Energy can neither be created nor destroyed*, states the first law of thermodynamics. *Entropy cannot be destroyed, but it can be created*, states the second law.

2.5.4 Entropy Equation of the Second Law of Thermodynamics

For the general case of an open system, the second law can be written

$$dS_{\text{system}} = (\delta Q/T)_{\text{rev}} + \delta m_i s_i \; - \; \delta m_e s_e + dS_{\text{irr}} \qquad (2\text{-}52)$$

where

$\delta m_i s_i$ = entropy increase due to the mass entering
$\delta m_e s_e$ = entropy decrease due to the mass leaving
$\delta Q/T$ = entropy change due to reversible heat transfer between the system and surroundings
dS_{irr} = entropy created due to irreversibilities

The equation accounts for all the entropy changes in the system. Rearranging,

$$\delta Q = T[\delta m_e s_e \; - \; \delta m_i s_i) + dS_{\text{sys}} - dS_{\text{irr}}] \qquad (2\text{-}53)$$

In integrated form—subject to the restrictions that inlet and outlet properties, mass flow rates, and interactions with the surroundings do not vary with time—the general equation for the second law is

$$\begin{aligned}(S_f - S_i)_{\text{out}} &= \int_{\text{rev}} \delta Q/T + \Sigma(ms)_{\text{in}} \\ &\quad - \Sigma(ms)_{\text{out}} + \Delta S_{\text{produced}}\end{aligned} \qquad (2\text{-}54)$$

Example 2-3 A contact feedwater heater operates on the principle of mixing steam and water. Steam enters the heater at 100 psia and 98% quality. Water enters the heater at 100 psia, 80°F. As a result, 25,000 lb_m/h of water at 95 psia and 290°F leave the heater. No heat transfers between the heater and the surroundings. Evaluate each term in the general entropy equation for the second law.

Solution:

First law

$$m_1 h_1 + m_2 h_2 - m_3 h_3 = 0$$

From Table 2-1

$$h_1 = 298.61 + 0.98(889.2) = 1170$$

$$h_2 = 48.05$$

$$h_3 = 259.4 \text{ Btu/lb}$$

$$s_1 = 0.47439 + 0.98(1.1290) = 1.581$$

$$s_2 = 0.09325$$

$$s_3 = 0.4236 \text{ Btu/lb}\cdot{}^\circ\text{R}$$

$$m_1(1170) + (25000 - m_1)(48.05) = 25000(259.4)$$

$$m_1 = 4710 \text{ lb}_m\text{/h}; \quad m_2 = 20{,}290 \text{ lb}_m\text{/h}$$

Second law

$$m_f s_f - m_i s_i = \int \delta Q/T + m_1 s_1 + m_2 s_2 - m_3 s_3 + \Delta S_{irr}$$

$$\begin{aligned}0 = 0 + 4710(1.581) &+ 20290(0.9325) \\ &- 25000(0.4236) + \Delta S_{irr}\end{aligned}$$

$$\Delta S_{\text{irr}} = 1150$$

$$m_f s_f - m_i s_i = \int \delta Q/T + m_1 s_1 + m_2 s_2 - m_3 s_3 + \Delta S_{irr}$$

$$0 = 0 + 7450 + 2000 - 10600 + 1150$$

First Law of Thermodynamics (Energy Balance)

$$\sum m \int_{\text{in}} \left(u + \frac{Pv}{J} + \frac{V^2}{2g_c J} + \frac{g}{g_c}\frac{z}{J} \right)_{\text{in}}$$

$$- \sum m_{\text{out}} \left(u + \frac{Pv}{J} + \frac{V^2}{2g_c J} + \frac{g}{g_c}\frac{z}{J} \right)_{\text{out}} + Q - W$$

$$= \left[m_f \left(u + \frac{V^2}{2g_c J} + \frac{g}{g_c}\frac{z}{J} \right)_f - m_i \left(u + \frac{V^2}{2g_c J} + \frac{g}{g_c}\frac{z}{J} \right)_i \right]_{\text{syster}}$$

Second Law of Thermodynamics (Availability)

General Equation:

$$(S_f - S_i)_{\text{system}} = \int_{\text{rev}} \frac{\delta Q}{T} + \sum (ms)_{\text{in}}$$

$$- \sum (ms)_{\text{out}} + \Delta S_{\text{produced}}$$

Mass Flow (Continuity)

$$m = \frac{AV}{v} = \rho AV$$

Perfect Gas Relations

$$Pv = RT;\ R = \frac{1544}{M};\ c_p - c_v = \frac{R}{J};\ K = \frac{c_p}{c_v}$$

$$\Delta h = h_2 - h_1 = \int_{T_1}^{T_2} c_p dT = c_p(T_2 - T_1)$$

$$\Delta u = u_2 - u_1 = \int_{T_1}^{T_2} c_v dT = c_v(T_2 - T_1)$$

$$\Delta s = s_2 - s_1 = \int_{T_1}^{T_2} c_p \frac{dT}{T} - \frac{R}{J} \ln \frac{P_2}{P_1}$$

$$= c_p \ln \frac{T_2}{T_1} - \frac{R}{J} \ln \frac{P_2}{P_1} \text{ or } c_v \ln \frac{T_2}{T_1} + \frac{R}{J} \ln \frac{v_2}{v_1}$$

Reversible Polytropic Processes

$$pv^n = C = p_1 v_1^n = p_2 v_2^n = \ldots$$

$$\frac{T_2}{T_1} = \left(\frac{P_1}{P_2} \right)^{\frac{n-1}{n}} = \left(\frac{v_1}{v_2} \right)^{n-1}$$

where, if

isentropic, $n = k$ constant volume, $n = \infty$

constant pressure, $n = 0$

constant temperature, $n = 1$ or for general case, $n = n$ as specified by problem

Definitions

Enthalpy, $h = u + Pv/J$

Work, $W = \int Fdx$

$= \int Pdv$ (reversible, closed system)

$= -\int vdP - \Delta ke - \Delta pe$ (reversible, steady flow system)

Fig. 2-15 Basic Equations of Thermodynamics

2.6 Third Law of Thermodynamics

In the late 1800s, Amontons found that a given volume of air, when heated or cooled, expands or contracts by the same amount for each degree of temperature change. He determined this change as about 1/240 of the gas volume at 0°C, which suggests that at –240°C, the volume becomes zero, and at still lower temperature, it has to become negative. Since this observation makes no sense, he concluded that an ultimate low temperature of absolute zero exists. Today, it is known that absolute zero is –273.15°C.

In 1889, Dewar approached absolute zero within 20°C. In 1906, Nernst formulated the third law of thermodynamics that states, *While absolute zero can be approached to an arbitrary degree, it can never be reached*. Postulates to this law conclude that it is not the energy, but the entropy that tends to zero, and that a residual amount of energy is left in any substance, even at absolute zero.

2.7 Basic Equations of Thermodynamics

Figure 2-15 contains the basic set of equations required for most applications of thermodynamics.

2.8 Thermodynamics Applied to Refrigeration

Continuous refrigeration can be accomplished by several processes. In most applications, and almost exclusively in the smaller power range, the vapor compression system (commonly termed the mechanical vapor compression cycle) is used for the refrigeration process. However, absorption systems and steam-jet vacuum systems are being successfully used in many applications. In larger equipment, centrifugal systems are an adaptation of the compression cycle.

A larger number of working fluids (refrigerants) are used in vapor-compression refrigeration systems than in vapor power cycles. Ammonia and sulfur dioxide were first used as vapor-compression refrigerants. Today, the main refrigerants are the halogenated hydrocarbons. Two important considerations in selecting a refrigerant are the temperature at which refrigeration is desired and the type of equipment to be used.

Refrigerants used in most mechanical refrigeration systems are R-22, which boils at –41.4°F (–40.8°C), R-134a, which boils at –15.1°F (–26.2°C) at atmospheric pressure, and R-123 with a boiling point of +82.2°F (+27.9°C) at atmospheric pressure. Past favorites R-12 and R-11 are phased out due to adverse effects on the ozone in the stratosphere. R-22 will be phased out in 2010.

The basic vapor compression cycle is illustrated in Figure 2-16. Cool, low-pressure liquid refrigerant enters the evaporator and evaporates. As it does so, it absorbs heat from another substance, such as air or water, thereby accomplishing refrigeration. The refrigerant then leaves the evaporator as a cool, low-pressure gas and proceeds to the compressor. Here, its pressure and temperature are increased, and this hot,

high-pressure gas is discharged to the condenser. In the condenser, the hot gas is condensed into a liquid. The condensing agent, air or water, is at a temperature lower than the refrigerant gas. This hot, high-pressure liquid flows from the condenser through the expansion valve to the evaporator. The expansion valve reduces the pressure and meters the liquid flow, reducing the hot, high-pressure liquid to a cool, low-pressure liquid as it enters the evaporator.

The basic refrigeration cycle is plotted on the Pressure-Enthalpy diagram as Figures 2-17. Subcooled liquid, at Point A, begins losing pressure as it goes through the metering valve, located at the point where the vertical liquid line meets the saturation curve. As it leaves the metering point, some of the liquid flashes into vapor and cools the liquid entering the evaporator at Point B. Notice that there is additional reduction in pressure from the metering point to Point B, but no change in enthalpy.

As it passes from Point B to C, the remaining liquid picks up heat and changes from a liquid to a gas but does not increase in pressure. Enthalpy, however, does increase. Superheat is added between Point C, where the vapor passes the saturation curve, and Point D.

As it passes through the compressor, Point D to E, the temperature and the pressure are markedly increased, as is the enthalpy, due to the heat of compression. Line E-F indicates that the vapor must be desuperheated within the condenser before it attains a saturated condition and begins condensing. Line F-G represents the change from vapor to liquid within the condenser. Line G-A represents subcooling within the liquid line prior to flow through the metering device.

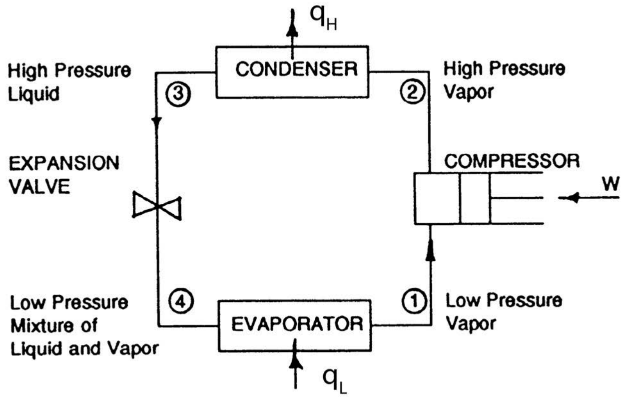


Fig. 2-16 Basic Vapor Compression Refrigeration Cycle

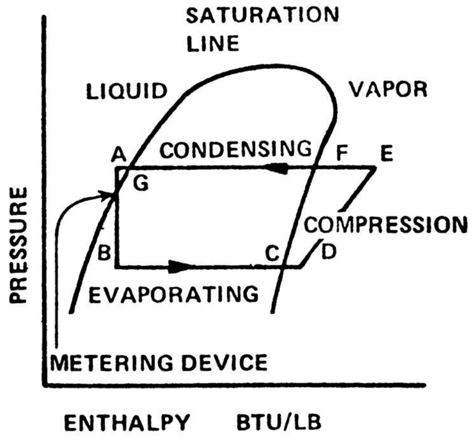


Fig. 2-17 Simplified Pressure-Enthalpy Plot

Note that the pressure remains essentially constant as the refrigerant passes through the evaporator, but due to superheat, its temperature is increased beyond the saturation point before it enters the compressor.

Likewise, the pressure remains constant as the refrigerant enters the condenser as a vapor and leaves as a liquid. While the temperature is constant through the condenser, it is reduced as the liquid is subcooled before entering the metering valve.

The change in enthalpy as the refrigerant passes through the evaporator is almost all latent heat since the temperature does not change appreciably.

Figure 2-18 provides a somewhat more completely labeled *p-h* diagram of the refrigeration cycle.

Applying the first law of thermodynamics to the vapor compression refrigeration system as a whole requires that the sum of all energy in must equal the sum of all energy out when the unit is operating at a steady state rate; hence

$$Q_L + W = Q_H$$

The rate of heat being rejected at the condenser Q_H is numerically greater than the rate at which work is delivered to drive the compressor W and is also greater than the rate of heat absorption into the evaporator Q_L.

This relation shows that every refrigeration cycle operates as a heat pump. The household refrigerator absorbs a quantity of heat Q_L at a low temperature in the vicinity of the ice-making section (the evaporator) and rejects heat Q_H at a higher temperature to the air in the room. The rate of heat rejection Q_H is greater than the rate of absorption Q_L by the power input W to drive the compressor.

In air-conditioning applications the desired effect for cooling is the heat absorbed at the evaporator located inside the conditioned space. Heat is rejected through the condenser outside the conditioned space.

A heat pump uses this same basic cycle for heating. In this application, the condenser is located inside the building and the evaporator is located outside the building where it absorbs heat. Both modes of the cycle are in Figure 2-19.

Example 2-4 An R-134a air-conditioning unit contains a 1 hp (0.746 kW) motor and operates between pressures of 0.2 MPa (in evaporator) and 1.02 MPa (in condenser). Estimate the maximum cooling effect, in kW, that can be expected from this unit.

Solution:

$$\text{COP} = Q_i/W = Q_i/(Q_o - Q_i) = 1/[(Q_o/Q_i) - 1]$$

$$\text{COP}_{\text{max}} = 1/(T_o/T_i - 1) = 1/(313/263 - 1) = 1/0.19 = 5.26$$

$$Q_i = (\text{COP})W = 5.26(0.746)$$
$$= 3.92 \text{ kW } (13{,}400 \text{ Btu/h})$$

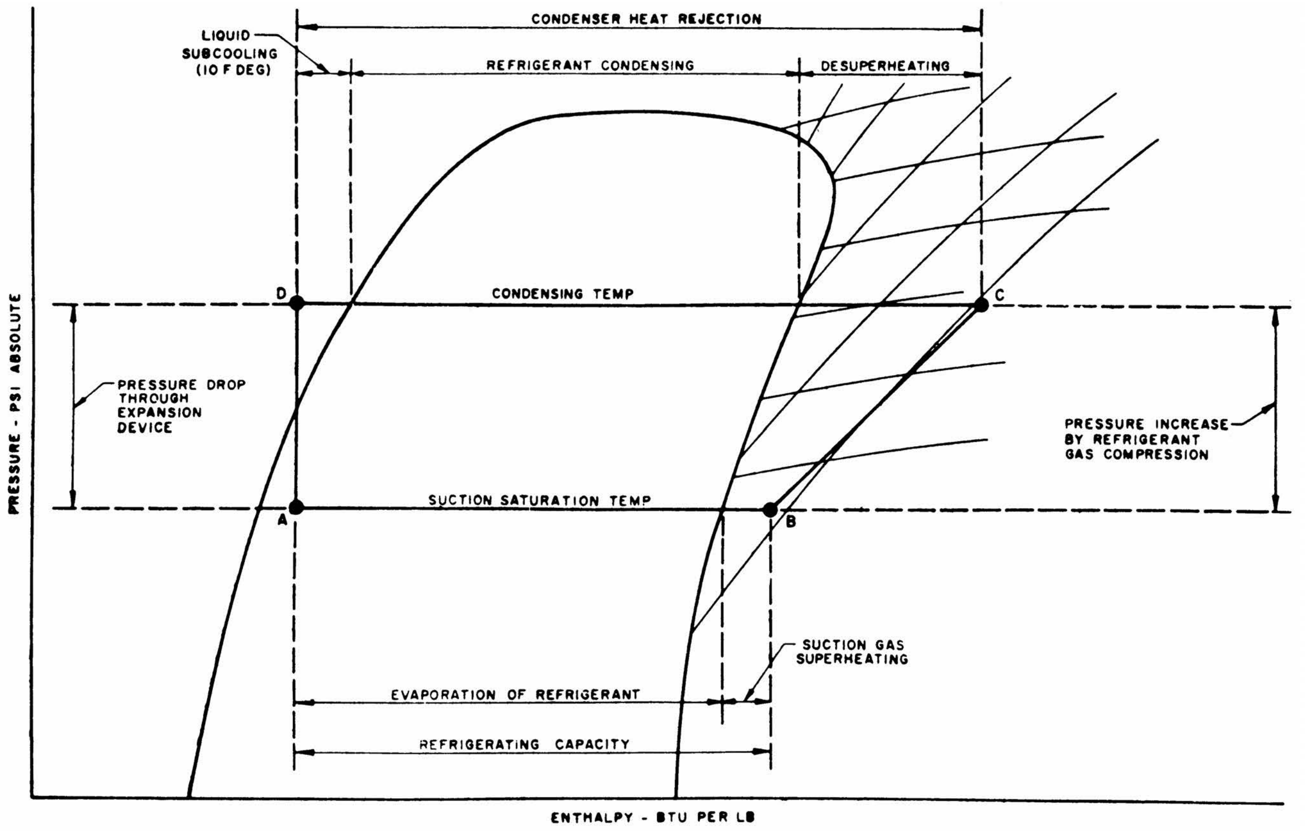


Fig. 2-18 Typical p-h Diagram for the Refrigeration Cycle

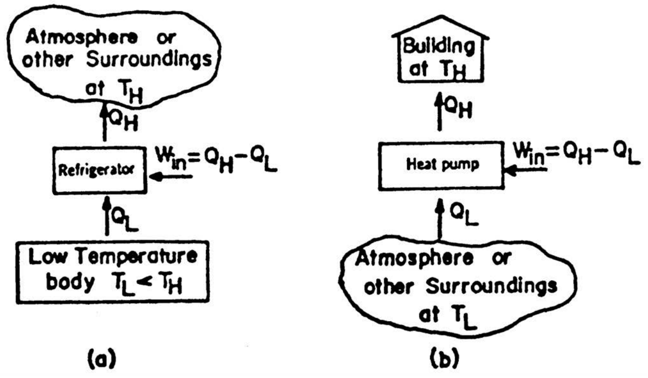


Fig. 2-19 Refrigeration Cycle (a) as Refrigerator, (b) as Heat Pump

2.8.1 Energy Relations for the Basic Refrigeration Cycle

The first law of thermodynamics can be applied to each component of the system individually, since energy must be conserved at each of these, as well as for the entire system.

Compressor. The mass and major energy flows for a compressor are shown in Figure 2-20.

The rate of energy inflow must equal the rate of energy outflow during steady state operation; hence

$$mh_1 + W = mh_2$$

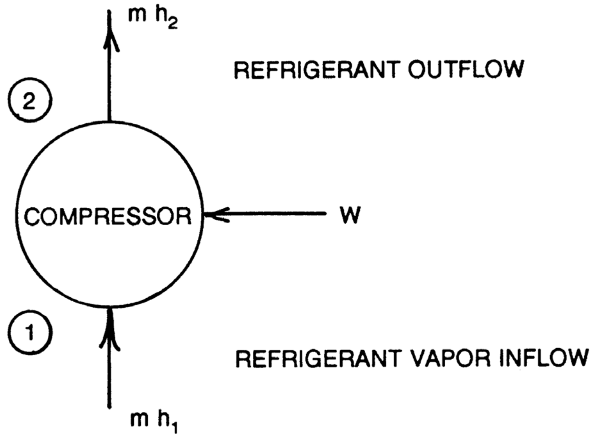


Fig. 2-20 Compressor Flows

$$W = m(h_2 - h_1) \tag{2-55}$$

where

m = rate of refrigerant flow

h_1, h_2 = enthalpies of refrigerant at compressor inlet and outlet

Condenser. The mass and energy flow for a condenser are illustrated in Figure 2-21.

$$mh_2 = mh_3 + Q_R \quad \text{or} \quad Q_R = m(h_2 - h_3) \tag{2-56}$$

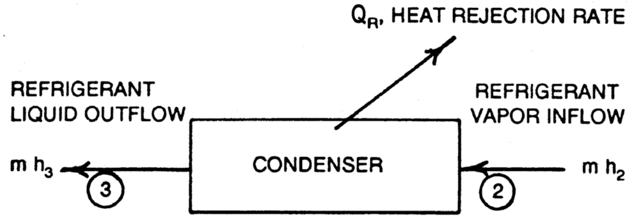


Fig. 2-21 Condenser Flows

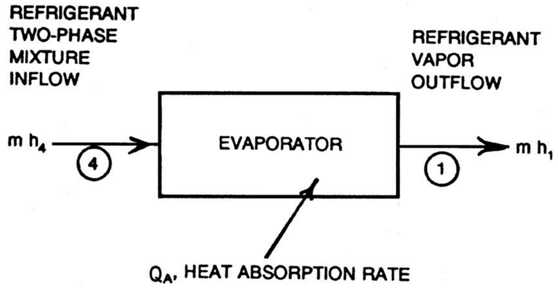
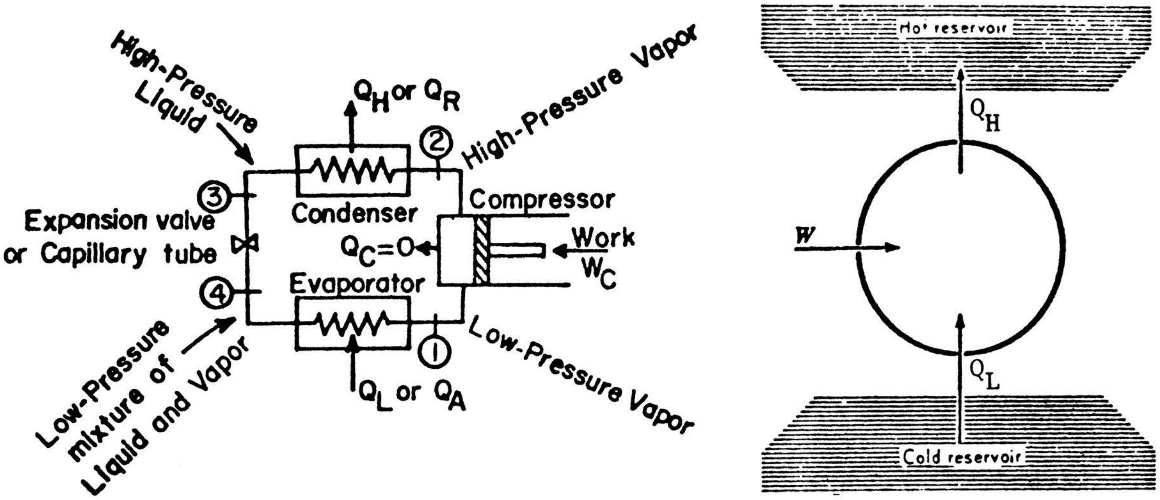


Fig. 2-22 Evaporator Flows



Compressor: $m(h_1 - h_2) + {}_1Q_2 - {}_1W_2 = 0 \qquad W_C = -{}_1W_2$

Condenser: $m(h_2 - h_3) + {}_2Q_3 = 0 \qquad Q_R = -{}_2Q_3 = Q_H$

Expansion Device: $h_3 - h_4 = 0$

Evaporator: $m(h_4 - h_1) + {}_4Q_1 = 0 \qquad Q_A = {}_4Q_1 = Q_L$

Overall: $Q_A + W_C = Q_C + Q_R$

or $({}_1Q_2 + {}_2Q_3 + {}_4Q_1) - ({}_1W_2) = 0$

Coefficient of Performance for Cooling:

$$\mathrm{COP}_c = \frac{\text{Useful effect}}{\text{Input that costs}} = \frac{Q_L}{W} = \frac{1}{Q_H/Q_L - 1} \qquad \mathrm{COP}_{c,\max} = \frac{1}{T_H/T_L - 1}$$

Coefficient of Performance for Heating:

$$\mathrm{COP}_h = \frac{Q_h}{W} \qquad \mathrm{COP}_{h,\max} = \frac{1}{1 - T_L/T_H}$$

Fig. 2-23 The Vapor Compression System

Typically, heat from the condensing refrigerant Q_R is rejected to another fluid. The Q_R leaving the condensing refrigerant must equal the heat absorbed by the fluid receiving it. Thus, from the viewpoint of the condenser cooling fluid,

$$Q_R = m_{\text{fluid}}(c_p)_{\text{fluid}}(t_{\text{out}} - t_{\text{in}}) \qquad (2\text{-}57)$$

Expansion Device. An expansion device is a throttling device or a flow restrictor—a small valve seat opening or a long length of small bore tubing—so neither work nor any significant amount of heat transfer occurs. Hence

$$mh_3 = mh_4$$

or, dividing both sides by m gives

$$h_3 = h_4 \qquad (2\text{-}58)$$

Evaporator. Major mass and energy flows for an evaporator are shown in Figure 2-22.

$$mh_4 + Q_A = mh_1 \text{ or}$$

$$Q_A = m(h_1 - h_4) \qquad (2\text{-}59)$$

Typically, the evaporator receives the heat flow quantity Q_A by heat transfer to it from another fluid—usually water or air. From the viewpoint of that other fluid,

$$Q_A = m_{\text{fluid}}\,(c_p)_{\text{fluid}}\,(t_{\text{in}} - t_{\text{out}})_{\text{fluid}}$$

The results of applying the laws of thermodynamics to the basic vapor-compression refrigeration system are summarized in Figure 2-23.

Example 2-5 A window air conditioner using R-134A is rated at 24,000 Btu/h when operating between an evaporating temperature of 40°F and a condensing temperature of 125°F. A thermostatic expansion valve is used so that the refrigerant vapor leaving the evaporator is superheated by 20°F to safeguard against any liquid entering the compressor. If the compressor efficiency is 62%, determine the adiabatic discharge temperature from the compressor (°F) and the power input (kW). Show the cycle on the *p-h* diagram.

Solution:

Fig. 2-16 provides a sketch with notation for the cycle. Using the *p-h* diagram for R-134A shown below, the following thermodynamic properties are found:

$p_H = 200$ psia (saturation pressure at 125°F)

$p_L = 50$ psia (saturation pressure at 40°F)

$h_3 = h_4 = h_{f\text{ at }125°\text{F}} = 54.2$ Btu/lb

h_1 (50 psia, 60°F) = 113.0 Btu/lb

$s_1 = 0.230$

$h_{2,\text{ ideal}}$ (200 psia, $s = 0.230$) = 126 Btu/lb

From first law, $m(h_1 - h_2) - W = 0$ for the compressor and, $m(h_4 - h_1) + q_L = 0$ for the evaporator.

Thus, $m(54.2 - 113.0) + 24000 = 0$, or

$$m = 408 \text{ lb/h}$$

The ideal compressor work is

$$W_i = 408(113 - 126) = -5304 \text{ Btu/h,}$$

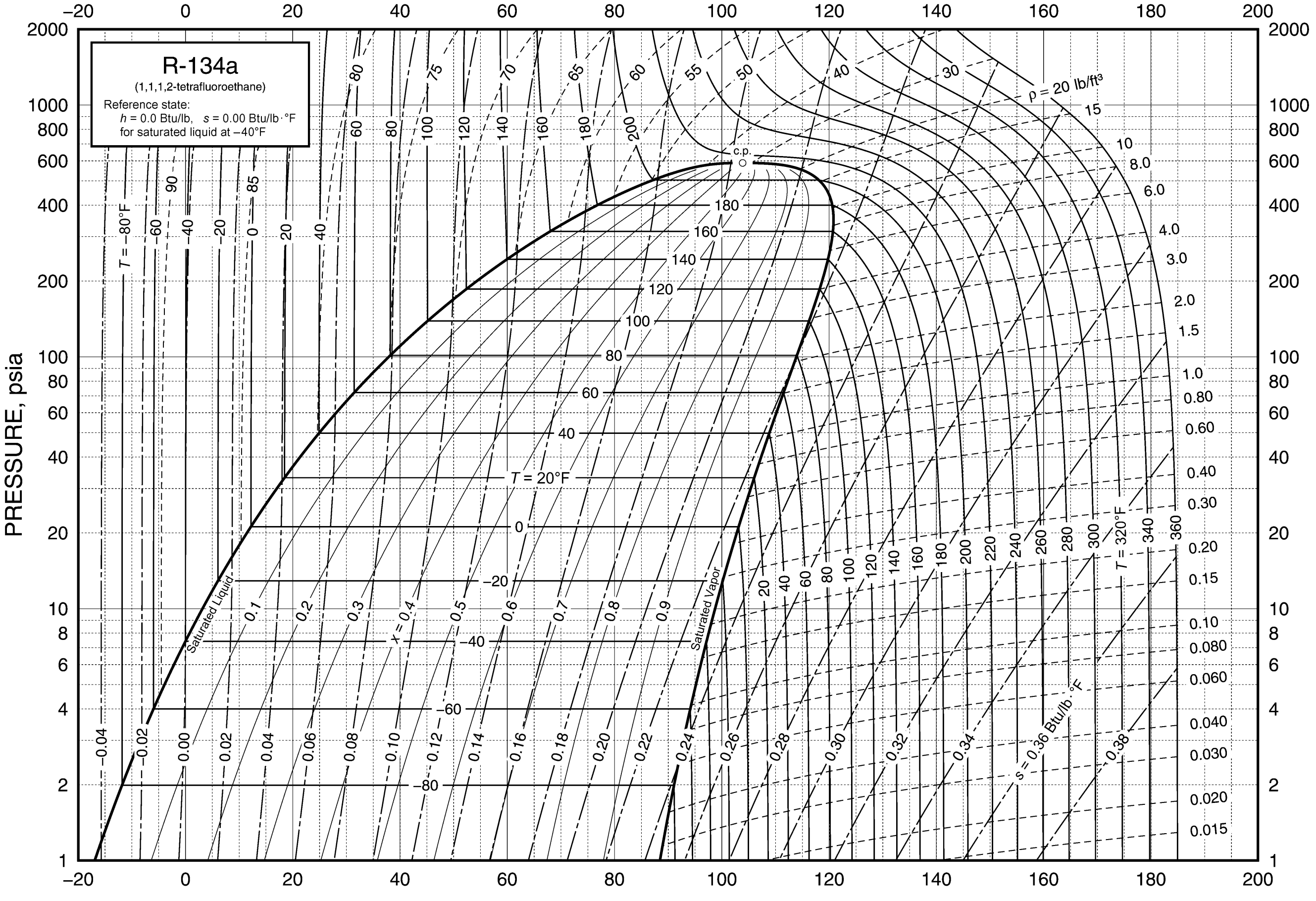
and the actual work using the compressor efficiency is

$$W_a = W_i/\eta = -5304/.62 = -8555 \text{ Btu/h or } 2.5 \text{ kW}$$

The actual enthalpy leaving the compressor can be determined using the efficiency as

$$h_{2a} = h_1 + (h_{2i} - h_i)/\eta = 113 + (126 - 113)/0.62$$
$$= 134 \text{ Btu/lb}$$

$$t_2\ \{200 \text{ psia}, h = 134\} = 180°\text{F (adiabatic discharge)}$$



Properties computed with: **NIST REFPROP** version 7.0

Based on formulation of Tillner-Roth and Baehr (1994)

2.9 Applying Thermodynamics to Heat Pumps

A heat pump is a thermodynamic device that operates in a cycle requiring work; it transfers heat from a low-temperature body to a high-temperature body. The heat pump cycle is identical to a refrigeration cycle in principle, but differs in that the primary purpose of the heat pump is to *supply* heat rather than *remove* it from an enclosed space.

The heat pump cycle can be reversed to provide space cooling (Figure 2-24). A four-way valve switches the heat exchangers so that the indoor exchanger becomes the evaporator and the outdoor heat exchanger becomes the condenser.

The four basic components of a heat pump are the compressor, condenser, expansion device, and evaporator (Figure 2-16). The thermodynamic cycle for a heat pump is identical to the conventional vapor-compression refrigeration cycle (Figure 2-25).

Superheated refrigerant vapor with low pressure and temperature at state 1 is compressed to a much higher pressure and temperature at state 2. The high- pressure, high-temperature gas then passes through the condenser (indoor coil of a heat pump), where it transfers heat to the high-temperature environment and changes from vapor to liquid at high pressure. At state 3, the refrigerant exits the condenser (usually as a subcooled liquid). Next, the refrigerant passes through an expansion device where its pressure drops. This drop in pressure is accompanied by a drop in temperature such that the refrigerant leaves the expansion device and enters the evaporator (outdoor coil of a heat pump) as a low-pressure, low-temperature mixture of liquid and vapor at state 4. Finally, the refrigerant passes through the evaporator, where it picks up heat from the low-temperature environment, changes to all vapor, and exits at state 1.

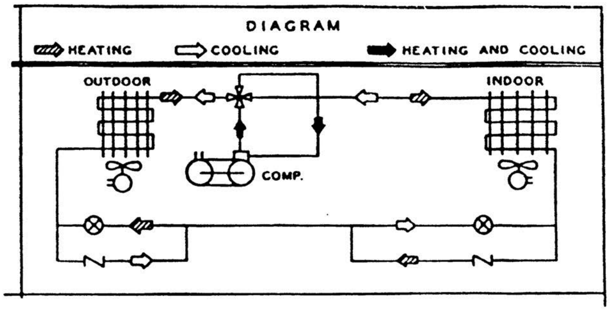


Fig. 2-24 Basic Heat Pump Cycle

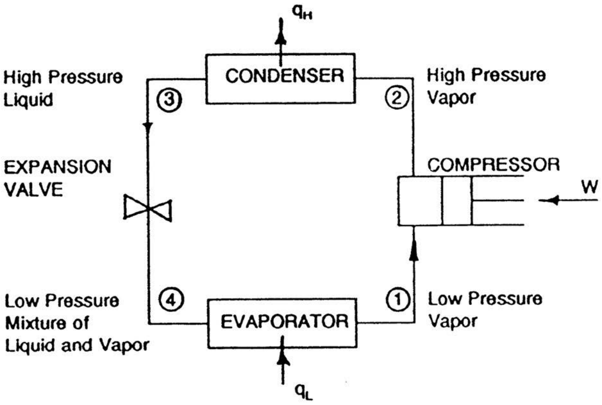


Fig. 2-25 Basic Heat Pump Components

An energy balance on the system shown in Figure 2-25 gives

$$Q_H = Q_L + W$$

where

Q_H = heat energy rejected to the high-temperature environment

Q_L = heat energy taken from the low-temperature environment

W = input work required to move Q_L from the low-temperature environment to the high-temperature environment

The coefficient of performance (COP) for heating equals the heat output divided by the work input:

$$\text{COP} = Q_H/W = (Q_L + W)/W$$

$$\text{COP} = 1 + Q_L/W \tag{2-60}$$

The COP of a heat pump is always greater than one. That is, a heat pump always produces more heat energy than work energy consumed because a net gain of energy Q_L is transferred from the low-temperature to the high-temperature environment.

The heat pump is a reverse heat engine and is therefore limited by the Carnot cycle COP:

$$\text{COP}_{\text{Carnot}} = T_H/(T_H - T_L) \tag{2-61}$$

where

T_L = low temperature in cycle

T_H = high temperature in cycle

The maximum possible COP for a heat pump maintaining a fixed temperature in the heated space is therefore a function of source temperature (Figure 2-26). However, any real heat transfer system must have finite temperature differences across the heat exchangers. The Carnot COP for a typical air-to-air heat pump, as well as the actual COP for the same heat pump, are illustrated in Figure 2-26. The difference between Carnot and actual COPs is due to the nature of real working fluids, flow losses, and compressor efficiency.

2.10 Absorption Refrigeration Cycle

Absorption refrigeration cycles are heat-operated cycles in which a secondary fluid, the absorbent, is used to absorb the primary fluid, gaseous refrigerant, that has been vaporized in the evaporator. The basic absorption cycle is shown in Figure 2-27.

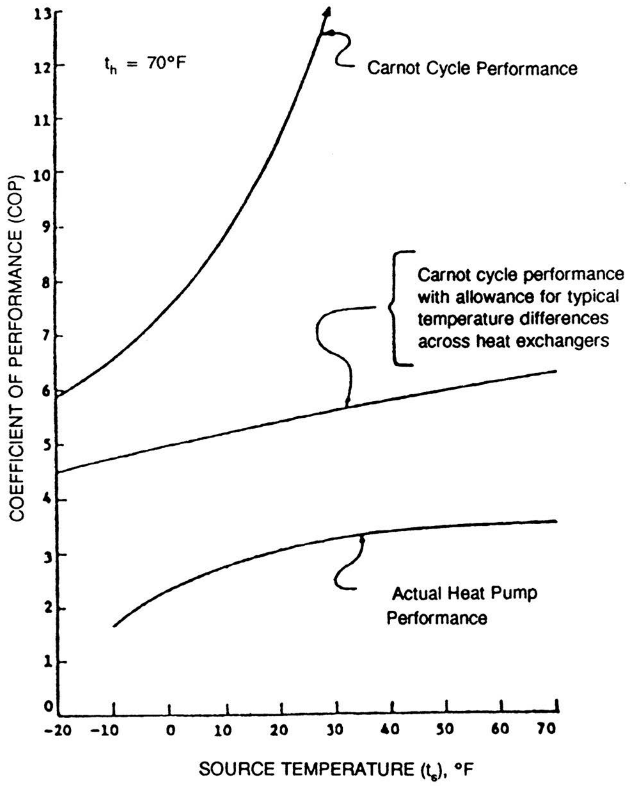


Fig. 2-26 Actual versus Ideal Heat Pump COPs

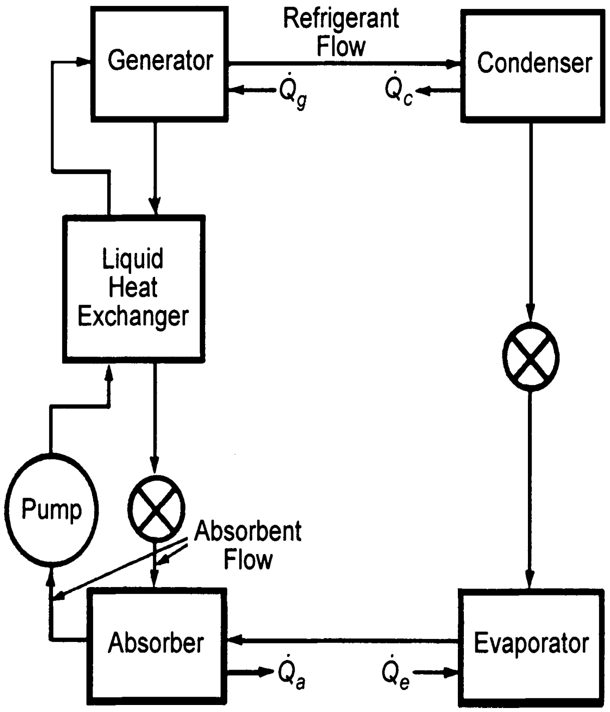


Fig. 2-27 The Basic Absorption Cycle

The low-pressure vapor leaving the evaporator enters the absorber where it is absorbed in the weak solution. Since the temperature is slightly above that of the surroundings, heat is transferred to the surroundings during this process. The strong solution is then pumped through a heat exchanger to the generator at a higher pressure and temperature. As a result of heat transfer, the refrigerant evaporates from the solution. The refrigerant vapor goes to the condenser (as in a vapor-compression system) and then to the expansion valve and evaporator. The weak solution is returned to the absorber through the heat exchanger.

The distinctive feature of the absorption system is that little work input is required because liquids (rather than vapors) are pumped. However, more equipment is involved in an absorption system than in the vapor-compression cycle. Thus, an absorption system is economically feasible only where a low-cost source of heat is available. Additional material on absorption systems is given in Chapter 18 with much greater details available in the 2017 *ASHRAE Handbook—Fundamentals*, Chapter 2, and in the 2014 *ASHRAE Handbook—Refrigeration*, Chapter 18.

2.11 Problems

2.1 Write the first law of thermodynamics in general integrated form.

2.2 Write the second law of thermodynamics in general integrated form.

2.3 Write the following perfect gas relations:

(a) the equation of state
(b) the equation for entropy change
(c) the equation for enthalpy change
(d) the equation for internal energy change

2.4 Write the continuity (mass flow) equation.

2.5 Write the equations for work for

(a) a reversible, closed system
(b) a reversible, steady-flow system

2.6 Two pounds of air contained in a cylinder expand without friction against a piston. The pressure on the back side of the piston is constant at 200 psia. The air initially occupies a volume of 0.50 ft^3. What is the work done by the air in ft·lb$_f$ if the expansion continues until the temperature of the air reaches 100°F? [Ans: 45300 ft·lb$_f$]

2.7 Determine the specific volume, enthalpy, and entropy of 1 kg of R-134a at a saturation temperature of –5°C and a quality of 14%.

2.8 Saturated R-134a vapor at 42°C is superheated at constant pressure to a final temperature of 72°C. What is the pressure? What are the changes in specific volume, enthalpy, entropy, and internal energy?

2.9 A tank having a volume of 200 ft^3 contains saturated vapor (steam) at a pressure of 20 psia. Attached to this tank is a line in which vapor at 100 psia, 400°F flows. Steam from this line enters the vessel until the pressure is 100 psia. If there

is no heat transfer from the tank and the heat capacity of the tank is neglected, calculate the mass of steam that enters the tank. [Ans: 24 lb_m]

2.10 Determine the heat required to vaporize 50 kg of water at a saturation temperature of 100°C.

2.11 The temperature of 150 kg of water is raised from 15°C to 85°C by the addition of heat. How much heat is supplied?

2.12 Three cubic meters per second of water are cooled from 30°C to 2°C. Compute the rate of heat transfer in kilojoules per second (kilowatts). [Ans: 351 000 kW]

2.13 Consider 10 lb_m of air that is initially at 14.7 psia, 100°F. Heat is transferred to the air until the temperature reaches 500°F. Determine the change of internal energy, the change in enthalpy, the heat transfer, and the work done for

(a) a constant-volume process
(b) a constant-pressure process.

2.14 The discharge of a pump is 10 ft above the inlet. Water enters at a pressure of 20 psia and leaves at a pressure of 200 psia. The specific volume of the water is 0.016 ft^3/lb. If there is no heat transfer and no change in kinetic or internal energy, what is the work per pound? [Ans: –0.546 Btu]

2.15 The discharge of a pump is 3 m above the inlet. Water enters at a pressure of 138 kPa and leaves at a pressure of 1380 kPa. The specific volume of the water is 0.001 m^3/kg. If there is no heat transfer and no change in kinetic or internal energy, what is the work per unit mass? [Ans: –30.7 J/kg]

2.16 Air is compressed in a reversible, isothermal, steady-flow process from 15 psia, 100°F to 100 psia. Calculate the work of compression per pound, the change of entropy, and the heat transfer per pound of air compressed.

2.17 Liquid nitrogen at a temperature of –240°F exists in a container, and both the liquid and vapor phases are present. The volume of the container is 3 ft^3 and the mass of nitrogen in the container has been determined as 44.5 lb_m. What is the mass of liquid and the mass of vapor present in the container?

2.18 A fan in an air-conditioning system is drawing 1.25 hp at 1760 rpm. The capacity through the fan is 0.85 m^3/s of 24°C air and the inlet and outlet ducts are 0.31 m in diameter. What is the temperature rise of the air due to this fan? [Ans: 0.9°C]

2.19 Air is contained in a cylinder. Initially, the cylinder contains 1.5 m^3 of air at 150 kPa, 20°C. The air is then compressed reversibly according to the relationship pv^n = constant until the final pressure is 600 kPa, at which point the temperature is 120°C. For this process, determine

(a) the polytropic exponent n
(b) the final volume of the air
(c) the work done on the air and the heat transfer

2.20 Water at 20°C is pumped from ground level to an elevated storage tank above ground level; the volume of the tank is 50 m^3. Initially, the tank contains air at 100 kPa, 20°C, and the tank is closed so that the air is compressed as the water enters the bottom of the tank. The pump is operated until the tank is three-quarters full. The temperature of the air and water remain constant at 20°C. Determine the work input to the pump.

2.21 A centrifugal pump delivers liquid oxygen to a rocket engine at the rate of 100 lb_m/s. The oxygen enters the pump as liquid at 15 psia and the discharge pressure is 500 psia. The density of liquid oxygen is 66.7 lb_m/ft^3. Determine the minimum size motor (in horsepower) to drive this pump. [Ans: 190.4 hp]

2.22 Air undergoes a steady-flow, reversible adiabatic process. The initial state is 200 psia, 1500°F, and the final pressure is 20 psia. Changes in kinetic and potential energy are negligible. Determine

(a) final temperature
(b) final specific volume
(c) change in internal energy per lb_m
(d) change in enthalpy per lb_m
(e) work per lb_m

2.23 Air undergoes a steady-flow, reversible adiabatic process. The initial state is 1400 kPa, 815°C, and the final pressure is 140 kPa. Changes in kinetic and potential energy are negligible. Determine

(a) final temperature
(b) final specific volume
(c) change in specific internal energy
(d) change in specific enthalpy
(e) specific work

2.24 A fan provides fresh air to the welding area in an industrial plant. The fan takes in outdoor air at 80°F and 14.7 psia at the rate of 1200 cfm with negligible inlet velocity. In the 10 ft^2 duct leaving the fan, air pressure is 1 psig. If the process is assumed to be reversible and adiabatic, determine the size motor needed to drive the fan. [Ans: W = 5.1 hp]

2.25 If the fan in the previous problem has an efficiency of 64% and is driven by a motor having an efficiency of 78%, determine the required power, kW.

2.26 A fan provides fresh air to the welding area in an industrial plant. The fan takes in outdoor air at 32.2°C and 101.4 kPa at the rate of 566 L/s with negligible inlet velocity. In the 0.93 m^2 duct leaving the fan, air pressure is 102 kPa. Determine the minimum size motor needed to drive the fan.

2.27 In an insulated feedwater heater, steam condenses at a constant temperature of 220°F. The feedwater is heated from 60°F to 150°F at constant pressure.

(a) Assuming the specific heat at constant pressure of the feedwater is unity, how many Btu are absorbed by each pound in its passage through the heater? [Ans: 90 Btu/lb]
(b) What is the change in entropy of the condensing steam per pound of feedwater heated? [Ans: –0.1324 Btu/lb·R]
(c) What is the change in entropy of 1 lb of feedwater as it passes through the heater? [Ans: +0.1595 Btu/lb·R]
(d) What is the change in entropy of the combined system? Does this violate the second law? Explain. [Ans: +0.0271 Btu/lb·R, No]

2.28 Steam at 124 kPa and 96% quality enters a radiator. The steam is condensed as it flows through the radiator and leaves as condensate at 88°C. If the radiator is to have a heating capacity of 1.85 kW, how many kilograms per hour of steam must be supplied to the radiator?

2.29 Solve the following:

(a) Air at 50 psia and 90°F flows through a restriction in a 2 in. ID pipe. The velocity of the air upstream from the restriction is 450 fpm. If 58°F air is desired, what must the velocity downstream of the restriction be? Comment on this as a method of cooling.
(b) Air at 50 psia and 90°F flows through an insulated turbine at the rate of 1.6 lb_m/s. If the air delivers 11.5 hp to the turbine blades, at what temperature does the air leave the turbine?
(c) Air at 50 psia and 90°F flows through an insulated turbine at the rate of 1.6 lb_m/s to an exit pressure of 14.7 psia. What is the lowest temperature attainable at exit?

2.30 Liquid water at a pressure of 10 psia and a temperature of 80°F enters a 1 in. diameter tube at the rate of 0.8 ft^3/min. Heat is transferred to the water so that it leaves as saturated vapor at 9 psia. Determine the heat transfer per minute. [Ans: 95,800 Btu/min]

2.31 A refrigerator uses R-134a as the refrigerant and handles 200 lb_m/h. Condensing temperature is 110°F and evaporating temperature is 5°F. For a cooling effect of 11,000 Btu/h, determine the minimum size motor (hp) required to drive the compressor.

2.32 A heat pump is used in place of a furnace for heating a house. In winter, when the outdoor air temperature is 10°F, the heat loss from the house is 60,000 Btu/h if the inside is maintained at 70°F. Determine the minimum electric power required to operate the heat pump (in kW).

2.33 A heat pump is used in place of a furnace for heating a house. In winter, when the outdoor air temperature is –10°C, the heat loss from the house is 200 kW if the inside is maintained at 21°C. Determine the minimum electric power required to operate the heat pump. [Ans: 21.1 kW]

2.34 Refrigerant-134a vapor enters a compressor at 25 psia, 40°F, and the mass rate of flow is 5 lb_m/min. What is the smallest diameter tubing that can be used if the velocity of refrigerant must not exceed 20 ft/s?

2.35 An R-134a refrigerating system is operating with a condensing temperature of 86°F and evaporating temperature of 25°F. If the liquid line from the condenser is soldered to the suction line from the evaporator to form a simple heat exchanger, and if as a result of this, saturated liquid leaving the condenser is subcooled 6°F, how many degrees will the saturated vapor leaving the evaporator be superheated? (Use tables.)

2.36 Ammonia is heated in the evaporator of a refrigeration system from inlet conditions of 10°F, 10% quality, to saturated vapor. The pressure remains constant during the process. For each pound, determine the changes in enthalpy and volume. [Ans: 505 Btu/lb; 6.55 ft^3/lb]

2.37 For a compressor using R-134a with an evaporator temperature of 20°F and a condensing temperature of 80°F, calculate per ton of refrigeration

(a) displacement
(b) mass flow
(c) horsepower required

2.38 For a compressor using an R-22 system operating between 100°F condensing temperature and –10°F evaporator temperature, calculate per ton

(a) displacement
(b) mass flow
(c) horsepower required

2.39 An industrial plant has available a 4 cylinder, 3 in. bore by 4 in. stroke, 800 rpm, single-acting compressor for use with R-134a. Proposed operating conditions for the compressor are 100°F condensing temperature and 40°F evaporating temperature. It is estimated that the refrigerant will enter the expansion valve as a saturated liquid, that vapor will leave the evaporator at a temperature of 45°F, and that vapor will enter the compressor at a temperature of 55°F. Assume a compressor-volumetric efficiency of 70% and frictionless flow. Calculate the refrigerating capacity in tons for a system equipped with this compressor. Plot the cycle on the *p-h* diagram. [Ans: 12 tons]

2.40 A mechanical refrigeration system with R-134a is operating under such conditions that the evaporator pressure is 160 kPa and the liquid approaching the refrigerant control valve is at a temperature of 41°C. If the system has a capacity of 15 kW, determine

(a) the refrigerating effect per kilogram of refrigerant circulated
(b) the mass flow rate in kilograms per second per kilowatt
(c) the volume flow rate in liters per second per kilowatt at the compressor inlet
(d) the total mass flow rate in kilograms per second

(e) the total volume flow rate in liters per second at the compressor inlet

2.41 A vapor-compression R-22 refrigeration system is being designed to provide 50 kW of cooling when operating between evaporating and condensing temperatures of 0°C and 34°C, respectively. The refrigerant leaving the condenser is subcooled 3 degrees and the vapor leaving the evaporator is superheated 5 degrees. Determine

(a) ideal compressor discharge temperature, °C
(b) refrigerant flow rate, kg/s
(c) compressor motor size, kW
(d) COP for cooling
(e) compressor discharge temperature if compression efficiency is 60%

2.42 For a line of ammonia compressors, the actual volumetric efficiency is given by

$$\eta_{va} = 94 - 6.1(p_d/p_s), \%$$

The compression efficiency is fairly constant at 82%. A compressor in this line has two cylinders, each having a 92 mm bore and a 74 mm stroke. The compressor has 4.5% clearance and operates at 28 r/s. The system is being selected for an air-conditioning unit and will therefore operate between an evaporating temperature of 0°C and a condensing temperature of 35°C. There is 5°C of subcooling in the condenser and 10°C of superheating in the evaporator. Sketch and label the system, including appropriate values for the thermodynamic properties, starting with state 1 at the compressor inlet. Determine

(a) refrigerant flow rate [Ans: 0.0676 kg/s]
(b) refrigerating capacity [Ans: 77.4 kW]
(c) compressor motor size [Ans: 14.4 kW]
(d) compressor discharge temperature [Ans: 110°C]
(e) COP_c [Ans: 5.4]

2.43 An ammonia refrigerating system is operating with a condensing temperature of 30°C and an evaporating temperature of –4°C. For the ideal standard vapor compression cycle, determine

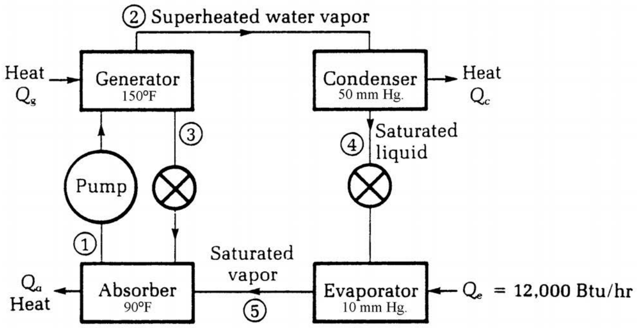
(a) refrigerating effect
(b) COP

Sketch and label a *p-h* diagram showing values.

2.44 A single-cylinder R-22 compressor has a 50 mm bore, a 40 mm stroke, and operates at 1725 rpm. Clearance volume is 4%. Determine as close as possible the actual refrigerating capacity, kW, and the required motor size, in hp, if the compressor is used in a system operating between 10°C and 40°C, evaporating and condensing temperatures, respectively.

2.45 For the lithium-bromide/water absorption refrigeration system shown below, determine

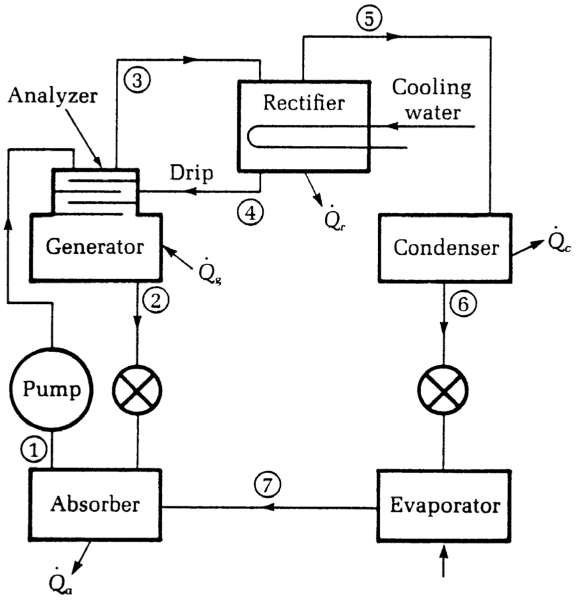
(a) heat required at the generator per ton of cooling [Ans: 516 Btu/min]



(b) COP [Ans: 0.39]
(c) heat rejection ratio $(Q_{\text{absorber}} + Q_{\text{condenser}})/Q_{\text{evaporator}}$ [Ans: 3.58]

2.46 In the basic lithium-bromide water absorption system, the generator operates at 170°F while the evaporator is at 47°F. The absorbing temperature is 75°F and the condensing temperature is 88°F. Calculate the heat rejection ratio for these conditions.

2.47 For the aqua-ammonia absorption refrigeration system shown in the sketch below, complete the table of properties.



Point	*p*, psia	*t*, °F	*x*, lb NH_3/lb mix	*h*, Btu/lb
1		80		
2				
3	200	260		
4				
5		160		
6				
7	25	20		

2.48 Solar energy is to be used to warm a large collector plate. This energy will, in turn, be transferred as heat to a fluid within a heat engine, and the engine will reject energy as heat to the atmosphere. Experiments indicate that about 200 Btu/h·ft^2 of energy can be collected when the plate is operating at 190°F. Estimate the minimum collector area that will be required for a plant producing 1 kW of useful shaft power, when the atmospheric temperature is 70°F. [Ans: 92.7 ft^2]

2.49 What are the Seebeck, Peltier, Thomson, Joule, and Fourier effects? Which are reversible and which are irreversible?

2.50 A 20 ft by 12 ft by 8 ft (6.1 m by 3.6 m by 2.4 m) room contains an air-water vapor mixture at 80°F (26.7°C). The barometric pressure is standard and the partial pressure of the water vapor is measured to be 0.2 psia (1.38 kPa). Calculate

(a) relative humidity
(b) humidity ratio
(c) dew-point temperature
(d) mass of water vapor contained in the room

2.51 Given room conditions of 75°F (23.9°C) dry bulb and 60% rh, determine for the air vapor mixture *without using* the psychrometric charts

(a) humidity ratio
(b) enthalpy
(c) dew-point temperature
(d) specific volume
(e) degree of saturation

2.52 For the conditions of *Problem 2.51* (above), *using* the ASHRAE Psychrometric Chart, find

(a) wet-bulb temperature [Ans: 65.2°F (18.4°C)]
(b) enthalpy [Ans: 30.2 Btu/lb$_m$ (70.2 J/g)]
(c) humidity ratio [Ans: 0.0112 lb/lb (0.0112 kg/kg)]

2.53 Using the ASHRAE Psychrometric Chart, complete the following table.

Dry Bulb, °F	Wet Bulb, °F	Dew Point, °F	Humidity W, lb/lb$_{air}$	Enthalpy h, Btu/lb$_{air}$	Relative Humidity ϕ, %	Specific Volume v, ft^3/lb$_{air}$
85	60					
75		50				
				30	60	
	70		0.01143			
		82		50		

2.54 Using the ASHRAE Psychrometric Chart complete the following table:

Dry Bulb, °F	Wet Bulb, °F	Dew Point, °F	Humidity Ratio, lb$_v$/lb$_a$	Relative Humidity, %	Enthalpy, Btu/lb$_{air}$	Specific Volume, ft^3/lb$_{air}$
80						13.8
70	55					
100		70				
				40	40	
			0.01			13.8
	60	40				
40				20		
		60			30	
85			0.012			
80	80					

2.55 Complete the following table using the Psychrometric Chart.

Dry Bulb, °C	Wet Bulb, °C	Dew Point, °C	Humidity Ratio, kg/kg	Relative Humidity, %	Enthalpy, kJ/kg	Specific Volume, m^3/kg
26.5						0.86
21	13					
38		21				
				40	95	
			0.01			0.85
	16	4				
4				20		
		16			70	
30			0.012			
27	27					

2.56 Complete the following table.

Dry Bulb, °C	Wet Bulb, °C	Dew Point, °C	Humidity Ratio, kg/kg	Relative Humidity, %	Enthalpy, kJ/kg	Specific Volume, m^3/kg
32	24					
40					81	
		18		30		
			0.022			0.9
7	7					

2.57 Without using the psychrometric chart, determine the humidity ratio and relative humidity of an air-water vapor mixture with a dry-bulb temperature of 90°F and thermodynamic wet-bulb temperature of 78°F. The barometric pressure is 14.7 psia. Check your result using the psychrometric chart. [Ans: $W = 0.018$ lb/lb, relative humidity $\phi = 59\%$]

2.12 Bibliography

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SI Tables and Figures

Table 2-5 SI Standard Atmospheric Data with Altitude

(Table 1, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Altitude, m	Temperature, °C	Pressure, kPa
–500	18.2	107.478
0	15.0	101.325
500	11.8	95.461
1000	8.5	89.875
1500	5.2	84.556
2000	2.0	79.495
2500	–1.2	74.682
3000	–4.5	70.108
4000	–11.0	61.640
5000	–17.5	54.020
6000	–24.0	47.181
7000	–30.5	41.061
8000	–37.0	35.600
9000	–43.5	30.742
10000	–50	26.436
12000	–63	19.284
14000	–76	13.786
16000	–89	9.632
18000	–102	6.556
20000	–115	4.328

Data adapted from NASA (1976).

Table 2-1 SI Thermodynamic Properties of Water

(Table 3, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °C t	Absolute Pressure, kPa p	Specific Volume, m^3/kg (water)			Specific Enthalpy, kJ/kg (water)			Specific Entropy, kJ/(kg·K) (water)			Temp., °C t
		Sat. Solid v_i	Evap. v_{ig}	Sat. Vapor v_g	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
–60	0.00108	0.001082	90942.00	90942.00	–446.40	2836.27	2389.87	–1.6854	13.3065	11.6211	–60
–59	0.00124	0.001082	79858.69	79858.69	–444.74	2836.46	2391.72	–1.7667	13.2452	11.5677	–59
–58	0.00141	0.001082	70212.37	70212.37	–443.06	2836.64	2393.57	–1.6698	13.8145	11.5147	–58
–57	0.00161	0.001082	61805.35	61805.35	–441.38	2836.81	2395.43	–1.6620	13.1243	11.4623	–57
–56	0.00184	0.001082	54469.39	54469.39	–439.69	2836.97	2397.28	–1.6542	13.0646	11.4104	–56
–55	0.00209	0.001082	48061.05	48061.05	–438.00	2837.13	2399.12	–1.6464	13.0054	11.3590	–55
–54	0.00238	0.001082	42455.57	42455.57	–436.29	2837.27	2400.98	–1.6386	12.9468	11.3082	–54
–53	0.00271	0.001083	37546.09	37546.09	–434.59	2837.42	2402.83	–1.6308	12.8886	11.2578	–53
–52	0.00307	0.001083	33242.14	33242.14	–432.87	2837.55	2404.68	–1.6230	12.8309	11.2079	–52
–51	0.00348	0.001083	29464.67	29464.67	–431.14	2837.68	2406.53	–1.6153	12.7738	11.1585	–51
–50	0.00394	0.001083	26145.01	26145.01	–429.41	2837.80	2408.39	–1.6075	12.7170	11.1096	–50
–49	0.00445	0.001083	23223.69	23223.70	–427.67	2837.91	2410.24	–1.5997	12.6608	11.0611	–49
–48	0.00503	0.001083	20651.68	20651.69	–425.93	2838.02	2412.09	–1.5919	12.6051	11.0131	–48
–47	0.00568	0.001083	18383.50	18383.51	–424.27	2838.12	2413.94	–1.5842	12.5498	10.9656	–47
–46	0.00640	0.001083	16381.35	16381.36	–422.41	2838.21	2415.79	–1.5764	12.4949	10.9185	–46
–45	0.00721	0.001984	14612.35	14512.36	–420.65	2838.29	2417.65	–1.5686	12.4405	10.8719	–45
–44	0.00811	0.001084	13047.65	13047.66	–418.87	2838.37	2419.50	–1.5609	12.3866	10.8257	–44
–43	0.00911	0.001084	11661.85	11661.85	–417.09	2838.44	2421.35	–1.5531	12.3330	10.7799	–43
–42	0.01022	0.001084	10433.85	10433.85	–415.30	2838.50	2423.20	–1.5453	12.2799	10.7346	–42
–41	0.01147	0.001084	9344.25	9344.25	–413.50	2838.55	2425.05	–1.5376	12.2273	10.6897	–41
–40	0.01285	0.001084	8376.33	8376.33	–411.70	2838.60	2426.90	–1.5298	12.1750	10.6452	–40
–39	0.01438	0.001085	7515.86	7515.87	–409.88	2838.64	2428.76	–1.5221	12.1232	10.6011	–39
–38	0.01608	0.001085	6750.36	6750.36	–508.07	2838.67	1430.61	–1.5143	12.0718	10.5575	–38
–37	0.01796	0.001085	6068.16	6068.17	–406.24	2838.70	2432.46	–1.5066	12.0208	10.5142	–37
–36	0.02004	0.001085	5459.82	5459.82	–404.40	2838.71	2434.31	–1.4988	11.9702	10.4713	–36
–35	0.02235	0.001085	4917.09	4917.10	–402.56	2838.73	2436.16	–1.4911	11.9199	10.4289	–35
–34	0.02490	0.001085	4432.36	4432.37	–400.72	2838.73	2438.01	–1.4833	11.8701	10.3868	–34
–33	0.02771	0.001085	3998.71	3998.71	–398.86	2838.72	2439.86	–1.4756	11.8207	10.3451	–33
–32	0.03082	0.001086	3610.71	3610.71	–397.00	2838.71	2441.72	–1.4678	11.7716	10.3037	–32
–31	0.03424	0.001086	3263.20	3263.20	–395.12	2838.69	2443.57	–1.4601	11.7229	10.2628	–31
–30	0.03802	0.001086	2951.64	2951.64	–393.25	2838.66	2445.42	–1.4524	11.6746	10.2222	–30
–29	0.04217	0.001086	2672.03	2672.03	–391.36	2838.63	2447.27	–1.4446	11.6266	10.1820	–29
–28	0.04673	0.001086	2420.89	2420.89	–389.47	2838.59	2449.12	–1.4369	11.4790	10.1421	–28
–27	0.05174	0.001086	2195.23	2195.23	–387.57	2838.53	2450.97	–1.4291	11.5318	10.1026	–27
–26	0.05725	0.001087	1992.15	1992.15	–385.66	2838.48	2452.82	–1.4214	11.4849	10.0634	–26
–25	0.06329	0.001087	1809.35	1809.35	–383.74	2838.41	2454.67	–1.4137	11.4383	10.0246	–25
–24	0.06991	0.001087	1644.59	1644.59	–381.34	2838.34	2456.52	–1.4059	11.3921	9.9862	–24
–23	0.07716	0.001087	1495.98	1495.98	–379.89	2838.26	2458.37	–1.3982	11.3462	9.9480	–23
–22	0.08510	0.001087	1361.94	1361.94	–377.95	2838.17	2460.22	–1.3905	11.3007	9.9102	–22
–21	0.09378	0.001087	1240.77	1240.77	–376.01	2838.07	2462.06	–1.3828	11.2555	9.8728	–21
–20	0.10326	0.001087	1131.27	1131.27	–374.06	2837.97	2463.91	–1.3750	11.2106	9.8356	–20
–19	0.11362	0.001088	1032.18	1032.18	–372.10	2837.86	2465.76	–1.3673	11.1661	9.7988	–19
–18	0.12492	0.001088	942.46	942.47	–370.13	2837.74	2467.61	–1.3596	11.1218	9.7623	–18
–17	0.13725	0.001088	861.17	861.18	–368.15	2837.61	2469.46	–1.3518	11.0779	9.7261	–17
–16	0.15068	0.001088	787.48	787.49	–366.17	2837.47	2471.30	–1.3441	11.0343	9.6902	–16
–15	0.16530	0.001088	720.59	720.59	–364.18	2837.33	2473.15	–1.3364	10.9910	9.6546	–15
–14	0.18122	0.001088	659.86	659.86	–362.18	2837.18	2474.99	–1.3287	10.9480	9.6193	–14
–13	0.19852	0.001089	604.65	604.65	–360.18	2837.02	2476.84	–1.3210	10.9053	9.5844	–13
–12	0.21732	0.001089	554.45	554.45	–358.17	2836.85	2478.68	–1.3232	10.8629	9.5497	–12
–11	0.23774	0.001089	508.75	508.75	–356.15	2836.68	2480.53	–1.3055	10.8208	9.5153	–11
–10	0.25990	0.001089	467.14	467.14	–354.12	2836.49	2482.37	–1.2978	10.7790	9.4812	–10
–9	0.28393	0.001089	429.21	429.21	–352.08	2836.30	2484.22	–1.2901	10.7375	9.4474	–9
–8	0.30998	0.001090	394.64	394.64	–350.04	2836.10	2486.06	–1.2824	10.6962	9.4139	–8
–7	0.33819	0.001090	363.07	363.07	–347.99	2835.89	2487.90	–1.2746	10.6552	9.3806	–7
–6	0.36874	0.001090	334.25	334.25	–345.93	2835.68	2489.74	–1.2669	10.6145	9.3476	–6
–5	0.40176	0.001090	307.91	307.91	–343.87	2835.45	2491.58	–2.2592	10.4741	9.3149	–5
–4	0.43747	0.001090	283.83	283.83	–341.80	2835.22	2493.42	–1.2515	10.5340	9.2825	–4
–3	0.47606	0.001090	261.79	261.79	–339.72	2834.98	2495.26	–1.2438	10.4941	9.2503	–3
–2	0.51772	0.001091	241.60	241.60	–337.63	2834.72	2497.10	–1.2361	10.4544	9.2184	–2
–1	0.56267	0.001091	223.11	223.11	–335.53	2834.47	2498.93	–1.2284	10.4151	9.1867	–1
0	0.61115	0.001091	206.16	206.16	–333.43	2834.20	2500.77	–1.2206	10.3760	9.1553	0

Table 2-1 SI Thermodynamic Properties of Water (*Continued*)

(Table 3, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °C	Absolute Pressure, kPa	Specific Volume, m^3/kg (water)			Specific Enthalpy, kJ/kg (water)			Specific Entropy, kJ/(kg·K) (water)			Temp., °C
		Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
t	p	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	t
0	0.6112	0.001000	206.141	206.143	–0.04	2500.81	2500.77	–0.0002	9.1555	9.1553	0
1	0.6571	0.001000	192.455	192.456	4.18	2498.43	2502.61	0.0153	9.1134	9.1286	1
2	0.7060	0.001000	179.769	179.770	8.39	2496.05	2504.45	0.0306	9.0716	9.1022	2
3	0.7580	0.001000	168.026	168.027	12.60	2493.68	2506.28	0.0459	9.0302	9.0761	3
4	0.8135	0.001000	157.137	157.138	16.81	2491.31	2508.12	0.0611	8.9890	9.0501	4
5	0.8725	0.001000	147.032	147.033	21.02	2488.94	2509.96	0.0763	8.9482	9.0244	5
6	0.9353	0.001000	137.653	137.654	25.22	2486.57	2511.79	0.0913	8.9077	8.9990	6
7	1.0020	0.001000	128.947	128.948	29.42	2484.20	2513.62	0.1064	8.8674	8.9738	7
8	1.0728	0.001000	120.850	120.851	33.62	2481.84	2515.46	0.1213	8.8273	8.9488	8
9	1.1481	0.001000	113.326	113.327	37.82	2479.47	2517.29	0.1362	8.7878	8.9245	9
10	1.2280	0.001000	106.328	106.329	42.01	2477.11	2519.12	0.1511	8.7484	8.8995	10
11	1.3127	0.001000	99.812	99.813	46.21	2474.74	2520.95	0.1659	8.7093	8.8752	11
12	1.4026	0.001001	93.743	93.744	50.40	2472.38	2522.78	0.1806	8.6705	8.8511	12
13	1.4978	0.001001	88.088	88.089	54.59	2470.02	2524.61	0.1953	8.6319	8.8272	13
14	1.5987	0.001001	82.815	82.816	58.78	2467.66	2526.44	0.2099	8.5936	3.8035	14
15	1.7055	0.001001	77.897	77.898	62.97	2465.30	2528.26	0.2244	8.5556	8.7801	15
16	1.8184	0.001001	73.307	73.308	67.16	2462.93	2530.09	0.2389	8.5178	8.7568	16
17	1.9380	0.001001	69.021	69.022	71.34	2460.57	2531.92	0.2534	8.4804	8.7338	17
18	2.0643	0.001002	65.017	65.018	75.53	2458.21	2533.74	0.2678	8.4431	8.7109	18
19	2.1978	0.001002	65.274	61.273	79.72	2455.85	2535.56	0.2821	8.4061	8.6883	19
20	2.3388	0.001002	57.774	57.773	83.90	2453.48	2537.38	0.2964	8.3694	8.6658	20
21	2.4877	0.001002	54.450	54.500	88.08	2451.12	2539.20	0.3107	8.3329	8.6436	21
22	2.6448	0.001002	51.433	51.434	92.27	2448.75	2541.02	0.3249	8.2967	8.6215	22
23	2.8104	0.001003	48.562	48.563	96.45	2446.39	2542.84	0.3390	8.2607	8.5996	23
24	2.9851	0.001003	45.872	45.873	100.63	2444.02	2544.65	0.3531	8.2249	8.5780	24
25	3.1692	0.001003	43.350	43.351	104.81	2441.66	2546.47	0.3672	8.1894	8.5565	25
26	3.3631	0.001003	40.985	40.986	108.99	2439.29	2548.28	0.3812	8.1541	8.5352	26
27	3.5673	0.001004	38.766	38.767	113.18	2436.92	2550.09	0.3951	8.1190	8.5141	27
28	3.7822	0.001004	36.682	36.683	117.36	2434.55	2551.90	0.4090	8.0842	8.4932	28
29	4.0083	0.001004	34.726	34.727	121.54	2432.17	2553.71	0.4229	8.0496	8.4724	29
30	4.2460	0.001004	32.889	32.889	125.72	2429.80	2555.52	0.4367	8.0152	8.4519	30
31	4.4959	0.001005	31.160	31.161	129.90	2427.43	2557.32	0.4505	7.9810	8.4315	31
32	4.7585	0.001005	29.535	29.536	134.08	2425.05	2559.13	0.4642	7.9471	8.4112	32
33	5.0343	0.001005	28.006	28.007	138.26	2422.67	2560.93	0.4779	7.9133	8.3912	33
34	5.3239	0.001006	26.567	26.568	142.44	2410.29	2562.73	0.4915	7.8790	8.3713	34
35	5.6278	0.001006	25.212	25.213	146.62	2417.91	2564.53	0.5051	7.8465	8.3516	35
36	5.9466	0.001006	23.935	23.936	150.80	2415.53	2566.33	0.5186	7.8134	8.3320	36
37	6.2810	0.001007	22.733	22.734	154.98	2413.14	2568.12	0.5321	7.7805	8.3127	37
38	6.6315	0.001007	21.599	21.600	159.16	2410.76	2569.91	0.5456	7.7479	8.2934	38
39	6.9987	0.001008	20.529	20.530	163.34	2408.37	2571.71	0.5590	7.7154	8.2744	39
40	7.3835	0.001008	19.520	19.521	167.52	2405.98	2573.50	0.5724	7.6831	8.2555	40
41	7.7863	0.001008	18.567	18.568	171.70	2403.58	2575.28	0.5857	7.6510	8.2367	41
42	8.2080	0.001009	17.667	17.668	175.88	2401.19	2577.07	0.5990	7.6191	8.2181	42
43	8.6492	0.001009	16.818	16.819	180.06	2398.79	2578.85	0.6122	7.5875	8.1997	43
44	9.1107	0.001010	16.014	16.015	184.24	2396.39	2580.63	0.6254	7.3560	8.1814	44
45	9.5932	0.001010	15.255	15.256	188.42	2393.99	2582.41	0.6386	7.5247	8.1632	45
46	10.0976	0.001010	14.537	14.538	192.60	2391.59	2584.19	0.6517	7.4936	8.1452	46
47	10.6246	0.001011	13.858	13.859	196.78	2389.18	2585.96	0.6648	7.4626	8.1274	47
48	11.1751	0.001011	13.214	13.215	200.97	2386.77	2587.74	0.6778	7.4319	8.1097	48
49	11.7500	0.001012	12.606	12.607	205.15	2384.36	2589.51	0.6908	7.4013	8.0921	49
50	12.3499	0.001012	12.029	12.029	209.33	2381.94	2591.27	0.7038	7.3709	8.0747	50
51	12.9759	0.001013	11.482	11.483	213.51	2379.53	2593.04	0.7167	7.3407	8.0574	51
52	13.6290	0.001013	10.964	10.965	217.70	2377.10	2594.80	0.7296	7.3107	8.0403	52
53	14.3100	0.001014	10.473	10.474	221.88	2374.68	2596.56	0.7424	7.2809	8.0233	53
54	15.0200	0.001014	10.001	10.008	226.06	2372.26	2598.32	0.7552	7.2512	8.0064	54
55	15.7597	0.001015	9.563	9.5663	230.25	2369.83	2600.07	0.7680	7.2217	7.9897	55
56	16.5304	0.001015	9.147	9.1468	234.43	2367.39	2601.82	0.7807	7.1924	7.9731	56
57	17.3331	0.001016	8.744	8.7489	238.61	2364.96	2603.57	0.7934	7.1632	7.9566	57
58	18.1690	0.001016	8.3690	8.3700	242.80	2362.52	2605.32	0.8061	7.1342	7.9403	58
59	19.0387	0.001017	8.0094	8.0114	246.99	2360.08	2607.06	0.8187	7.1054	7.9240	59
60	19.944	0.001017	7.6677	7.6697	251.17	2357.63	2608.80	0.8313	7.0767	7.9079	60
61	20.885	0.001018	7.3428	7.3438	255.36	2355.19	2610.54	0.8438	7.0482	7.8920	61
62	21.864	0.001018	7.0337	7.0347	259.54	2352.73	2612.28	0.8563	7.0198	7.8761	62
63	22.882	0.001019	6.7397	6.7407	263.73	2350.28	2614.01	0.8688	6.9916	7.8604	63
64	23.940	0.001019	6.4599	6.4609	267.92	2347.82	2615.74	0.8812	6.9636	7.8448	64
65	25.040	0.001020	6.1935	6.1946	272.11	2345.36	2617.46	0.8936	6.9357	7.8293	65
66	26.180	0.001020	5.9397	5.9409	276.30	2342.89	2619.19	0.9060	6.9080	7.8140	66
67	27.366	0.001021	5.6982	5.6992	280.49	2340.42	2620.90	0.9183	6.8804	7.7987	67
68	28.596	0.001022	5.4680	5.4690	284.68	2337.95	2622.62	0.9306	2.8530	7.7836	68
69	29.873	0.001022	5.2485	5.2495	288.87	2335.47	2624.33	0.9429	6.8257	7.7686	69

Table 2-1 SI Thermodynamic Properties of Water (*Continued*)

(Table 3, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °C t	Absolute Pressure, kPa p	Specific Volume, m^3/kg (water) Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Specific Enthalpy, kJ/kg (water) Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Specific Entropy, kJ/(kg·K) (water) Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	Temp., °C t
70	31.198	0.001023	5.0392	5.0402	293.06	2332.99	2626.04	0.9551	6.7986	7.7537	70
71	32.572	0.001023	4.8396	4.8407	297.25	2330.50	2627.75	0.9673	6.7716	7.7389	71
72	33.997	0.001024	4.6492	4.6502	301.44	2328.01	2629.45	0.9795	6.7448	7.7242	72
73	35.475	0.001025	4.4675	4.4685	305.63	2325.51	2631.15	0.9916	6.7181	7.7097	73
74	37.006	0.001025	4.2940	4.2951	309.83	2323.02	2632.84	1.0037	6.6915	7.6952	74
75	38.592	0.001026	4.1284	4.1294	314.02	2320.51	2634.53	1.0157	6.6651	7.6809	75
76	40.236	0.001026	3.9702	3.9712	318.22	2318.01	2636.22	1.0278	6.6389	7.6666	76
77	41.938	0.001027	3.8190	3.8201	322.41	2315.49	2637.90	1.0398	6.6127	7.6525	77
78	43.700	0.001028	3.6746	3.6756	326.61	2312.98	2639.58	1.0517	6.5867	7.6384	78
79	45.524	0.001028	3.5365	3.5375	330.81	2310.46	2641.26	1.0636	6.5609	7.6245	79
80	47.412	0.001029	3.4044	3.4055	335.00	2307.93	2642.93	1.0755	6.5351	7.6107	80
81	49.364	0.001030	3.2781	3.2792	339.20	2305.40	2644.60	1.0874	6.5095	7.5969	81
82	51.384	0.001030	3.1573	3.1583	343.40	2902.86	2646.26	1.0993	6.4841	7.5833	82
83	53.473	0.001031	3.0417	3.0427	347.60	2300.32	2647.92	1.1111	6.4587	7.5698	83
84	55.633	0.001032	2.9310	2.9320	351.80	2297.78	2649.58	1.1228	6.4335	7.5563	84
85	57.865	0.001032	2.8250	2.8260	356.01	2295.22	2651.23	1.1346	6.4084	7.5430	85
86	60.171	0.001033	2.7235	2.7245	350.21	2292.67	2652.88	1.1463	6.3834	7.5297	86
87	62.554	0.001034	2.6263	2.6273	364.41	2290.11	2654.52	1.1580	6.3586	7.5166	87
88	65.015	0.001035	2.5331	2.5341	368.62	2287.54	2656.16	1.1696	6.3339	7.5035	88
89	67.556	0.001035	2.4438	2.4448	372.82	2284.97	2657.79	1.1812	6.3093	7.4905	89
90	70.180	0.001036	2.3582	2.3592	377.03	2282.39	2659.42	1.1928	6.2848	7.4776	90
91	72.888	0.001037	2.2760	2.2771	381.24	2279.81	2661.04	1.2044	6.2605	7.4648	91
92	75.683	0.001037	2.1973	2.1983	385.45	2277.22	2662.66	1.2159	6.2362	7.4521	92
93	78.566	0.001038	2.1217	2.1228	389.66	2274.62	2664.28	1.2274	6.2121	7.4395	93
94	81.541	0.001039	2.0492	2.0502	393.87	2272.02	2665.89	1.2389	6.1881	7.4270	94
95	84.608	0.001040	1.9796	1.9806	398.08	2269.41	2667.49	1.2504	6.1642	7.4146	95
96	87.770	0.001040	1.9128	1.9138	402.29	2266.80	2669.09	1.2618	6.1404	7.4022	96
97	91.030	0.001041	1.8486	1.8496	406.51	2264.18	2670.69	1.2732	6.1168	7.3899	97
98	94.390	0.001042	1.7869	1.7880	410.72	2261.55	2672.28	1.2845	6.0932	7.3777	98
99	97.852	0.001044	1.7277	1.7287	414.94	2258.92	2673.86	1.2959	6.0697	7.3656	99
100	101.419	0.001044	1.6708	1.6718	419.16	2256.28	2675.44	1.3072	6.0464	7.3536	100
101	105.092	0.001044	1.6161	1.6171	423.38	2253.64	2677.02	1.3185	6.0232	7.3416	101
102	108.875	0.001045	1.5635	1.5645	427.60	2250.99	2678.58	1.3297	6.0000	7.3298	102
103	112.770	0.001046	1.5129	1.5139	431.82	2248.33	2680.15	1.3410	5.9770	7.3180	103
104	116.779	0.001047	1.4642	1.4652	436.04	2245.66	2681.71	1.3522	5.9541	7.3062	104
105	120.906	0.001047	1.4174	1.4184	440.27	2242.99	2683.26	1.3634	5.9313	7.2946	105
106	125.152	0.001048	1.3723	1.3734	444.49	2240.31	2684.80	1.3745	5.9086	7.2830	106
107	129.520	0.001049	1.3290	1.3300	448.72	2237.63	2686.35	1.3856	5.8860	7.2716	107
108	134.012	0.001050	1.2872	1.2883	452.95	2234.93	2687.88	1.3967	5.8635	7.2601	108
109	138.633	0.001051	1.2470	1.2481	457.18	2232.23	2689.41	1.4078	5.8410	7.2488	109
110	143.384	0.001052	1.2083	1.2093	461.41	2229.52	2690.93	1.4188	5.8187	7.2375	110
111	148.267	0.001052	1.1710	1.1720	465.64	2226.81	2692.45	1.4298	5.7965	7.2263	111
112	153.287	0.001053	1.1350	1.1361	469.88	2224.09	2693.96	1.4408	5.7744	7.2152	112
113	158.445	0.001054	1.1004	1.1015	474.11	2221.35	2695.47	1.4518	5.7524	7.2402	113
114	163.745	0.001055	1.0670	1.0681	478.35	2218.62	2696.97	1.4627	5.7304	7.1931	114
115	169.190	0.001056	1.0348	1.0359	482.59	2215.87	2698.46	1.4737	5.7086	7.1822	115
116	174.782	0.001057	1.0038	1.0048	486.83	2213.12	2699.95	1.4846	5.6868	7.1714	116
117	180.525	0.001058	0.9739	0.9749	491.07	2210.35	2701.43	1.4954	5.6652	7.1606	117
118	186.420	0.001059	0.9450	0.9460	495.32	2207.58	2702.90	1.5063	5.6436	7.1499	118
119	192.473	0.001059	0.9171	0.9182	499.56	2204.80	2704.37	1.5171	5.6221	7.1392	119
120	198.685	0.001060	0.8902	0.8913	503.81	2202.02	2705.83	1.5279	5.6007	7.1286	120
122	211.601	0.001062	0.8391	0.8402	512.31	2196.42	2706.73	1.5494	5.5582	7.1076	122
124	225.194	0.001064	0.7916	0.7927	520.82	2190.78	2711.60	1.5709	5.5160	7.0869	124
126	239.490	0.001066	0.7472	0.7483	529.33	2185.11	2714.44	1.5922	5.4742	7.0664	126
128	254.515	0.001068	0.7057	0.7068	537.86	2179.40	2717.26	1.6135	5.4326	7.0461	128
130	270.298	0.001070	0.6670	0.6681	546.39	2173.66	2720.05	1.6347	5.3914	7.0261	130
132	286.866	0.001072	0.6308	0.6318	554.93	2167.87	2722.80	1.6557	5.3505	7.0063	132
134	304.247	0.001074	0.5969	0.5979	563.48	2162.05	2725.53	1.6767	5.3099	6.9867	134
136	322.470	0.001076	0.5651	0.5662	572.04	2156.18	2728.22	1.6977	5.2697	6.9673	136
138	341.566	0.001078	0.5354	0.5364	580.60	2150.28	2730.88	1.7185	5.2296	6.9481	138
140	361.565	0.001080	0.5075	0.5085	589.18	2144.33	2733.51	1.7393	5.1899	6.9292	140
142	382.497	0.001082	0.4813	0.4824	597.76	2138.34	2736.11	1.7599	5.1505	6.9104	142
144	404.394	0.001084	0.4567	0.4578	606.36	2132.31	2738.67	1.7805	5.1113	6.8918	144
146	427.288	0.001086	0.4336	0.4347	614.97	2126.23	2741.19	1.8011	5.0724	6.8735	146
148	451.211	0.001088	0.4119	0.4130	623.58	2120.10	2743.68	1.8215	5.0338	6.8553	148
150	476.198	0.001091	0.3914	0.3925	632.21	2113.92	2746.13	1.8419	4.9954	6.8373	150
152	502.281	0.001093	0.3722	0.3733	640.85	2107.70	2748.55	1.8622	4.9573	6.8194	152
154	529.495	0.001095	0.3541	0.3552	649.50	2101.43	2750.93	1.8824	4.9194	6.8017	154
156	557.875	0.001097	0.3370	0.3381	658.16	2095.11	2753.27	1.9026	4.8817	6.7842	156
158	587.456	0.001100	0.3209	0.3220	666.83	2088.73	2755.57	1.9226	4.8443	6.7669	158
160	618.275	0.001102	0.3058	0.3069	675.52	2082.31	2757.82	1.9427	4.8070	6.7497	160

Table 2-2 SI Refrigerant 134a Properties of Saturated Liquid and Saturated Vapor

(Table Refrigerant 134a, Chapter 30, 2017 ASHRAE Handbook—Fundamentals)

Temp.,* °C	Pressure, MPa	Density, kg/m^3	Volume, m^3/kg	Enthalpy, kJ/kg		Entropy, kJ/(kg·K)		Specific Heat c_p, kJ/(kg·K)		c_p/c_v	Velocity of Sound, m/s		Viscosity, μPa·s		Thermal Cond., mW/(m·K)		Surface Tension, mN/m	Temp., °C
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor		
–103.30a	0.00039	1591.1	35.496	71.46	334.94	0.4126	1.9639	1.184	0.585	1.164	1120.	126.8	2175.	6.46	145.2	3.08	28.07	–103.30
–100.00	0.00056	1582.4	25.193	75.36	336.85	0.4354	1.9456	1.184	0.593	1.162	1103.	127.9	1893.	6.60	143.2	3.34	27.50	–100.00
–90.00	0.00152	1555.8	9.7698	87.23	342.76	0.5020	1.8972	1.189	0.617	1.156	1052.	131.0	1339.	7.03	137.3	4.15	25.79	–90.00
–80.00	0.00367	1529.0	4.2682	99.16	348.83	0.5654	1.8580	1.198	0.642	1.151	1002.	134.0	1018.	7.46	131.5	4.95	24.10	–80.00
–70.00	0.00798	1501.9	2.0590	111.20	355.02	0.6262	1.8264	1.210	0.667	1.148	952.	136.8	809.2	7.89	126.0	5.75	22.44	–70.00
–60.00	0.01591	1474.3	1.0790	123.36	361.31	0.6846	1.8010	1.223	0.692	1.146	903.	139.4	663.1	8.30	120.7	6.56	20.80	–60.00
–50.00	0.02945	1446.3	0.60620	135.67	367.65	0.7410	1.7806	1.238	0.720	1.146	855.	141.7	555.1	8.72	115.6	7.36	19.18	–50.00
–40.00	0.05121	1417.7	0.36108	148.14	374.00	0.7956	1.7643	1.255	0.749	1.148	807.	143.6	472.2	9.12	110.6	8.17	17.60	–40.00
–30.00	0.08438	1388.4	0.22594	160.79	380.32	0.8486	1.7515	1.273	0.781	1.152	760.	145.2	406.4	9.52	105.8	8.99	16.04	–30.00
–28.00	0.09270	1382.4	0.20680	163.34	381.57	0.8591	1.7492	1.277	0.788	1.153	751.	145.4	394.9	9.60	104.8	9.15	15.73	–28.00
–26.07b	0.10133	1376.7	0.19018	165.81	382.78	0.8690	1.7472	1.281	0.794	1.154	742.	145.7	384.2	9.68	103.9	9.31	15.44	–26.07
–26.00	0.10167	1376.5	0.18958	165.90	382.82	0.8694	1.7471	1.281	0.794	1.154	742.	145.7	383.8	9.68	103.9	9.32	15.43	–26.00
–24.00	0.11130	1370.4	0.17407	168.47	384.07	0.8798	1.7451	1.285	0.801	1.155	732.	145.9	373.1	9.77	102.9	9.48	15.12	–24.00
–22.00	0.12165	1364.4	0.16006	171.05	385.32	0.8900	1.7432	1.289	0.809	1.156	723.	146.1	362.9	9.85	102.0	9.65	14.82	–22.00
–20.00	0.13273	1358.3	0.14739	173.64	386.55	0.9002	1.7413	1.293	0.816	1.158	714.	146.3	353.0	9.92	101.1	9.82	14.51	–20.00
–18.00	0.14460	1352.1	0.13592	176.23	387.79	0.9104	1.7396	1.297	0.823	1.159	705.	146.4	343.5	10.01	100.1	9.98	14.21	–18.00
–16.00	0.15728	1345.9	0.12551	178.83	389.02	0.9205	1.7379	1.302	0.831	1.161	695.	146.6	334.3	10.09	99.2	10.15	13.91	–16.00
–14.00	0.17082	1339.7	0.11605	181.44	390.24	0.9306	1.7363	1.306	0.838	1.163	686.	146.7	325.4	10.17	98.3	10.32	13.61	–14.00
–12.00	0.18524	1333.4	0.10744	184.07	391.46	0.9407	1.7348	1.311	0.846	1.165	677.	146.8	316.9	10.25	97.4	10.49	13.32	–12.00
–10.00	0.20060	1327.1	0.09959	186.70	392.66	0.9506	1.7334	1.316	0.854	1.167	668.	146.9	308.6	10.33	96.5	10.66	13.02	–10.00
–8.00	0.21693	1320.8	0.09242	189.34	393.87	0.9606	1.7320	1.320	0.863	1.169	658.	146.9	300.6	10.41	95.6	10.83	12.72	–8.00
–6.00	0.23428	1314.3	0.08587	191.99	395.06	0.9705	1.7307	1.325	0.871	1.171	649.	147.0	292.9	10.49	94.7	11.00	12.43	–6.00
–4.00	0.25268	1307.9	0.07987	194.65	396.25	0.9804	1.7294	1.330	0.880	1.174	640.	147.0	285.4	10.57	93.8	11.17	12.14	–4.00
–2.00	0.27217	1301.4	0.07436	197.32	397.43	0.9902	1.7282	1.336	0.888	1.176	631.	147.0	278.1	10.65	92.9	11.34	11.85	–2.00
0.00	0.29280	1294.8	0.06931	200.00	398.60	1.0000	1.7271	1.341	0.897	1.179	622.	146.9	271.1	10.73	92.0	11.51	11.56	0.00
2.00	0.31462	1288.1	0.06466	202.69	399.77	1.0098	1.7260	1.347	0.906	1.182	612.	146.9	264.3	10.81	91.1	11.69	11.27	2.00
4.00	0.33766	1281.4	0.06039	205.40	400.92	1.0195	1.7250	1.352	0.916	1.185	603.	146.8	257.6	10.90	90.2	11.86	10.99	4.00
6.00	0.36198	1274.7	0.05644	208.11	402.06	1.0292	1.7240	1.358	0.925	1.189	594.	146.7	251.2	10.98	89.4	12.04	10.70	6.00
8.00	0.38761	1267.9	0.05280	210.84	403.20	1.0388	1.7230	1.364	0.935	1.192	585.	146.5	244.9	11.06	88.5	12.22	10.42	8.00
10.00	0.41461	1261.0	0.04944	213.58	404.32	1.0485	1.7221	1.370	0.945	1.196	576.	146.4	238.8	11.15	87.6	12.40	10.14	10.00
12.00	0.44301	1254.0	0.04633	216.33	405.43	1.0581	1.7212	1.377	0.956	1.200	566.	146.2	232.9	11.23	86.7	12.58	9.86	12.00
14.00	0.47288	1246.9	0.04345	219.09	406.53	1.0677	1.7204	1.383	0.967	1.204	557.	146.0	227.1	11.32	85.9	12.77	9.58	14.00
16.00	0.50425	1239.8	0.04078	221.87	407.61	1.0772	1.7196	1.390	0.978	1.209	548.	145.7	221.5	11.40	85.0	12.95	9.30	16.00
18.00	0.53718	1232.6	0.03830	224.66	408.69	1.0867	1.7188	1.397	0.989	1.214	539.	145.5	216.0	11.49	84.1	13.14	9.03	18.00
20.00	0.57171	1225.3	0.03600	227.47	409.75	1.0962	1.7180	1.405	1.001	1.219	530.	145.1	210.7	11.58	83.3	13.33	8.76	20.00
22.00	0.60789	1218.0	0.03385	230.29	410.79	1.1057	1.7173	1.413	1.013	1.224	520.	144.8	205.5	11.67	82.4	13.53	8.48	22.00
24.00	0.64578	1210.5	0.03186	233.12	411.82	1.1152	1.7166	1.421	1.025	1.230	511.	144.5	200.4	11.76	81.6	13.72	8.21	24.00
26.00	0.68543	1202.9	0.03000	235.97	412.84	1.1246	1.7159	1.429	1.038	1.236	502.	144.1	195.4	11.85	80.7	13.92	7.95	26.00
28.00	0.72688	1195.2	0.02826	238.84	413.84	1.1341	1.7152	1.437	1.052	1.243	493.	143.6	190.5	11.95	79.8	14.13	7.68	28.00
30.00	0.77020	1187.5	0.02664	241.72	414.82	1.1435	1.7145	1.446	1.065	1.249	483.	143.2	185.8	12.04	79.0	14.33	7.42	30.00
32.00	0.81543	1179.6	0.02513	244.62	415.78	1.1529	1.7138	1.456	1.080	1.257	474.	142.7	181.1	12.14	78.1	14.54	7.15	32.00
34.00	0.86263	1171.6	0.02371	247.54	416.72	1.1623	1.7131	1.466	1.095	1.265	465.	142.1	176.6	12.24	77.3	14.76	6.89	34.00
36.00	0.91185	1163.4	0.02238	250.48	417.65	1.1717	1.7124	1.476	1.111	1.273	455.	141.6	172.1	12.34	76.4	14.98	6.64	36.00
38.00	0.96315	1155.1	0.02113	253.43	418.55	1.1811	1.7118	1.487	1.127	1.282	446.	141.0	167.7	12.44	75.6	15.21	6.38	38.00
40.00	1.0166	1146.7	0.01997	256.41	419.43	1.1905	1.7111	1.498	1.145	1.292	436.	140.3	163.4	12.55	74.7	15.44	6.13	40.00
42.00	1.0722	1138.2	0.01887	259.41	420.28	1.1999	1.7103	1.510	1.163	1.303	427.	139.7	159.2	12.65	73.9	15.68	5.88	42.00
44.00	1.1301	1129.5	0.01784	262.43	421.11	1.2092	1.7096	1.523	1.182	1.314	418.	138.9	155.1	12.76	73.0	15.93	5.63	44.00
46.00	1.1903	1120.6	0.01687	265.47	421.92	1.2186	1.7089	1.537	1.202	1.326	408.	138.2	151.0	12.88	72.1	16.18	5.38	46.00
48.00	1.2529	1111.5	0.01595	268.53	422.69	1.2280	1.7081	1.551	1.223	1.339	399.	137.4	147.0	13.00	71.3	16.45	5.13	48.00
50.00	1.3179	1102.3	0.01509	271.62	423.44	1.2375	1.7072	1.566	1.246	1.354	389.	136.6	143.1	13.12	70.4	16.72	4.89	50.00
52.00	1.3854	1092.9	0.01428	274.74	424.15	1.2469	1.7064	1.582	1.270	1.369	379.	135.7	139.2	13.24	69.6	17.01	4.65	52.00
54.00	1.4555	1083.2	0.01351	277.89	424.83	1.2563	1.7055	1.600	1.296	1.386	370.	134.7	135.4	13.37	68.7	17.31	4.41	54.00
56.00	1.5282	1073.4	0.01278	281.06	425.47	1.2658	1.7045	1.618	1.324	1.405	360.	133.8	131.6	13.51	67.8	17.63	4.18	56.00
58.00	1.6036	1063.2	0.01209	284.27	426.07	1.2753	1.7035	1.638	1.354	1.425	350.	132.7	127.9	13.65	67.0	17.96	3.95	58.00
60.00	1.6818	1052.9	0.01144	287.50	426.63	1.2848	1.7024	1.660	1.387	1.448	340.	131.7	124.2	13.79	66.1	18.31	3.72	60.00
62.00	1.7628	1042.2	0.01083	290.78	427.14	1.2944	1.7013	1.684	1.422	1.473	331.	130.5	120.6	13.95	65.2	18.68	3.49	62.00
64.00	1.8467	1031.2	0.01024	294.09	427.61	1.3040	1.7000	1.710	1.461	1.501	321.	129.4	117.0	14.11	64.3	19.07	3.27	64.00
66.00	1.9337	1020.0	0.00969	297.44	428.02	1.3137	1.6987	1.738	1.504	1.532	311.	128.1	113.5	14.28	63.4	19.50	3.05	66.00
68.00	2.0237	1008.3	0.00916	300.84	428.36	1.3234	1.6972	1.769	1.552	1.567	301.	126.8	109.9	14.46	62.6	19.95	2.83	68.00
70.00	2.1168	996.2	0.00865	304.28	428.65	1.3332	1.6956	1.804	1.605	1.607	290.	125.5	106.4	14.65	61.7	20.45	2.61	70.00
72.00	2.2132	983.8	0.00817	307.78	428.86	1.3430	1.6939	1.843	1.665	1.653	280.	124.0	102.9	14.85	60.8	20.98	2.40	72.00
74.00	2.3130	970.8	0.00771	311.33	429.00	1.3530	1.6920	1.887	1.734	1.705	269.	122.6	99.5	15.07	59.9	21.56	2.20	74.00
76.00	2.4161	957.3	0.00727	314.94	429.04	1.3631	1.6899	1.938	1.812	1.766	259.	121.0	96.0	15.30	59.0	22.21	1.99	76.00
78.00	2.5228	943.1	0.00685	318.63	428.98	1.3733	1.6876	1.996	1.904	1.838	248.	119.4	92.5	15.56	58.1	22.92	1.80	78.00
80.00	2.6332	928.2	0.00645	322.39	428.81	1.3836	1.6850	2.065	2.012	1.924	237.	117.7	89.0	15.84	57.2	23.72	1.60	80.00
85.00	2.9258	887.2	0.00550	332.22	427.76	1.4104	1.6771	2.306	2.397	2.232	207.	113.1	80.2	16.67	54.9	26.22	1.14	85.00
90.00	3.2442	837.8	0.00461	342.93	425.42	1.4390	1.6662	2.756	3.121	2.820	176.	107.9	70.9	17.81	52.8	29.91	0.71	90.00
95.00	3.5912	772.7	0.00374	355.25	420.67	1.4715	1.6492	3.938	5.020	4.369	141.	101.9	60.4	19.61	51.7	36.40	0.33	95.00
100.00	3.9724	651.2	0.00268	373.30	407.68	1.5188	1.6109	17.59	25.35	20.81	101.	94.0	45.1	24.21	59.9	60.58	0.04	100.00
101.06c	4.0593	511.9	0.00195	389.64	389.64	1.5621	1.5621	•	•	•	0.	0.0	—	—	•	•	0.00	101.06

*Temperatures are on the ITS-90 scale a = triple point b = normal boiling point c = critical point

Table 2-4 SI Thermodynamic Properties of Moist Air at Standard Atmospheric Pressure, 101.325 kPa

(Table 2, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °C	Humidity Ratio, kg(w)/kg(da)	Specific Volume, m^3/kg (dry air)			Specific Enthalpy, kJ/kg (dry air)			Specific Entropy, kJ/(kg·K) (dry air)			Condensed Water			Temp., °C
											Specific Enthalpy, kJ/kg	Specific Entropy, kJ/(kg·K)	Vapor Pressure, kPa	
t	W_s	v_{da}	v_{as}	v_s	h_{da}	h_{as}	h_s	s_{da}	s_{as}	s_s	h_w	s_w	p_s	t
–60	0.0000067	0.6027	0.0000	0.6027	–60.351	0.017	–60.334	–0.2495	0.0001	–0.2494	–446.29	–1.6854	0.00108	–60
–59	0.0000076	0.6056	0.0000	0.6056	–59.344	0.018	–59.326	–0.2448	0.0001	–0.2447	–444.63	–1.6776	0.00124	–59
–58	0.0000087	0.6084	0.0000	0.6084	–58.338	0.021	–58.317	–0.2401	0.0001	–0.2400	–442.95	–1.6698	0.00141	–58
–57	0.0000100	0.6113	0.0000	0.6113	–57.332	0.024	–57.308	–0.2354	0.0001	–0.2353	–441.27	–1.6620	0.00161	–57
–56	0.0000114	0.6141	0.0000	0.6141	–56.326	0.028	–56.298	–0.2308	0.0001	–0.2306	–439.58	–1.6542	0.00184	–56
–55	0.0000129	0.6170	0.0000	0.6170	–55.319	0.031	–55.288	–0.2261	0.0002	–0.2260	–437.89	–1.6464	0.00209	–55
–54	0.0000147	0.6198	0.0000	0.6198	–54.313	0.036	–54.278	–0.2215	0.0002	–0.2214	–436.19	–1.6386	0.00238	–54
–53	0.0000167	0.6226	0.0000	0.6227	–53.307	0.041	–53.267	–0.2170	0.0002	–0.2168	–434.48	–1.6308	0.00271	–53
–52	0.0000190	0.6255	0.0000	0.6255	–52.301	0.046	–52.255	–0.2124	0.0002	–0.2122	–432.76	–1.6230	0.00307	–52
–51	0.0000215	0.6283	0.0000	0.6284	–51.295	0.052	–51.243	–0.2079	0.0002	–0.2076	–431.03	–1.6153	0.00348	–51
–50	0.0000243	0.6312	0.0000	0.6312	–50.289	0.059	–50.230	–0.2033	0.0003	–0.2031	–429.30	–1.6075	0.00394	–50
–49	0.0000275	0.6340	0.0000	0.6341	–49.283	0.067	–49.216	–0.1988	0.0003	–0.1985	–427.56	–1.5997	0.00445	–49
–48	0.0000311	0.6369	0.0000	0.6369	–48.277	0.075	–48.202	–0.1944	0.0004	–0.1940	–425.82	–1.5919	0.00503	–48
–47	0.0000350	0.6397	0.0000	0.6398	–47.271	0.085	–47.186	–0.1899	0.0004	–0.1895	–424.06	–1.5842	0.00568	–47
–46	0.0000395	0.6426	0.0000	0.6426	–46.265	0.095	–46.170	–0.1855	0.0004	–0.1850	–422.30	–1.5764	0.00640	–46
–45	0.0000445	0.6454	0.0000	0.6455	–45.259	0.108	–45.151	–0.1811	0.0005	–0.1805	–420.54	–1.5686	0.00721	–45
–44	0.0000500	0.6483	0.0001	0.6483	–44.253	0.121	–44.132	–0.1767	0.0006	–0.1761	–418.76	–1.5609	0.00811	–44
–43	0.0000562	0.6511	0.0001	0.6512	–43.247	0.137	–43.111	–0.1723	0.0006	–0.1716	–416.98	–1.5531	0.00911	–43
–42	0.0000631	0.6540	0.0001	0.6540	–42.241	0.153	–42.088	–0.1679	0.0007	–0.1672	–415.19	–1.5453	0.01022	–42
–41	0.0000708	0.6568	0.0001	0.6569	–41.235	0.172	–41.063	–0.1636	0.0008	–0.1628	–413.39	–1.5376	0.01147	–41
–40	0.0000793	0.6597	0.0001	0.6597	–40.229	0.192	–40.037	–0.1592	0.0009	–0.1584	–411.59	–1.5298	0.01285	–40
–39	0.0000887	0.6625	0.0001	0.6626	–39.224	0.216	–39.007	–0.1549	0.0010	–0.1540	–409.77	–1.5221	0.01438	–39
–38	0.0000992	0.6653	0.0001	0.6654	–38.218	0.241	–37.976	–0.1507	0.0011	–0.1496	–407.96	–1.5143	0.01608	–38
–37	0.0001108	0.6682	0.0001	0.6683	–37.212	0.270	–36.942	–0.1464	0.0012	–0.1452	–406.13	–1.5066	0.01796	–37
–36	0.0001237	0.6710	0.0001	0.6712	–36.206	0.302	–35.905	–0.1421	0.0014	–0.1408	–404.29	–1.4988	0.02005	–36
–35	0.0001379	0.6739	0.0001	0.6740	–35.200	0.336	–34.864	–0.1379	0.0015	–0.1364	–402.45	–1.4911	0.02235	–35
–34	0.0001536	0.6767	0.0002	0.6769	–34.195	0.375	–33.820	–0.1337	0.0017	–0.1320	–400.60	–1.4833	0.02490	–34
–33	0.0001710	0.6796	0.0002	0.6798	–33.189	0.417	–32.772	–0.1295	0.0018	–0.1276	–398.75	–1.4756	0.02772	–33
–32	0.0001902	0.6824	0.0002	0.6826	–32.183	0.464	–31.718	–0.1253	0.0020	–0.1233	–396.89	–1.4678	0.03082	–32
–31	0.0002113	0.6853	0.0002	0.6855	–31.178	0.517	–30.661	–0.1212	0.0023	–0.1189	–395.01	–1.4601	0.03425	–31
–30	0.0002346	0.6881	0.0003	0.6884	–30.171	0.574	–29.597	–0.1170	0.0025	–0.1145	–393.14	–1.4524	0.03802	–30
–29	0.0002602	0.6909	0.0003	0.6912	–29.166	0.636	–28.529	–0.1129	0.0028	–0.1101	–391.25	–1.4446	0.04217	–29
–28	0.0002883	0.6938	0.0003	0.6941	–28.160	0.707	–27.454	–0.1088	0.0031	–0.1057	–389.36	–1.4369	0.04673	–28
–27	0.0003193	0.6966	0.0004	0.6970	–27.154	0.782	–26.372	–0.1047	0.0034	–0.1013	–387.46	–1.4291	0.05175	–27
–26	0.0003533	0.6995	0.0004	0.6999	–26.149	0.867	–25.282	–0.1006	0.0037	–0.0969	–385.55	–1.4214	0.05725	–26
–25	0.0003905	0.7023	0.0004	0.7028	–25.143	0.959	–24.184	–0.0965	0.0041	–0.0924	–383.63	–1.4137	0.06329	–25
–24	0.0004314	0.7052	0.0005	0.7057	–24.137	1.059	–23.078	–0.0925	0.0045	–0.0880	–381.71	–1.4059	0.06991	–24
–23	0.0004762	0.7080	0.0005	0.7086	–23.132	1.171	–21.961	–0.0885	0.0050	–0.0835	–379.78	–1.3982	0.07716	–23
–22	0.0005251	0.7109	0.0006	0.7115	–22.126	1.292	–20.834	–0.0845	0.0054	–0.0790	–377.84	–1.3905	0.08510	–22
–21	0.0005787	0.7137	0.0007	0.7144	–21.120	1.425	–19.695	–0.0805	0.0060	–0.0745	–375.90	–1.3828	0.09378	–21
–20	0.0006373	0.7165	0.0007	0.7173	–20.115	1.570	–18.545	–0.0765	0.0066	–0.0699	–373.95	–1.3750	0.10326	–20
–19	0.0007013	0.7194	0.0008	0.7202	–19.109	1.729	–17.380	–0.0725	0.0072	–0.0653	–371.99	–1.3673	0.11362	–19
–18	0.0007711	0.7222	0.0009	0.7231	–18.103	1.902	–16.201	–0.0686	0.0079	–0.0607	–370.02	–1.3596	0.12492	–18
–17	0.0008473	0.7251	0.0010	0.7261	–17.098	2.092	–15.006	–0.0646	0.0086	–0.0560	–368.04	–1.3518	0.13725	–17
–16	0.0009303	0.7279	0.0011	0.7290	–16.092	2.299	–13.793	–0.0607	0.0094	–0.0513	–366.06	–1.3441	0.15068	–16
–15	0.0010207	0.7308	0.0012	0.7320	–15.086	2.524	–12.562	–0.0568	0.0103	–0.0465	–364.07	–1.3364	0.16530	–15
–14	0.0011191	0.7336	0.0013	0.7349	–14.080	2.769	–11.311	–0.0529	0.0113	–0.0416	–362.07	–1.3287	0.18122	–14
–13	0.0012262	0.7364	0.0014	0.7379	–13.075	3.036	–10.039	–0.0490	0.0123	–0.0367	–360.07	–1.3210	0.19852	–13
–12	0.0013425	0.7393	0.0016	0.7409	–12.069	3.327	–8.742	–0.0452	0.0134	–0.0318	–358.06	–1.3132	0.21732	–12
–11	0.0014690	0.7421	0.0017	0.7439	–11.063	3.642	–7.421	–0.0413	0.0146	–0.0267	–356.04	–1.3055	0.23775	–11
–10	0.0016062	0.7450	0.0019	0.7469	–10.057	3.986	–6.072	–0.0375	0.0160	–0.0215	–354.01	–1.2978	0.25991	–10
–9	0.0017551	0.7478	0.0021	0.7499	–9.052	4.358	–4.693	–0.0337	0.0174	–0.0163	–351.97	–1.2901	0.28395	–9
–8	0.0019166	0.7507	0.0023	0.7530	–8.046	4.764	–3.283	–0.0299	0.0189	–0.0110	–349.93	–1.2824	0.30999	–8
–7	0.0020916	0.7535	0.0025	0.7560	–7.040	5.202	–1.838	–0.0261	0.0206	–0.0055	–347.88	–1.2746	0.33821	–7
–6	0.0022811	0.7563	0.0028	0.7591	–6.035	5.677	–0.357	–0.0223	0.0224	–0.0000	–345.82	–1.2669	0.36874	–6
–5	0.0024862	0.7592	0.0030	0.7622	–5.029	6.192	1.164	–0.0186	0.0243	–0.0057	–343.76	–1.2592	0.40178	–5
–4	0.0027081	0.7620	0.0033	0.7653	–4.023	6.751	2.728	–0.0148	0.0264	–0.0115	–341.69	–1.2515	0.43748	–4
–3	0.0029480	0.7649	0.0036	0.7685	–3.017	7.353	4.336	–0.0111	0.0286	–0.0175	–339.61	–1.2438	0.47606	–3
–2	0.0032074	0.7677	0.0039	0.7717	–2.011	8.007	5.995	–0.0074	0.0310	–0.0236	–337.52	–1.2361	0.51773	–2
–1	0.0034874	0.7705	0.0043	0.7749	–1.006	8.712	7.706	–0.0037	0.0336	–0.0299	–335.42	–1.2284	0.56268	–1
0	0.0037895	0.7734	0.0047	0.7781	0.000	9.473	9.473	0.0000	0.0364	0.0364	–333.32	–1.2206	0.61117	0
0*	0.003789	0.7734	0.0047	0.7781	0.000	9.473	9.473	0.0000	0.0364	0.0364	0.06	–0.0001	0.6112	0
1	0.004076	0.7762	0.0051	0.7813	1.006	10.197	11.203	0.0037	0.0391	0.0427	4.28	0.0153	0.6571	1
2	0.004381	0.7791	0.0055	0.7845	2.012	10.970	12.982	0.0073	0.0419	0.0492	8.49	0.0306	0.7060	2
3	0.004707	0.7819	0.0059	0.7878	3.018	11.793	14.811	0.0110	0.0449	0.0559	12.70	0.0459	0.7581	3
4	0.005054	0.7848	0.0064	0.7911	4.024	12.672	16.696	0.0146	0.0480	0.0627	16.91	0.0611	0.8135	4
5	0.005424	0.7876	0.0068	0.7944	5.029	13.610	18.639	0.0182	0.0514	0.0697	21.12	0.0762	0.8725	5
6	0.005818	0.7904	0.0074	0.7978	6.036	14.608	20.644	0.0219	0.0550	0.0769	25.32	0.0913	0.9353	6
7	0.006237	0.7933	0.0079	0.8012	7.041	15.671	22.713	0.0255	0.0588	0.0843	29.52	0.1064	1.0020	7
8	0.006683	0.7961	0.0085	0.8046	8.047	16.805	24.852	0.0290	0.0628	0.0919	33.72	0.1213	1.0729	8
9	0.007157	0.7990	0.0092	0.8081	9.053	18.010	27.064	0.0326	0.0671	0.0997	37.92	0.1362	1.1481	9
10	0.007661	0.8018	0.0098	0.8116	10.059	19.293	29.352	0.0362	0.0717	0.1078	42.11	0.1511	1.2280	10
11	0.008197	0.8046	0.0106	0.8152	11.065	20.658	31.724	0.0397	0.0765	0.1162	46.31	0.1659	1.3128	11
12	0.008766	0.8075	0.0113	0.8188	12.071	22.108	34.179	0.0433	0.0816	0.1248	50.50	0.1806	1.4026	12
13	0.009370	0.8103	0.0122	0.8225	13.077	23.649	36.726	0.0468	0.0870	0.1337	54.69	0.1953	1.4979	13

*Extrapolated to represent metastable equilibrium with undercooled liquid.

Table 2-4 SI Thermodynamic Properties of Moist Air at Standard Atmospheric Pressure, 101.325 kPa (*Continued*)

(Table 2, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °C	Humidity Ratio, kg(w)/kg(da)	Specific Volume, m^3/kg (dry air)			Specific Enthalpy, kJ/kg (dry air)			Specific Entropy, kJ/(kg·K) (dry air)			Condensed Water			Temp., °C
											Specific Enthalpy, kJ/kg	Specific Entropy, kJ/(kg·K)	Vapor Pressure, kPa	
t	W_s	v_{da}	v_{as}	v_s	h_{da}	h_{as}	h_s	s_{da}	s_{as}	s_s	h_w	s_w	p_s	t
14	0.010012	0.8132	0.0131	0.8262	14.084	25.286	39.370	0.0503	0.0927	0.1430	58.88	0.2099	1.5987	14
15	0.010692	0.8160	0.0140	0.8300	15.090	27.023	42.113	0.0538	0.0987	0.1525	63.07	0.2244	1.7055	15
16	0.011413	0.8188	0.0150	0.8338	16.096	28.867	44.963	0.0573	0.1051	0.1624	67.26	0.2389	1.8185	16
17	0.012178	0.8217	0.0160	0.8377	17.102	30.824	47.926	0.0607	0.1119	0.1726	71.44	0.2534	1.9380	17
18	0.012989	0.8245	0.0172	0.8417	18.108	32.900	51.008	0.0642	0.1190	0.1832	75.63	0.2678	2.0643	18
19	0.013848	0.8274	0.0184	0.8457	19.114	35.101	54.216	0.0677	0.1266	0.1942	79.81	0.2821	2.1979	19
20	0.014758	0.8302	0.0196	0.8498	20.121	37.434	57.555	0.0711	0.1346	0.2057	84.00	0.2965	2.3389	20
21	0.015721	0.8330	0.0210	0.8540	21.127	39.908	61.035	0.0745	0.1430	0.2175	88.18	0.3107	2.4878	21
22	0.016741	0.8359	0.0224	0.8583	22.133	42.527	64.660	0.0779	0.1519	0.2298	92.36	0.3249	2.6448	22
23	0.017821	0.8387	0.0240	0.8627	23.140	45.301	68.440	0.0813	0.1613	0.2426	96.55	0.3390	2.8105	23
24	0.018963	0.8416	0.0256	0.8671	24.146	48.239	72.385	0.0847	0.1712	0.2559	100.73	0.3531	2.9852	24
25	0.020170	0.8444	0.0273	0.8717	25.153	51.347	76.500	0.0881	0.1817	0.2698	104.91	0.3672	3.1693	25
26	0.021448	0.8472	0.0291	0.8764	26.159	54.638	80.798	0.0915	0.1927	0.2842	109.09	0.3812	3.3633	26
27	0.022798	0.8501	0.0311	0.8811	27.165	58.120	85.285	0.0948	0.2044	0.2992	113.27	0.3951	3.5674	27
28	0.024226	0.8529	0.0331	0.8860	28.172	61.804	89.976	0.0982	0.2166	0.3148	117.45	0.4090	3.7823	28
29	0.025735	0.8558	0.0353	0.8910	29.179	65.699	94.878	0.1015	0.2296	0.3311	121.63	0.4229	4.0084	29
30	0.027329	0.8586	0.0376	0.8962	30.185	69.820	100.006	0.1048	0.2432	0.3481	125.81	0.4367	4.2462	30
31	0.029014	0.8614	0.0400	0.9015	31.192	74.177	105.369	0.1082	0.2576	0.3658	129.99	0.4505	4.4961	31
32	0.030793	0.8643	0.0426	0.9069	32.198	78.780	110.979	0.1115	0.2728	0.3842	134.17	0.4642	4.7586	32
33	0.032674	0.8671	0.0454	0.9125	33.205	83.652	116.857	0.1148	0.2887	0.4035	138.35	0.4779	5.0345	33
34	0.034660	0.8700	0.0483	0.9183	34.212	88.799	123.011	0.1180	0.3056	0.4236	142.53	0.4915	5.3242	34
35	0.036756	0.8728	0.0514	0.9242	35.219	94.236	129.455	0.1213	0.3233	0.4446	146.71	0.5051	5.6280	35
36	0.038971	0.8756	0.0546	0.9303	36.226	99.983	136.209	0.1246	0.3420	0.4666	150.89	0.5186	5.9468	36
37	0.041309	0.8785	0.0581	0.9366	37.233	106.058	143.290	0.1278	0.3617	0.4895	155.07	0.5321	6.2812	37
38	0.043778	0.8813	0.0618	0.9431	38.239	112.474	150.713	0.1311	0.3824	0.5135	159.25	0.5456	6.6315	38
39	0.046386	0.8842	0.0657	0.9498	39.246	119.258	158.504	0.1343	0.4043	0.5386	163.43	0.5590	6.9988	39
40	0.049141	0.8870	0.0698	0.9568	40.253	126.430	166.683	0.1375	0.4273	0.5649	167.61	0.5724	7.3838	40
41	0.052049	0.8898	0.0741	0.9640	41.261	134.005	175.265	0.1407	0.4516	0.5923	171.79	0.5857	7.7866	41
42	0.055119	0.8927	0.0788	0.9714	42.268	142.007	184.275	0.1439	0.4771	0.6211	175.97	0.5990	8.2081	42
43	0.058365	0.8955	0.0837	0.9792	43.275	150.475	193.749	0.1471	0.5041	0.6512	180.15	0.6122	8.6495	43
44	0.061791	0.8983	0.0888	0.9872	44.282	159.417	203.699	0.1503	0.5325	0.6828	184.33	0.6254	9.1110	44
45	0.065411	0.9012	0.0943	0.9955	45.289	168.874	214.164	0.1535	0.5624	0.7159	188.51	0.6386	9.5935	45
46	0.069239	0.9040	0.1002	1.0042	46.296	178.882	225.179	0.1566	0.5940	0.7507	192.69	0.6517	10.0982	46
47	0.073282	0.9069	0.1063	1.0132	47.304	189.455	236.759	0.1598	0.6273	0.7871	196.88	0.6648	10.6250	47
48	0.077556	0.9097	0.1129	1.0226	48.311	200.644	248.955	0.1629	0.6624	0.8253	201.06	0.6778	11.1754	48
49	0.082077	0.9125	0.1198	1.0323	49.319	212.485	261.803	0.1661	0.6994	0.8655	205.24	0.6908	11.7502	49
50	0.086858	0.9154	0.1272	1.0425	50.326	225.019	275.345	0.1692	0.7385	0.9077	209.42	0.7038	12.3503	50
51	0.091918	0.9182	0.1350	1.0532	51.334	238.290	289.624	0.1723	0.7798	0.9521	213.60	0.7167	12.9764	51
52	0.097272	0.9211	0.1433	1.0643	52.341	252.340	304.682	0.1754	0.8234	0.9988	217.78	0.7296	13.6293	52
53	0.102948	0.9239	0.1521	1.0760	53.349	267.247	320.596	0.1785	0.8695	1.0480	221.97	0.7424	14.3108	53
54	0.108954	0.9267	0.1614	1.0882	54.357	283.031	337.388	0.1816	0.9182	1.0998	226.15	0.7552	15.0205	54
55	0.115321	0.9296	0.1713	1.1009	55.365	299.772	355.137	0.1847	0.9698	1.1544	230.33	0.7680	15.7601	55
56	0.122077	0.9324	0.1819	1.1143	56.373	317.549	373.922	0.1877	1.0243	1.2120	234.52	0.7807	16.5311	56
57	0.129243	0.9353	0.1932	1.1284	57.381	336.417	393.798	0.1908	1.0820	1.2728	238.70	0.7934	17.3337	57
58	0.136851	0.9381	0.2051	1.1432	58.389	356.461	414.850	0.1938	1.1432	1.3370	242.88	0.8061	18.1691	58
59	0.144942	0.9409	0.2179	1.1588	59.397	377.788	437.185	0.1969	1.2081	1.4050	247.07	0.8187	19.0393	59
60	0.15354	0.9438	0.2315	1.1752	60.405	400.458	460.863	0.1999	1.2769	1.4768	251.25	0.8313	19.9439	60
61	0.16269	0.9466	0.2460	1.1926	61.413	424.624	486.036	0.2029	1.3500	1.5530	255.44	0.8438	20.8858	61
62	0.17244	0.9494	0.2614	1.2109	62.421	450.377	512.798	0.2059	1.4278	1.6337	259.62	0.8563	21.8651	62
63	0.18284	0.9523	0.2780	1.2303	63.429	477.837	541.266	0.2089	1.5104	1.7194	263.81	0.8688	22.8826	63
64	0.19393	0.9551	0.2957	1.2508	64.438	507.177	571.615	0.2119	1.5985	1.8105	268.00	0.8812	23.9405	64
65	0.20579	0.9580	0.3147	1.2726	65.446	538.548	603.995	0.2149	1.6925	1.9074	272.18	0.8936	25.0397	65
66	0.21848	0.9608	0.3350	1.2958	66.455	572.116	638.571	0.2179	1.7927	2.0106	276.37	0.9060	26.1810	66
67	0.23207	0.9636	0.3568	1.3204	67.463	608.103	675.566	0.2209	1.8999	2.1208	280.56	0.9183	27.3664	67
68	0.24664	0.9665	0.3803	1.3467	68.472	646.724	715.196	0.2238	2.0147	2.2385	284.75	0.9306	28.5967	68
69	0.26231	0.9693	0.4055	1.3749	69.481	688.261	757.742	0.2268	2.1378	2.3646	288.94	0.9429	29.8741	69
70	0.27916	0.9721	0.4328	1.4049	70.489	732.959	803.448	0.2297	2.2699	2.4996	293.13	0.9551	31.1986	70
71	0.29734	0.9750	0.4622	1.4372	71.498	781.208	852.706	0.2327	2.4122	2.6448	297.32	0.9673	32.5734	71
72	0.31698	0.9778	0.4941	1.4719	72.507	833.335	905.842	0.2356	2.5655	2.8010	301.51	0.9794	33.9983	72
73	0.33824	0.9807	0.5287	1.5093	73.516	889.807	963.323	0.2385	2.7311	2.9696	305.70	0.9916	35.4759	73
74	0.36130	0.9835	0.5662	1.5497	74.525	951.077	1025.603	0.2414	2.9104	3.1518	309.89	1.0037	37.0063	74
75	0.38641	0.9863	0.6072	1.5935	75.535	1017.841	1093.375	0.2443	3.1052	3.3496	314.08	1.0157	38.5940	75
76	0.41377	0.9892	0.6519	1.6411	76.543	1090.628	1167.172	0.2472	3.3171	3.5644	318.28	1.0278	40.2369	76
77	0.44372	0.9920	0.7010	1.6930	77.553	1170.328	1247.881	0.2501	3.5486	3.7987	322.47	1.0398	41.9388	77
78	0.47663	0.9948	0.7550	1.7498	78.562	1257.921	1336.483	0.2530	3.8023	4.0553	326.67	1.0517	43.7020	78
79	0.51284	0.9977	0.8145	1.8121	79.572	1354.347	1433.918	0.2559	4.0810	4.3368	330.86	1.0636	45.5248	79
80	0.55295	1.0005	0.8805	1.8810	80.581	1461.200	1541.781	0.2587	4.3890	4.6477	335.06	1.0755	47.4135	80
81	0.59751	1.0034	0.9539	1.9572	81.591	1579.961	1661.552	0.2616	4.7305	4.9921	339.25	1.0874	49.3670	81
82	0.64724	1.0062	1.0360	2.0422	82.600	1712.547	1795.148	0.2644	5.1108	5.3753	343.45	1.0993	51.3860	82
83	0.70311	1.0090	1.1283	2.1373	83.610	1861.548	1945.158	0.2673	5.5372	5.8045	347.65	1.1111	53.4746	83
84	0.76624	1.0119	1.2328	2.2446	84.620	2029.983	2114.603	0.2701	6.0181	6.2882	351.85	1.1228	55.6337	84
85	0.83812	1.0147	1.3518	2.3666	85.630	2221.806	2307.436	0.2729	6.5644	6.8373	356.05	1.1346	57.8658	85
86	0.92062	1.0175	1.4887	2.5062	86.640	2442.036	2528.677	0.2757	7.1901	7.4658	360.25	1.1463	60.1727	86
87	1.01611	1.0204	1.6473	2.6676	87.650	2697.016	2784.666	0.2785	7.9128	8.1914	364.45	1.1580	62.5544	87
88	1.12800	1.0232	1.8333	2.8565	88.661	2995.890	3084.551	0.2813	8.7580	9.0393	368.65	1.1696	65.0166	88
89	1.26064	1.0261	2.0540	3.0800	89.671	3350.254	3439.925	0.2841	9.7577	10.0419	372.86	1.1812	67.5581	89
90	1.42031	1.0289	2.3199	3.3488	90.681	3776.918	3867.599	0.2869	10.9586	11.2455	377.06	1.1928	70.1817	90

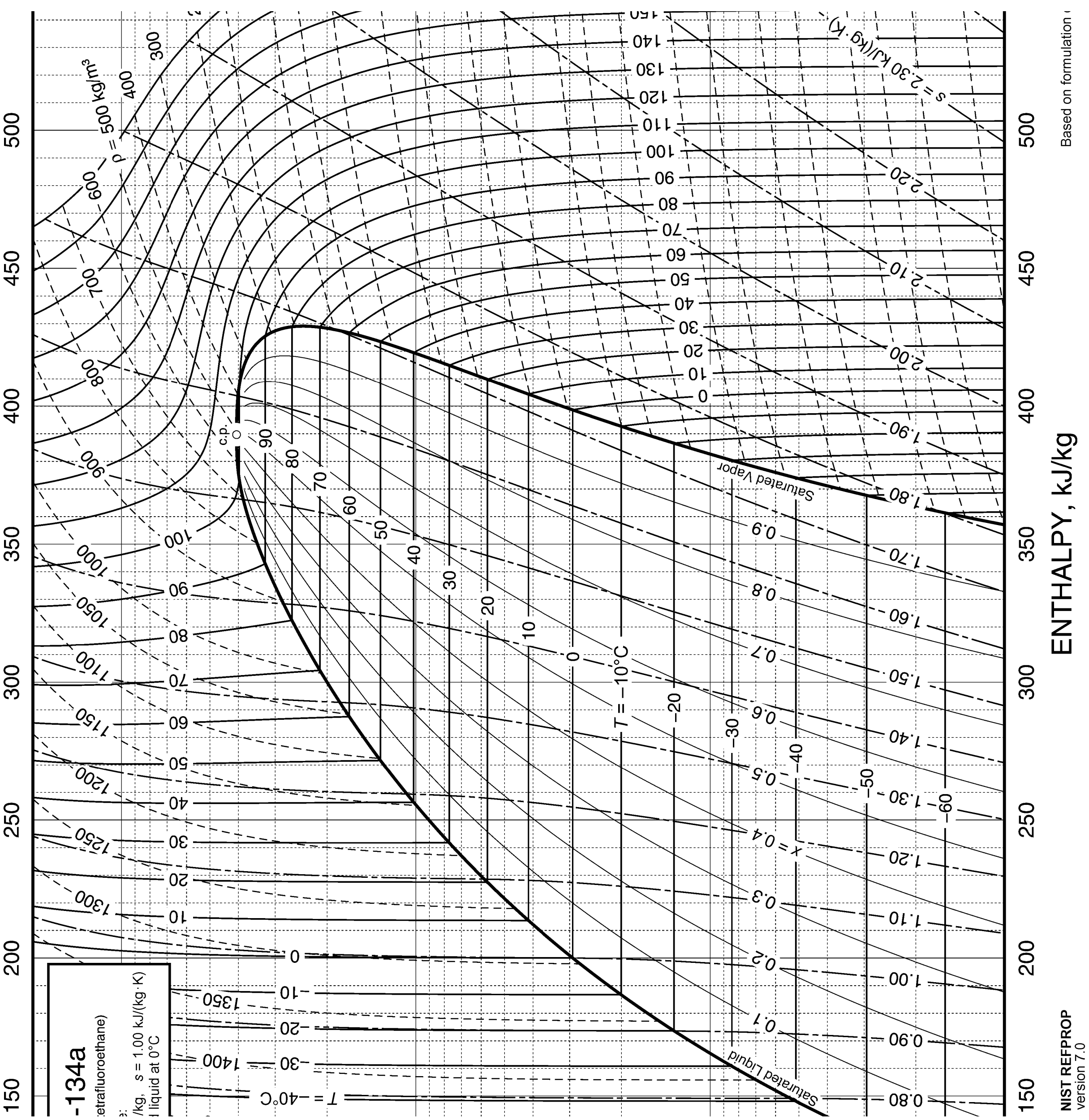


Fig. 2-5 SI Pressure-Enthalpy Diagram for Refrigerant 134a

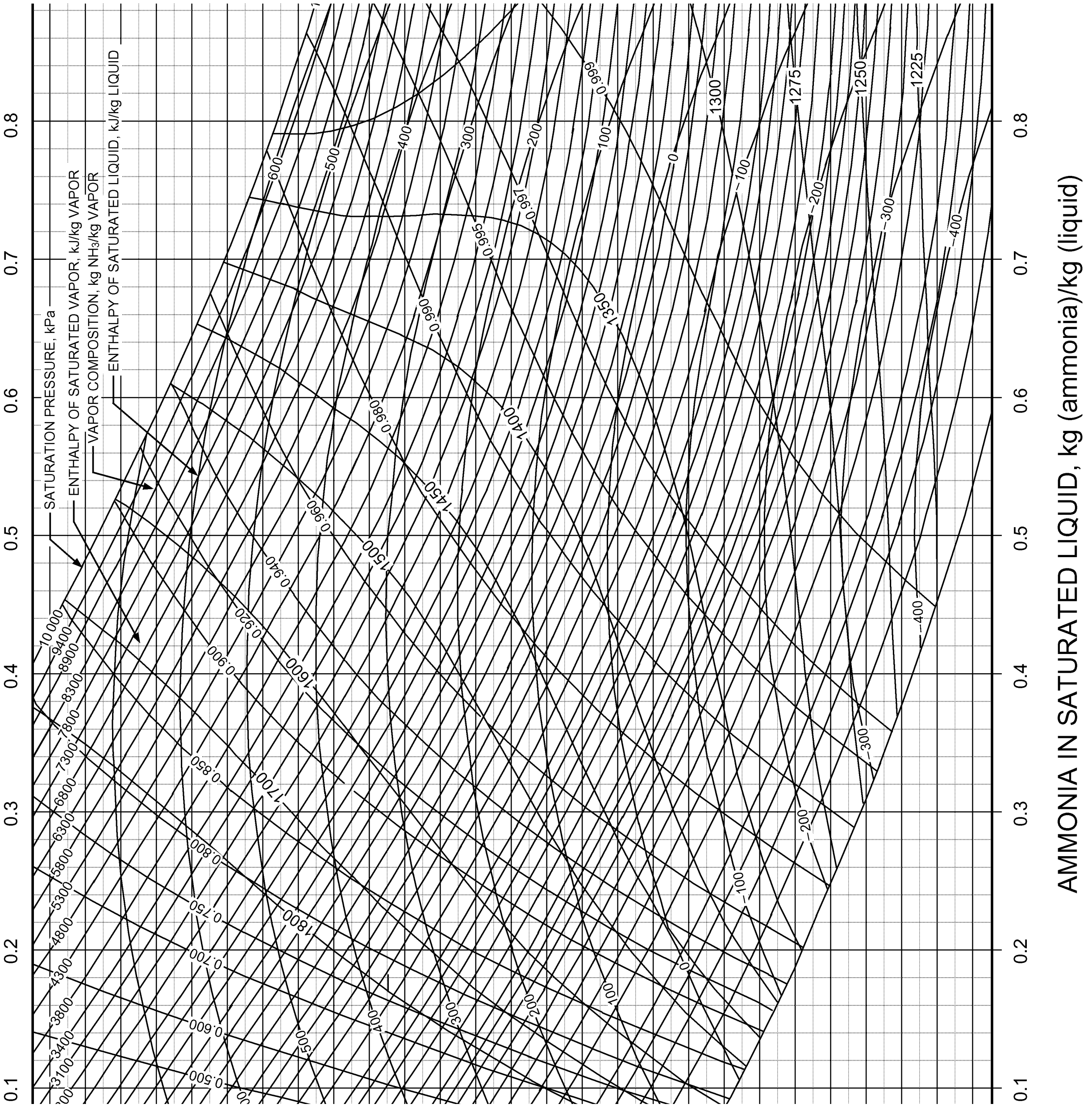
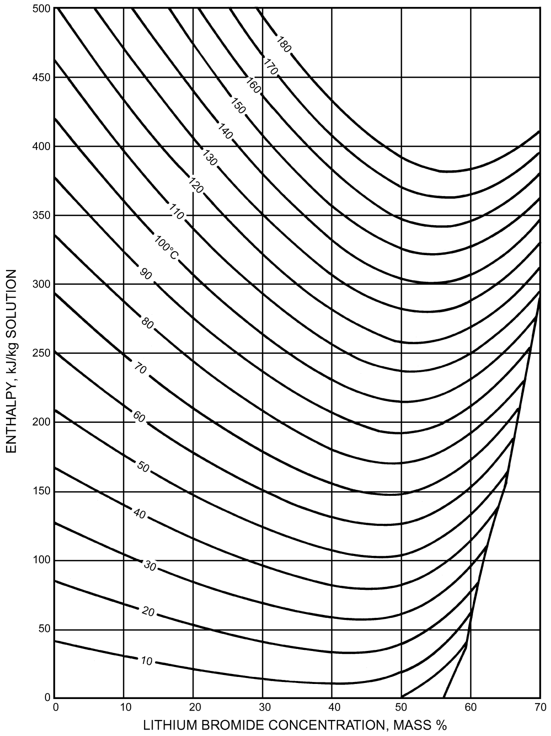


Fig. 2-6 SI Enthalpy-Concentration Diagram for Aqua-Ammonia



EQUATIONS **CONCENTRATION RANGE $40 < X < 70\%$ LiBr** **TEMPERATURE RANGE $15 < t < 165$°C**

$h = \Sigma_0^4 A_n X^n + t\Sigma_0^4 B_n X^n + t^2\Sigma_0^4 C_n X^n$ in kJ/kg, where t = °C and X = %LiBr

$A_0 = -2024.33$
$A_1 = 163.309$
$A_2 = -4.88161$
$A_3 = 6.302948$ E-2
$A_4 = -2.913705$ E-4

$B_0 = 18.2829$
$B_1 = -1.1691757$
$B_2 = 3.248041$ E-2
$B_3 = -4.034184$ E-4
$B_4 = 1.8520569$ E-6

$C_0 = -3.7008214$ E-2
$C_1 = 2.8877666$ E-3
$C_2 = -8.1313015$ E-5
$C_3 = 9.9116628$ E-7
$C_4 = -4.4441207$ E-9

EQUATIONS

1. $t = \Sigma_0^3 B_n X^n + t'\Sigma_0^3 A_n X^n$ Solution Temperature, °C
2. $t' = (t - \Sigma_0^3 B_n X^n)/\Sigma_0^3 A_n X^n$ Refrigerant Temperature, °C
3. $\log P = C + D/T' + E/T'^2$ P = kPa; T' = K
4. $T' = \dfrac{-2E}{D + [D^2 - 4E(C - \log P)]^{0.5}}$

$A_0 = -2.007\ 55$
$A_1 = 0.169\ 76$
$A_2 = -3.133\ 362$ E-03
$A_3 = 1.976\ 68$ E-05

$B_0 = 124.937$
$B_1 = -7.716\ 49$
$B_2 = 0.152\ 286$
$B_3 = -7.950\ 90$ E-04

$C = 7.05$
$D = -1596.49$
$E = -104\ 095.5$

RANGE: $-15 < t' < 110$°C
$5 < t < 175$°C
$45 < X < 70\%$ LiBr

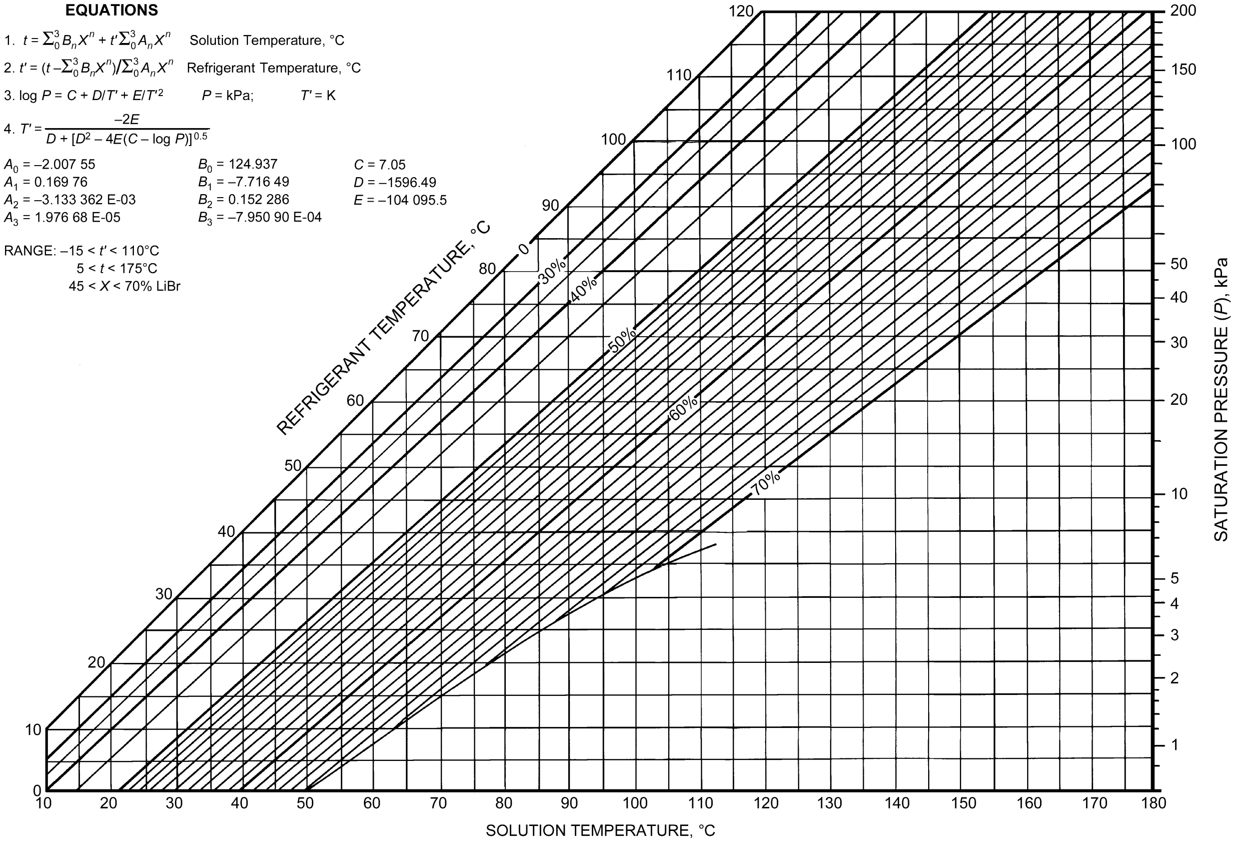


Fig. 2-7 SI Enthalpy-Concentration and Equilibrium Charts for Li-Br/Water

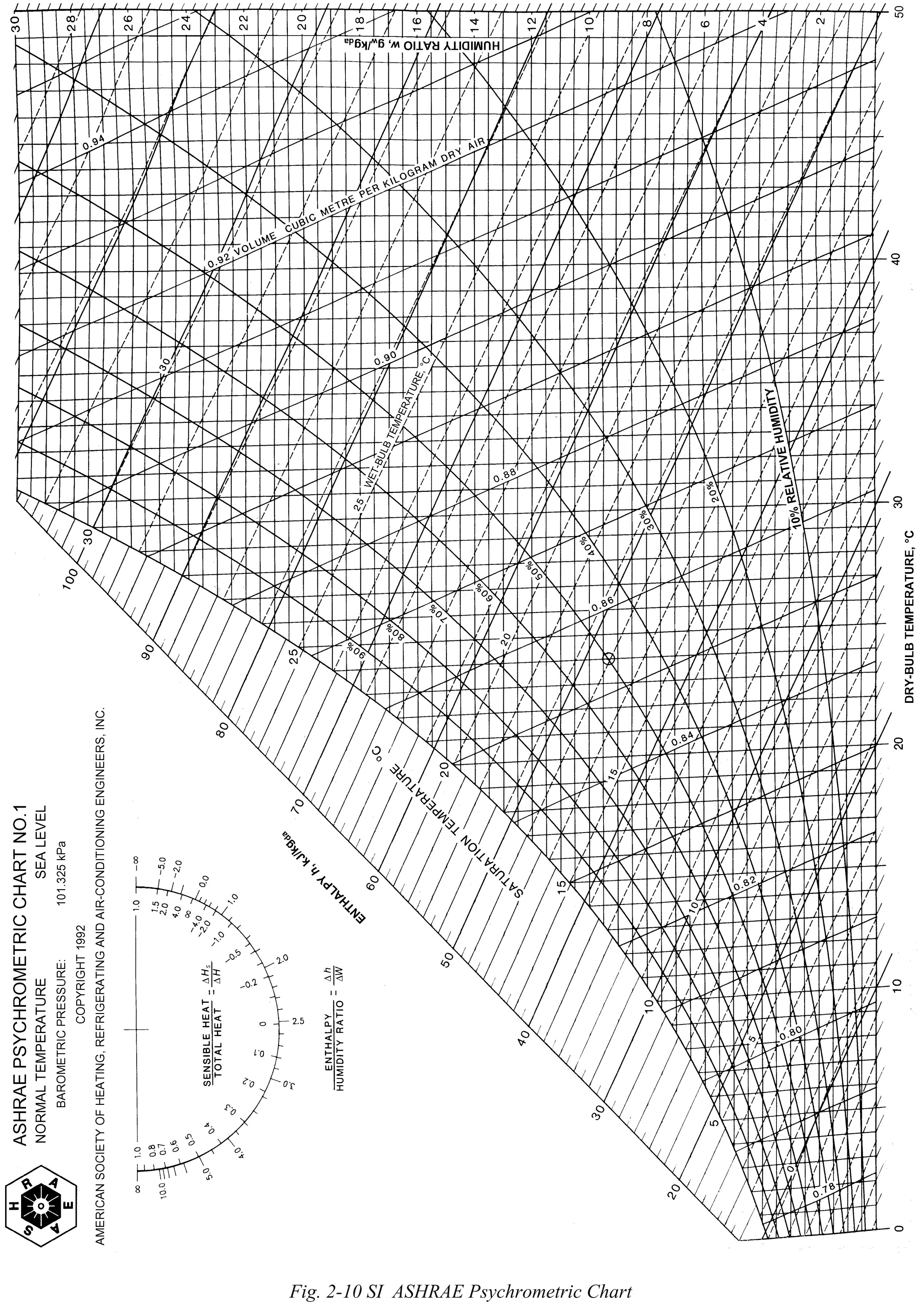


Fig. 2-10 SI ASHRAE Psychrometric Chart

Chapter 3

BASIC HVAC SYSTEM CALCULATIONS

This chapter illustrates the application of the principles of thermodynamics and psychrometrics to the various processes found in air-conditioning systems. It relates to material in Chapter 1 of the 2017 *ASHRAE Handbook—Fundamentals.*

3.1 Applying Thermodynamics to HVAC Processes

A simple but complete air-conditioning system is given schematically in Figure 3-1, which shows various space heat and moisture transfers. The symbol q_S represents a sensible heat transfer rate; the symbol m_w represents a moisture transfer rate. The symbol q_L designates the transfer of energy that accompanies moisture transfer and is given by $\Sigma m_w h_w$, where h_w is the specific enthalpy of the added (or removed) moisture. Solar radiation and internal loads are always gains for the space. Heat transmission through solid construction components due to a temperature difference, as well as energy transfer because of infiltration, may represent a gain or a loss.

Note that the energy q_c and moisture m_c transfers at the conditioner cannot be determined from the space heat and moisture transfers alone. The effect of the outdoor ventilation air, as well as other system load components, must be included. The designer must recognize that items such as fan energy, duct transmission, roof and ceiling transmissions, heat from lights, bypass and leakage, type of return air system, location of main fans, and actual versus design room conditions are all related to one another, to component sizing, and to system arrangement.

The first law of thermodynamics (energy balance) and the conservation of mass (mass balance) are the basis for the analysis of moist air processes. The following sections demonstrate the application of these laws to specific HVAC processes.

In many air-conditioning systems, air is removed from the room, returned to the air-conditioning apparatus where it is reconditioned, and then supplied again to the room. In most systems, the return air from the room is mixed with outdoor air required for ventilation. A typical air-conditioning system and the corresponding psychrometric representation of the processes for cooling are illustrated in Figure 3-2. Outdoor air o is mixed with return air r from the room and enters the apparatus at condition m. Air flows through the conditioner and is supplied to the space s. The air supplied to the space absorbs heat q_s and moisture m_w, and the cycle is repeated.

A typical psychrometric representation of the previous system operating under conditions of heating followed by humidification is given as Figure 3-3.

3.1.1 Absorption of Space Heat and Moisture Gains

The problem of air conditioning a space usually reduces to determining the quantity of moist air that must be supplied and the condition it must have to remove given amounts of energy and water from the space and be withdrawn at a specified condition. A space with incident rates of energy and moisture gains is shown in Figure 3-4. The quantity q_s denotes

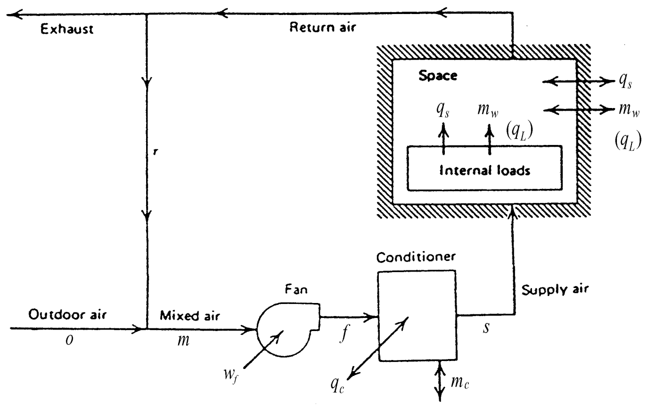


Fig. 3-1 Schematic of Air-Conditioning System

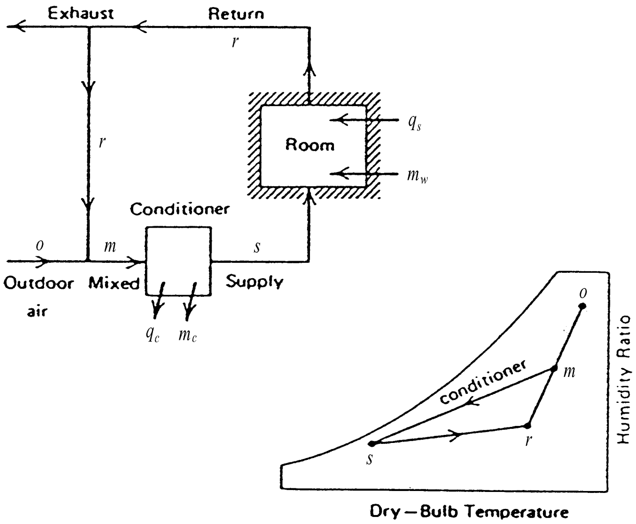


Fig. 3-2 Typical Air-Conditioning Cooling Processes

the net sum of all rates of heat gain upon the space arising from transfers through boundaries and from sources within the space. This **sensible heat gain** involves the addition of energy alone, and does not include energy contributions due to the addition of water (or water vapor). The quantity m_w denotes the net sum of all rates of moisture gain upon the space arising from transfers through boundaries and from sources within the space. Each pound (kilogram) of moisture injected into the space adds an amount of energy equal to its specific enthalpy. A typical value of h_w is 1100 Btu/lb.

Assuming steady-state conditions, the governing equations are

$$m_a h_1 + m_w h_w - m_a h_2 + q_s = 0 \tag{3-1}$$

$$m_a W_1 + m_w = m_a W_2 \tag{3-2}$$

3.1.2 Heating or Cooling of Air

When air is heated or cooled without moisture loss or gain, the process yields a straight horizontal line on the psychrometric chart because the humidity ratio is constant. Such processes can occur when moist air flows through a heat exchanger (Figure 3-5).

For steady-flow conditions, the governing equations are

$$m_a h_1 - m_a h_2 + q = 0 \tag{3-3}$$

$$W_2 = W_1 \tag{3-4}$$

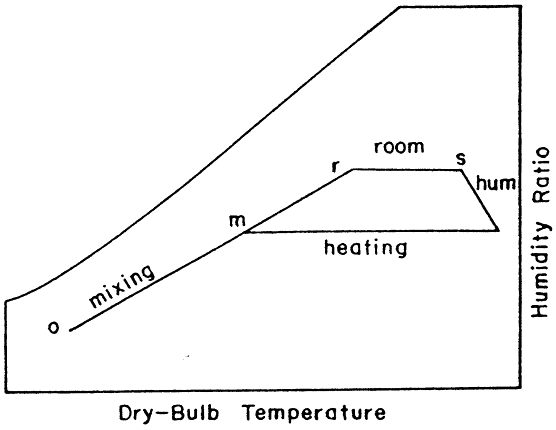


Fig. 3-3 Psychrometric Representation of Heating/ Humidifying Process

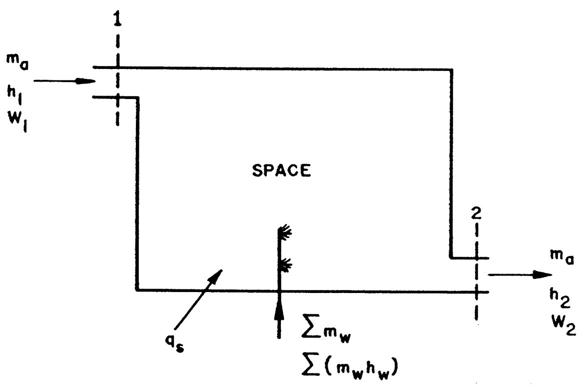


Fig. 3-4 Space HVAC Process

3.1.3 Cooling and Dehumidifying Air

When moist air is cooled to a temperature below its dew point, some of the water vapor condenses and leaves the air stream. A schematic cooling and dehumidifying device is shown in Figure 3-6.

Although the actual process path varies depending on the type of surface, surface temperature, and flow conditions, the heat and mass transfer can be expressed in terms of the initial and final states.

Although water may condense out at various temperatures ranging from the initial dew point to the final saturation temperature, it is assumed that condensed water is cooled to the final air temperature t_2 before it drains from the system. For the system shown in Figure 3-6, the steady-flow energy and material balance equations are

$$m_a h_1 = m_a h_2 + q + m_w h_{w_2} \tag{3-5}$$

$$m_a W_1 = m_a W_2 + m_w \tag{3-6}$$

Thus

$$m_w = m_a (W_1 - W_2) \tag{3-7}$$

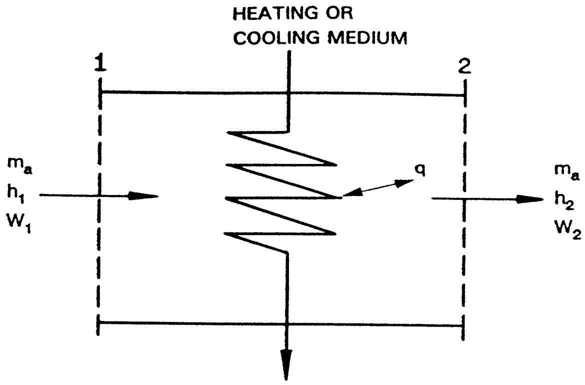


Fig. 3-5 Schematic Heating or Cooling Device

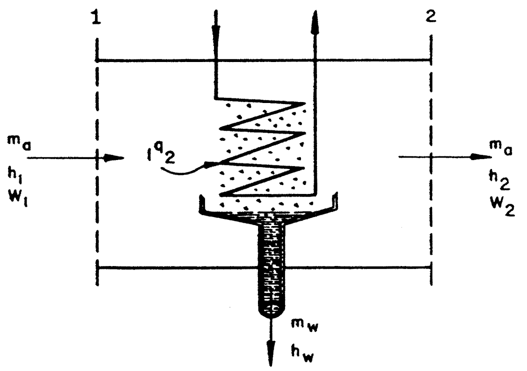


Fig. 3-6 Schematic Cooling and Dehumidifying Device

$$q = m_a[(h_1 - h_2) - (W_1 - W_2)h_{w_2}] \tag{3-8}$$

The cooling and dehumidifying process involves both sensible and latent heat transfer where sensible heat transfer is associated with the decrease in dry-bulb temperature and the latent heat transfer is associated with the decrease in humidity ratio. These quantities may be expressed as

$$q_S = m_a c_p(t_1 - t_2) \tag{3-9}$$

$$q_L = m_a(W_1 - W_2)h_{fg} \tag{3-10}$$

Example 3-1 In a steady-flow process, a cooling and dehumidifying coil receives an air-water vapor mixture at 16 psia, 95°F, and 83% rh and discharges the air-water vapor mixture at 14.7 psia, 50°F, and 96% rh. The condensate leaves the unit at 50°F. Calculate the heat transfer per pound of dry air flowing through the unit.

Solution:

The first law of thermodynamics for a steady-state, steady-flow system is given by

$${}_1\dot{q}_2 + \left(\frac{V_1^2}{2g_c} + \frac{g}{g_c}z_1 + h_1\right)m_{a_1} = {}_1W_2 + \left(\frac{V_2^2}{2g_c} + \frac{g}{g_c}z_2 + h_2\right)m_{a_2}$$

The continuity equation is

$$m = A\rho V$$

For the air flowing through the apparatus this becomes

$$m_{a_1} = m_{a_2} = m_a$$

For the water vapor this becomes

$$m_{w_1} - m_{w_2} = m_{\text{condensate}}$$

Neglecting any kinetic or potential energy changes of the flowing fluid and noting that there is no mechanical work being done on or by the system, the First Law equation reduces to

$${}_1\dot{q}_2 = m_a(h_2 - h_1) + m_{\text{condensate}}h_{\text{condensate}}$$

and the enthalpy terms may be expanded, so that

$$\frac{{}_1\dot{q}_2}{m_a} = h_{a_2} - h_{a_1} + W_2h_{w_2} - W_1h_{w_1} + \frac{m_{\text{condensate}}}{m_a}h_{\text{condensate}}$$

With a mass balance on the water this becomes

$$\frac{{}_1\dot{q}_2}{m_a} = h_{a_2} - h_{a_1} + W_2h_{w_2} - W_1h_{w_1} + (W_1 - W_2)h_{\text{condensate}}$$

From Table 3, Chapter 1 of the 2017 *ASHRAE Handbook—Fundamentals*,

$$h_{w_2} = 1083.07 \text{ Btu/lb}$$

$$h_{w_1} = 1102.57 \text{ Btu/lb}$$

$$h_{\text{condensate}} = 18.07 \text{ Btu/lb}$$

W_1 and W_2 can be calculated as

$$p_{w_1} = \phi_1 p_{ws_1} = 0.83(0.8156) = 0.678 \text{ psia}$$

$$W_1 = 0.622p_{w_1}/(p_1 - p_{w_1})$$
$$= 0.622 \times 0.678/(16 - 0.678) = 0.0275$$

$$p_{w_2} = 0.96(0.178) = 0.171 \text{ psia}$$

$$W_2 = 0.622p_{w_2}(p_2 - p_{w_2})$$
$$= 0.622 \times 0.171/(14.7 - 0.171) = 0.0073$$

Substituting into the energy equation,

$$\frac{{}_1\dot{q}_2}{m_a} = 0.24(50 - 95) + 0.0073(1083.07) - 0.0275(1102.57)$$
$$+ (0.0275 - 0.0073)18.07$$
$$= -10.8 + 7.91 - 30.3 \div 0.36 = 32.8 \text{ Btu/lb}_{\text{air}}$$

3.1.4 Heating and Humidifying Air

A device to heat and humidify moist air is shown in Figure 3-7. This process is generally required during the cold months of the year. An energy balance on the device yields

$$m_ah_1 + q + m_wh_w = m_ah_2 \tag{3-11}$$

and a mass balance on the water gives

$$m_aW_1 + m_w = m_aW_2 \tag{3-12}$$

3.1.5 Adiabatic Mixing of Two Streams of Air

A common process involved in air conditioning is the adiabatic mixing of two streams of moist air (Figure 3-8).

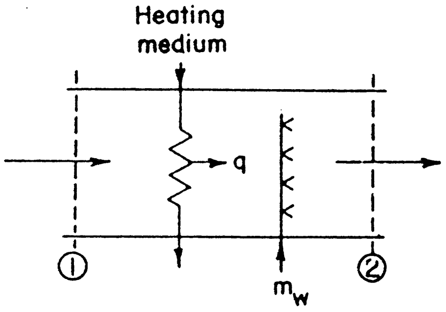


Fig. 3-7 Schematic Heating and Humidifying Device

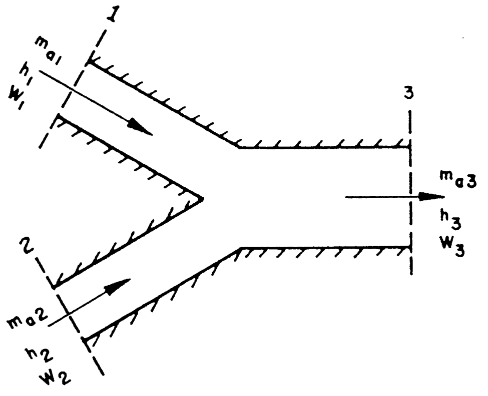


Fig. 3-8 Adiabatic Mixing of Two Streams of Moist Air

If the mixing is adiabatic, it must be governed by these three equations:

$$m_{a_1}h_1 + m_{a_2}h_2 = m_{a_3}h_3 \tag{3-13}$$

$$m_{a_1} + m_{a_2} = m_{a_3} \tag{3-14}$$

$$m_{a_1}W_1 + m_{a_2}W_2 = m_{a_3}W_3 \tag{3-15}$$

3.1.6 Adiabatic Mixing of Moist Air with Injected Water

Injecting steam or liquid water into a moist air stream to raise the humidity ratio of the moist air is a frequent air-conditioning process (Figure 3-9). If the mixing is adiabatic, the following equations apply:

$$m_a h_1 + m_w h_w = m_a h_2 \tag{3-16}$$

$$m_a W_1 + m_w = m_a W_2 \tag{3-17}$$

3.1.7 Moving Air

In all HVAC systems, a fan or blower is required to move the air. Under steady-flow conditions for the fan shown in Figure 3-10, the conservation equations are

$$m_a h_1 - m_a h_2 - W_k = 0 \tag{3-18}$$

$$W_1 = W_2 \tag{3-19}$$

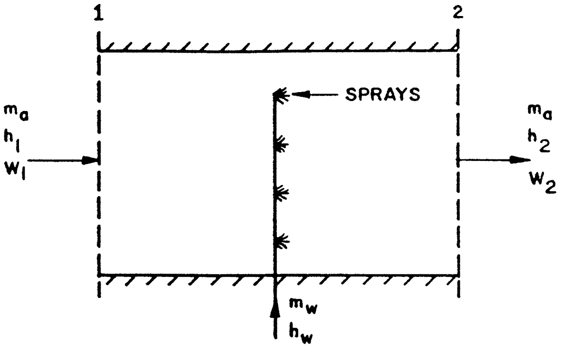


Fig. 3-9 Schematic Injection of Water into Moist Air

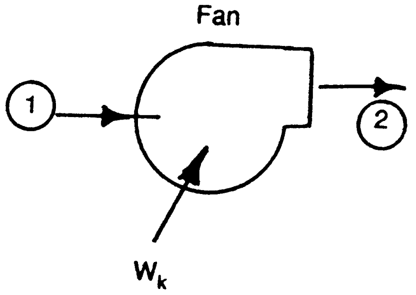


Fig. 3-10 Air Moving

3.1.8 Approximate Equations Using Volume Flow Rates

Since the specific volume of air varies appreciably with temperature, all calculations should be made with the mass of air instead of the volume. Nevertheless, volume values are required when selecting coils, fans, ducts, and other components.

In Chapter 1 of the 2017 *ASHRAE Handbook—Fundamentals*, basic calculations for air system loads, flow rates, and psychrometric representations are based on mass flow. Chapters 17 and 18 on load estimating give equations based on volumetric flow. These volumetric equations are commonly used and generally apply to most air systems.

One method of using volume while still including mass is to use volume values based on measurement at standard air conditions. ASHRAE defines one standard condition as dry air at 20°C and 101.325 kPa (68°F and 14.7 psia). Under that condition the density of dry air is about 1.204 kg/m^3 (0.075 lb/ft^3) and the specific volume is 0.83 m^3/kg (13.3 ft^3/lb). Saturated air at 15°C (59.5°F) has about the same density or specific volume. Thus, in the range at which air usually passes through the coils, fans, ducts, and other equipment, its density is close to standard and is not likely to require correction.

When the actual volumetric airflow is desired at any particular condition or point, the corresponding specific volume is obtained from the psychrometric chart and the volume at standard conditions is multiplied by the ratio of the actual specific volume to the standard value of 0.83 (13.3). To illustrate, assume the outdoor airflow rate at ASHRAE standard conditions is 470 L/s (1000 cfm). The actual outdoor air condition is 35°C (95°F) dry bulb and 23.8°C (75°F) wet bulb [v = 0.89 m^3/kg (14.3 ft^3/lb)]. The actual volume flow rate at this condition would be 470(0.89/0.83) = 500 L/s [1000(14.3/13.3) = 1080 cfm].

Air-conditioning design often requires the calculation of sensible, latent, and total energy gains as follow:

1. *Sensible heat gain corresponding to the change of dry-bulb temperature for a given airflow (standard conditions).* The sensible heat gain in watts (Btu/h) as a result of a difference in temperature Δt between the incoming air and leaving air flowing at ASHRAE standard conditions is

$$q_s = Q(1.204)(1.00 + 1.872W)\ \Delta t \tag{3-20}$$

where

q_s = sensible heat gain, W
Q = airflow, L/s
1.204 = density of standard dry air, kg/m^3
1.00 = specific heat of dry air, kJ/(kg·K)
1.872 = specific heat of water vapor, kJ/(kg·K)
W = humidity ratio, mass of water per mass of dry air, kg/kg
Δt = temperature difference, °C

In I-P units

$$q_s = Q(60)(0.075)(0.24 + 0.45W)\ \Delta t \tag{3-21}$$

where

q_s = sensible heat gain, Btu/h
Q = airflow, ft^3/min (cfm)
60 = minutes per hour
0.075 = density of dry air, lb/ft^3
0.24 = specific heat of dry air, Btu/lb·°F
0.45 = specific heat of water vapor, Btu/lb·°F
W = humidity ratio, pounds of water per pound of dry air
Δt = temperature difference, °F

Since $W \cong 0.01$ in many air-conditioning problems, the sensible heat gain may be approximated by

$$q_s \approx 1.23Q\,\Delta t \text{ in which } Q \text{ is in L/s} \quad (3\text{-}22)$$

and in I-P units

$$q_s \approx 1.10Q\,\Delta t \text{ in which } Q \text{ is in CFM} \quad (3\text{-}23)$$

2. *Latent heat gain corresponding to the change of humidity ratio* W *for given airflow (standard conditions).* The latent heat gain in watts (Btu/h) as a result of a difference in humidity ratio ΔW between the incoming and leaving air flowing at ASHRAE standard conditions is

$$q_l = (\text{L/s})(1000)(0.001204)(2500)\,\Delta W \quad (3\text{-}24)$$

and in I-P units

$$q_l = (\text{cfm})(60)(0.075)(1076)\,\Delta W \quad (3\text{-}25)$$

In Equations 3-24 and 3-25, respectively, 2500 kJ/kg (1076 Btu/lb) is the approximate energy content of the superheated water vapor at 23.8°C (75°F) (1093.95 Btu/lb), less the energy content of water at 10°C (50°F) (18.07 Btu/lb). This difference is rounded up to 1076 Btu/lb (2500 kJ/kg) in Equation 3-25. A temperature of 24°C (75°F) is a common design condition for the space and 10°C (50°F) is normal condensate temperature from cooling and dehumidifying coils. Combining the constants, the latent heat gain is

$$q_l = 3010Q\,\Delta W \quad (3\text{-}26)$$

In I-P units

$$q_l = 4840Q\,\Delta W \quad (3\text{-}27)$$

3. *Total heat gain corresponding to the change of dry-bulb temperature and humidity ratio* W *for given airflow (standard conditions).* Total heat gain in watts (Btu/h) as a result of a difference in enthalpy Δh between the incoming and leaving air flowing at ASHRAE standard conditions is

$$q = Q(1000)(0.001204)\,\Delta h \quad (3\text{-}28)$$

In I-P units

$$q = Q(60)(0.075)\,\Delta h \quad (3\text{-}29)$$

If the product of the two constants is used as a single number, the total energy exchange is

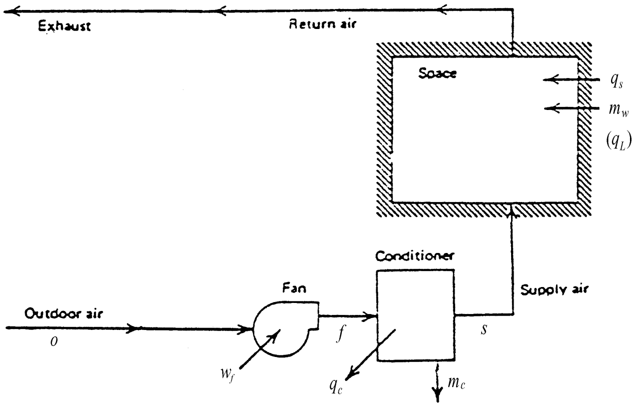
$$q = 1.204Q\,\Delta h \quad (3\text{-}30)$$

In I-P units

$$q = 4.5Q\,\Delta h \quad (3\text{-}31)$$

The values 1.23 (1.10), 3010 (4840), and 1.204 (4.5) are useful in air-conditioning calculations for an atmospheric pressure of approximately 101.3 kPa (14.7 psia) and normal temperatures and moisture ratios. For other conditions, calculations should use more precise values. For frequent computations at other altitudes, it may be desirable to calculate appropriate values in the same manner. For example, at an altitude of 1500 m (5000 ft) with a pressure of 84.15 kPa (12.2 psia), appropriate values are 1.03 (0.92), 2500 (4020), and 0.998 (3.73), respectively.

Example 3-2 A hospital operating room is being designed to use the type of HVAC system shown in the following sketch. To avoid recirculating bacteria, 100% outdoor air is used. For summer operation, the air leaving the cooling coil and supplied to the space is at 55°F, 100% relative humidity. The summer design loads are: 37,500 Btu/h (sensible) and 8,800 Btu/h (latent). The indoor design temperature is 75°F. Outdoor design conditions are: 94°F (dry bulb) and 75°F (wet bulb). Atmospheric pressure is close to sea level standard of 14.7 psia. Neglecting the effect of the fan, determine the size of cooling unit required, Btu/h, and determine the relative humidity of the air leaving the operating room, %.



Solution:

Using either the psychrometric chart or corresponding psychrometric equations, the following properties are determined:

$$h_o = 38.4 \text{ Btu/lb};\ W_o = 0.0144 \text{ lb/lb}$$

$$h_s = 23.2 \text{ Btu/lb};\ W_s = 0.0092 \text{ lb/lb}$$

Using $\quad q_s = 0.244m_a(t_r - t_s)$

$$m_a = 37500/[0.244(75 - 55)] = 7680 \text{ lb/h}$$

A water vapor balance across the space may be written

$$W_r = W_s + m_w/m_a$$

where

$$m_w = q_l / 1100 = 8800/1100 = 8 \text{ lb/h}$$

where 1100 Btu/lb approximates the enthalpy of the moisture added, causing the latent heat load. It is an approximation for 1076 Btu/lb in Equation 3-25.

Thus, $W_r = 0.0092 + 8/7680 = 0.01024$ lb/lb

At 75°F and 0.01024 lb/lb humidity ratio, the relative humidity is found to be 55%.

For the conditioner (cooling coil), the first law of thermodynamics may be written

$$m_a[h_o - h_s - (W_o - W_s)h_c] + q_c = 0$$

$$7684\ [38.4 - 23.2 - (0.0144 - 0.0092)23] + Q_c = 0$$

$$q_c = -115{,}900 \text{ Btu/h}$$

Alternatively,

$$q_c = Q[1.10\ (t_o - t_s) + 4840\ (W_o - W_s)]$$

where

$$Q = m_a 13.33/60 = 7680(13.33/60) = 1706 \text{ cfm}$$

and

$$q_c = 1706\ [1.10(94 - 55) + 4840(0.0144 - 0.0092)]$$

$$= 116{,}100 \text{ Btu/h}$$

3.2 Single-Path Systems

In the following discussions various pieces of HVAC equipment and systems will be mentioned. These are discussed later in the text when the various HVAC systems are described and analyzed. The reader can jump ahead to Chapter 11 or read Chapters 1 and 4 in the 2016 *ASHRAE Handbook—Systems and Equipment* or review items in the 1991 *ASHRAE Terminology of Heating, Ventilation, Air Conditioning, and Refrigeration.*

The simplest form of an all-air HVAC system is a single conditioner serving a single temperature control zone. The unit may be installed within, or remote from, the space it serves, and it may operate either with or without distributing ductwork. Well-designed systems can maintain temperature and humidity closely and efficiently and can be shut down when desired without affecting the operation of adjacent areas.

A single-zone system responds to only one set of space conditions. Its use is limited to situations where variations occur approximately uniformly throughout the zone served or where the load is stable; but when installed in multiple, it can handle a variety of conditions efficiently. Single-zone systems are used in such applications as small department stores, small individual shops in a shopping center, individual classrooms for a small school, and computer rooms. For example, a rooftop unit complete with a refrigeration system serving an individual space is a single-zone system. The refrigeration system, however, may be remote and serving several single-zone units in a larger installation.

A schematic of the single-zone central unit is shown in Figure 3-11. The return fan is necessary if 100% outdoor air is used for cooling purposes, but may be eliminated if air may be relieved from the space with very little pressure loss through the relief system. In general, a return air fan is needed if the resistance of the return air system (grilles and ductwork) exceeds 60 Pa (0.25 in. water gage).

Control of the single-zone system can be affected by on-off operation, varying the quantity of cooling medium providing reheat, face and bypass dampers, or a combination of these. The single-duct system with reheat satisfies variations in load by providing independent sources of heating and cooling. When a humidifier is included in the system, humidity control completely responsive to space needs is available. Since control is directly from space temperature and humidity, close regulation of the system conditions may be achieved. Single-duct systems without reheat offer cooling flexibility but cannot control summer humidity independent of temperature requirements.

3.3 Air-Volume Equations for Single-Path Systems

Basic equations for individual rooms (zones) are the same for all single-path systems. Air supplied to each room must be adequate to take care of each room's peak load conditions whether or not it occurs simultaneously in all rooms. The peak may be governed by sensible or latent room cooling loads, heating loads, outdoor air requirements, air motion, or exhaust.

Supply Air for Cooling:

$$(Q_{sRS1})^S = \frac{(q_{SR1})^S}{C_1(t_R - t_s)} \tag{3-32}$$

Supply Air for Dehumidification:

$$(Q_{sRL1})^S = \frac{(q_{LR1})^S}{C_2(W_R - W_s)} \tag{3-33}$$

Supply Air for Heating:

$$(Q_{sRS1})^W = \frac{(q_{SR1})^W}{C_1(t_s - t_R)} \tag{3-34}$$

where

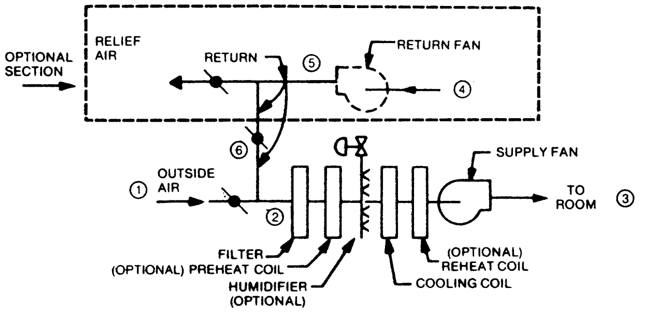


Fig. 3-11 Single-Duct System

$(Q_{sRS1})^S$ = summer room supply air volume required to satisfy peak sensible load of each room
$(Q_{sRL1})^S$ = summer room supply air volume required to satisfy peak latent load of each room
$(q_{SR1})^S$ = peak summer sensible room load for each room
$(q_{LR1})^S$ = peak summer latent room load for each room
$(Q_{sRS1})^W$ = winter room supply air volume required to satisfy peak heating load of each room
$(q_{SR1})^W$ = peak winter sensible room load (less any auxiliary heat) for each room
W_R = room humidity ratio
W_S = humidity ratio of dehumidified supply air
t_R = room air temperature
t_s = supply air temperature required to satisfy the summer or winter peak loads
C_1 = 1.23 (SI); 1.10 (I-P)
C_2 = 3010 (SI); 4840 (I-P)

Supply Air for Ventilation. Ventilation requirements, rather than sensible or latent loads, may govern when the supply air is deficient in any of the following ways:

1. If it does not contain adequate outdoor air, as determined by the outdoor ratio $X_o = Q_o/Q_s$, then, for such a room, supply air for outdoor air ventilation must be determined from the required room outdoor air Q_{oR} and

$$Q_{sRv} = Q_{oR}/X_o \qquad (3\text{-}35)$$

where
Q_{sRv} = room supply air required to satisfy ventilation requirements
Q_{oR} = minimum outdoor air required in a particular room
X_o = ratio of the system's total outdoor air to its total supply air that satisfies outdoor air requirements in most rooms

2. The supply air may not be adequate to serve as make-up for exhaust requirements in the room. In such cases, no return air comes from the room, and only conditioned supply air is assumed to be used as make-up (no supplementary ventilation supply system). This entire volume of make-up ventilation air would become an outdoor air burden to the system, in the form of a larger X_o distributed to all rooms, even though all the air supplied to this particular room is not outdoor air and

$$Q_{sRv} = Q_{sR} \qquad (3\text{-}36)$$

where
Q_{sR} = air exhausted or relieved from a room and not returned to the conditioned air system

3. If the desired rate of air change in the room is not satisfied, then supply air is determined as follows (in SI units):

$$Q_{sRv}\ (\text{L/s}) = (\text{Room volume in m}^3) \times (\text{No. of air changes/hour})/3.6 \qquad (3\text{-}37\ \text{SI})$$

and in I-P units

$$Q_{sRv}\ (\text{cfm}) = (\text{Room volume in ft}^3) \times (\text{No. of air changes/hour})/60 \qquad (3\text{-}37\ \text{I-P})$$

4. If air movement, as measured by an area index instead of an air change index, is not satisfied, or

$$Q_{sRv} = K \times [(\text{L/s})/\text{m}^2] \qquad (3\text{-}38\ \text{SI})$$

$$Q_{sRv} = K \times (\text{cfm/ft}^2) \qquad (3\text{-}38\ \text{I-P})$$

where K = a constant, greater or less than one.

Both the rate of air change and K are empirical values that vary according to designers' experiences and local building code requirements. For example, 5 air changes per hour in a room with a 12 ft (3.7 m) ceiling corresponds to 1 cfm/ft^2 (5 L/s·m^2), while the same air change rate with an 8 ft (2.4 m) ceiling is only 0.66 cfm/ft^2 (3.3 L/s·m^2). Physiologically, one may have no advantage over the other.

Case 1 is to be used when outdoor air Q_o governs, Cases 3 and 4 when air movement governs, and Case 2 when exhaust governs.

Example 3-3 A space is designed to have a summer inside temperature of 75°F and relative humidity of 50% and a winter inside temperature of 72°F and relative humidity of 25%. The summer supply air conditions are 55°F, 90% rh, while the winter supply air temperature is 110°F with a humidity ratio of 0.0065 lb/lb. The summer design loads are:

$$q_{\text{sensible},s} = 10{,}000\ \text{Btu/h}$$

$$q_{\text{latent},s} = 1500\ \text{Btu/h}$$

The winter design loads are:

$$q_{\text{sensible},w} = 12{,}000\ \text{Btu/h}$$

$$q_{\text{latent},w} = 1{,}000\ \text{Btu/h}$$

The outdoor air requirement is 80 cfm and the ratio of outdoor air to total supply air is 0.33. Determine the required supply air in CFM to satisfy summer, winter, and ventilation conditions.

Solution:

Summer: Sensible load

$$q_{\text{sensible},s} = 1.1\ \text{CFM}_{s,s}\ \Delta t$$

$$10{,}000 = 1.1 \times \text{CFM}_{s,s} \times (75 - 55)$$

$$\text{CFM}_{s,s} = 455\ \text{cfm}$$

Latent load

$$q_{\text{latent},s} = 4840 \times \text{CFM}_{s,L} \times \Delta W$$

$$W_i = 0.0093\ \text{lb/lb at 75°F, 50\% rh}$$

$$W_s = 0.0083\ \text{lb/lb at 55°F, 90\% rh}$$

$$1500 = 4840 \times \text{CFM}_{s,L} \times (0.0093 - 0.0083)$$

$$\text{CFM}_{s,l} = 310\ \text{cfm}$$

Winter: Sensible load

$$q_{\text{sensible},w} = 1.1 \times \text{CFM}_{w,s} \times \Delta t$$

$$12{,}000 = 1.1 \times \text{CFM}_{w,s} \times (110 - 72)$$

$$\text{CFM}_{w,s} = 287\ \text{cfm}$$

Latent load

$$q_{\text{latent},w} = 4840 \times \text{CFM}_{L,w} \times (\Delta W)$$

$$W_s = 0.0065 \text{ lb/lb}$$

$$W_r = 0.0042 \text{ lb/lb at } 72°\text{F, } 25\% \text{ rh}$$

$$1000 = 4840 \times \text{CFM}_{L,w} \times (0.0065 - 0.0042)$$

$$\text{CFM}_{L,w} = 90 \text{ cfm}$$

Ventilation:

$$Q_{sRV} = Q_{o,r} / X_o$$

$$Q_{sRV} = 80 / 0.33$$

$$Q_{sRV} = 242 \text{ cfm}$$

For satisfaction of all design parameters, the design volume flow rate should be selected as the maximum flow requirement of 455 cfm. This is what is required for the summer sensible design load. All of the other design parameters will be satisfied with this volume flow but would require some form of control to maintain temperature, relative humidity, and outside ventilation air quality.

3.4 Psychrometric Representation of Single-Path Systems

The operation of a single path system is illustrated in Figure 3-12. Each state point is shown with corresponding nomenclature in the cycle diagram and in the summer and winter representations. Each change in temperature Δt or humidity ratio ΔW is a result of sensible or latent heat loss or gain q_S or q_L. The cycle diagram in Figure 3-12 shows the full symbol and subscript for each state, while psychrometric charts show only the corresponding state points. This section describes the flow paths in a typical single-duct, single-zone, draw-through system. In this illustration all return air is assumed to pass from the room through a hung- ceiling return air plenum.

In Figure 3-12, supply air cfm$_S$ at the fan discharge temperature t_{sf} in the summer mode absorbs transmitted supply duct heat q_{sd} and supply air fan velocity pressure energy q_{sfvp}, thereby raising the temperature to t_s. Room supply air absorbs room sensible and latent heat q_{SR} and q_{LR} along the room sensible heat factor (SHF) line **s-R**, reaching desired room state, t_R and W_R. Room (internal) sensible loads which determine cfm$_S$ consist of

1. Ceiling transmission c_{clg} (shown) from the hung ceiling above the room, and floor transmission q_D (not shown) from the floor deck below (the q_D shown is the heat gain to the hung ceiling from the roof deck above, not the floor deck below);
2. Direct light heat emissions to the room q_{lR} (without the direct light heat emission to the plenum q_{lp});
3. Transmissions q_{tran} from other surfaces such as walls, windows, etc.;
4. Appliance heat $(q_S)_{\text{aux}}$ and occupancy heat $(q_S)_{\text{occ}}$; and
5. Infiltration load $(q_S)_{\text{inf}}$ taken here as zero, with exfiltration air cfm$_{eR}$ shown instead.

The remaining air returned from the room (CFM$_S$ – CFM$_{er}$ = CFM$_R$) passes through the ceiling plenum along line **R-rp**, absorbing some ceiling heat q_{rp} while simultaneously retransmitting some of it back to the room q_{clg}. Air is picked up by the return duct system at t_{rp}, after passing through the ceiling at average temperature t_p.

Return volume cfm$_R$ picks up return duct transmissions q_{rd} and return fan heat gains q_{rf} (including both static and velocity energy) by the time it reaches the intake plenum entrance at t_r. CFM$_r$ may be less than cfm$_R$ if leakage occurs from the ducts or through the exhaust damper from the return air system. Mixing of outdoor air, cfm$_o$ at state *o*, with final return air occurs along process line **r-o** to mixture state *m*. Total system air cfm$_s$ passes through the cooling coil releasing the total sensible heat factor, SHR$_{cc}$ line **m-cc**, terminating in state *cc*. The temperature rise through the supply fan from $(q_{sf})_{sp}$ results in t_{sf}, which completes the cycle.

The winter cycle (Figure 3-12) is similar, except for the interposition of the heating coil energy q_{hc} added in process line **m-hc**; the deletion of the cooling coil action; and the temperature drop in process line **sf-s**, resulting from duct transmission losses q_{sd} instead of gains.

3.5 Sensible Heat Factor (Sensible Heat Ratio)

The **sensible heat factor** (SHF), also called the **sensible heat ratio** (SHR), is the ratio of the sensible heat for a process to the summation of the sensible and latent heat for the process. The sum of sensible and latent heat is also called the **total heat**. On the ASHRAE Psychrometric Chart, values of sensible heat factors are given on the protractor as $\Delta H_s/\Delta H_T$, and they may be used to establish the process line for changes in the condition of the air across either the room or the conditioner on the psychrometric chart.

The supply air to a conditioned space must have the capacity to offset simultaneously both the room sensible and room latent heat loads. The room and the supply air conditions to the space may be plotted on the psychrometric chart and connected with a straight line called the room sensible heat factor line.

As air passes through the cooling coil of an air conditioner, the sensible heat factor line represents the simultaneous cooling and dehumidifying that occurs. If the cooling process involves removing only sensible and no latent heat, i.e., no moisture removal, the sensible heat factor line is horizontal and the sensible heat factor is 1.0. If 50% is sensible and 50% latent, the SHF is 0.5.

Example 3-4 An air-conditioned space has a summer sensible design heat load of 100,000 Btu/h, a summer design latent load of 20,000 Btu/h, and is maintained at 75°F and 55% rh. Conditioned air leaves the apparatus and enters the room at 58°F. The outdoor air is at 96°F, 77°F wet bulb, and is 20% of total flow to the conditioning apparatus.

(a) Draw and label the schematic flow diagram for the system.

(b) Complete a table of properties and flow rates at various locations in the system.

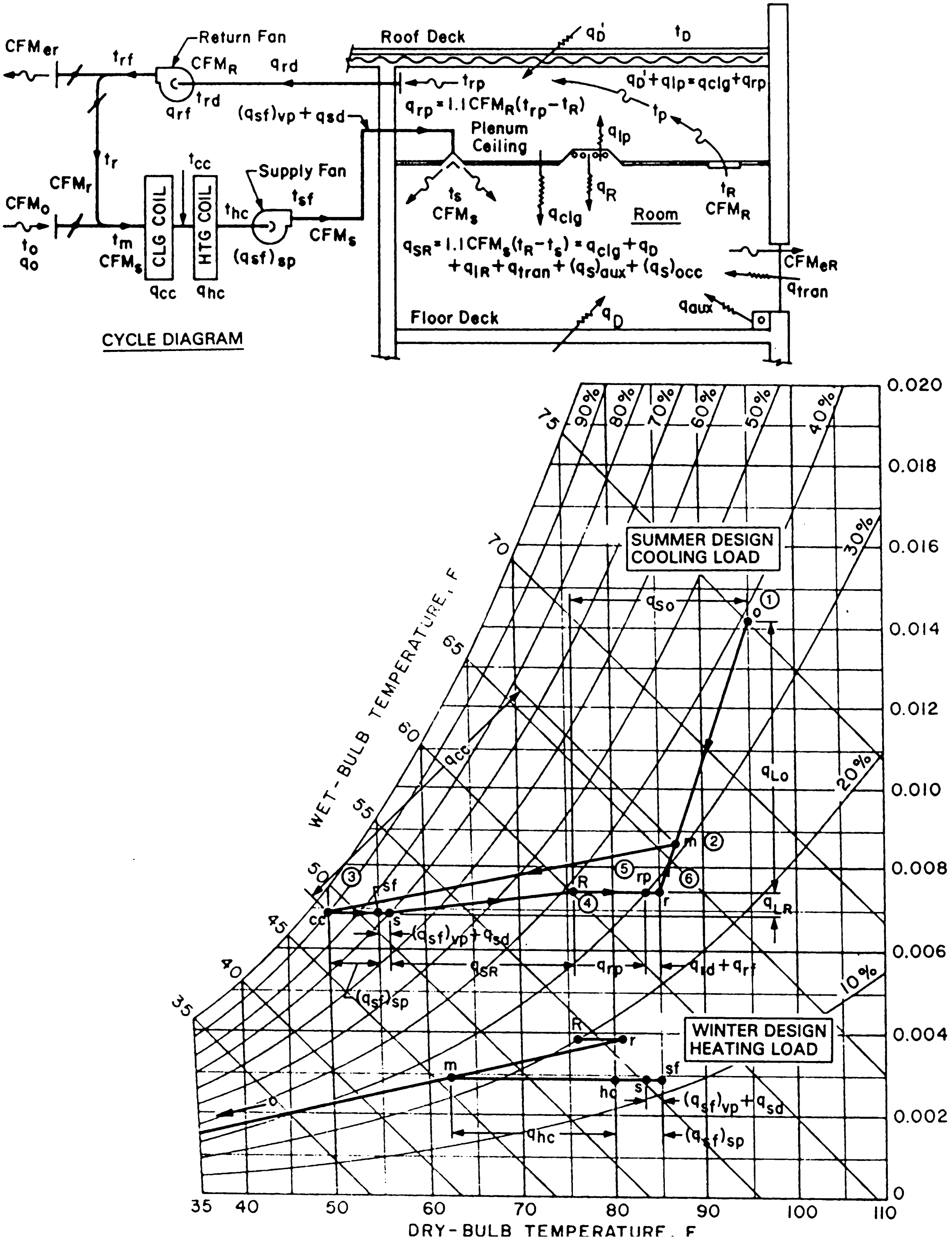


Fig. 3-12 Single-Duct, Single-Zone Cycle and Psychrometric Chart

(c) Show all the processes on the psychrometric coordinates.

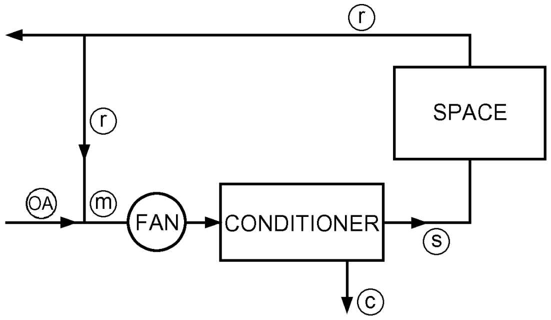
(d) Determine the size of cooling unit needed in Btu/h and tons.

(e) What percent of the required cooling is for sensible cooling and what percent is for dehumidification?

(f) What percent of the required cooling is due to the outdoor air load?

Solution:

(a)



(b)

Point	*t*, °F	*ϕ*, %	WB, °F	h, Btu/lb	*W*, lb/lb	v_a, ft³/lb	*m*, lb/h	CFM	SCFM
OA	**96**	43	**77**	40.2	0.0157	14.36	4822	1151	1071
r	**75**	**55**	64	29.2	0.0102	13.7	19,286	4403	4285
m	79.1	52	66.8	31.4	0.0113	13.83	24,108	5557	5356
s	**58**	89	56.1	24	0.00944	13.24	24,108	5320	5356

Note: 1. Assume that the thermal conditions leaving the fan are the same as those entering the fan.
2. The properties in bold are given values.

The psychrometric properties for OA and r are read directly from the psychrometric chart.

$$q_{\text{sensible}} = m \times c_{pa} \times (t_r - t_s)$$

$$100{,}000 = m \times 0.244 \text{ Btu/lb°F} \times (75 - 58)$$

$$m = 24{,}108 \text{ lb/h}$$

Mass flow rate for outdoor air is

$$24{,}108 \times 0.2 = 4822 \text{ lb/h}$$

Mass flow rate of return air mixed with outdoor air is

$$24{,}108 - 4822 = 19{,}286 \text{ lb/h}$$

For the mixed air state, *m*:

Energy balance at *m* $\quad 0.8\,(29.2) + 0.2\,(40.2) = 31.4$ Btu/lb

Moisture balance at *m* $\quad 0.8\,(0.0102) + 0.2\,(0.015) = 0.0113$ lb/lb

The other conditions at *m* come from the psychrometric chart.

A moisture balance on the space provides W_s.

$$W_s = W_r - \frac{m_w}{m_a} = 0.0102 - \frac{20{,}000/1100 \text{ Btu/lb}}{24{,}108}$$

$$W_s = 0.0102 - 0.000\,7542$$

$$W_s = 0.00944 \text{ lb/lb}$$

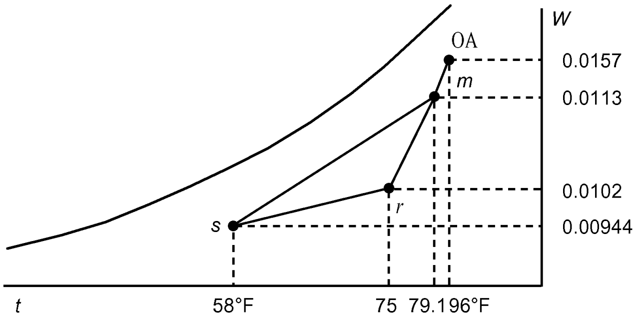
With W_s and t_s the other properties are found from the psychrometric chart from

$$m = \text{cfm}/\nu \Rightarrow \text{cfm} = m \times \nu/60$$

The scfm values at the four states are found from the same equation; however, using

$$\nu = \nu_s = 13.33 \text{ ft}^3\text{/lb}.$$

(c)



(d)

$$q = m\,[h_m - h_s - (w_m - w_s)\,h_c]$$

$$q = 24{,}108\,[31.4 - 24 - (0.0113 - 0.00944)\,26]$$

$$q = 177{,}240 \text{ Btu/h or 14.8 tons}$$

(e)

$$\%\text{ sensible} = \frac{24{,}108(0.244)(79.1 - 58)}{177{,}240} = 0.70$$

% latent = 0.30, 30%

(f)

$$\%\text{ Due to OA} = \frac{1071[1.1(96-75) + 4840(0.0157 - 0.0102)]}{177{,}240}$$

$$= 0.30,\ 30\%$$

3.6 Problems

3.1 One of the many methods used for drying air is to cool the air below the dew point so that condensation or freezing of the moisture takes place. To what temperature must atmospheric air be cooled in order to have a humidity ratio of 0.000017 lb/lb (17 mg/kg)? To what temperature must this air be cooled if its pressure is 10 atm?

3.2 One method of removing moisture from atmospheric air is to cool the air so that the moisture condenses or freezes out. Suppose an experiment requires a humidity ratio of 0.0001. To what temperature must the air be cooled at a pressure of 0.1 kPa in order to achieve this humidity?

3.3 A room of dimensions 4 m by 6 m by 2.4 m contains an air-water vapor mixture at a total pressure of 100 kPa and a

temperature of 25°C. The partial pressure of the water vapor is 1.4 kPa. Calculate

(a) humidity ratio [Ans: 0.0088 kg/kg_{air}
(b) dew point [Ans: 11.8°C]
(c) total mass of water vapor in the room [Ans: 0.584 kg]

3.4 The air conditions at the intake of an air compressor are 70°F (21.1°C), 50% rh, and 14.7 psia (101.3 kPa). The air is compressed to 50 psia (344.7 kPa), then sent to an intercooler. If condensation of water vapor from the air is to be prevented, what is the lowest temperature to which the air can be cooled in the intercooler?

3.5 Humid air enters a dehumidifier with an enthalpy of 21.6 Btu/lb_m of dry air and 1100 Btu/lb_m of water vapor. There are 0.02 lb_m of vapor per pound of dry air at entrance and 0.009 lb_m of vapor per pound of dry air at exit. The dry air at exit has an enthalpy of 13.2 Btu/lb_m, and the vapor at exit has an enthalpy of 1085 Btu/lb_m. Condensate leaves with an enthalpy of 22 Btu/lb_m. The rate of flow of dry air is 287 lb_m/min. Determine

(a) the amount of moisture removed from the air (lb_m/min)
(b) the rate of heat removal required
[Ans: 3.16 lb/min, −5860 Btu/min]

3.6 Air is supplied to a room from the outside, where the temperature is 20°F (–6.7°C) and the relative humidity is 60%. The room is to be maintained at 70°F (21.1°C) and 50% rh. How many pounds of water must be supplied per pound of air supplied to the room?

3.7 Air is heated to 80°F (26.7°C) without adding water, from 60°F (15.6°C) dry-bulb and 50°F (10°C) wet-bulb temperature. Use the psychrometric chart to find

(a) relative humidity of the original mixture
(b) original dew-point temperature
(c) original humidity ratio
(d) initial enthalpy
(e) final enthalpy
(f) the heat added
(g) final relative humidity

3.8 Saturated air at 40°F (4.4°C) is first preheated and then saturated adiabatically. This saturated air is then heated to a final condition of 105°F (40.6°C) and 28% rh. To what temperature must the air initially be heated in the preheat coil?
[Ans: 101°F (37.8°C)]

3.9 Atmospheric air at 100°F (37.8°C) dry-bulb and 65°F (18.3°C) wet-bulb temperature is humidified adiabatically with steam. The supply steam contains 10% moisture and is at 16 psia (110.3 kPa). What is the dry-bulb temperature of the humidified air if enough steam is added to bring the air to 70% rh?

3.10 The conditions on a day in New Orleans, Louisiana, are 95°F (35°C) dry-bulb and 80°F (26.7°C) wet-bulb temperature. In Tucson, Arizona, the air conditions are 105°F (40.6°C) dry-bulb and 72°F (22.2°C) wet-bulb temperature. What is the lowest air temperature that could theoretically be attained in an evaporative cooler at these summer conditions in these two cities?

3.11 Air at 29.92 in. Hg enters an adiabatic saturator at 80°F dry-bulb and 66°F wet-bulb temperature. Water is supplied at 66°F. Find (without using the psychrometric chart) the humidity ratio, degree of saturation, enthalpy, and specific volume of entering air.

[Ans: 0.0104 lb/lb air, $\mu = 0.47$, h = 30.7 Btu/lb air, $v =$ 13.83 ft^3/lb air]

3.12 An air-water vapor mixture enters an air-conditioning unit at a pressure of 150 kPa, a temperature of 30°C, and a relative humidity of 80%. The mass flow of dry air entering is 1 kg/s. The air-vapor mixture leaves the air-conditioning unit at 125 kPa, 10°C, 100% rh. The moisture condensed leaves at 10°C. Determine the heat transfer rate for the process.

3.13 Air at 40°C, 300 kPa, with a relative humidity of 35% is to be expanded in a reversible adiabatic nozzle. How low a pressure can the gas be expanded to if no condensation is to take place? What is the exit velocity at this condition?

3.14 By using basic definitions and Dalton's Law of partial pressure, show that

$$v = R_a T/(p - p_w)$$

3.15 In an air-conditioning unit, 71,000 cfm at 80°F dry bulb, 60% rh, and standard atmospheric pressure, enter the unit. The air leaves the unit at 57°F dry bulb and 90% relative humidity. Calculate the following:

(a) cooling capacity of the air-conditioning unit, in Btu/h
(b) rate of water removal from the unit
(c) sensible heat load on the conditioner, in Btu/h
(d) latent heat load on the conditioner, in Btu/h
(e) the dew point of the air leaving the conditioner

3.16 Four pounds of air at 80°F (26.7°C) dry bulb and 50% rh are mixed with one pound of air at 50°F (15.6°C) and 50% rh. Determine

(a) relative humidity of the mixture
(b) dew-point temperature of the mixture
[Ans: 52%, 55.5°F (13°C)]

3.17 Air is compressed from 85°F, 60% rh, 14.7 psia to 60 psia and then cooled in an intercooler before entering a second stage of compression. What is the minimum temperature to which the air can be cooled so that condensation does not take place?

3.18 An air-water vapor mixture flowing at a rate of 4000 cfm (1890 L/s) enters a perfect refrigeration coil at 84°F (28.9°C) and 70°F (21.1°C) wet-bulb temperature. The air leaves the coil at 53°F (11.7°C). How many Btu/h of refrigeration are required?

3.19 Air at 40°F dry bulb and 35°F wet bulb is mixed with air at 100°F dry bulb and 77°F wet bulb in the ratio of 2 lb of cool air to 1 lb of warm air. Compute the resultant humidity ratio and enthalpy of the mixed air. [Ans: 0.007 lb/lb, 22.2 Btu/lb]

3.20 Outdoor air at 90°F (32.2°C) and 78°F (25.6°C) wet bulb is mixed with return air at 75°F (23.9°C) and 52% rh. There are 1000 lb (454 kg) of outdoor air for every 5000 lb (2265 kg) of return air. What are the dry- and wet-bulb temperatures for the mixed airstream?

3.21 In a mixing process of two streams of air, 10,000 cfm of air at 75°F and 50% rh mix with 4000 cfm of air at 98°F dry-bulb and 78°F wet-bulb temperature. Calculate the following conditions after mixing at atmospheric pressure:

(a) dry-bulb temperature
(b) humidity ratio
(c) relative humidity
(d) enthalpy
(e) dew-point temperature

3.22 Solve the following:

(a) Determine the humidity ratio and relative humidity of an air-water vapor mixture that has a dry-bulb temperature of 30°C, an adiabatic saturation temperature of 25°C, and a pressure of 100 kPa.
(b) Use the psychrometric chart to determine the humidity ratio and relative humidity of an air-water vapor mixture that has a dry-bulb temperature of 30°C, a wet-bulb temperature of 25°C, and a pressure of 100 kPa.
[Ans: 0.0183 kg/kg, 67%, 0.018 kg/kg, 67%]

3.23 An air-water vapor mixture at 100 kPa, 35°C, and 70% rh is contained in a 0.5 m^3 closed tank. The tank is cooled until the water just begins to condense. Determine the temperature at which condensation begins and the heat transfer for the process.

3.24 A room is to be maintained at 76°F and 40% rh. Supply air at 39°F is to absorb 100,000 Btu sensible heat and 35 lb of moisture per hour. Assume the moisture has an enthalpy of 1100 Btu/lb. How many pounds of dry air per hour are required? What should the dew-point temperature and relative humidity of the supply air be? Assume $h_w = 1100\ \text{Btu/lb}_m$.
[Ans: 11,260 lb/h, 36°F, 90%]

3.25 Moist air enters a chamber at 40°F dry-bulb and 36°F wet-bulb temperature at a rate of 3000 cfm. In passing through the chamber, the air absorbs sensible heat at a rate of 116,000 Btu/h and picks up 83 lb/h of saturated steam at 230°F. Determine the dry-bulb and wet-bulb temperatures of the leaving air. [Ans: 74°F db, 64°F wb]

3.26 In an auditorium maintained at a temperature not to exceed 77°F, and a relative humidity not to exceed 55%, a sensible-heat load of 350,000 Btu and 1,000,000 grains of moisture per hour must be removed. Air is supplied to the auditorium at 67°F.

(a) How much air must be supplied, in lb/h?
(b) What is the dew-point temperature of the entering air, and what is its relative humidity?
(c) How much latent heat is picked up in the auditorium?
(d) What is the sensible heat ratio?

3.27 A meeting hall is maintained at 75°F dry bulb and 65°F wet bulb. The barometric pressure is 29.92 in. Hg. The space has a load of 200,000 Btu/h sensible, and 200,000 Btu/h latent. The temperature of the supply air to the space cannot be lower than 65°F dry bulb.

(a) How much air must be supplied, in lb/h?
(b) What is the required wet-bulb temperature of the supply air?
(c) What is the sensible heat ratio?
[Ans: 81,970 lb/h, 58°F, 0.5]

3.28 A structure to be air conditioned has a sensible heat load of 20,000 Btu/h at a time when the total load is 100,000 Btu/h. If the inside state is to be at 80°F, 50% rh, is it possible to meet the load conditions by supplying air to the room at 100°F and 60% rh? If not, discuss the direction in which the inside state would be expected to move if such air were supplied.

3.29 A flow rate of 30,000 lb/h of conditioned air at 60°F and 85% rh is added to a space that has a sensible load of 120,000 Btu/h and a latent load of 30,000 Btu/h.

(a) What are the dry- and wet-bulb temperatures in the space?
(b) If a mixture of 50% return air and 50% outdoor air at 98°F dry bulb and 77°F wet bulb enters the air conditioner, what is the refrigeration load?

3.30 An air-water vapor mixture enters a heater-humidifier unit at 5°C, 100 kPa, 50% rh. The flow rate of dry air is 0.1 kg/s. Liquid water at 10°C is sprayed into the mixture at the rate of 0.0022 kg/s. The mixture leaves the unit at 30°C, 100 kPa. Calculate

(a) the relative humidity at the outlet
(b) the rate of heat transfer to the unit
[Ans: 91%, 7.94 kW]

3.31 A room is being maintained at 75°F and 50% rh. The outdoor air conditions are 40°F and 50% rh at this time. Return air from the room is cooled and dehumidified by mixing it with fresh ventilation air from the outside. The total airflow to the room is 60% outdoor and 40% return air by mass. Determine the temperature, relative humidity, and humidity content of the mixed air going to the room.
[Ans: 54.5°F, 58%, 0.00524 lb/lb]

3.32 A room with a sensible load of 20,000 Btu/h is maintained at 75°F and 50% rh. Outdoor air at 95°F and 80°F wet bulb was mixed with the room return air. The outdoor air, which is mixed, is 25% by mass of the total flow going to the conditioner. This air is then cooled and dehumidified by a coil and leaves the coil saturated at 50°F, which is on the condition line for the room. The air is then mixed with some room return

air so that the temperature of the air entering the room is at 60°F. Find the following:

(a) the air-conditioning processes on the psychrometric chart
(b) ratio of latent to sensible load
(c) airflow rate
(d) the percent by mass of room return air mixed with air leaving the cooling coil

3.33 An air-water vapor mixture at 14.7 psia (101.5 kPa), 85°F (29.4°C), and 50% rh is contained within a 15 ft^3 (0.425 m^3) tank. At what temperature will condensation begin? If the tank and mixture are cooled an additional 15°F (8.3°C), how much water will condense from the mixture?

3.34 Air flowing at 1000 cfm and at 14.7 psia, 90°F, and 60% rh passes over a coil with a mean surface temperature of 40°F. A spray on the coil assures that the leaving air is saturated at the coil temperature. What is the required cooling capacity of the coil? [Ans: 9.3 tons]

3.35 An air-vapor mixture at 100°F (37.8°C) dry bulb contains 0.02 lb water vapor per pound of dry air (20 g/kg). The barometric pressure is 28.561 in. Hg (96.7 kPa). Calculate the relative humidity, dew-point temperature, and degree of saturation.

3.36 Air enters a space at 20°F and 80% rh. Within the space, sensible heat is added at the rate of 45,000 Btu/h and latent heat is added at the rate of 20,000 Btu/h. The conditions to be maintained inside the space are 50°F and 75% rh. What must the air exhaust rate (lb/h) from the space be to maintain a 50°F temperature? What must be the air exhaust rate (lb/h) from the space to maintain a 75% rh? Discuss the difference.

3.37 Moist air at a low pressure of 11 psia is flowing through a duct at a low velocity of 200 fpm. The duct is 1 ft in diameter and has negligible heat transfer to the surroundings. The dry-bulb temperature is 85°F and the wet-bulb temperature is 70°F. Calculate the following:

(a) humidity ratio, lb/lb
(b) dew-point temperature, °F
(c) relative humidity, %
[Ans: 0.0177 lb/lb, 52°F, 32%]

3.38 If an air compressor takes in moist air (at about 90% rh) at room temperature and pressure and compresses this to 120 psig (827 kPa) (and slightly higher temperature), would you expect some condensation to occur? Why? If yes, where would the condensation form? How would you remove it?

3.39 Does a sling psychrometer give an accurate reading of the adiabatic saturation temperature? Explain.

3.40 An air processor handles 2000 cfm of air with initial conditions of 50°F and 50% rh. The air is heated with a finned heat exchanger with 78 ft^2 of heat transfer surface area and a *UA* value of 210 Btu/h·°F.

Also, a steam spray system adds moisture to the air from saturated steam at 16 psia. The outlet air is at 100°F and 50% rh. Do the following:

(a) Show the processes on the psychrometric chart.
(b) Calculate the mass flow rate, lb/min.
(c) Calculate the pounds per minute of steam required.
(d) Calculate the heat added by the coil, Btu/min.
[Ans: 155 lb/min, 2.65 lb/min, 1900 Btu/min]

3.41 At an altitude of 5000 ft (1500 m), a sling psychrometer reads 80°F (26.7°C) and 67°F (19.4°C) wet bulb. Determine correct values of relative humidity and enthalpy from the chart. Compare these to the corresponding values for the same readings at sea level.

3.42 The average person gives off sensible heat at the rate of 250 Btu/h and perspires and respires about 0.27 lb/h of moisture. Estimate the sensible and latent load for a room with 25 people in it (the lights give off 9000 Btu/h). If the room conditions are to be 78°F and 50% rh, what flow rate of air would be required if the supply air came in at 63°F? What would be the supply air relative humidity?

3.43 A space in an industrial building has a winter sensible heat loss of 200,000 Btu/h and a negligible latent heat load (latent losses to outside are made up by latent gains within the space). The space is to be maintained precisely at 75°F and 50% rh. Due to the nature of the process, 100% outdoor air is required for ventilation. The outdoor air conditions can be taken as saturated air at 20°F. The amount of ventilation air is 7000 scfm and the air is to be preheated, humidified with an adiabatic saturator to the desired humidity, and then reheated. The temperature out of the adiabatic saturator is to be maintained at 60°F dry bulb. Determine the following:

(a) temperature of air entering the space to be heated, °F
(b) heat supplied to preheat coil, Btu/h
(c) heat supplied to reheat coil, Btu/h
(d) amount of water required for humidification, gpm

3.44 Using the SI psychrometric chart at standard atmospheric pressure, find

(a) dew point and humidity ratio for air at 28°C dry bulb and 22°C wet bulb [Ans: 19.5°C, 0.014 kg/kg]
(b) enthalpy and specific volume [Ans: 64.7 kJ/kg, 0.87 m^3/kg]

3.45 Using the SI chart, find

(a) moisture that must be removed in cooling air from 24°C dry bulb, 21°C wet bulb to 13°C dry bulb, saturated
(b) total, sensible, and latent heat removal for the process

3.46 An air-conditioned space has a sensible heat load of 200,000 Btu/h, a latent load of 50,000 Btu/h, and is maintained at 78°F dry bulb and 60% rh. On a mass basis, 25% outdoor air is mixed with return air. Outdoor air is at 95°F dry bulb and 76°F wet bulb. Conditioned air leaves the apparatus and enters the room at 60°F dry bulb. The fan must produce a

pressure increase of 3.5 in. water to overcome the system pressure loss. Fan efficiency is estimated as 55%.

(a) Draw and label the schematic flow diagram for the complete system. (Hint: See Fig. 3-1)
(b) Complete the table below.
(c) Plot and draw all processes on a psychrometric chart.
(d) Specify the fan size, scfm, and fan motor rating, HP.
(e) Determine the size refrigeration unit needed, in Btu/h and tons.
(f) What percent of the required refrigeration is for (1) sensible cooling and (2) for dehumidification?
(g) What percent of the required refrigeration is due to the outdoor air load?

Point	**Dry Bulb t, °F**	**ϕ, %**	**Enthalpy h, Btu/lb**	**W, lb/lb**	**m_a, lb/h**	**scfm**	**CFM**
OA							
r							
m							
f							
s							

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Chapter 4

DESIGN CONDITIONS

This chapter covers the selection, specification, and determination of the indoor and outdoor environmental conditions that are to be expected at "design time" or the conditions that will govern the sizing of the heating, cooling, and ventilating equipment. Additional details related to design conditions are provided in Chapters 9, 10, and 14 of the 2017 *ASHRAE Handbook—Fundamentals*.

4.1 Indoor Design Conditions

4.1.1 Physiological Principles

A principal purpose of HVAC is to provide conditions for human comfort, "that condition of mind that expresses satisfaction with the thermal environment" (ASHRAE Standard 55). This definition leaves open what is meant by "condition of mind" or "satisfaction," but it correctly emphasizes that judgment of comfort is a cognitive process involving many inputs influenced by physical, physiological, psychological, and other processes. The conscious mind appears to reach conclusions about thermal comfort and discomfort from direct temperature and moisture sensations from the skin, deep body temperatures, and the effects necessary to regulate body temperatures. Surprisingly, although climates, living conditions, and cultures differ widely throughout the world, the temperatures that people choose for comfort under similar conditions of clothing, activity, humidity, and air movement have been found to be very similar. Definitions of comfort do vary. Comfort encompasses perception of the environment (e.g., hot, cold, noisy) and a value rating of affective implications (e.g., too high, too cold). Acceptability is the foundation of a number of standards covering thermal comfort and acoustics.

Operational definitions of health and discomfort are controversial. However, the most generally accepted definition is that in the constitution of the World Health Organization (WHO): "Health is a state of complete physical, mental, and social well-being and not merely the absence of disease or infirmity." Concern about the health effects associated with indoor air dates back several hundred years, and it has increased dramatically in recent decades, particularly since the energy crisis in the early 1970s. This attention was partially the result of increased reporting by building occupants about poor health associated with exposure to indoor air. Since then, two types of diseases associated with exposure to indoor air have been identified: **sick building syndrome (SBS)** and **building-related illness (BRI)**.

SBS describes a number of adverse health symptoms related to occupancy in a "sick" building, including mucosal irritation, fatigue, headache, and, occasionally, lower respiratory symptoms and nausea. Sick building syndrome is characterized by an absence of routine physical signs and clinical laboratory abnormalities. The term *nonspecific* is sometimes used to imply that the pattern of symptoms reported by afflicted building occupants is not consistent with the pattern of symptoms for a particular disease. Additional symptoms can include nosebleeds, chest tightness, and fever.

Building-related illnesses, in contrast, have a known origin, may have a different set of symptoms, and are often accompanied by physical signs and abnormalities that can be clinically identified with laboratory measurements. For example, hypersensitivity illnesses, including humidifier fever, asthma, and allergic rhinitis, are caused by individual sensitization to bioaerosols.

The thermal environment affects human health in that it affects body temperature regulation and heat exchange with the environment. In the normal, healthy, resting adult, internal or core body temperatures are very stable, with variations seldom exceeding 1°F. The internal temperature of a resting adult, measured orally, averages about 98.6°F; measured rectally, it is about 1°F higher. In contrast, skin temperature is basically unregulated and can (depending on environmental temperature) vary from about 88°F to 96.8°F in normal environments and activities.

To design an environmental control system that is effective for comfort and health, the engineer must understand physiological principles. In its broadest sense, the term air conditioning implies control of any or all of the physical and chemical qualities of air. Herein, the definition of air conditioning pertains only to those conditions of air relating to health and comfort requirements of the occupants of the conditioned space.

Significant variations in the percentage composition of the normal constituents of air may make it unfit for human use. The presence of foreign materials classified as contaminants may also make air unfit. Air conditioning can control most climatological environmental factors for the service and comfort of people.

The objective of a comfort air-conditioning system is to provide a comfortable environment for the occupants of residential or commercial buildings. A comfortable environment is created by simultaneously controlling temperature, humidity, air cleanliness, and air distribution within the occupant's vicinity. These factors include mean radiant temperature, as well as air temperature, odor control, and control of the proper acoustic level within the occupant's vicinity.

Both the air and surfaces of the enclosure surrounding the occupant are sinks for the metabolic heat emitted by the occupant (Figure 4-1). Air circulates around the occupant and the surfaces. The occupant also exchanges radiant heat with the surrounding surfaces (e.g., glass and outside walls). Air is brought into motion within a given space thermally or by mechanical forces.

Whether nude or clothed, humans feel comfortable at a mean skin temperature of 91.5°F (33°C). The range of skin temperature within which no discomfort is experienced is about ±2.5°F (±1.4°C). The necessary criteria, indices, and standards for use where human occupancy is concerned are given in Chapter 9, "Thermal Comfort" and in Chapter 10, "Indoor Environmental Health," of the 2017 *ASHRAE Handbook—Fundamentals*.

The environmental indices used to evaluate the sensation of comfort for the human body are classified as direct, rationally derived, and empirical indices, which include the following factors:

Direct Indices
- Dry-bulb temperature
- Dew-point temperature
- Wet-bulb temperature
- Relative humidity
- Air movement

Rationally Derived Indices
- Mean radiant temperature
- Operative temperature
- Humid operative temperature
- Heat stress index
- Index of skin wettedness

Empirical Indices
- Effective temperature
- Black globe temperature
- Corrected effective temperature
- Wet-bulb-globe temperature index
- Wind chill index

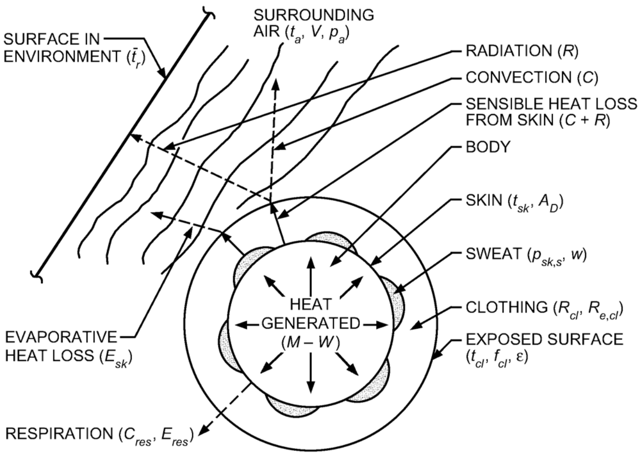


Fig. 4-1 Cylindrical Model of Body's Interaction with Environment.

(Figure 1, Chapter 9, 2017 ASHRAE Handbook—Fundamentals)

The **mean radiant temperature** $\bar{T}_r$ is a key variable in thermal calculations for the human body. It is the uniform temperature of an imaginary enclosure in which radiant heat transfer from the human body equals the radiant heat transfer in the actual uniform enclosure. It is calculated from the measured temperature of surrounding walls and surfaces and their positions with respect to the person. For most building surfaces, the emittance is high and the following equation can be used:

$$\bar{T}_r^4 = T_1^4 F_{p-1} + T_2^4 F_{p-2} + \dots T_N^4 F_{p-N}$$

where

$\bar{T}_r$ = mean radiant temperature, °R
T_N = surface temperature of surface N, °R
F_{p-N} = angle factor between a person and surface N

(See Chapter 8 in 2013 ASHRAE Handbook—Fundamentals*)*

A simplification is often used for many HVAC applications:

$$\text{MRT} = \bar{T}_r = \frac{A_1 t_1 + A_2 t_2 + \dots A_N t_N}{A_1 + A_2 + \dots A_N}$$

where

MRT = approximate mean radiant temperature for surface temperatures not significantly different from each other, °F
t_N = surface temperature of surface N, °F
A_N = area of surface N

The **operative temperature** is the uniform temperature of a radiantly black enclosure in which an occupant exchanges the same amount of heat by radiation plus convection as in the actual nonuniform environment. Numerically, operative temperature is the average, weighted by respective heat-transfer coefficients, of the air and mean radiant temperatures. At air speeds of 0.4 m/s (80 fpm) or less and mean radiant temperature less than 50°C (120°F), operative temperature is approximately the simple average of the air and mean radiant temperatures:

$$t_o = \frac{\text{MRT} + t_a}{2}$$

Physiologists recognize that sensations of comfort and temperature may have different physiological and physical bases, thus each type should be considered separately. This dichotomy was recognized in ANSI/ASHRAE Standard 55, where thermal comfort is defined as *"that state of mind which expresses satisfaction with the thermal environment."* In contrast, most current predictive charts are based on comfort defined as a sensation *"that is neither slightly warm nor slightly cool."*

Most research on comfort has been limited to lightly clothed, sedentary people. This research has proven sound since about 90% of people's indoor occupation and leisure time is spent at or near the sedentary activity level. The predictive methods to be described are all believed to be accurate depending upon the limitations stated. During physical activity, a change occurs in a person's physiology. Physiological

thermal neutrality in the sedentary sense does not exist. Some form of thermal regulation is always occurring during the sedentary condition. The same skin and body temperatures, if used as indices of comfort, prove false during moderate to heavy activity.

The main purpose of the HVAC system is to maintain comfortable indoor conditions for occupants. However, the purpose of load calculations is to obtain data for sizing the system components. In most cases, the system will rarely be set to operate at design conditions. Therefore, the use and occupancy of the space is a general consideration from the design temperature point of view. Later, when the energy requirements of the building are computed, the actual conditions in the space and outdoor environment must be considered.

The indoor design temperature should be selected at the lower end of the acceptable temperature range so that the heating equipment will not be oversized. Even properly sized equipment usually operates under partial load, at reduced efficiency, most of the time; therefore, any oversizing aggravates this condition and lowers the overall system efficiency. A maximum design dry-bulb temperature of 72°F is recommended for most occupancies. The indoor design value of relative humidity should be compatible with a healthful environment and the thermal and moisture integrity of the building envelope. A maximum relative humidity of 30% is recommended for most situations.

The conscious mind appears to reach conclusions about thermal comfort and discomfort from direct temperature and moisture sensations from the skin, deep body temperatures, and the efforts necessary to regulate body temperatures. In general, comfort occurs when body temperatures are held within narrow ranges, skin moisture is low, and the physiological effort of regulation is minimized.

The ASHRAE thermal sensation scale (Y), with its numerical representation, is

+3	hot
+2	warm
+1	slightly warm
0	neutral
−1	slightly cool
−2	cool
−3	cold

Experience indicates that women are more sensitive to temperature and less sensitive to humidity than men. However, in general, about a 5.4°F change in temperature or a 0.44 psi change in water vapor pressure is necessary to change a thermal sensation vote by one unit or temperature category.

4.1.2 Metabolic Rate

In choosing optimal conditions for comfort and health, the energy expended during the course of routine physical activities must be known, since body heat production increases in proportion to exercise intensity. Table 4-1 provides metabolic rates for various activities, on a per unit body surface area basis. The most useful measure of body surface area, proposed by DuBois, is described by

$$A_D = 0.108 m^{0.425} l^{0.725} \tag{4-1}$$

where

A_D = body surface area, ft^2
m = mass of body, lb
l = height of body, in.

An average-sized male has a mass of 70 kg (154 lb) and a height of 1.73 m (5 ft, 8 in.), so his body surface area is 1.83 m^2 (19.7 ft^2).

In choosing optimal conditions for comfort and health, the rate of work done during routine physical activities must be

Table 4-1 Typical Heat Generation Rates

(Table 4, Chapter 9, 2017 ASHRAE Handbook—Fundamentals)

	Btu/h·ft^2	met[a]
Resting		
Sleeping	13	0.7
Reclining	15	0.8
Seated, quiet	18	1.0
Standing, relaxed	22	1.2
Walking (on level surface)		
2.9 ft/s (2 mph)	37	2.0
4.4 ft/s (3 mph)	48	2.6
5.9 ft/s (4 mph)	70	3.8
Office Activities		
Reading, seated	18	1.0
Writing	18	1.0
Typing	20	1.1
Filing, seated	22	1.2
Filing, standing	26	1.4
Walking about	31	1.7
Lifting/packing	39	2.1
Driving/Flying		
Car	18 to 37	1.0 to 2.0
Aircraft, routine	22	1.2
Aircraft, instrument landing	33	1.8
Aircraft, combat	44	2.4
Heavy vehicle	59	3.2
Miscellaneous Occupational Activities		
Cooking	29 to 37	1.6 to 2.0
Housecleaning	37 to 63	2.0 to 3.4
Seated, heavy limb movement	41	2.2
Machine work		
sawing (table saw)	33	1.8
light (electrical industry)	37 to 44	2.0 to 2.4
heavy	74	4.0
Handling 110 lb bags	74	4.0
Pick and shovel work	74 to 88	4.0 to 4.8
Miscellaneous Leisure Activities		
Dancing, social	44 to 81	2.4 to 4.4
Calisthenics/exercise	55 to 74	3.0 to 4.0
Tennis, singles	66 to 74	3.6 to 4.0
Basketball	90 to 140	5.0 to 7.6
Wrestling, competitive	130 to 160	7.0 to 8.7

[a] 1 met = 18.4 Btu/h·ft^2

known, because metabolic power increases in proportion to exercise intensity. Metabolic rate varies over a wide range, depending on the activity, the person, and the conditions under which the activity is performed. Table 4-1 lists typical metabolic rates for an average adult male ($A_D = 19.7$ ft^2) for activities performed continuously. The highest power a person can maintain for any continuous period is approximately 50% of the maximal capacity to use oxygen (maximum energy capacity).

A unit used to express the metabolic rate per unit DuBois area is the met, defined as the metabolic rate of a sedentary person (seated, quiet): 1 met = 18.4 Btu/h·ft^2) = 50 kcal/(h·m^2). A normal, healthy man has a maximum capacity of approximately M_{act} = 12 met at age 20, which drops to 7 met at age 70. Maximum rates for women are about 30% lower. Long-distance runners and trained athletes have maximum rates as high as 20 met. An average 35 year-old who does not exercise has a maximum rate of about 10 met and activities with $M_{act} > 5$ met are likely to prove exhausting.

The metabolic activities of the body result almost completely in heat that must be continuously dissipated and regulated to prevent abnormal body temperatures. Insufficient heat loss leads to over-heating, called hyperthermia, and excessive heat loss results in body cooling, called hypothermia. Skin temperatures associated with comfort at sedentary activities are 91.5 to 93°F and decrease with increasing activity.

The heat produced by a resting adult is about 340 Btu/h. Because most of this heat is transferred to the environment through the skin, it is often convenient to characterize metabolic activity in terms of heat production per unit area of skin. For the resting person, this is about 18.4 Btu/h·ft^2 (58 W/m^2) (the average male has a skin surface area of about 19.7 ft^2) and is called 1 met. Higher metabolic rates are often described in terms of the resting rate. Thus, a person working at a metabolic rate five times the resting rate would have a metabolic rate of 5 met.

4.1.3 Clothing Level

Clothing insulation value may be expressed in clo units. In order to avoid confusion, the symbol I is used with the clo unit instead of the symbol R, normally used for thermal resistance per unit area. The relationship between the two is

$$R = 0.88I \tag{4-2}$$

or 1.0 clo is equivalent to 0.88 ft^2·h·°F/Btu. (0.155 (m^2·K)/W).

Often it is not possible to find an already measured clothing ensemble that matches the one in question. In this case, the ensemble insulation can be estimated from the insulation of individual garments. Table 4-2 gives a list of individual garments commonly worn. The insulation of an ensemble is estimated from the individual values using a summation formula:

$$I_{cl} = \sum_i I_{clu,i} \tag{4-3}$$

The main source of inaccuracy is in determining the appropriate values for individual garments. Overall accuracies are on the order of ±25% if the tables are used carefully. Where $I_{clu,i}$ is the effective insulation of garment i, and I_{cl}, as before is the insulation for the entire ensemble.

4.1.4 Conditions for Thermal Comfort

Environmental conditions for good thermal comfort minimize effort of the physiological control system. For a resting person wearing trousers and a long-sleeved shirt, thermal comfort in a steady state is experienced in a still-air environment at 75°F. A zone of comfort extends about 3°F above and below this optimum level.

ANSI/ASHRAE Standard 55-2013*, Thermal Environmental Conditions for Human Occupancy* specifies conditions of the indoor thermal environment that a majority of the occupants will find acceptable. The body of the standard clearly defines “majority” such that the requirements are based on 80% overall acceptability, while specific dissatisfaction limits vary for different sources of local discomfort. A space that meets the criteria of the standard likely will have individual occupants who are not satisfied due to large individual differences in preference and sensitivity. The standard is intended for use in designing, commissioning, and testing of buildings and other occupied spaces and their HVAC systems and for the evaluation of existing thermal environments. Standard 55 deals exclusively with thermal comfort in the indoor environment. The scope is not limited to any specific building type, so it may be used for residential or commercial buildings and for new or existing buildings. It also can apply to other occupied spaces such as cars, trains, planes, and ships.

The standard does not cover hot or cold stress in thermally extreme environments or comfort in outdoor spaces. It also does not address nonthermal environmental conditions (e.g., air quality or acoustics) or the effect of any environmental factors on nonthermal human responses (e.g., the effect of humidity on health). The scope clearly states that its criteria are based only on thermal comfort. Thus, a minimum humidity level is not specified since no lower humidity limits relate exclusively to thermal comfort. The form of the upper limit of humidity has changed throughout the standard’s history. The current upper limit is specified in terms of absolute humidity as the limiting parameter at a humidity ratio of 0.012.

The winter and summer comfort zones specified in ANSI/ASHRAE Standard 55 are given in Figure 4-2. The temperature ranges are appropriate for current seasonal clothing habits in the United States. Summer clothing is considered to be light slacks and a short-sleeved shirt or a comparable ensemble with an insulation value of 0.5 clo. Winter clothing is considered heavy slacks, long-sleeved shirt, and sweater or jacket with an insulation value of 0.9 clo. The temperature ranges are for sedentary and slightly active people.

The winter zone is for air speeds less than 0.15 m/s; the summer zone is for air movements less than 0.25 m/s. The standard allows the summer comfort zone to extend above 26°C if the average air movement is increased 0.275 m/s for each °C of temperature increase to a maximum temperature of 28°C and air movement of 0.8 m/s.

Table 4-2 Garment Insulation Values
(Table 9, Chapter 9, 2017 Handbook—Fundamentals)

Garment Description[a]	$I_{clu,i}$, clo[b]	Garment Description[a]	$I_{clu,i}$, clo[b]	Garment Description[a]	$I_{clu,i}$, clo[b]
Underwear		Long-sleeve, flannel shirt	0.34	Long-sleeve (thin)	0.25
Men's briefs	0.04	Short-sleeve, knit sport shirt	0.17	Long-sleeve (thick)	0.36
Panties	0.03	Long-sleeve, sweat shirt	0.34	**Dresses and skirts[c]**	
Bra	0.01	**Trousers and Coveralls**	0.06	Skirt (thin)	0.14
T-shirt	0.08	Short shorts	0.08	Skirt (thick)	0.23
Full slip	0.16	Walking shorts	0.15	Long-sleeve shirtdress (thin)	0.33
Half slip	0.14	Straight trousers (thin)	0.24	Long-sleeve shirtdress (thick)	0.47
Long underwear top	0.20	Straight trousers (thick)	0.28	Short-sleeve shirtdress (thin)	0.29
Long underwear bottoms	0.15	Sweatpants	0.30	Sleeveless, scoop neck (thin)	0.23
Footwear		Overalls	0.49	Sleeveless, scoop neck (thick), i.e., jumper	0.27
Ankle-length athletic socks	0.02	Coveralls			
Calf-length socks	0.03	**Suit jackets and vests (lined)**		**Sleepwear and Robes**	
Knee socks (thick)	0.06	Single-breasted (thin)	0.36	Sleeveless, short gown (thin)	0.18
Panty hose	0.02	Single-breasted (thick)	0.44	Sleeveless, long gown (thin)	0.20
Sandals/thongs	0.02	Double-breasted (thin)	0.42	Short-sleeve hospital gown	0.31
Slippers (quilted, pile-lined)	0.03	Double-breasted (thick)	0.48	Long-sleeve, long gown (thick)	0.46
Boots	0.10	Sleeveless vest (thin)	0.10	Long-sleeve pajamas (thick)	0.57
Shirts and Blouses		Sleeveless vest (thick)	0.17	Short-sleeve pajamas (thin)	0.42
Sleeveless, scoop-neck blouse	0.12	**Sweaters**		Long-sleeve, long wrap robe (thick)	0.69
Short-sleeve, dress shirt	0.19	Sleeveless vest (thin)	0.13	Long-sleeve, short wrap robe (thick)	0.48
Long-sleeve, dress shirt	0.25	Sleeveless vest (thick)	0.22	Short-sleeve, short robe (thin)	0.34

[a] "Thin" garments are made of light, thin fabrics worn in summer; "thick" garments are made of heavy, thick fabrics worn in winter.

[b] 1 clo = 0.880°F·ft^2·h/Btu

[c] Knee-length

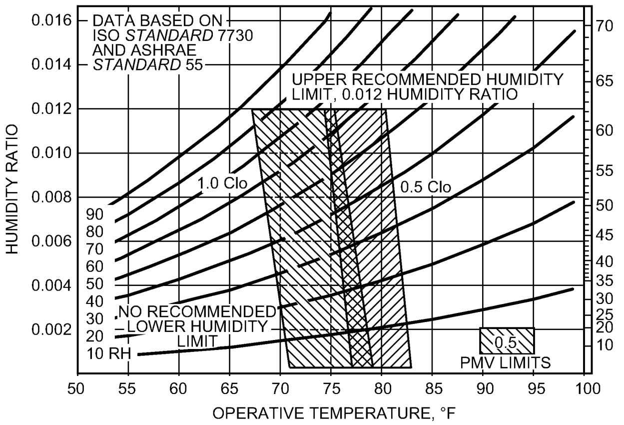


Fig. 4-2 ASHRAE Summer and Winter Comfort Zones

(Acceptable ranges of operative temperature and humidity with air speed ≤ 40 fpm for people wearing 1.0 and 0.5 clothing during primarily sedentary activity.)

(Figure 5, Chapter 9, 2017 ASHRAE Handbook—Fundamentals)

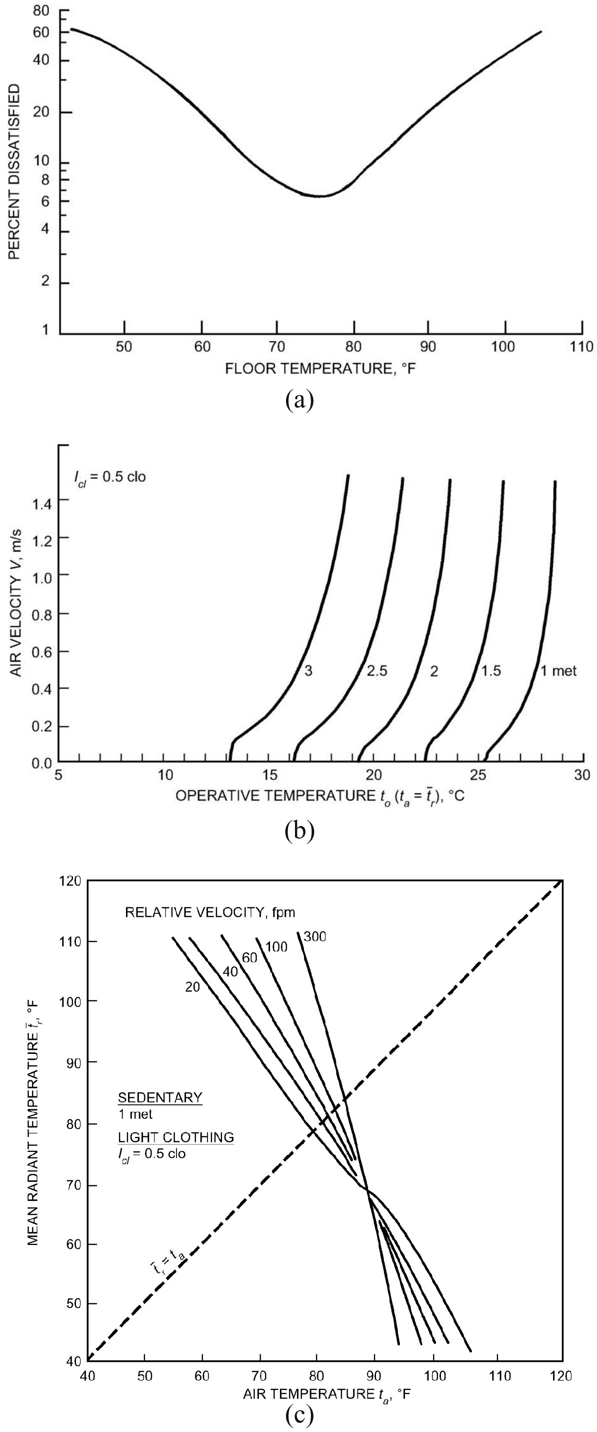
The temperature boundaries of the comfort zones in Figure 4-2 can be shifted –1°F (–0.6°C) per 0.1 clo for clothing levels other than 0.5 and 0.9. The zones can also be shifted lower for increased activity levels.

Thermal comfort conditions must be fairly uniform over the body to prevent local discomfort. The radiant temperature asymmetry should be less than 5°C in the vertical direction and 10°C in the horizontal direction. The vertical temperature difference between head and foot should not exceed 3°C.

Figure 4-2 applies generally to altitudes from sea level to 3000 m (10,000 ft), and to the most common indoor thermal environments in which mean radiant temperature is nearly equal to the dry-bulb air temperature and where the air velocity is less than 30 fpm in winter and 50 fpm in summer. For these cases, the thermal environment can be specified by operative temperature and humidity variables.

Comfort zones for other clothing levels can be approximated by decreasing the temperature borders of the zone by 1°F for each 0.1 clo increase in clothing insulation and vise-versa. Similarly, a zone's temperatures can be decreased by 2.5°F per met increase in activity above 1.2 met. The upper and lower humidity levels of the comfort zones are less precise. Low humidity can lead to drying of the skin and mucous

surfaces. Comfort complaints about dry nose, throat, eyes, and skin occur in low humidity conditions, typically when the dew point is less than 32°F. In compliance with these and other discomfort observations, Standard 55 recommends that the dew point temperature of occupied spaces not be less than 35°F. In contrast, at high humidity levels too much skin moisture tends to increase discomfort, particularly skin moisture that is physiological in origin (water diffusion and perspiration). On the warm side of the comfort zone, the relative humidity should not exceed 60%.



(a)

(b)

(c)

Fig. 4-3 Examples of Fanger's Charts

(Figures 13, 14, and 15, Chapter 9, 2017 ASHRAE Handbook—Fundamentals)

Table 4-3 provides design criteria covering factors that apply to many different building types.

4.1.5 Adjustments for Clothing and/or Activity Levels

Activity. The indoor designated temperatures of Figure 4-2 should be decreased when the average steady-state activity level of the occupants is higher than light, primarily sedentary (>1.2 met).

This temperature can be calculated from the operative temperature at sedentary conditions with the following equation:

$$t_{i,\text{active}} = t_{i,\text{sedentary}} - 3(1 + \text{clo})(\text{met} - 1.2)(°\text{C}) \qquad (4\text{-}4)$$

$$t_{i,\text{active}} = t_{i,\text{sedentary}} - 5.4(1 + \text{clo})(\text{met} - 1.2)(°\text{F}) \qquad (4\text{-}5)$$

The equation is only appropriate between 1.2 and 3 met. The minimum allowable operative temperature for these activities is 15°C (59°F). The acceptable range (based on a 10% dissatisfaction criterion) will increase with activity and clothing. The ranges are approximately ±1.4°C (2.7°F) for 0.1 clo, ±2°C (3.5°F) for 0.5 clo, and ±3°C (5.4°F) for 0.9 clo.

Clothing. The temperatures of Figure 4.2, after being corrected for activity level, can be corrected for clothing level using a decrease of 1°F for each 0.1 clo increase.

A wide range of environmental applications are covered by the ANSI/ASHRAE Standard 55. The comfort envelope defined by the standard applies only for sedentary and slightly active, normally clothed people at low air velocities, when the mean radiant temperature (MRT) equals the air temperature. For other clothing, activities, air temperatures, etc., the standard recommends use of Fanger's General Comfort Charts. Examples of these are shown in Figure 4-3.

Example 4-1 Determine the optimal comfort conditions for a conference room under summer conditions. Occupants wear light clothing (0.5 clo) during summer. Air movement is 0.1 m/s (20 fpm), MRT = air temperature, and summer rh = 70%.

Solution:

From Figure 4-3(b):
Air Temperature = MRT = 78°F (26°C)

Example 4-2 If, in winter, neither the air temperature nor the level of clothing for Example 4.1 change, what activity level must the occupant move up to in order to be comfortable?

Solution:

From Figure 4-3(a): Given conditions of 0.33 ft/s and 78°F fall on comfort line at approximately 1 met (only slightly more active, e.g., reading or writing).

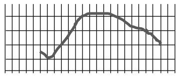
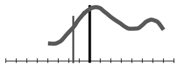
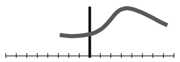
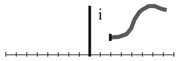
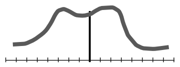
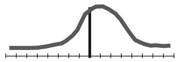
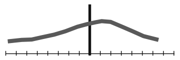
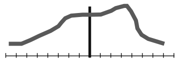
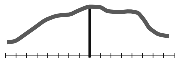
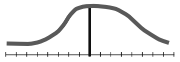
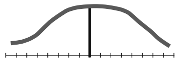
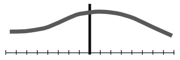
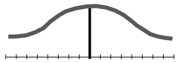
The indoor conditions to be maintained within a building are the dry-bulb temperature and relative humidity of the air at the breathing line, 1 to 1.5 m (3 to 5 ft) above the floor, in an area that would indicate average conditions at that level and which would not be affected by abnormal or unusual heat gains or losses from the interior or exterior.

4.1.6 Moisture and Humidity

Too often, the behavior of moisture is given insufficient attention in building design and construction. Moisture is

Table 4-3 General Design Criteria

(Adapted from Table 1, Chapter 3, 2007 Handbook—HVAC Applications)

General Category	Specific Category	Inside Design Conditions		Air Movement	Circulation, Air Changes per Hour	Load Profile
		Winter	Summer			
Dining and Entertainment Centers	Cafeterias and Luncheonettes	70°F to 74°F 20% to 30% rh	78°F[e] 50% rh	50 fpm at 6 ft above floor	12 to 15	Peak at 1 to 2 p.m. 
	Restaurants	70°F to 74°F 20% to 30% rh	74°F to 78°F 55% to 60% rh	25 to 30 fpm	8 to 12	Peak at 1 to 2 p.m. 
	Bars	70°F to 74°F 20% to 30% rh	74°F to 78°F 50% to 60% rh	30 fpm at 6 ft above floor	15 to 20	Peak at 5 to 7 p.m. 
	Nightclubs	70°F to 74°F 20% to 30% rh	74°F to 78°F 50% to 60% rh	below 25 fpm at 5 ft above floor	20 to 30	Peak after 8 p.m., off from 2 a.m. to 4 p.m.
	Kitchens	70°F to 74°F	85°F to 88°F	30 to 50 fpm	12 to 15[h]	
Office Buildings		70°F to 74°F 20% to 30% rh	74°F to 78°F 50% to 60% rh	25 to 45 fpm 0.75 to 2 cfm/ft^2	4 to 10	Peak at 4 p.m. 
Libraries and Museums	Average	68°F to 72°F 40% to 55% rh		below 25 fpm	8 to 12	Peak at 3 p.m. 
	Archival	See Chapter 20, 1999 *Handbook—HVAC Applications*		below 25 fpm	8 to 12	Peak at 3 p.m. 
Bowling Centers		70°F to 74°F 20% to 30% rh	75°F to 78°F 50% to 55% rh	50 fpm at 6 ft above floor	10 to 15	Peak at 6 to 8 p.m. 
Communication Centers	Telephone Terminal Rooms	72°F to 78°F 40% to 50% rh	72°F to 78°F 40% to 50% rh	25 to 30 fpm	8 to 20	Varies with location and use
	Teletype Centers	70°F to 74°F 40% to 50% rh	74°F to 78°F 45% to 55% rh	25 to 30 fpm	8 to 20	Varies with location and use
	Radio and Television Studios	74°F to 78°F 30% to 40% rh	74°F to 78°F 40% to 55% rh	below 25 fpm at 12 ft above floor	15 to 40	Varies widely due to changes in lighting and people
Transportation Centers	Airport Terminals	70°F to 74°F 20% to 30% rh	74°F to 78°F 50% to 60% rh	25 to 30 fpm at 6 ft above floor	8 to 12	Peak at 10 a.m. to 9 p.m. 
	Ship Docks	70°F to 74°F 20% to 30% rh	74°F to 78°F 50% to 60% rh	25 to 30 fpm at 6 ft above floor	8 to 12	Peak at 10 a.m. to 5 p.m. 
	Bus Terminals	70°F to 74°F 20% to 30% rh	74°F to 78°F 50% to 60% rh	25 to 30 fpm at 6 ft above floor	8 to 12	Peak at 10 a.m. to 5 p.m. 
	Garages[l]	40°F to 55°F	80°F to 100°F	30 to 75 fpm	4 to 6 Refer to NFPA	Peak at 10 a.m. to 5 p.m. 
Warehouses		Inside design temperatures for warehouses often depend on the materials stored.			1 to 4	Peak at 10 a.m. to 3 p.m. 

present as a vapor in all air and as absorbed moisture in most building materials. Problems involving moisture may arise from changes in moisture content, from the presence of excessive moisture, or from effects associated with its changes in state.

Water vapor originates from such activities as cooking, laundering, bathing, and people breathing and perspiring. Some typical values of moisture production are given in Table 4-4.

Exterior and interior building materials should allow vapor to pass five times more rapidly than materials inside the wall. Provided this condition is met, any moisture that may get into a wall will move on through it. There is a complete discussion of moisture behavior in Chapter 36, “Moisture Management in Buildings” in the 2017 ASHRAE Handbook—*Fundamentals.*

Selecting and applying humidification or dehumidification equipment involves considering both the environmental requirements of the occupancy or process and the limitations imposed by the thermal and permeable characteristics of the building enclosure. As these may not always be compatible, a compromise solution may be necessary, particularly in the case of existing buildings.

The environmental requirements for a particular occupancy or process may dictate a specific relative humidity, a required range of relative humidity, or certain limiting maximum or minimum values. The following classifications give guidance for most applications.

Human Comfort. The effect of relative humidity on human comfort has not been completely established. Nevertheless, humidity extremes are assumed to be undesirable and, for human comfort, relative humidities should be kept within a broad range of 30 to 60%.

Static Electricity. Electrostatic charges are generated when materials of high electrical resistance move against each other. Such charges may cause unpleasant sparks for people walking over carpets; difficulties in handling sheets of paper, fibers, and fabrics; objectionable clinging of dust to oppositely charged objects; or dangerous situations when explosive gases are present. Increasing the relative humidity of an environment tends to prevent the accumulation of such charges, but the optimum level of humidity depends to some extent on the materials involved. With many materials, relative humidities of 45% or more are usually required to reduce or eliminate electrostatic effects. Hospital operating rooms, where explosive mixtures of anesthetics are used, constitute a special and critical case in regard to electrostatic charges. A relative humidity of 50% or more is usually required and other special grounding arrangements and restrictions are imposed as to types of clothing the occupants wear. From a consideration of both comfort and safety, conditions of 72°F (22°C) and 55% rh are usually recommended in operating rooms.

Prevention and Treatment of Disease. Relative humidity has a significant effect on the control of airborne infection. At 50% rh, the mortality rate of certain organisms is highest (e.g., influenza virus loses much of its virulence). The mortality rate decreases both above and below this value. A relative humidity of 65% is regarded as optimum for nurseries for premature infants, while a value of 50% is suitable for full-terms and observational nurseries. In the treatment of allergic disorders, humidities well below 50% have proven satisfactory.

Visible Condensation. Condensation occurs on any interior surface when the dew point of the air in contact with it exceeds the surface temperature. The maximum permissible relative humidity that may be maintained without condensation is thus influenced by the thermal properties of the enclosure and the interior and exterior environment. In general, windows present the lowest surface temperature in most buildings and provide the best guide to permissible indoor humidity levels for no condensation (Table 4-5).

Concealed Condensation. The humidity level a building can tolerate without serious difficulties from concealed condensation may be much lower than indicated by visible condensation criteria. The migration of water vapor through the inner envelope by diffusion or air leakage brings it into contact with surfaces at temperatures approaching the outside temperature. Unless the building has been designed to eliminate or effectively reduce this possibility, the permissible humidity may be limited by the ability of the building enclosure to handle internal moisture rather than prevent the occurrence of moisture.

4.2 Outdoor Design Conditions: Weather Data

The 2017 *ASHRAE Handbook—Fundamentals*, Chapter 14, “Climatic Design Information,” and its accompanying CD-ROM provide design weather information for 8118 locations in the United States, Canada, and around the world. The large number of stations made printing all the tables imprac-

Table 4-4 Moisture Production in Residences

(Table 1, Chapter 36, 2017 ASHRAE Handbook—Fundamentals)

Source		Units	Release, × 10^{-3}
Humans	Light activity	lb/h	66-130
	Medium activity	lb/h	270-440
	Hard work	lb/h	440-660
Bathroom	Bath (15 min)	lb	130
	Shower (15 min)	lb	1460
Breakfast preparation for 4 people		lb	350-600
Lunch preparation for 4 people		lb	550-710
Dinner preparation for 4 people		lb	1210-1590
Breakfast dish washing for 4 people		lb	220
Lunch dish washing for 4 people		lb	150
Dinner dish washing for 4 people		lb	680
Simmering pot (diameter 5.9 in., 10 min)		lb	130
Boiling pot (diameter 5.9 in., 10 min)		lb	570
Potted flowers		lb/h per pot	5-20
Potted plants		lb/h per pot	7-30
Laundry			
Already spin-dried, until dry		lb/h	40-440
Dripping wet, until dry		lb/h	220-11001
Unvented drier, until dry		lb/h	4700-6390

Sources: IEA-ECB (1990), Kumaran and Sanders (2008), Sanders (1996), and TenWolde and Walker (2001).

Table 4-5 Maximum Relative Humidity without Window Condensation

Natural Convection, Indoor Air at 23.3°C (74°F)			
Outdoor Temp., °C (°F)		**Single Glazing**	**Double Glazing**
4.4	(40)	39	59
−1.1	(30)	29	50
−6.7	(20)	21	43
−12.2	(10)	15	36
−17.8	(0)	10	30
−23.3	(–10)	7	26
−28.9	(–20)	5	21
−34.4	(–30)	3	17

Table 4-6 Nomenclature for Tables of Climatic Design Conditions

(Table 1A, Chapter 14, 2013 ASHRAE Handbook—Fundamentals)

CDD*n*	Cooling degree-days base *n*°F, °F-day
CDH*n*	Cooling degree-hours base *n*°F, °F-hour
DB	Dry-bulb temperature, °F
DBAvg	Average daily dry-bulb temperature, °F
DBStd	Standard deviation of daily average dry-bulb temperature, °F
DP	Dew-point temperature, °F
Ebn,noon	Clear sky beam normal irradiances at solar noon, Btu/h·ft^2
Edh,noon	Clear sky diffuse horizontal irradiance at solar noon, Btu/h·ft^2
Elev	Elevation, ft
Enth	Enthalpy, Btu/lb
HDD*n*	Heating degree-days base n°F, °F-day
HR	Humidity ratio, grains of moisture per lb of dry air
Lat	Latitude, °
Long	Longitude, °
MCDB	Mean coincident dry bulb temperature, °F
MCDBR	Mean coincident dry bulb temp. range, °F
MCWB	Mean coincident wet bulb temperature, °F
MCWBR	Mean coincident wet bulb temp. range, °F
MCWS	Mean coincident wind speed, mph
MDBR	Mean dry bulb temp. range, °F
PCWD	Prevailing coincident wind direction, ° (0 = North; 90 = East)
Period	Years used to calculate the design conditions
PrecAvg	Average precipitation, in.
PrecMax	Maximum precipitation, in.
PrecMin	Minimum precipitation, in.
PrecStd	Standard deviation of precipitation, in.
RadAvg	Average daily all sky solar radiation, Btu/ft^2
RadStd	Standard devation of average daily all sky solar radiation, Btu/ft^2
StdP	Standard pressure at station elevation, psi
taub	Clear sky optical depth for beam irradiance
taud	Clear sky optical depth for diffuse irradiance
Time Zone	Hours ahead or behind UTC, and time zone code
WB	Wet bulb temperature, °F
WBAN	Weather Bureau Army Navy number
WMO#	Station identifier from the World Meteorological Organization
WS	Wind speed, mph
WSAvg	Average wind speed, mph

Note: Numbers *(1)* to *(45)* and letters *(a)* to *(p)* are row and column references to quickly point to an element in the table. For example, the 5% design wet-bulb temperature for July can be found in row *(31)*, column *(k)*.

tical. However, 31 of the locations required for the solution of the problems in this textbook have been included. These 31 locations make up Figure 4-4. The complete tables are contained on a CD-ROM distributed with the 2017 *ASHRAE Handbook—Fundamentals*.

This climatic design information is commonly used for design, sizing, distribution, installation, and marketing of heating, ventilating, air-conditioning, and dehumidification equipment, as well as for other energy-related processes in residential, agricultural, commercial, and industrial applications. These summaries include values of dry-bulb, wet-bulb, and dew-point temperature, and wind speed with direction at various frequencies of occurrence. Also included in this edition are monthly degree-days to various bases, parameters to calculate clear-sky irradiance, and monthly averages of daily all-sky solar radification.

Design information in this chapter includes design values of dry-bulb with mean coincident wet-bulb temperature, design wet-bulb with mean coincident dry-bulb temperature, and design dew-point with mean coincident dry-bulb temperature and corresponding humidity ratio. These data allow the designer to consider various operational peak conditions. Design values of wind speed facilitate the design of smoke management systems in buildings.

Warm-season temperature and humidity conditions are based on annual percentiles of 0.4, 1.0, and 2.0. Cold-season conditions are based on annual percentiles of 99.6 and 99.0. The use of annual percentiles to define design conditions ensures that they represent the same probability of occurrence in any climate, regardless of the seasonal distribution of extreme temperature and humidity.

Monthly information including percentiles is compiled in addition to annual percentiles, to provide seasonally representative combinations of temperature, humidity, and solar conditions. The tables also list heating and cooling degree-days for bases 65°F and 50°F, as well as cooling degree-hours for bases 74°F and 80°F. The calculation of daily dry-bulb and wet-bulb temperature profiles, which are useful for generating 24 h weather data sequences suitable as input to many HVAC analysis methods, has been significantly updated, with the inclusion of mean dry-bulb and wet-bulb temperature ranges coincident with the 5% monthly dry-bulb and wet-bulb design temperatures.

Design conditions are provided for locations for which long-term hourly observations were available (1990–2014 for most stations in the United States and Canada).

Figure 4-4 shows climatic design conditions for 31 locations, to illustrate the format of the data available on the CD-ROM. A subset of the United States and Canada weather stations containing 23 annual data elements is provided for conveniences in Table 4-7.

The top part of Figure 4-4 contains station information as follows:

- Name of the observing station, state (USA) or province (Canada), country.
- World Meteorological Organization (WMO) station identifier.

2017 ASHRAE Handbook - Fundamentals (IP)

ATLANTA HARTSFIELD-JACKSON, GA, USA

WMO#: **722190**

Lat: **33.640N** Long: **84.430W** Elev: **1027** StdP: **14.16** Time Zone: **-5.00 (NAE)** Period: **90-14** WBAN: **13874**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	21.9	26.5	4.9	7.3	29.3	9.3	9.2	32.8	24.8	39.7	23.3	39.2	11.8	320	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	16.7	94.0	74.2	91.6	73.8	89.5	73.3	77.3	88.3	76.3	86.5	75.4	84.8	8.7	300	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	74.3	133.3	81.3	73.4	128.9	80.3	72.6	125.5	79.6	41.3	88.3	40.3	86.7	39.5	85.4	82.4	(3)

Extreme Annual Design Conditions

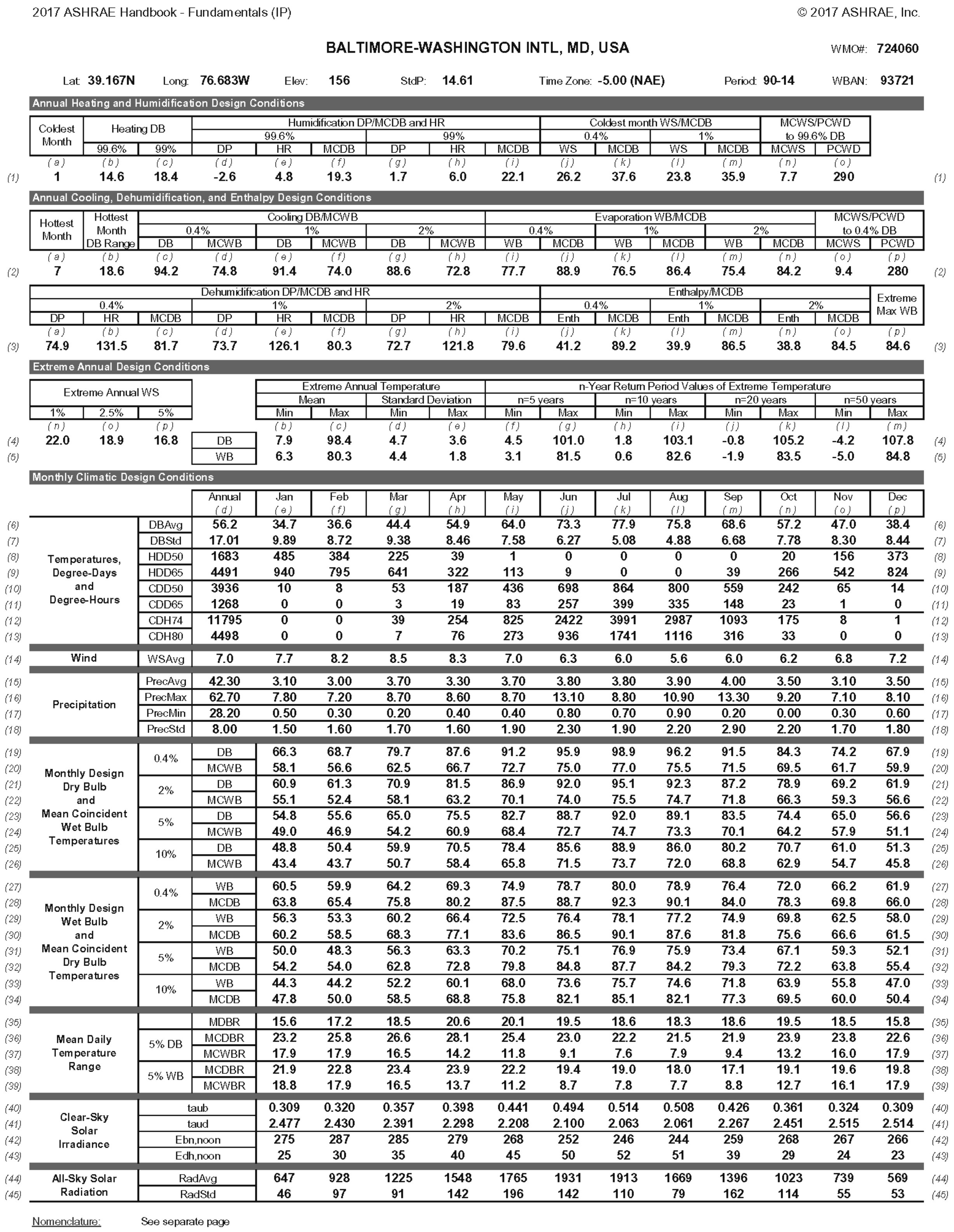
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	21.3	18.9	17.0	DB	15.0	96.6	4.6	3.7	11.6	99.3	8.9	101.4	6.3	103.5	3.0	106.2	(4)
(5)				WB	12.6	79.0	4.4	1.5	9.5	80.1	6.9	81.0	4.4	81.8	1.2	82.9	(5)

Monthly Climatic Design Conditions

				Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	
				(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**		DBAvg	63.0	44.5	48.0	54.9	62.6	70.5	77.3	80.1	79.5	73.9	63.6	53.5	46.4	(6)
(7)			DBStd	14.60	9.66	8.88	9.00	7.40	6.17	4.51	3.50	3.84	5.47	7.07	8.10	8.82	(7)
(8)			HDD50	653	224	132	55	6	0	0	0	0	0	4	56	176	(8)
(9)			HDD65	2640	635	477	329	127	25	1	0	0	7	111	352	576	(9)
(10)			CDD50	5391	54	76	205	385	636	819	933	914	717	425	161	66	(10)
(11)			CDD65	1901	0	1	14	56	195	370	468	449	273	67	7	1	(11)
(12)			CDH74	16445	0	3	103	435	1509	3337	4488	4173	2014	360	22	1	(12)
(13)			CDH80	6151	0	0	8	66	419	1333	1922	1726	637	40	0	0	(13)
(14)	**Wind**		WSAvg	8.2	9.2	9.4	9.5	8.8	7.9	7.3	7.0	6.7	7.5	7.8	8.2	8.9	(14)
(15)	**Precipitation**		PrecAvg	50.80	4.70	4.80	5.80	4.30	4.30	3.60	5.00	3.70	3.40	3.10	3.90	4.30	(15)
(16)			PrecMax	64.90	10.20	12.80	11.70	11.90	8.40	7.40	8.50	8.70	6.10	7.50	7.20	9.90	(16)
(17)			PrecMin	37.70	1.70	0.80	2.40	1.50	0.40	1.00	0.70	0.50	0.70	0.10	0.90	0.70	(17)
(18)			PrecStd	7.20	2.10	2.70	2.70	2.40	2.30	1.80	2.20	2.20	1.60	2.10	1.60	2.40	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB		70.5	73.4	80.8	85.0	90.2	94.5	97.6	97.4	92.8	83.5	77.4	72.3	(19)
(20)			MCWB		60.8	61.3	62.6	66.5	71.3	72.9	74.8	74.3	72.0	68.3	64.1	63.1	(20)
(21)		2%	DB		66.3	69.1	76.9	82.0	87.0	92.0	94.1	93.7	89.0	80.9	73.3	68.6	(21)
(22)			MCWB		58.7	58.8	60.6	64.9	69.8	72.9	74.7	74.6	71.4	66.7	61.8	61.6	(22)
(23)		5%	DB		63.0	65.9	73.5	79.2	84.6	89.8	91.6	90.9	86.4	78.2	70.3	64.6	(23)
(24)			MCWB		57.2	56.9	59.1	63.4	68.8	72.5	74.3	74.2	70.7	64.7	60.6	59.0	(24)
(25)		10%	DB		59.5	62.7	69.8	76.1	82.1	87.4	89.1	88.3	83.8	75.3	67.1	60.9	(25)
(26)			MCWB		53.9	54.5	57.7	61.8	67.8	71.9	74.2	73.4	70.2	64.1	58.4	55.2	(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB		64.2	65.5	66.4	70.8	74.9	77.3	78.9	78.4	76.4	72.0	69.2	66.7	(27)
(28)			MCDB		67.5	67.9	72.8	79.2	83.3	88.4	89.6	89.9	86.1	78.8	71.8	69.8	(28)
(29)		2%	WB		61.6	62.4	64.1	68.3	72.7	75.9	77.5	77.2	74.6	70.2	66.1	63.5	(29)
(30)			MCDB		64.6	66.6	71.5	76.3	82.3	86.5	88.3	88.4	83.2	75.9	70.0	66.4	(30)
(31)		5%	WB		58.6	59.5	62.1	66.4	71.3	74.9	76.5	76.2	73.5	68.7	63.5	60.3	(31)
(32)			MCDB		61.9	64.0	70.0	74.4	80.6	85.0	86.7	86.1	80.9	73.8	67.7	64.1	(32)
(33)		10%	WB		55.2	56.3	60.0	64.5	69.9	73.8	75.4	75.2	72.6	66.9	60.5	56.3	(33)
(34)			MCDB		58.3	60.6	67.0	72.3	78.6	83.1	84.9	84.2	79.5	72.3	65.4	59.8	(34)
(35)	**Mean Daily Temperature Range**		MDBR		17.3	18.2	19.2	19.8	18.2	17.2	16.7	16.4	16.5	18.1	18.7	16.6	(35)
(36)		5% DB	MCDBR		19.5	20.9	22.5	22.0	20.1	20.1	20.2	19.5	19.1	20.4	20.9	19.5	(36)
(37)			MCWBR		13.6	13.3	11.0	9.5	7.5	6.6	6.1	6.0	6.8	8.9	11.7	13.2	(37)
(38)		5% WB	MCDBR		16.0	17.5	17.9	18.2	17.3	17.3	17.5	16.8	15.3	14.9	16.8	16.8	(38)
(39)			MCWBR		13.2	13.8	11.0	9.8	7.7	6.9	6.4	6.1	6.9	8.9	12.9	13.6	(39)
(40)	**Clear-Sky Solar Irradiance**		taub		0.310	0.315	0.347	0.386	0.440	0.473	0.515	0.515	0.417	0.363	0.333	0.311	(40)
(41)			taud		2.538	2.521	2.453	2.324	2.213	2.168	2.066	2.052	2.312	2.460	2.484	2.554	(41)
(42)			Ebn,noon		288	298	295	286	270	260	249	246	268	277	276	281	(42)
(43)			Edh,noon		26	29	34	40	46	48	53	52	39	31	27	24	(43)
(44)	**All-Sky Solar Radiation**		RadAvg		852	1054	1407	1751	1904	1954	1858	1726	1496	1275	963	738	(44)
(45)			RadStd		50	122	109	126	164	162	134	103	148	163	82	73	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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BALTIMORE-WASHINGTON INTL, MD, USA WMO#: **724060**

Lat: **39.167N** Long: **76.683W** Elev: **156** StdP: **14.61** Time Zone: **-5.00 (NAE)** Period: **90-14** WBAN: **93721**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	14.6	18.4	-2.6	4.8	19.3	1.7	6.0	22.1	26.2	37.6	23.8	35.9	7.7	290	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	18.6	94.2	74.8	91.4	74.0	88.6	72.8	77.7	88.9	76.5	86.4	75.4	84.2	9.4	280	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	74.9	131.5	81.7	73.7	126.1	80.3	72.7	121.8	79.6	41.2	89.2	39.9	86.5	38.8	84.5	84.6	(3)

Extreme Annual Design Conditions

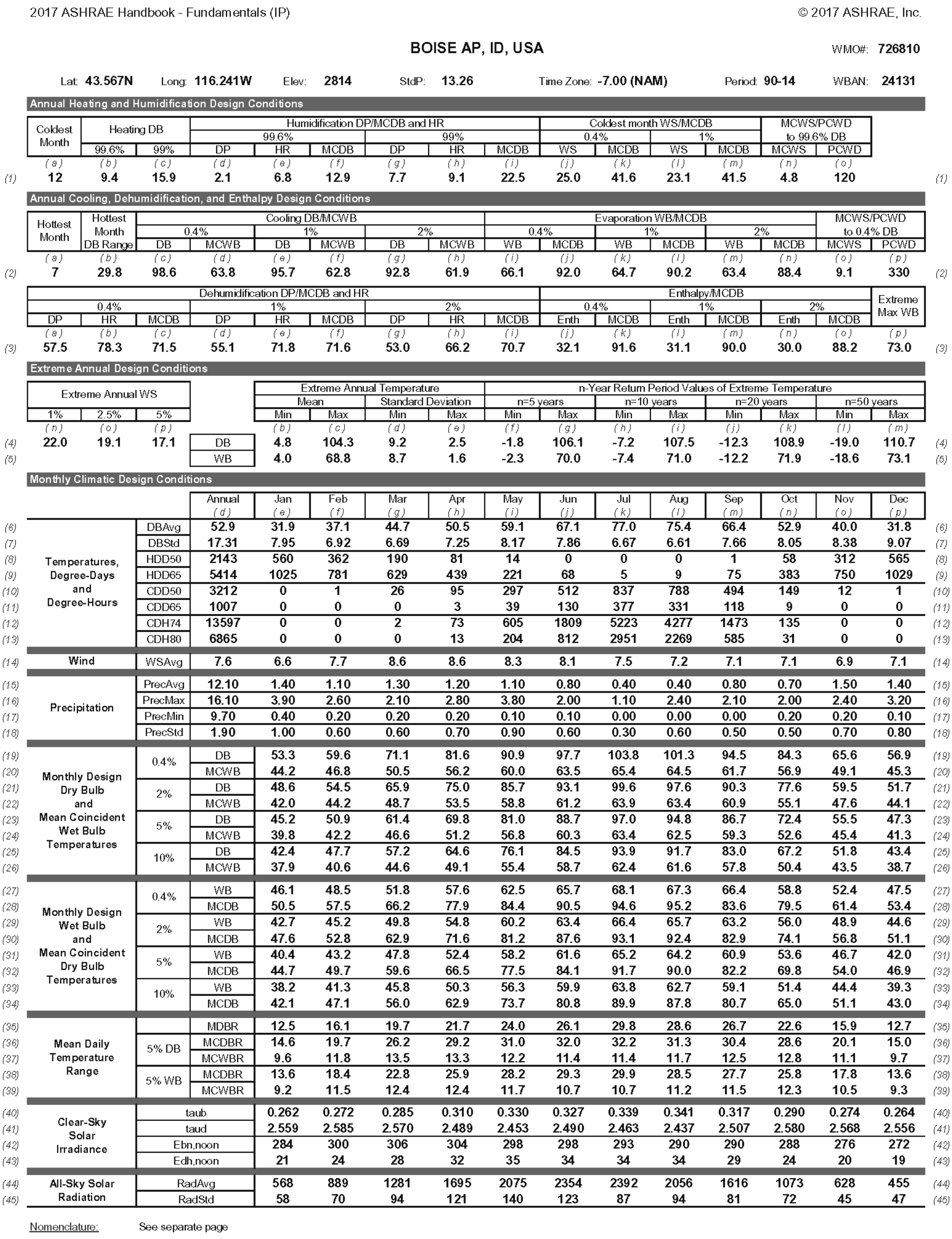
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	22.0	18.9	16.8	DB	7.9	98.4	4.7	3.6	4.5	101.0	1.8	103.1	-0.8	105.2	-4.2	107.8	(4)
(5)				WB	6.3	80.3	4.4	1.8	3.1	81.5	0.6	82.6	-1.9	83.5	-5.0	84.8	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	56.2	34.7	36.6	44.4	54.9	64.0	73.3	77.9	75.8	68.6	57.2	47.0	38.4	(6)
(7)			DBStd	17.01	9.89	8.72	9.38	8.46	7.58	6.27	5.08	4.88	6.68	7.78	8.30	8.44	(7)
(8)			HDD50	1683	485	384	225	39	1	0	0	0	0	20	156	373	(8)
(9)			HDD65	4491	940	795	641	322	113	9	0	0	39	266	542	824	(9)
(10)			CDD50	3936	10	8	53	187	436	698	864	800	559	242	65	14	(10)
(11)			CDD65	1268	0	0	3	19	83	257	399	335	148	23	1	0	(11)
(12)			CDH74	11795	0	0	39	254	825	2422	3991	2987	1093	175	8	1	(12)
(13)			CDH80	4498	0	0	7	76	273	936	1741	1116	316	33	0	0	(13)
(14)	Wind		WSAvg	7.0	7.7	8.2	8.5	8.3	7.0	6.3	6.0	5.6	6.0	6.2	6.8	7.2	(14)
(15)	Precipitation		PrecAvg	42.30	3.10	3.00	3.70	3.30	3.70	3.80	3.80	3.90	4.00	3.50	3.10	3.50	(15)
(16)			PrecMax	62.70	7.80	7.20	8.70	8.60	8.70	13.10	8.80	10.90	13.30	9.20	7.10	8.10	(16)
(17)			PrecMin	28.20	0.50	0.30	0.20	0.40	0.40	0.80	0.70	0.90	0.20	0.00	0.30	0.60	(17)
(18)			PrecStd	8.00	1.50	1.60	1.70	1.60	1.90	2.30	1.90	2.20	2.90	2.20	1.70	1.80	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		66.3	68.7	79.7	87.6	91.2	95.9	98.9	96.2	91.5	84.3	74.2	67.9	(19)
(20)			MCWB		58.1	56.6	62.5	66.7	72.7	75.0	77.0	75.5	71.5	69.5	61.7	59.9	(20)
(21)		2%	DB		60.9	61.3	70.9	81.5	86.9	92.0	95.1	92.3	87.2	78.9	69.2	61.9	(21)
(22)			MCWB		55.1	52.4	58.1	63.2	70.1	74.0	75.5	74.7	71.8	66.3	59.3	56.6	(22)
(23)		5%	DB		54.8	55.6	65.0	75.5	82.7	88.7	92.0	89.1	83.5	74.4	65.0	56.6	(23)
(24)			MCWB		49.0	46.9	54.2	60.9	68.4	72.7	74.7	73.3	70.1	64.2	57.9	51.1	(24)
(25)		10%	DB		48.8	50.4	59.9	70.5	78.4	85.6	88.9	86.0	80.2	70.7	61.0	51.3	(25)
(26)			MCWB		43.4	43.7	50.7	58.4	65.8	71.5	73.7	72.0	68.8	62.9	54.7	45.8	(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		60.5	59.9	64.2	69.3	74.9	78.7	80.0	78.9	76.4	72.0	66.2	61.9	(27)
(28)			MCDB		63.8	65.4	75.8	80.2	87.5	88.7	92.3	90.1	84.0	78.3	69.8	66.0	(28)
(29)		2%	WB		56.3	53.3	60.2	66.4	72.5	76.4	78.1	77.2	74.9	69.8	62.5	58.0	(29)
(30)			MCDB		60.2	58.5	68.3	77.1	83.6	86.5	90.1	87.6	81.8	75.6	66.6	61.5	(30)
(31)		5%	WB		50.0	48.3	56.3	63.3	70.2	75.1	76.9	75.9	73.4	67.1	59.3	52.1	(31)
(32)			MCDB		54.2	54.0	62.8	72.8	79.8	84.8	87.7	84.2	79.3	72.2	63.8	55.4	(32)
(33)		10%	WB		44.3	44.2	52.2	60.1	68.0	73.6	75.7	74.6	71.8	63.9	55.8	47.0	(33)
(34)			MCDB		47.8	50.0	58.5	68.8	75.8	82.1	85.1	82.1	77.3	69.5	60.0	50.4	(34)
(35)	Mean Daily Temperature Range		MDBR		15.6	17.2	18.5	20.6	20.1	19.5	18.6	18.3	18.6	19.5	18.5	15.8	(35)
(36)		5% DB	MCDBR		23.2	25.8	26.6	28.1	25.4	23.0	22.2	21.5	21.9	23.9	23.8	22.6	(36)
(37)			MCWBR		17.9	17.9	16.5	14.2	11.8	9.1	7.6	7.9	9.4	13.2	16.0	17.9	(37)
(38)		5% WB	MCDBR		21.9	22.8	23.4	23.9	22.2	19.4	19.0	18.0	17.1	19.1	19.6	19.8	(38)
(39)			MCWBR		18.8	17.9	16.5	13.7	11.2	8.7	7.8	7.7	8.8	12.7	16.1	17.9	(39)
(40)	Clear-Sky Solar Irradiance		taub		0.309	0.320	0.357	0.398	0.441	0.494	0.514	0.508	0.426	0.361	0.324	0.309	(40)
(41)			taud		2.477	2.430	2.391	2.298	2.208	2.100	2.063	2.061	2.267	2.451	2.515	2.514	(41)
(42)			Ebn,noon		275	287	285	279	268	252	246	244	259	268	267	266	(42)
(43)			Edh,noon		25	30	35	40	45	50	52	51	39	29	24	23	(43)
(44)	All-Sky Solar Radiation		RadAvg		647	928	1225	1548	1765	1931	1913	1669	1396	1023	739	569	(44)
(45)			RadStd		46	97	91	142	196	142	110	79	162	114	55	53	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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BOISE AP, ID, USA

WMO#: **726810**

Lat: **43.567N** Long: **116.241W** Elev: **2814** StdP: **13.26** Time Zone: **-7.00 (NAM)** Period: **90-14** WBAN: **24131**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	12	9.4	15.9	2.1	6.8	12.9	7.7	9.1	22.5	25.0	41.6	23.1	41.5	4.8	120	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	29.8	98.6	63.8	95.7	62.8	92.8	61.9	66.1	92.0	64.7	90.2	63.4	88.4	9.1	330	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	57.5	78.3	71.5	55.1	71.8	71.6	53.0	66.2	70.7	32.1	91.6	31.1	90.0	30.0	88.2	73.0	(3)

Extreme Annual Design Conditions

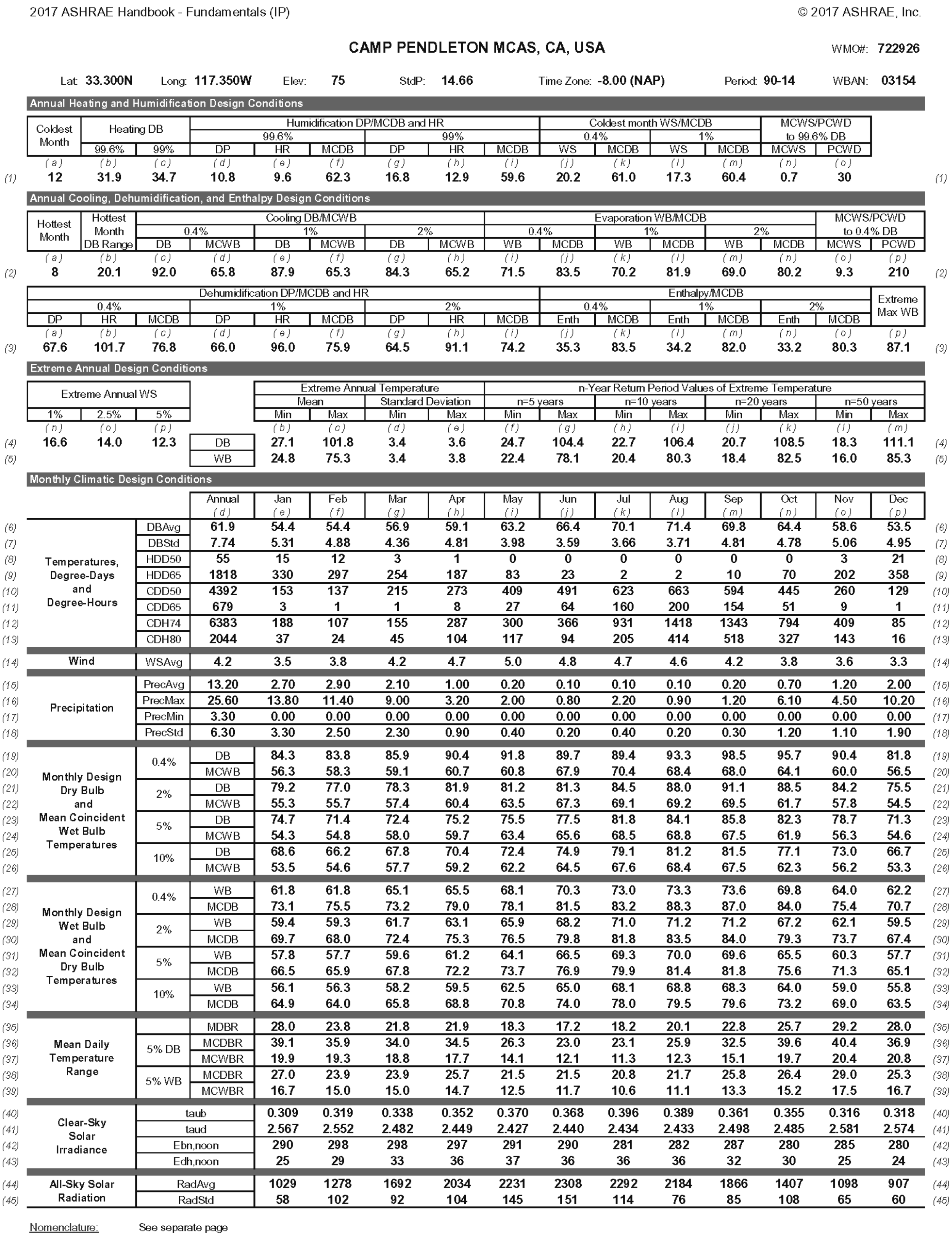
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	22.0	19.1	17.1	DB	4.8	104.3	9.2	2.5	-1.8	106.1	-7.2	107.5	-12.3	108.9	-19.0	110.7	(4)
(5)				WB	4.0	68.8	8.7	1.6	-2.3	70.0	-7.4	71.0	-12.2	71.9	-18.6	73.1	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	Temperatures, Degree-Days and Degree-Hours	DBAvg		52.9	31.9	37.1	44.7	50.5	59.1	67.1	77.0	75.4	66.4	52.9	40.0	31.8	(6)
(7)		DBStd		17.31	7.95	6.92	6.69	7.25	8.17	7.86	6.67	6.61	7.66	8.05	8.38	9.07	(7)
(8)		HDD50		2143	560	362	190	81	14	0	0	0	1	58	312	565	(8)
(9)		HDD65		5414	1025	781	629	439	221	68	5	9	75	383	750	1029	(9)
(10)		CDD50		3212	0	1	26	95	297	512	837	788	494	149	12	1	(10)
(11)		CDD65		1007	0	0	0	3	39	130	377	331	118	9	0	0	(11)
(12)		CDH74		13597	0	0	2	73	605	1809	5223	4277	1473	135	0	0	(12)
(13)		CDH80		6865	0	0	0	13	204	812	2951	2269	585	31	0	0	(13)
(14)	Wind	WSAvg		7.6	6.6	7.7	8.6	8.6	8.3	8.1	7.5	7.2	7.1	7.1	6.9	7.1	(14)
(15)	Precipitation	PrecAvg		12.10	1.40	1.10	1.30	1.20	1.10	0.80	0.40	0.40	0.80	0.70	1.50	1.40	(15)
(16)		PrecMax		16.10	3.90	2.60	2.10	2.80	3.80	2.00	1.10	2.40	2.10	2.00	2.40	3.20	(16)
(17)		PrecMin		9.70	0.40	0.20	0.20	0.20	0.10	0.10	0.00	0.00	0.00	0.20	0.20	0.10	(17)
(18)		PrecStd		1.90	1.00	0.60	0.60	0.70	0.90	0.60	0.30	0.60	0.50	0.50	0.70	0.80	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		53.3	59.6	71.1	81.6	90.9	97.7	103.8	101.3	94.5	84.3	65.6	56.9	(19)
(20)			MCWB		44.2	46.8	50.5	56.2	60.0	63.5	65.4	64.5	61.7	56.9	49.1	45.3	(20)
(21)		2%	DB		48.6	54.5	65.9	75.0	85.7	93.1	99.6	97.6	90.3	77.6	59.5	51.7	(21)
(22)			MCWB		42.0	44.2	48.7	53.5	58.8	61.2	63.9	63.4	60.9	55.1	47.6	44.1	(22)
(23)		5%	DB		45.2	50.9	61.4	69.8	81.0	88.7	97.0	94.8	86.7	72.4	55.5	47.3	(23)
(24)			MCWB		39.8	42.2	46.6	51.2	56.8	60.3	63.4	62.5	59.3	52.6	45.4	41.3	(24)
(25)		10%	DB		42.4	47.7	57.2	64.6	76.1	84.5	93.9	91.7	83.0	67.2	51.8	43.4	(25)
(26)			MCWB		37.9	40.6	44.6	49.1	55.4	58.7	62.4	61.6	57.8	50.4	43.5	38.7	(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		46.1	48.5	51.8	57.6	62.5	65.7	68.1	67.3	66.4	58.8	52.4	47.5	(27)
(28)			MCDB		50.5	57.5	66.2	77.9	84.4	90.5	94.6	95.2	83.6	79.5	61.4	53.4	(28)
(29)		2%	WB		42.7	45.2	49.8	54.8	60.2	63.4	66.4	65.7	63.2	56.0	48.9	44.6	(29)
(30)			MCDB		47.6	52.8	62.9	71.6	81.2	87.6	93.1	92.4	82.9	74.1	56.8	51.1	(30)
(31)		5%	WB		40.4	43.2	47.8	52.4	58.2	61.6	65.2	64.2	60.9	53.6	46.7	42.0	(31)
(32)			MCDB		44.7	49.7	59.6	66.5	77.5	84.1	91.7	90.0	82.2	69.8	54.0	46.9	(32)
(33)		10%	WB		38.2	41.3	45.8	50.3	56.3	59.9	63.8	62.7	59.1	51.4	44.4	39.3	(33)
(34)			MCDB		42.1	47.1	56.0	62.9	73.7	80.8	89.9	87.8	80.7	65.0	51.1	43.0	(34)
(35)	Mean Daily Temperature Range		MDBR		12.5	16.1	19.7	21.7	24.0	26.1	29.8	28.6	26.7	22.6	15.9	12.7	(35)
(36)		5% DB	MCDBR		14.6	19.7	26.2	29.2	31.0	32.0	32.2	31.3	30.4	28.6	20.1	15.0	(36)
(37)			MCWBR		9.6	11.8	13.5	13.3	12.2	11.4	11.4	11.7	12.5	12.8	11.1	9.7	(37)
(38)		5% WB	MCDBR		13.6	18.4	22.8	25.9	28.2	29.3	29.9	28.5	27.7	25.8	17.8	13.6	(38)
(39)			MCWBR		9.2	11.5	12.4	12.4	11.7	10.7	10.7	11.2	11.5	12.3	10.5	9.3	(39)
(40)	Clear-Sky Solar Irradiance	taub			0.262	0.272	0.285	0.310	0.330	0.327	0.339	0.341	0.317	0.290	0.274	0.264	(40)
(41)		taud			2.559	2.585	2.570	2.489	2.453	2.490	2.463	2.437	2.507	2.580	2.568	2.556	(41)
(42)		Ebn,noon			284	300	306	304	298	298	293	290	290	288	276	272	(42)
(43)		Edh,noon			21	24	28	32	35	34	34	34	29	24	20	19	(43)
(44)	All-Sky Solar Radiation	RadAvg			568	889	1281	1695	2075	2354	2392	2056	1616	1073	628	455	(44)
(45)		RadStd			58	70	94	121	140	123	87	94	81	72	45	47	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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CAMP PENDLETON MCAS, CA, USA

WMO#: **722926**

Lat: **33.300N** Long: **117.350W** Elev: **75** StdP: **14.66** Time Zone: **-8.00 (NAP)** Period: **90-14** WBAN: **03154**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	**12**	**31.9**	**34.7**	**10.8**	**9.6**	**62.3**	**16.8**	**12.9**	**59.6**	**20.2**	**61.0**	**17.3**	**60.4**	**0.7**	**30**	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	**8**	**20.1**	**92.0**	**65.8**	**87.9**	**65.3**	**84.3**	**65.2**	**71.5**	**83.5**	**70.2**	**81.9**	**69.0**	**80.2**	**9.3**	**210**	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	**67.6**	**101.7**	**76.8**	**66.0**	**96.0**	**75.9**	**64.5**	**91.1**	**74.2**	**35.3**	**83.5**	**34.2**	**82.0**	**33.2**	**80.3**	**87.1**	(3)

Extreme Annual Design Conditions

	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	**16.6**	**14.0**	**12.3**	DB	**27.1**	**101.8**	**3.4**	**3.6**	**24.7**	**104.4**	**22.7**	**106.4**	**20.7**	**108.5**	**18.3**	**111.1**	(4)
(5)				WB	**24.8**	**75.3**	**3.4**	**3.8**	**22.4**	**78.1**	**20.4**	**80.3**	**18.4**	**82.5**	**16.0**	**85.3**	(5)

Monthly Climatic Design Conditions

				Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	
				(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**		DBAvg	**61.9**	**54.4**	**54.4**	**56.9**	**59.1**	**63.2**	**66.4**	**70.1**	**71.4**	**69.8**	**64.4**	**58.6**	**53.5**	(6)
(7)			DBStd	**7.74**	**5.31**	**4.88**	**4.36**	**4.81**	**3.98**	**3.59**	**3.66**	**3.71**	**4.81**	**4.78**	**5.06**	**4.95**	(7)
(8)			HDD50	**55**	**15**	**12**	**3**	**1**	**0**	**0**	**0**	**0**	**0**	**0**	**3**	**21**	(8)
(9)			HDD65	**1818**	**330**	**297**	**254**	**187**	**83**	**23**	**2**	**2**	**10**	**70**	**202**	**358**	(9)
(10)			CDD50	**4392**	**153**	**137**	**215**	**273**	**409**	**491**	**623**	**663**	**594**	**445**	**260**	**129**	(10)
(11)			CDD65	**679**	**3**	**1**	**1**	**8**	**27**	**64**	**160**	**200**	**154**	**51**	**9**	**1**	(11)
(12)			CDH74	**6383**	**188**	**107**	**155**	**287**	**300**	**366**	**931**	**1418**	**1343**	**794**	**409**	**85**	(12)
(13)			CDH80	**2044**	**37**	**24**	**45**	**104**	**117**	**94**	**205**	**414**	**518**	**327**	**143**	**16**	(13)
(14)	**Wind**		WSAvg	**4.2**	**3.5**	**3.8**	**4.2**	**4.7**	**5.0**	**4.8**	**4.7**	**4.6**	**4.2**	**3.8**	**3.6**	**3.3**	(14)
(15)	**Precipitation**		PrecAvg	**13.20**	**2.70**	**2.90**	**2.10**	**1.00**	**0.20**	**0.10**	**0.10**	**0.10**	**0.20**	**0.70**	**1.20**	**2.00**	(15)
(16)			PrecMax	**25.60**	**13.80**	**11.40**	**9.00**	**3.20**	**2.00**	**0.80**	**2.20**	**0.90**	**1.20**	**6.10**	**4.50**	**10.20**	(16)
(17)			PrecMin	**3.30**	**0.00**	**0.00**	**0.00**	**0.00**	**0.00**	**0.00**	**0.00**	**0.00**	**0.00**	**0.00**	**0.00**	**0.00**	(17)
(18)			PrecStd	**6.30**	**3.30**	**2.50**	**2.30**	**0.90**	**0.40**	**0.20**	**0.40**	**0.20**	**0.30**	**1.20**	**1.10**	**1.90**	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB		**84.3**	**83.8**	**85.9**	**90.4**	**91.8**	**89.7**	**89.4**	**93.3**	**98.5**	**95.7**	**90.4**	**81.8**	(19)
(20)			MCWB		**56.3**	**58.3**	**59.1**	**60.7**	**60.8**	**67.9**	**70.4**	**68.4**	**68.0**	**64.1**	**60.0**	**56.5**	(20)
(21)		2%	DB		**79.2**	**77.0**	**78.3**	**81.9**	**81.2**	**81.3**	**84.5**	**88.0**	**91.1**	**88.5**	**84.2**	**75.5**	(21)
(22)			MCWB		**55.3**	**55.7**	**57.4**	**60.4**	**63.5**	**67.3**	**69.1**	**69.2**	**69.5**	**61.7**	**57.8**	**54.5**	(22)
(23)		5%	DB		**74.7**	**71.4**	**72.4**	**75.2**	**75.5**	**77.5**	**81.8**	**84.1**	**85.8**	**82.3**	**78.7**	**71.3**	(23)
(24)			MCWB		**54.3**	**54.8**	**58.0**	**59.7**	**63.4**	**65.6**	**68.5**	**68.8**	**67.5**	**61.9**	**56.3**	**54.6**	(24)
(25)		10%	DB		**68.6**	**66.2**	**67.8**	**70.4**	**72.4**	**74.9**	**79.1**	**81.2**	**81.5**	**77.1**	**73.0**	**66.7**	(25)
(26)			MCWB		**53.5**	**54.6**	**57.7**	**59.2**	**62.2**	**64.5**	**67.6**	**68.4**	**67.5**	**62.3**	**56.2**	**53.3**	(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB		**61.8**	**61.8**	**65.1**	**65.5**	**68.1**	**70.3**	**73.0**	**73.3**	**73.6**	**69.8**	**64.0**	**62.2**	(27)
(28)			MCDB		**73.1**	**75.5**	**73.2**	**79.0**	**78.1**	**81.5**	**83.2**	**88.3**	**87.0**	**84.0**	**75.4**	**70.7**	(28)
(29)		2%	WB		**59.4**	**59.3**	**61.7**	**63.1**	**65.9**	**68.2**	**71.0**	**71.2**	**71.2**	**67.2**	**62.1**	**59.5**	(29)
(30)			MCDB		**69.7**	**68.0**	**72.4**	**75.3**	**76.5**	**79.8**	**81.8**	**83.5**	**84.0**	**79.3**	**73.7**	**67.4**	(30)
(31)		5%	WB		**57.8**	**57.7**	**59.6**	**61.2**	**64.1**	**66.5**	**69.3**	**70.0**	**69.6**	**65.5**	**60.3**	**57.7**	(31)
(32)			MCDB		**66.5**	**65.9**	**67.8**	**72.2**	**73.7**	**76.9**	**79.9**	**81.4**	**81.8**	**75.6**	**71.3**	**65.1**	(32)
(33)		10%	WB		**56.1**	**56.3**	**58.2**	**59.5**	**62.5**	**65.0**	**68.1**	**68.8**	**68.3**	**64.0**	**59.0**	**55.8**	(33)
(34)			MCDB		**64.9**	**64.0**	**65.8**	**68.8**	**70.8**	**74.0**	**78.0**	**79.5**	**79.6**	**73.2**	**69.0**	**63.5**	(34)
(35)	**Mean Daily Temperature Range**		MDBR		**28.0**	**23.8**	**21.8**	**21.9**	**18.3**	**17.2**	**18.2**	**20.1**	**22.8**	**25.7**	**29.2**	**28.0**	(35)
(36)		5% DB	MCDBR		**39.1**	**35.9**	**34.0**	**34.5**	**26.3**	**23.0**	**23.1**	**25.9**	**32.5**	**39.6**	**40.4**	**36.9**	(36)
(37)			MCWBR		**19.9**	**19.3**	**18.8**	**17.7**	**14.1**	**12.1**	**11.3**	**12.3**	**15.1**	**19.7**	**20.4**	**20.8**	(37)
(38)		5% WB	MCDBR		**27.0**	**23.9**	**23.9**	**25.7**	**21.5**	**21.5**	**20.8**	**21.7**	**25.8**	**26.4**	**29.0**	**25.3**	(38)
(39)			MCWBR		**16.7**	**15.0**	**15.0**	**14.7**	**12.5**	**11.7**	**10.6**	**11.1**	**13.3**	**15.2**	**17.5**	**16.7**	(39)
(40)	**Clear-Sky Solar Irradiance**	taub			**0.309**	**0.319**	**0.338**	**0.352**	**0.370**	**0.368**	**0.396**	**0.389**	**0.361**	**0.355**	**0.316**	**0.318**	(40)
(41)		taud			**2.567**	**2.552**	**2.482**	**2.449**	**2.427**	**2.440**	**2.434**	**2.433**	**2.498**	**2.485**	**2.581**	**2.574**	(41)
(42)		Ebn,noon			**290**	**298**	**298**	**297**	**291**	**290**	**281**	**282**	**287**	**280**	**285**	**280**	(42)
(43)		Edh,noon			**25**	**29**	**33**	**36**	**37**	**36**	**36**	**36**	**32**	**30**	**25**	**24**	(43)
(44)	**All-Sky Solar Radiation**	RadAvg			**1029**	**1278**	**1692**	**2034**	**2231**	**2308**	**2292**	**2184**	**1866**	**1407**	**1098**	**907**	(44)
(45)		RadStd			**58**	**102**	**92**	**104**	**145**	**151**	**114**	**76**	**85**	**108**	**65**	**60**	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals

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THE EASTERN IOWA AP, IA, USA

WMO#: **725450**

Lat: **41.883N** Long: **91.717W** Elev: **868** StdP: **14.24** Time Zone: **-6.00 (NAC)** Period: **90-14** WBAN: **14990**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	**1**	**-8.2**	**-2.6**	**-14.5**	**2.6**	**-7.5**	**-9.5**	**3.4**	**-2.1**	**32.7**	**20.7**	**28.6**	**17.8**	**10.2**	**310**	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	**7**	**18.9**	**90.4**	**76.3**	**87.6**	**74.5**	**84.8**	**72.7**	**78.6**	**86.7**	**76.9**	**84.6**	**75.1**	**82.3**	**10.6**	**180**	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	**76.3**	**141.9**	**84.1**	**74.5**	**133.2**	**82.0**	**72.7**	**125.4**	**80.0**	**42.8**	**86.9**	**40.9**	**84.2**	**39.1**	**82.7**	**88.3**	(3)

Extreme Annual Design Conditions

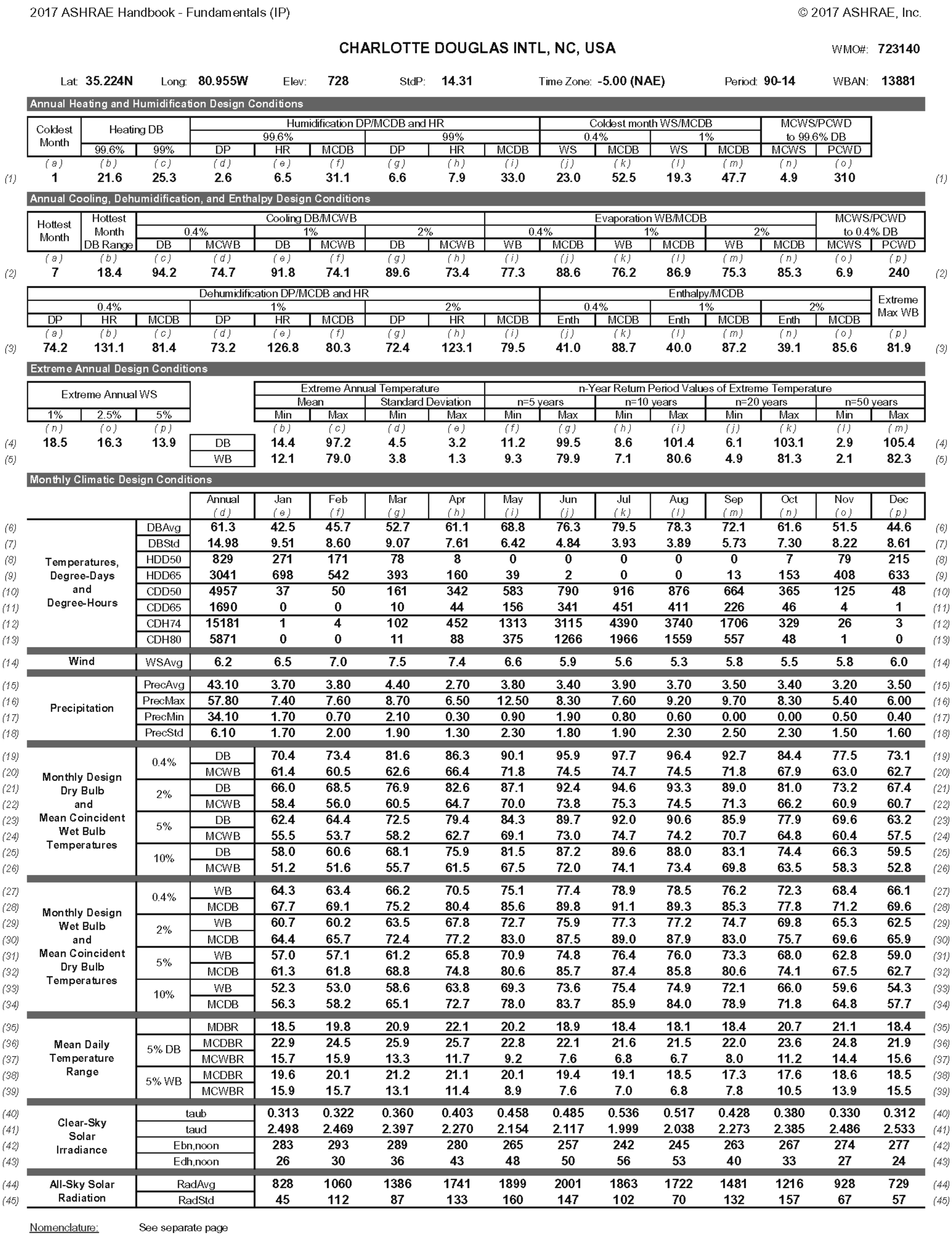
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	**26.7**	**23.7**	**20.5**	DB	**-15.2**	**94.2**	**5.7**	**3.5**	**-19.3**	**96.7**	**-22.7**	**98.8**	**-25.9**	**100.7**	**-30.1**	**103.2**	(4)
(5)				WB	**-15.6**	**81.8**	**5.7**	**1.8**	**-19.7**	**83.1**	**-23.0**	**84.2**	**-26.2**	**85.2**	**-30.3**	**86.5**	(5)

Monthly Climatic Design Conditions

				Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	
				(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**		DBAvg	**48.6**	**20.2**	**24.8**	**36.7**	**49.3**	**60.3**	**69.8**	**73.0**	**71.2**	**63.1**	**51.2**	**37.4**	**24.8**	(6)
(7)			DBStd	**20.86**	**12.42**	**12.08**	**11.75**	**9.62**	**8.34**	**6.45**	**5.67**	**5.52**	**8.20**	**9.57**	**10.50**	**11.79**	(7)
(8)			HDD50	**3490**	**923**	**707**	**438**	**125**	**12**	**0**	**0**	**0**	**5**	**102**	**395**	**783**	(8)
(9)			HDD65	**6755**	**1388**	**1127**	**878**	**479**	**195**	**28**	**5**	**11**	**133**	**438**	**827**	**1246**	(9)
(10)			CDD50	**2984**	**0**	**0**	**27**	**104**	**331**	**595**	**712**	**658**	**397**	**140**	**18**	**2**	(10)
(11)			CDD65	**774**	**0**	**0**	**1**	**8**	**49**	**173**	**252**	**204**	**75**	**12**	**0**	**0**	(11)
(12)			CDH74	**6764**	**0**	**0**	**10**	**107**	**435**	**1500**	**2263**	**1645**	**687**	**116**	**1**	**0**	(12)
(13)			CDH80	**2099**	**0**	**0**	**0**	**24**	**100**	**463**	**789**	**498**	**200**	**25**	**0**	**0**	(13)
(14)	**Wind**		WSAvg	**10.0**	**11.5**	**11.3**	**11.6**	**12.0**	**10.5**	**8.8**	**7.5**	**6.9**	**8.0**	**9.8**	**10.9**	**10.7**	(14)
(15)	**Precipitation**		PrecAvg	**34.40**	**1.00**	**1.00**	**2.20**	**3.30**	**3.90**	**4.60**	**4.50**	**4.10**	**3.70**	**2.30**	**2.10**	**1.60**	(15)
(16)			PrecMax	**60.50**	**2.60**	**3.30**	**6.10**	**5.70**	**8.70**	**11.60**	**17.00**	**13.10**	**10.30**	**5.60**	**6.40**	**3.60**	(16)
(17)			PrecMin	**19.30**	**0.00**	**0.10**	**0.10**	**0.70**	**0.30**	**0.10**	**0.40**	**0.40**	**0.40**	**0.00**	**0.20**	**0.20**	(17)
(18)			PrecStd	**9.10**	**0.70**	**0.70**	**1.40**	**1.40**	**2.10**	**2.70**	**3.40**	**2.90**	**2.30**	**1.40**	**1.40**	**0.90**	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB	**52.9**	**57.1**	**75.2**	**83.7**	**87.4**	**91.4**	**95.6**	**93.1**	**89.8**	**83.4**	**70.2**	**57.6**		(19)
(20)			MCWB	**45.7**	**49.6**	**61.6**	**65.1**	**70.6**	**75.5**	**78.4**	**76.5**	**74.1**	**68.0**	**58.0**	**52.9**		(20)
(21)		2%	DB	**44.5**	**50.3**	**66.9**	**76.7**	**82.7**	**88.3**	**90.8**	**88.7**	**85.3**	**76.9**	**63.1**	**50.1**		(21)
(22)			MCWB	**40.2**	**44.9**	**56.1**	**61.4**	**67.5**	**73.7**	**77.9**	**76.4**	**71.1**	**63.2**	**55.0**	**45.8**		(22)
(23)		5%	DB	**39.2**	**44.2**	**61.2**	**70.7**	**79.0**	**85.4**	**87.5**	**85.4**	**81.7**	**72.1**	**57.4**	**44.3**		(23)
(24)			MCWB	**36.2**	**39.8**	**54.2**	**58.1**	**65.4**	**72.2**	**76.0**	**74.7**	**69.0**	**61.1**	**51.1**	**40.9**		(24)
(25)		10%	DB	**35.6**	**39.3**	**54.9**	**65.5**	**75.2**	**82.2**	**84.2**	**82.2**	**77.7**	**67.0**	**53.0**	**39.5**		(25)
(26)			MCWB	**33.4**	**36.3**	**47.8**	**54.9**	**63.8**	**70.3**	**73.7**	**72.5**	**66.6**	**58.0**	**47.7**	**36.5**		(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB	**46.3**	**50.6**	**63.1**	**66.8**	**73.7**	**78.5**	**82.1**	**80.9**	**76.6**	**70.4**	**60.5**	**54.7**		(27)
(28)			MCDB	**48.6**	**54.8**	**72.7**	**79.1**	**83.2**	**87.8**	**90.5**	**89.0**	**85.9**	**78.8**	**66.8**	**56.9**		(28)
(29)		2%	WB	**40.0**	**45.3**	**59.2**	**63.2**	**71.0**	**76.5**	**79.5**	**78.1**	**73.5**	**66.2**	**56.6**	**47.2**		(29)
(30)			MCDB	**44.3**	**49.9**	**64.5**	**72.6**	**78.8**	**85.1**	**87.8**	**85.4**	**81.9**	**73.4**	**61.5**	**48.9**		(30)
(31)		5%	WB	**36.5**	**40.2**	**54.8**	**60.1**	**68.3**	**74.5**	**77.4**	**76.3**	**71.2**	**63.1**	**52.5**	**41.0**		(31)
(32)			MCDB	**39.1**	**43.8**	**60.9**	**69.0**	**75.9**	**82.5**	**85.1**	**83.1**	**78.3**	**68.7**	**56.9**	**43.8**		(32)
(33)		10%	WB	**33.4**	**36.5**	**48.4**	**56.3**	**65.8**	**72.5**	**75.4**	**74.5**	**68.9**	**59.5**	**47.6**	**36.8**		(33)
(34)			MCDB	**35.2**	**39.2**	**54.5**	**64.2**	**72.7**	**79.9**	**82.5**	**80.4**	**75.4**	**66.7**	**52.1**	**39.4**		(34)
(35)	**Mean Daily Temperature Range**		MDBR	**15.8**	**15.8**	**18.3**	**20.7**	**20.0**	**19.0**	**18.9**	**19.2**	**22.0**	**20.7**	**17.6**	**15.2**		(35)
(36)		5% DB	MCDBR	**20.2**	**20.7**	**26.3**	**28.7**	**23.8**	**21.7**	**21.2**	**21.5**	**25.4**	**26.5**	**23.8**	**20.0**		(36)
(37)			MCWBR	**16.1**	**15.4**	**16.6**	**16.3**	**12.5**	**10.3**	**10.7**	**10.9**	**12.4**	**14.6**	**16.7**	**15.9**		(37)
(38)		5% WB	MCDBR	**18.5**	**20.0**	**23.2**	**25.1**	**20.5**	**19.1**	**19.2**	**18.9**	**21.1**	**21.8**	**21.7**	**18.7**		(38)
(39)			MCWBR	**15.2**	**15.4**	**15.7**	**15.5**	**12.5**	**10.6**	**10.9**	**11.0**	**11.9**	**14.6**	**17.2**	**15.7**		(39)
(40)	**Clear-Sky Solar Irradiance**	taub		**0.286**	**0.304**	**0.331**	**0.377**	**0.416**	**0.422**	**0.436**	**0.416**	**0.385**	**0.338**	**0.299**	**0.286**		(40)
(41)		taud		**2.400**	**2.361**	**2.390**	**2.278**	**2.235**	**2.268**	**2.213**	**2.277**	**2.326**	**2.460**	**2.494**	**2.427**		(41)
(42)		Ebn,noon		**275**	**286**	**291**	**283**	**273**	**271**	**266**	**267**	**268**	**271**	**269**	**265**		(42)
(43)		Edh,noon		**25**	**30**	**34**	**41**	**43**	**42**	**44**	**40**	**36**	**28**	**23**	**23**		(43)
(44)	**All-Sky Solar Radiation**	RadAvg		**593**	**890**	**1200**	**1492**	**1755**	**1933**	**1980**	**1755**	**1448**	**964**	**623**	**456**		(44)
(45)		RadStd		**88**	**108**	**101**	**115**	**128**	**132**	**146**	**112**	**105**	**111**	**51**	**49**		(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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CHARLOTTE DOUGLAS INTL, NC, USA WMO#: **723140**

Lat: **35.224N** Long: **80.955W** Elev: **728** StdP: **14.31** Time Zone: **-5.00 (NAE)** Period: **90-14** WBAN: **13881**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	**1**	**21.6**	**25.3**	**2.6**	**6.5**	**31.1**	**6.6**	**7.9**	**33.0**	**23.0**	**52.5**	**19.3**	**47.7**	**4.9**	**310**	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	**7**	**18.4**	**94.2**	**74.7**	**91.8**	**74.1**	**89.6**	**73.4**	**77.3**	**88.6**	**76.2**	**86.9**	**75.3**	**85.3**	**6.9**	**240**	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	**74.2**	**131.1**	**81.4**	**73.2**	**126.8**	**80.3**	**72.4**	**123.1**	**79.5**	**41.0**	**88.7**	**40.0**	**87.2**	**39.1**	**85.6**	**81.9**	(3)

Extreme Annual Design Conditions

	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	**18.5**	**16.3**	**13.9**	DB	**14.4**	**97.2**	**4.5**	**3.2**	**11.2**	**99.5**	**8.6**	**101.4**	**6.1**	**103.1**	**2.9**	**105.4**	(4)
(5)				WB	**12.1**	**79.0**	**3.8**	**1.3**	**9.3**	**79.9**	**7.1**	**80.6**	**4.9**	**81.3**	**2.1**	**82.3**	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**		DBAvg	**61.3**	**42.5**	**45.7**	**52.7**	**61.1**	**68.8**	**76.3**	**79.5**	**78.3**	**72.1**	**61.6**	**51.5**	**44.6**	(6)
(7)			DBStd	**14.98**	**9.51**	**8.60**	**9.07**	**7.61**	**6.42**	**4.84**	**3.93**	**3.89**	**5.73**	**7.30**	**8.22**	**8.61**	(7)
(8)			HDD50	**829**	**271**	**171**	**78**	**8**	**0**	**0**	**0**	**0**	**0**	**7**	**79**	**215**	(8)
(9)			HDD65	**3041**	**698**	**542**	**393**	**160**	**39**	**2**	**0**	**0**	**13**	**153**	**408**	**633**	(9)
(10)			CDD50	**4957**	**37**	**50**	**161**	**342**	**583**	**790**	**916**	**876**	**664**	**365**	**125**	**48**	(10)
(11)			CDD65	**1690**	**0**	**0**	**10**	**44**	**156**	**341**	**451**	**411**	**226**	**46**	**4**	**1**	(11)
(12)			CDH74	**15181**	**1**	**4**	**102**	**452**	**1313**	**3115**	**4390**	**3740**	**1706**	**329**	**26**	**3**	(12)
(13)			CDH80	**5871**	**0**	**0**	**11**	**88**	**375**	**1266**	**1966**	**1559**	**557**	**48**	**1**	**0**	(13)
(14)	**Wind**		WSAvg	**6.2**	**6.5**	**7.0**	**7.5**	**7.4**	**6.6**	**5.9**	**5.6**	**5.3**	**5.8**	**5.5**	**5.8**	**6.0**	(14)
(15)	**Precipitation**		PrecAvg	**43.10**	**3.70**	**3.80**	**4.40**	**2.70**	**3.80**	**3.40**	**3.90**	**3.70**	**3.50**	**3.40**	**3.20**	**3.50**	(15)
(16)			PrecMax	**57.80**	**7.40**	**7.60**	**8.70**	**6.50**	**12.50**	**8.30**	**7.60**	**9.20**	**9.70**	**8.30**	**5.40**	**6.00**	(16)
(17)			PrecMin	**34.10**	**1.70**	**0.70**	**2.10**	**0.30**	**0.90**	**1.90**	**0.80**	**0.60**	**0.00**	**0.00**	**0.50**	**0.40**	(17)
(18)			PrecStd	**6.10**	**1.70**	**2.00**	**1.90**	**1.30**	**2.30**	**1.80**	**1.90**	**2.30**	**2.50**	**2.30**	**1.50**	**1.60**	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB		**70.4**	**73.4**	**81.6**	**86.3**	**90.1**	**95.9**	**97.7**	**96.4**	**92.7**	**84.4**	**77.5**	**73.1**	(19)
(20)			MCWB		**61.4**	**60.5**	**62.6**	**66.4**	**71.8**	**74.5**	**74.7**	**74.5**	**71.8**	**67.9**	**63.0**	**62.7**	(20)
(21)		2%	DB		**66.0**	**68.5**	**76.9**	**82.6**	**87.1**	**92.4**	**94.6**	**93.3**	**89.0**	**81.0**	**73.2**	**67.4**	(21)
(22)			MCWB		**58.4**	**56.0**	**60.5**	**64.7**	**70.0**	**73.8**	**75.3**	**74.5**	**71.3**	**66.2**	**60.9**	**60.7**	(22)
(23)		5%	DB		**62.4**	**64.4**	**72.5**	**79.4**	**84.3**	**89.7**	**92.0**	**90.6**	**85.9**	**77.9**	**69.6**	**63.2**	(23)
(24)			MCWB		**55.5**	**53.7**	**58.2**	**62.7**	**69.1**	**73.0**	**74.7**	**74.2**	**70.7**	**64.8**	**60.4**	**57.5**	(24)
(25)		10%	DB		**58.0**	**60.6**	**68.1**	**75.9**	**81.5**	**87.2**	**89.6**	**88.0**	**83.1**	**74.4**	**66.3**	**59.5**	(25)
(26)			MCWB		**51.2**	**51.6**	**55.7**	**61.5**	**67.5**	**72.0**	**74.1**	**73.4**	**69.8**	**63.5**	**58.3**	**52.8**	(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB		**64.3**	**63.4**	**66.2**	**70.5**	**75.1**	**77.4**	**78.9**	**78.5**	**76.2**	**72.3**	**68.4**	**66.1**	(27)
(28)			MCDB		**67.7**	**69.1**	**75.2**	**80.4**	**85.6**	**89.8**	**91.1**	**89.3**	**85.3**	**77.8**	**71.2**	**69.6**	(28)
(29)		2%	WB		**60.7**	**60.2**	**63.5**	**67.8**	**72.7**	**75.9**	**77.3**	**77.2**	**74.7**	**69.8**	**65.3**	**62.5**	(29)
(30)			MCDB		**64.4**	**65.7**	**72.4**	**77.2**	**83.0**	**87.5**	**89.0**	**87.9**	**83.0**	**75.7**	**69.6**	**65.9**	(30)
(31)		5%	WB		**57.0**	**57.1**	**61.2**	**65.8**	**70.9**	**74.8**	**76.4**	**76.0**	**73.3**	**68.0**	**62.8**	**59.0**	(31)
(32)			MCDB		**61.3**	**61.8**	**68.8**	**74.8**	**80.6**	**85.7**	**87.4**	**85.8**	**80.6**	**74.1**	**67.5**	**62.7**	(32)
(33)		10%	WB		**52.3**	**53.0**	**58.6**	**63.8**	**69.3**	**73.6**	**75.4**	**74.9**	**72.1**	**66.0**	**59.6**	**54.3**	(33)
(34)			MCDB		**56.3**	**58.2**	**65.1**	**72.7**	**78.0**	**83.7**	**85.9**	**84.0**	**78.9**	**71.8**	**64.8**	**57.7**	(34)
(35)	**Mean Daily Temperature Range**		MDBR		**18.5**	**19.8**	**20.9**	**22.1**	**20.2**	**18.9**	**18.4**	**18.1**	**18.4**	**20.7**	**21.1**	**18.4**	(35)
(36)		5% DB	MCDBR		**22.9**	**24.5**	**25.9**	**25.7**	**22.8**	**22.1**	**21.6**	**21.5**	**22.0**	**23.6**	**24.8**	**21.9**	(36)
(37)			MCWBR		**15.7**	**15.9**	**13.3**	**11.7**	**9.2**	**7.6**	**6.8**	**6.7**	**8.0**	**11.2**	**14.4**	**15.6**	(37)
(38)		5% WB	MCDBR		**19.6**	**20.1**	**21.2**	**21.1**	**20.1**	**19.4**	**19.1**	**18.5**	**17.3**	**17.6**	**18.6**	**18.5**	(38)
(39)			MCWBR		**15.9**	**15.7**	**13.1**	**11.4**	**8.9**	**7.6**	**7.0**	**6.8**	**7.8**	**10.5**	**13.9**	**15.5**	(39)
(40)	**Clear-Sky Solar Irradiance**	taub			**0.313**	**0.322**	**0.360**	**0.403**	**0.458**	**0.485**	**0.536**	**0.517**	**0.428**	**0.380**	**0.330**	**0.312**	(40)
(41)		taud			**2.498**	**2.469**	**2.397**	**2.270**	**2.154**	**2.117**	**1.999**	**2.038**	**2.273**	**2.385**	**2.486**	**2.533**	(41)
(42)		Ebn,noon			**283**	**293**	**289**	**280**	**265**	**257**	**242**	**245**	**263**	**267**	**274**	**277**	(42)
(43)		Edh,noon			**26**	**30**	**36**	**43**	**48**	**50**	**56**	**53**	**40**	**33**	**27**	**24**	(43)
(44)	**All-Sky Solar Radiation**	RadAvg			**828**	**1060**	**1386**	**1741**	**1899**	**2001**	**1863**	**1722**	**1481**	**1216**	**928**	**729**	(44)
(45)		RadStd			**45**	**112**	**87**	**133**	**160**	**147**	**102**	**70**	**132**	**157**	**67**	**57**	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals

2017 ASHRAE Handbook - Fundamentals (IP)

CHICAGO O'HARE INTL, IL, USA WMO#: **725300**

Lat: **41.995N** Long: **87.934W** Elev: **662** StdP: **14.35** Time Zone: **-6.00 (NAC)** Period: **90-14** WBAN: **94846**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	-1.0	4.4	-10.7	3.2	0.8	-6.0	4.1	5.6	26.9	30.6	24.9	30.0	10.4	270	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	17.9	91.3	74.2	88.5	72.9	85.9	71.6	77.5	87.3	75.7	84.7	74.1	82.6	11.4	220	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	74.5	132.2	83.5	72.8	124.7	81.4	71.1	117.7	79.4	41.3	87.1	39.6	85.3	37.9	82.9	83.3	(3)

Extreme Annual Design Conditions

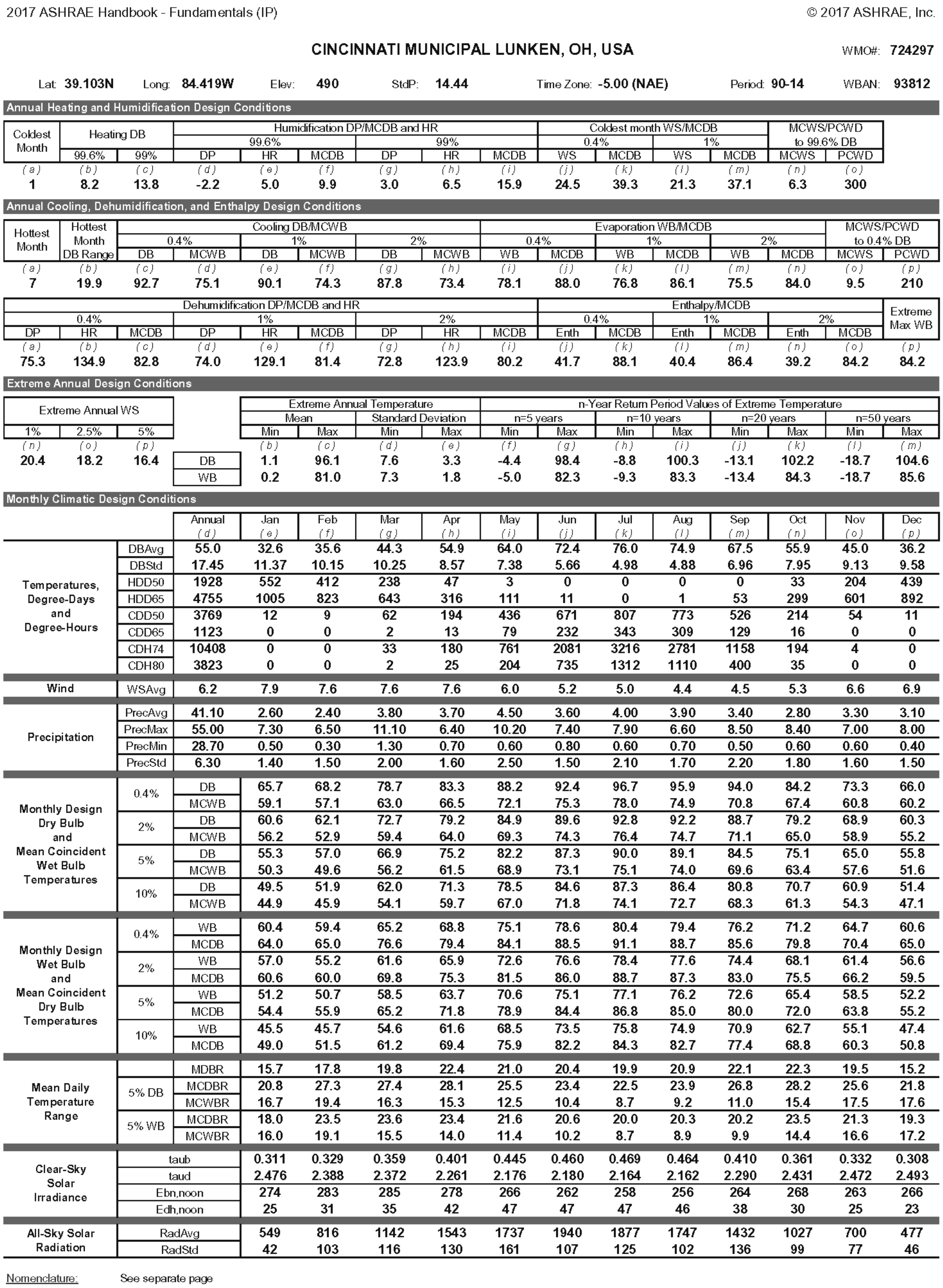
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	24.6	21.0	19.1	DB	-7.3	95.8	6.4	3.8	-11.8	98.5	-15.6	100.7	-19.1	102.8	-23.8	105.5	(4)
(5)				WB	-8.0	80.0	6.1	1.9	-12.4	81.4	-16.0	82.5	-19.4	83.5	-23.8	84.9	(5)

Monthly Climatic Design Conditions

				Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	
				(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**		DBAvg	50.5	24.6	28.2	38.2	49.2	59.6	69.5	74.2	72.7	64.9	53.1	40.6	29.3	(6)
(7)			DBStd	19.51	11.26	10.61	11.05	9.26	8.86	7.56	6.02	5.54	7.78	8.69	9.56	10.52	(7)
(8)			HDD50	2955	788	612	397	123	14	0	0	0	2	67	307	645	(8)
(9)			HDD65	6190	1253	1030	832	483	221	42	3	6	98	381	733	1108	(9)
(10)			CDD50	3122	0	2	33	98	310	585	751	704	448	163	25	3	(10)
(11)			CDD65	882	0	0	3	9	52	177	290	245	93	13	0	0	(11)
(12)			CDH74	7871	0	0	20	97	531	1715	2732	1921	754	101	0	0	(12)
(13)			CDH80	2622	0	0	3	18	155	592	1026	602	209	17	0	0	(13)
(14)	**Wind**		WSAvg	9.8	11.1	10.9	11.0	11.3	10.0	8.7	8.3	7.8	8.3	9.7	10.6	10.3	(14)
(15)	**Precipitation**		PrecAvg	35.80	1.50	1.40	2.70	3.60	3.30	3.80	3.70	4.20	3.80	2.40	2.90	2.50	(15)
(16)			PrecMax	45.60	4.10	3.10	5.90	5.50	7.10	8.70	7.60	8.50	14.20	4.70	5.00	5.40	(16)
(17)			PrecMin	20.80	0.30	0.20	1.00	1.00	1.40	1.60	1.20	0.70	0.00	0.20	0.60	0.30	(17)
(18)			PrecStd	7.00	1.20	0.80	1.20	1.30	1.50	1.90	1.80	2.70	3.50	1.20	1.10	1.40	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB		56.8	59.4	76.7	82.5	88.7	92.9	96.9	93.1	90.0	82.6	68.8	60.3	(19)
(20)			MCWB		53.8	51.0	63.1	64.2	70.0	73.3	76.5	76.2	72.8	67.5	57.3	56.9	(20)
(21)		2%	DB		49.0	52.4	68.6	76.1	84.4	89.8	92.1	89.4	85.4	76.4	63.6	52.9	(21)
(22)			MCWB		44.8	46.4	58.2	61.7	68.2	72.4	75.9	74.5	69.9	63.8	56.4	49.1	(22)
(23)		5%	DB		43.1	45.9	61.9	70.8	80.3	86.4	88.6	86.1	81.9	72.1	60.0	47.6	(23)
(24)			MCWB		39.4	41.0	54.0	58.4	66.0	71.1	73.8	73.0	68.1	61.3	53.9	43.6	(24)
(25)		10%	DB		39.1	41.6	55.6	65.5	75.6	83.3	85.7	83.1	78.3	67.7	55.5	43.0	(25)
(26)			MCWB		36.4	38.0	48.5	54.5	63.7	69.9	72.3	70.8	66.2	59.2	49.6	39.5	(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB		54.3	53.2	64.3	67.0	74.2	77.5	80.4	79.2	75.4	70.3	61.5	57.7	(27)
(28)			MCDB		56.1	55.2	74.5	78.9	84.7	86.6	90.6	89.5	84.9	78.2	64.9	59.7	(28)
(29)		2%	WB		45.8	46.7	60.1	63.6	71.0	75.2	78.1	76.9	72.6	66.4	58.1	50.2	(29)
(30)			MCDB		48.0	51.4	66.5	71.9	79.8	85.3	88.0	85.8	80.7	72.8	62.8	51.9	(30)
(31)		5%	WB		40.1	41.6	54.8	59.8	68.4	73.3	76.4	75.0	70.8	63.4	54.0	44.2	(31)
(32)			MCDB		42.5	45.5	61.0	69.7	76.9	83.3	85.9	82.8	78.2	70.1	58.8	47.1	(32)
(33)		10%	WB		36.5	38.0	48.5	55.8	65.3	71.4	74.5	73.3	68.7	60.1	50.3	39.7	(33)
(34)			MCDB		39.0	41.3	55.0	64.5	73.7	80.9	83.3	80.2	75.9	66.5	55.0	42.6	(34)
(35)	**Mean Daily Temperature Range**		MDBR		13.3	13.7	15.9	18.3	19.6	19.4	17.9	16.8	18.5	17.5	14.4	12.5	(35)
(36)		5% DB	MCDBR		16.8	19.4	24.3	26.2	24.6	22.4	20.4	19.9	21.9	23.6	20.0	16.7	(36)
(37)			MCWBR		14.6	14.9	16.3	15.4	12.7	10.4	8.9	8.8	9.7	13.4	15.0	14.0	(37)
(38)		5% WB	MCDBR		16.2	17.9	22.3	23.7	21.9	20.2	18.8	17.1	18.4	20.2	18.0	15.3	(38)
(39)			MCWBR		14.9	14.6	16.6	15.8	13.0	11.1	9.7	8.9	9.7	13.7	15.3	14.2	(39)
(40)	**Clear-Sky Solar Irradiance**		taub		0.298	0.314	0.346	0.401	0.428	0.438	0.445	0.439	0.410	0.357	0.315	0.297	(40)
(41)			taud		2.436	2.385	2.384	2.254	2.233	2.250	2.231	2.245	2.294	2.443	2.509	2.472	(41)
(42)			Ebn,noon		270	283	285	275	270	266	263	261	260	263	262	260	(42)
(43)			Edh,noon		24	30	34	42	44	43	43	42	37	28	23	22	(43)
(44)	**All-Sky Solar Radiation**		RadAvg		513	784	1147	1481	1693	1944	1945	1716	1393	928	567	425	(44)
(45)			RadStd		59	97	72	123	129	120	108	104	126	105	49	42	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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CINCINNATI MUNICIPAL LUNKEN, OH, USA WMO#: **724297**

Lat: **39.103N** Long: **84.419W** Elev: **490** StdP: **14.44** Time Zone: **-5.00 (NAE)** Period: **90-14** WBAN: **93812**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	8.2	13.8	-2.2	5.0	9.9	3.0	6.5	15.9	24.5	39.3	21.3	37.1	6.3	300	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	19.9	92.7	75.1	90.1	74.3	87.8	73.4	78.1	88.0	76.8	86.1	75.5	84.0	9.5	210	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	75.3	134.9	82.8	74.0	129.1	81.4	72.8	123.9	80.2	41.7	88.1	40.4	86.4	39.2	84.2	84.2	(3)

Extreme Annual Design Conditions

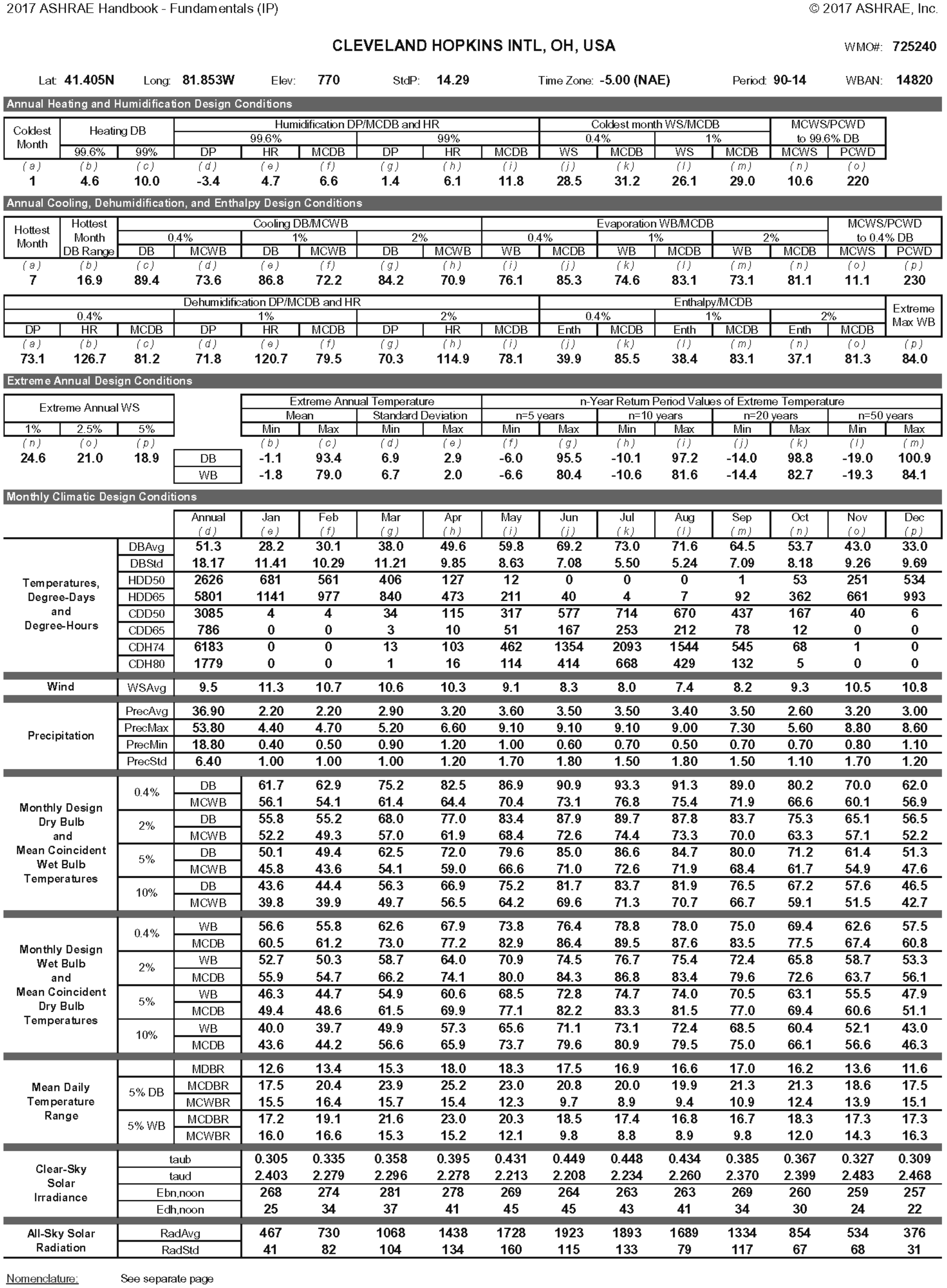
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	20.4	18.2	16.4	DB	1.1	96.1	7.6	3.3	-4.4	98.4	-8.8	100.3	-13.1	102.2	-18.7	104.6	(4)
(5)				WB	0.2	81.0	7.3	1.8	-5.0	82.3	-9.3	83.3	-13.4	84.3	-18.7	85.6	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	55.0	32.6	35.6	44.3	54.9	64.0	72.4	76.0	74.9	67.5	55.9	45.0	36.2	(6)
(7)			DBStd	17.45	11.37	10.15	10.25	8.57	7.38	5.66	4.98	4.88	6.96	7.95	9.13	9.58	(7)
(8)			HDD50	1928	552	412	238	47	3	0	0	0	0	33	204	439	(8)
(9)			HDD65	4755	1005	823	643	316	111	11	0	1	53	299	601	892	(9)
(10)			CDD50	3769	12	9	62	194	436	671	807	773	526	214	54	11	(10)
(11)			CDD65	1123	0	0	2	13	79	232	343	309	129	16	0	0	(11)
(12)			CDH74	10408	0	0	33	180	761	2081	3216	2781	1158	194	4	0	(12)
(13)			CDH80	3823	0	0	2	25	204	735	1312	1110	400	35	0	0	(13)
(14)	Wind		WSAvg	6.2	7.9	7.6	7.6	7.6	6.0	5.2	5.0	4.4	4.5	5.3	6.6	6.9	(14)
(15)	Precipitation		PrecAvg	41.10	2.60	2.40	3.80	3.70	4.50	3.60	4.00	3.90	3.40	2.80	3.30	3.10	(15)
(16)			PrecMax	55.00	7.30	6.50	11.10	6.40	10.20	7.40	7.90	6.60	8.50	8.40	7.00	8.00	(16)
(17)			PrecMin	28.70	0.50	0.30	1.30	0.70	0.60	0.80	0.60	0.70	0.50	0.60	0.60	0.40	(17)
(18)			PrecStd	6.30	1.40	1.50	2.00	1.60	2.50	1.50	2.10	1.70	2.20	1.80	1.60	1.50	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		65.7	68.2	78.7	83.3	88.2	92.4	96.7	95.9	94.0	84.2	73.3	66.0	(19)
(20)			MCWB		59.1	57.1	63.0	66.5	72.1	75.3	78.0	74.9	70.8	67.4	60.8	60.2	(20)
(21)		2%	DB		60.6	62.1	72.7	79.2	84.9	89.6	92.8	92.2	88.7	79.2	68.9	60.3	(21)
(22)			MCWB		56.2	52.9	59.4	64.0	69.3	74.3	76.4	74.7	71.1	65.0	58.9	55.2	(22)
(23)		5%	DB		55.3	57.0	66.9	75.2	82.2	87.3	90.0	89.1	84.5	75.1	65.0	55.8	(23)
(24)			MCWB		50.3	49.6	56.2	61.5	68.9	73.1	75.1	74.0	69.6	63.4	57.6	51.6	(24)
(25)		10%	DB		49.5	51.9	62.0	71.3	78.5	84.6	87.3	86.4	80.8	70.7	60.9	51.4	(25)
(26)			MCWB		44.9	45.9	54.1	59.7	67.0	71.8	74.1	72.7	68.3	61.3	54.3	47.1	(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		60.4	59.4	65.2	68.8	75.1	78.6	80.4	79.4	76.2	71.2	64.7	60.6	(27)
(28)			MCDB		64.0	65.0	76.6	79.4	84.1	88.5	91.1	88.7	85.6	79.8	70.4	65.0	(28)
(29)		2%	WB		57.0	55.2	61.6	65.9	72.6	76.6	78.4	77.6	74.4	68.1	61.4	56.6	(29)
(30)			MCDB		60.6	60.0	69.8	75.3	81.5	86.0	88.7	87.3	83.0	75.5	66.2	59.5	(30)
(31)		5%	WB		51.2	50.7	58.5	63.7	70.6	75.1	77.1	76.2	72.6	65.4	58.5	52.2	(31)
(32)			MCDB		54.4	55.9	65.2	71.8	78.9	84.4	86.8	85.0	80.0	72.0	63.8	55.2	(32)
(33)		10%	WB		45.5	45.7	54.6	61.6	68.5	73.5	75.8	74.9	70.9	62.7	55.1	47.4	(33)
(34)			MCDB		49.0	51.5	61.2	69.4	75.9	82.2	84.3	82.7	77.4	68.8	60.3	50.8	(34)
(35)	Mean Daily Temperature Range		MDBR		15.7	17.8	19.8	22.4	21.0	20.4	19.9	20.9	22.1	22.3	19.5	15.2	(35)
(36)		5% DB	MCDBR		20.8	27.3	27.4	28.1	25.5	23.4	22.5	23.9	26.8	28.2	25.6	21.8	(36)
(37)			MCWBR		16.7	19.4	16.3	15.3	12.5	10.4	8.7	9.2	11.0	15.4	17.5	17.6	(37)
(38)		5% WB	MCDBR		18.0	23.5	23.6	23.4	21.6	20.6	20.0	20.3	20.2	23.5	21.3	19.3	(38)
(39)			MCWBR		16.0	19.1	15.5	14.0	11.4	10.2	8.7	8.9	9.9	14.4	16.6	17.2	(39)
(40)	Clear-Sky Solar Irradiance		taub		0.311	0.329	0.359	0.401	0.445	0.460	0.469	0.464	0.410	0.361	0.332	0.308	(40)
(41)			taud		2.476	2.388	2.372	2.261	2.176	2.180	2.164	2.162	2.290	2.431	2.472	2.493	(41)
(42)			Ebn,noon		274	283	285	278	266	262	258	256	264	268	263	266	(42)
(43)			Edh,noon		25	31	35	42	47	47	47	46	38	30	25	23	(43)
(44)	All-Sky Solar Radiation		RadAvg		549	816	1142	1543	1737	1940	1877	1747	1432	1027	700	477	(44)
(45)			RadStd		42	103	116	130	161	107	125	102	136	99	77	46	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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CLEVELAND HOPKINS INTL, OH, USA WMO#: **725240**

Lat: **41.405N** Long: **81.853W** Elev: **770** StdP: **14.29** Time Zone: **-5.00 (NAE)** Period: **90-14** WBAN: **14820**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	**1**	**4.6**	**10.0**	**-3.4**	**4.7**	**6.6**	**1.4**	**6.1**	**11.8**	**28.5**	**31.2**	**26.1**	**29.0**	**10.6**	**220**	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	**7**	**16.9**	**89.4**	**73.6**	**86.8**	**72.2**	**84.2**	**70.9**	**76.1**	**85.3**	**74.6**	**83.1**	**73.1**	**81.1**	**11.1**	**230**	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	**73.1**	**126.7**	**81.2**	**71.8**	**120.7**	**79.5**	**70.3**	**114.9**	**78.1**	**39.9**	**85.5**	**38.4**	**83.1**	**37.1**	**81.3**	**84.0**	(3)

Extreme Annual Design Conditions

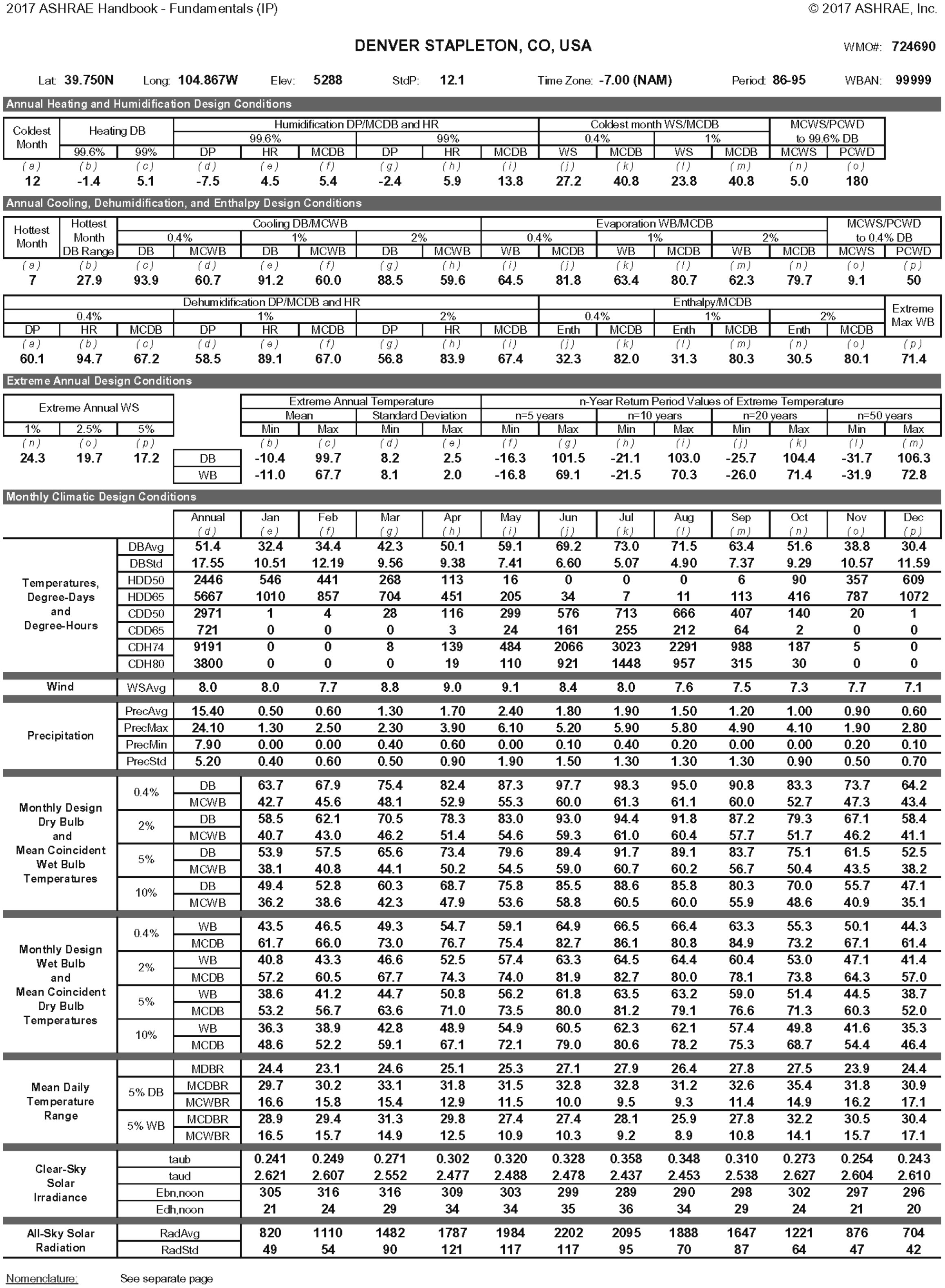
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	**24.6**	**21.0**	**18.9**	DB	**-1.1**	**93.4**	**6.9**	**2.9**	**-6.0**	**95.5**	**-10.1**	**97.2**	**-14.0**	**98.8**	**-19.0**	**100.9**	(4)
(5)				WB	**-1.8**	**79.0**	**6.7**	**2.0**	**-6.6**	**80.4**	**-10.6**	**81.6**	**-14.4**	**82.7**	**-19.3**	**84.1**	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**		DBAvg	**51.3**	**28.2**	**30.1**	**38.0**	**49.6**	**59.8**	**69.2**	**73.0**	**71.6**	**64.5**	**53.7**	**43.0**	**33.0**	(6)
(7)			DBStd	**18.17**	**11.41**	**10.29**	**11.21**	**9.85**	**8.63**	**7.08**	**5.50**	**5.24**	**7.09**	**8.18**	**9.26**	**9.69**	(7)
(8)			HDD50	**2626**	**681**	**561**	**406**	**127**	**12**	**0**	**0**	**0**	**1**	**53**	**251**	**534**	(8)
(9)			HDD65	**5801**	**1141**	**977**	**840**	**473**	**211**	**40**	**4**	**7**	**92**	**362**	**661**	**993**	(9)
(10)			CDD50	**3085**	**4**	**4**	**34**	**115**	**317**	**577**	**714**	**670**	**437**	**167**	**40**	**6**	(10)
(11)			CDD65	**786**	**0**	**0**	**3**	**10**	**51**	**167**	**253**	**212**	**78**	**12**	**0**	**0**	(11)
(12)			CDH74	**6183**	**0**	**0**	**13**	**103**	**462**	**1354**	**2093**	**1544**	**545**	**68**	**1**	**0**	(12)
(13)			CDH80	**1779**	**0**	**0**	**1**	**16**	**114**	**414**	**668**	**429**	**132**	**5**	**0**	**0**	(13)
(14)	**Wind**		WSAvg	**9.5**	**11.3**	**10.7**	**10.6**	**10.3**	**9.1**	**8.3**	**8.0**	**7.4**	**8.2**	**9.3**	**10.5**	**10.8**	(14)
(15)	**Precipitation**		PrecAvg	**36.90**	**2.20**	**2.20**	**2.90**	**3.20**	**3.60**	**3.50**	**3.50**	**3.40**	**3.50**	**2.60**	**3.20**	**3.00**	(15)
(16)			PrecMax	**53.80**	**4.40**	**4.70**	**5.20**	**6.60**	**9.10**	**9.10**	**9.10**	**9.00**	**7.30**	**5.60**	**8.80**	**8.60**	(16)
(17)			PrecMin	**18.80**	**0.40**	**0.50**	**0.90**	**1.20**	**1.00**	**0.60**	**0.70**	**0.50**	**0.70**	**0.70**	**0.80**	**1.10**	(17)
(18)			PrecStd	**6.40**	**1.00**	**1.00**	**1.00**	**1.20**	**1.70**	**1.80**	**1.50**	**1.80**	**1.50**	**1.10**	**1.70**	**1.20**	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB		**61.7**	**62.9**	**75.2**	**82.5**	**86.9**	**90.9**	**93.3**	**91.3**	**89.0**	**80.2**	**70.0**	**62.0**	(19)
(20)			MCWB		**56.1**	**54.1**	**61.4**	**64.4**	**70.4**	**73.1**	**76.8**	**75.4**	**71.9**	**66.6**	**60.1**	**56.9**	(20)
(21)		2%	DB		**55.8**	**55.2**	**68.0**	**77.0**	**83.4**	**87.9**	**89.7**	**87.8**	**83.7**	**75.3**	**65.1**	**56.5**	(21)
(22)			MCWB		**52.2**	**49.3**	**57.0**	**61.9**	**68.4**	**72.6**	**74.4**	**73.3**	**70.0**	**63.3**	**57.1**	**52.2**	(22)
(23)		5%	DB		**50.1**	**49.4**	**62.5**	**72.0**	**79.6**	**85.0**	**86.6**	**84.7**	**80.0**	**71.2**	**61.4**	**51.3**	(23)
(24)			MCWB		**45.8**	**43.6**	**54.1**	**59.0**	**66.6**	**71.0**	**72.6**	**71.9**	**68.4**	**61.7**	**54.9**	**47.6**	(24)
(25)		10%	DB		**43.6**	**44.4**	**56.3**	**66.9**	**75.2**	**81.7**	**83.7**	**81.9**	**76.5**	**67.2**	**57.6**	**46.5**	(25)
(26)			MCWB		**39.8**	**39.9**	**49.7**	**56.5**	**64.2**	**69.6**	**71.3**	**70.7**	**66.7**	**59.1**	**51.5**	**42.7**	(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB		**56.6**	**55.8**	**62.6**	**67.9**	**73.8**	**76.4**	**78.8**	**78.0**	**75.0**	**69.4**	**62.6**	**57.5**	(27)
(28)			MCDB		**60.5**	**61.2**	**73.0**	**77.2**	**82.9**	**86.4**	**89.5**	**87.6**	**83.5**	**77.5**	**67.4**	**60.8**	(28)
(29)		2%	WB		**52.7**	**50.3**	**58.7**	**64.0**	**70.9**	**74.5**	**76.7**	**75.4**	**72.4**	**65.8**	**58.7**	**53.3**	(29)
(30)			MCDB		**55.9**	**54.7**	**66.2**	**74.1**	**80.0**	**84.3**	**86.8**	**83.4**	**79.6**	**72.6**	**63.7**	**56.1**	(30)
(31)		5%	WB		**46.3**	**44.7**	**54.9**	**60.6**	**68.5**	**72.8**	**74.7**	**74.0**	**70.5**	**63.1**	**55.5**	**47.9**	(31)
(32)			MCDB		**49.4**	**48.6**	**61.5**	**69.9**	**77.1**	**82.2**	**83.3**	**81.5**	**77.0**	**69.4**	**60.6**	**51.1**	(32)
(33)		10%	WB		**40.0**	**39.7**	**49.9**	**57.3**	**65.6**	**71.1**	**73.1**	**72.4**	**68.5**	**60.4**	**52.1**	**43.0**	(33)
(34)			MCDB		**43.6**	**44.2**	**56.6**	**65.9**	**73.7**	**79.6**	**80.9**	**79.5**	**75.0**	**66.1**	**56.6**	**46.3**	(34)
(35)	**Mean Daily Temperature Range**		MDBR		**12.6**	**13.4**	**15.3**	**18.0**	**18.3**	**17.5**	**16.9**	**16.6**	**17.0**	**16.2**	**13.6**	**11.6**	(35)
(36)		5% DB	MCDBR		**17.5**	**20.4**	**23.9**	**25.2**	**23.0**	**20.8**	**20.0**	**19.9**	**21.3**	**21.3**	**18.6**	**17.5**	(36)
(37)			MCWBR		**15.5**	**16.4**	**15.7**	**15.4**	**12.3**	**9.7**	**8.9**	**9.4**	**10.9**	**12.4**	**13.9**	**15.1**	(37)
(38)		5% WB	MCDBR		**17.2**	**19.1**	**21.6**	**23.0**	**20.3**	**18.5**	**17.4**	**16.8**	**16.7**	**18.3**	**17.3**	**17.3**	(38)
(39)			MCWBR		**16.0**	**16.6**	**15.3**	**15.2**	**12.1**	**9.8**	**8.8**	**8.9**	**9.8**	**12.0**	**14.3**	**16.3**	(39)
(40)	**Clear-Sky Solar Irradiance**		taub		**0.305**	**0.335**	**0.358**	**0.395**	**0.431**	**0.449**	**0.448**	**0.434**	**0.385**	**0.367**	**0.327**	**0.309**	(40)
(41)			taud		**2.403**	**2.279**	**2.296**	**2.278**	**2.213**	**2.208**	**2.234**	**2.260**	**2.370**	**2.399**	**2.483**	**2.468**	(41)
(42)			Ebn,noon		**268**	**274**	**281**	**278**	**269**	**264**	**263**	**263**	**269**	**260**	**259**	**257**	(42)
(43)			Edh,noon		**25**	**34**	**37**	**41**	**45**	**45**	**43**	**41**	**34**	**30**	**24**	**22**	(43)
(44)	**All-Sky Solar Radiation**		RadAvg		**467**	**730**	**1068**	**1438**	**1728**	**1923**	**1893**	**1689**	**1334**	**854**	**534**	**376**	(44)
(45)			RadStd		**41**	**82**	**104**	**134**	**160**	**115**	**133**	**79**	**117**	**67**	**68**	**31**	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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DENVER STAPLETON, CO, USA

WMO#: **724690**

Lat: **39.750N** Long: **104.867W** Elev: **5288** StdP: **12.1** Time Zone: **-7.00 (NAM)** Period: **86-95** WBAN: **99999**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	12	-1.4	5.1	-7.5	4.5	5.4	-2.4	5.9	13.8	27.2	40.8	23.8	40.8	5.0	180	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	27.9	93.9	60.7	91.2	60.0	88.5	59.6	64.5	81.8	63.4	80.7	62.3	79.7	9.1	50	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	60.1	94.7	67.2	58.5	89.1	67.0	56.8	83.9	67.4	32.3	82.0	31.3	80.3	30.5	80.1	71.4	(3)

Extreme Annual Design Conditions

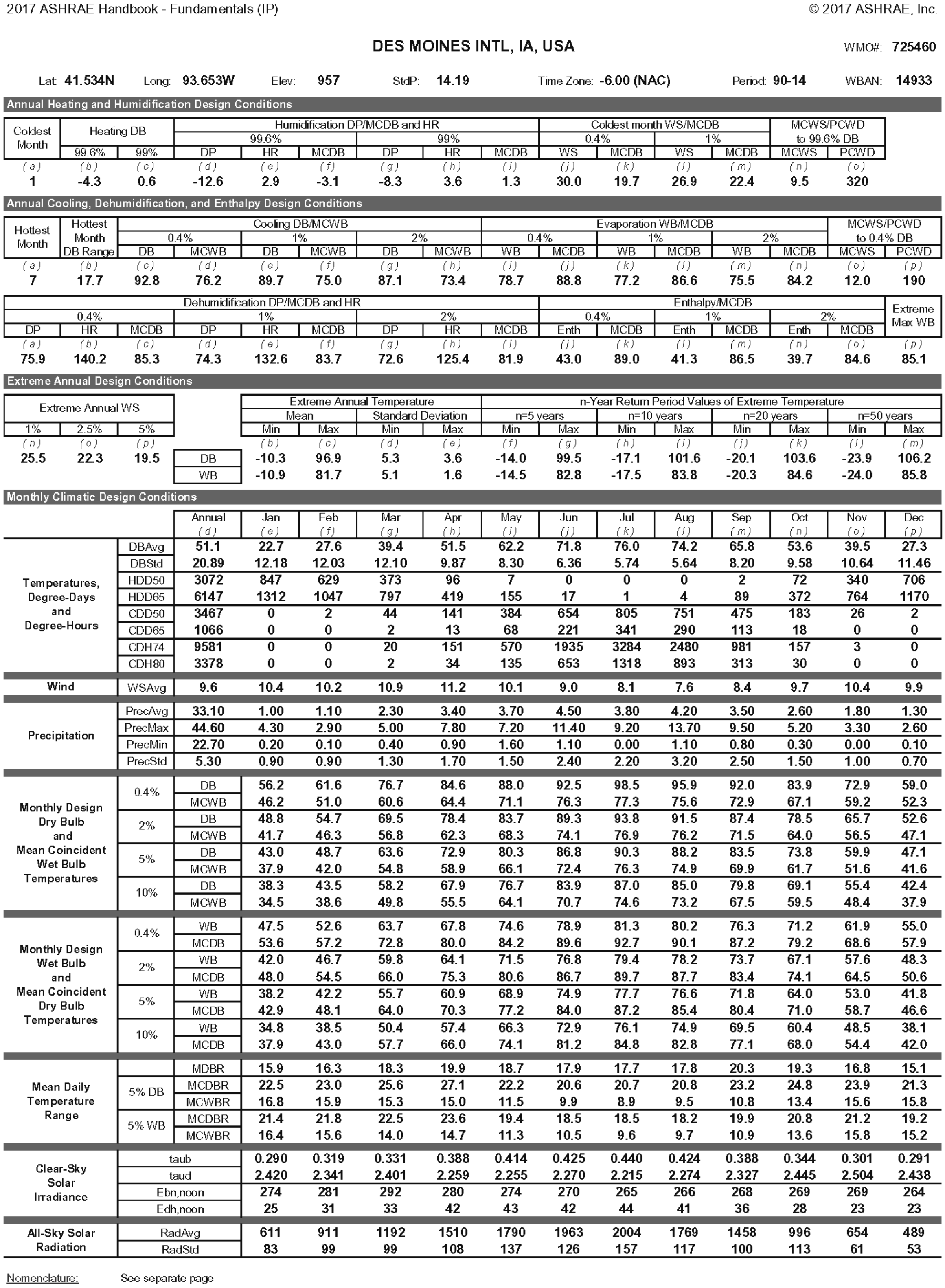
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	24.3	19.7	17.2	DB	-10.4	99.7	8.2	2.5	-16.3	101.5	-21.1	103.0	-25.7	104.4	-31.7	106.3	(4)
(5)				WB	-11.0	67.7	8.1	2.0	-16.8	69.1	-21.5	70.3	-26.0	71.4	-31.9	72.8	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	51.4	32.4	34.4	42.3	50.1	59.1	69.2	73.0	71.5	63.4	51.6	38.8	30.4	(6)
(7)			DBStd	17.55	10.51	12.19	9.56	9.38	7.41	6.60	5.07	4.90	7.37	9.29	10.57	11.59	(7)
(8)			HDD50	2446	546	441	268	113	16	0	0	0	6	90	357	609	(8)
(9)			HDD65	5667	1010	857	704	451	205	34	7	11	113	416	787	1072	(9)
(10)			CDD50	2971	1	4	28	116	299	576	713	666	407	140	20	1	(10)
(11)			CDD65	721	0	0	0	3	24	161	255	212	64	2	0	0	(11)
(12)			CDH74	9191	0	0	8	139	484	2066	3023	2291	988	187	5	0	(12)
(13)			CDH80	3800	0	0	0	19	110	921	1448	957	315	30	0	0	(13)
(14)	Wind		WSAvg	8.0	8.0	7.7	8.8	9.0	9.1	8.4	8.0	7.6	7.5	7.3	7.7	7.1	(14)
(15)	Precipitation		PrecAvg	15.40	0.50	0.60	1.30	1.70	2.40	1.80	1.90	1.50	1.20	1.00	0.90	0.60	(15)
(16)			PrecMax	24.10	1.30	2.50	2.30	3.90	6.10	5.20	5.90	5.80	4.90	4.10	1.90	2.80	(16)
(17)			PrecMin	7.90	0.00	0.00	0.40	0.60	0.00	0.10	0.40	0.20	0.00	0.00	0.20	0.10	(17)
(18)			PrecStd	5.20	0.40	0.60	0.50	0.90	1.90	1.50	1.30	1.30	1.30	0.90	0.50	0.70	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB	63.7	67.9	75.4	82.4	87.3	97.7	98.3	95.0	90.8	83.3	73.7	64.2		(19)
(20)			MCWB	42.7	45.6	48.1	52.9	55.3	60.0	61.3	61.1	60.0	52.7	47.3	43.4		(20)
(21)		2%	DB	58.5	62.1	70.5	78.3	83.0	93.0	94.4	91.8	87.2	79.3	67.1	58.4		(21)
(22)			MCWB	40.7	43.0	46.2	51.4	54.6	59.3	61.0	60.4	57.7	51.7	46.2	41.1		(22)
(23)		5%	DB	53.9	57.5	65.6	73.4	79.6	89.4	91.7	89.1	83.7	75.1	61.5	52.5		(23)
(24)			MCWB	38.1	40.8	44.1	50.2	54.5	59.0	60.7	60.2	56.7	50.4	43.5	38.2		(24)
(25)		10%	DB	49.4	52.8	60.3	68.7	75.8	85.5	88.6	85.8	80.3	70.0	55.7	47.1		(25)
(26)			MCWB	36.2	38.6	42.3	47.9	53.6	58.8	60.5	60.0	55.9	48.6	40.9	35.1		(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB	43.5	46.5	49.3	54.7	59.1	64.9	66.5	66.4	63.3	55.3	50.1	44.3		(27)
(28)			MCDB	61.7	66.0	73.0	76.7	75.4	82.7	86.1	80.8	84.9	73.2	67.1	61.4		(28)
(29)		2%	WB	40.8	43.3	46.6	52.5	57.4	63.3	64.5	64.4	60.4	53.0	47.1	41.4		(29)
(30)			MCDB	57.2	60.5	67.7	74.3	74.0	81.9	82.7	80.0	78.1	73.8	64.3	57.0		(30)
(31)		5%	WB	38.6	41.2	44.7	50.8	56.2	61.8	63.5	63.2	59.0	51.4	44.5	38.7		(31)
(32)			MCDB	53.2	56.7	63.6	71.0	73.5	80.0	81.2	79.1	76.6	71.3	60.3	52.0		(32)
(33)		10%	WB	36.3	38.9	42.8	48.9	54.9	60.5	62.3	62.1	57.4	49.8	41.6	35.3		(33)
(34)			MCDB	48.6	52.2	59.1	67.1	72.1	79.0	80.6	78.2	75.3	68.7	54.4	46.4		(34)
(35)	Mean Daily Temperature Range		MDBR	24.4	23.1	24.6	25.1	25.3	27.1	27.9	26.4	27.8	27.5	23.9	24.4		(35)
(36)		5% DB	MCDBR	29.7	30.2	33.1	31.8	31.5	32.8	32.8	31.2	32.6	35.4	31.8	30.9		(36)
(37)			MCWBR	16.6	15.8	15.4	12.9	11.5	10.0	9.5	9.3	11.4	14.9	16.2	17.1		(37)
(38)		5% WB	MCDBR	28.9	29.4	31.3	29.8	27.4	27.4	28.1	25.9	27.8	32.2	30.5	30.4		(38)
(39)			MCWBR	16.5	15.7	14.9	12.5	10.9	10.3	9.2	8.9	10.8	14.1	15.7	17.1		(39)
(40)	Clear-Sky Solar Irradiance		taub		0.241	0.249	0.271	0.302	0.320	0.328	0.358	0.348	0.310	0.273	0.254	0.243	(40)
(41)			taud		2.621	2.607	2.552	2.477	2.488	2.478	2.437	2.453	2.538	2.627	2.604	2.610	(41)
(42)			Ebn,noon		305	316	316	309	303	299	289	290	298	302	297	296	(42)
(43)			Edh,noon		21	24	29	34	34	35	36	34	29	24	21	20	(43)
(44)	All-Sky Solar Radiation		RadAvg		820	1110	1482	1787	1984	2202	2095	1888	1647	1221	876	704	(44)
(45)			RadStd		49	54	90	121	117	117	95	70	87	64	47	42	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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DES MOINES INTL, IA, USA WMO#: **725460**

Lat: **41.534N** Long: **93.653W** Elev: **957** StdP: **14.19** Time Zone: **-6.00 (NAC)** Period: **90-14** WBAN: **14933**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	-4.3	0.6	-12.6	2.9	-3.1	-8.3	3.6	1.3	30.0	19.7	26.9	22.4	9.5	320	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	17.7	92.8	76.2	89.7	75.0	87.1	73.4	78.7	88.8	77.2	86.6	75.5	84.2	12.0	190	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	75.9	140.2	85.3	74.3	132.6	83.7	72.6	125.4	81.9	43.0	89.0	41.3	86.5	39.7	84.6	85.1	(3)

Extreme Annual Design Conditions

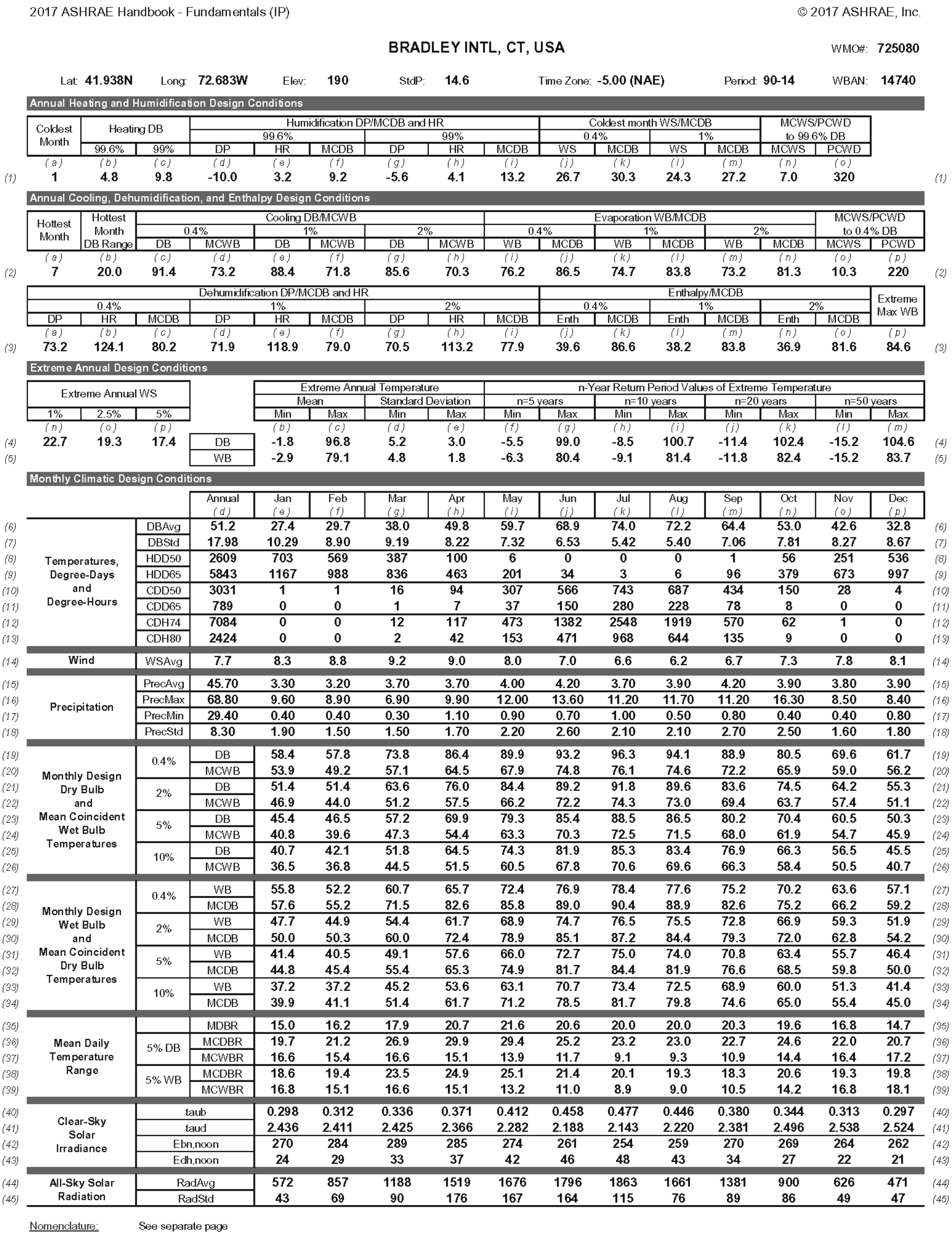
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	25.5	22.3	19.5	DB	-10.3	96.9	5.3	3.6	-14.0	99.5	-17.1	101.6	-20.1	103.6	-23.9	106.2	(4)
(5)				WB	-10.9	81.7	5.1	1.6	-14.5	82.8	-17.5	83.8	-20.3	84.6	-24.0	85.8	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	51.1	22.7	27.6	39.4	51.5	62.2	71.8	76.0	74.2	65.8	53.6	39.5	27.3	(6)
(7)			DBStd	20.89	12.18	12.03	12.10	9.87	8.30	6.36	5.74	5.64	8.20	9.58	10.64	11.46	(7)
(8)			HDD50	3072	847	629	373	96	7	0	0	0	2	72	340	706	(8)
(9)			HDD65	6147	1312	1047	797	419	155	17	1	4	89	372	764	1170	(9)
(10)			CDD50	3467	0	2	44	141	384	654	805	751	475	183	26	2	(10)
(11)			CDD65	1066	0	0	2	13	68	221	341	290	113	18	0	0	(11)
(12)			CDH74	9581	0	0	20	151	570	1935	3284	2480	981	157	3	0	(12)
(13)			CDH80	3378	0	0	2	34	135	653	1318	893	313	30	0	0	(13)
(14)	Wind		WSAvg	9.6	10.4	10.2	10.9	11.2	10.1	9.0	8.1	7.6	8.4	9.7	10.4	9.9	(14)
(15)	Precipitation		PrecAvg	33.10	1.00	1.10	2.30	3.40	3.70	4.50	3.80	4.20	3.50	2.60	1.80	1.30	(15)
(16)			PrecMax	44.60	4.30	2.90	5.00	7.80	7.20	11.40	9.20	13.70	9.50	5.20	3.30	2.60	(16)
(17)			PrecMin	22.70	0.20	0.10	0.40	0.90	1.60	1.10	0.00	1.10	0.80	0.30	0.00	0.10	(17)
(18)			PrecStd	5.30	0.90	0.90	1.30	1.70	1.50	2.40	2.20	3.20	2.50	1.50	1.00	0.70	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		56.2	61.6	76.7	84.6	88.0	92.5	98.5	95.9	92.0	83.9	72.9	59.0	(19)
(20)			MCWB		46.2	51.0	60.6	64.4	71.1	76.3	77.3	75.6	72.9	67.1	59.2	52.3	(20)
(21)		2%	DB		48.8	54.7	69.5	78.4	83.7	89.3	93.8	91.5	87.4	78.5	65.7	52.6	(21)
(22)			MCWB		41.7	46.3	56.8	62.3	68.3	74.1	76.9	76.2	71.5	64.0	56.5	47.1	(22)
(23)		5%	DB		43.0	48.7	63.6	72.9	80.3	86.8	90.3	88.2	83.5	73.8	59.9	47.1	(23)
(24)			MCWB		37.9	42.0	54.8	58.9	66.1	72.4	76.3	74.9	69.9	61.7	51.6	41.6	(24)
(25)		10%	DB		38.3	43.5	58.2	67.9	76.7	83.9	87.0	85.0	79.8	69.1	55.4	42.4	(25)
(26)			MCWB		34.5	38.6	49.8	55.5	64.1	70.7	74.6	73.2	67.5	59.5	48.4	37.9	(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		47.5	52.6	63.7	67.8	74.6	78.9	81.3	80.2	76.3	71.2	61.9	55.0	(27)
(28)			MCDB		53.6	57.2	72.8	80.0	84.2	89.6	92.7	90.1	87.2	79.2	68.6	57.9	(28)
(29)		2%	WB		42.0	46.7	59.8	64.1	71.5	76.8	79.4	78.2	73.7	67.1	57.6	48.3	(29)
(30)			MCDB		48.0	54.5	66.0	75.3	80.6	86.7	89.7	87.7	83.4	74.1	64.5	50.6	(30)
(31)		5%	WB		38.2	42.2	55.7	60.9	68.9	74.9	77.7	76.6	71.8	64.0	53.0	41.8	(31)
(32)			MCDB		42.9	48.1	64.0	70.3	77.2	84.0	87.2	85.4	80.4	71.0	58.7	46.6	(32)
(33)		10%	WB		34.8	38.5	50.4	57.4	66.3	72.9	76.1	74.9	69.5	60.4	48.5	38.1	(33)
(34)			MCDB		37.9	43.0	57.7	66.0	74.1	81.2	84.8	82.8	77.1	68.0	54.4	42.0	(34)
(35)	Mean Daily Temperature Range		MDBR		15.9	16.3	18.3	19.9	18.7	17.9	17.7	17.8	20.3	19.3	16.8	15.1	(35)
(36)		5% DB	MCDBR		22.5	23.0	25.6	27.1	22.2	20.6	20.7	20.8	23.2	24.8	23.9	21.3	(36)
(37)			MCWBR		16.8	15.9	15.3	15.0	11.5	9.9	8.9	9.5	10.8	13.4	15.6	15.8	(37)
(38)		5% WB	MCDBR		21.4	21.8	22.5	23.6	19.4	18.5	18.5	18.2	19.9	20.8	21.2	19.2	(38)
(39)			MCWBR		16.4	15.6	14.0	14.7	11.3	10.5	9.6	9.7	10.9	13.6	15.8	15.2	(39)
(40)	Clear-Sky Solar Irradiance		taub		0.290	0.319	0.331	0.388	0.414	0.425	0.440	0.424	0.388	0.344	0.301	0.291	(40)
(41)			taud		2.420	2.341	2.401	2.259	2.255	2.270	2.215	2.274	2.327	2.445	2.504	2.438	(41)
(42)			Ebn,noon		274	281	292	280	274	270	265	266	268	269	269	264	(42)
(43)			Edh,noon		25	31	33	42	43	42	44	41	36	28	23	23	(43)
(44)	All-Sky Solar Radiation		RadAvg		611	911	1192	1510	1790	1963	2004	1769	1458	996	654	489	(44)
(45)			RadStd		83	99	99	108	137	126	157	117	100	113	61	53	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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BRADLEY INTL, CT, USA

WMO#: **725080**

Lat: **41.938N** Long: **72.683W** Elev: **190** StdP: **14.6** Time Zone: **-5.00 (NAE)** Period: **90-14** WBAN: **14740**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	4.8	9.8	-10.0	3.2	9.2	-5.6	4.1	13.2	26.7	30.3	24.3	27.2	7.0	320	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	20.0	91.4	73.2	88.4	71.8	85.6	70.3	76.2	86.5	74.7	83.8	73.2	81.3	10.3	220	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	73.2	124.1	80.2	71.9	118.9	79.0	70.5	113.2	77.9	39.6	86.6	38.2	83.8	36.9	81.6	84.6	(3)

Extreme Annual Design Conditions

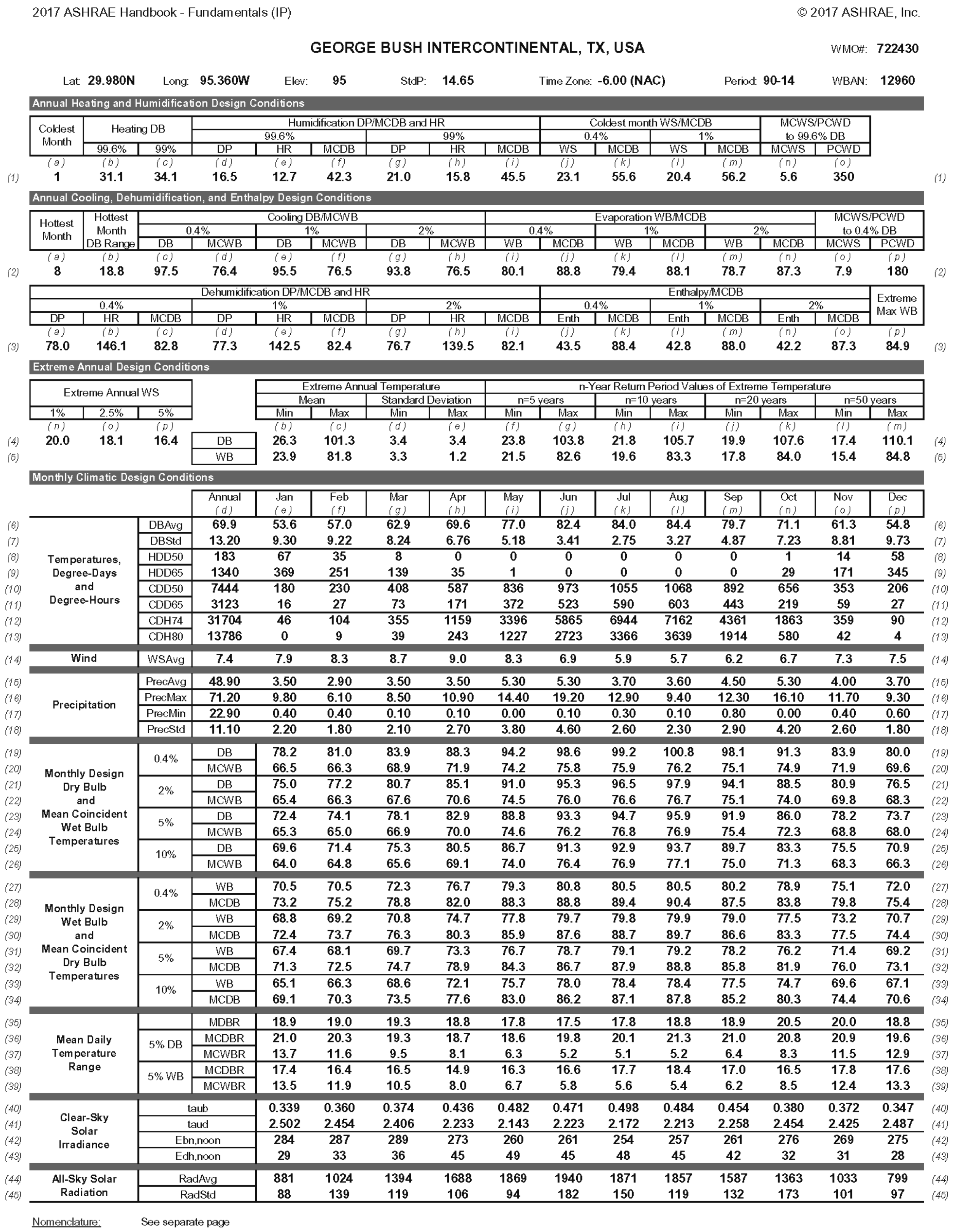
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	22.7	19.3	17.4	DB	-1.8	96.8	5.2	3.0	-5.5	99.0	-8.5	100.7	-11.4	102.4	-15.2	104.6	(4)
(5)				WB	-2.9	79.1	4.8	1.8	-6.3	80.4	-9.1	81.4	-11.8	82.4	-15.2	83.7	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**	DBAvg		51.2	27.4	29.7	38.0	49.8	59.7	68.9	74.0	72.2	64.4	53.0	42.6	32.8	(6)
(7)		DBStd		17.98	10.29	8.90	9.19	8.22	7.32	6.53	5.42	5.40	7.06	7.81	8.27	8.67	(7)
(8)		HDD50		2609	703	569	387	100	6	0	0	0	1	56	251	536	(8)
(9)		HDD65		5843	1167	988	836	463	201	34	3	6	96	379	673	997	(9)
(10)		CDD50		3031	1	1	16	94	307	566	743	687	434	150	28	4	(10)
(11)		CDD65		789	0	0	1	7	37	150	280	228	78	8	0	0	(11)
(12)		CDH74		7084	0	0	12	117	473	1382	2548	1919	570	62	1	0	(12)
(13)		CDH80		2424	0	0	2	42	153	471	968	644	135	9	0	0	(13)
(14)	**Wind**	WSAvg		7.7	8.3	8.8	9.2	9.0	8.0	7.0	6.6	6.2	6.7	7.3	7.8	8.1	(14)
(15)	**Precipitation**	PrecAvg		45.70	3.30	3.20	3.70	3.70	4.00	4.20	3.70	3.90	4.20	3.90	3.80	3.90	(15)
(16)		PrecMax		68.80	9.60	8.90	6.90	9.90	12.00	13.60	11.20	11.70	11.20	16.30	8.50	8.40	(16)
(17)		PrecMin		29.40	0.40	0.40	0.30	1.10	0.90	0.70	1.00	0.50	0.80	0.40	0.40	0.80	(17)
(18)		PrecStd		8.30	1.90	1.50	1.50	1.70	2.20	2.60	2.10	2.10	2.70	2.50	1.60	1.80	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB		58.4	57.8	73.8	86.4	89.9	93.2	96.3	94.1	88.9	80.5	69.6	61.7	(19)
(20)			MCWB		53.9	49.2	57.1	64.5	67.9	74.8	76.1	74.6	72.2	65.9	59.0	56.2	(20)
(21)		2%	DB		51.4	51.4	63.6	76.0	84.4	89.2	91.8	89.6	83.6	74.5	64.2	55.3	(21)
(22)			MCWB		46.9	44.0	51.2	57.5	66.2	72.2	74.3	73.0	69.4	63.7	57.4	51.1	(22)
(23)		5%	DB		45.4	46.5	57.2	69.9	79.3	85.4	88.5	86.5	80.2	70.4	60.5	50.3	(23)
(24)			MCWB		40.8	39.6	47.3	54.4	63.3	70.3	72.5	71.5	68.0	61.9	54.7	45.9	(24)
(25)		10%	DB		40.7	42.1	51.8	64.5	74.3	81.9	85.3	83.4	76.9	66.3	56.5	45.5	(25)
(26)			MCWB		36.5	36.8	44.5	51.5	60.5	67.8	70.6	69.6	66.3	58.4	50.5	40.7	(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB		55.8	52.2	60.7	65.7	72.4	76.9	78.4	77.6	75.2	70.2	63.6	57.1	(27)
(28)			MCDB		57.6	55.2	71.5	82.6	85.8	89.0	90.4	88.9	82.6	75.2	66.2	59.2	(28)
(29)		2%	WB		47.7	44.9	54.4	61.7	68.9	74.7	76.5	75.5	72.8	66.9	59.3	51.9	(29)
(30)			MCDB		50.0	50.3	60.0	72.4	78.9	85.1	87.2	84.4	79.3	72.0	62.8	54.2	(30)
(31)		5%	WB		41.4	40.5	49.1	57.6	66.0	72.7	75.0	74.0	70.8	63.4	55.7	46.4	(31)
(32)			MCDB		44.8	45.4	55.4	65.3	74.9	81.7	84.4	81.9	76.6	68.5	59.8	50.0	(32)
(33)		10%	WB		37.2	37.2	45.2	53.6	63.1	70.7	73.4	72.5	68.9	60.0	51.3	41.4	(33)
(34)			MCDB		39.9	41.1	51.4	61.7	71.2	78.5	81.7	79.8	74.6	65.0	55.4	45.0	(34)
(35)	**Mean Daily Temperature Range**		MDBR		15.0	16.2	17.9	20.7	21.6	20.6	20.0	20.0	20.3	19.6	16.8	14.7	(35)
(36)		5% DB	MCDBR		19.7	21.2	26.9	29.9	29.4	25.2	23.2	23.0	22.7	24.6	22.0	20.7	(36)
(37)			MCWBR		16.6	15.4	16.6	15.1	13.9	11.7	9.1	9.3	10.9	14.4	16.4	17.2	(37)
(38)		5% WB	MCDBR		18.6	19.4	23.5	24.9	25.1	21.4	20.1	19.3	18.3	20.6	19.3	19.8	(38)
(39)			MCWBR		16.8	15.1	16.6	15.1	13.2	11.0	8.9	9.0	10.5	14.2	16.8	18.1	(39)
(40)	**Clear-Sky Solar Irradiance**	taub			0.298	0.312	0.336	0.371	0.412	0.458	0.477	0.446	0.380	0.344	0.313	0.297	(40)
(41)		taud			2.436	2.411	2.425	2.366	2.282	2.188	2.143	2.220	2.381	2.496	2.538	2.524	(41)
(42)		Ebn,noon			270	284	289	285	274	261	254	259	270	269	264	262	(42)
(43)		Edh,noon			24	29	33	37	42	46	48	43	34	27	22	21	(43)
(44)	**All-Sky Solar Radiation**	RadAvg			572	857	1188	1519	1676	1796	1863	1661	1381	900	626	471	(44)
(45)		RadStd			43	69	90	176	167	164	115	76	89	86	49	47	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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GEORGE BUSH INTERCONTINENTAL, TX, USA WMO#: **722430**

Lat: **29.980N** Long: **95.360W** Elev: **95** StdP: **14.65** Time Zone: **-6.00 (NAC)** Period: **90-14** WBAN: **12960**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	31.1	34.1	16.5	12.7	42.3	21.0	15.8	45.5	23.1	55.6	20.4	56.2	5.6	350	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	8	18.8	97.5	76.4	95.5	76.5	93.8	76.5	80.1	88.8	79.4	88.1	78.7	87.3	7.9	180	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	78.0	146.1	82.8	77.3	142.5	82.4	76.7	139.5	82.1	43.5	88.4	42.8	88.0	42.2	87.3	84.9	(3)

Extreme Annual Design Conditions

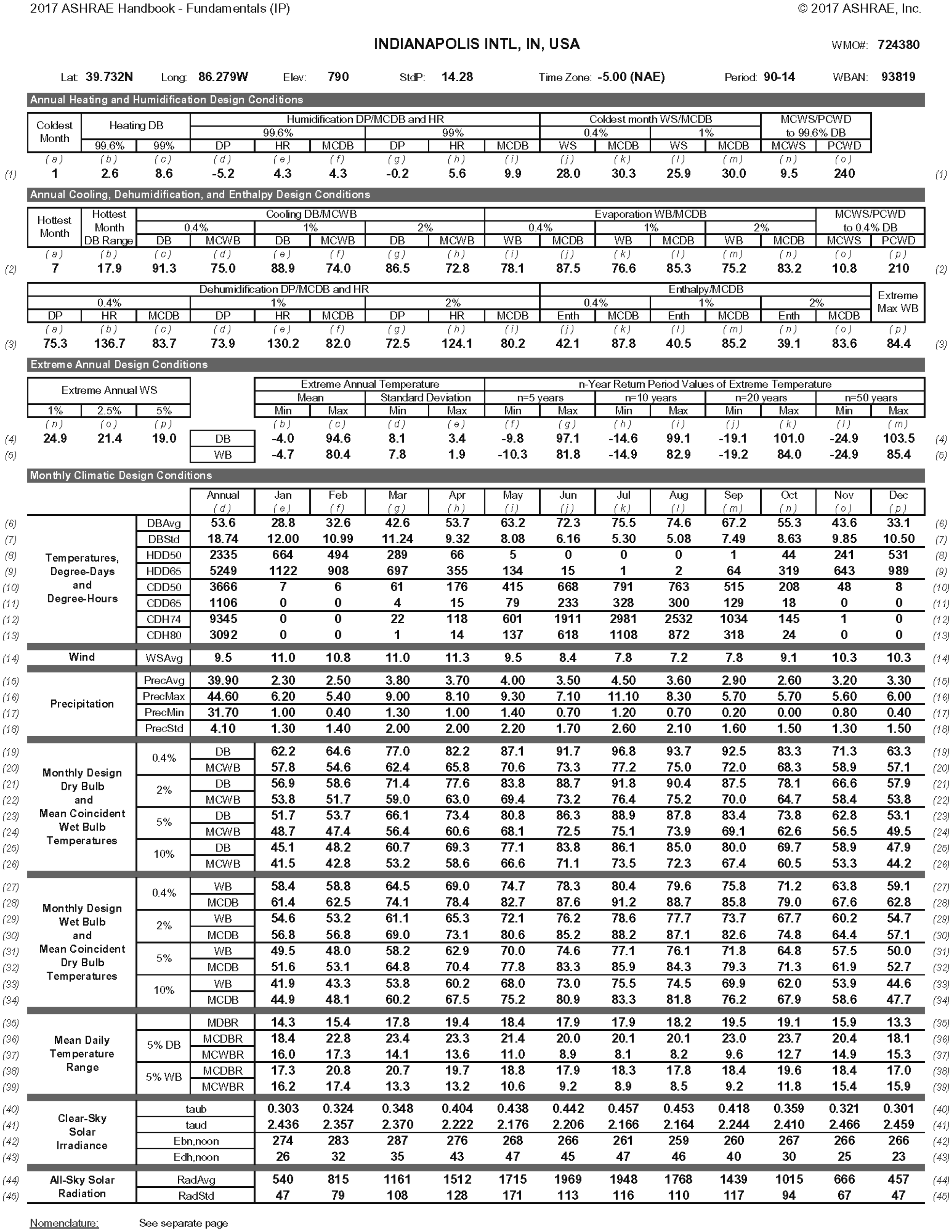
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	20.0	18.1	16.4	DB	26.3	101.3	3.4	3.4	23.8	103.8	21.8	105.7	19.9	107.6	17.4	110.1	(4)
(5)				WB	23.9	81.8	3.3	1.2	21.5	82.6	19.6	83.3	17.8	84.0	15.4	84.8	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**		DBAvg	69.9	53.6	57.0	62.9	69.6	77.0	82.4	84.0	84.4	79.7	71.1	61.3	54.8	(6)
(7)			DBStd	13.20	9.30	9.22	8.24	6.76	5.18	3.41	2.75	3.27	4.87	7.23	8.81	9.73	(7)
(8)			HDD50	183	67	35	8	0	0	0	0	0	0	1	14	58	(8)
(9)			HDD65	1340	369	251	139	35	1	0	0	0	0	29	171	345	(9)
(10)			CDD50	7444	180	230	408	587	836	973	1055	1068	892	656	353	206	(10)
(11)			CDD65	3123	16	27	73	171	372	523	590	603	443	219	59	27	(11)
(12)			CDH74	31704	46	104	355	1159	3396	5865	6944	7162	4361	1863	359	90	(12)
(13)			CDH80	13786	0	9	39	243	1227	2723	3366	3639	1914	580	42	4	(13)
(14)	**Wind**		WSAvg	7.4	7.9	8.3	8.7	9.0	8.3	6.9	5.9	5.7	6.2	6.7	7.3	7.5	(14)
(15)	**Precipitation**		PrecAvg	48.90	3.50	2.90	3.50	3.50	5.30	5.30	3.70	3.60	4.50	5.30	4.00	3.70	(15)
(16)			PrecMax	71.20	9.80	6.10	8.50	10.90	14.40	19.20	12.90	9.40	12.30	16.10	11.70	9.30	(16)
(17)			PrecMin	22.90	0.40	0.40	0.10	0.10	0.00	0.10	0.30	0.10	0.80	0.00	0.40	0.60	(17)
(18)			PrecStd	11.10	2.20	1.80	2.10	2.70	3.80	4.60	2.60	2.30	2.90	4.20	2.60	1.80	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB		78.2	81.0	83.9	88.3	94.2	98.6	99.2	100.8	98.1	91.3	83.9	80.0	(19)
(20)			MCWB		66.5	66.3	68.9	71.9	74.2	75.8	75.9	76.2	75.1	74.9	71.9	69.6	(20)
(21)		2%	DB		75.0	77.2	80.7	85.1	91.0	95.3	96.5	97.9	94.1	88.5	80.9	76.5	(21)
(22)			MCWB		65.4	66.3	67.6	70.6	74.5	76.0	76.6	76.7	75.1	74.0	69.8	68.3	(22)
(23)		5%	DB		72.4	74.1	78.1	82.9	88.8	93.3	94.7	95.9	91.9	86.0	78.2	73.7	(23)
(24)			MCWB		65.3	65.0	66.9	70.0	74.6	76.2	76.8	76.9	75.4	72.3	68.8	68.0	(24)
(25)		10%	DB		69.6	71.4	75.3	80.5	86.7	91.3	92.9	93.7	89.7	83.3	75.5	70.9	(25)
(26)			MCWB		64.0	64.8	65.6	69.1	74.0	76.4	76.9	77.1	75.0	71.3	68.3	66.3	(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB		70.5	70.5	72.3	76.7	79.3	80.8	80.5	80.5	80.2	78.9	75.1	72.0	(27)
(28)			MCDB		73.2	75.2	78.8	82.0	88.3	88.8	89.4	90.4	87.5	83.8	79.8	75.4	(28)
(29)		2%	WB		68.8	69.2	70.8	74.7	77.8	79.7	79.8	79.9	79.0	77.5	73.2	70.7	(29)
(30)			MCDB		72.4	73.7	76.3	80.3	85.9	87.6	88.7	89.7	86.6	83.3	77.5	74.4	(30)
(31)		5%	WB		67.4	68.1	69.7	73.3	76.7	78.7	79.1	79.2	78.2	76.2	71.4	69.2	(31)
(32)			MCDB		71.3	72.5	74.7	78.9	84.3	86.7	87.9	88.8	85.8	81.9	76.0	73.1	(32)
(33)		10%	WB		65.1	66.3	68.6	72.1	75.7	78.0	78.4	78.4	77.5	74.7	69.6	67.1	(33)
(34)			MCDB		69.1	70.3	73.5	77.6	83.0	86.2	87.1	87.8	85.2	80.3	74.4	70.6	(34)
(35)	**Mean Daily Temperature Range**		MDBR		18.9	19.0	19.3	18.8	17.8	17.5	17.8	18.8	18.9	20.5	20.0	18.8	(35)
(36)		5% DB	MCDBR		21.0	20.3	19.3	18.7	18.6	19.8	20.1	21.3	21.0	20.8	20.9	19.6	(36)
(37)			MCWBR		13.7	11.6	9.5	8.1	6.3	5.2	5.1	5.2	6.4	8.3	11.5	12.9	(37)
(38)		5% WB	MCDBR		17.4	16.4	16.5	14.9	16.3	16.6	17.7	18.4	17.0	16.5	17.8	17.6	(38)
(39)			MCWBR		13.5	11.9	10.5	8.0	6.7	5.8	5.6	5.4	6.2	8.5	12.4	13.3	(39)
(40)	**Clear-Sky Solar Irradiance**		taub		0.339	0.360	0.374	0.436	0.482	0.471	0.498	0.484	0.454	0.380	0.372	0.347	(40)
(41)			taud		2.502	2.454	2.406	2.233	2.143	2.223	2.172	2.213	2.258	2.454	2.425	2.487	(41)
(42)			Ebn,noon		284	287	289	273	260	261	254	257	261	276	269	275	(42)
(43)			Edh,noon		29	33	36	45	49	45	48	45	42	32	31	28	(43)
(44)	**All-Sky Solar Radiation**		RadAvg		881	1024	1394	1688	1869	1940	1871	1857	1587	1363	1033	799	(44)
(45)			RadStd		88	139	119	106	94	182	150	119	132	173	101	97	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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INDIANAPOLIS INTL, IN, USA WMO#: **724380**

Lat: **39.732N** Long: **86.279W** Elev: **790** StdP: **14.28** Time Zone: **-5.00 (NAE)** Period: **90-14** WBAN: **93819**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	2.6	8.6	-5.2	4.3	4.3	-0.2	5.6	9.9	28.0	30.3	25.9	30.0	9.5	240	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	17.9	91.3	75.0	88.9	74.0	86.5	72.8	78.1	87.5	76.6	85.3	75.2	83.2	10.8	210	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	75.3	136.7	83.7	73.9	130.2	82.0	72.5	124.1	80.2	42.1	87.8	40.5	85.2	39.1	83.6	84.4	(3)

Extreme Annual Design Conditions

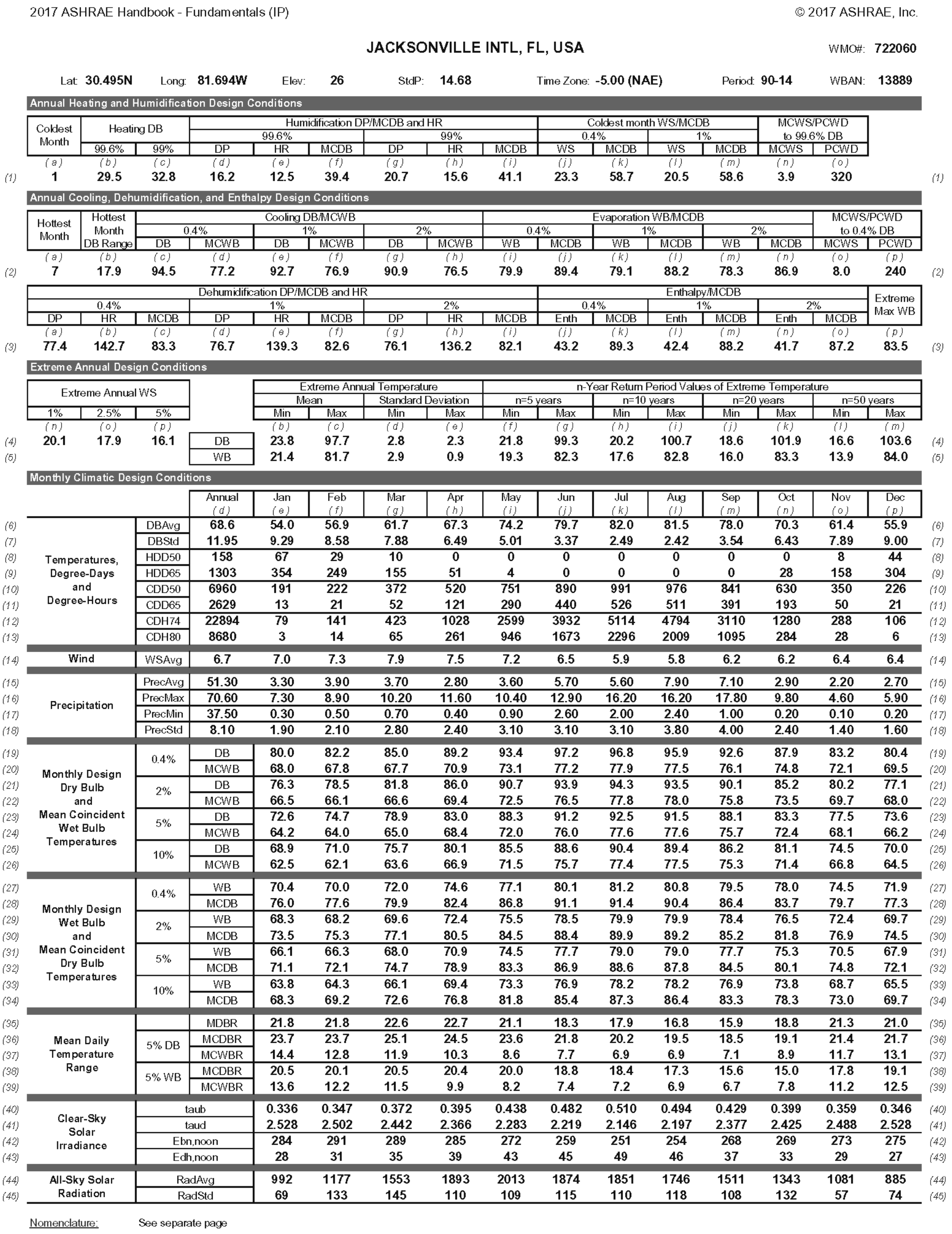
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	24.9	21.4	19.0	DB	-4.0	94.6	8.1	3.4	-9.8	97.1	-14.6	99.1	-19.1	101.0	-24.9	103.5	(4)
(5)				WB	-4.7	80.4	7.8	1.9	-10.3	81.8	-14.9	82.9	-19.2	84.0	-24.9	85.4	(5)

Monthly Climatic Design Conditions

				Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	
				(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	53.6	28.8	32.6	42.6	53.7	63.2	72.3	75.5	74.6	67.2	55.3	43.6	33.1	(6)
(7)			DBStd	18.74	12.00	10.99	11.24	9.32	8.08	6.16	5.30	5.08	7.49	8.63	9.85	10.50	(7)
(8)			HDD50	2335	664	494	289	66	5	0	0	0	1	44	241	531	(8)
(9)			HDD65	5249	1122	908	697	355	134	15	1	2	64	319	643	989	(9)
(10)			CDD50	3666	7	6	61	176	415	668	791	763	515	208	48	8	(10)
(11)			CDD65	1106	0	0	4	15	79	233	328	300	129	18	0	0	(11)
(12)			CDH74	9345	0	0	22	118	601	1911	2981	2532	1034	145	1	0	(12)
(13)			CDH80	3092	0	0	1	14	137	618	1108	872	318	24	0	0	(13)
(14)	Wind		WSAvg	9.5	11.0	10.8	11.0	11.3	9.5	8.4	7.8	7.2	7.8	9.1	10.3	10.3	(14)
(15)	Precipitation		PrecAvg	39.90	2.30	2.50	3.80	3.70	4.00	3.50	4.50	3.60	2.90	2.60	3.20	3.30	(15)
(16)			PrecMax	44.60	6.20	5.40	9.00	8.10	9.30	7.10	11.10	8.30	5.70	5.70	5.60	6.00	(16)
(17)			PrecMin	31.70	1.00	0.40	1.30	1.00	1.40	0.70	1.20	0.70	0.20	0.00	0.80	0.40	(17)
(18)			PrecStd	4.10	1.30	1.40	2.00	2.00	2.20	1.70	2.60	2.10	1.60	1.50	1.30	1.50	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		62.2	64.6	77.0	82.2	87.1	91.7	96.8	93.7	92.5	83.3	71.3	63.3	(19)
(20)			MCWB		57.8	54.6	62.4	65.8	70.6	73.3	77.2	75.0	72.0	68.3	58.9	57.1	(20)
(21)		2%	DB		56.9	58.6	71.4	77.6	83.8	88.7	91.8	90.4	87.5	78.1	66.6	57.9	(21)
(22)			MCWB		53.8	51.7	59.0	63.0	69.4	73.2	76.4	75.2	70.0	64.7	58.4	53.8	(22)
(23)		5%	DB		51.7	53.7	66.1	73.4	80.8	86.3	88.9	87.8	83.4	73.8	62.8	53.1	(23)
(24)			MCWB		48.7	47.4	56.4	60.6	68.1	72.5	75.1	73.9	69.1	62.6	56.5	49.5	(24)
(25)		10%	DB		45.1	48.2	60.7	69.3	77.1	83.8	86.1	85.0	80.0	69.7	58.9	47.9	(25)
(26)			MCWB		41.5	42.8	53.2	58.6	66.6	71.1	73.5	72.3	67.4	60.5	53.3	44.2	(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		58.4	58.8	64.5	69.0	74.7	78.3	80.4	79.6	75.8	71.2	63.8	59.1	(27)
(28)			MCDB		61.4	62.5	74.1	78.4	82.7	87.6	91.2	88.7	85.8	79.0	67.6	62.8	(28)
(29)		2%	WB		54.6	53.2	61.1	65.3	72.1	76.2	78.6	77.7	73.7	67.7	60.2	54.7	(29)
(30)			MCDB		56.8	56.8	69.0	73.1	80.6	85.2	88.2	87.1	82.6	74.8	64.4	57.1	(30)
(31)		5%	WB		49.5	48.0	58.2	62.9	70.0	74.6	77.1	76.1	71.8	64.8	57.5	50.0	(31)
(32)			MCDB		51.6	53.1	64.8	70.4	77.8	83.3	85.9	84.3	79.3	71.3	61.9	52.7	(32)
(33)		10%	WB		41.9	43.3	53.8	60.2	68.0	73.0	75.5	74.5	69.9	62.0	53.9	44.6	(33)
(34)			MCDB		44.9	48.1	60.2	67.5	75.2	80.9	83.3	81.8	76.2	67.9	58.6	47.7	(34)
(35)	Mean Daily Temperature Range		MDBR		14.3	15.4	17.8	19.4	18.4	17.9	17.9	18.2	19.5	19.1	15.9	13.3	(35)
(36)		5% DB	MCDBR		18.4	22.8	23.4	23.3	21.4	20.0	20.1	20.1	23.0	23.7	20.4	18.1	(36)
(37)			MCWBR		16.0	17.3	14.1	13.6	11.0	8.9	8.1	8.2	9.6	12.7	14.9	15.3	(37)
(38)		5% WB	MCDBR		17.3	20.8	20.7	19.7	18.8	17.9	18.3	17.8	18.4	19.6	18.4	17.0	(38)
(39)			MCWBR		16.2	17.4	13.3	13.2	10.6	9.2	8.9	8.5	9.2	11.8	15.4	15.9	(39)
(40)	Clear-Sky Solar Irradiance		taub		0.303	0.324	0.348	0.404	0.438	0.442	0.457	0.453	0.418	0.359	0.321	0.301	(40)
(41)			taud		2.436	2.357	2.370	2.222	2.176	2.206	2.166	2.164	2.244	2.410	2.466	2.459	(41)
(42)			Ebn,noon		274	283	287	276	268	266	261	259	260	267	266	266	(42)
(43)			Edh,noon		26	32	35	43	47	45	47	46	40	30	25	23	(43)
(44)	All-Sky Solar Radiation		RadAvg		540	815	1161	1512	1715	1969	1948	1768	1439	1015	666	457	(44)
(45)			RadStd		47	79	108	128	171	113	116	110	117	94	67	47	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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JACKSONVILLE INTL, FL, USA WMO#: **722060**

Lat: **30.495N** Long: **81.694W** Elev: **26** StdP: **14.68** Time Zone: **-5.00 (NAE)** Period: **90-14** WBAN: **13889**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	29.5	32.8	16.2	12.5	39.4	20.7	15.6	41.1	23.3	58.7	20.5	58.6	3.9	320	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	17.9	94.5	77.2	92.7	76.9	90.9	76.5	79.9	89.4	79.1	88.2	78.3	86.9	8.0	240	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	77.4	142.7	83.3	76.7	139.3	82.6	76.1	136.2	82.1	43.2	89.3	42.4	88.2	41.7	87.2	83.5	(3)

Extreme Annual Design Conditions

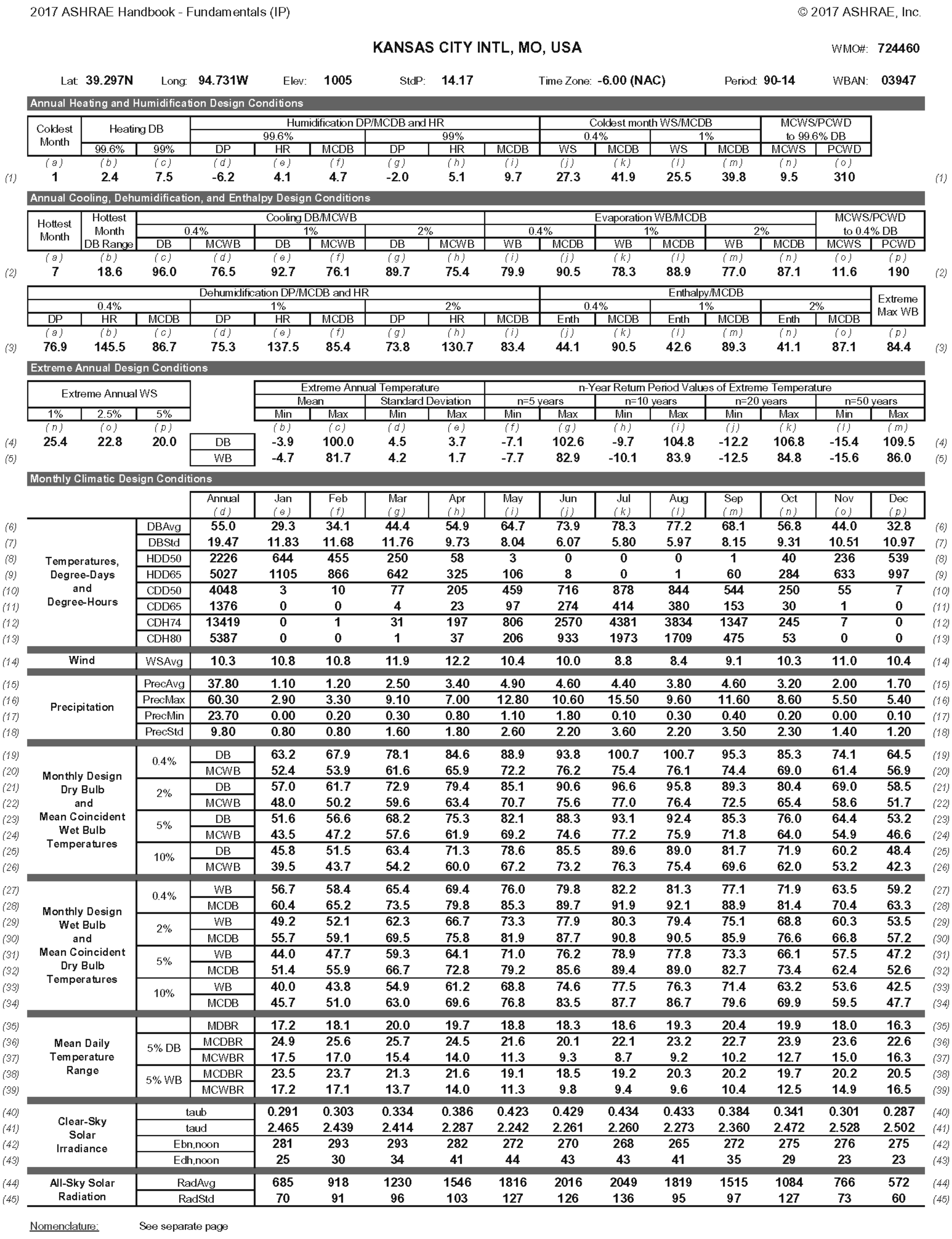
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	20.1	17.9	16.1	DB	23.8	97.7	2.8	2.3	21.8	99.3	20.2	100.7	18.6	101.9	16.6	103.6	(4)
(5)				WB	21.4	81.7	2.9	0.9	19.3	82.3	17.6	82.8	16.0	83.3	13.9	84.0	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	Temperatures, Degree-Days and Degree-Hours	DBAvg		68.6	54.0	56.9	61.7	67.3	74.2	79.7	82.0	81.5	78.0	70.3	61.4	55.9	(6)
(7)		DBStd		11.95	9.29	8.58	7.88	6.49	5.01	3.37	2.49	2.42	3.54	6.43	7.89	9.00	(7)
(8)		HDD50		158	67	29	10	0	0	0	0	0	0	0	8	44	(8)
(9)		HDD65		1303	354	249	155	51	4	0	0	0	0	28	158	304	(9)
(10)		CDD50		6960	191	222	372	520	751	890	991	976	841	630	350	226	(10)
(11)		CDD65		2629	13	21	52	121	290	440	526	511	391	193	50	21	(11)
(12)		CDH74		22894	79	141	423	1028	2599	3932	5114	4794	3110	1280	288	106	(12)
(13)		CDH80		8680	3	14	65	261	946	1673	2296	2009	1095	284	28	6	(13)
(14)	Wind	WSAvg		6.7	7.0	7.3	7.9	7.5	7.2	6.5	5.9	5.8	6.2	6.2	6.4	6.4	(14)
(15)	Precipitation	PrecAvg		51.30	3.30	3.90	3.70	2.80	3.60	5.70	5.60	7.90	7.10	2.90	2.20	2.70	(15)
(16)		PrecMax		70.60	7.30	8.90	10.20	11.60	10.40	12.90	16.20	16.20	17.80	9.80	4.60	5.90	(16)
(17)		PrecMin		37.50	0.30	0.50	0.70	0.40	0.90	2.60	2.00	2.40	1.00	0.20	0.10	0.20	(17)
(18)		PrecStd		8.10	1.90	2.10	2.80	2.40	3.10	3.10	3.10	3.80	4.00	2.40	1.40	1.60	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		80.0	82.2	85.0	89.2	93.4	97.2	96.8	95.9	92.6	87.9	83.2	80.4	(19)
(20)			MCWB		68.0	67.8	67.7	70.9	73.1	77.2	77.9	77.5	76.1	74.8	72.1	69.5	(20)
(21)		2%	DB		76.3	78.5	81.8	86.0	90.7	93.9	94.3	93.5	90.1	85.2	80.2	77.1	(21)
(22)			MCWB		66.5	66.1	66.6	69.4	72.5	76.5	77.8	78.0	75.8	73.5	69.7	68.0	(22)
(23)		5%	DB		72.6	74.7	78.9	83.0	88.3	91.2	92.5	91.5	88.1	83.3	77.5	73.6	(23)
(24)			MCWB		64.2	64.0	65.0	68.4	72.0	76.0	77.6	77.6	75.7	72.4	68.1	66.2	(24)
(25)		10%	DB		68.9	71.0	75.7	80.1	85.5	88.6	90.4	89.4	86.2	81.1	74.5	70.0	(25)
(26)			MCWB		62.5	62.1	63.6	66.9	71.5	75.7	77.4	77.5	75.3	71.4	66.8	64.5	(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		70.4	70.0	72.0	74.6	77.1	80.1	81.2	80.8	79.5	78.0	74.5	71.9	(27)
(28)			MCDB		76.0	77.6	79.9	82.4	86.8	91.1	91.4	90.4	86.4	83.7	79.7	77.3	(28)
(29)		2%	WB		68.3	68.2	69.6	72.4	75.5	78.5	79.9	79.9	78.4	76.5	72.4	69.7	(29)
(30)			MCDB		73.5	75.3	77.1	80.5	84.5	88.4	89.9	89.2	85.2	81.8	76.9	74.5	(30)
(31)		5%	WB		66.1	66.3	68.0	70.9	74.5	77.7	79.0	79.0	77.7	75.3	70.5	67.9	(31)
(32)			MCDB		71.1	72.1	74.7	78.9	83.3	86.9	88.6	87.8	84.5	80.1	74.8	72.1	(32)
(33)		10%	WB		63.8	64.3	66.1	69.4	73.3	76.9	78.2	78.2	76.9	73.8	68.7	65.5	(33)
(34)			MCDB		68.3	69.2	72.6	76.8	81.8	85.4	87.3	86.4	83.3	78.3	73.0	69.7	(34)
(35)	Mean Daily Temperature Range		MDBR		21.8	21.8	22.6	22.7	21.1	18.3	17.9	16.8	15.9	18.8	21.3	21.0	(35)
(36)		5% DB	MCDBR		23.7	23.7	25.1	24.5	23.6	21.8	20.2	19.5	18.5	19.1	21.4	21.7	(36)
(37)			MCWBR		14.4	12.8	11.9	10.3	8.6	7.7	6.9	6.9	7.1	8.9	11.7	13.1	(37)
(38)		5% WB	MCDBR		20.5	20.1	20.5	20.4	20.0	18.8	18.4	17.3	15.6	15.0	17.8	19.1	(38)
(39)			MCWBR		13.6	12.2	11.5	9.9	8.2	7.4	7.2	6.9	6.7	7.8	11.2	12.5	(39)
(40)	Clear-Sky Solar Irradiance	taub			0.336	0.347	0.372	0.395	0.438	0.482	0.510	0.494	0.429	0.399	0.359	0.346	(40)
(41)		taud			2.528	2.502	2.442	2.366	2.283	2.219	2.146	2.197	2.377	2.425	2.488	2.528	(41)
(42)		Ebn,noon			284	291	289	285	272	259	251	254	268	269	273	275	(42)
(43)		Edh,noon			28	31	35	39	43	45	49	46	37	33	29	27	(43)
(44)	All-Sky Solar Radiation	RadAvg			992	1177	1553	1893	2013	1874	1851	1746	1511	1343	1081	885	(44)
(45)		RadStd			69	133	145	110	109	115	110	118	108	132	57	74	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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KANSAS CITY INTL, MO, USA WMO#: **724460**

Lat: **39.297N** Long: **94.731W** Elev: **1005** StdP: **14.17** Time Zone: **-6.00 (NAC)** Period: **90-14** WBAN: **03947**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB 99.6%	Heating DB 99%	Humidification DP/MCDB and HR 99.6% DP	99.6% HR	99.6% MCDB	99% DP	99% HR	99% MCDB	Coldest month WS/MCDB 0.4% WS	0.4% MCDB	1% WS	1% MCDB	MCWS/PCWD to 99.6% DB MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	2.4	7.5	-6.2	4.1	4.7	-2.0	5.1	9.7	27.3	41.9	25.5	39.8	9.5	310	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB 0.4% DB	0.4% MCWB	1% DB	1% MCWB	2% DB	2% MCWB	Evaporation WB/MCDB 0.4% WB	0.4% MCDB	1% WB	1% MCDB	2% WB	2% MCDB	MCWS/PCWD to 0.4% DB MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	18.6	96.0	76.5	92.7	76.1	89.7	75.4	79.9	90.5	78.3	88.9	77.0	87.1	11.6	190	(2)

	Dehumidification DP/MCDB and HR 0.4% DP	0.4% HR	0.4% MCDB	1% DP	1% HR	1% MCDB	2% DP	2% HR	2% MCDB	Enthalpy/MCDB 0.4% Enth	0.4% MCDB	1% Enth	1% MCDB	2% Enth	2% MCDB	Extreme Max WB	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	76.9	145.5	86.7	75.3	137.5	85.4	73.8	130.7	83.4	44.1	90.5	42.6	89.3	41.1	87.1	84.4	(3)

Extreme Annual Design Conditions

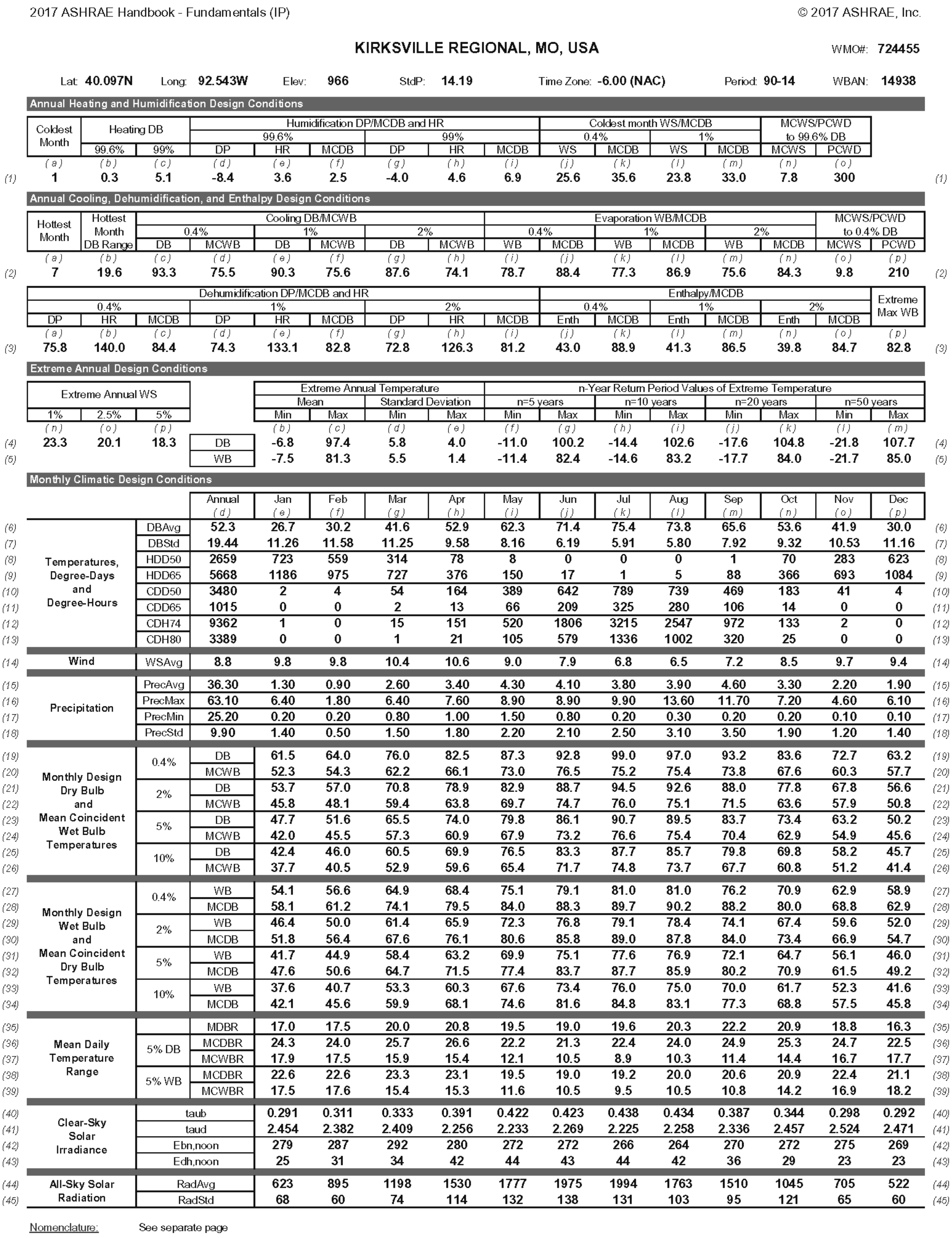
	Extreme Annual WS 1%	2.5%	5%		Extreme Annual Temperature Mean Min	Mean Max	Standard Deviation Min	Standard Deviation Max	n-Year Return Period Values of Extreme Temperature n=5 years Min	n=5 years Max	n=10 years Min	n=10 years Max	n=20 years Min	n=20 years Max	n=50 years Min	n=50 years Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	25.4	22.8	20.0	DB	-3.9	100.0	4.5	3.7	-7.1	102.6	-9.7	104.8	-12.2	106.8	-15.4	109.5	(4)
(5)				WB	-4.7	81.7	4.2	1.7	-7.7	82.9	-10.1	83.9	-12.5	84.8	-15.6	86.0	(5)

Monthly Climatic Design Conditions

				Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	
				(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	55.0	29.3	34.1	44.4	54.9	64.7	73.9	78.3	77.2	68.1	56.8	44.0	32.8	(6)
(7)			DBStd	19.47	11.83	11.68	11.76	9.73	8.04	6.07	5.80	5.97	8.15	9.31	10.51	10.97	(7)
(8)			HDD50	2226	644	455	250	58	3	0	0	0	1	40	236	539	(8)
(9)			HDD65	5027	1105	866	642	325	106	8	0	1	60	284	633	997	(9)
(10)			CDD50	4048	3	10	77	205	459	716	878	844	544	250	55	7	(10)
(11)			CDD65	1376	0	0	4	23	97	274	414	380	153	30	1	0	(11)
(12)			CDH74	13419	0	1	31	197	806	2570	4381	3834	1347	245	7	0	(12)
(13)			CDH80	5387	0	0	1	37	206	933	1973	1709	475	53	0	0	(13)
(14)	Wind		WSAvg	10.3	10.8	10.8	11.9	12.2	10.4	10.0	8.8	8.4	9.1	10.3	11.0	10.4	(14)
(15)	Precipitation		PrecAvg	37.80	1.10	1.20	2.50	3.40	4.90	4.60	4.40	3.80	4.60	3.20	2.00	1.70	(15)
(16)			PrecMax	60.30	2.90	3.30	9.10	7.00	12.80	10.60	15.50	9.60	11.60	8.60	5.50	5.40	(16)
(17)			PrecMin	23.70	0.00	0.20	0.30	0.80	1.10	1.80	0.10	0.30	0.40	0.20	0.00	0.10	(17)
(18)			PrecStd	9.80	0.80	0.80	1.60	1.80	2.60	2.20	3.60	2.20	3.50	2.30	1.40	1.20	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		63.2	67.9	78.1	84.6	88.9	93.8	100.7	100.7	95.3	85.3	74.1	64.5	(19)
(20)			MCWB		52.4	53.9	61.6	65.9	72.2	76.2	75.4	76.1	74.4	69.0	61.4	56.9	(20)
(21)		2%	DB		57.0	61.7	72.9	79.4	85.1	90.6	96.6	95.8	89.3	80.4	69.0	58.5	(21)
(22)			MCWB		48.0	50.2	59.6	63.4	70.7	75.6	77.0	76.4	72.5	65.4	58.6	51.7	(22)
(23)		5%	DB		51.6	56.6	68.2	75.3	82.1	88.3	93.1	92.4	85.3	76.0	64.4	53.2	(23)
(24)			MCWB		43.5	47.2	57.6	61.9	69.2	74.6	77.2	75.9	71.8	64.0	54.9	46.6	(24)
(25)		10%	DB		45.8	51.5	63.4	71.3	78.6	85.5	89.6	89.0	81.7	71.9	60.2	48.4	(25)
(26)			MCWB		39.5	43.7	54.2	60.0	67.2	73.2	76.3	75.4	69.6	62.0	53.2	42.3	(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		56.7	58.4	65.4	69.4	76.0	79.8	82.2	81.3	77.1	71.9	63.5	59.2	(27)
(28)			MCDB		60.4	65.2	73.5	79.8	85.3	89.7	91.9	92.1	88.9	81.4	70.4	63.3	(28)
(29)		2%	WB		49.2	52.1	62.3	66.7	73.3	77.9	80.3	79.4	75.1	68.8	60.3	53.5	(29)
(30)			MCDB		55.7	59.1	69.5	75.8	81.9	87.7	90.8	90.5	85.9	76.6	66.8	57.2	(30)
(31)		5%	WB		44.0	47.7	59.3	64.1	71.0	76.2	78.9	77.8	73.3	66.1	57.5	47.2	(31)
(32)			MCDB		51.4	55.9	66.7	72.8	79.2	85.6	89.4	89.0	82.7	73.4	62.4	52.6	(32)
(33)		10%	WB		40.0	43.8	54.9	61.2	68.8	74.6	77.5	76.3	71.4	63.2	53.6	42.5	(33)
(34)			MCDB		45.7	51.0	63.0	69.6	76.8	83.5	87.7	86.7	79.6	69.9	59.5	47.7	(34)
(35)	Mean Daily Temperature Range		MDBR		17.2	18.1	20.0	19.7	18.8	18.3	18.6	19.3	20.4	19.9	18.0	16.3	(35)
(36)		5% DB	MCDBR		24.9	25.6	25.7	24.5	21.6	20.1	22.1	23.2	22.7	23.9	23.6	22.6	(36)
(37)			MCWBR		17.5	17.0	15.4	14.0	11.3	9.3	8.7	9.2	10.2	12.7	15.0	16.3	(37)
(38)		5% WB	MCDBR		23.5	23.7	21.3	21.6	19.1	18.5	19.2	20.3	20.2	19.7	20.2	20.5	(38)
(39)			MCWBR		17.2	17.1	13.7	14.0	11.3	9.8	9.4	9.6	10.4	12.5	14.9	16.5	(39)
(40)	Clear-Sky Solar Irradiance		taub		0.291	0.303	0.334	0.386	0.423	0.429	0.434	0.433	0.384	0.341	0.301	0.287	(40)
(41)			taud		2.465	2.439	2.414	2.287	2.242	2.261	2.260	2.273	2.360	2.472	2.528	2.502	(41)
(42)			Ebn,noon		281	293	293	282	272	270	268	265	272	275	276	275	(42)
(43)			Edh,noon		25	30	34	41	44	43	43	41	35	29	23	23	(43)
(44)	All-Sky Solar Radiation		RadAvg		685	918	1230	1546	1816	2016	2049	1819	1515	1084	766	572	(44)
(45)			RadStd		70	91	96	103	127	126	136	95	97	127	73	60	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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KIRKSVILLE REGIONAL, MO, USA WMO#: **724455**

Lat: **40.097N** Long: **92.543W** Elev: **966** StdP: **14.19** Time Zone: **-6.00 (NAC)** Period: **90-14** WBAN: **14938**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	**1**	**0.3**	**5.1**	**-8.4**	**3.6**	**2.5**	**-4.0**	**4.6**	**6.9**	**25.6**	**35.6**	**23.8**	**33.0**	**7.8**	**300**	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	**7**	**19.6**	**93.3**	**75.5**	**90.3**	**75.6**	**87.6**	**74.1**	**78.7**	**88.4**	**77.3**	**86.9**	**75.6**	**84.3**	**9.8**	**210**	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	**75.8**	**140.0**	**84.4**	**74.3**	**133.1**	**82.8**	**72.8**	**126.3**	**81.2**	**43.0**	**88.9**	**41.3**	**86.5**	**39.8**	**84.7**	**82.8**	(3)

Extreme Annual Design Conditions

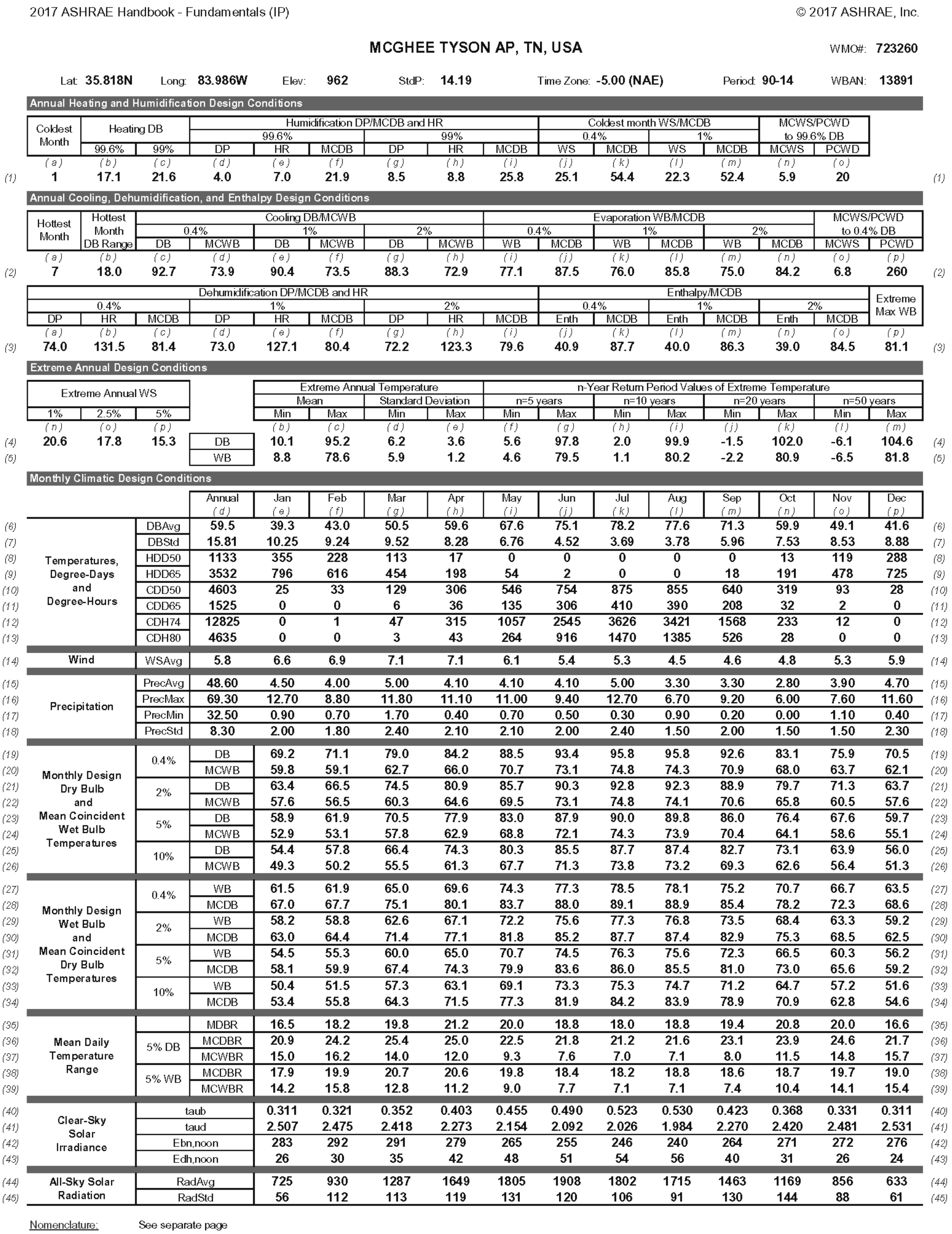
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	**23.3**	**20.1**	**18.3**	DB	**-6.8**	**97.4**	**5.8**	**4.0**	**-11.0**	**100.2**	**-14.4**	**102.6**	**-17.6**	**104.8**	**-21.8**	**107.7**	(4)
(5)				WB	**-7.5**	**81.3**	**5.5**	**1.4**	**-11.4**	**82.4**	**-14.6**	**83.2**	**-17.7**	**84.0**	**-21.7**	**85.0**	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**	DBAvg		**52.3**	**26.7**	**30.2**	**41.6**	**52.9**	**62.3**	**71.4**	**75.4**	**73.8**	**65.6**	**53.6**	**41.9**	**30.0**	(6)
(7)		DBStd		**19.44**	**11.26**	**11.58**	**11.25**	**9.58**	**8.16**	**6.19**	**5.91**	**5.80**	**7.92**	**9.32**	**10.53**	**11.16**	(7)
(8)		HDD50		**2659**	**723**	**559**	**314**	**78**	**8**	**0**	**0**	**0**	**1**	**70**	**283**	**623**	(8)
(9)		HDD65		**5668**	**1186**	**975**	**727**	**376**	**150**	**17**	**1**	**5**	**88**	**366**	**693**	**1084**	(9)
(10)		CDD50		**3480**	**2**	**4**	**54**	**164**	**389**	**642**	**789**	**739**	**469**	**183**	**41**	**4**	(10)
(11)		CDD65		**1015**	**0**	**0**	**2**	**13**	**66**	**209**	**325**	**280**	**106**	**14**	**0**	**0**	(11)
(12)		CDH74		**9362**	**1**	**0**	**15**	**151**	**520**	**1806**	**3215**	**2547**	**972**	**133**	**2**	**0**	(12)
(13)		CDH80		**3389**	**0**	**0**	**1**	**21**	**105**	**579**	**1336**	**1002**	**320**	**25**	**0**	**0**	(13)
(14)	**Wind**	WSAvg		**8.8**	**9.8**	**9.8**	**10.4**	**10.6**	**9.0**	**7.9**	**6.8**	**6.5**	**7.2**	**8.5**	**9.7**	**9.4**	(14)
(15)	**Precipitation**	PrecAvg		**36.30**	**1.30**	**0.90**	**2.60**	**3.40**	**4.30**	**4.10**	**3.80**	**3.90**	**4.60**	**3.30**	**2.20**	**1.90**	(15)
(16)		PrecMax		**63.10**	**6.40**	**1.80**	**6.40**	**7.60**	**8.90**	**8.90**	**9.90**	**13.60**	**11.70**	**7.20**	**4.60**	**6.10**	(16)
(17)		PrecMin		**25.20**	**0.20**	**0.20**	**0.80**	**1.00**	**1.50**	**0.80**	**0.20**	**0.30**	**0.20**	**0.20**	**0.10**	**0.10**	(17)
(18)		PrecStd		**9.90**	**1.40**	**0.50**	**1.50**	**1.80**	**2.20**	**2.10**	**2.50**	**3.10**	**3.50**	**1.90**	**1.20**	**1.40**	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB		**61.5**	**64.0**	**76.0**	**82.5**	**87.3**	**92.8**	**99.0**	**97.0**	**93.2**	**83.6**	**72.7**	**63.2**	(19)
(20)			MCWB		**52.3**	**54.3**	**62.2**	**66.1**	**73.0**	**76.5**	**75.2**	**75.4**	**73.8**	**67.6**	**60.3**	**57.7**	(20)
(21)		2%	DB		**53.7**	**57.0**	**70.8**	**78.9**	**82.9**	**88.7**	**94.5**	**92.6**	**88.0**	**77.8**	**67.8**	**56.6**	(21)
(22)			MCWB		**45.8**	**48.1**	**59.4**	**63.8**	**69.7**	**74.7**	**76.0**	**75.1**	**71.5**	**63.6**	**57.9**	**50.8**	(22)
(23)		5%	DB		**47.7**	**51.6**	**65.5**	**74.0**	**79.8**	**86.1**	**90.7**	**89.5**	**83.7**	**73.4**	**63.2**	**50.2**	(23)
(24)			MCWB		**42.0**	**45.5**	**57.3**	**60.9**	**67.9**	**73.2**	**76.6**	**75.4**	**70.4**	**62.9**	**54.9**	**45.6**	(24)
(25)		10%	DB		**42.4**	**46.0**	**60.5**	**69.9**	**76.5**	**83.3**	**87.7**	**85.7**	**79.8**	**69.8**	**58.2**	**45.7**	(25)
(26)			MCWB		**37.7**	**40.5**	**52.9**	**59.6**	**65.4**	**71.7**	**74.8**	**73.7**	**67.7**	**60.8**	**51.2**	**41.4**	(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB		**54.1**	**56.6**	**64.9**	**68.4**	**75.1**	**79.1**	**81.0**	**81.0**	**76.2**	**70.9**	**62.9**	**58.9**	(27)
(28)			MCDB		**58.1**	**61.2**	**74.1**	**79.5**	**84.0**	**88.3**	**89.7**	**90.2**	**88.2**	**80.0**	**68.8**	**62.9**	(28)
(29)		2%	WB		**46.4**	**50.0**	**61.4**	**65.9**	**72.3**	**76.8**	**79.1**	**78.4**	**74.1**	**67.4**	**59.6**	**52.0**	(29)
(30)			MCDB		**51.8**	**56.4**	**67.6**	**76.1**	**80.6**	**85.8**	**89.0**	**87.8**	**84.0**	**73.4**	**66.9**	**54.7**	(30)
(31)		5%	WB		**41.7**	**44.9**	**58.4**	**63.2**	**69.9**	**75.1**	**77.6**	**76.9**	**72.1**	**64.7**	**56.1**	**46.0**	(31)
(32)			MCDB		**47.6**	**50.6**	**64.7**	**71.5**	**77.4**	**83.7**	**87.7**	**85.9**	**80.2**	**70.9**	**61.5**	**49.2**	(32)
(33)		10%	WB		**37.6**	**40.7**	**53.3**	**60.3**	**67.6**	**73.4**	**76.0**	**75.0**	**70.0**	**61.7**	**52.3**	**41.6**	(33)
(34)			MCDB		**42.1**	**45.6**	**59.9**	**68.1**	**74.6**	**81.6**	**84.8**	**83.1**	**77.3**	**68.8**	**57.5**	**45.8**	(34)
(35)	**Mean Daily Temperature Range**		MDBR		**17.0**	**17.5**	**20.0**	**20.8**	**19.5**	**19.0**	**19.6**	**20.3**	**22.2**	**20.9**	**18.8**	**16.3**	(35)
(36)		5% DB	MCDBR		**24.3**	**24.0**	**25.7**	**26.6**	**22.2**	**21.3**	**22.4**	**24.0**	**24.9**	**25.3**	**24.7**	**22.5**	(36)
(37)			MCWBR		**17.9**	**17.5**	**15.9**	**15.4**	**12.1**	**10.5**	**8.9**	**10.3**	**11.4**	**14.4**	**16.7**	**17.7**	(37)
(38)		5% WB	MCDBR		**22.6**	**22.6**	**23.3**	**23.1**	**19.5**	**19.0**	**19.2**	**20.0**	**20.6**	**20.9**	**22.4**	**21.1**	(38)
(39)			MCWBR		**17.5**	**17.6**	**15.4**	**15.3**	**11.6**	**10.5**	**9.5**	**10.5**	**10.8**	**14.2**	**16.9**	**18.2**	(39)
(40)	**Clear-Sky Solar Irradiance**	taub			**0.291**	**0.311**	**0.333**	**0.391**	**0.422**	**0.423**	**0.438**	**0.434**	**0.387**	**0.344**	**0.298**	**0.292**	(40)
(41)		taud			**2.454**	**2.382**	**2.409**	**2.256**	**2.233**	**2.269**	**2.225**	**2.258**	**2.336**	**2.457**	**2.524**	**2.471**	(41)
(42)		Ebn,noon			**279**	**287**	**292**	**280**	**272**	**272**	**266**	**264**	**270**	**272**	**275**	**269**	(42)
(43)		Edh,noon			**25**	**31**	**34**	**42**	**44**	**43**	**44**	**42**	**36**	**29**	**23**	**23**	(43)
(44)	**All-Sky Solar Radiation**	RadAvg			**623**	**895**	**1198**	**1530**	**1777**	**1975**	**1994**	**1763**	**1510**	**1045**	**705**	**522**	(44)
(45)		RadStd			**68**	**60**	**74**	**114**	**132**	**138**	**131**	**103**	**95**	**121**	**65**	**60**	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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MCGHEE TYSON AP, TN, USA WMO#: **723260**

Lat: **35.818N** Long: **83.986W** Elev: **962** StdP: **14.19** Time Zone: **-5.00 (NAE)** Period: **90-14** WBAN: **13891**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	17.1	21.6	4.0	7.0	21.9	8.5	8.8	25.8	25.1	54.4	22.3	52.4	5.9	20	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	18.0	92.7	73.9	90.4	73.5	88.3	72.9	77.1	87.5	76.0	85.8	75.0	84.2	6.8	260	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	74.0	131.5	81.4	73.0	127.1	80.4	72.2	123.3	79.6	40.9	87.7	40.0	86.3	39.0	84.5	81.1	(3)

Extreme Annual Design Conditions

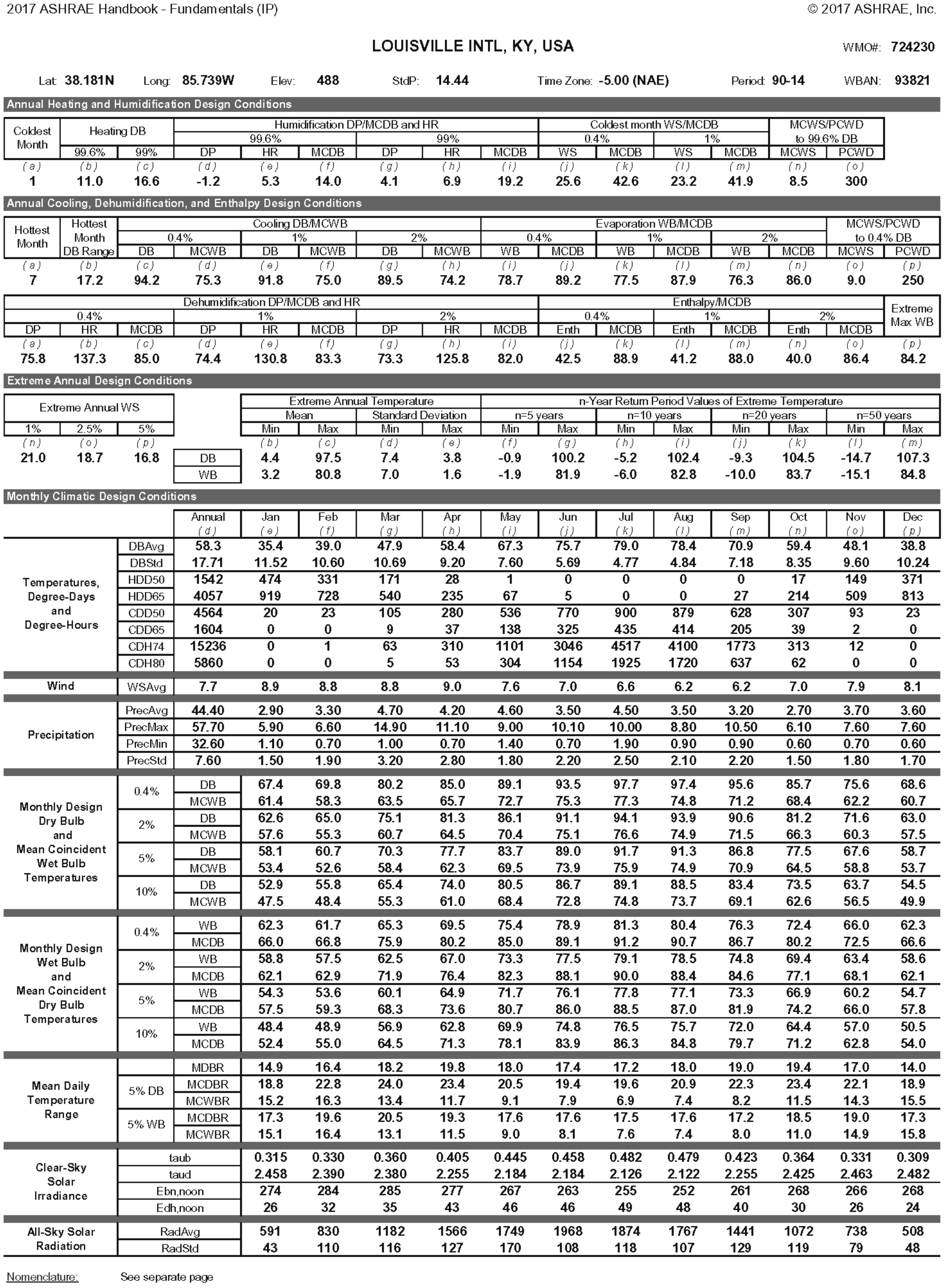
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	20.6	17.8	15.3	DB	10.1	95.2	6.2	3.6	5.6	97.8	2.0	99.9	-1.5	102.0	-6.1	104.6	(4)
(5)				WB	8.8	78.6	5.9	1.2	4.6	79.5	1.1	80.2	-2.2	80.9	-6.5	81.8	(5)

Monthly Climatic Design Conditions

				Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	
				(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	59.5	39.3	43.0	50.5	59.6	67.6	75.1	78.2	77.6	71.3	59.9	49.1	41.6	(6)
(7)			DBStd	15.81	10.25	9.24	9.52	8.28	6.76	4.52	3.69	3.78	5.96	7.53	8.53	8.88	(7)
(8)			HDD50	1133	355	228	113	17	0	0	0	0	0	13	119	288	(8)
(9)			HDD65	3532	796	616	454	198	54	2	0	0	18	191	478	725	(9)
(10)			CDD50	4603	25	33	129	306	546	754	875	855	640	319	93	28	(10)
(11)			CDD65	1525	0	0	6	36	135	306	410	390	208	32	2	0	(11)
(12)			CDH74	12825	0	1	47	315	1057	2545	3626	3421	1568	233	12	0	(12)
(13)			CDH80	4635	0	0	3	43	264	916	1470	1385	526	28	0	0	(13)
(14)	Wind		WSAvg	5.8	6.6	6.9	7.1	7.1	6.1	5.4	5.3	4.5	4.6	4.8	5.3	5.9	(14)
(15)	Precipitation		PrecAvg	48.60	4.50	4.00	5.00	4.10	4.10	4.10	5.00	3.30	3.30	2.80	3.90	4.70	(15)
(16)			PrecMax	69.30	12.70	8.80	11.80	11.10	11.00	9.40	12.70	6.70	9.20	6.00	7.60	11.60	(16)
(17)			PrecMin	32.50	0.90	0.70	1.70	0.40	0.70	0.50	0.30	0.90	0.20	0.00	1.10	0.40	(17)
(18)			PrecStd	8.30	2.00	1.80	2.40	2.10	2.10	2.00	2.40	1.50	2.00	1.50	1.50	2.30	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		69.2	71.1	79.0	84.2	88.5	93.4	95.8	95.8	92.6	83.1	75.9	70.5	(19)
(20)			MCWB		59.8	59.1	62.7	66.0	70.7	73.1	74.8	74.3	70.9	68.0	63.7	62.1	(20)
(21)		2%	DB		63.4	66.5	74.5	80.9	85.7	90.3	92.8	92.3	88.9	79.7	71.3	63.7	(21)
(22)			MCWB		57.6	56.5	60.3	64.6	69.5	73.1	74.8	74.1	70.6	65.8	60.5	57.6	(22)
(23)		5%	DB		58.9	61.9	70.5	77.9	83.0	87.9	90.0	89.8	86.0	76.4	67.6	59.7	(23)
(24)			MCWB		52.9	53.1	57.8	62.9	68.8	72.1	74.3	73.9	70.4	64.1	58.6	55.1	(24)
(25)		10%	DB		54.4	57.8	66.4	74.3	80.3	85.5	87.7	87.4	82.7	73.1	63.9	56.0	(25)
(26)			MCWB		49.3	50.2	55.5	61.3	67.7	71.3	73.8	73.2	69.3	62.6	56.4	51.3	(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		61.5	61.9	65.0	69.6	74.3	77.3	78.5	78.1	75.2	70.7	66.7	63.5	(27)
(28)			MCDB		67.0	67.7	75.1	80.1	83.7	88.0	89.1	88.9	85.4	78.2	72.3	68.6	(28)
(29)		2%	WB		58.2	58.8	62.6	67.1	72.2	75.6	77.3	76.8	73.5	68.4	63.3	59.2	(29)
(30)			MCDB		63.0	64.4	71.4	77.1	81.8	85.2	87.7	87.4	82.9	75.3	68.5	62.5	(30)
(31)		5%	WB		54.5	55.3	60.0	65.0	70.7	74.5	76.3	75.6	72.3	66.5	60.3	56.2	(31)
(32)			MCDB		58.1	59.9	67.4	74.3	79.9	83.6	86.0	85.5	81.0	73.0	65.6	59.2	(32)
(33)		10%	WB		50.4	51.5	57.3	63.1	69.1	73.3	75.3	74.7	71.2	64.7	57.2	51.6	(33)
(34)			MCDB		53.4	55.8	64.3	71.5	77.3	81.9	84.2	83.9	78.9	70.9	62.8	54.6	(34)
(35)	Mean Daily Temperature Range		MDBR		16.5	18.2	19.8	21.2	20.0	18.8	18.0	18.8	19.4	20.8	20.0	16.6	(35)
(36)		5% DB	MCDBR		20.9	24.2	25.4	25.0	22.5	21.8	21.2	21.6	23.1	23.9	24.6	21.7	(36)
(37)			MCWBR		15.0	16.2	14.0	12.0	9.3	7.6	7.0	7.1	8.0	11.5	14.8	15.7	(37)
(38)		5% WB	MCDBR		17.9	19.9	20.7	20.6	19.8	18.4	18.2	18.8	18.6	18.7	19.7	19.0	(38)
(39)			MCWBR		14.2	15.8	12.8	11.2	9.0	7.7	7.1	7.1	7.4	10.4	14.1	15.4	(39)
(40)	Clear-Sky Solar Irradiance		taub		0.311	0.321	0.352	0.403	0.455	0.490	0.523	0.530	0.423	0.368	0.331	0.311	(40)
(41)			taud		2.507	2.475	2.418	2.273	2.154	2.092	2.026	1.984	2.270	2.420	2.481	2.531	(41)
(42)			Ebn,noon		283	292	291	279	265	255	246	240	264	271	272	276	(42)
(43)			Edh,noon		26	30	35	42	48	51	54	56	40	31	26	24	(43)
(44)	All-Sky Solar Radiation		RadAvg		725	930	1287	1649	1805	1908	1802	1715	1463	1169	856	633	(44)
(45)			RadStd		56	112	113	119	131	120	106	91	130	144	88	61	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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LOUISVILLE INTL, KY, USA WMO#: **724230**

Lat: **38.181N** Long: **85.739W** Elev: **488** StdP: **14.44** Time Zone: **-5.00 (NAE)** Period: **90-14** WBAN: **93821**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	11.0	16.6	-1.2	5.3	14.0	4.1	6.9	19.2	25.6	42.6	23.2	41.9	8.5	300	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	17.2	94.2	75.3	91.8	75.0	89.5	74.2	78.7	89.2	77.5	87.9	76.3	86.0	9.0	250	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	75.8	137.3	85.0	74.4	130.8	83.3	73.3	125.8	82.0	42.5	88.9	41.2	88.0	40.0	86.4	84.2	(3)

Extreme Annual Design Conditions

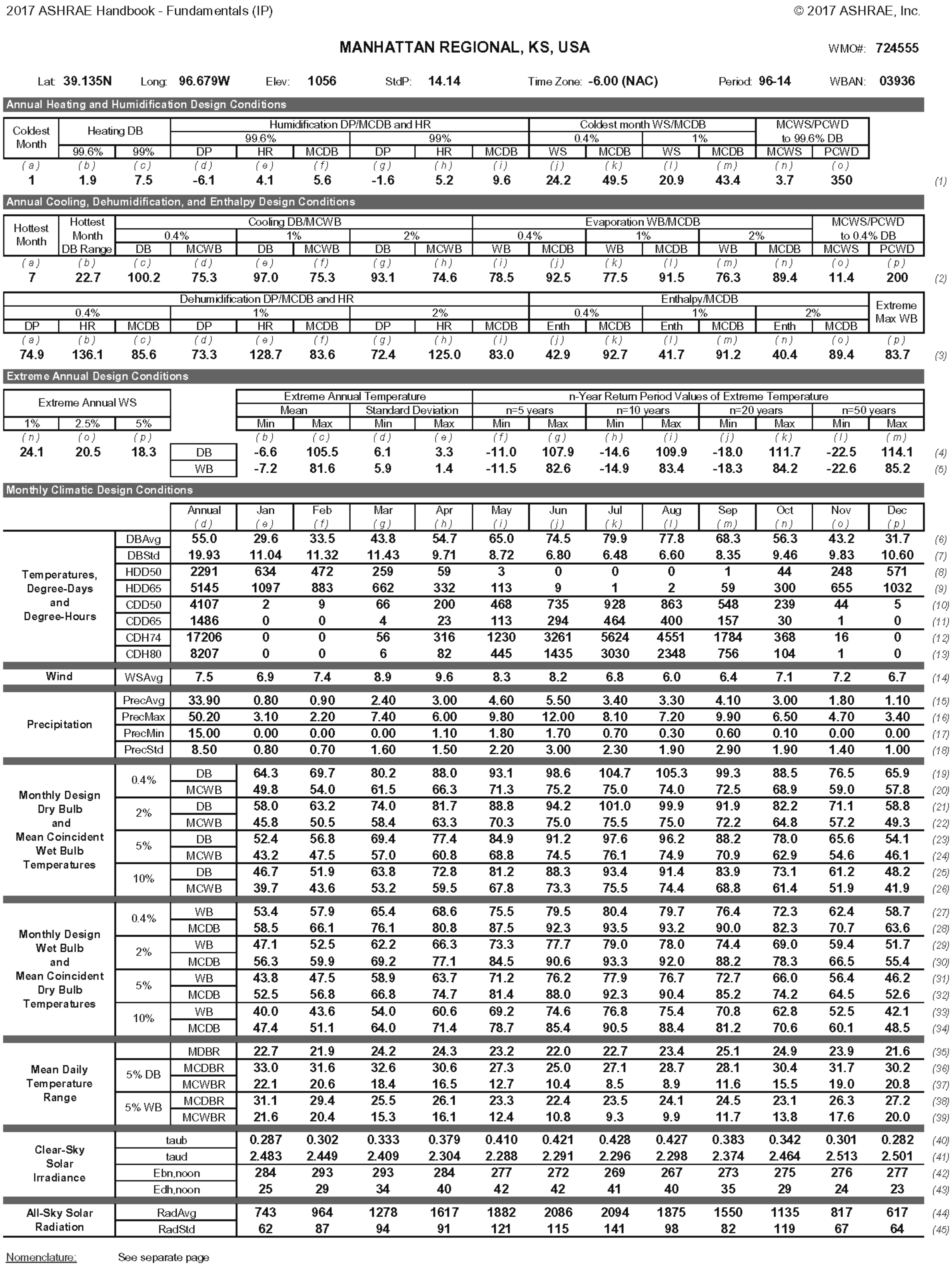
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	21.0	18.7	16.8	DB	4.4	97.5	7.4	3.8	-0.9	100.2	-5.2	102.4	-9.3	104.5	-14.7	107.3	(4)
(5)				WB	3.2	80.8	7.0	1.6	-1.9	81.9	-6.0	82.8	-10.0	83.7	-15.1	84.8	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**		DBAvg	58.3	35.4	39.0	47.9	58.4	67.3	75.7	79.0	78.4	70.9	59.4	48.1	38.8	(6)
(7)			DBStd	17.71	11.52	10.60	10.69	9.20	7.60	5.69	4.77	4.84	7.18	8.35	9.60	10.24	(7)
(8)			HDD50	1542	474	331	171	28	1	0	0	0	0	17	149	371	(8)
(9)			HDD65	4057	919	728	540	235	67	5	0	0	27	214	509	813	(9)
(10)			CDD50	4564	20	23	105	280	536	770	900	879	628	307	93	23	(10)
(11)			CDD65	1604	0	0	9	37	138	325	435	414	205	39	2	0	(11)
(12)			CDH74	15236	0	1	63	310	1101	3046	4517	4100	1773	313	12	0	(12)
(13)			CDH80	5860	0	0	5	53	304	1154	1925	1720	637	62	0	0	(13)
(14)	**Wind**		WSAvg	7.7	8.9	8.8	8.8	9.0	7.6	7.0	6.6	6.2	6.2	7.0	7.9	8.1	(14)
(15)	**Precipitation**		PrecAvg	44.40	2.90	3.30	4.70	4.20	4.60	3.50	4.50	3.50	3.20	2.70	3.70	3.60	(15)
(16)			PrecMax	57.70	5.90	6.60	14.90	11.10	9.00	10.10	10.00	8.80	10.50	6.10	7.60	7.60	(16)
(17)			PrecMin	32.60	1.10	0.70	1.00	0.70	1.40	0.70	1.90	0.90	0.90	0.60	0.70	0.60	(17)
(18)			PrecStd	7.60	1.50	1.90	3.20	2.80	1.80	2.20	2.50	2.10	2.20	1.50	1.80	1.70	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB		67.4	69.8	80.2	85.0	89.1	93.5	97.7	97.4	95.6	85.7	75.6	68.6	(19)
(20)			MCWB		61.4	58.3	63.5	65.7	72.7	75.3	77.3	74.8	71.2	68.4	62.2	60.7	(20)
(21)		2%	DB		62.6	65.0	75.1	81.3	86.1	91.1	94.1	93.9	90.6	81.2	71.6	63.0	(21)
(22)			MCWB		57.6	55.3	60.7	64.5	70.4	75.1	76.6	74.9	71.5	66.3	60.3	57.5	(22)
(23)		5%	DB		58.1	60.7	70.3	77.7	83.7	89.0	91.7	91.3	86.8	77.5	67.6	58.7	(23)
(24)			MCWB		53.4	52.6	58.4	62.3	69.5	73.9	75.9	74.9	70.9	64.5	58.8	53.7	(24)
(25)		10%	DB		52.9	55.8	65.4	74.0	80.5	86.7	89.1	88.5	83.4	73.5	63.7	54.5	(25)
(26)			MCWB		47.5	48.4	55.3	61.0	68.4	72.8	74.8	73.7	69.1	62.6	56.5	49.9	(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB		62.3	61.7	65.3	69.5	75.4	78.9	81.3	80.4	76.3	72.4	66.0	62.3	(27)
(28)			MCDB		66.0	66.8	75.9	80.2	85.0	89.1	91.2	90.7	86.7	80.2	72.5	66.6	(28)
(29)		2%	WB		58.8	57.5	62.5	67.0	73.3	77.5	79.1	78.5	74.8	69.4	63.4	58.6	(29)
(30)			MCDB		62.1	62.9	71.9	76.4	82.3	88.1	90.0	88.4	84.6	77.1	68.1	62.1	(30)
(31)		5%	WB		54.3	53.6	60.1	64.9	71.7	76.1	77.8	77.1	73.3	66.9	60.2	54.7	(31)
(32)			MCDB		57.5	59.3	68.3	73.6	80.7	86.0	88.5	87.0	81.9	74.2	66.0	57.8	(32)
(33)		10%	WB		48.4	48.9	56.9	62.8	69.9	74.8	76.5	75.7	72.0	64.4	57.0	50.5	(33)
(34)			MCDB		52.4	55.0	64.5	71.3	78.1	83.9	86.3	84.8	79.7	71.2	62.8	54.0	(34)
(35)	**Mean Daily Temperature Range**		MDBR		14.9	16.4	18.2	19.8	18.0	17.4	17.2	18.0	19.0	19.4	17.0	14.0	(35)
(36)		5% DB	MCDBR		18.8	22.8	24.0	23.4	20.5	19.4	19.6	20.9	22.3	23.4	22.1	18.9	(36)
(37)			MCWBR		15.2	16.3	13.4	11.7	9.1	7.9	6.9	7.4	8.2	11.5	14.3	15.5	(37)
(38)		5% WB	MCDBR		17.3	19.6	20.5	19.3	17.6	17.6	17.5	17.6	17.2	18.5	19.0	17.3	(38)
(39)			MCWBR		15.1	16.4	13.1	11.5	9.0	8.1	7.6	7.4	8.0	11.0	14.9	15.8	(39)
(40)	**Clear-Sky Solar Irradiance**		taub		0.315	0.330	0.360	0.405	0.445	0.458	0.482	0.479	0.423	0.364	0.331	0.309	(40)
(41)			taud		2.458	2.390	2.380	2.255	2.184	2.184	2.126	2.122	2.255	2.425	2.463	2.482	(41)
(42)			Ebn,noon		274	284	285	277	267	263	255	252	261	268	266	268	(42)
(43)			Edh,noon		26	32	35	43	46	46	49	48	40	30	26	24	(43)
(44)	**All-Sky Solar Radiation**		RadAvg		591	830	1182	1566	1749	1968	1874	1767	1441	1072	738	508	(44)
(45)			RadStd		43	110	116	127	170	108	118	107	129	119	79	48	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



MANHATTAN REGIONAL, KS, USA

WMO#: 724555

Lat: 39.135N Long: 96.679W Elev: 1056 StdP: 14.14 Time Zone: -6.00 (NAC) Period: 96-14 WBAN: 03936

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	1.9	7.5	-6.1	4.1	5.6	-1.6	5.2	9.6	24.2	49.5	20.9	43.4	3.7	350	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	22.7	100.2	75.3	97.0	75.3	93.1	74.6	78.5	92.5	77.5	91.5	76.3	89.4	11.4	200	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	74.9	136.1	85.6	73.3	128.7	83.6	72.4	125.0	83.0	42.9	92.7	41.7	91.2	40.4	89.4	83.7	(3)

Extreme Annual Design Conditions

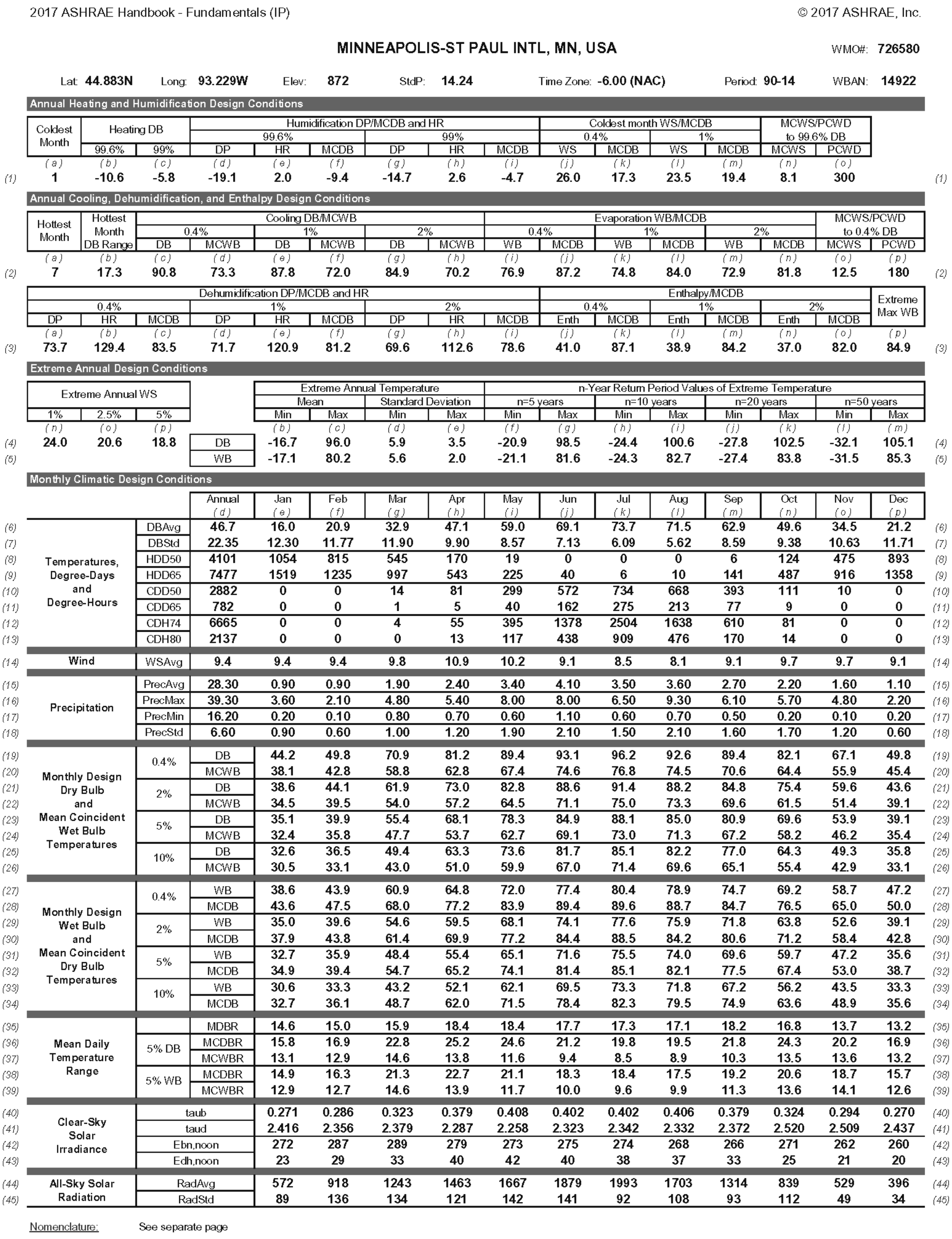
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	24.1	20.5	18.3	DB	-6.6	105.5	6.1	3.3	-11.0	107.9	-14.6	109.9	-18.0	111.7	-22.5	114.1	(4)
(5)				WB	-7.2	81.6	5.9	1.4	-11.5	82.6	-14.9	83.4	-18.3	84.2	-22.6	85.2	(5)

Monthly Climatic Design Conditions

				Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	
				(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	55.0	29.6	33.5	43.8	54.7	65.0	74.5	79.9	77.8	68.3	56.3	43.2	31.7	(6)
(7)			DBStd	19.93	11.04	11.32	11.43	9.71	8.72	6.80	6.48	6.60	8.35	9.46	9.83	10.60	(7)
(8)			HDD50	2291	634	472	259	59	3	0	0	0	1	44	248	571	(8)
(9)			HDD65	5145	1097	883	662	332	113	9	1	2	59	300	655	1032	(9)
(10)			CDD50	4107	2	9	66	200	468	735	928	863	548	239	44	5	(10)
(11)			CDD65	1486	0	0	4	23	113	294	464	400	157	30	1	0	(11)
(12)			CDH74	17206	0	0	56	316	1230	3261	5624	4551	1784	368	16	0	(12)
(13)			CDH80	8207	0	0	6	82	445	1435	3030	2348	756	104	1	0	(13)
(14)	Wind		WSAvg	7.5	6.9	7.4	8.9	9.6	8.3	8.2	6.8	6.0	6.4	7.1	7.2	6.7	(14)
(15)	Precipitation		PrecAvg	33.90	0.80	0.90	2.40	3.00	4.60	5.50	3.40	3.30	4.10	3.00	1.80	1.10	(15)
(16)			PrecMax	50.20	3.10	2.20	7.40	6.00	9.80	12.00	8.10	7.20	9.90	6.50	4.70	3.40	(16)
(17)			PrecMin	15.00	0.00	0.00	0.00	1.10	1.80	1.70	0.70	0.30	0.60	0.10	0.00	0.00	(17)
(18)			PrecStd	8.50	0.80	0.70	1.60	1.50	2.20	3.00	2.30	1.90	2.90	1.90	1.40	1.00	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB	64.3	69.7	80.2	88.0	93.1	98.6	104.7	105.3	99.3	88.5	76.5	65.9		(19)
(20)			MCWB	49.8	54.0	61.5	66.3	71.3	75.2	75.0	74.0	72.5	68.9	59.0	57.8		(20)
(21)		2%	DB	58.0	63.2	74.0	81.7	88.8	94.2	101.0	99.9	91.9	82.2	71.1	58.8		(21)
(22)			MCWB	45.8	50.5	58.4	63.3	70.3	75.0	75.5	75.0	72.2	64.8	57.2	49.3		(22)
(23)		5%	DB	52.4	56.8	69.4	77.4	84.9	91.2	97.6	96.2	88.2	78.0	65.6	54.1		(23)
(24)			MCWB	43.2	47.5	57.0	60.8	68.8	74.5	76.1	74.9	70.9	62.9	54.6	46.1		(24)
(25)		10%	DB	46.7	51.9	63.8	72.8	81.2	88.3	93.4	91.4	83.9	73.1	61.2	48.2		(25)
(26)			MCWB	39.7	43.6	53.2	59.5	67.8	73.3	75.5	74.4	68.8	61.4	51.9	41.9		(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB	53.4	57.9	65.4	68.6	75.5	79.5	80.4	79.7	76.4	72.3	62.4	58.7		(27)
(28)			MCDB	58.5	66.1	76.1	80.8	87.5	92.3	93.5	93.2	90.0	82.3	70.7	63.6		(28)
(29)		2%	WB	47.1	52.5	62.2	66.3	73.3	77.7	79.0	78.0	74.4	69.0	59.4	51.7		(29)
(30)			MCDB	56.3	59.9	69.2	77.1	84.5	90.6	93.3	92.0	88.2	78.3	66.5	55.4		(30)
(31)		5%	WB	43.8	47.5	58.9	63.7	71.2	76.2	77.9	76.7	72.7	66.0	56.4	46.2		(31)
(32)			MCDB	52.5	56.8	66.8	74.7	81.4	88.0	92.3	90.4	85.2	74.2	64.5	52.6		(32)
(33)		10%	WB	40.0	43.6	54.0	60.6	69.2	74.6	76.8	75.4	70.8	62.8	52.5	42.1		(33)
(34)			MCDB	47.4	51.1	64.0	71.4	78.7	85.4	90.5	88.4	81.2	70.6	60.1	48.5		(34)
(35)	Mean Daily Temperature Range		MDBR		22.7	21.9	24.2	24.3	23.2	22.0	22.7	23.4	25.1	24.9	23.9	21.6	(35)
(36)		5% DB	MCDBR		33.0	31.6	32.6	30.6	27.3	25.0	27.1	28.7	28.1	30.4	31.7	30.2	(36)
(37)			MCWBR		22.1	20.6	18.4	16.5	12.7	10.4	8.5	8.9	11.6	15.5	19.0	20.8	(37)
(38)		5% WB	MCDBR		31.1	29.4	25.5	26.1	23.3	22.4	23.5	24.1	24.5	23.1	26.3	27.2	(38)
(39)			MCWBR		21.6	20.4	15.3	16.1	12.4	10.8	9.3	9.9	11.7	13.8	17.6	20.0	(39)
(40)	Clear-Sky Solar Irradiance		taub		0.287	0.302	0.333	0.379	0.410	0.421	0.428	0.427	0.383	0.342	0.301	0.282	(40)
(41)			taud		2.483	2.449	2.409	2.304	2.288	2.291	2.296	2.298	2.374	2.464	2.513	2.501	(41)
(42)			Ebn,noon		284	293	293	284	277	272	269	267	273	275	276	277	(42)
(43)			Edh,noon		25	29	34	40	42	42	41	40	35	29	24	23	(43)
(44)	All-Sky Solar Radiation		RadAvg		743	964	1278	1617	1882	2086	2094	1875	1550	1135	817	617	(44)
(45)			RadStd		62	87	94	91	121	115	141	98	82	119	67	64	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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MINNEAPOLIS-ST PAUL INTL, MN, USA WMO#: **726580**

Lat: **44.883N** Long: **93.229W** Elev: **872** StdP: **14.24** Time Zone: **-6.00 (NAC)** Period: **90-14** WBAN: **14922**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	-10.6	-5.8	-19.1	2.0	-9.4	-14.7	2.6	-4.7	26.0	17.3	23.5	19.4	8.1	300	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	17.3	90.8	73.3	87.8	72.0	84.9	70.2	76.9	87.2	74.8	84.0	72.9	81.8	12.5	180	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	73.7	129.4	83.5	71.7	120.9	81.2	69.6	112.6	78.6	41.0	87.1	38.9	84.2	37.0	82.0	84.9	(3)

Extreme Annual Design Conditions

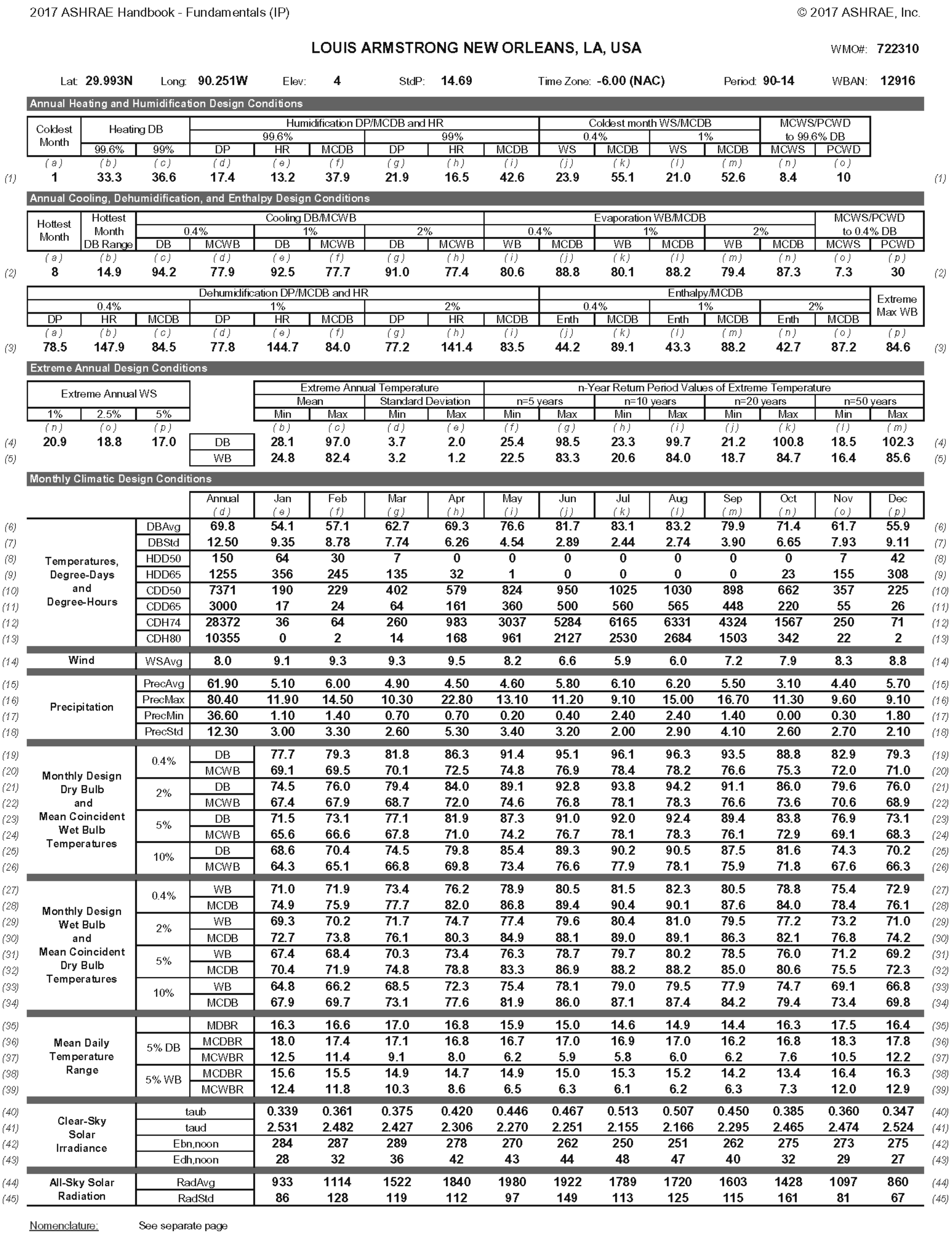
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	24.0	20.6	18.8	DB	-16.7	96.0	5.9	3.5	-20.9	98.5	-24.4	100.6	-27.8	102.5	-32.1	105.1	(4)
(5)				WB	-17.1	80.2	5.6	2.0	-21.1	81.6	-24.3	82.7	-27.4	83.8	-31.5	85.3	(5)

Monthly Climatic Design Conditions

				Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	
				(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	46.7	16.0	20.9	32.9	47.1	59.0	69.1	73.7	71.5	62.9	49.6	34.5	21.2	(6)
(7)			DBStd	22.35	12.30	11.77	11.90	9.90	8.57	7.13	6.09	5.62	8.59	9.38	10.63	11.71	(7)
(8)			HDD50	4101	1054	815	545	170	19	0	0	0	6	124	475	893	(8)
(9)			HDD65	7477	1519	1235	997	543	225	40	6	10	141	487	916	1358	(9)
(10)			CDD50	2882	0	0	14	81	299	572	734	668	393	111	10	0	(10)
(11)			CDD65	782	0	0	1	5	40	162	275	213	77	9	0	0	(11)
(12)			CDH74	6665	0	0	4	55	395	1378	2504	1638	610	81	0	0	(12)
(13)			CDH80	2137	0	0	0	13	117	438	909	476	170	14	0	0	(13)
(14)	Wind		WSAvg	9.4	9.4	9.4	9.8	10.9	10.2	9.1	8.5	8.1	9.1	9.7	9.7	9.1	(14)
(15)	Precipitation		PrecAvg	28.30	0.90	0.90	1.90	2.40	3.40	4.10	3.50	3.60	2.70	2.20	1.60	1.10	(15)
(16)			PrecMax	39.30	3.60	2.10	4.80	5.40	8.00	8.00	6.50	9.30	6.10	5.70	4.80	2.20	(16)
(17)			PrecMin	16.20	0.20	0.10	0.80	0.70	0.60	1.10	0.60	0.70	0.50	0.20	0.10	0.20	(17)
(18)			PrecStd	6.60	0.90	0.60	1.00	1.20	1.90	2.10	1.50	2.10	1.60	1.70	1.20	0.60	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		44.2	49.8	70.9	81.2	89.4	93.1	96.2	92.6	89.4	82.1	67.1	49.8	(19)
(20)			MCWB		38.1	42.8	58.8	62.8	67.4	74.6	76.8	74.5	70.6	64.4	55.9	45.4	(20)
(21)		2%	DB		38.6	44.1	61.9	73.0	82.8	88.6	91.4	88.2	84.8	75.4	59.6	43.6	(21)
(22)			MCWB		34.5	39.5	54.0	57.2	64.5	71.1	75.0	73.3	69.6	61.5	51.4	39.1	(22)
(23)		5%	DB		35.1	39.9	55.4	68.1	78.3	84.9	88.1	85.0	80.9	69.6	53.9	39.1	(23)
(24)			MCWB		32.4	35.8	47.7	53.7	62.7	69.1	73.0	71.3	67.2	58.2	46.2	35.4	(24)
(25)		10%	DB		32.6	36.5	49.4	63.3	73.6	81.7	85.1	82.2	77.0	64.3	49.3	35.8	(25)
(26)			MCWB		30.5	33.1	43.0	51.0	59.9	67.0	71.4	69.6	65.1	55.4	42.9	33.1	(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		38.6	43.9	60.9	64.8	72.0	77.4	80.4	78.9	74.7	69.2	58.7	47.2	(27)
(28)			MCDB		43.6	47.5	68.0	77.2	83.9	89.4	89.6	88.7	84.7	76.5	65.0	50.0	(28)
(29)		2%	WB		35.0	39.6	54.6	59.5	68.1	74.1	77.6	75.9	71.8	63.8	52.6	39.1	(29)
(30)			MCDB		37.9	43.8	61.4	69.9	77.2	84.4	88.5	84.2	80.6	71.2	58.4	42.8	(30)
(31)		5%	WB		32.7	35.9	48.4	55.4	65.1	71.6	75.5	74.0	69.6	59.7	47.2	35.6	(31)
(32)			MCDB		34.9	39.4	54.7	65.2	74.1	81.4	85.1	82.1	77.5	67.4	53.0	38.7	(32)
(33)		10%	WB		30.6	33.3	43.2	52.1	62.1	69.5	73.3	71.8	67.2	56.2	43.5	33.3	(33)
(34)			MCDB		32.7	36.1	48.7	62.0	71.5	78.4	82.3	79.5	74.9	63.6	48.9	35.6	(34)
(35)	Mean Daily Temperature Range		MDBR		14.6	15.0	15.9	18.4	18.4	17.7	17.3	17.1	18.2	16.8	13.7	13.2	(35)
(36)		5% DB	MCDBR		15.8	16.9	22.8	25.2	24.6	21.2	19.8	19.5	21.8	24.3	20.2	16.9	(36)
(37)			MCWBR		13.1	12.9	14.6	13.8	11.6	9.4	8.5	8.9	10.3	13.5	13.6	13.2	(37)
(38)		5% WB	MCDBR		14.9	16.3	21.3	22.7	21.1	18.3	18.4	17.5	19.2	20.6	18.7	15.7	(38)
(39)			MCWBR		12.9	12.7	14.6	13.9	11.7	10.0	9.6	9.9	11.3	13.6	14.1	12.6	(39)
(40)	Clear-Sky Solar Irradiance	taub			0.271	0.286	0.323	0.379	0.408	0.402	0.402	0.406	0.379	0.324	0.294	0.270	(40)
(41)		taud			2.416	2.356	2.379	2.287	2.258	2.323	2.342	2.332	2.372	2.520	2.509	2.437	(41)
(42)		Ebn,noon			272	287	289	279	273	275	274	268	266	271	262	260	(42)
(43)		Edh,noon			23	29	33	40	42	40	38	37	33	25	21	20	(43)
(44)	All-Sky Solar Radiation	RadAvg			572	918	1243	1463	1667	1879	1993	1703	1314	839	529	396	(44)
(45)		RadStd			89	136	134	121	142	141	92	108	93	112	49	34	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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LOUIS ARMSTRONG NEW ORLEANS, LA, USA WMO#: **722310**

Lat: **29.993N** Long: **90.251W** Elev: **4** StdP: **14.69** Time Zone: **-6.00 (NAC)** Period: **90-14** WBAN: **12916**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	**1**	**33.3**	**36.6**	**17.4**	**13.2**	**37.9**	**21.9**	**16.5**	**42.6**	**23.9**	**55.1**	**21.0**	**52.6**	**8.4**	**10**	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	**8**	**14.9**	**94.2**	**77.9**	**92.5**	**77.7**	**91.0**	**77.4**	**80.6**	**88.8**	**80.1**	**88.2**	**79.4**	**87.3**	**7.3**	**30**	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	**78.5**	**147.9**	**84.5**	**77.8**	**144.7**	**84.0**	**77.2**	**141.4**	**83.5**	**44.2**	**89.1**	**43.3**	**88.2**	**42.7**	**87.2**	**84.6**	(3)

Extreme Annual Design Conditions

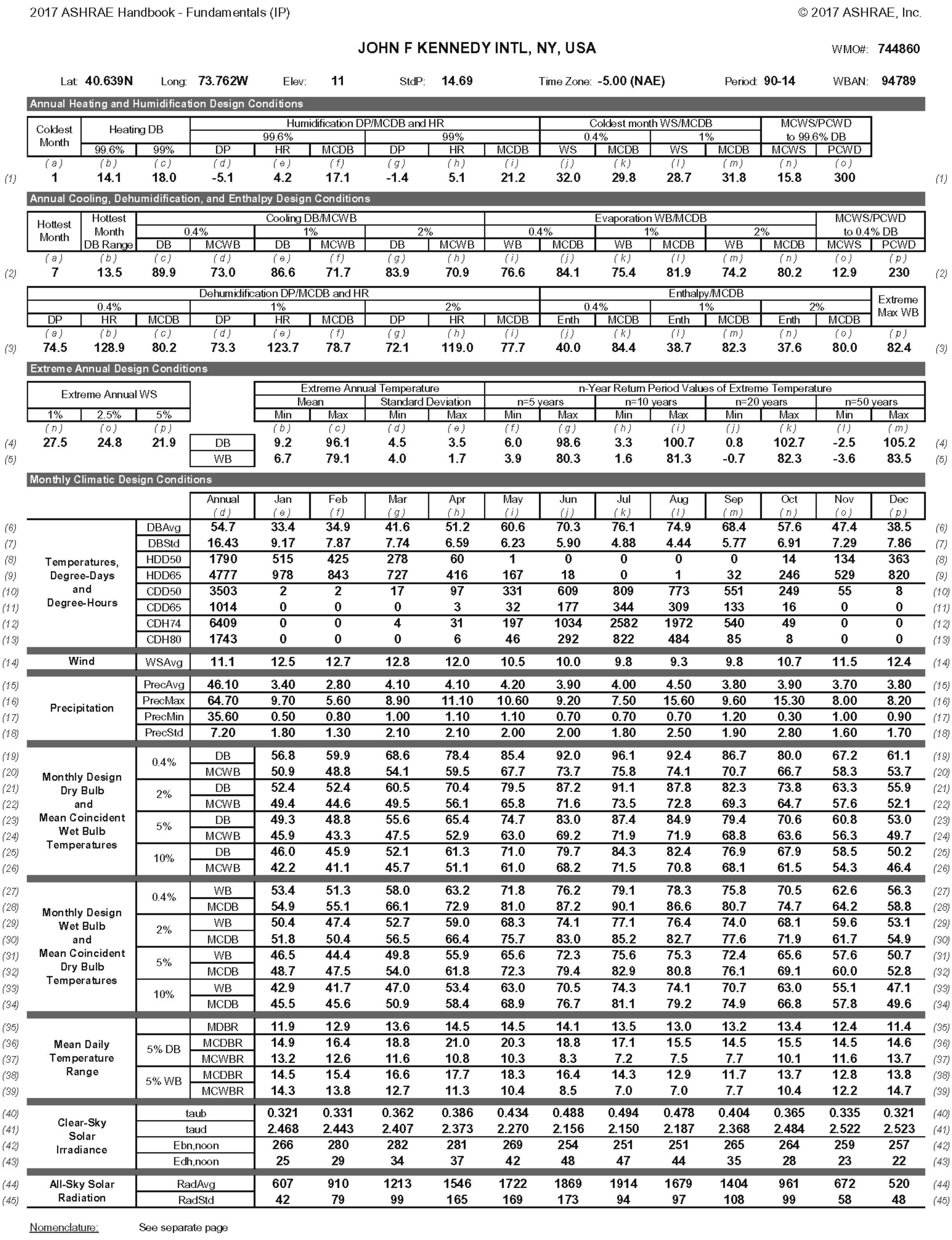
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	**20.9**	**18.8**	**17.0**	DB	**28.1**	**97.0**	**3.7**	**2.0**	**25.4**	**98.5**	**23.3**	**99.7**	**21.2**	**100.8**	**18.5**	**102.3**	(4)
(5)				WB	**24.8**	**82.4**	**3.2**	**1.2**	**22.5**	**83.3**	**20.6**	**84.0**	**18.7**	**84.7**	**16.4**	**85.6**	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**		DBAvg	**69.8**	**54.1**	**57.1**	**62.7**	**69.3**	**76.6**	**81.7**	**83.1**	**83.2**	**79.9**	**71.4**	**61.7**	**55.9**	(6)
(7)			DBStd	**12.50**	**9.35**	**8.78**	**7.74**	**6.26**	**4.54**	**2.89**	**2.44**	**2.74**	**3.90**	**6.65**	**7.93**	**9.11**	(7)
(8)			HDD50	**150**	**64**	**30**	**7**	**0**	**0**	**0**	**0**	**0**	**0**	**0**	**7**	**42**	(8)
(9)			HDD65	**1255**	**356**	**245**	**135**	**32**	**1**	**0**	**0**	**0**	**0**	**23**	**155**	**308**	(9)
(10)			CDD50	**7371**	**190**	**229**	**402**	**579**	**824**	**950**	**1025**	**1030**	**898**	**662**	**357**	**225**	(10)
(11)			CDD65	**3000**	**17**	**24**	**64**	**161**	**360**	**500**	**560**	**565**	**448**	**220**	**55**	**26**	(11)
(12)			CDH74	**28372**	**36**	**64**	**260**	**983**	**3037**	**5284**	**6165**	**6331**	**4324**	**1567**	**250**	**71**	(12)
(13)			CDH80	**10355**	**0**	**2**	**14**	**168**	**961**	**2127**	**2530**	**2684**	**1503**	**342**	**22**	**2**	(13)
(14)	**Wind**		WSAvg	**8.0**	**9.1**	**9.3**	**9.3**	**9.5**	**8.2**	**6.6**	**5.9**	**6.0**	**7.2**	**7.9**	**8.3**	**8.8**	(14)
(15)	**Precipitation**		PrecAvg	**61.90**	**5.10**	**6.00**	**4.90**	**4.50**	**4.60**	**5.80**	**6.10**	**6.20**	**5.50**	**3.10**	**4.40**	**5.70**	(15)
(16)			PrecMax	**80.40**	**11.90**	**14.50**	**10.30**	**22.80**	**13.10**	**11.20**	**9.10**	**15.00**	**16.70**	**11.30**	**9.60**	**9.10**	(16)
(17)			PrecMin	**36.60**	**1.10**	**1.40**	**0.70**	**0.70**	**0.20**	**0.40**	**2.40**	**2.40**	**1.40**	**0.00**	**0.30**	**1.80**	(17)
(18)			PrecStd	**12.30**	**3.00**	**3.30**	**2.60**	**5.30**	**3.40**	**3.20**	**2.00**	**2.90**	**4.10**	**2.60**	**2.70**	**2.10**	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB		**77.7**	**79.3**	**81.8**	**86.3**	**91.4**	**95.1**	**96.1**	**96.3**	**93.5**	**88.8**	**82.9**	**79.3**	(19)
(20)			MCWB		**69.1**	**69.5**	**70.1**	**72.5**	**74.8**	**76.9**	**78.4**	**78.2**	**76.6**	**75.3**	**72.0**	**71.0**	(20)
(21)		2%	DB		**74.5**	**76.0**	**79.4**	**84.0**	**89.1**	**92.8**	**93.8**	**94.2**	**91.1**	**86.0**	**79.6**	**76.0**	(21)
(22)			MCWB		**67.4**	**67.9**	**68.7**	**72.0**	**74.6**	**76.8**	**78.1**	**78.3**	**76.6**	**73.6**	**70.6**	**68.9**	(22)
(23)		5%	DB		**71.5**	**73.1**	**77.1**	**81.9**	**87.3**	**91.0**	**92.0**	**92.4**	**89.4**	**83.8**	**76.9**	**73.1**	(23)
(24)			MCWB		**65.6**	**66.6**	**67.8**	**71.0**	**74.2**	**76.7**	**78.1**	**78.3**	**76.1**	**72.9**	**69.1**	**68.3**	(24)
(25)		10%	DB		**68.6**	**70.4**	**74.5**	**79.8**	**85.4**	**89.3**	**90.2**	**90.5**	**87.5**	**81.6**	**74.3**	**70.2**	(25)
(26)			MCWB		**64.3**	**65.1**	**66.8**	**69.8**	**73.4**	**76.6**	**77.9**	**78.1**	**75.9**	**71.8**	**67.6**	**66.3**	(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB		**71.0**	**71.9**	**73.4**	**76.2**	**78.9**	**80.5**	**81.5**	**82.3**	**80.5**	**78.8**	**75.4**	**72.9**	(27)
(28)			MCDB		**74.9**	**75.9**	**77.7**	**82.0**	**86.8**	**89.4**	**90.4**	**90.1**	**87.6**	**84.0**	**78.4**	**76.1**	(28)
(29)		2%	WB		**69.3**	**70.2**	**71.7**	**74.7**	**77.4**	**79.6**	**80.4**	**81.0**	**79.5**	**77.2**	**73.2**	**71.0**	(29)
(30)			MCDB		**72.7**	**73.8**	**76.1**	**80.3**	**84.9**	**88.1**	**89.0**	**89.1**	**86.3**	**82.1**	**76.8**	**74.2**	(30)
(31)		5%	WB		**67.4**	**68.4**	**70.3**	**73.4**	**76.3**	**78.7**	**79.7**	**80.2**	**78.5**	**76.0**	**71.2**	**69.2**	(31)
(32)			MCDB		**70.4**	**71.9**	**74.8**	**78.8**	**83.3**	**86.9**	**88.2**	**88.2**	**85.0**	**80.6**	**75.5**	**72.3**	(32)
(33)		10%	WB		**64.8**	**66.2**	**68.5**	**72.3**	**75.4**	**78.1**	**79.0**	**79.5**	**77.9**	**74.7**	**69.1**	**66.8**	(33)
(34)			MCDB		**67.9**	**69.7**	**73.1**	**77.6**	**81.9**	**86.0**	**87.1**	**87.4**	**84.2**	**79.4**	**73.4**	**69.8**	(34)
(35)	**Mean Daily Temperature Range**		MDBR		**16.3**	**16.6**	**17.0**	**16.8**	**15.9**	**15.0**	**14.6**	**14.9**	**14.4**	**16.3**	**17.5**	**16.4**	(35)
(36)		5% DB	MCDBR		**18.0**	**17.4**	**17.1**	**16.8**	**16.7**	**17.0**	**16.9**	**17.0**	**16.2**	**16.8**	**18.3**	**17.8**	(36)
(37)			MCWBR		**12.5**	**11.4**	**9.1**	**8.0**	**6.2**	**5.9**	**5.8**	**6.0**	**6.2**	**7.6**	**10.5**	**12.2**	(37)
(38)		5% WB	MCDBR		**15.6**	**15.5**	**14.9**	**14.7**	**14.9**	**15.0**	**15.3**	**15.2**	**14.2**	**13.4**	**16.4**	**16.3**	(38)
(39)			MCWBR		**12.4**	**11.8**	**10.3**	**8.6**	**6.5**	**6.3**	**6.1**	**6.2**	**6.3**	**7.3**	**12.0**	**12.9**	(39)
(40)	**Clear-Sky Solar Irradiance**		taub		**0.339**	**0.361**	**0.375**	**0.420**	**0.446**	**0.467**	**0.513**	**0.507**	**0.450**	**0.385**	**0.360**	**0.347**	(40)
(41)			taud		**2.531**	**2.482**	**2.427**	**2.306**	**2.270**	**2.251**	**2.155**	**2.166**	**2.295**	**2.465**	**2.474**	**2.524**	(41)
(42)			Ebn,noon		**284**	**287**	**289**	**278**	**270**	**262**	**250**	**251**	**262**	**275**	**273**	**275**	(42)
(43)			Edh,noon		**28**	**32**	**36**	**42**	**43**	**44**	**48**	**47**	**40**	**32**	**29**	**27**	(43)
(44)	**All-Sky Solar Radiation**		RadAvg		**933**	**1114**	**1522**	**1840**	**1980**	**1922**	**1789**	**1720**	**1603**	**1428**	**1097**	**860**	(44)
(45)			RadStd		**86**	**128**	**119**	**112**	**97**	**149**	**113**	**125**	**115**	**161**	**81**	**67**	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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JOHN F KENNEDY INTL, NY, USA

WMO#: **744860**

Lat: **40.639N** Long: **73.762W** Elev: **11** StdP: **14.69** Time Zone: **-5.00 (NAE)** Period: **90-14** WBAN: **94789**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	**1**	**14.1**	**18.0**	**-5.1**	**4.2**	**17.1**	**-1.4**	**5.1**	**21.2**	**32.0**	**29.8**	**28.7**	**31.8**	**15.8**	**300**	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	**7**	**13.5**	**89.9**	**73.0**	**86.6**	**71.7**	**83.9**	**70.9**	**76.6**	**84.1**	**75.4**	**81.9**	**74.2**	**80.2**	**12.9**	**230**	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	**74.5**	**128.9**	**80.2**	**73.3**	**123.7**	**78.7**	**72.1**	**119.0**	**77.7**	**40.0**	**84.4**	**38.7**	**82.3**	**37.6**	**80.0**	**82.4**	(3)

Extreme Annual Design Conditions

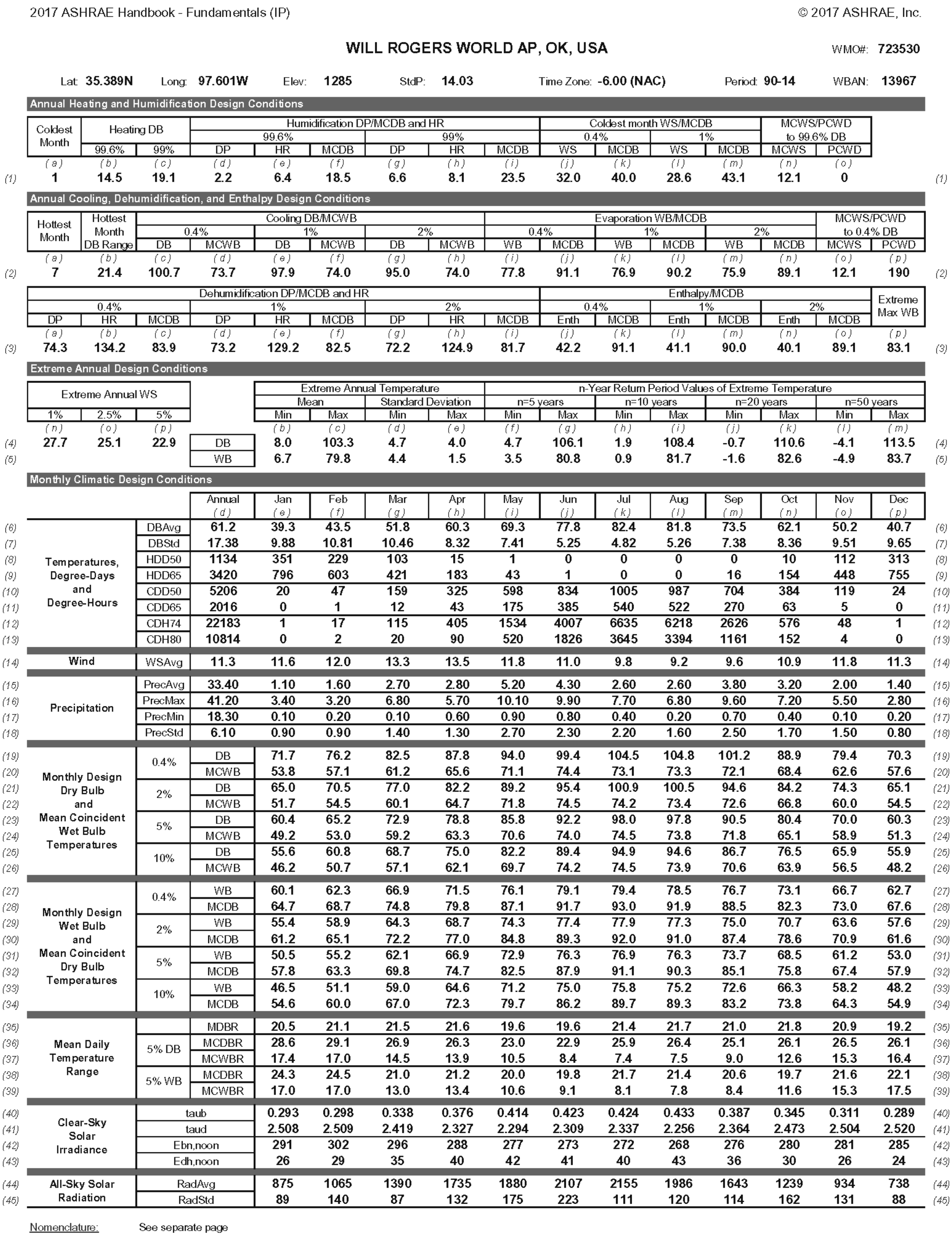
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	**27.5**	**24.8**	**21.9**	DB	**9.2**	**96.1**	**4.5**	**3.5**	**6.0**	**98.6**	**3.3**	**100.7**	**0.8**	**102.7**	**-2.5**	**105.2**	(4)
(5)				WB	**6.7**	**79.1**	**4.0**	**1.7**	**3.9**	**80.3**	**1.6**	**81.3**	**-0.7**	**82.3**	**-3.6**	**83.5**	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**		DBAvg	**54.7**	**33.4**	**34.9**	**41.6**	**51.2**	**60.6**	**70.3**	**76.1**	**74.9**	**68.4**	**57.6**	**47.4**	**38.5**	(6)
(7)			DBStd	**16.43**	**9.17**	**7.87**	**7.74**	**6.59**	**6.23**	**5.90**	**4.88**	**4.44**	**5.77**	**6.91**	**7.29**	**7.86**	(7)
(8)			HDD50	**1790**	**515**	**425**	**278**	**60**	**1**	**0**	**0**	**0**	**0**	**14**	**134**	**363**	(8)
(9)			HDD65	**4777**	**978**	**843**	**727**	**416**	**167**	**18**	**0**	**1**	**32**	**246**	**529**	**820**	(9)
(10)			CDD50	**3503**	**2**	**2**	**17**	**97**	**331**	**609**	**809**	**773**	**551**	**249**	**55**	**8**	(10)
(11)			CDD65	**1014**	**0**	**0**	**0**	**3**	**32**	**177**	**344**	**309**	**133**	**16**	**0**	**0**	(11)
(12)			CDH74	**6409**	**0**	**0**	**4**	**31**	**197**	**1034**	**2582**	**1972**	**540**	**49**	**0**	**0**	(12)
(13)			CDH80	**1743**	**0**	**0**	**0**	**6**	**46**	**292**	**822**	**484**	**85**	**8**	**0**	**0**	(13)
(14)	**Wind**		WSAvg	**11.1**	**12.5**	**12.7**	**12.8**	**12.0**	**10.5**	**10.0**	**9.8**	**9.3**	**9.8**	**10.7**	**11.5**	**12.4**	(14)
(15)	**Precipitation**		PrecAvg	**46.10**	**3.40**	**2.80**	**4.10**	**4.10**	**4.20**	**3.90**	**4.00**	**4.50**	**3.80**	**3.90**	**3.70**	**3.80**	(15)
(16)			PrecMax	**64.70**	**9.70**	**5.60**	**8.90**	**11.10**	**10.60**	**9.20**	**7.50**	**15.60**	**9.60**	**15.30**	**8.00**	**8.20**	(16)
(17)			PrecMin	**35.60**	**0.50**	**0.80**	**1.00**	**1.10**	**1.10**	**0.70**	**0.70**	**0.70**	**1.20**	**0.30**	**1.00**	**0.90**	(17)
(18)			PrecStd	**7.20**	**1.80**	**1.30**	**2.10**	**2.10**	**2.00**	**2.00**	**1.80**	**2.50**	**1.90**	**2.80**	**1.60**	**1.70**	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB		**56.8**	**59.9**	**68.6**	**78.4**	**85.4**	**92.0**	**96.1**	**92.4**	**86.7**	**80.0**	**67.2**	**61.1**	(19)
(20)			MCWB		**50.9**	**48.8**	**54.1**	**59.5**	**67.7**	**73.7**	**75.8**	**74.1**	**70.7**	**66.7**	**58.3**	**53.7**	(20)
(21)		2%	DB		**52.4**	**52.4**	**60.5**	**70.4**	**79.5**	**87.2**	**91.1**	**87.8**	**82.3**	**73.8**	**63.3**	**55.9**	(21)
(22)			MCWB		**49.4**	**44.6**	**49.5**	**56.1**	**65.8**	**71.6**	**73.5**	**72.8**	**69.3**	**64.7**	**57.6**	**52.1**	(22)
(23)		5%	DB		**49.3**	**48.8**	**55.6**	**65.4**	**74.7**	**83.0**	**87.4**	**84.9**	**79.4**	**70.6**	**60.8**	**53.0**	(23)
(24)			MCWB		**45.9**	**43.3**	**47.5**	**52.9**	**63.0**	**69.2**	**71.9**	**71.9**	**68.8**	**63.6**	**56.3**	**49.7**	(24)
(25)		10%	DB		**46.0**	**45.9**	**52.1**	**61.3**	**71.0**	**79.7**	**84.3**	**82.4**	**76.9**	**67.9**	**58.5**	**50.2**	(25)
(26)			MCWB		**42.2**	**41.1**	**45.7**	**51.1**	**61.0**	**68.2**	**71.5**	**70.8**	**68.1**	**61.5**	**54.3**	**46.4**	(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB		**53.4**	**51.3**	**58.0**	**63.2**	**71.8**	**76.2**	**79.1**	**78.3**	**75.8**	**70.5**	**62.6**	**56.3**	(27)
(28)			MCDB		**54.9**	**55.1**	**66.1**	**72.9**	**81.0**	**87.2**	**90.1**	**86.6**	**80.7**	**74.7**	**64.2**	**58.8**	(28)
(29)		2%	WB		**50.4**	**47.4**	**52.7**	**59.0**	**68.3**	**74.1**	**77.1**	**76.4**	**74.0**	**68.1**	**59.6**	**53.1**	(29)
(30)			MCDB		**51.8**	**50.4**	**56.5**	**66.4**	**75.7**	**83.0**	**85.2**	**82.7**	**77.6**	**71.9**	**61.7**	**54.9**	(30)
(31)		5%	WB		**46.5**	**44.4**	**49.8**	**55.9**	**65.6**	**72.3**	**75.6**	**75.3**	**72.4**	**65.6**	**57.6**	**50.7**	(31)
(32)			MCDB		**48.7**	**47.5**	**54.0**	**61.8**	**72.3**	**79.4**	**82.9**	**80.8**	**76.1**	**69.1**	**60.0**	**52.8**	(32)
(33)		10%	WB		**42.9**	**41.7**	**47.0**	**53.4**	**63.0**	**70.5**	**74.3**	**74.1**	**70.7**	**63.0**	**55.1**	**47.1**	(33)
(34)			MCDB		**45.5**	**45.6**	**50.9**	**58.4**	**68.9**	**76.7**	**81.1**	**79.2**	**74.9**	**66.8**	**57.8**	**49.6**	(34)
(35)	**Mean Daily Temperature Range**		MDBR		**11.9**	**12.9**	**13.6**	**14.5**	**14.5**	**14.1**	**13.5**	**13.0**	**13.2**	**13.4**	**12.4**	**11.4**	(35)
(36)		5% DB	MCDBR		**14.9**	**16.4**	**18.8**	**21.0**	**20.3**	**18.8**	**17.1**	**15.5**	**14.5**	**15.5**	**14.5**	**14.6**	(36)
(37)			MCWBR		**13.2**	**12.6**	**11.6**	**10.8**	**10.3**	**8.3**	**7.2**	**7.5**	**7.7**	**10.1**	**11.6**	**13.7**	(37)
(38)		5% WB	MCDBR		**14.5**	**15.4**	**16.6**	**17.7**	**18.3**	**16.4**	**14.3**	**12.9**	**11.7**	**13.7**	**12.8**	**13.8**	(38)
(39)			MCWBR		**14.3**	**13.8**	**12.7**	**11.3**	**10.4**	**8.5**	**7.0**	**7.0**	**7.7**	**10.4**	**12.2**	**14.7**	(39)
(40)	**Clear-Sky Solar Irradiance**		taub		**0.321**	**0.331**	**0.362**	**0.386**	**0.434**	**0.488**	**0.494**	**0.478**	**0.404**	**0.365**	**0.335**	**0.321**	(40)
(41)			taud		**2.468**	**2.443**	**2.407**	**2.373**	**2.270**	**2.156**	**2.150**	**2.187**	**2.368**	**2.484**	**2.522**	**2.523**	(41)
(42)			Ebn,noon		**266**	**280**	**282**	**281**	**269**	**254**	**251**	**251**	**265**	**264**	**259**	**257**	(42)
(43)			Edh,noon		**25**	**29**	**34**	**37**	**42**	**48**	**47**	**44**	**35**	**28**	**23**	**22**	(43)
(44)	**All-Sky Solar Radiation**		RadAvg		**607**	**910**	**1213**	**1546**	**1722**	**1869**	**1914**	**1679**	**1404**	**961**	**672**	**520**	(44)
(45)			RadStd		**42**	**79**	**99**	**165**	**169**	**173**	**94**	**97**	**108**	**99**	**58**	**48**	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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WILL ROGERS WORLD AP, OK, USA WMO#: **723530**

Lat: **35.389N** Long: **97.601W** Elev: **1285** StdP: **14.03** Time Zone: **-6.00 (NAC)** Period: **90-14** WBAN: **13967**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	14.5	19.1	2.2	6.4	18.5	6.6	8.1	23.5	32.0	40.0	28.6	43.1	12.1	0	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	21.4	100.7	73.7	97.9	74.0	95.0	74.0	77.8	91.1	76.9	90.2	75.9	89.1	12.1	190	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	74.3	134.2	83.9	73.2	129.2	82.5	72.2	124.9	81.7	42.2	91.1	41.1	90.0	40.1	89.1	83.1	(3)

Extreme Annual Design Conditions

	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	27.7	25.1	22.9	DB	8.0	103.3	4.7	4.0	4.7	106.1	1.9	108.4	-0.7	110.6	-4.1	113.5	(4)
(5)				WB	6.7	79.8	4.4	1.5	3.5	80.8	0.9	81.7	-1.6	82.6	-4.9	83.7	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	61.2	39.3	43.5	51.8	60.3	69.3	77.8	82.4	81.8	73.5	62.1	50.2	40.7	(6)
(7)			DBStd	17.38	9.88	10.81	10.46	8.32	7.41	5.25	4.82	5.26	7.38	8.36	9.51	9.65	(7)
(8)			HDD50	1134	351	229	103	15	1	0	0	0	0	10	112	313	(8)
(9)			HDD65	3420	796	603	421	183	43	1	0	0	16	154	448	755	(9)
(10)			CDD50	5206	20	47	159	325	598	834	1005	987	704	384	119	24	(10)
(11)			CDD65	2016	0	1	12	43	175	385	540	522	270	63	5	0	(11)
(12)			CDH74	22183	1	17	115	405	1534	4007	6635	6218	2626	576	48	1	(12)
(13)			CDH80	10814	0	2	20	90	520	1826	3645	3394	1161	152	4	0	(13)
(14)	Wind		WSAvg	11.3	11.6	12.0	13.3	13.5	11.8	11.0	9.8	9.2	9.6	10.9	11.8	11.3	(14)
(15)	Precipitation		PrecAvg	33.40	1.10	1.60	2.70	2.80	5.20	4.30	2.60	2.60	3.80	3.20	2.00	1.40	(15)
(16)			PrecMax	41.20	3.40	3.20	6.80	5.70	10.10	9.90	7.70	6.80	9.60	7.20	5.50	2.80	(16)
(17)			PrecMin	18.30	0.10	0.20	0.10	0.60	0.90	0.80	0.40	0.20	0.70	0.40	0.10	0.20	(17)
(18)			PrecStd	6.10	0.90	0.90	1.40	1.30	2.70	2.30	2.20	1.60	2.50	1.70	1.50	0.80	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		71.7	76.2	82.5	87.8	94.0	99.4	104.5	104.8	101.2	88.9	79.4	70.3	(19)
(20)			MCWB		53.8	57.1	61.2	65.6	71.1	74.4	73.1	73.3	72.1	68.4	62.6	57.6	(20)
(21)		2%	DB		65.0	70.5	77.0	82.2	89.2	95.4	100.9	100.5	94.6	84.2	74.3	65.1	(21)
(22)			MCWB		51.7	54.5	60.1	64.7	71.8	74.5	74.2	73.4	72.6	66.8	60.0	54.5	(22)
(23)		5%	DB		60.4	65.2	72.9	78.8	85.8	92.2	98.0	97.8	90.5	80.4	70.0	60.3	(23)
(24)			MCWB		49.2	53.0	59.2	63.3	70.6	74.0	74.5	73.8	71.8	65.1	58.9	51.3	(24)
(25)		10%	DB		55.6	60.8	68.7	75.0	82.2	89.4	94.9	94.6	86.7	76.5	65.9	55.9	(25)
(26)			MCWB		46.2	50.7	57.1	62.1	69.7	74.2	74.5	73.9	70.6	63.9	56.5	48.2	(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		60.1	62.3	66.9	71.5	76.1	79.1	79.4	78.5	76.7	73.1	66.7	62.7	(27)
(28)			MCDB		64.7	68.7	74.8	79.8	87.1	91.7	93.0	91.9	88.5	82.3	73.0	67.6	(28)
(29)		2%	WB		55.4	58.9	64.3	68.7	74.3	77.4	77.9	77.3	75.0	70.7	63.6	57.6	(29)
(30)			MCDB		61.2	65.1	72.2	77.0	84.8	89.3	92.0	91.0	87.4	78.6	70.9	61.6	(30)
(31)		5%	WB		50.5	55.2	62.1	66.9	72.9	76.3	76.9	76.3	73.7	68.5	61.2	53.0	(31)
(32)			MCDB		57.8	63.3	69.8	74.7	82.5	87.9	91.1	90.3	85.1	75.8	67.4	57.9	(32)
(33)		10%	WB		46.5	51.1	59.0	64.6	71.2	75.0	75.8	75.2	72.6	66.3	58.2	48.2	(33)
(34)			MCDB		54.6	60.0	67.0	72.3	79.7	86.2	89.7	89.3	83.2	73.8	64.3	54.9	(34)
(35)	Mean Daily Temperature Range		MDBR		20.5	21.1	21.5	21.6	19.6	19.6	21.4	21.7	21.0	21.8	20.9	19.2	(35)
(36)		5% DB	MCDBR		28.6	29.1	26.9	26.3	23.0	22.9	25.9	26.4	25.1	26.1	26.5	26.1	(36)
(37)			MCWBR		17.4	17.0	14.5	13.9	10.5	8.4	7.4	7.5	9.0	12.6	15.3	16.4	(37)
(38)		5% WB	MCDBR		24.3	24.5	21.0	21.2	20.0	19.8	21.7	21.4	20.6	19.7	21.6	22.1	(38)
(39)			MCWBR		17.0	17.0	13.0	13.4	10.6	9.1	8.1	7.8	8.4	11.6	15.3	17.5	(39)
(40)	Clear-Sky Solar Irradiance	taub			0.293	0.298	0.338	0.376	0.414	0.423	0.424	0.433	0.387	0.345	0.311	0.289	(40)
(41)		taud			2.508	2.509	2.419	2.327	2.294	2.309	2.337	2.256	2.364	2.473	2.504	2.520	(41)
(42)		Ebn,noon			291	302	296	288	277	273	272	268	276	280	281	285	(42)
(43)		Edh,noon			26	29	35	40	42	41	40	43	36	30	26	24	(43)
(44)	All-Sky Solar Radiation	RadAvg			875	1065	1390	1735	1880	2107	2155	1986	1643	1239	934	738	(44)
(45)		RadStd			89	140	87	132	175	223	111	120	114	162	131	88	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals

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PURDUE UNIVERSITY AP, IN, USA

WMO#: **724386**

Lat: **40.412N** Long: **86.937W** Elev: **599** StdP: **14.38** Time Zone: **-5.00 (NAE)** Period: **90-14** WBAN: **14835**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	**1**	**0.6**	**6.3**	**-7.8**	**3.7**	**1.9**	**-2.0**	**5.0**	**7.4**	**26.7**	**26.9**	**24.1**	**27.1**	**8.0**	**270**	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	**7**	**19.5**	**91.6**	**75.6**	**89.5**	**74.3**	**86.9**	**72.8**	**78.5**	**87.9**	**76.9**	**85.7**	**75.3**	**83.6**	**9.4**	**230**	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	**75.7**	**137.7**	**84.5**	**74.1**	**130.2**	**82.3**	**72.6**	**123.7**	**80.4**	**42.5**	**88.0**	**40.6**	**85.7**	**39.1**	**83.9**	**88.5**	(3)

Extreme Annual Design Conditions

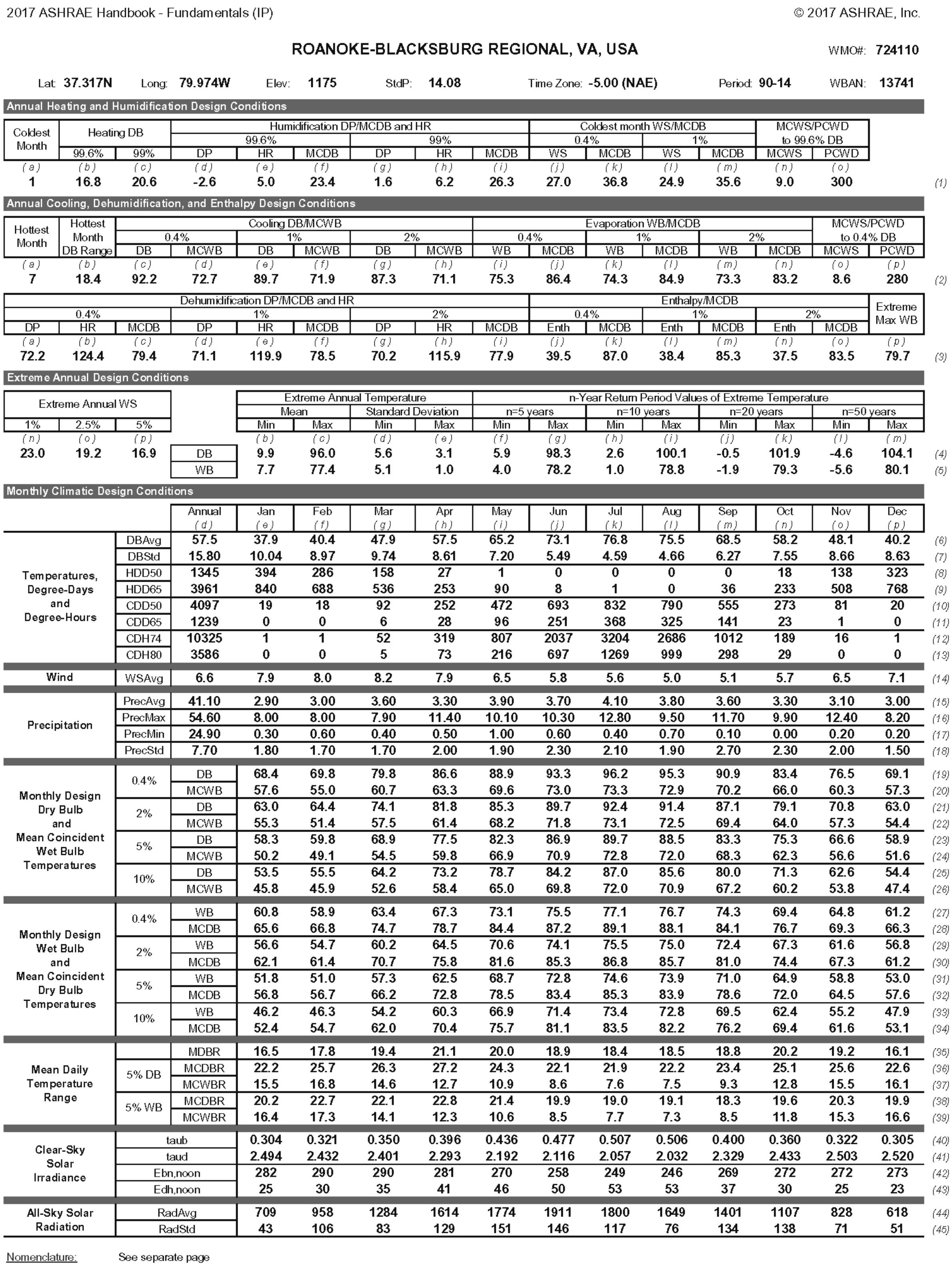
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	**22.7**	**19.8**	**18.0**	DB	**-6.4**	**96.0**	**7.5**	**3.3**	**-11.8**	**98.4**	**-16.2**	**100.3**	**-20.3**	**102.2**	**-25.8**	**104.6**	(4)
(5)				WB	**-7.1**	**81.8**	**7.2**	**2.7**	**-12.4**	**83.7**	**-16.6**	**85.3**	**-20.7**	**86.9**	**-25.9**	**88.8**	(5)

Monthly Climatic Design Conditions

				Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	
				(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**		DBAvg	**52.6**	**27.4**	**31.1**	**41.3**	**52.7**	**62.6**	**71.7**	**74.5**	**73.2**	**66.1**	**54.7**	**42.7**	**31.8**	(6)
(7)			DBStd	**18.96**	**12.09**	**11.18**	**11.28**	**9.49**	**8.46**	**6.51**	**5.59**	**5.36**	**7.62**	**8.87**	**10.09**	**10.62**	(7)
(8)			HDD50	**2528**	**704**	**533**	**322**	**78**	**6**	**0**	**0**	**0**	**1**	**51**	**264**	**569**	(8)
(9)			HDD65	**5526**	**1164**	**949**	**739**	**383**	**151**	**19**	**2**	**4**	**78**	**339**	**670**	**1028**	(9)
(10)			CDD50	**3474**	**4**	**4**	**52**	**159**	**396**	**650**	**760**	**719**	**485**	**196**	**43**	**6**	(10)
(11)			CDD65	**998**	**0**	**0**	**4**	**14**	**76**	**219**	**297**	**258**	**111**	**19**	**0**	**0**	(11)
(12)			CDH74	**9160**	**0**	**0**	**25**	**128**	**715**	**2018**	**2785**	**2259**	**1040**	**188**	**2**	**0**	(12)
(13)			CDH80	**3185**	**0**	**0**	**3**	**18**	**204**	**723**	**1060**	**791**	**347**	**39**	**0**	**0**	(13)
(14)	**Wind**		WSAvg	**8.0**	**9.6**	**9.5**	**9.7**	**9.9**	**8.0**	**6.6**	**6.0**	**5.6**	**6.2**	**7.6**	**8.8**	**8.8**	(14)
(15)	**Precipitation**		PrecAvg	**35.60**	**1.70**	**1.50**	**2.60**	**3.70**	**3.60**	**4.10**	**3.40**	**4.10**	**3.00**	**2.40**	**2.80**	**2.90**	(15)
(16)			PrecMax	**42.90**	**4.40**	**3.90**	**5.70**	**8.80**	**6.50**	**8.20**	**7.10**	**12.20**	**8.80**	**5.00**	**9.00**	**7.70**	(16)
(17)			PrecMin	**29.60**	**0.10**	**0.30**	**0.40**	**0.60**	**1.00**	**0.30**	**0.60**	**0.90**	**0.00**	**0.20**	**0.30**	**0.30**	(17)
(18)			PrecStd	**4.20**	**1.30**	**0.90**	**1.30**	**1.90**	**1.50**	**2.10**	**1.80**	**2.80**	**2.20**	**1.20**	**1.80**	**1.70**	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB		**61.6**	**63.6**	**77.7**	**82.3**	**89.0**	**93.4**	**96.9**	**93.4**	**92.4**	**84.5**	**72.1**	**63.3**	(19)
(20)			MCWB		**57.2**	**55.1**	**62.6**	**65.8**	**70.6**	**75.7**	**78.9**	**76.1**	**72.6**	**67.2**	**58.4**	**58.5**	(20)
(21)		2%	DB		**55.4**	**57.2**	**71.5**	**78.1**	**85.2**	**90.2**	**91.8**	**90.5**	**87.8**	**79.3**	**66.4**	**56.9**	(21)
(22)			MCWB		**52.0**	**51.0**	**59.6**	**63.3**	**68.6**	**73.7**	**76.8**	**76.2**	**70.5**	**64.2**	**58.1**	**53.1**	(22)
(23)		5%	DB		**49.9**	**52.0**	**66.0**	**73.3**	**81.9**	**87.5**	**89.0**	**87.8**	**83.9**	**74.6**	**62.9**	**52.0**	(23)
(24)			MCWB		**46.2**	**46.1**	**56.6**	**60.6**	**67.7**	**72.5**	**75.5**	**74.5**	**68.9**	**62.6**	**56.5**	**48.0**	(24)
(25)		10%	DB		**43.3**	**46.0**	**60.4**	**69.5**	**78.0**	**84.3**	**85.9**	**84.3**	**80.6**	**70.1**	**58.7**	**46.1**	(25)
(26)			MCWB		**39.6**	**41.3**	**52.4**	**58.6**	**65.8**	**70.9**	**73.6**	**72.5**	**67.2**	**60.4**	**52.4**	**42.8**	(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB		**58.0**	**58.0**	**64.5**	**69.1**	**74.8**	**78.4**	**81.8**	**80.2**	**76.1**	**70.9**	**63.5**	**59.0**	(27)
(28)			MCDB		**61.0**	**62.1**	**75.0**	**79.2**	**84.4**	**89.3**	**91.6**	**89.6**	**87.3**	**80.2**	**67.5**	**62.9**	(28)
(29)		2%	WB		**53.5**	**51.8**	**61.2**	**65.5**	**72.0**	**76.6**	**79.4**	**78.1**	**73.7**	**67.4**	**60.1**	**54.0**	(29)
(30)			MCDB		**55.4**	**55.7**	**69.5**	**74.4**	**81.4**	**86.4**	**88.8**	**87.5**	**82.5**	**74.5**	**64.7**	**56.4**	(30)
(31)		5%	WB		**46.3**	**45.9**	**58.0**	**62.7**	**69.8**	**74.9**	**77.4**	**76.4**	**71.5**	**64.4**	**57.2**	**48.1**	(31)
(32)			MCDB		**49.0**	**50.8**	**65.2**	**70.8**	**78.0**	**84.1**	**85.6**	**84.7**	**78.5**	**71.7**	**62.0**	**50.8**	(32)
(33)		10%	WB		**39.8**	**41.7**	**53.0**	**59.9**	**67.7**	**73.0**	**75.5**	**74.6**	**69.5**	**61.7**	**52.9**	**43.2**	(33)
(34)			MCDB		**43.0**	**46.1**	**59.6**	**67.9**	**75.5**	**81.3**	**83.7**	**81.6**	**77.0**	**68.9**	**57.9**	**46.4**	(34)
(35)	**Mean Daily Temperature Range**		MDBR		**14.4**	**15.7**	**18.1**	**20.6**	**20.6**	**20.1**	**19.5**	**19.8**	**22.1**	**20.7**	**16.5**	**13.6**	(35)
(36)		5% DB	MCDBR		**18.8**	**23.1**	**24.6**	**24.5**	**24.1**	**23.2**	**21.8**	**22.0**	**25.8**	**26.1**	**21.2**	**18.5**	(36)
(37)			MCWBR		**15.7**	**17.9**	**15.1**	**14.2**	**11.9**	**10.2**	**9.6**	**9.9**	**11.2**	**13.6**	**15.2**	**15.5**	(37)
(38)		5% WB	MCDBR		**17.9**	**21.0**	**21.3**	**21.3**	**20.4**	**19.8**	**19.2**	**19.1**	**20.6**	**21.5**	**19.1**	**17.2**	(38)
(39)			MCWBR		**16.4**	**17.6**	**14.1**	**14.0**	**11.4**	**10.0**	**10.0**	**9.8**	**10.8**	**12.9**	**15.6**	**15.5**	(39)
(40)	**Clear-Sky Solar Irradiance**		taub		**0.299**	**0.326**	**0.350**	**0.404**	**0.434**	**0.441**	**0.459**	**0.458**	**0.422**	**0.364**	**0.321**	**0.299**	(40)
(41)			taud		**2.427**	**2.339**	**2.359**	**2.218**	**2.188**	**2.208**	**2.157**	**2.147**	**2.227**	**2.384**	**2.454**	**2.451**	(41)
(42)			Ebn,noon		**274**	**281**	**286**	**275**	**269**	**266**	**260**	**257**	**258**	**263**	**263**	**265**	(42)
(43)			Edh,noon		**25**	**32**	**35**	**43**	**46**	**45**	**47**	**46**	**40**	**30**	**25**	**23**	(43)
(44)	**All-Sky Solar Radiation**		RadAvg		**533**	**820**	**1166**	**1512**	**1702**	**1970**	**1990**	**1760**	**1444**	**990**	**642**	**448**	(44)
(45)			RadStd		**62**	**83**	**95**	**132**	**175**	**115**	**111**	**110**	**110**	**95**	**57**	**45**	(45)

<u>Nomenclature:</u> See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



2017 ASHRAE Handbook - Fundamentals (IP)

ROANOKE-BLACKSBURG REGIONAL, VA, USA

WMO#: **724110**

Lat: **37.317N** Long: **79.974W** Elev: **1175** StdP: **14.08** Time Zone: **-5.00 (NAE)** Period: **90-14** WBAN: **13741**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	16.8	20.6	-2.6	5.0	23.4	1.6	6.2	26.3	27.0	36.8	24.9	35.6	9.0	300	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	18.4	92.2	72.7	89.7	71.9	87.3	71.1	75.3	86.4	74.3	84.9	73.3	83.2	8.6	280	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	72.2	124.4	79.4	71.1	119.9	78.5	70.2	115.9	77.9	39.5	87.0	38.4	85.3	37.5	83.5	79.7	(3)

Extreme Annual Design Conditions

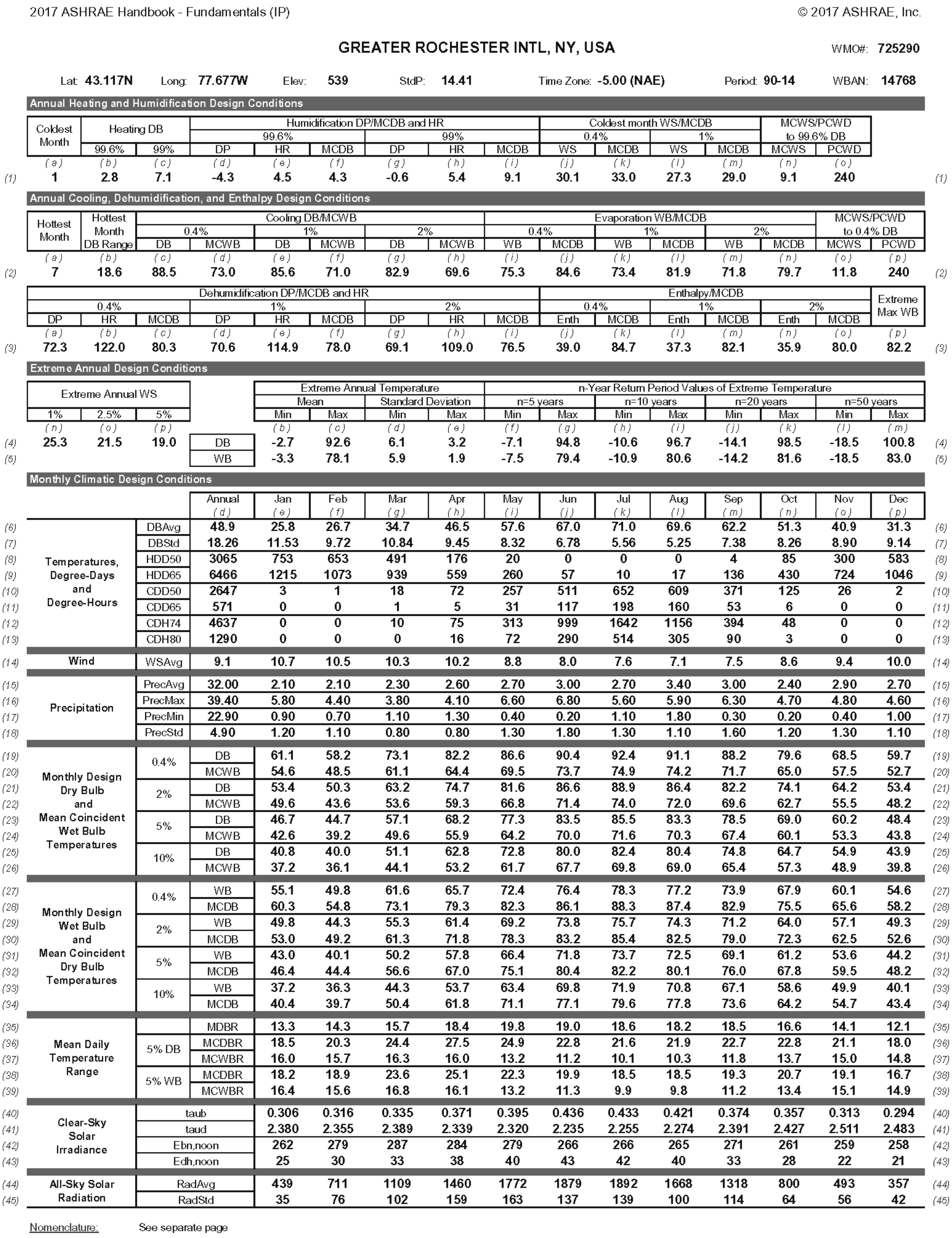
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	23.0	19.2	16.9	DB	9.9	96.0	5.6	3.1	5.9	98.3	2.6	100.1	-0.5	101.9	-4.6	104.1	(4)
(5)				WB	7.7	77.4	5.1	1.0	4.0	78.2	1.0	78.8	-1.9	79.3	-5.6	80.1	(5)

Monthly Climatic Design Conditions

				Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	
				(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	57.5	37.9	40.4	47.9	57.5	65.2	73.1	76.8	75.5	68.5	58.2	48.1	40.2	(6)
(7)			DBStd	15.80	10.04	8.97	9.74	8.61	7.20	5.49	4.59	4.66	6.27	7.55	8.66	8.63	(7)
(8)			HDD50	1345	394	286	158	27	1	0	0	0	0	18	138	323	(8)
(9)			HDD65	3961	840	688	536	253	90	8	1	0	36	233	508	768	(9)
(10)			CDD50	4097	19	18	92	252	472	693	832	790	555	273	81	20	(10)
(11)			CDD65	1239	0	0	6	28	96	251	368	325	141	23	1	0	(11)
(12)			CDH74	10325	1	1	52	319	807	2037	3204	2686	1012	189	16	1	(12)
(13)			CDH80	3586	0	0	5	73	216	697	1269	999	298	29	0	0	(13)
(14)	Wind		WSAvg	6.6	7.9	8.0	8.2	7.9	6.5	5.8	5.6	5.0	5.1	5.7	6.5	7.1	(14)
(15)	Precipitation		PrecAvg	41.10	2.90	3.00	3.60	3.30	3.90	3.70	4.10	3.80	3.60	3.30	3.10	3.00	(15)
(16)			PrecMax	54.60	8.00	8.00	7.90	11.40	10.10	10.30	12.80	9.50	11.70	9.90	12.40	8.20	(16)
(17)			PrecMin	24.90	0.30	0.60	0.40	0.50	1.00	0.60	0.40	0.70	0.10	0.00	0.20	0.20	(17)
(18)			PrecStd	7.70	1.80	1.70	1.70	2.00	1.90	2.30	2.10	1.90	2.70	2.30	2.00	1.50	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB	68.4	69.8	79.8	86.6	88.9	93.3	96.2	95.3	90.9	83.4	76.5	69.1		(19)
(20)			MCWB	57.6	55.0	60.7	63.3	69.6	73.0	73.3	72.9	70.2	66.0	60.3	57.3		(20)
(21)		2%	DB	63.0	64.4	74.1	81.8	85.3	89.7	92.4	91.4	87.1	79.1	70.8	63.0		(21)
(22)			MCWB	55.3	51.4	57.5	61.4	68.2	71.8	73.1	72.5	69.4	64.0	57.3	54.4		(22)
(23)		5%	DB	58.3	59.8	68.9	77.5	82.3	86.9	89.7	88.5	83.3	75.3	66.6	58.9		(23)
(24)			MCWB	50.2	49.1	54.5	59.8	66.9	70.9	72.8	72.0	68.3	62.3	56.6	51.6		(24)
(25)		10%	DB	53.5	55.5	64.2	73.2	78.7	84.2	87.0	85.6	80.0	71.3	62.6	54.4		(25)
(26)			MCWB	45.8	45.9	52.6	58.4	65.0	69.8	72.0	70.9	67.2	60.2	53.8	47.4		(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB	60.8	58.9	63.4	67.3	73.1	75.5	77.1	76.7	74.3	69.4	64.8	61.2		(27)
(28)			MCDB	65.6	66.8	74.7	78.7	84.4	87.2	89.1	88.1	84.1	76.7	69.3	66.3		(28)
(29)		2%	WB	56.6	54.7	60.2	64.5	70.6	74.1	75.5	75.0	72.4	67.3	61.6	56.8		(29)
(30)			MCDB	62.1	61.4	70.7	75.8	81.6	85.3	86.8	85.7	81.0	74.4	67.3	61.2		(30)
(31)		5%	WB	51.8	51.0	57.3	62.5	68.7	72.8	74.6	73.9	71.0	64.9	58.8	53.0		(31)
(32)			MCDB	56.8	56.7	66.2	72.8	78.5	83.4	85.3	83.9	78.6	72.0	64.5	57.6		(32)
(33)		10%	WB	46.2	46.3	54.2	60.3	66.9	71.4	73.4	72.8	69.5	62.4	55.2	47.9		(33)
(34)			MCDB	52.4	54.7	62.0	70.4	75.7	81.1	83.5	82.2	76.2	69.4	61.6	53.1		(34)
(35)	Mean Daily Temperature Range		MDBR	16.5	17.8	19.4	21.1	20.0	18.9	18.4	18.5	18.8	20.2	19.2	16.1		(35)
(36)		5% DB	MCDBR	22.2	25.7	26.3	27.2	24.3	22.1	21.9	22.2	23.4	25.1	25.6	22.6		(36)
(37)			MCWBR	15.5	16.8	14.6	12.7	10.9	8.6	7.6	7.5	9.3	12.8	15.5	16.1		(37)
(38)		5% WB	MCDBR	20.2	22.7	22.1	22.8	21.4	19.9	19.0	19.1	18.3	19.6	20.3	19.9		(38)
(39)			MCWBR	16.4	17.3	14.1	12.3	10.6	8.5	7.7	7.3	8.5	11.8	15.3	16.6		(39)
(40)	Clear-Sky Solar Irradiance		taub		0.304	0.321	0.350	0.396	0.436	0.477	0.507	0.506	0.400	0.360	0.322	0.305	(40)
(41)			taud		2.494	2.432	2.401	2.293	2.192	2.116	2.057	2.032	2.329	2.433	2.503	2.520	(41)
(42)			Ebn,noon		282	290	290	281	270	258	249	246	269	272	272	273	(42)
(43)			Edh,noon		25	30	35	41	46	50	53	53	37	30	25	23	(43)
(44)	All-Sky Solar Radiation		RadAvg		709	958	1284	1614	1774	1911	1800	1649	1401	1107	828	618	(44)
(45)			RadStd		43	106	83	129	151	146	117	76	134	138	71	51	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



2017 ASHRAE Handbook - Fundamentals (IP)

GREATER ROCHESTER INTL, NY, USA

WMO#: **725290**

Lat: **43.117N** Long: **77.677W** Elev: **539** StdP: **14.41** Time Zone: **-5.00 (NAE)** Period: **90-14** WBAN: **14768**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	2.8	7.1	-4.3	4.5	4.3	-0.6	5.4	9.1	30.1	33.0	27.3	29.0	9.1	240	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	18.6	88.5	73.0	85.6	71.0	82.9	69.6	75.3	84.6	73.4	81.9	71.8	79.7	11.8	240	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	72.3	122.0	80.3	70.6	114.9	78.0	69.1	109.0	76.5	39.0	84.7	37.3	82.1	35.9	80.0	82.2	(3)

Extreme Annual Design Conditions

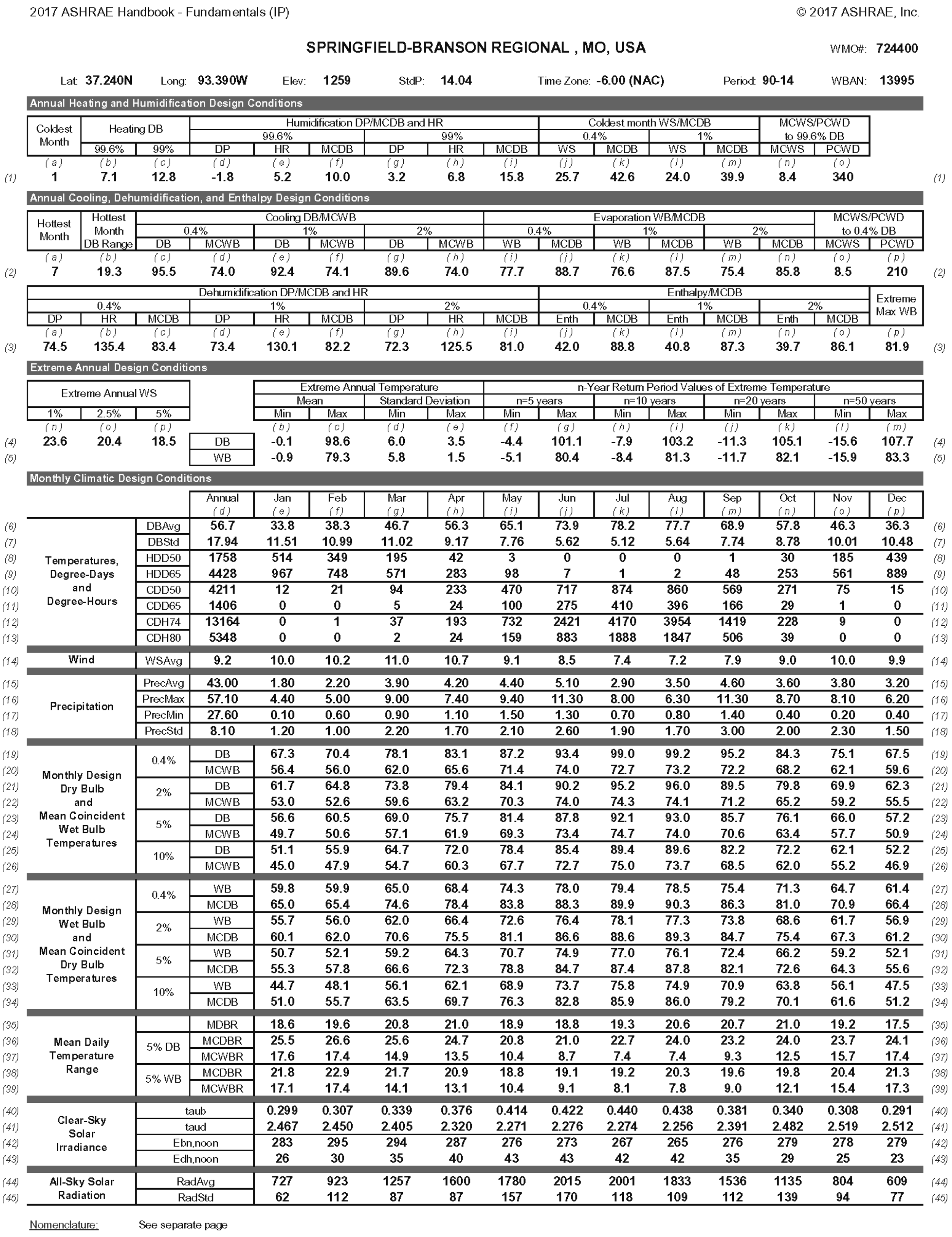
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	25.3	21.5	19.0	DB	-2.7	92.6	6.1	3.2	-7.1	94.8	-10.6	96.7	-14.1	98.5	-18.5	100.8	(4)
(5)				WB	-3.3	78.1	5.9	1.9	-7.5	79.4	-10.9	80.6	-14.2	81.6	-18.5	83.0	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	48.9	25.8	26.7	34.7	46.5	57.6	67.0	71.0	69.6	62.2	51.3	40.9	31.3	(6)
(7)			DBStd	18.26	11.53	9.72	10.84	9.45	8.32	6.78	5.56	5.25	7.38	8.26	8.90	9.14	(7)
(8)			HDD50	3065	753	653	491	176	20	0	0	0	4	85	300	583	(8)
(9)			HDD65	6466	1215	1073	939	559	260	57	10	17	136	430	724	1046	(9)
(10)			CDD50	2647	3	1	18	72	257	511	652	609	371	125	26	2	(10)
(11)			CDD65	571	0	0	1	5	31	117	198	160	53	6	0	0	(11)
(12)			CDH74	4637	0	0	10	75	313	999	1642	1156	394	48	0	0	(12)
(13)			CDH80	1290	0	0	0	16	72	290	514	305	90	3	0	0	(13)
(14)	Wind		WSAvg	9.1	10.7	10.5	10.3	10.2	8.8	8.0	7.6	7.1	7.5	8.6	9.4	10.0	(14)
(15)	Precipitation		PrecAvg	32.00	2.10	2.10	2.30	2.60	2.70	3.00	2.70	3.40	3.00	2.40	2.90	2.70	(15)
(16)			PrecMax	39.40	5.80	4.40	3.80	4.10	6.60	6.80	5.60	5.90	6.30	4.70	4.80	4.60	(16)
(17)			PrecMin	22.90	0.90	0.70	1.10	1.30	0.40	0.20	1.10	1.80	0.30	0.20	0.40	1.00	(17)
(18)			PrecStd	4.90	1.20	1.10	0.80	0.80	1.30	1.80	1.30	1.10	1.60	1.20	1.30	1.10	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		61.1	58.2	73.1	82.2	86.6	90.4	92.4	91.1	88.2	79.6	68.5	59.7	(19)
(20)			MCWB		54.6	48.5	61.1	64.4	69.5	73.7	74.9	74.2	71.7	65.0	57.5	52.7	(20)
(21)		2%	DB		53.4	50.3	63.2	74.7	81.6	86.6	88.9	86.4	82.2	74.1	64.2	53.4	(21)
(22)			MCWB		49.6	43.6	53.6	59.3	66.8	71.4	74.0	72.0	69.6	62.7	55.5	48.2	(22)
(23)		5%	DB		46.7	44.7	57.1	68.2	77.3	83.5	85.5	83.3	78.5	69.0	60.2	48.4	(23)
(24)			MCWB		42.6	39.2	49.6	55.9	64.2	70.0	71.6	70.3	67.4	60.1	53.3	43.8	(24)
(25)		10%	DB		40.8	40.0	51.1	62.8	72.8	80.0	82.4	80.4	74.8	64.7	54.9	43.9	(25)
(26)			MCWB		37.2	36.1	44.1	53.2	61.7	67.7	69.8	69.0	65.4	57.3	48.9	39.8	(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		55.1	49.8	61.6	65.7	72.4	76.4	78.3	77.2	73.9	67.9	60.1	54.6	(27)
(28)			MCDB		60.3	54.8	73.1	79.3	82.3	86.1	88.3	87.4	82.9	75.5	65.6	58.2	(28)
(29)		2%	WB		49.8	44.3	55.3	61.4	69.2	73.8	75.7	74.3	71.2	64.0	57.1	49.3	(29)
(30)			MCDB		53.0	49.2	61.3	71.8	78.3	83.2	85.4	82.5	79.0	72.3	62.5	52.6	(30)
(31)		5%	WB		43.0	40.1	50.2	57.8	66.4	71.8	73.7	72.5	69.1	61.2	53.6	44.2	(31)
(32)			MCDB		46.4	44.4	56.6	67.0	75.1	80.4	82.2	80.1	76.0	67.8	59.5	48.2	(32)
(33)		10%	WB		37.2	36.3	44.3	53.7	63.4	69.8	71.9	70.8	67.1	58.6	49.9	40.1	(33)
(34)			MCDB		40.4	39.7	50.4	61.8	71.1	77.1	79.6	77.8	73.6	64.2	54.7	43.4	(34)
(35)	Mean Daily Temperature Range		MDBR		13.3	14.3	15.7	18.4	19.8	19.0	18.6	18.2	18.5	16.6	14.1	12.1	(35)
(36)		5% DB	MCDBR		18.5	20.3	24.4	27.5	24.9	22.8	21.6	21.9	22.7	22.8	21.1	18.0	(36)
(37)			MCWBR		16.0	15.7	16.3	16.0	13.2	11.2	10.1	10.3	11.8	13.7	15.0	14.8	(37)
(38)		5% WB	MCDBR		18.2	18.9	23.6	25.1	22.3	19.9	18.5	18.5	19.3	20.7	19.1	16.7	(38)
(39)			MCWBR		16.4	15.6	16.8	16.1	13.2	11.3	9.9	9.8	11.2	13.4	15.1	14.9	(39)
(40)	Clear-Sky Solar Irradiance		taub		0.306	0.316	0.335	0.371	0.395	0.436	0.433	0.421	0.374	0.357	0.313	0.294	(40)
(41)			taud		2.380	2.355	2.389	2.339	2.320	2.235	2.255	2.274	2.391	2.427	2.511	2.483	(41)
(42)			Ebn,noon		262	279	287	284	279	266	266	265	271	261	259	258	(42)
(43)			Edh,noon		25	30	33	38	40	43	42	40	33	28	22	21	(43)
(44)	All-Sky Solar Radiation		RadAvg		439	711	1109	1460	1772	1879	1892	1668	1318	800	493	357	(44)
(45)			RadStd		35	76	102	159	163	137	139	100	114	64	56	42	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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SPRINGFIELD-BRANSON REGIONAL , MO, USA WMO#: **724400**

Lat: **37.240N** Long: **93.390W** Elev: **1259** StdP: **14.04** Time Zone: **-6.00 (NAC)** Period: **90-14** WBAN: **13995**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	7.1	12.8	-1.8	5.2	10.0	3.2	6.8	15.8	25.7	42.6	24.0	39.9	8.4	340	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	19.3	95.5	74.0	92.4	74.1	89.6	74.0	77.7	88.7	76.6	87.5	75.4	85.8	8.5	210	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	74.5	135.4	83.4	73.4	130.1	82.2	72.3	125.5	81.0	42.0	88.8	40.8	87.3	39.7	86.1	81.9	(3)

Extreme Annual Design Conditions

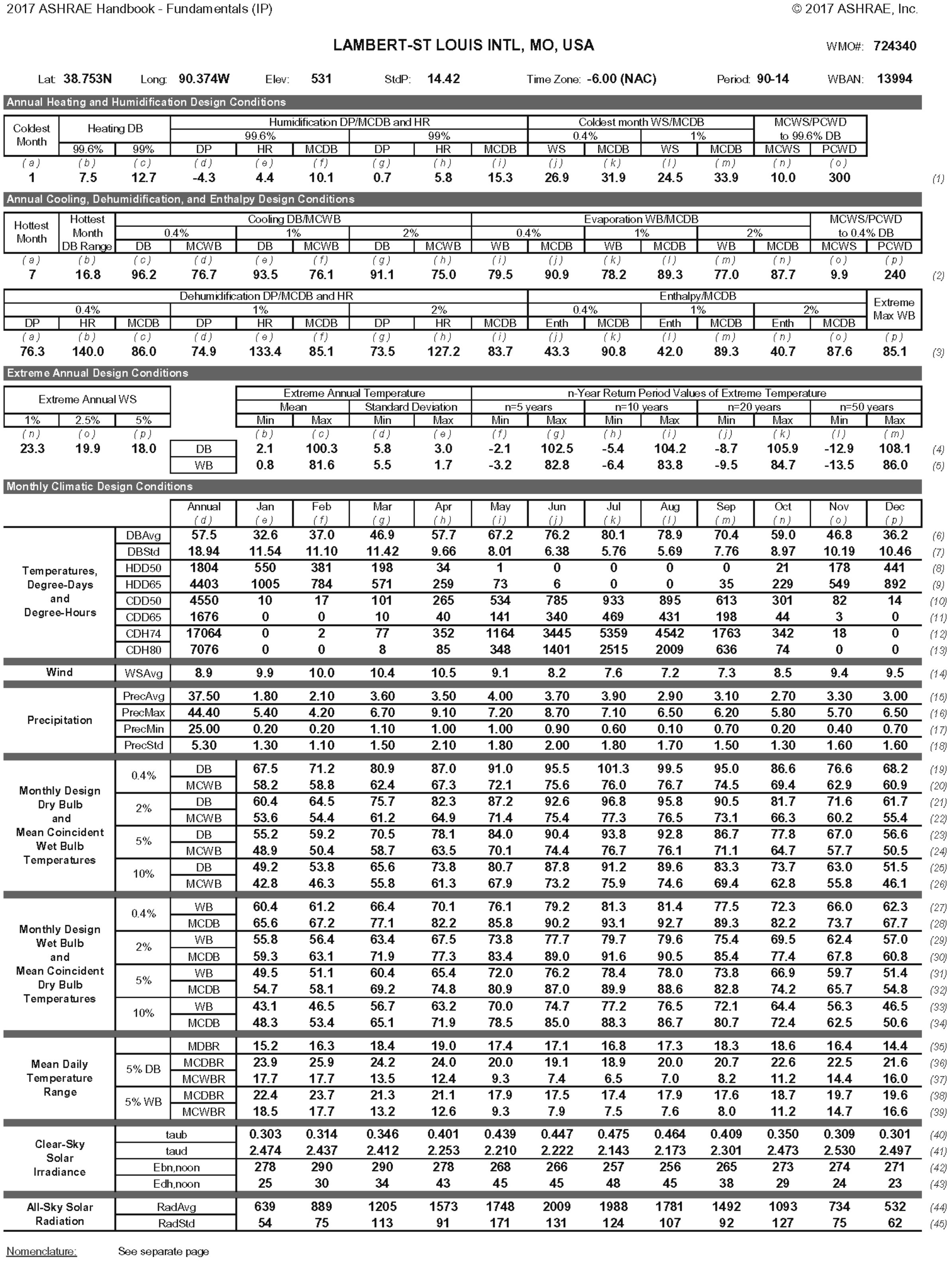
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	23.6	20.4	18.5	DB	-0.1	98.6	6.0	3.5	-4.4	101.1	-7.9	103.2	-11.3	105.1	-15.6	107.7	(4)
(5)				WB	-0.9	79.3	5.8	1.5	-5.1	80.4	-8.4	81.3	-11.7	82.1	-15.9	83.3	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**		DBAvg	56.7	33.8	38.3	46.7	56.3	65.1	73.9	78.2	77.7	68.9	57.8	46.3	36.3	(6)
(7)			DBStd	17.94	11.51	10.99	11.02	9.17	7.76	5.62	5.12	5.64	7.74	8.78	10.01	10.48	(7)
(8)			HDD50	1758	514	349	195	42	3	0	0	0	1	30	185	439	(8)
(9)			HDD65	4428	967	748	571	283	98	7	1	2	48	253	561	889	(9)
(10)			CDD50	4211	12	21	94	233	470	717	874	860	569	271	75	15	(10)
(11)			CDD65	1406	0	0	5	24	100	275	410	396	166	29	1	0	(11)
(12)			CDH74	13164	0	1	37	193	732	2421	4170	3954	1419	228	9	0	(12)
(13)			CDH80	5348	0	0	2	24	159	883	1888	1847	506	39	0	0	(13)
(14)	**Wind**		WSAvg	9.2	10.0	10.2	11.0	10.7	9.1	8.5	7.4	7.2	7.9	9.0	10.0	9.9	(14)
(15)	**Precipitation**		PrecAvg	43.00	1.80	2.20	3.90	4.20	4.40	5.10	2.90	3.50	4.60	3.60	3.80	3.20	(15)
(16)			PrecMax	57.10	4.40	5.00	9.00	7.40	9.40	11.30	8.00	6.30	11.30	8.70	8.10	6.20	(16)
(17)			PrecMin	27.60	0.10	0.60	0.90	1.10	1.50	1.30	0.70	0.80	1.40	0.40	0.20	0.40	(17)
(18)			PrecStd	8.10	1.20	1.00	2.20	1.70	2.10	2.60	1.90	1.70	3.00	2.00	2.30	1.50	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB		67.3	70.4	78.1	83.1	87.2	93.4	99.0	99.2	95.2	84.3	75.1	67.5	(19)
(20)			MCWB		56.4	56.0	62.0	65.6	71.4	74.0	72.7	73.2	72.2	68.2	62.1	59.6	(20)
(21)		2%	DB		61.7	64.8	73.8	79.4	84.1	90.2	95.2	96.0	89.5	79.8	69.9	62.3	(21)
(22)			MCWB		53.0	52.6	59.6	63.2	70.3	74.0	74.3	74.1	71.2	65.2	59.2	55.5	(22)
(23)		5%	DB		56.6	60.5	69.0	75.7	81.4	87.8	92.1	93.0	85.7	76.1	66.0	57.2	(23)
(24)			MCWB		49.7	50.6	57.1	61.9	69.3	73.4	74.7	74.0	70.6	63.4	57.7	50.9	(24)
(25)		10%	DB		51.1	55.9	64.7	72.0	78.4	85.4	89.4	89.6	82.2	72.2	62.1	52.2	(25)
(26)			MCWB		45.0	47.9	54.7	60.3	67.7	72.7	75.0	73.7	68.5	62.0	55.2	46.9	(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB		59.8	59.9	65.0	68.4	74.3	78.0	79.4	78.5	75.4	71.3	64.7	61.4	(27)
(28)			MCDB		65.0	65.4	74.6	78.4	83.8	88.3	89.9	90.3	86.3	81.0	70.9	66.4	(28)
(29)		2%	WB		55.7	56.0	62.0	66.4	72.6	76.4	78.1	77.3	73.8	68.6	61.7	56.9	(29)
(30)			MCDB		60.1	62.0	70.6	75.5	81.1	86.6	88.6	89.3	84.7	75.4	67.3	61.2	(30)
(31)		5%	WB		50.7	52.1	59.2	64.3	70.7	74.9	77.0	76.1	72.4	66.2	59.2	52.1	(31)
(32)			MCDB		55.3	57.8	66.6	72.3	78.8	84.7	87.4	87.8	82.1	72.6	64.3	55.6	(32)
(33)		10%	WB		44.7	48.1	56.1	62.1	68.9	73.7	75.8	74.9	70.9	63.8	56.1	47.5	(33)
(34)			MCDB		51.0	55.7	63.5	69.7	76.3	82.8	85.9	86.0	79.2	70.1	61.6	51.2	(34)
(35)	**Mean Daily Temperature Range**		MDBR		18.6	19.6	20.8	21.0	18.9	18.8	19.3	20.6	20.7	21.0	19.2	17.5	(35)
(36)		5% DB	MCDBR		25.5	26.6	25.6	24.7	20.8	21.0	22.7	24.0	23.2	24.0	23.7	24.1	(36)
(37)			MCWBR		17.6	17.4	14.9	13.5	10.4	8.7	7.4	7.4	9.3	12.5	15.7	17.4	(37)
(38)		5% WB	MCDBR		21.8	22.9	21.7	20.9	18.8	19.1	19.2	20.3	19.6	19.8	20.4	21.3	(38)
(39)			MCWBR		17.1	17.4	14.1	13.1	10.4	9.1	8.1	7.8	9.0	12.1	15.4	17.3	(39)
(40)	**Clear-Sky Solar Irradiance**	taub			0.299	0.307	0.339	0.376	0.414	0.422	0.440	0.438	0.381	0.340	0.308	0.291	(40)
(41)		taud			2.467	2.450	2.405	2.320	2.271	2.276	2.274	2.256	2.391	2.482	2.519	2.512	(41)
(42)		Ebn,noon			283	295	294	287	276	273	267	265	276	279	278	279	(42)
(43)		Edh,noon			26	30	35	40	43	43	42	42	35	29	25	23	(43)
(44)	**All-Sky Solar Radiation**	RadAvg			727	923	1257	1600	1780	2015	2001	1833	1536	1135	804	609	(44)
(45)		RadStd			62	112	87	87	157	170	118	109	112	139	94	77	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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LAMBERT-ST LOUIS INTL, MO, USA WMO#: **724340**

Lat: **38.753N** Long: **90.374W** Elev: **531** StdP: **14.42** Time Zone: **-6.00 (NAC)** Period: **90-14** WBAN: **13994**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	7.5	12.7	-4.3	4.4	10.1	0.7	5.8	15.3	26.9	31.9	24.5	33.9	10.0	300	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	16.8	96.2	76.7	93.5	76.1	91.1	75.0	79.5	90.9	78.2	89.3	77.0	87.7	9.9	240	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	76.3	140.0	86.0	74.9	133.4	85.1	73.5	127.2	83.7	43.3	90.8	42.0	89.3	40.7	87.6	85.1	(3)

Extreme Annual Design Conditions

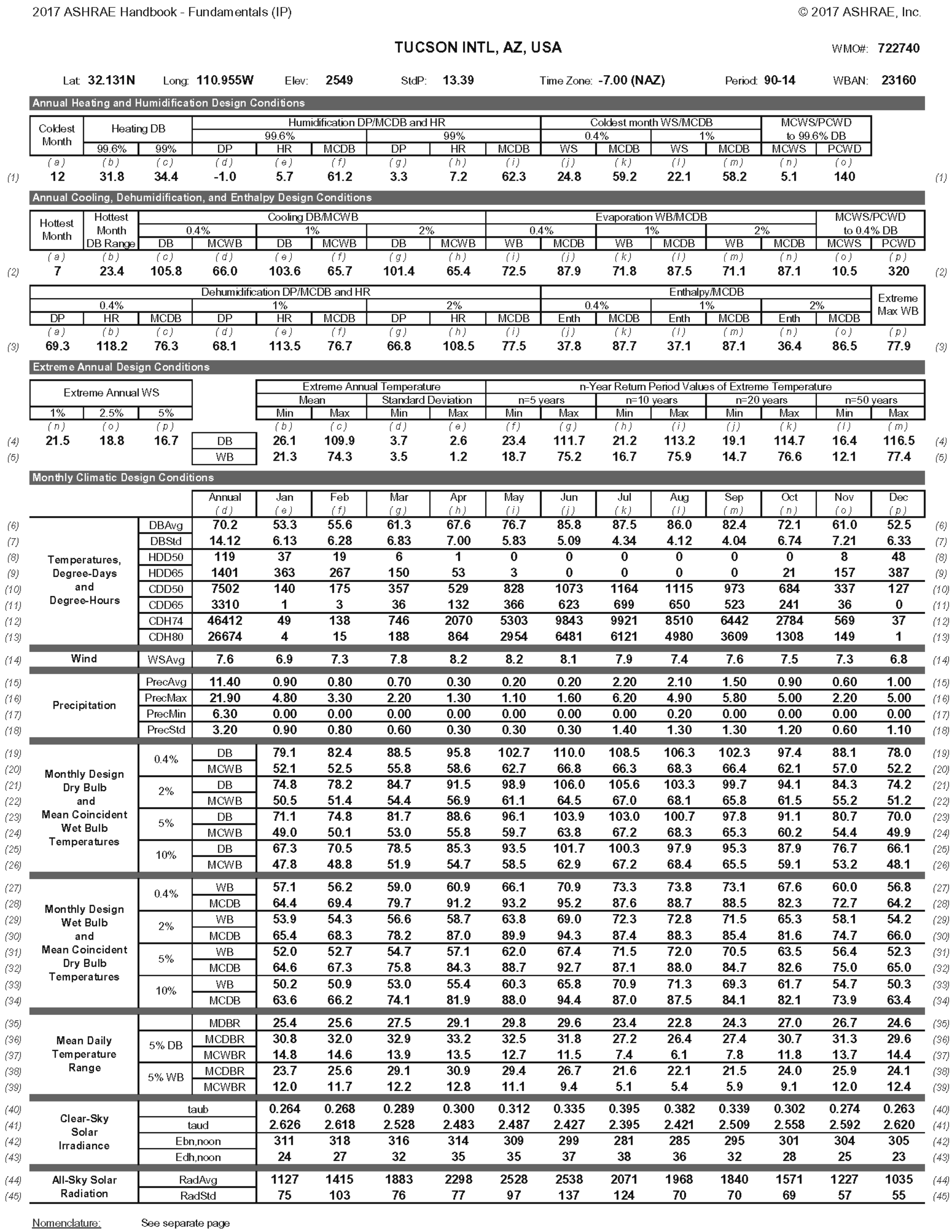
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	23.3	19.9	18.0	DB	2.1	100.3	5.8	3.0	-2.1	102.5	-5.4	104.2	-8.7	105.9	-12.9	108.1	(4)
(5)				WB	0.8	81.6	5.5	1.7	-3.2	82.8	-6.4	83.8	-9.5	84.7	-13.5	86.0	(5)

Monthly Climatic Design Conditions

				Annual (d)	Jan (e)	Feb (f)	Mar (g)	Apr (h)	May (i)	Jun (j)	Jul (k)	Aug (l)	Sep (m)	Oct (n)	Nov (o)	Dec (p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	57.5	32.6	37.0	46.9	57.7	67.2	76.2	80.1	78.9	70.4	59.0	46.8	36.2	(6)
(7)			DBStd	18.94	11.54	11.10	11.42	9.66	8.01	6.38	5.76	5.69	7.76	8.97	10.19	10.46	(7)
(8)			HDD50	1804	550	381	198	34	1	0	0	0	0	21	178	441	(8)
(9)			HDD65	4403	1005	784	571	259	73	6	0	0	35	229	549	892	(9)
(10)			CDD50	4550	10	17	101	265	534	785	933	895	613	301	82	14	(10)
(11)			CDD65	1676	0	0	10	40	141	340	469	431	198	44	3	0	(11)
(12)			CDH74	17064	0	2	77	352	1164	3445	5359	4542	1763	342	18	0	(12)
(13)			CDH80	7076	0	0	8	85	348	1401	2515	2009	636	74	0	0	(13)
(14)	Wind		WSAvg	8.9	9.9	10.0	10.4	10.5	9.1	8.2	7.6	7.2	7.3	8.5	9.4	9.5	(14)
(15)	Precipitation		PrecAvg	37.50	1.80	2.10	3.60	3.50	4.00	3.70	3.90	2.90	3.10	2.70	3.30	3.00	(15)
(16)			PrecMax	44.40	5.40	4.20	6.70	9.10	7.20	8.70	7.10	6.50	6.20	5.80	5.70	6.50	(16)
(17)			PrecMin	25.00	0.20	0.20	1.10	1.00	1.00	0.90	0.60	0.10	0.70	0.20	0.40	0.70	(17)
(18)			PrecStd	5.30	1.30	1.10	1.50	2.10	1.80	2.00	1.80	1.70	1.50	1.30	1.60	1.60	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		67.5	71.2	80.9	87.0	91.0	95.5	101.3	99.5	95.0	86.6	76.6	68.2	(19)
(20)			MCWB		58.2	58.8	62.4	67.3	72.1	75.6	76.0	76.7	74.5	69.4	62.9	60.9	(20)
(21)		2%	DB		60.4	64.5	75.7	82.3	87.2	92.6	96.8	95.8	90.5	81.7	71.6	61.7	(21)
(22)			MCWB		53.6	54.4	61.2	64.9	71.4	75.4	77.3	76.5	73.1	66.3	60.2	55.4	(22)
(23)		5%	DB		55.2	59.2	70.5	78.1	84.0	90.4	93.8	92.8	86.7	77.8	67.0	56.6	(23)
(24)			MCWB		48.9	50.4	58.7	63.5	70.1	74.4	76.7	76.1	71.1	64.7	57.7	50.5	(24)
(25)		10%	DB		49.2	53.8	65.6	73.8	80.7	87.8	91.2	89.6	83.3	73.7	63.0	51.5	(25)
(26)			MCWB		42.8	46.3	55.8	61.3	67.9	73.2	75.9	74.6	69.4	62.8	55.8	46.1	(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		60.4	61.2	66.4	70.1	76.1	79.2	81.3	81.4	77.5	72.3	66.0	62.3	(27)
(28)			MCDB		65.6	67.2	77.1	82.2	85.8	90.2	93.1	92.7	89.3	82.2	73.7	67.7	(28)
(29)		2%	WB		55.8	56.4	63.4	67.5	73.8	77.7	79.7	79.6	75.4	69.5	62.4	57.0	(29)
(30)			MCDB		59.3	63.1	71.9	77.3	83.4	89.0	91.6	90.5	85.4	77.4	67.8	60.8	(30)
(31)		5%	WB		49.5	51.1	60.4	65.4	72.0	76.2	78.4	78.0	73.8	66.9	59.7	51.4	(31)
(32)			MCDB		54.7	58.1	69.2	74.8	80.9	87.0	89.9	88.6	82.8	74.2	65.7	54.8	(32)
(33)		10%	WB		43.1	46.5	56.7	63.2	70.0	74.7	77.2	76.5	72.1	64.4	56.3	46.5	(33)
(34)			MCDB		48.3	53.4	65.1	71.9	78.5	85.0	88.3	86.7	80.7	72.4	62.5	50.6	(34)
(35)	Mean Daily Temperature Range		MDBR		15.2	16.3	18.4	19.0	17.4	17.1	16.8	17.3	18.3	18.6	16.4	14.4	(35)
(36)		5% DB	MCDBR		23.9	25.9	24.2	24.0	20.0	19.1	18.9	20.0	20.7	22.6	22.5	21.6	(36)
(37)			MCWBR		17.7	17.7	13.5	12.4	9.3	7.4	6.5	7.0	8.2	11.2	14.4	16.0	(37)
(38)		5% WB	MCDBR		22.4	23.7	21.3	21.1	17.9	17.5	17.4	17.9	17.6	18.7	19.7	19.6	(38)
(39)			MCWBR		18.5	17.7	13.2	12.6	9.3	7.9	7.5	7.6	8.0	11.2	14.7	16.6	(39)
(40)	Clear-Sky Solar Irradiance		taub		0.303	0.314	0.346	0.401	0.439	0.447	0.475	0.464	0.409	0.350	0.309	0.301	(40)
(41)			taud		2.474	2.437	2.412	2.253	2.210	2.222	2.143	2.173	2.301	2.473	2.530	2.497	(41)
(42)			Ebn,noon		278	290	290	278	268	266	257	256	265	273	274	271	(42)
(43)			Edh,noon		25	30	34	43	45	45	48	45	38	29	24	23	(43)
(44)	All-Sky Solar Radiation		RadAvg		639	889	1205	1573	1748	2009	1988	1781	1492	1093	734	532	(44)
(45)			RadStd		54	75	113	91	171	131	124	107	92	127	75	62	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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TUCSON INTL, AZ, USA WMO#: **722740**

Lat: **32.131N** Long: **110.955W** Elev: **2549** StdP: **13.39** Time Zone: **-7.00 (NAZ)** Period: **90-14** WBAN: **23160**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	12	31.8	34.4	-1.0	5.7	61.2	3.3	7.2	62.3	24.8	59.2	22.1	58.2	5.1	140	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	23.4	105.8	66.0	103.6	65.7	101.4	65.4	72.5	87.9	71.8	87.5	71.1	87.1	10.5	320	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	69.3	118.2	76.3	68.1	113.5	76.7	66.8	108.5	77.5	37.8	87.7	37.1	87.1	36.4	86.5	77.9	(3)

Extreme Annual Design Conditions

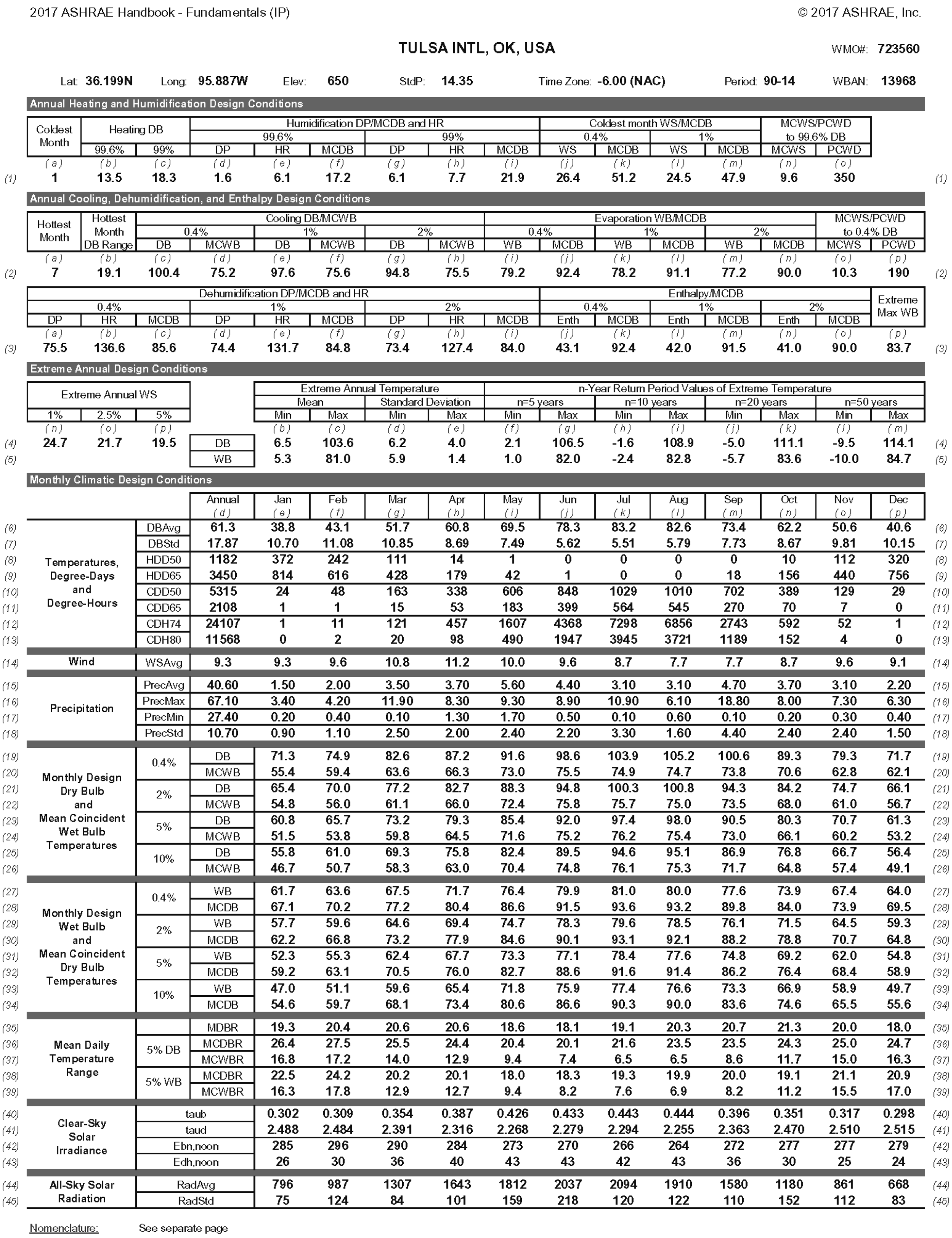
	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	21.5	18.8	16.7	DB	26.1	109.9	3.7	2.6	23.4	111.7	21.2	113.2	19.1	114.7	16.4	116.5	(4)
(5)				WB	21.3	74.3	3.5	1.2	18.7	75.2	16.7	75.9	14.7	76.6	12.1	77.4	(5)

Monthly Climatic Design Conditions

				Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	
				(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(6)	Temperatures, Degree-Days and Degree-Hours		DBAvg	70.2	53.3	55.6	61.3	67.6	76.7	85.8	87.5	86.0	82.4	72.1	61.0	52.5	(6)
(7)			DBStd	14.12	6.13	6.28	6.83	7.00	5.83	5.09	4.34	4.12	4.04	6.74	7.21	6.33	(7)
(8)			HDD50	119	37	19	6	1	0	0	0	0	0	0	8	48	(8)
(9)			HDD65	1401	363	267	150	53	3	0	0	0	0	21	157	387	(9)
(10)			CDD50	7502	140	175	357	529	828	1073	1164	1115	973	684	337	127	(10)
(11)			CDD65	3310	1	3	36	132	366	623	699	650	523	241	36	0	(11)
(12)			CDH74	46412	49	138	746	2070	5303	9843	9921	8510	6442	2784	569	37	(12)
(13)			CDH80	26674	4	15	188	864	2954	6481	6121	4980	3609	1308	149	1	(13)
(14)	Wind		WSAvg	7.6	6.9	7.3	7.8	8.2	8.2	8.1	7.9	7.4	7.6	7.5	7.3	6.8	(14)
(15)	Precipitation		PrecAvg	11.40	0.90	0.80	0.70	0.30	0.20	0.20	2.20	2.10	1.50	0.90	0.60	1.00	(15)
(16)			PrecMax	21.90	4.80	3.30	2.20	1.30	1.10	1.60	6.20	4.90	5.80	5.00	2.20	5.00	(16)
(17)			PrecMin	6.30	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.20	0.00	0.00	0.00	0.00	(17)
(18)			PrecStd	3.20	0.90	0.80	0.60	0.30	0.30	0.30	1.40	1.30	1.30	1.20	0.60	1.10	(18)
(19)	Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures	0.4%	DB		79.1	82.4	88.5	95.8	102.7	110.0	108.5	106.3	102.3	97.4	88.1	78.0	(19)
(20)			MCWB		52.1	52.5	55.8	58.6	62.7	66.8	66.3	68.3	66.4	62.1	57.0	52.2	(20)
(21)		2%	DB		74.8	78.2	84.7	91.5	98.9	106.0	105.6	103.3	99.7	94.1	84.3	74.2	(21)
(22)			MCWB		50.5	51.4	54.4	56.9	61.1	64.5	67.0	68.1	65.8	61.5	55.2	51.2	(22)
(23)		5%	DB		71.1	74.8	81.7	88.6	96.1	103.9	103.0	100.7	97.8	91.1	80.7	70.0	(23)
(24)			MCWB		49.0	50.1	53.0	55.8	59.7	63.8	67.2	68.3	65.3	60.2	54.4	49.9	(24)
(25)		10%	DB		67.3	70.5	78.5	85.3	93.5	101.7	100.3	97.9	95.3	87.9	76.7	66.1	(25)
(26)			MCWB		47.8	48.8	51.9	54.7	58.5	62.9	67.2	68.4	65.5	59.1	53.2	48.1	(26)
(27)	Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures	0.4%	WB		57.1	56.2	59.0	60.9	66.1	70.9	73.3	73.8	73.1	67.6	60.0	56.8	(27)
(28)			MCDB		64.4	69.4	79.7	91.2	93.2	95.2	87.6	88.7	88.5	82.3	72.7	64.2	(28)
(29)		2%	WB		53.9	54.3	56.6	58.7	63.8	69.0	72.3	72.8	71.5	65.3	58.1	54.2	(29)
(30)			MCDB		65.4	68.3	78.2	87.0	89.9	94.3	87.4	88.3	85.4	81.6	74.7	66.0	(30)
(31)		5%	WB		52.0	52.7	54.7	57.1	62.0	67.4	71.5	72.0	70.5	63.5	56.4	52.3	(31)
(32)			MCDB		64.6	67.3	75.8	84.3	88.7	92.7	87.1	88.0	84.7	82.6	75.0	65.0	(32)
(33)		10%	WB		50.2	50.9	53.0	55.4	60.3	65.8	70.9	71.3	69.3	61.7	54.7	50.3	(33)
(34)			MCDB		63.6	66.2	74.1	81.9	88.0	94.4	87.0	87.5	84.1	82.1	73.9	63.4	(34)
(35)	Mean Daily Temperature Range		MDBR		25.4	25.6	27.5	29.1	29.8	29.6	23.4	22.8	24.3	27.0	26.7	24.6	(35)
(36)		5% DB	MCDBR		30.8	32.0	32.9	33.2	32.5	31.8	27.2	26.4	27.4	30.7	31.3	29.6	(36)
(37)			MCWBR		14.8	14.6	13.9	13.5	12.7	11.5	7.4	6.1	7.8	11.8	13.7	14.4	(37)
(38)		5% WB	MCDBR		23.7	25.6	29.1	30.9	29.4	26.7	21.6	22.1	21.5	24.0	25.9	24.1	(38)
(39)			MCWBR		12.0	11.7	12.2	12.8	11.1	9.4	5.1	5.4	5.9	9.1	12.0	12.4	(39)
(40)	Clear-Sky Solar Irradiance		taub		0.264	0.268	0.289	0.300	0.312	0.335	0.395	0.382	0.339	0.302	0.274	0.263	(40)
(41)			taud		2.626	2.618	2.528	2.483	2.487	2.427	2.395	2.421	2.509	2.558	2.592	2.620	(41)
(42)			Ebn,noon		311	318	316	314	309	299	281	285	295	301	304	305	(42)
(43)			Edh,noon		24	27	32	35	35	37	38	36	32	28	25	23	(43)
(44)	All-Sky Solar Radiation		RadAvg		1127	1415	1883	2298	2528	2538	2071	1968	1840	1571	1227	1035	(44)
(45)			RadStd		75	103	76	77	97	137	124	70	70	69	57	55	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals



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TULSA INTL, OK, USA WMO#: **723560**

Lat: **36.199N** Long: **95.887W** Elev: **650** StdP: **14.35** Time Zone: **-6.00 (NAC)** Period: **90-14** WBAN: **13968**

Annual Heating and Humidification Design Conditions

	Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		
				99.6%			99%			0.4%		1%				
		99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	
(1)	1	13.5	18.3	1.6	6.1	17.2	6.1	7.7	21.9	26.4	51.2	24.5	47.9	9.6	350	(1)

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

	Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB		
			0.4%		1%		2%		0.4%		1%		2%				
			DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD	
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(2)	7	19.1	100.4	75.2	97.6	75.6	94.8	75.5	79.2	92.4	78.2	91.1	77.2	90.0	10.3	190	(2)

	Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB	
	0.4%			1%			2%			0.4%		1%		2%			
	DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB		
	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(3)	75.5	136.6	85.6	74.4	131.7	84.8	73.4	127.4	84.0	43.1	92.4	42.0	91.5	41.0	90.0	83.7	(3)

Extreme Annual Design Conditions

	Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature								
					Mean		Standard Deviation		n=5 years		n=10 years		n=20 years		n=50 years		
	1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	
	(n)	(o)	(p)		(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	
(4)	24.7	21.7	19.5	DB	6.5	103.6	6.2	4.0	2.1	106.5	-1.6	108.9	-5.0	111.1	-9.5	114.1	(4)
(5)				WB	5.3	81.0	5.9	1.4	1.0	82.0	-2.4	82.8	-5.7	83.6	-10.0	84.7	(5)

Monthly Climatic Design Conditions

			Annual	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	
			(d)	(e)	(f)	(g)	(h)	(i)	(j)	(k)	(l)	(m)	(n)	(o)	(p)	
(6)	**Temperatures, Degree-Days and Degree-Hours**	DBAvg	61.3	38.8	43.1	51.7	60.8	69.5	78.3	83.2	82.6	73.4	62.2	50.6	40.6	(6)
(7)		DBStd	17.87	10.70	11.08	10.85	8.69	7.49	5.62	5.51	5.79	7.73	8.67	9.81	10.15	(7)
(8)		HDD50	1182	372	242	111	14	1	0	0	0	0	10	112	320	(8)
(9)		HDD65	3450	814	616	428	179	42	1	0	0	18	156	440	756	(9)
(10)		CDD50	5315	24	48	163	338	606	848	1029	1010	702	389	129	29	(10)
(11)		CDD65	2108	1	1	15	53	183	399	564	545	270	70	7	0	(11)
(12)		CDH74	24107	1	11	121	457	1607	4368	7298	6856	2743	592	52	1	(12)
(13)		CDH80	11568	0	2	20	98	490	1947	3945	3721	1189	152	4	0	(13)
(14)	**Wind**	WSAvg	9.3	9.3	9.6	10.8	11.2	10.0	9.6	8.7	7.7	7.7	8.7	9.6	9.1	(14)
(15)	**Precipitation**	PrecAvg	40.60	1.50	2.00	3.50	3.70	5.60	4.40	3.10	3.10	4.70	3.70	3.10	2.20	(15)
(16)		PrecMax	67.10	3.40	4.20	11.90	8.30	9.30	8.90	10.90	6.10	18.80	8.00	7.30	6.30	(16)
(17)		PrecMin	27.40	0.20	0.40	0.10	1.30	1.70	0.50	0.10	0.60	0.10	0.20	0.30	0.40	(17)
(18)		PrecStd	10.70	0.90	1.10	2.50	2.00	2.40	2.20	3.30	1.60	4.40	2.40	2.40	1.50	(18)
(19)	**Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures**	0.4%	DB	71.3	74.9	82.6	87.2	91.6	98.6	103.9	105.2	100.6	89.3	79.3	71.7	(19)
(20)			MCWB	55.4	59.4	63.6	66.3	73.0	75.5	74.9	74.7	73.8	70.6	62.8	62.1	(20)
(21)		2%	DB	65.4	70.0	77.2	82.7	88.3	94.8	100.3	100.8	94.3	84.2	74.7	66.1	(21)
(22)			MCWB	54.8	56.0	61.1	66.0	72.4	75.8	75.7	75.0	73.5	68.0	61.0	56.7	(22)
(23)		5%	DB	60.8	65.7	73.2	79.3	85.4	92.0	97.4	98.0	90.5	80.3	70.7	61.3	(23)
(24)			MCWB	51.5	53.8	59.8	64.5	71.6	75.2	76.2	75.4	73.0	66.1	60.2	53.2	(24)
(25)		10%	DB	55.8	61.0	69.3	75.8	82.4	89.5	94.6	95.1	86.9	76.8	66.7	56.4	(25)
(26)			MCWB	46.7	50.7	58.3	63.0	70.4	74.8	76.1	75.3	71.7	64.8	57.4	49.1	(26)
(27)	**Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures**	0.4%	WB	61.7	63.6	67.5	71.7	76.4	79.9	81.0	80.0	77.6	73.9	67.4	64.0	(27)
(28)			MCDB	67.1	70.2	77.2	80.4	86.6	91.5	93.6	93.2	89.8	84.0	73.9	69.5	(28)
(29)		2%	WB	57.7	59.6	64.6	69.4	74.7	78.3	79.6	78.5	76.1	71.5	64.5	59.3	(29)
(30)			MCDB	62.2	66.8	73.2	77.9	84.6	90.1	93.1	92.1	88.2	78.8	70.7	64.8	(30)
(31)		5%	WB	52.3	55.3	62.4	67.7	73.3	77.1	78.4	77.6	74.8	69.2	62.0	54.8	(31)
(32)			MCDB	59.2	63.1	70.5	76.0	82.7	88.6	91.6	91.4	86.2	76.4	68.4	58.9	(32)
(33)		10%	WB	47.0	51.1	59.6	65.4	71.8	75.9	77.4	76.6	73.3	66.9	58.9	49.7	(33)
(34)			MCDB	54.6	59.7	68.1	73.4	80.6	86.6	90.3	90.0	83.6	74.6	65.5	55.6	(34)
(35)	**Mean Daily Temperature Range**		MDBR	19.3	20.4	20.6	20.6	18.6	18.1	19.1	20.3	20.7	21.3	20.0	18.0	(35)
(36)		5% DB	MCDBR	26.4	27.5	25.5	24.4	20.4	20.1	21.6	23.5	23.5	24.3	25.0	24.7	(36)
(37)			MCWBR	16.8	17.2	14.0	12.9	9.4	7.4	6.5	6.5	8.6	11.7	15.0	16.3	(37)
(38)		5% WB	MCDBR	22.5	24.2	20.2	20.1	18.0	18.3	19.3	19.9	20.0	19.1	21.1	20.9	(38)
(39)			MCWBR	16.3	17.8	12.9	12.7	9.4	8.2	7.6	6.9	8.2	11.2	15.5	17.0	(39)
(40)	**Clear-Sky Solar Irradiance**	taub		0.302	0.309	0.354	0.387	0.426	0.433	0.443	0.444	0.396	0.351	0.317	0.298	(40)
(41)		taud		2.488	2.484	2.391	2.316	2.268	2.279	2.294	2.255	2.363	2.470	2.510	2.515	(41)
(42)		Ebn,noon		285	296	290	284	273	270	266	264	272	277	277	279	(42)
(43)		Edh,noon		26	30	36	40	43	43	42	43	36	30	25	24	(43)
(44)	**All-Sky Solar Radiation**	RadAvg		796	987	1307	1643	1812	2037	2094	1910	1580	1180	861	668	(44)
(45)		RadStd		75	124	84	101	159	218	120	122	110	152	112	83	(45)

Nomenclature: See separate page

Fig. 4-4 Example tables of climatic design information included on CD-ROM of 2017 ASHRAE Handbook—Fundamentals

- Weather Bureau Army Navy (WBAN) number (–99999 denotes missing).
- Latitude of station, °N/S.
- Longitude of station, °E/W.
- Elevation of station, ft.
- Standard pressure at elevation, in psia (see Chapter 2 for equations used to calculate standard pressure).
- Time zone, h ± UTC
- Time zone code (e.g., NAE = Eastern Time, USA and Canada). The CD-ROM contains a list of all time zone codes used in the tables.
- Period analyzed (e.g., 90–14 = data from 1990 to 2014 were used).
- Table 4-6 gives the nomenclature for tables of climatic design conditions given in Figure 4-4.

Annual Design Conditions. Annual climatic design conditions are contained in the first three sections following the top part of the table. They contain information as follows:

Annual Heating and Humidification Design Conditions.

- Coldest month (i.e., month with lowest average dry-bulb temperature; 1 = January, 12 = December).
- Dry-bulb temperature corresponding to 99.6 and 99.0% annual cumulative frequency of occurrence (cold conditions), °F.
- Dew-point temperature corresponding to 99.6 and 99.0% annual cumulative frequency of occurrence, °F; corresponding humidity ratio, calculated at standard atmospheric pressure at elevation of station, grains of moisture per lb of dry air; mean coincident dry-bulb temperature, °F.
- Wind speed corresponding to 0.4 and 1.0% cumulative frequency of occurrence for coldest month, mph; mean coincident dry-bulb temperature, °F.
- Mean wind speed coincident with 99.6% dry-bulb temperature, mph; corresponding most frequent wind direction, degrees from north (east = 90°).

Annual Cooling, Dehumidification, and Enthalpy Design Conditions.

- Hottest month (i.e., month with highest average dry-bulb temperature; 1 = January, 12 = December).
- Daily temperature range for hottest month, °F [defined as mean of the difference between daily maximum and daily minimum dry-bulb temperatures for hottest month].
- Dry-bulb temperature corresponding to 0.4, 1.0, and 2.0% annual cumulative frequency of occurrence (warm conditions), °F; mean coincident wet-bulb temperature, °F.
- Wet-bulb temperature corresponding to 0.4, 1.0, and 2.0% annual cumulative frequency of occurrence, °F; mean coincident dry-bulb temperature, °F.
- Mean wind speed coincident with 0.4% dry-bulb temperature, mph; corresponding most frequent wind direction, degrees true from north (east = 90°).
- Dew-point temperature corresponding to 0.4, 1.0, and 2.0% annual cumulative frequency of occurrence, °F; corresponding humidity ratio, calculated at the standard atmospheric pressure at elevation of station, grains of moisture per lb of dry air; mean coincident dry-bulb temperature, °F.
- Enthalpy corresponding to 0.4, 1.0, and 2.0% annual cumulative frequency of occurrence, Btu/lb; mean coincident dry-bulb temperature, °F.
- Extreme maximum wet-bulb temperature provides the highest wet-bulb temperature observed over the entire period of record.

Extreme Annual Design Conditions.

- Wind speed corresponding to 1.0, 2.5, and 5.0% annual cumulative frequency of occurrence, mph.
- Mean and standard deviation of extreme annual minimum and maximum dry-bulb temperature, °F.
- 5-, 10-, 20-, and 50-year return period values for minimum and maximum extreme dry-bulb temperature, °F.

Monthly Design Conditions. Monthly design conditions are divided into subsections as follows:

Temperatures, Degree-Days, and Degree-Hours.

- Average temperature, °F. This parameter is a prime indicator of climate and is also useful to calculate heating and cooling degree-days to any base.
- Standard deviation of average daily temperature, °F. This parameter is useful to calculate heating and cooling degree-days to any base. Its use is explained in the section on Estimation of Degree-Days.
- Heating and cooling degree-days (bases 50°F and 65°F). These parameters are useful in energy estimating methods. They are also used to classify locations into climate zones in ASHRAE Standard 169.
- Cooling degree-hours (bases 74°F and 80°F). These are used in various standards, such as Standard 90.2-2007.

Precipitation.

- Average precipitation, in. This parameter is used to calculate climate zones for Standard 169, and is of interest in some green building technologies (e.g., vegetative roofs).
- Standard deviation of precipitation, in. This parameter indicates the variability of precipitation at the site.
- Minimum and maximum precipitation, in. These parameters give extremes of precipitation and are useful for green building technologies and stormwater management.

Monthly Design Dry-Bulb, Wet-Bulb, and Mean Coincident Temperatures. These values are derived from the same analysis that results in the annual design conditions. The monthly summaries are useful when seasonal variations in solar geometry and intensity, building or facility occupancy, or building use patterns require consideration. In particular, these values can be used when determining air-conditioning loads during periods of maximum solar radiation. The values listed in the tables include

- Dry-bulb temperature corresponding to 0.4, 2.0, 5.0, and 10.0% cumulative frequency of occurrence for indicated month, °F; mean coincident wet-bulb temperature, °F.

Table 4-7 Design Conditions for Selected Locations

Meaning of acronyms:
DB: Dry bulb temperature, °F *WB: Wet bulb temperature, °F*
MCWB: Mean coincident wet bulb temperature, °F
Lat: Latitude, °
DP: Dew point temperature, °F
MCDB: Mean coincident dry bulb temperature, °F
Long: Longitude, °
HR: Humidity ratio, grains of moisture per lb of dry air
Elev: Elevation, ft
WS: Wind speed, mph
HDD and CDD 65: Annual heating and cooling degree-days, base 65°F, °F-day

Station	Lat	Long	Elev	Heating DB		Cooling DB/MCWB						Evaporation WB/MCDB				Dehumidification DP/HR/MCDB						Extreme Annual WS			Heat./Cool. Degree-Days	
						0.4%		1%		2%		0.4%		1%		0.4%			1%							
				99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	DP	HR	MCDB	DP	HR	MCDB	1%	2.5%	5%	HDD	CDD 65
United States of America																										*542 sites, 1410 more on CD-ROM*
Alabama																										*11 sites, 28 more on CD-ROM*
AUBURN UNIVERSITY REGIONAL	32.616N	85.433W	776	23.2	27.5	93.4	73.5	91.2	73.9	90.1	73.6	77.6	88.3	76.6	86.7	74.8	134.1	81.5	73.3	127.4	80.3	17.9	15.8	13.1	2416	1916
BIRMINGHAM SHUTTLESWORTH INTL	33.566N	86.745W	615	20.8	25.1	95.6	74.6	93.2	74.5	91.1	74.2	78.4	88.5	77.5	87.5	75.9	138.7	82.8	74.7	133.0	81.8	18.5	16.5	14.6	2610	2054
CAIRNS AAF	31.267N	85.717W	301	26.8	29.8	95.6	76.2	93.4	76.0	91.4	75.6	80.5	88.7	79.3	87.7	78.8	151.2	83.6	77.2	143.2	82.1	17.5	15.2	12.8	1797	2378
DOTHAN REGIONAL	31.317N	85.450W	374	27.4	30.9	96.4	75.7	93.9	75.3	91.9	75.0	79.8	89.5	78.6	88.0	77.4	144.3	83.3	76.3	139.3	82.5	19.7	17.6	15.6	1735	2521
HUNTSVILLE INTL	34.644N	86.786W	624	18.7	22.9	95.3	75.0	92.9	74.6	90.7	74.0	78.4	88.4	77.5	87.4	75.9	138.6	82.7	74.8	133.4	81.7	20.4	18.2	16.3	3052	1846
MAXWELL AFB	32.383N	86.350W	171	25.6	29.6	97.5	76.1	95.5	76.3	93.5	76.3	80.2	91.2	79.3	90.2	77.4	143.4	84.2	76.6	139.6	83.6	18.0	15.8	13.1	1957	2546
MOBILE REGIONAL	30.688N	88.246W	215	27.7	31.1	94.1	76.8	92.2	76.5	90.5	76.1	80.1	88.5	79.1	87.2	78.0	146.4	83.4	77.1	142.0	82.5	20.0	17.9	16.2	1642	2516
MONTGOMERY REGIONAL	32.300N	86.408W	202	24.3	27.9	96.8	76.0	94.7	75.9	92.7	75.6	79.6	90.7	78.5	89.1	76.6	139.9	84.1	75.7	135.4	83.1	18.7	16.5	14.3	2109	2352
NORTHEAST ALABAMA AP	33.967N	86.083W	569	18.8	22.6	93.4	74.4	91.2	74.4	90.0	74.2	78.0	89.0	77.0	87.7	74.8	133.2	83.5	73.3	126.3	82.1	16.6	14.1	12.2	3179	1590
NORTHWEST ALABAMA REGIONAL	34.744N	87.600W	540	19.8	23.8	96.2	75.4	93.7	75.1	91.4	74.7	78.9	89.6	77.9	88.4	76.0	138.7	83.6	75.0	133.8	82.5	18.9	16.8	14.7	2997	1906
TUSCALOOSA REGIONAL	33.212N	87.616W	150	22.5	26.6	97.3	75.4	94.6	75.7	92.4	75.4	79.4	90.1	78.4	88.9	76.8	140.4	83.3	75.4	134.0	82.5	17.2	15.0	12.8	2450	2200
Alaska																										*7 sites, 120 more on CD-ROM*
BRYANT AAF	61.266N	149.653W	388	-19.6	-14.0	74.8	60.3	71.6	58.9	68.3	57.1	61.7	72.6	59.6	69.5	56.3	68.6	65.4	54.9	65.1	62.6	19.3	14.6	11.5	10677	5
ELMENDORF AFB	61.253N	149.794W	212	-15.4	-10.6	74.2	58.7	71.6	57.8	68.2	56.4	60.7	70.3	59.3	67.6	57.2	70.3	61.2	55.5	66.0	60.4	19.6	16.3	13.1	10324	12
FAIRBANKS INTL	64.804N	147.876W	432	-42.8	-38.6	81.2	60.9	78.1	59.8	74.8	58.5	63.2	77.0	61.5	74.0	58.3	73.7	65.5	56.5	69.1	64.3	16.9	14.3	12.0	13577	69
JUNEAU INTL	58.357N	134.564W	16	4.7	8.9	73.9	59.5	70.1	58.0	66.5	56.5	61.0	71.3	59.4	67.4	57.2	69.9	61.3	56.0	66.8	60.4	26.5	23.4	19.6	8377	3
LAKE HOOD SEAPLANE BASE	61.178N	149.966W	90	-8.2	-3.7	74.0	59.5	70.7	58.1	67.9	56.7	61.0	71.4	59.4	68.1	56.5	68.3	63.3	55.1	64.7	62.3	18.4	15.9	12.9	9741	15
MERRILL FIELD	61.217N	149.855W	138	-10.7	-7.0	73.2	59.3	70.6	58.2	68.1	56.9	61.2	70.5	59.7	67.9	57.2	70.0	62.8	55.5	65.8	62.2	15.5	12.3	10.5	9961	13
TED STEVENS ANCHORAGE INTL	61.169N	150.028W	120	-8.7	-4.5	71.5	58.9	68.4	57.4	66.0	56.2	60.4	69.0	58.9	66.3	56.4	68.2	62.9	55.2	65.1	61.7	20.8	18.6	16.6	10115	5
Arizona																										*9 sites, 14 more on CD-ROM*
CASA GRANDE MUNICIPAL	32.950N	111.767W	1462	31.7	34.9	108.5	69.5	106.5	69.1	104.5	68.6	74.0	92.9	73.2	93.9	70.2	117.3	79.4	67.9	108.3	79.6	20.5	17.7	15.1	1498	3575
DAVIS-MONTHAN AFB	32.167N	110.883W	2704	32.4	35.5	105.2	65.2	102.8	64.9	100.4	64.6	72.6	85.5	71.7	86.1	70.0	122.2	76.2	68.3	115.0	76.8	20.6	18.2	16.1	1457	3222
FLAGSTAFF PULLIAM AP	35.144N	111.666W	7003	3.9	9.4	85.8	55.3	83.5	54.9	81.3	54.7	61.4	72.6	60.2	72.2	58.3	94.7	63.8	56.8	89.6	63.1	24.8	20.7	18.3	6801	129
LUKE AFB	33.550N	112.367W	1085	34.8	37.3	110.9	70.0	108.6	69.9	106.4	69.6	76.3	94.1	75.3	93.9	72.5	125.6	81.2	70.5	116.9	82.8	20.7	18.0	15.8	1222	4041
PHOENIX SKY HARBOR INTL	33.428N	112.004W	1107	39.2	41.8	110.3	69.5	108.3	69.3	106.4	69.0	75.7	95.0	74.9	94.6	71.3	120.2	82.0	69.4	112.8	84.1	18.8	16.4	13.5	912	4636
PRESCOTT MUNICIPAL	34.652N	112.421W	5042	17.7	20.8	94.5	60.6	92.1	60.1	90.1	59.7	66.5	80.9	65.3	79.5	63.0	104.2	70.4	61.3	98.0	69.7	21.4	18.7	16.9	4145	1021
TUCSON INTL	32.131N	110.955W	2549	31.8	34.4	105.8	66.0	103.6	65.7	101.4	65.4	72.5	87.9	71.8	87.5	69.3	118.2	76.3	68.1	113.5	76.7	21.5	18.8	16.7	1401	3310
WINDOW ROCK AP	35.658N	109.061W	6739	0.1	5.2	89.8	56.2	87.7	55.7	85.1	55.0	61.6	75.5	60.5	74.7	57.9	92.2	63.4	56.7	88.5	63.4	25.7	21.2	18.3	6395	306
YUMA INTL AP	32.650N	114.600W	207	42.3	44.8	111.0	72.6	108.9	72.3	107.5	71.7	79.3	96.4	78.1	95.4	75.1	132.7	86.9	73.2	124.4	87.8	21.0	18.6	16.6	657	4749
Arkansas																										*11 sites, 20 more on CD-ROM*
BENTONVILLE MUNICIPAL	36.350N	94.217W	1296	10.1	15.9	96.6	74.5	92.6	74.4	90.2	74.1	77.5	90.0	76.4	88.5	73.3	130.1	84.2	72.6	126.9	83.3	19.6	17.5	15.7	4043	1437
CLINTON NATL	34.727N	92.239W	258	20.6	24.5	98.8	77.0	96.0	77.0	93.4	76.6	80.1	92.0	79.1	90.8	77.1	142.4	85.5	76.0	137.3	84.6	19.3	17.2	15.6	2897	2207
DRAKE FIELD	36.010N	94.169W	1251	10.4	16.3	95.9	74.3	92.8	74.4	90.4	74.3	77.9	88.9	76.8	87.9	74.7	136.3	83.4	73.3	129.5	82.1	20.8	18.7	16.9	3962	1426
FORT SMITH REGIONAL	35.333N	94.363W	449	17.9	22.4	100.3	75.8	97.3	76.2	94.5	75.9	79.6	92.4	78.5	90.9	76.4	139.9	84.7	75.3	134.9	83.9	20.4	17.9	16.0	3122	2131
GRIDER FIELD	34.175N	91.935W	206	22.4	25.8	97.5	77.5	95.2	77.3	93.0	76.9	80.4	91.7	79.4	90.9	77.4	143.4	85.7	76.4	138.6	84.9	19.4	17.5	15.8	2745	2223
JONESBORO MUNICIPAL	35.831N	90.646W	262	17.1	20.7	97.1	76.9	94.7	76.2	92.6	75.8	80.0	91.4	78.8	90.1	77.0	141.9	85.4	75.4	134.4	84.2	22.6	19.4	17.4	3504	1958
LITTLE ROCK AFB	34.917N	92.150W	311	18.1	22.2	99.8	77.3	97.0	77.5	94.3	77.2	81.3	91.7	80.1	90.7	78.9	151.9	85.3	77.3	143.9	84.7	18.3	16.0	13.5	3135	2098
NORTH LITTLE ROCK MUNICIPAL	34.835N	92.260W	563	18.4	23.2	95.4	76.6	93.0	76.3	90.9	75.6	79.1	90.2	78.1	88.8	76.0	138.9	84.7	75.0	134.1	83.9	18.4	16.5	14.7	3175	1937
ROGERS MUNICIPAL	36.372N	94.107W	1353	9.9	15.7	95.1	73.3	91.9	73.7	89.8	73.2	76.9	88.2	75.7	86.9	73.3	130.3	82.6	72.4	126.2	81.9	22.2	19.3	17.3	4046	1429
SMITH FIELD	36.191N	94.491W	1193	10.2	16.1	97.2	74.3	93.1	74.3	90.5	74.1	77.4	90.4	76.3	88.8	73.2	129.1	84.5	72.5	125.8	83.7	23.5	20.3	18.2	3954	1489
TEXARKANA REGIONAL	33.454N	94.007W	361	23.8	27.2	99.3	75.7	96.8	75.9	94.2	75.8	79.4	91.2	78.4	90.0	76.4	139.5	84.0	75.4	134.8	83.2	18.8	16.8	14.8	2448	2369
California																										*55 sites, 60 more on CD-ROM*
ALAMEDA	37.772N	122.298W	7	40.4	42.4	81.1	64.0	77.2	62.9	73.9	62.0	66.5	78.6	65.2	76.3	62.4	84.4	68.9	61.0	80.0	67.9	20.3	17.7	15.7	2508	160
BEALE AFB	39.133N	121.433W	113	31.1	33.9	101.1	70.3	98.2	69.1	95.2	67.9	73.0	95.9	71.3	93.3	64.8	92.1	82.2	63.2	87.1	79.7	23.3	19.4	16.9	2398	1548
BOB HOPE AP	34.201N	118.358W	775	38.8	41.1	97.6	67.9	94.1	66.8	91.1	66.4	72.6	89.6	71.0	87.4	66.8	101.5	77.7	65.7	97.5	76.8	18.2	14.8	12.7	1381	1449
BROWN FIELD MUNICIPAL	32.572N	116.979W	515	39.0	41.5	89.5	64.1	85.2	64.6	82.0	64.6	71.1	81.9	69.6	79.2	67.6	103.5	75.4	66.2	98.2	74.0	16.3	13.0	11.9	1647	667
CAMARILLO AP	34.217N	119.083W	77	37.5	39.7	86.2	62.2	82.4	63.1	79.9	63.1	69.1	79.1	67.8	77.6	64.9	92.5	73.6	63.7	88.7	71.7	25.0	20.0	16.2	1819	421
CAMP PENDLETON MCAS	33.300N	117.350W	75	31.9	34.7	92.0	65.8	87.9	65.3	84.3	65.2	71.5	83.5	70.2	81.9	67.6	101.7	76.8	66.0	96.0	75.9	16.6	14.0	12.3	1818	679
CASTLE AFB	37.383N	120.567W	191	30.0	31.9	102.5	69.6	99.9	68.2	97.7	67.3	72.1	96.0	70.7	94.3	64.5	91.4	74.4	63.3	87.6	72.7	20.4	17.6	14.6	2384	1762
DESERT RESORTS REGIONAL	33.627N	116.159W	-118	31.1	34.4	111.5	72.2	109.1	71.7	107.1	71.2	79.7	97.0	78.2	96.8	75.0	130.6	88.8	73.0	121.7	89.0	20.2	17.8	15.7	1092	3915
EL TORO MCAS	33.667N	117.733W	384	43.2	45.3	91.9	67.8	88.9	67.5	86.0	66.8	71.7	86.1	70.4	84.5	66.4	98.5	79.1	65.0	93.8	77.8	15.5	12.3	10.6	1111	1172
FRESNO YOSEMITE INTL	36.780N	119.719W	333	31.9	34.1	103.7	70.3	101.0	69.0	98.5	68.0	73.4	97.3	71.7	94.9	65.3	94.5	85.7	63.0	87.2	84.0	18.4	16.4	14.5	2216	2166
FULLERTON MUNICIPAL	33.872N	117.979W	96	39.8	42.8	95.0	66.9	91.3	66.8	88.3	66.3	72.3	87.3	70.9	85.0	66.9	99.2	78.9	65.8	95.4	77.8	12.8	10.9	10.1	1113	1352
HAWTHORN MUNICIPAL	33.923N	118.334W	63	44.5	45.8	87.9	62.8	84.0	63.0	81.1	63.5	69.7	79.6	68.6	77.9	66.2	96.7	74.4	64.6	91.3	72.3	16.2	13.9	12.4	1105	819
HAYWARD EXECUTIVE	37.654N	122.115W	43	36.8	38.9	87.1	65.2	82.4	64.1	78.5	63.3	67.6	81.7	66.0	78.9	62.6	85.0	71.6	61.2	80.8	69.3	19.4	17.6	16.1	2542	293
IMPERIAL COUNTY AP	32.834N	115.579W	-58	35.7	37.9	111.3	72.8	109.1	72.4	107.5	72.1	81.0	97.2	79.5	96.3	77.0	140.4	88.3	75.0	131.0	88.9	26.0	22.0	18.7	937	4164

Table 4-8 Design Conditions for Selected Locations *(Continued)*

Meaning of acronyms:
DB: Dry bulb temperature, °F
WB: Wet bulb temperature, °F
MCWB: Mean coincident wet bulb temperature, °F
Lat: Latitude, °
DP: Dew point temperature, °F
MCDB: Mean coincident dry bulb temperature, °F
Long: Longitude, °
HR: Humidity ratio, grains of moisture per lb of dry air
Elev: Elevation, ft
WS: Wind speed, mph
HDD and CDD 65: Annual heating and cooling degree-days, base 65°F, °F-day

Station	Lat	Long	Elev	Heating DB		Cooling DB/MCWB						Evaporation WB/MCDB				Dehumidification DP/HR/MCDB						Extreme Annual WS			Heat./Cool. Degree-Days	
						0.4%		1%		2%		0.4%		1%		0.4%			1%							
				99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		WB / MCDB		WB / MCDB		DP / HR / MCDB			DP / HR / MCDB			1%	2.5%	5%	HDD / CDD 65	
LEMOORE NAS	36.333N	119.950W	232	28.0	31.0	103.4	70.7	100.6	69.7	98.4	68.7	73.8	97.1	71.9	95.8	65.6	95.4	87.4	63.0	86.8	84.7	20.8	18.0	15.6	2296	1836
LIVERMORE MUNICIPAL	37.693N	121.814W	393	30.6	33.1	98.7	67.6	94.8	66.0	90.9	64.8	69.5	93.5	67.8	90.3	60.9	81.0	76.1	59.2	76.1	72.6	19.3	17.8	16.3	2671	836
LOMPOC AP	34.667N	120.467W	88	32.4	35.5	81.7	61.1	77.4	60.9	74.1	60.4	65.3	75.7	63.8	73.7	61.2	81.1	69.4	59.3	75.6	67.8	20.3	18.5	17.1	2809	60
LONG BEACH AP	33.812N	118.146W	31	41.6	43.7	91.4	66.2	87.8	66.3	84.5	65.6	72.0	83.0	70.5	81.0	68.6	105.2	75.9	67.0	99.5	75.0	16.8	14.1	12.3	1188	1098
LOS ANGELES INTL	33.938N	118.389W	97	44.9	46.8	83.8	63.1	80.4	63.9	77.8	64.1	70.0	77.4	68.8	75.7	67.3	100.8	73.8	66.1	96.4	72.7	20.0	17.4	16.0	1287	614
LOS ANGELES ONTARIO INTL	34.056N	117.600W	949	38.4	40.6	100.2	69.7	97.5	68.4	94.6	67.8	73.5	93.1	71.9	91.1	67.6	105.2	80.3	65.7	98.1	78.3	21.7	18.0	15.8	1381	1797
MARCH AFB	33.900N	117.250W	1536	32.1	35.0	101.3	67.9	98.8	66.9	95.9	66.2	72.1	93.7	70.7	91.8	65.8	100.6	78.1	63.8	93.9	75.3	18.0	15.6	13.1	1895	1584
MCCLELLAN-PALOMAR AP	33.128N	117.279W	328	42.9	44.8	83.8	62.0	80.7	63.8	77.9	64.0	70.2	77.1	69.0	75.4	67.5	102.4	74.3	66.2	97.8	73.1	13.5	12.0	10.7	1623	536
MEADOWS FIELD	35.434N	119.054W	489	32.6	35.3	102.9	70.4	100.4	69.2	98.1	68.2	73.4	96.9	71.7	95.4	65.0	94.2	87.4	62.6	86.3	85.1	18.4	15.8	13.0	2050	2319
MINETA SAN JOSE INTL	37.359N	121.924W	51	35.5	37.7	91.4	66.1	87.8	65.3	83.9	64.3	68.9	85.9	67.5	83.0	62.8	85.6	75.6	61.3	81.0	73.9	19.5	17.9	16.2	2143	624
MIRAMAR MCAS	32.867N	117.133W	477	39.1	41.4	90.8	65.9	87.5	65.9	84.1	65.5	71.5	83.4	70.3	81.6	67.7	103.5	76.2	66.2	98.2	74.9	15.3	12.6	11.2	1489	845
MODESTO CITY-COUNTY AP	37.624N	120.951W	73	30.7	33.3	101.4	69.9	98.2	68.4	95.4	67.1	71.8	96.4	70.1	94.2	62.6	85.0	83.4	60.5	78.9	80.6	19.1	17.0	15.4	2362	1631
MOFFETT FEDERAL AIRFIELD	37.406N	122.048W	39	36.0	38.5	88.1	65.5	83.7	64.5	80.5	64.1	68.2	82.5	66.8	79.8	63.0	86.3	73.9	61.4	81.5	71.9	18.8	17.0	15.4	2190	467
MONTEREY REGIONAL	36.588N	121.845W	165	36.6	38.8	78.7	59.9	73.5	59.2	71.1	58.9	63.0	72.0	61.9	70.2	59.4	76.1	65.2	58.0	72.2	64.1	16.8	14.6	12.6	3231	50
MONTGOMERY FIELD	32.816N	117.139W	417	40.7	43.0	90.3	65.4	86.5	64.9	83.3	64.9	71.2	82.8	69.7	80.3	67.2	101.6	76.4	65.9	97.1	74.8	15.7	12.9	11.9	1465	854
NAPA COUNTY AP	38.210N	122.285W	14	29.2	31.8	91.1	65.8	86.5	64.8	82.4	63.9	68.4	86.1	66.7	82.8	61.4	81.3	74.5	60.3	78.1	72.8	21.3	19.1	17.5	3182	242
NORTH ISLAND NAS	32.700N	117.200W	26	44.8	46.2	84.6	64.0	81.3	65.2	78.9	65.8	71.2	77.8	70.0	76.6	69.1	107.0	74.9	67.5	101.1	73.9	18.7	16.5	14.5	1118	783
OAKLAND INTL	37.721N	122.221W	6	36.6	38.9	83.4	64.2	79.1	63.0	75.0	62.2	66.5	78.1	65.1	75.4	62.7	85.1	69.2	61.1	80.6	68.1	23.6	20.1	18.3	2692	164
PALM SPRINGS INTL	33.822N	116.504W	409	41.1	43.5	111.4	70.7	109.2	70.4	107.6	69.9	78.7	98.3	77.1	97.1	73.1	124.7	90.7	71.2	116.9	91.0	23.0	20.0	17.9	757	4394
POINT ARGUELLO	34.580N	120.650W	106	45.7	47.6	71.6	N/A	67.9	N/A	65.5	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	41.6	34.9	31.9	3405	23
POINT MUGU NAS	34.117N	119.117W	13	38.9	41.0	81.5	61.2	78.4	62.4	75.4	62.9	69.5	75.4	67.9	73.9	67.5	101.2	72.3	65.5	94.1	71.4	23.2	19.1	16.6	2078	213
PORTERVILLE MUNICIPAL	36.029N	119.063W	442	30.1	33.5	100.5	70.0	99.2	69.3	97.1	68.1	72.8	96.6	71.1	94.3	63.7	89.8	85.9	62.6	86.2	84.7	12.9	11.4	10.4	2520	1704
REDDING MUNICIPAL	40.518N	122.299W	497	28.5	30.9	105.3	68.4	102.0	67.1	99.1	66.0	71.4	96.5	69.9	94.8	63.5	89.2	79.9	61.3	82.6	78.1	25.0	19.8	16.8	2702	1889
RIVERSIDE MUNICIPAL	33.952N	117.439W	804	36.4	38.7	100.2	69.3	97.7	68.4	94.8	67.4	72.9	93.3	71.4	91.7	66.0	98.8	81.5	63.9	91.7	79.1	19.0	16.1	13.3	1448	1718
SACRAMENTO EXECUTIVE	38.507N	121.495W	15	30.9	33.6	100.1	69.8	96.9	68.4	93.6	67.2	72.3	95.5	70.3	92.6	63.6	88.0	83.5	61.5	81.6	79.4	20.1	17.9	15.8	2491	1222
SACRAMENTO INTL	38.696N	121.590W	23	30.3	33.2	100.5	70.4	97.6	69.3	94.6	68.0	72.9	96.2	71.1	93.7	64.0	89.5	84.4	62.6	85.0	82.6	23.9	19.8	17.4	2496	1364
SACRAMENTO MATHER AP	38.567N	121.300W	99	27.7	30.1	101.2	68.8	98.2	67.4	94.8	66.5	70.9	96.8	69.3	93.6	61.3	81.4	75.4	59.9	77.4	75.8	20.6	17.0	13.7	2771	1190
SACRAMENTO MCCLELLAN AFB	38.667N	121.400W	77	30.1	33.0	102.1	70.0	99.3	68.7	95.8	67.4	72.2	97.7	70.5	95.0	63.2	87.0	79.6	61.3	81.2	78.7	20.8	17.6	14.9	2323	1588
SALINAS MUNICIPAL	36.664N	121.608W	74	33.9	36.4	83.2	62.1	78.6	61.0	75.0	60.7	65.0	77.5	63.6	74.5	60.6	79.2	67.6	59.2	75.4	66.4	20.7	18.6	17.0	2707	112
SAN BERNARDINO INTL	34.095N	117.235W	1159	33.9	36.5	102.9	69.7	100.2	69.5	97.4	68.8	74.5	95.1	73.0	93.6	68.1	107.7	83.1	66.1	100.5	83.3	16.7	12.9	10.9	1652	1811
SAN DIEGO INTL	32.734N	117.183W	15	45.0	46.9	83.7	64.7	80.6	65.5	78.3	65.6	71.1	77.6	69.9	76.2	68.5	104.8	74.6	67.4	100.9	73.8	16.9	15.1	12.9	1171	715
SAN FRANCISCO INTL	37.620N	122.365W	8	39.5	41.6	82.6	62.7	77.9	62.0	74.3	61.4	65.6	77.2	64.2	74.3	61.4	81.3	68.4	60.1	77.7	67.2	28.5	25.5	23.3	2679	157
SAN LUIS OBISPO CO REGIONAL	35.237N	120.641W	200	34.0	36.3	89.1	64.0	84.3	63.2	81.2	62.8	67.3	82.8	65.8	80.2	61.5	82.2	71.3	60.5	79.2	70.1	25.0	22.3	19.6	2223	294
SANTA BARBARA MUNICIPAL	34.426N	119.843W	9	35.0	37.0	82.9	63.3	79.7	63.7	77.0	63.1	68.4	77.0	67.0	75.4	64.9	92.1	71.5	63.7	88.3	70.1	18.9	16.3	13.1	2224	218
SANTA MARIA PUBLIC AP	34.899N	120.449W	242	33.2	35.5	84.2	61.9	80.0	61.3	76.6	60.8	66.0	78.0	64.4	75.7	61.4	82.0	69.0	60.0	77.8	67.9	24.3	20.5	18.3	2668	115
SONOMA COUNTY AP	38.504N	122.810W	114	28.9	31.2	94.7	66.4	90.9	65.7	87.2	64.4	69.1	89.9	67.4	87.1	61.0	80.4	75.7	59.2	75.4	73.3	17.1	15.0	12.6	2986	366
SOUTHERN CALIFORNIA LOGISTICS	34.583N	117.383W	2885	27.6	30.5	100.7	65.3	98.4	64.7	96.0	63.9	69.8	88.4	68.4	88.2	64.5	101.2	76.9	61.9	92.1	78.6	22.7	18.9	16.7	2661	1911
STOCKTON METROPOLITAN	37.889N	121.226W	26	30.3	32.7	101.1	70.0	97.9	68.8	94.8	67.9	73.2	96.1	70.8	94.0	64.6	91.3	84.8	61.6	82.0	80.2	22.7	19.3	17.3	2448	1393
TRAVIS AFB	38.267N	121.933W	62	30.0	33.0	99.0	67.7	95.1	66.6	91.1	65.6	70.4	93.2	68.6	90.2	62.7	85.3	73.6	60.8	79.8	72.2	28.8	26.8	24.9	2516	993
VISALIA MUNICIPAL	36.317N	119.400W	295	29.8	32.4	100.0	71.8	98.7	70.9	96.5	69.9	74.7	95.4	73.0	93.4	67.8	103.4	85.2	65.5	95.3	84.5	15.4	12.5	10.9	2509	1641
WILLIAM J FOX AP	34.741N	118.212W	2338	21.3	24.7	102.8	65.8	100.3	64.4	98.0	63.5	68.3	96.3	66.9	95.0	58.0	78.4	80.6	55.2	70.6	80.5	29.9	27.0	24.9	2937	1890
Colorado																									*10 sites, 30 more on CD-ROM*	
BUCKLEY AFB	39.717N	104.750W	5663	0.4	7.2	93.2	58.4	90.6	58.4	88.0	58.3	63.9	77.8	62.5	77.4	60.8	98.5	65.8	58.8	91.6	65.4	24.1	20.1	17.4	5798	687
CENTENNIAL AP	39.570N	104.849W	5883	-0.1	5.8	91.9	59.5	89.9	59.2	87.2	58.8	64.8	80.3	63.2	78.8	60.9	99.7	68.1	58.7	91.9	67.5	24.7	20.9	18.4	6052	628
COLORADO SPRINGS MUNICIPAL	38.810N	104.688W	6181	1.4	7.0	90.9	58.7	88.3	58.4	85.7	58.3	63.4	78.5	62.3	77.4	59.5	95.7	65.9	58.2	91.3	65.5	28.2	24.8	21.0	6117	505
DENVER INTL	39.833N	104.658W	5414	-0.8	5.3	94.8	59.7	92.2	59.7	89.5	59.5	64.8	80.9	63.7	80.5	60.7	97.1	67.9	59.1	91.6	67.6	27.1	23.7	20.0	5935	812
DENVER STAPLETON	39.750N	104.867W	5288	-1.4	5.1	93.9	60.7	91.2	60.0	88.5	59.6	64.5	81.8	63.4	80.7	60.1	94.7	67.2	58.5	89.1	67.0	24.3	19.7	17.2	5667	721
FORT COLLINS DOWNTOWN	40.588N	105.042W	4939	-2.6	4.8	90.1	60.9	87.2	60.4	84.4	60.1	64.6	80.7	63.6	80.0	59.7	92.1	69.7	58.4	87.9	69.2	20.0	16.8	13.6	6096	462
FORT COLLINS LOVELAND MUNI	40.450N	105.017W	5015	-0.2	5.5	93.5	60.9	90.9	60.8	88.3	60.5	65.4	82.5	64.2	82.0	61.0	96.7	69.9	58.6	88.5	69.6	25.9	21.4	18.0	6196	624
GRAND JUNCTION REGIONAL	39.134N	108.540W	4858	3.7	9.3	97.7	61.2	95.2	60.3	92.7	59.7	65.1	84.9	64.0	83.8	60.7	95.1	68.0	58.7	88.3	68.5	23.6	19.5	17.0	5481	1228
GREELEY-WELD COUNTY AP	40.436N	104.632W	4697	-7.8	0.0	96.6	62.6	92.6	62.2	90.0	62.1	67.2	84.7	65.9	84.1	62.6	101.1	73.2	60.8	94.9	72.1	28.2	24.0	19.4	6579	643
PUEBLO MEMORIAL	38.290N	104.498W	4720	0.0	6.5	98.5	62.2	95.8	61.9	93.1	61.6	66.8	84.9	65.8	84.1	62.7	101.9	69.0	61.3	96.6	69.0	28.7	24.7	20.4	5488	951
Connecticut																									*5 sites, 5 more on CD-ROM*	
BRADLEY INTL	41.938N	72.683W	190	4.8	9.8	91.4	73.2	88.4	71.8	85.6	70.3	76.2	86.5	74.7	83.8	73.2	124.1	80.2	71.9	118.9	79.0	22.7	19.3	17.4	5843	789
HARTFORD-BRAINARD AP	41.736N	72.651W	19	8.2	12.2	90.9	73.1	88.2	72.2	85.1	70.7	76.6	85.7	75.2	83.3	73.7	125.7	80.2	72.6	120.9	79.2	19.2	17.5	15.7	5500	863
IGOR SIKORSKY MEMORIAL	41.158N	73.129W	5	11.4	15.6	87.9	73.1	84.9	71.7	82.5	70.6	76.1	83.2	74.9	81.0	73.8	126.1	79.5	72.8	121.6	78.3	24.3	20.7	18.5	5230	858
WATERBURY-OXFORD AP	41.483N	73.133W	726	3.2	8.8	87.6	72.9	83.7	71.1	81.3	69.5	75.4	83.2	73.7	80.4	72.9	125.4	78.9	72.0	121.7	77.8	19.5	17.0	14.9	6416	468
WINDHAM AP	41.742N	72.184W	247	3.6	9.3	89.7	73.2	86.5	71.9	83.8	70.4	76.1	84.9	74.6	82.2	73.2	124.6	79.5	72.3	120.8	78.5	19.2	17.0	15.3	5950	642

Table 4-8 Design Conditions for Selected Locations *(Continued)*

Meaning of acronyms:
DB: Dry bulb temperature, °F
WB: Wet bulb temperature, °F
MCWB: Mean coincident wet bulb temperature, °F
Lat: Latitude, °
DP: Dew point temperature, °F
MCDB: Mean coincident dry bulb temperature, °F
Long: Longitude, °
HR: Humidity ratio, grains of moisture per lb of dry air
HDD and CDD 65: Annual heating and cooling degree-days, base 65°F, °F-day
Elev: Elevation, ft
WS: Wind speed, mph

Station	Lat	Long	Elev	Heating DB		Cooling DB/MCWB						Evaporation WB/MCDB				Dehumidification DP/HR/MCDB						Extreme Annual WS			Heat./Cool. Degree-Days	
						0.4%		1%		2%		0.4%		1%		0.4%			1%							
				99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		WB / MCDB		WB / MCDB		DP / HR / MCDB			DP / HR / MCDB			1%	2.5%	5%	HDD / CDD 65	
Delaware																						*2 sites, 3 more on CD-ROM*				
DOVER AFB	39.133N	75.467W	28	14.7	18.7	92.5	75.7	90.0	74.8	87.3	74.0	78.7	87.0	77.5	85.5	76.7	139.4	82.3	75.2	132.4	80.9	24.7	20.9	18.7	4461	1194
NEW CASTLE AP	39.673N	75.601W	79	13.8	17.7	92.3	75.1	89.4	73.8	86.8	72.9	78.0	87.5	76.7	85.2	75.4	133.4	81.9	74.2	128.1	80.6	24.7	20.8	18.6	4684	1160
Florida																						*32 sites, 45 more on CD-ROM*				
CECIL FIELD	30.219N	81.876W	81	30.3	34.0	96.1	76.7	93.9	76.4	92.0	75.9	79.6	89.6	78.6	88.5	77.2	142.1	82.7	76.2	137.1	82.1	19.0	16.9	15.0	1164	2730
CRAIG MUNICIPAL	30.336N	81.515W	41	32.7	36.1	94.3	77.0	92.2	76.8	90.4	76.6	80.0	88.8	79.1	87.8	77.6	143.8	83.5	76.9	140.2	82.9	19.0	17.4	15.8	1158	2745
DAYTONA BEACH INTL	29.183N	81.048W	31	35.5	39.6	92.7	76.9	90.9	76.8	89.3	76.7	80.0	88.1	79.1	87.1	77.6	143.7	83.6	76.9	140.3	83.0	20.3	18.0	16.2	728	3020
FT LAUDERDALE HOLLYWOOD INTL	26.072N	80.154W	11	47.8	52.0	91.8	78.3	90.6	78.3	89.8	78.2	81.1	87.9	80.4	87.2	79.3	151.9	84.7	78.6	148.4	84.3	21.9	19.6	18.1	124	4609
GAINESVILLE REGIONAL	29.692N	82.276W	123	29.6	33.2	93.8	75.9	92.2	75.8	90.6	75.5	79.5	87.9	78.5	86.8	77.3	142.8	82.6	76.4	138.5	81.9	18.3	16.3	14.2	1148	2670
HOMESTEAD AFB	25.483N	80.383W	5	46.0	50.2	91.2	79.2	90.4	79.0	89.6	78.9	81.4	87.5	80.7	87.0	79.4	152.8	84.0	79.1	150.9	83.8	20.1	18.1	16.3	144	4228
JACKSONVILLE INTL	30.495N	81.694W	26	29.5	32.8	94.5	77.2	92.7	76.9	90.9	76.5	79.9	89.4	79.1	88.2	77.4	142.7	83.3	76.7	139.3	82.6	20.1	17.9	16.1	1303	2629
JACKSONVILLE NAS	30.233N	81.667W	20	34.2	37.5	95.8	76.8	93.7	76.3	91.9	76.0	80.4	88.4	79.4	87.8	78.7	149.1	83.7	77.3	142.0	83.1	20.9	18.6	16.7	946	3259
KENNEDY SPACE CENTER	28.617N	80.683W	10	39.2	43.1	91.7	78.1	90.4	78.0	89.3	77.8	80.7	87.5	79.9	86.6	79.2	151.5	83.6	78.3	147.2	83.2	18.7	16.7	14.7	525	3176
MACDILL AFB	27.850N	82.517W	14	38.8	43.0	93.3	78.4	92.2	78.1	91.0	77.8	82.2	88.7	81.3	87.9	80.6	159.1	85.1	79.5	152.9	84.6	18.9	16.9	15.4	491	3653
MAYPORT NAF	30.400N	81.417W	16	35.2	39.0	93.9	77.2	91.6	77.1	89.9	76.8	80.5	88.4	79.5	87.6	78.7	148.8	85.0	77.2	141.7	84.1	21.4	18.7	16.7	1007	2961
MELBOURNE INTL	28.101N	80.644W	27	38.6	43.0	91.9	77.5	90.4	77.5	89.5	77.5	80.3	87.8	79.5	87.1	78.5	148.2	84.0	77.3	141.9	83.5	20.9	18.9	17.6	462	3473
MIAMI EXECUTIVE	25.648N	80.433W	10	45.6	49.6	92.8	77.9	91.4	77.7	90.5	77.6	80.4	87.9	79.8	87.4	78.9	149.9	83.2	77.5	143.1	83.0	20.7	18.8	17.5	164	4147
MIAMI INTL	25.791N	80.316W	29	48.8	52.6	91.9	77.6	90.8	77.6	89.8	77.5	80.3	86.9	79.7	86.8	78.4	147.6	83.5	77.6	143.7	83.3	20.0	18.3	16.7	113	4578
NAPLES MUNICIPAL	26.152N	81.775W	9	43.6	47.5	91.6	77.7	90.5	77.8	89.7	77.8	80.9	87.4	80.1	87.0	79.1	151.3	84.2	78.2	146.5	83.7	18.9	17.0	15.2	273	3807
ORLANDO EXECUTIVE	28.545N	81.333W	108	38.9	43.2	93.5	76.1	92.4	75.9	91.0	75.7	79.7	86.6	78.8	86.0	78.4	148.3	82.4	77.1	141.6	81.6	19.4	17.6	15.9	505	3543
ORLANDO INTL	28.434N	81.325W	90	38.3	42.3	93.8	76.5	92.4	76.2	91.1	75.9	79.6	87.3	78.8	86.5	77.7	144.6	81.7	77.1	141.3	81.2	20.3	18.2	16.5	529	3405
ORLANDO SANFORD INTL	28.780N	81.244W	55	36.9	40.9	94.4	75.7	92.9	75.5	91.2	75.4	79.0	88.0	78.3	87.1	76.9	140.5	82.0	76.0	136.0	81.5	20.1	17.9	16.1	617	3342
PAGE FIELD	26.585N	81.861W	15	42.7	46.6	93.5	76.7	92.4	76.7	91.2	76.7	80.2	88.0	79.4	87.3	78.2	146.6	83.1	77.4	142.4	82.6	18.6	16.8	15.0	267	3943
PALM BEACH INTL	26.685N	80.099W	19	44.5	48.5	91.7	77.7	90.4	77.7	89.3	77.7	80.2	87.8	79.5	87.2	78.0	145.7	83.6	77.3	142.4	83.4	22.6	19.9	18.4	208	4138
PANAMA CITY BAY COUNTY INTL	30.212N	85.683W	21	31.8	35.7	92.8	76.8	91.1	76.9	90.1	76.7	81.4	87.0	80.2	86.4	79.5	153.1	83.9	78.9	150.0	83.7	18.7	16.7	15.0	1242	2847
PENSACOLA INTL	30.478N	87.187W	112	30.0	33.8	93.9	77.5	92.0	77.3	90.3	77.0	81.0	88.4	80.0	87.4	79.1	151.8	84.4	78.1	146.4	83.6	20.1	18.1	16.4	1422	2706
PENSACOLA NAS	30.350N	87.317W	28	29.6	33.1	93.1	78.7	91.3	78.5	90.0	78.1	81.9	88.5	80.8	87.7	80.1	156.2	85.5	79.0	150.6	84.8	20.6	18.4	16.6	1474	2618
SARASOTA BRADENTON INTL	27.401N	82.559W	28	39.6	44.1	92.4	78.5	91.1	78.4	90.3	78.2	82.6	88.6	81.4	87.5	81.2	162.2	86.5	79.5	153.2	85.0	20.7	18.4	16.6	456	3477
SOUTHWEST FLORIDA INTL	26.536N	81.755W	31	41.3	45.4	93.4	76.6	92.2	76.6	91.0	76.5	80.2	87.4	79.4	86.8	78.8	149.8	82.7	77.4	142.6	82.3	20.3	18.1	16.1	306	3743
ST PETE-CLEARWATER INTL	27.911N	82.688W	11	42.2	45.4	92.2	77.6	91.0	77.6	90.2	77.4	81.4	86.9	80.4	86.4	80.0	155.7	84.2	79.0	150.5	83.6	20.9	18.8	17.1	443	3672
TALLAHASSEE REGIONAL	30.393N	84.353W	55	26.2	29.7	96.2	76.1	94.2	75.6	92.4	75.3	79.7	89.0	78.8	87.8	77.4	142.9	82.8	76.5	138.4	82.1	18.2	16.1	13.7	1493	2662
TAMPA INTL	27.962N	82.540W	19	39.6	43.4	92.5	77.0	91.3	77.0	90.3	77.0	80.4	87.9	79.8	87.5	78.4	147.5	84.9	77.5	142.9	84.1	18.1	16.0	13.6	505	3609
TAYLOR FIELD	29.167N	82.233W	87	29.6	33.9	93.2	75.2	91.4	75.3	90.6	75.1	78.6	87.8	78.0	87.4	75.7	134.8	82.4	75.2	132.4	82.1	18.0	15.3	12.6	1043	2756
TYNDALL AFB	30.067N	85.583W	17	31.5	35.5	91.3	78.8	90.2	78.8	89.0	78.5	82.5	87.4	81.5	86.8	81.3	162.5	85.3	80.0	155.8	84.9	20.5	18.3	16.4	1324	2644
VENICE	27.070N	82.450W	0	41.8	45.8	88.1	76.4	86.9	77.0	86.2	77.2	82.1	83.7	80.9	83.6	81.8	165.4	83.0	80.2	156.9	82.4	28.0	23.8	19.8	477	3009
VERO BEACH REGIONAL	27.651N	80.420W	28	38.8	43.2	91.8	77.6	90.5	77.7	89.4	77.7	80.3	88.0	79.7	87.3	78.4	147.6	84.2	77.4	142.5	83.7	20.4	18.5	17.0	414	3460
Georgia																						*19 sites, 27 more on CD-ROM*				
ATHENS BEN EPPS AP	33.948N	83.328W	785	22.6	26.5	95.4	74.6	93.1	74.0	90.7	73.6	77.7	88.9	76.8	87.4	74.7	133.9	81.8	73.8	129.5	80.6	18.2	16.1	13.6	2755	1804
ATLANTA HARTSFIELD-JACKSON	33.640N	84.430W	1027	21.9	26.5	94.0	74.2	91.6	73.8	89.5	73.3	77.3	88.3	76.3	86.5	74.3	133.3	81.3	73.4	128.9	80.3	21.3	18.9	17.0	2640	1901
ATLANTA REGIONAL	33.355N	84.567W	798	19.4	23.4	93.4	73.8	91.3	73.6	89.8	73.4	77.5	87.4	76.5	86.0	74.9	134.9	81.2	73.5	128.2	79.6	17.3	15.0	12.3	3002	1578
AUGUSTA REGIONAL	33.364N	81.963W	132	22.6	26.1	97.3	76.0	94.9	75.6	92.6	75.2	79.4	90.7	78.3	89.1	76.5	139.1	83.4	75.5	134.2	82.4	18.8	16.5	14.1	2384	2074
COLUMBUS AP	32.516N	84.942W	392	25.9	29.6	96.4	74.6	94.2	74.4	92.3	74.1	78.2	89.3	77.3	88.0	75.3	134.6	82.1	74.4	130.6	81.2	18.4	16.3	14.1	2047	2359
DANIEL FIELD	33.467N	82.039W	423	27.2	29.9	96.8	74.4	93.9	73.7	91.8	73.4	77.5	89.1	76.9	87.9	74.8	132.2	80.8	73.4	126.4	79.8	16.6	14.5	12.5	2106	2329
DEKALB-PEACHTREE AP	33.875N	84.302W	1002	21.1	25.5	94.2	73.6	91.8	73.3	90.2	72.8	76.9	88.0	75.9	86.4	73.4	129.1	79.3	73.0	127.0	79.0	18.6	16.5	14.0	2867	1816
DOBBINS AFB	33.917N	84.517W	1068	19.5	24.5	93.1	74.4	91.1	74.2	89.2	73.6	77.3	87.7	76.3	86.3	74.4	133.9	81.2	73.4	129.3	80.2	18.9	16.7	14.5	2924	1757
FULTON COUNTY AP	33.779N	84.521W	840	21.2	25.7	94.1	74.4	91.9	74.0	90.1	73.5	77.4	88.4	76.5	86.9	74.4	132.7	81.6	73.3	127.9	80.6	17.6	15.4	13.1	2807	1765
HUNTER AAF	32.017N	81.133W	41	27.9	31.7	95.6	77.3	93.3	76.9	91.2	76.6	81.1	88.3	79.9	88.0	79.3	152.0	83.6	77.5	143.2	82.9	19.2	16.8	14.7	1610	2579
LAWSON AAF	32.332N	84.989W	232	22.5	25.9	96.7	75.9	94.6	75.8	92.4	75.5	80.3	89.6	79.0	89.0	77.9	146.4	83.7	76.6	139.8	82.6	17.0	14.5	12.3	2312	2074
LEE GILMER MEMORIAL	34.272N	83.830W	1275	21.3	26.1	92.5	73.3	90.5	73.1	88.4	72.5	76.4	86.4	75.4	85.0	73.4	130.3	79.6	72.7	127.1	79.1	19.0	16.9	15.2	3006	1643
MIDDLE GEORGIA REGIONAL	32.685N	83.653W	343	23.9	27.4	96.6	75.3	94.4	75.1	92.3	74.7	79.0	90.0	78.0	88.7	76.2	138.5	83.0	75.3	134.1	82.0	18.2	16.1	13.5	2265	2153
MOODY AFB	30.967N	83.200W	233	29.1	32.6	96.1	76.5	94.2	76.2	92.6	75.8	80.1	90.6	78.9	89.4	77.4	143.7	83.8	76.4	138.8	82.9	17.8	15.3	12.8	1433	2667
RICHARD B RUSSELL REGIONAL	34.348N	85.161W	639	19.4	23.4	95.8	74.5	93.2	73.9	91.1	73.8	78.0	89.6	77.1	88.3	75.0	134.4	82.0	73.7	128.7	81.1	16.2	13.3	11.8	3030	1777
ROBINS AFB	32.633N	83.600W	294	24.8	28.0	97.0	75.8	94.9	75.8	92.7	75.2	79.7	90.4	78.4	88.9	77.1	142.7	83.5	75.7	135.8	82.4	19.4	16.8	14.3	2135	2228
SAVANNAH HILTON HEAD INTL	32.130N	81.210W	46	27.6	30.8	95.5	77.1	93.3	76.8	91.3	76.2	80.2	89.5	79.2	88.3	78.0	145.6	83.5	77.0	140.9	82.6	18.9	16.8	15.2	1723	2465

Table 4-8 Design Conditions for Selected Locations *(Continued)*

Meaning of acronyms:
DB: Dry bulb temperature, °F *WB: Wet bulb temperature, °F*
MCWB: Mean coincident wet bulb temperature, °F
Lat: Latitude, °
DP: Dew point temperature, °F
MCDB: Mean coincident dry bulb temperature, °F
Long: Longitude, °
HR: Humidity ratio, grains of moisture per lb of dry air
HDD and CDD 65: Annual heating and cooling degree-days, base 65°F, °F-day
Elev: Elevation, ft
WS: Wind speed, mph

Station	Lat	Long	Elev	Heating DB		Cooling DB/MCWB						Evaporation WB/MCDB				Dehumidification DP/HR/MCDB						Extreme Annual WS			Heat./Cool. Degree-Days	
						0.4%		1%		2%		0.4%		1%		0.4%			1%							
				99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		WB / MCDB		WB / MCDB		DP / HR / MCDB			DP / HR / MCDB			1%	2.5%	5%	HDD / CDD 65	
SW GEORGIA REGIONAL	31.536N	84.194W	190	26.6	29.6	96.8	75.9	94.7	75.7	92.7	75.4	79.7	90.4	78.5	88.8	77.2	142.4	83.1	75.9	136.4	82.2	18.4	16.4	14.2	1746	2547
VALDOSTA REGIONAL	30.783N	83.277W	198	27.7	30.8	96.6	76.6	94.5	76.2	92.7	75.8	80.2	90.0	79.2	88.8	77.6	144.6	83.1	76.9	141.1	82.5	16.5	14.0	12.4	1477	2627
Hawaii																										*4 sites, 8 more on CD-ROM*
HILO INTL	19.719N	155.053W	38	61.6	62.8	85.7	74.0	84.7	73.7	83.9	73.4	76.5	82.0	75.8	81.5	75.0	131.5	79.2	74.0	127.2	78.5	16.7	14.9	12.8	0	3245
HONOLULU INTL	21.324N	157.929W	7	62.5	64.5	89.4	73.8	88.5	73.4	87.7	73.0	77.1	84.5	76.2	83.8	74.9	130.9	81.0	73.7	125.6	80.3	22.5	20.4	18.9	0	4656
KALAELOA	21.317N	158.067W	33	60.4	62.5	90.1	73.4	88.8	73.2	87.9	73.0	77.4	84.9	76.3	84.2	75.1	131.9	81.5	73.5	124.7	80.5	18.2	16.4	14.7	0	4214
KANEOHE MCAS	21.450N	157.768W	24	64.0	65.9	84.9	74.3	84.1	74.1	83.4	73.8	77.0	81.6	76.1	81.3	75.3	132.9	79.8	74.3	128.5	79.5	18.5	16.7	15.4	0	4190
Idaho																										*7 sites, 14 more on CD-ROM*
BOISE AP	43.567N	116.241W	2814	9.4	15.9	98.6	63.8	95.7	62.8	92.8	61.9	66.1	92.0	64.7	90.2	57.5	78.3	71.5	55.1	71.8	71.6	22.0	19.1	17.1	5414	1007
CALDWELL INDUSTRIAL AP	43.650N	116.633W	2429	9.6	15.7	97.0	66.2	93.1	64.7	90.6	63.8	68.2	92.3	66.4	90.0	59.1	82.0	78.5	56.7	75.0	77.6	22.1	19.2	17.0	5739	692
COEUR D'ALENE AP	47.767N	116.817W	2307	5.8	10.4	91.3	63.1	88.5	62.6	84.2	61.1	66.3	85.6	64.3	83.4	59.2	81.7	72.0	56.8	75.0	70.6	22.3	18.9	16.8	6875	316
IDAHO FALLS REGIONAL	43.516N	112.067W	4729	-6.6	-0.3	92.1	60.9	89.6	60.5	86.7	59.6	64.7	83.2	63.0	81.9	58.7	87.9	71.0	56.2	80.3	68.9	27.2	24.3	20.7	7672	288
LEWISTON-NEZ PERCE CO REGL	46.375N	117.016W	1436	13.0	18.8	98.5	65.3	95.0	64.5	91.4	63.2	67.8	91.9	66.1	89.8	60.0	81.6	72.7	57.7	75.1	72.0	20.9	17.9	15.1	5044	868
MAGIC VALLEY REGIONAL	42.482N	114.487W	4151	7.6	12.0	95.0	62.6	92.2	62.1	89.8	61.6	66.3	88.7	64.7	86.1	58.8	86.4	74.6	56.6	79.6	74.2	27.9	24.6	20.9	6029	775
POCATELLO REGIONAL	42.920N	112.571W	4452	-2.0	3.8	94.9	61.4	91.8	60.8	88.8	60.0	65.2	86.3	63.5	84.5	58.7	86.9	70.7	55.8	78.2	70.5	28.6	25.5	22.6	6941	440
Illinois																										*14 sites, 48 more on CD-ROM*
ABRAHAM LINCOLN CAPITAL	39.845N	89.684W	594	1.1	6.9	92.6	76.7	90.4	75.6	88.0	74.1	79.5	89.5	77.9	87.3	76.5	141.4	86.2	75.0	134.3	84.3	24.7	21.4	19.1	5328	1144
AURORA MUNICIPAL	41.770N	88.481W	710	-4.9	0.7	90.7	74.1	88.2	73.2	85.5	72.0	77.7	87.1	76.0	84.3	74.8	134.0	83.4	73.1	126.2	81.3	26.0	23.0	19.9	6537	729
CHICAGO MIDWAY INTL	41.786N	87.752W	612	0.5	6.1	91.9	74.7	89.4	73.1	86.6	72.0	77.8	87.8	76.0	85.3	74.9	133.6	84.0	72.9	125.0	82.0	24.4	21.0	19.1	5850	1057
CHICAGO O'HARE INTL	41.995N	87.934W	662	-1.0	4.4	91.3	74.2	88.5	72.9	85.9	71.6	77.5	87.3	75.7	84.7	74.5	132.2	83.5	72.8	124.7	81.4	24.6	21.0	19.1	6190	882
CHICAGO ROCKFORD INTL	42.193N	89.093W	730	-5.4	0.3	90.9	74.4	88.0	72.9	85.4	71.6	77.8	87.0	75.8	84.0	75.0	134.8	83.3	73.2	126.5	81.7	24.5	20.9	18.9	6589	786
DECATUR AP	39.834N	88.866W	675	1.9	7.3	93.1	76.6	90.7	75.5	88.3	74.3	79.4	89.4	77.8	87.5	76.4	141.5	86.0	74.9	134.0	84.1	24.7	21.5	19.6	5388	1105
DUPAGE COUNT AP	41.914N	88.246W	754	-2.5	2.6	90.2	74.6	87.8	73.5	84.6	71.8	77.9	86.8	76.0	84.0	75.0	135.1	83.8	73.1	126.6	81.5	24.8	21.6	19.3	6430	750
GLENVIEW NAS	42.083N	87.817W	653	-0.7	4.8	93.7	75.0	90.2	73.3	87.1	72.1	77.9	90.2	76.2	87.0	74.2	130.7	85.1	72.4	123.1	83.6	20.2	18.0	16.2	6104	909
GREATER PEORIA REGIONAL	40.668N	89.684W	650	-0.9	4.3	92.0	76.5	89.6	75.2	87.0	73.6	79.3	88.3	77.5	86.5	76.6	142.1	85.3	74.8	133.7	83.2	23.1	19.7	17.8	5733	1057
QUAD CITY INTL	41.465N	90.523W	592	-3.5	1.8	92.4	76.2	89.7	74.9	87.1	73.2	79.1	88.8	77.4	86.5	76.3	140.5	85.3	74.6	132.4	83.2	23.9	20.2	18.2	6091	988
QUINCY REGIONAL	39.937N	91.192W	769	0.4	5.6	93.1	76.5	90.3	75.4	87.7	74.1	78.9	89.1	77.5	87.1	76.0	139.6	85.3	74.4	132.3	83.6	24.3	20.7	18.7	5497	1119
SCOTT AFB	38.550N	89.850W	459	7.2	12.1	95.2	77.7	92.7	76.9	90.3	75.9	80.7	90.2	79.2	88.6	78.1	148.5	85.8	76.7	141.4	84.5	22.7	19.6	17.5	4617	1413
ST LOUIS DOWNTOWN AP	38.571N	90.157W	413	8.6	12.7	95.4	76.6	92.7	76.3	90.4	75.2	79.9	90.5	78.3	88.9	77.0	142.9	85.4	75.1	133.9	83.9	20.8	18.6	16.6	4569	1432
U OF ILLINOIS WILLARD AP	40.040N	88.278W	754	-0.4	4.9	91.4	75.5	89.6	74.8	86.9	73.4	79.2	87.8	77.4	85.9	76.7	143.1	85.1	74.8	134.2	82.7	27.5	24.6	21.4	5693	973
Indiana																										*8 sites, 9 more on CD-ROM*
EVANSVILLE REGIONAL	38.044N	87.521W	400	9.0	14.6	94.0	76.1	91.7	75.6	89.5	74.7	79.2	89.9	78.0	88.3	76.2	138.7	85.1	75.0	133.0	83.7	20.5	18.2	16.3	4388	1452
FORT WAYNE INTL	40.971N	85.206W	791	-0.5	5.4	90.8	74.5	88.1	73.0	85.5	71.7	77.6	87.0	75.8	84.1	74.7	133.8	83.0	73.1	126.8	81.1	25.1	21.8	19.2	5978	826
GRISSOM AFB	40.650N	86.150W	812	-1.4	5.5	90.6	75.4	88.0	74.0	85.6	72.8	79.4	85.9	77.4	84.2	77.8	148.7	83.1	75.5	137.5	81.0	25.9	22.6	19.5	5868	899
INDIANAPOLIS INTL	39.732N	86.279W	790	2.6	8.6	91.3	75.0	88.9	74.0	86.5	72.8	78.1	87.5	76.6	85.3	75.3	136.7	83.7	73.9	130.2	82.0	24.9	21.4	19.0	5249	1106
MONROE COUNTY AP	39.133N	86.617W	844	3.9	9.8	91.4	74.9	89.6	74.9	87.1	73.4	78.4	86.9	77.2	85.9	75.5	138.0	83.3	74.5	133.1	82.2	19.5	17.3	15.6	5071	1026
PURDUE UNIVERSITY AP	40.412N	86.937W	599	0.6	6.3	91.6	75.6	89.5	74.3	86.9	72.8	78.5	87.9	76.9	85.7	75.7	137.7	84.5	74.1	130.2	82.3	22.7	19.8	18.0	5526	998
SOUTH BEND INTL	41.707N	86.316W	773	0.4	6.3	90.4	74.0	87.7	72.4	85.0	71.1	77.2	86.1	75.3	83.7	74.5	132.6	82.8	72.7	124.6	80.5	24.0	20.5	18.5	6160	799
TERRE HAUTE INTL	39.452N	87.309W	575	2.2	8.8	92.5	76.2	90.1	75.6	87.8	74.2	79.2	88.6	77.7	86.8	76.5	141.2	85.1	74.9	133.5	83.2	22.5	19.3	17.7	5170	1084
Iowa																										*9 sites, 46 more on CD-ROM*
AMES MUNICIPAL	41.991N	93.619W	955	-6.4	-1.5	90.9	76.2	88.3	74.7	85.8	73.1	79.1	87.3	77.2	85.4	76.7	144.1	84.9	74.7	134.7	82.6	26.3	23.5	20.2	6625	808
ANKENY REGIONAL	41.691N	93.566W	910	-3.7	1.0	93.0	75.6	90.3	75.1	87.8	73.7	79.3	88.1	77.4	86.8	76.9	145.1	84.8	74.5	133.6	83.2	22.9	19.6	17.7	6125	975
BOONE MUNICIPAL	42.049N	93.848W	1160	-5.7	0.2	91.2	76.5	89.5	75.8	86.1	73.5	80.2	87.9	78.1	85.2	77.5	149.6	85.4	75.4	139.1	83.0	26.2	23.4	20.3	6484	878
DAVENPORT MUNICIPAL	41.614N	90.591W	753	-6.9	-1.4	91.1	75.0	88.2	73.9	85.8	72.6	77.9	87.7	76.4	85.6	75.0	135.0	84.3	73.3	127.2	82.8	26.7	24.0	20.6	6474	878
DES MOINES INTL	41.534N	93.653W	957	-4.3	0.6	92.8	76.2	89.7	75.0	87.1	73.4	78.7	88.8	77.2	86.6	75.9	140.2	85.3	74.3	132.6	83.7	25.5	22.3	19.5	6147	1066
DUBUQUE REGIONAL	42.398N	90.704W	1056	-8.1	-2.7	88.4	75.0	85.6	73.3	83.1	71.4	77.5	85.3	75.6	82.6	75.2	137.6	82.5	73.3	128.7	80.5	25.8	22.8	19.9	7042	641
SIOUX GATEWAY AP	42.391N	96.379W	1095	-7.1	-2.5	92.9	75.2	90.1	74.3	87.4	73.0	78.8	88.3	77.2	86.7	76.2	142.3	85.2	74.2	133.1	83.6	28.5	25.1	22.0	6733	914
THE EASTERN IOWA AP	41.883N	91.717W	868	-8.2	-2.6	90.4	76.3	87.6	74.5	84.8	72.7	78.6	86.7	76.9	84.6	76.3	141.9	84.1	74.5	133.2	82.0	26.7	23.7	20.5	6755	774
WATERLOO MUNICIPAL	42.554N	92.401W	868	-9.5	-4.5	91.0	75.5	88.1	73.7	85.5	72.3	78.5	87.0	76.6	84.6	76.1	140.9	84.4	74.1	131.5	82.0	26.1	23.4	20.2	7016	769
Kansas																										*10 sites, 22 more on CD-ROM*
COLONEL JAMES JABARA AP	37.746N	97.221W	1421	6.7	11.0	100.0	73.6	97.2	73.9	93.2	73.6	77.4	91.0	76.4	90.0	73.4	130.8	83.5	72.5	126.9	82.4	27.8	25.1	22.8	4504	1632
FORBES FIELD	38.950N	95.664W	1067	3.5	8.9	99.0	76.0	94.8	75.4	91.4	74.6	78.9	91.8	77.7	90.4	75.2	137.7	85.8	73.6	130.0	84.0	26.2	23.6	20.5	4977	1422
JOHNSON COUNTY EXECUTIVE	38.850N	94.739W	1070	4.3	9.2	96.5	75.5	92.7	75.3	90.1	74.9	78.9	89.7	77.7	88.6	75.4	138.6	85.3	74.5	134.2	84.1	23.3	20.1	18.3	4837	1413
LAWRENCE MUNICIPAL	39.008N	95.212W	832	3.4	8.6	99.2	76.6	95.5	75.8	92.0	75.4	79.8	92.3	78.3	90.8	76.3	141.4	87.1	74.8	134.5	85.6	25.1	21.9	19.2	4991	1475
MANHATTAN REGIONAL	39.135N	96.679W	1056	1.9	7.5	100.2	75.3	97.0	75.3	93.1	74.6	78.5	92.5	77.5	91.5	74.9	136.1	85.6	73.3	128.7	83.6	24.1	20.5	18.3	5145	1486
MARSHALL AIRFIELD	39.050N	96.767W	1065	4.8	9.2	100.7	75.0	96.8	75.1	93.6	74.8	78.8	91.6	77.5	90.5	75.5	139.1	85.0	74.0	132.2	83.6	24.4	20.6	18.5	4940	1593
MCCONNELL AFB	37.617N	97.267W	1371	8.3	12.8	100.2	72.7	97.2	73.3	93.8	73.4	77.7	90.0	76.6	88.8	74.8	137.4	82.1	73.2	129.6	81.3	27.4	24.8	22.2	4284	1745
PHILIP BILLARD MUNICIPAL	39.073N	95.626W	876	4.0	8.9	98.3	75.9	94.9	75.9	91.8	75.0	79.1	91.6	77.9	90.4	75.5	138.0	86.0	74.2	132.1	84.8	23.2	20.0	18.1	4885	1506

Table 4-8 Design Conditions for Selected Locations *(Continued)*

Meaning of acronyms:
DB: Dry bulb temperature, °F *WB: Wet bulb temperature, °F*
MCWB: Mean coincident wet bulb temperature, °F
Lat: Latitude, °
DP: Dew point temperature, °F
MCDB: Mean coincident dry bulb temperature, °F
Long: Longitude, °
HR: Humidity ratio, grains of moisture per lb of dry air
HDD and CDD 65: Annual heating and cooling degree-days, base 65°F, °F-day
Elev: Elevation, ft
WS: Wind speed, mph

Station	Lat	Long	Elev	Heating DB		Cooling DB/MCWB						Evaporation WB/MCDB				Dehumidification DP/HR/MCDB						Extreme Annual WS			Heat./Cool. Degree-Days	
						0.4%		1%		2%		0.4%		1%		0.4%			1%							
				99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	DP	HR	MCDB	DP	HR	MCDB	1%	2.5%	5%	HDD	CDD 65
SALINA MUNICIPAL	38.800N	97.650W	1269	4.3	9.2	101.9	73.5	98.8	73.7	95.3	73.2	77.3	92.4	76.1	91.0	73.1	129.1	83.9	72.1	124.3	83.0	28.1	25.2	22.6	4799	1706
WICHITA EISENHOWER NATL	37.648N	97.430W	1321	7.6	12.3	101.1	73.2	97.8	73.6	94.4	73.5	77.6	90.7	76.6	89.9	74.2	134.2	83.7	72.9	128.2	82.2	28.3	25.7	23.5	4444	1743
Kentucky																										*8 sites, 9 more on CD-ROM*
BLUE GRASS AP	38.041N	84.606W	980	8.8	14.1	91.5	74.0	89.2	73.6	87.0	72.7	77.3	87.2	76.0	85.3	74.3	132.8	82.6	73.1	127.5	81.0	20.7	18.4	16.5	4535	1197
BOWLING GREEN AP	36.965N	86.424W	528	12.4	17.6	94.1	75.1	91.6	75.0	89.6	74.5	78.4	88.8	77.4	87.5	75.4	135.6	83.9	74.3	130.9	82.7	19.8	17.8	15.8	3979	1475
BOWMAN FIELD	38.228N	85.664W	540	10.7	16.3	93.4	75.1	91.2	74.7	89.3	73.9	78.4	88.4	77.3	87.2	75.5	136.1	83.2	74.5	131.7	82.4	18.5	16.5	14.4	4156	1486
CAMPBELL AAF	36.667N	87.483W	573	12.0	17.7	94.6	76.1	92.1	75.8	90.2	75.3	79.4	89.3	78.3	87.8	77.0	143.8	83.4	75.5	136.3	82.5	20.2	17.7	15.6	3773	1615
CINCINNATI NORTHERN KY INTL	39.043N	84.672W	869	5.9	11.9	91.6	74.3	89.0	73.4	86.7	72.5	77.5	87.0	76.1	85.0	74.6	133.8	82.5	73.3	128.1	80.8	21.6	18.9	17.0	4918	1115
HENDERSON CITY-COUNTY AP	37.800N	87.683W	387	8.8	14.3	93.4	76.3	91.2	75.9	90.1	75.3	79.4	91.3	78.0	89.2	75.4	135.1	86.9	74.5	130.9	85.8	21.3	18.8	16.7	4473	1396
LAKE CUMBERLAND REGIONAL	37.054N	84.615W	927	12.2	18.0	94.1	74.6	91.3	74.0	90.0	73.5	77.9	90.0	76.5	87.8	73.5	128.9	82.6	72.9	126.5	81.8	17.9	15.5	12.6	3945	1388
LOUISVILLE INTL	38.181N	85.739W	488	11.0	16.6	94.2	75.3	91.8	75.0	89.5	74.2	78.7	89.2	77.5	87.9	75.8	137.3	85.0	74.4	130.8	83.3	21.0	18.7	16.8	4057	1604
Louisiana																										*12 sites, 16 more on CD-ROM*
ALEXANDRIA ESLER REGIONAL	31.395N	92.291W	118	26.0	28.4	98.0	76.7	95.6	77.0	93.5	76.9	80.3	89.9	79.5	89.7	78.0	146.1	83.7	77.0	141.3	83.3	16.3	13.6	11.9	1988	2533
ALEXANDRIA INTL	31.335N	92.559W	84	27.0	29.9	97.4	76.9	95.1	77.0	93.0	76.7	80.4	89.6	79.5	89.5	78.5	148.5	84.1	77.0	141.1	83.5	18.8	16.8	14.6	1850	2659
BARKSDALE AFB	32.500N	93.667W	166	24.6	27.6	99.0	75.5	96.4	75.8	93.9	76.0	79.3	90.7	78.4	89.5	76.5	139.1	83.2	75.4	134.1	82.4	20.3	18.1	16.1	2238	2424
BATON ROUGE METROPOLITAN	30.537N	91.147W	64	28.4	31.6	94.8	77.4	93.3	77.4	91.8	77.0	80.4	88.8	79.7	88.1	78.4	148.0	83.7	77.5	143.3	83.0	18.8	16.7	14.7	1576	2721
LAFAYETTE REGIONAL	30.205N	91.988W	38	30.2	33.7	94.8	77.7	93.1	77.5	91.5	77.2	80.5	88.7	79.9	88.2	78.8	149.5	83.7	77.7	144.0	83.2	20.3	18.3	16.5	1443	2857
LAKE CHARLES REGIONAL	30.125N	93.228W	9	30.5	33.8	94.7	77.7	93.0	77.7	91.4	77.6	81.3	88.5	80.4	87.6	79.4	152.6	84.1	78.6	148.3	83.6	20.3	18.2	16.5	1434	2849
LAKEFRONT AP	30.049N	90.029W	9	36.0	39.2	93.5	78.7	92.6	78.4	91.1	77.9	81.4	89.7	80.6	88.8	79.3	151.9	86.0	78.7	148.9	85.6	26.2	23.1	19.9	1072	3385
LOUIS ARMSTRONG NEW ORLEANS	29.993N	90.251W	4	33.3	36.6	94.2	77.9	92.5	77.7	91.0	77.4	80.6	88.8	80.1	88.2	78.5	147.9	84.5	77.8	144.7	84.0	20.9	18.8	17.0	1255	3000
MONROE REGIONAL	32.516N	92.041W	79	25.4	28.4	98.3	77.7	95.8	77.4	93.6	77.1	80.8	91.6	80.0	90.8	78.1	146.6	85.3	77.1	141.7	84.5	19.1	17.1	15.0	2183	2506
NEW ORLEANS NAS	29.817N	90.017W	2	31.2	34.5	93.1	77.6	91.6	77.4	90.3	77.2	81.2	87.5	80.1	86.9	79.4	152.6	83.6	78.4	147.5	83.2	18.6	16.4	14.1	1375	2728
SHREVEPORT DOWNTOWN AP	32.543N	93.745W	179	26.5	29.1	99.5	76.2	97.1	76.1	94.6	76.0	79.4	91.1	78.5	89.9	76.6	139.6	83.0	75.5	134.5	82.7	18.8	16.7	14.8	2173	2654
SHREVEPORT REGIONAL	32.447N	93.824W	254	25.9	29.0	99.2	75.7	96.7	76.0	94.4	76.0	79.3	91.2	78.5	90.0	76.3	138.8	83.4	75.6	135.2	82.8	19.9	17.8	16.1	2093	2612
Maine																										*5 sites, 24 more on CD-ROM*
AUBURN LEWISTON MUNICIPAL	44.050N	70.283W	288	-5.9	0.1	87.8	71.2	83.6	69.7	81.1	67.6	73.8	83.4	71.7	80.4	70.5	113.4	78.6	69.6	109.8	77.3	20.8	18.5	16.4	7585	314
BANGOR INTL	44.798N	68.819W	148	-7.0	-1.8	87.7	70.5	84.2	69.0	81.1	67.0	73.0	83.2	71.2	80.6	69.7	109.8	77.9	68.0	103.3	75.4	23.4	19.8	17.9	7614	360
BRUNSWICK NAS	43.900N	69.933W	70	-2.2	2.3	86.2	70.3	82.8	68.9	80.3	67.1	73.4	82.1	71.4	79.6	70.4	112.1	77.9	68.7	105.8	75.7	23.5	19.8	17.7	7180	369
PORTLAND INTL JETPORT	43.650N	70.300W	45	0.5	5.2	86.7	71.2	83.3	69.9	80.2	68.1	73.9	83.2	72.1	80.0	70.8	113.6	78.5	69.3	107.9	76.2	23.2	19.6	17.5	6930	382
SANFORD SEACOAST REGIONAL	43.394N	70.708W	244	-6.1	0.3	89.7	71.7	85.7	70.0	82.2	68.1	74.5	85.2	72.4	82.3	71.6	117.9	80.9	69.7	110.3	78.6	20.9	18.5	16.3	7423	366
Maryland																										*3 sites, 10 more on CD-ROM*
ANDREWS AFB	38.817N	76.867W	282	15.1	18.7	94.1	75.0	91.0	74.0	88.5	72.9	78.0	88.5	76.6	86.3	75.1	133.2	81.8	73.5	125.9	80.2	24.2	20.5	18.2	4378	1251
BALTIMORE-WASHINGTON INTL	39.167N	76.683W	156	14.6	18.4	94.2	74.8	91.4	74.0	88.6	72.8	77.7	88.9	76.5	86.4	74.9	131.5	81.7	73.7	126.1	80.3	22.0	18.9	16.8	4491	1268
THOMAS POINT	38.900N	76.440W	0	18.2	22.0	86.9	76.4	84.9	75.9	83.1	75.2	81.9	83.3	80.0	81.9	81.5	164.0	82.5	79.4	152.7	81.0	37.6	31.6	26.5	4135	1245
Massachusetts																										*11 sites, 14 more on CD-ROM*
BARNSTABLE MUNICIPAL	41.669N	70.280W	55	10.3	15.4	84.2	72.9	81.5	71.5	79.3	70.4	75.6	81.1	74.3	78.7	73.5	124.7	77.8	72.8	121.7	77.0	24.7	21.2	19.1	5791	526
BOSTON LOGAN INTL	42.361N	71.011W	12	8.5	13.4	90.6	72.6	87.6	71.6	84.3	70.0	75.8	85.7	74.2	83.0	72.6	121.0	80.4	71.3	115.3	78.6	26.8	24.0	20.7	5512	776
BUZZARDS BAY	41.400N	71.030W	0	13.1	17.2	76.4	N/A	74.7	N/A	73.4	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	44.0	38.2	33.9	5411	339
CHATHAM MUNICIPAL	41.688N	69.993W	68	12.1	17.4	83.0	72.8	80.7	71.7	78.4	70.6	75.2	80.1	73.9	78.0	73.3	124.2	77.6	72.5	120.6	76.6	20.4	17.9	16.1	5598	493
LAWRENCE MUNICIPAL	42.717N	71.124W	149	4.9	9.8	90.5	73.0	87.9	71.9	84.2	70.2	75.7	85.5	74.3	83.2	72.7	121.8	79.9	71.8	118.3	79.3	20.4	18.1	16.2	6007	684
MARTHAS VINEYARD AP	41.393N	70.615W	68	9.8	14.4	84.1	72.5	81.4	71.3	79.2	70.2	75.3	80.9	74.0	78.8	73.3	124.0	78.1	72.4	120.4	76.9	26.0	23.4	20.3	5777	451
NEW BEDFORD REGIONAL	41.676N	70.958W	80	8.6	12.4	88.2	73.4	84.5	71.5	82.0	70.2	75.9	83.9	74.5	81.0	73.3	124.0	78.9	72.4	120.5	77.9	23.1	19.8	17.8	5770	586
NORWOOD MEMORIAL	42.191N	71.174W	50	3.1	9.0	90.6	73.5	87.9	72.4	84.4	70.4	76.6	86.0	75.0	83.1	73.4	124.4	79.9	72.5	120.5	79.0	20.3	18.0	16.1	6140	606
PLYMOUTH MUNICIPAL	41.910N	70.729W	149	6.1	10.3	88.7	73.0	85.3	71.7	82.3	70.0	75.8	84.3	74.3	81.4	73.1	123.8	79.0	72.3	120.1	78.1	23.4	19.8	17.8	6052	575
SOUTH WEYMOUTH NAS	42.150N	70.933W	161	5.9	10.4	91.2	73.8	87.7	72.3	84.7	70.7	76.9	86.8	74.9	83.8	74.1	127.9	81.9	72.2	119.7	79.4	18.5	16.5	14.5	5832	646
WORCESTER REGIONAL	42.271N	71.873W	1000	2.4	7.1	85.8	70.9	83.2	69.5	80.6	68.0	73.7	81.5	72.3	79.3	71.4	120.2	77.1	70.0	114.6	75.7	25.9	23.0	19.8	6603	485
Michigan																										*15 sites, 72 more on CD-ROM*
BISHOP INTL	42.967N	83.749W	770	-0.2	4.5	89.4	73.4	86.6	71.7	84.0	70.1	76.0	85.4	74.1	83.2	72.9	125.7	81.9	71.2	118.3	79.2	23.9	20.4	18.4	6692	610
DETROIT COLEMAN YOUNG INTL	42.409N	83.010W	626	5.2	9.6	90.6	73.4	88.0	71.9	85.4	70.5	76.3	86.2	74.5	83.7	73.1	125.9	81.7	71.6	119.4	80.1	20.5	18.6	17.1	5977	883
DETROIT METRO WAYNE COUNTY AP	42.231N	83.331W	631	3.3	8.2	90.3	74.0	87.4	72.5	84.7	70.9	76.8	86.4	74.9	83.5	73.7	128.6	82.2	72.1	121.7	80.1	25.0	21.5	19.1	6066	821
GERALD R FORD INTL	42.883N	85.524W	803	2.8	7.3	89.2	73.1	86.5	71.6	83.9	70.1	76.2	85.2	74.4	82.8	73.4	128.1	81.2	71.5	119.9	79.0	24.9	21.3	19.1	6544	663
GROSSE ILE MUNICIPAL	42.099N	83.161W	587	6.5	9.8	89.8	74.5	86.1	73.7	83.5	72.6	77.9	84.9	76.2	82.9	75.3	135.5	81.8	73.4	126.9	79.9	20.9	18.7	16.8	5887	858
JACKSON COUNTY AP	42.267N	84.467W	998	0.5	5.4	88.6	73.2	85.9	71.8	83.2	70.2	75.9	84.7	74.2	82.3	73.1	127.7	80.8	71.6	121.1	79.2	20.2	18.3	16.8	6596	581
KALAMAZOO BATTLE CREEK INTL	42.235N	85.552W	868	2.5	7.4	90.2	73.1	87.7	71.7	84.3	70.2	76.1	85.4	74.5	83.2	73.0	126.4	81.5	71.9	121.8	80.1	21.7	19.0	17.2	6288	731
LANSING CAPITAL REGION INTL	42.780N	84.579W	841	-0.5	4.7	89.0	73.2	86.2	71.6	83.5	70.0	76.0	85.2	74.2	82.6	73.1	126.8	81.4	71.3	119.1	79.0	24.1	20.4	18.5	6774	579
MBS INTL	43.533N	84.080W	660	0.4	4.7	89.4	73.0	86.3	71.2	83.6	70.0	75.9	85.4	74.1	82.8	73.0	125.3	81.2	71.3	118.4	78.9	24.5	20.9	18.9	6854	586
MUSKEGON COUNTY AP	43.171N	86.237W	625	5.5	9.4	86.2	72.2	83.7	70.7	81.6	69.6	75.3	82.5	73.8	80.4	73.0	125.5	79.6	71.4	118.7	78.0	25.2	22.7	19.9	6565	538
OAKLAND COUNTY INTL	42.665N	83.418W	976	1.0	5.4	89.7	73.0	86.4	71.1	83.7	69.6	75.3	84.9	73.5	82.3	72.4	124.4	80.8	70.6	116.9	78.5	24.1	20.6	18.6	6632	654

Table 4-8 Design Conditions for Selected Locations *(Continued)*

Meaning of acronyms:
DB: Dry bulb temperature, °F
WB: Wet bulb temperature, °F
MCWB: Mean coincident wet bulb temperature, °F
Lat: Latitude, °
DP: Dew point temperature, °F
MCDB: Mean coincident dry bulb temperature, °F
Long: Longitude, °
HR: Humidity ratio, grains of moisture per lb of dry air
Elev: Elevation, ft
WS: Wind speed, mph
HDD and CDD 65: Annual heating and cooling degree-days, base 65°F, °F-day

Station	Lat	Long	Elev	Heating DB		Cooling DB/MCWB						Evaporation WB/MCDB				Dehumidification DP/HR/MCDB						Extreme Annual WS			Heat./Cool. Degree-Days	
						0.4%		1%		2%		0.4%		1%		0.4%			1%							
				99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		WB / MCDB		WB / MCDB		DP / HR / MCDB			DP / HR / MCDB			1%	2.5%	5%	HDD / CDD 65	
SELFRIDGE ANGB	42.608N	82.818W	580	1.7	6.8	90.0	74.0	86.5	72.1	83.9	70.8	76.5	85.5	74.6	83.0	73.4	126.9	81.3	72.1	121.4	79.7	21.6	19.3	17.6	6413	663
ST CLAIR COUNTY INTL	42.911N	82.529W	640	0.0	5.0	89.9	73.5	85.9	71.3	82.3	69.6	75.9	84.5	73.9	81.8	73.0	125.3	79.8	72.0	121.0	78.7	18.7	16.6	14.7	6800	458
WEST MICHIGAN REGIONAL	42.746N	86.097W	689	6.8	10.1	89.5	73.4	86.4	71.9	83.6	70.7	76.2	85.3	74.5	83.0	73.1	125.9	81.6	72.0	121.6	80.0	25.6	22.1	19.1	6218	663
WILLOW RUN AP	42.233N	83.533W	777	1.0	6.4	91.2	73.8	88.3	72.4	85.5	70.7	76.8	87.2	74.9	84.3	73.4	127.7	82.1	72.1	122.1	80.4	24.7	21.3	19.0	6342	733
Minnesota																						*11 sites, 87 more on CD-ROM*				
ANOKA COUNTY AP	45.150N	93.217W	912	-8.8	-4.5	90.2	74.8	87.7	73.7	83.9	71.7	78.1	85.9	75.8	83.4	75.2	136.8	82.9	73.1	127.1	80.4	23.1	19.7	17.7	7530	617
CRYSTAL AP	45.062N	93.351W	861	-9.5	-6.0	90.7	73.4	88.1	72.0	84.5	69.9	76.9	86.9	74.7	84.2	73.3	127.8	83.5	71.9	121.6	81.5	21.1	19.0	17.2	7599	724
DULUTH INTL	46.837N	92.183W	1433	-17.2	-12.0	84.2	69.7	81.1	67.1	78.2	65.3	72.6	81.1	70.0	78.0	69.4	114.1	78.0	67.1	105.0	75.3	24.9	21.4	19.2	9286	218
FLYING CLOUD AP	44.832N	93.471W	907	-10.6	-6.3	90.8	74.2	88.2	72.7	85.0	70.9	77.6	87.4	75.4	84.2	74.7	134.2	83.9	72.4	124.3	81.8	21.8	19.2	17.4	7415	807
MANKATO MUNICIPAL	44.217N	93.917W	1021	-12.3	-8.2	89.9	73.7	86.3	71.9	83.5	70.6	77.0	85.5	74.7	83.0	73.5	129.4	82.3	72.2	123.6	80.9	26.8	24.1	20.8	7705	628
MINNEAPOLIS-ST PAUL INTL	44.883N	93.229W	872	-10.6	-5.8	90.8	73.3	87.8	72.0	84.9	70.2	76.9	87.2	74.8	84.0	73.7	129.4	83.5	71.7	120.9	81.2	24.0	20.6	18.8	7477	782
ROCHESTER INTL	43.904N	92.492W	1304	-12.4	-7.6	87.7	73.3	84.7	71.7	82.2	70.3	76.8	84.3	74.6	81.7	74.3	134.5	81.8	72.2	124.9	79.7	28.2	25.4	23.2	7832	528
SKY HARBOR AP	46.722N	92.043W	610	-10.5	-6.3	86.2	71.9	82.3	69.3	80.5	68.0	75.3	82.8	72.8	79.4	72.9	124.8	79.1	71.5	119.0	77.7	28.3	25.0	21.5	8561	310
SOUTH ST PAUL MUNICIPAL	44.857N	93.033W	820	-8.8	-4.4	90.8	73.1	88.1	71.6	84.6	69.8	77.1	86.2	74.9	83.6	74.5	132.8	82.6	72.2	123.0	80.4	18.8	16.7	14.8	7371	770
ST CLOUD REGIONAL	45.543N	94.051W	1009	-16.8	-11.3	89.4	72.5	86.3	70.7	83.4	68.7	76.2	85.4	74.0	83.0	73.2	128.1	82.1	70.9	118.0	79.5	23.7	20.1	18.1	8425	484
ST PAUL DOWNTOWN AP	44.932N	93.056W	700	-10.0	-5.8	90.6	73.8	87.9	72.5	84.3	70.5	77.3	86.6	75.2	84.0	74.4	132.0	83.5	72.3	122.8	81.3	23.2	20.1	18.3	7465	753
Mississippi																						*6 sites, 13 more on CD-ROM*				
HATTIESBURG-LAUREL AP	31.467N	89.333W	293	25.0	27.7	96.6	75.7	93.1	75.1	91.1	74.9	78.5	89.8	77.7	88.9	75.3	133.9	83.1	74.6	130.9	82.6	16.5	13.8	12.0	2086	2273
JACKSON INTL	32.321N	90.078W	330	23.6	26.9	96.3	76.1	94.2	76.1	92.3	75.9	79.6	90.2	78.5	88.7	77.0	142.1	83.7	76.1	137.8	82.8	18.5	16.4	14.2	2272	2315
KEESLER AFB	30.417N	88.917W	33	30.6	34.7	94.3	80.2	92.2	79.4	90.6	78.8	83.6	90.3	82.2	88.4	81.7	164.9	87.3	80.9	160.6	86.4	18.5	16.6	14.8	1432	2800
KEY FIELD	32.335N	88.744W	294	22.6	26.1	96.1	75.8	93.9	75.9	92.0	75.7	79.7	89.4	78.5	88.2	77.1	142.8	83.6	76.1	137.8	82.6	18.5	16.5	14.5	2351	2161
MERIDIAN NAS MCCAIN FIELD	32.550N	88.567W	271	22.5	26.6	96.8	76.2	94.6	76.2	92.6	75.9	80.0	90.8	78.8	89.7	77.1	142.5	85.6	75.8	136.2	83.9	16.5	13.0	11.3	2343	2232
TUPELO REGIONAL	34.262N	88.771W	361	19.5	23.8	96.2	76.0	93.8	75.7	91.7	75.5	79.3	89.9	78.3	88.7	76.4	139.7	84.0	75.4	135.0	83.2	18.9	16.8	15.2	2911	2010
Missouri																						*9 sites, 16 more on CD-ROM*				
CAPE GIRARDEAU REGIONAL	37.225N	89.571W	336	10.1	15.9	94.4	77.0	92.3	76.7	90.2	75.9	80.2	90.1	78.8	88.5	77.4	144.3	86.4	76.0	137.5	84.6	21.9	19.3	17.5	4193	1513
COLUMBIA REGIONAL	38.817N	92.218W	893	3.6	9.2	94.7	75.7	91.7	75.9	89.0	74.9	79.4	89.4	77.9	87.8	76.5	143.0	85.6	75.1	136.0	84.0	24.0	20.5	18.5	4917	1276
CR WHEELER DOWNTOWN	39.121N	94.597W	742	5.8	10.2	97.3	76.1	94.1	75.7	91.3	75.2	79.5	91.6	78.2	89.8	76.1	140.0	86.8	74.8	133.8	85.7	22.1	19.4	18.0	4567	1663
JEFFERSON CITY MEMORIAL	38.591N	92.156W	573	6.5	11.6	96.1	75.9	92.8	75.2	90.4	74.6	79.4	89.9	77.9	88.5	76.6	141.4	85.0	74.9	133.5	83.4	20.8	18.4	16.4	4574	1444
JOPLIN REGIONAL	37.147N	94.502W	980	8.7	14.0	97.4	74.8	94.4	75.3	91.6	74.8	78.5	90.3	77.6	89.4	75.3	137.5	85.5	74.0	131.7	84.1	25.1	22.0	19.5	4044	1669
KANSAS CITY INTL	39.297N	94.731W	1005	2.4	7.5	96.0	76.5	92.7	76.1	89.7	75.4	79.9	90.5	78.3	88.9	76.9	145.5	86.7	75.3	137.5	85.4	25.4	22.8	20.0	5027	1376
LAMBERT-ST LOUIS INTL	38.753N	90.374W	531	7.5	12.7	96.2	76.7	93.5	76.1	91.1	75.0	79.5	90.9	78.2	89.3	76.3	140.0	86.0	74.9	133.4	85.1	23.3	19.9	18.0	4403	1676
SPIRIT OF ST LOUIS AP	38.657N	90.656W	462	6.2	11.7	95.5	77.1	92.8	76.3	90.4	75.2	79.8	90.7	78.3	89.0	76.8	142.0	85.7	75.2	134.2	84.5	20.6	18.4	16.5	4672	1398
SPRINGFIELD-BRANSON REGIONAL	37.240N	93.390W	1259	7.1	12.8	95.5	74.0	92.4	74.1	89.6	74.0	77.7	88.7	76.6	87.5	74.5	135.4	83.4	73.4	130.1	82.2	23.6	20.4	18.5	4428	1406
Montana																						*7 sites, 18 more on CD-ROM*				
BERT MOONEY AP	45.965N	112.501W	5506	-17.3	-9.1	87.9	57.4	84.7	56.4	81.7	55.9	60.3	79.3	58.9	77.7	54.5	77.8	62.6	52.2	71.3	61.9	20.9	18.7	16.6	9095	79
BILLINGS LOGAN INTL	45.807N	108.542W	3581	-9.5	-3.2	94.7	62.7	91.3	61.9	88.1	61.4	66.3	85.2	64.6	83.8	60.3	89.2	72.0	58.1	82.3	70.7	27.6	24.7	21.7	6738	648
BOZEMAN YELLOWSTONE INTL	45.788N	111.161W	4427	-15.1	-7.7	92.1	61.1	88.7	60.4	85.2	59.4	64.3	83.2	62.5	81.7	58.1	85.2	69.1	55.9	78.5	68.1	21.2	18.2	15.3	8172	237
GREAT FALLS	47.450N	111.383W	3709	-13.8	-6.9	90.6	60.3	87.3	59.5	84.1	58.9	63.5	82.6	61.8	80.6	57.3	80.5	66.9	55.2	74.4	67.6	31.3	27.2	24.7	7706	333
GREAT FALLS INTL	47.473N	111.382W	3664	-16.2	-9.5	92.5	60.9	89.1	60.2	85.6	59.3	64.0	84.4	62.3	82.3	57.8	81.7	67.1	55.6	75.5	66.9	31.4	27.4	24.7	7557	336
MALMSTROM AFB	47.517N	111.183W	3472	-14.9	-8.5	94.5	62.7	90.8	61.5	87.4	60.5	65.8	87.0	63.7	84.6	58.5	83.4	71.7	56.3	76.7	69.8	30.9	27.1	24.4	6875	483
MISSOULA INTL	46.921N	114.093W	3192	-3.5	3.3	93.1	61.8	89.8	61.2	86.3	60.2	64.7	85.3	63.2	83.8	58.3	81.8	68.4	56.2	75.8	67.5	20.9	18.5	16.4	7353	336
Nebraska																						*5 sites, 37 more on CD-ROM*				
CENTRAL NEBRASKA REGIONAL	40.961N	98.314W	1840	-3.2	1.9	95.6	74.0	92.4	73.2	89.4	72.0	77.6	89.2	76.0	87.8	74.3	137.2	84.3	72.5	129.1	82.7	28.8	25.8	23.1	6092	1054
EPPLEY AIRFIELD	41.310N	95.899W	982	-3.2	1.4	94.8	76.0	91.8	75.3	88.9	73.8	79.4	89.5	77.6	87.7	76.5	143.4	86.1	74.7	134.7	84.1	26.5	23.9	20.6	6024	1165
LINCOLN MUNICIPAL	40.851N	96.748W	1190	-2.6	2.0	96.6	74.9	93.3	74.5	90.4	73.6	78.4	90.5	77.0	89.0	75.1	137.8	85.7	73.4	130.0	83.9	27.2	24.5	21.0	5969	1191
NORTH OMAHA AP	41.367N	96.017W	1332	-6.1	-0.1	94.0	75.0	90.9	74.6	88.0	73.0	77.7	89.0	76.3	87.3	74.4	135.3	84.2	72.9	128.2	83.0	23.3	19.2	17.8	5981	1093
OFFUTT AFB	41.117N	95.917W	1047	-2.5	1.7	95.0	76.5	91.4	75.8	89.0	74.5	80.0	89.3	78.2	87.4	77.4	148.2	85.1	75.5	138.7	83.6	24.7	20.8	18.5	5950	1166
Nevada																						*3 sites, 14 more on CD-ROM*				
MCCARRAN INTL	36.072N	115.163W	2180	31.7	34.3	108.5	67.5	106.3	66.7	104.1	65.9	72.5	96.1	71.1	94.4	66.0	103.9	81.4	63.5	95.0	83.3	25.6	22.6	19.7	1962	3557
NELLIS AFB	36.250N	115.033W	1870	28.5	31.7	109.2	67.3	107.2	66.7	104.9	65.9	72.3	94.9	71.1	94.7	66.1	103.3	80.1	63.5	94.1	83.3	26.9	24.0	20.4	2072	3454
RENO TAHOE INTL	39.484N	119.771W	4410	13.1	18.0	96.4	61.4	93.8	60.4	91.3	59.2	63.9	89.4	62.3	87.9	55.0	75.8	71.5	51.7	67.2	72.0	25.7	21.3	18.7	4961	872
New Hampshire																						*4 sites, 11 more on CD-ROM*				
CONCORD MUNICIPAL	43.195N	71.501W	346	-3.0	2.1	90.0	71.3	87.0	69.8	84.1	68.5	74.6	84.9	72.9	82.2	71.6	118.0	78.0	70.0	111.9	76.8	21.0	18.6	16.5	7061	477
JAFFREY-SILVER RANCH AP	42.805N	72.004W	1040	-2.3	2.1	87.1	70.0	83.8	68.6	81.3	67.2	73.0	81.4	71.6	79.2	70.6	116.9	75.9	69.8	113.8	75.2	16.3	13.6	12.0	7280	379

Table 4-8 Design Conditions for Selected Locations *(Continued)*

Meaning of acronyms:
DB: Dry bulb temperature, °F *WB: Wet bulb temperature, °F*
MCWB: Mean coincident wet bulb temperature, °F
Lat: Latitude, °
DP: Dew point temperature, °F
MCDB: Mean coincident dry bulb temperature, °F
Long: Longitude, °
HR: Humidity ratio, grains of moisture per lb of dry air
HDD and CDD 65: Annual heating and cooling degree-days, base 65°F, °F-day
Elev: Elevation, ft
WS: Wind speed, mph

Station	Lat	Long	Elev	Heating DB		Cooling DB/MCWB						Evaporation WB/MCDB				Dehumidification DP/HR/MCDB						Extreme Annual WS			Heat./Cool. Degree-Days	
						0.4%		1%		2%		0.4%		1%		0.4%			1%							
				99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	DP	HR	MCDB	DP	HR	MCDB	1%	2.5%	5%	HDD	CDD 65
MANCHESTER-BOSTON REGIONAL	42.933N	71.438W	225	2.2	7.2	91.1	71.9	88.5	70.6	85.6	69.3	75.3	85.5	73.6	83.0	72.3	120.6	79.8	71.0	115.1	78.3	19.8	17.8	15.8	6207	732
PORTSMOUTH INTL	43.083N	70.817W	100	2.7	7.7	89.5	72.5	86.0	71.1	82.8	69.5	75.3	85.1	73.5	82.0	72.3	120.2	79.6	70.8	114.0	77.8	23.4	19.9	17.7	6396	556
New Jersey																						*7 sites, 7 more on CD-ROM*				
ATLANTIC CITY INTL	39.449N	74.567W	60	12.3	16.5	92.3	75.2	89.3	73.8	86.5	72.7	77.9	87.6	76.6	84.9	75.2	132.5	81.9	74.1	127.7	80.6	25.0	21.6	18.9	4806	1035
MCGUIRE AFB	40.017N	74.600W	131	11.7	15.9	92.9	75.4	90.3	74.5	87.7	73.2	78.5	87.4	77.1	86.1	76.4	138.4	82.6	74.6	130.2	81.2	23.5	20.0	17.9	4849	1073
MILLVILLE MUNICIPAL	39.367N	75.067W	70	11.6	16.2	92.1	75.0	89.4	74.0	86.9	72.8	78.0	87.1	76.7	85.0	75.4	133.3	81.5	74.3	128.5	80.5	20.3	18.2	16.3	4845	1057
MONMOUTH EXECUTIVE	40.183N	74.133W	159	11.6	16.0	91.0	73.9	88.4	72.7	84.5	71.3	76.8	86.9	75.2	84.3	73.3	124.5	80.8	72.4	120.8	80.2	24.9	21.5	19.0	5052	919
NEWARK LIBERTY INTL	40.683N	74.169W	7	12.8	16.9	94.3	74.4	91.2	73.0	88.3	71.9	77.5	88.4	76.2	85.6	74.6	129.6	82.1	73.3	123.9	80.8	25.1	22.0	19.4	4633	1270
TETERBORO AP	40.850N	74.061W	9	11.9	16.1	92.6	74.1	89.9	72.9	87.3	71.5	77.1	87.3	75.6	84.7	74.0	126.7	80.8	72.8	121.8	79.9	20.7	18.5	16.8	4915	1090
TRENTON MERCER AP	40.277N	74.816W	184	11.8	16.1	92.6	74.3	89.9	73.2	87.3	72.3	77.2	87.9	75.8	85.0	74.0	127.5	81.1	72.9	123.0	80.0	20.3	18.1	16.3	4916	1057
New Mexico																						*8 sites, 20 more on CD-ROM*				
ALAMOGORDO-WHITE SANDS AP	32.840N	105.991W	4200	20.7	24.9	99.8	63.5	98.6	63.7	95.2	63.2	70.7	86.5	69.2	85.0	66.2	113.0	75.0	64.4	106.1	74.4	22.5	18.9	16.6	2897	1909
ALBUQUERQUE INTL	35.042N	106.616W	5310	18.3	21.9	95.5	59.9	93.3	59.7	91.0	59.5	65.2	81.4	64.3	80.6	61.4	99.4	68.0	60.1	94.6	68.9	28.3	24.7	20.6	3956	1438
CANNON AFB	34.383N	103.317W	4295	13.7	18.2	98.5	63.2	95.5	63.2	93.0	63.6	70.5	80.9	69.2	81.0	68.4	122.6	71.6	66.4	114.3	70.8	29.3	26.0	23.1	3718	1442
CLOVIS MUNICIPAL	34.433N	103.083W	4216	12.2	17.6	97.3	64.1	94.6	63.9	91.4	63.9	69.4	84.8	68.4	84.3	65.5	110.3	73.8	63.9	104.3	72.5	32.0	27.4	24.5	4053	1247
FOUR CORNERS REGIONAL	36.744N	108.229W	5495	7.8	12.3	95.9	59.6	93.3	59.1	91.0	58.9	64.6	81.4	63.6	81.2	61.1	98.7	67.1	58.9	91.4	67.4	25.2	21.9	18.8	5271	988
HOLLOMAN AFB	32.850N	106.100W	4158	18.7	22.1	99.8	62.7	97.5	62.5	95.2	62.3	68.6	84.6	67.8	84.0	65.3	109.4	71.1	63.7	103.1	71.7	24.8	20.5	17.9	3217	1838
ROSWELL INTL AIR CENTER	33.308N	104.508W	3649	17.7	21.6	101.1	64.4	98.5	64.7	96.2	64.6	70.7	86.6	69.6	85.7	67.1	114.2	74.1	65.9	109.4	73.6	26.6	22.7	18.9	3095	1976
WHITE SANDS	32.383N	106.483W	4081	18.4	22.5	99.0	63.7	96.5	63.9	94.2	63.8	69.8	87.4	68.9	86.1	65.9	111.2	72.1	64.6	106.4	72.3	18.7	16.2	13.3	2946	1811
New York																						*19 sites, 26 more on CD-ROM*				
ALBANY INTL	42.743N	73.809W	312	-0.3	4.7	88.9	72.6	86.1	71.0	83.4	69.7	75.3	84.8	73.7	82.0	72.4	121.2	79.9	70.9	115.3	78.4	23.9	20.3	18.3	6473	639
AMBROSE LIGHT	40.460N	73.830W	0	13.7	17.9	84.0	N/A	80.9	N/A	78.4	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	43.1	37.8	33.9	4850	720
BUFFALO NIAGARA INTL	42.941N	78.736W	716	3.5	7.4	86.3	71.2	83.9	69.9	81.6	68.9	74.7	81.7	73.1	80.2	72.3	122.9	78.9	70.5	115.4	77.5	27.0	24.1	20.6	6460	580
CHAUTAUQUA COUNTY AP	42.150N	79.250W	1723	0.6	4.8	82.4	69.7	81.1	68.6	78.8	66.9	72.3	80.0	70.6	77.8	70.1	118.1	77.4	68.0	109.9	75.3	21.4	18.9	17.2	7185	304
DUTCHESS COUNTY AP	41.627N	73.884W	166	2.7	8.4	91.3	73.5	88.4	72.3	85.6	70.9	76.4	86.8	75.0	84.2	73.2	124.0	81.2	72.1	119.6	80.1	18.6	16.9	14.4	6003	725
ELMIRA CORNING REGIONAL	42.159N	76.892W	955	-0.1	4.8	89.6	71.5	86.4	69.9	83.7	68.7	74.3	84.5	72.7	82.3	71.5	120.3	79.4	69.8	113.7	77.7	20.3	18.0	16.0	6726	470
GREATER BINGHAMTON AP	42.207N	75.980W	1595	0.0	4.5	85.2	69.8	82.3	68.3	79.8	67.2	72.6	80.8	71.1	78.6	70.0	117.3	76.5	68.5	111.0	75.0	20.7	18.6	16.9	7033	403
GREATER ROCHESTER INTL	43.117N	77.677W	539	2.8	7.1	88.5	73.0	85.6	71.0	82.9	69.6	75.3	84.6	73.4	81.9	72.3	122.0	80.3	70.6	114.9	78.0	25.3	21.5	19.0	6466	571
GRIFFIS AIRFIELD	43.234N	75.412W	519	-5.7	0.7	87.9	72.0	85.1	70.2	82.5	68.7	74.6	84.1	72.8	81.5	71.6	118.8	79.7	69.9	112.3	77.7	23.0	19.5	17.3	6992	485
JOHN F KENNEDY INTL	40.639N	73.762W	11	14.1	18.0	89.9	73.0	86.6	71.7	83.9	70.9	76.6	84.1	75.4	81.9	74.5	128.9	80.2	73.3	123.7	78.7	27.5	24.8	21.9	4777	1014
LAGUARDIA AP	40.779N	73.880W	11	14.3	18.4	92.5	73.9	89.6	72.5	86.9	71.4	76.8	86.9	75.5	84.2	74.0	126.7	80.9	72.8	121.5	80.1	27.0	24.4	21.1	4493	1284
LONG ISLAND MACARTHUR AP	40.794N	73.102W	84	11.9	15.9	88.6	73.5	85.7	71.9	83.1	71.0	76.6	83.7	75.3	81.3	74.5	129.4	79.7	73.3	123.9	78.4	24.3	20.6	18.7	5199	839
NIAGARA FALLS INTL	43.108N	78.938W	585	2.8	6.9	87.8	72.4	85.2	71.0	82.5	69.5	75.2	83.7	73.6	81.5	72.5	123.0	79.7	71.2	117.5	78.3	26.5	23.6	20.3	6595	581
ONEIDA COUNTY AP	43.145N	75.384W	711	-4.8	1.2	87.1	72.4	84.2	70.6	81.8	69.0	75.0	83.1	73.2	80.7	72.5	123.4	79.0	70.6	115.7	77.4	20.8	18.7	17.1	7010	471
PLATTSBURGH INTL	44.650N	73.467W	234	-8.7	-2.7	86.8	71.0	83.0	69.5	80.3	68.2	73.9	82.2	72.2	79.9	71.2	115.9	78.5	69.6	109.8	76.8	20.6	18.3	16.3	7534	374
REPUBLIC AP	40.734N	73.417W	81	12.4	17.0	89.9	73.8	86.3	71.9	83.6	71.3	76.6	84.3	75.4	81.9	74.6	129.8	79.9	73.2	123.6	78.4	24.5	20.9	18.8	5039	911
STEWART INTL	41.500N	74.100W	491	3.5	9.1	90.2	72.2	87.0	71.3	84.1	69.8	75.8	84.9	74.1	82.7	73.0	124.9	80.2	71.9	120.1	78.8	24.7	20.9	18.9	5940	713
SYRACUSE HANCOCK INTL	43.111N	76.104W	413	-0.7	4.9	89.2	73.0	86.4	71.2	83.7	69.8	75.2	85.6	73.5	82.6	72.0	120.3	80.8	70.3	113.4	78.8	24.4	20.6	18.4	6495	625
WESTCHESTER COUNTY AP	41.067N	73.708W	379	9.2	13.5	89.6	73.4	86.4	71.8	83.8	70.5	76.2	84.8	74.6	82.2	73.4	125.8	79.4	72.4	121.7	78.4	22.9	19.1	17.0	5471	773
North Carolina																						*14 sites, 55 more on CD-ROM*				
ALBERT J ELLIS AP	34.833N	77.617W	96	20.6	24.8	94.7	77.0	91.4	76.1	90.2	75.5	79.9	90.7	78.4	88.5	77.0	140.9	84.9	75.2	132.8	83.1	19.8	17.4	15.4	2950	1737
ASHEVILLE REGIONAL	35.432N	82.538W	2117	15.4	19.7	88.0	71.2	85.7	70.4	83.6	69.7	73.8	83.0	72.8	81.5	71.3	124.9	77.1	70.2	120.2	76.2	22.7	19.2	17.2	4076	857
CHARLOTTE DOUGLAS INTL	35.224N	80.955W	728	21.6	25.3	94.2	74.7	91.8	74.1	89.6	73.4	77.3	88.6	76.2	86.9	74.2	131.1	81.4	73.2	126.8	80.3	18.5	16.3	13.9	3041	1690
FAYETTEVILLE REGIONAL	34.991N	78.880W	186	22.9	26.7	96.3	75.9	93.4	74.9	91.2	74.4	78.7	89.6	77.8	88.2	75.7	135.5	81.9	75.0	132.4	81.3	20.4	17.9	15.9	2705	1984
HICKORY REGIONAL	35.743N	81.382W	1143	20.0	24.0	92.4	72.7	90.0	72.4	87.8	71.7	76.0	85.7	74.9	84.4	73.2	128.8	79.5	72.2	124.3	78.6	17.4	14.8	12.8	3454	1378
NEW RIVER MCAS	34.708N	77.440W	26	23.4	26.8	93.0	78.0	90.7	77.2	88.7	76.4	80.3	88.7	79.1	87.5	78.4	147.6	84.7	76.8	139.8	83.3	20.4	18.0	16.1	2536	1924
PIEDMONT TRIAD INTL	36.097N	79.943W	890	19.3	22.8	92.5	74.1	90.3	73.5	88.1	72.7	76.8	87.6	75.5	85.5	73.5	128.9	80.9	72.6	125.0	79.8	20.2	17.9	16.0	3533	1454
PITT-GREENVILLE AP	35.633N	77.383W	25	20.8	24.9	95.0	76.1	92.9	75.1	90.8	74.5	78.9	90.5	77.6	88.0	75.4	133.3	82.6	74.8	130.4	82.0	18.7	16.4	14.0	2994	1867
POPE AFB	35.174N	79.009W	218	21.1	25.0	97.1	76.3	94.6	75.7	91.8	74.8	79.9	88.9	78.5	87.8	77.6	144.6	82.9	76.5	139.1	82.2	19.0	16.7	14.5	2817	2034
RALEIGH-DURHAM INTL	35.892N	78.782W	416	20.4	24.2	95.0	75.5	92.5	75.0	90.1	74.3	78.2	89.8	77.3	88.2	75.2	134.2	82.7	74.2	129.6	81.4	18.9	16.7	14.7	3208	1685
SEYMOUR-JOHNSON AFB	35.344N	77.965W	109	22.7	26.5	96.6	76.2	93.7	75.7	91.3	75.1	80.1	88.3	78.9	87.1	78.4	148.1	82.2	77.1	141.6	81.6	19.3	16.9	14.9	2685	2013
SIMMONS AAF	35.133N	78.933W	244	22.4	26.2	96.7	75.7	94.0	75.2	91.4	74.5	78.9	89.7	77.8	88.1	76.3	138.6	82.4	75.2	133.4	81.5	18.2	15.6	12.9	2741	2034
SMITH REYNOLDS AP	36.134N	80.222W	970	20.2	24.4	92.2	73.7	90.2	73.0	88.1	72.4	76.5	86.9	75.4	85.2	73.3	128.3	80.1	72.5	124.7	79.4	17.8	15.3	12.8	3416	1469
WILMINGTON INTL	34.268N	77.900W	38	24.7	27.9	93.5	77.9	91.2	77.0	89.0	76.3	80.1	88.9	79.0	87.1	77.8	144.6	83.9	76.8	139.7	82.9	20.8	18.5	16.7	2364	2020
North Dakota																						*6 sites, 10 more on CD-ROM*				
BISMARCK MUNICIPAL	46.783N	100.757W	1651	-17.5	-12.2	93.1	69.9	89.4	68.9	86.1	67.6	74.5	85.9	72.2	84.1	70.9	121.2	81.8	68.4	110.8	78.7	27.1	24.2	20.7	8452	529
GRAND FORKS AFB	47.967N	97.400W	913	-20.1	-15.5	89.6	72.2	86.1	70.2	83.3	68.5	76.5	83.7	73.7	82.0	74.4	133.0	80.3	71.8	121.6	78.3	28.1	25.1	22.1	9311	420
GRAND FORKS INTL	47.943N	97.184W	842	-21.4	-16.6	89.0	71.7	85.9	69.5	83.2	67.9	74.9	84.6	72.6	82.3	71.7	120.8	81.2	69.2	110.8	78.6	26.8	24.2	20.9	9363	418

Table 4-8 Design Conditions for Selected Locations *(Continued)*

Meaning of acronyms:
DB: Dry bulb temperature, °F *WB: Wet bulb temperature, °F*
MCWB: Mean coincident wet bulb temperature, °F
Lat: Latitude, °
DP: Dew point temperature, °F
MCDB: Mean coincident dry bulb temperature, °F
Long: Longitude, °
HR: Humidity ratio, grains of moisture per lb of dry air
HDD and CDD 65: Annual heating and cooling degree-days, base 65°F, °F-day
Elev: Elevation, ft
WS: Wind speed, mph

Station	Lat	Long	Elev	Heating DB		Cooling DB/MCWB						Evaporation WB/MCDB				Dehumidification DP/HR/MCDB						Extreme Annual WS			Heat./Cool. Degree-Days	
						0.4%		1%		2%		0.4%		1%		0.4%			1%							
				99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	DP	HR	MCDB	DP	HR	MCDB	1%	2.5%	5%	HDD	CDD 65
HECTOR INTL	46.925N	96.811W	900	-18.7	-13.9	90.0	72.3	87.0	70.3	84.3	68.7	75.4	85.3	73.4	83.4	72.3	123.7	82.3	69.9	113.7	80.1	28.1	25.3	23.0	8740	551
MINOT AFB	48.417N	101.350W	1667	-22.3	-17.3	90.5	68.9	87.0	68.1	83.7	66.3	73.2	84.2	70.9	81.6	70.1	117.6	78.8	67.6	107.8	76.7	29.4	26.3	23.4	9348	355
MINOT INTL	48.255N	101.273W	1665	-18.1	-13.2	90.5	69.3	86.9	68.2	83.7	66.2	73.5	84.0	71.1	81.9	70.2	118.4	79.9	67.6	107.9	77.6	27.9	24.9	21.9	8794	431
Ohio																										*13 sites, 21 more on CD-ROM*
AKRON-CANTON REGIONAL	40.917N	81.433W	1208	3.1	8.2	88.6	72.6	86.0	71.4	83.5	69.9	75.4	84.4	73.7	82.0	72.6	126.3	80.5	71.2	120.2	78.1	23.3	19.9	18.1	5993	703
CINCINNATI MUNICIPAL LUNKEN	39.103N	84.419W	490	8.2	13.8	92.7	75.1	90.1	74.3	87.8	73.4	78.1	88.0	76.8	86.1	75.3	134.9	82.8	74.0	129.1	81.4	20.4	18.2	16.4	4755	1123
CLEVELAND HOPKINS INTL	41.405N	81.853W	770	4.6	10.0	89.4	73.6	86.8	72.2	84.2	70.9	76.1	85.3	74.6	83.1	73.1	126.7	81.2	71.8	120.7	79.5	24.6	21.0	18.9	5801	786
FAIRFIELD COUNTY AP	39.756N	82.657W	869	2.3	9.4	90.5	73.9	88.3	73.1	85.9	71.7	77.0	86.9	75.4	84.2	73.5	128.7	81.5	72.5	124.3	80.4	20.0	17.7	15.9	5449	816
FINDLAY AP	41.014N	83.669W	800	1.1	6.9	90.3	73.5	87.8	72.2	85.1	70.6	76.5	86.3	74.7	83.5	73.2	127.2	82.4	71.9	121.5	80.4	25.1	21.7	19.3	5912	807
JMCOX DAYTON INTL	39.906N	84.219W	1000	2.3	8.5	90.3	73.8	87.9	72.8	85.5	71.3	76.5	86.3	75.1	84.1	73.4	128.8	82.0	72.2	123.5	80.5	24.8	21.4	19.0	5481	952
MANSFIELD LAHM REGIONAL	40.820N	82.518W	1295	1.5	7.0	88.0	72.8	85.6	71.5	83.2	70.1	75.5	84.4	74.0	82.1	72.9	128.0	80.4	71.4	121.6	78.7	24.2	20.7	18.7	6110	670
OHIO STATE UNIVERSITY AP	40.078N	83.078W	905	4.4	9.9	90.5	73.5	88.2	72.7	85.8	71.3	76.5	86.4	75.0	84.0	73.2	127.5	81.6	72.2	123.4	80.2	22.1	19.2	17.3	5450	928
PORT COLUMBUS INTL	39.991N	82.881W	810	5.5	11.0	91.3	73.7	89.0	72.7	86.6	71.5	76.7	87.1	75.3	84.7	73.5	128.6	81.3	72.3	123.2	80.3	23.1	19.4	17.3	5185	1055
RICKENBACKER INTL	39.817N	82.933W	744	6.8	12.0	92.7	74.3	90.4	73.3	88.2	72.3	79.7	86.1	77.5	85.8	78.5	152.3	84.1	75.1	135.4	80.9	23.5	19.6	17.4	5025	1137
TOLEDO EXPRESS AP	41.589N	83.801W	669	1.5	7.0	91.2	74.1	88.4	72.3	85.7	70.9	77.0	86.9	75.2	84.1	73.9	129.7	82.3	72.3	122.7	80.1	24.5	20.8	18.6	6050	808
WRIGHT-PATTERSON AFB	39.833N	84.050W	823	3.2	9.6	91.1	74.4	88.6	73.2	86.2	71.8	77.0	86.8	75.5	84.9	74.1	131.3	81.5	72.7	125.2	80.1	22.5	19.4	17.4	5363	928
YOUNGSTOWN-WARREN REGIONAL	41.254N	80.674W	1180	3.3	8.1	88.2	72.4	85.6	70.7	83.2	69.4	74.9	84.2	73.3	81.8	72.0	123.8	79.6	70.5	117.2	77.5	21.1	18.8	17.0	6159	582
Oklahoma																										*9 sites, 33 more on CD-ROM*
HENRY POST AAF	34.650N	98.400W	1189	15.7	20.2	103.5	72.5	100.3	72.8	97.4	72.6	77.3	91.1	76.3	90.2	73.7	131.3	82.1	72.6	126.3	81.3	25.9	22.9	20.2	3154	2270
LAWTON-FORT SILL REGIONAL	34.558N	98.417W	1069	16.2	20.0	104.5	72.7	101.8	72.8	99.2	73.3	77.9	92.3	76.8	91.3	73.5	129.8	82.9	72.8	126.8	82.3	26.2	23.3	20.2	3157	2375
RICHARD L JONES JR AP	36.039N	95.984W	638	14.3	18.5	101.6	75.3	99.0	76.3	96.1	76.2	79.4	93.6	78.4	92.3	75.4	136.4	85.3	74.7	133.1	84.8	19.8	17.8	16.1	3513	2084
STILLWATER REGIONAL	36.162N	97.089W	984	12.1	17.7	102.5	74.4	99.5	74.9	96.7	74.8	78.9	93.4	77.8	92.1	75.0	136.3	86.2	73.4	128.8	84.1	24.8	22.1	19.7	3610	2072
TINKER AFB	35.417N	97.383W	1291	13.8	18.5	100.4	73.5	97.4	73.6	94.6	73.9	78.1	90.2	77.0	89.3	74.9	137.5	83.4	73.4	130.3	82.1	26.5	24.0	20.9	3387	1997
TULSA INTL	36.199N	95.887W	650	13.5	18.3	100.4	75.2	97.6	75.6	94.8	75.5	79.2	92.4	78.2	91.1	75.5	136.6	85.6	74.4	131.7	84.8	24.7	21.7	19.5	3450	2108
VANCE AFB	36.333N	97.917W	1306	10.3	15.6	101.9	73.1	99.4	73.3	96.6	73.5	77.5	91.7	76.5	90.5	73.9	132.9	82.1	72.8	127.8	81.4	27.9	25.1	22.4	3920	1960
WILEY POST AP	35.534N	97.647W	1297	13.8	18.6	100.4	73.3	98.3	73.5	95.1	73.6	77.4	91.1	76.3	90.0	73.3	129.8	83.6	72.5	126.2	82.7	27.2	25.0	22.9	3473	2111
WILL ROGERS WORLD AP	35.389N	97.601W	1285	14.5	19.1	100.7	73.7	97.9	74.0	95.0	74.0	77.8	91.1	76.9	90.2	74.3	134.2	83.9	73.2	129.2	82.5	27.7	25.1	22.9	3420	2016
Oregon																										*9 sites, 29 more on CD-ROM*
AURORA STATE AP	45.249N	122.769W	196	25.2	28.1	91.4	67.1	88.1	66.6	83.9	65.1	69.8	86.1	68.1	84.3	63.7	88.9	75.1	62.6	85.6	73.4	18.2	15.9	12.9	4461	375
CORVALLIS MUNICIPAL	44.500N	123.283W	250	24.7	27.5	92.7	67.0	89.7	66.0	85.5	64.4	68.8	88.5	67.2	86.7	61.2	81.5	75.6	59.1	75.4	74.6	19.7	17.7	16.0	4306	393
EUGENE AP	44.128N	123.221W	353	22.9	26.9	91.8	66.7	88.0	65.7	84.3	64.5	68.9	87.5	67.2	84.7	62.1	84.6	74.6	60.4	79.4	72.5	19.5	17.3	15.7	4664	282
MCMINNVILLE MUNICIPAL	45.195N	123.134W	159	25.8	28.1	92.2	66.7	88.4	66.0	84.2	64.7	68.9	87.3	67.2	85.2	62.4	84.8	73.4	60.8	80.0	72.3	20.6	17.6	15.4	4654	301
PORTLAND INTL	45.596N	122.609W	19	25.0	29.0	91.2	67.5	87.1	66.4	83.4	65.2	69.5	86.8	68.0	84.2	63.2	86.8	75.2	61.8	82.4	73.3	23.7	19.7	17.4	4232	447
PORTLAND-HILLSBORO AP	45.541N	122.949W	204	22.7	26.5	91.7	67.8	88.0	66.9	83.8	65.3	70.2	87.6	68.1	84.9	63.7	89.0	76.5	61.6	82.5	73.3	18.6	16.5	13.9	4792	276
ROBERTS FIELD	44.256N	121.139W	3043	5.1	12.0	93.1	61.4	90.2	60.7	87.0	59.4	63.6	88.2	62.0	85.8	54.9	71.7	66.9	52.8	66.4	66.6	20.9	18.7	16.7	6541	238
ROGUE VALLEY INTL	42.381N	122.872W	1297	22.8	26.0	99.0	67.1	95.5	65.7	92.3	64.6	68.9	94.1	67.4	91.4	60.3	81.9	74.1	58.3	76.2	73.8	18.2	15.3	12.3	4262	864
SALEM MUNICIPAL	44.905N	123.001W	205	23.7	27.3	92.2	66.9	88.1	65.9	84.3	64.6	68.8	88.3	67.2	85.1	61.7	82.8	73.4	60.1	78.1	72.3	20.9	18.2	16.1	4503	341
Pennsylvania																										*14 sites, 18 more on CD-ROM*
ALLEGHENY COUNTY AP	40.355N	79.922W	1248	6.7	10.9	89.2	72.1	86.5	70.9	84.2	69.7	75.0	84.4	73.5	82.3	72.3	125.2	79.6	70.7	118.6	78.0	20.5	18.4	16.7	5389	838
ALTOONA-BLAIR COUNTY AP	40.296N	78.320W	1480	6.4	10.4	88.3	71.8	85.5	70.6	82.9	69.5	74.7	84.0	73.2	81.8	72.0	124.9	79.7	70.3	118.0	77.8	23.4	19.5	17.3	5881	619
BUTLER COUNTY AP	40.777N	79.950W	1247	3.1	8.8	88.2	72.0	84.5	70.4	82.3	69.0	74.7	83.7	73.1	81.7	72.2	124.6	80.1	70.4	117.0	77.7	18.2	15.9	13.3	6100	561
CAPITAL CITY AP	40.217N	76.851W	340	12.3	16.2	92.3	74.0	89.5	72.5	86.9	71.5	76.5	87.0	75.2	84.9	73.3	125.3	80.5	72.3	121.1	79.6	20.4	18.1	16.0	5009	1066
ERIE INTL	42.080N	80.183W	730	6.1	10.4	86.7	73.0	84.2	71.7	81.9	70.6	75.3	82.8	73.8	81.2	72.8	125.0	80.6	71.2	118.3	78.7	24.4	21.1	19.2	6035	668
HARRISBURG INTL	40.196N	76.772W	312	11.7	15.7	92.4	75.1	89.5	73.8	86.8	72.6	77.9	87.8	76.4	85.4	75.0	132.9	83.0	73.4	125.6	81.0	25.6	22.5	19.0	5030	1119
LEHIGH VALLEY INTL	40.651N	75.449W	390	8.7	13.0	91.0	73.7	88.3	72.5	85.7	71.3	76.6	86.5	75.2	83.9	73.6	127.0	81.0	72.4	121.7	79.8	22.4	19.0	17.0	5482	851
NORTHEAST PHILADELPHIA AP	40.082N	75.011W	100	13.4	17.7	93.2	74.8	90.5	73.6	88.1	72.5	77.8	88.5	76.3	86.1	74.9	131.1	82.4	73.2	123.9	80.5	22.1	19.0	17.3	4692	1194
PHILADELPHIA INTL	39.868N	75.231W	10	14.5	18.6	93.6	74.9	90.9	74.0	88.3	72.6	78.1	88.5	76.7	86.0	75.3	132.5	82.5	74.0	126.7	81.1	24.8	21.0	18.9	4425	1357
PITTSBURGH INTL	40.485N	80.214W	1203	5.6	10.4	89.2	72.0	86.6	70.7	84.3	69.6	75.1	84.6	73.5	82.1	72.1	124.3	79.7	70.8	118.5	78.0	22.8	19.4	17.4	5532	785
READING REGIONAL	40.367N	75.967W	344	9.9	14.2	92.6	74.4	89.7	73.2	86.9	72.0	77.1	88.0	75.6	85.0	73.5	126.2	81.6	72.6	122.4	80.6	22.8	19.3	17.5	5170	1000
WASHINGTON COUNTY AP	40.133N	80.283W	1185	2.7	8.8	88.3	71.2	85.5	69.9	82.7	68.6	74.2	83.3	72.6	81.8	71.7	122.5	80.2	70.0	115.5	77.9	19.0	16.8	14.5	5958	540
WILKES-BARRE SCRANTON INTL	41.334N	75.727W	930	4.7	9.4	89.1	71.7	86.2	70.2	83.6	68.9	74.6	84.2	73.1	81.9	71.8	121.5	78.7	70.3	115.4	77.3	20.2	18.0	16.1	6007	653
WILLOW GROVE NAS	40.200N	75.150W	361	12.5	16.6	91.9	73.9	89.4	72.6	86.8	71.5	77.1	87.5	75.5	84.9	73.9	127.9	82.5	72.5	121.9	80.8	19.4	16.9	14.7	4935	1034
Rhode Island																										*1 site, 10 more on CD-ROM*
T F GREEN AP	41.722N	71.433W	60	8.8	13.3	90.1	73.2	86.7	71.8	83.9	70.4	76.3	85.3	74.8	82.1	73.6	125.5	80.1	72.4	120.3	78.7	24.2	20.5	18.5	5481	765

Table 4-8 Design Conditions for Selected Locations *(Continued)*

Meaning of acronyms:
DB: Dry bulb temperature, °F *WB: Wet bulb temperature, °F*
MCWB: Mean coincident wet bulb temperature, °F
Lat: Latitude, °
DP: Dew point temperature, °F
MCDB: Mean coincident dry bulb temperature, °F
Long: Longitude, °
HR: Humidity ratio, grains of moisture per lb of dry air
HDD and CDD 65: Annual heating and cooling degree-days, base 65°F, °F-day
Elev: Elevation, ft
WS: Wind speed, mph

Station	Lat	Long	Elev	Heating DB		Cooling DB/MCWB						Evaporation WB/MCDB				Dehumidification DP/HR/MCDB						Extreme Annual WS			Heat./Cool. Degree-Days	
						0.4%		1%		2%		0.4%		1%		0.4%			1%							
				99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		WB / MCDB		WB / MCDB		DP / HR / MCDB			DP / HR / MCDB			1%	2.5%	5%	HDD / CDD 65	
South Carolina																						*6 sites, 13 more on CD-ROM*				
CHARLESTON INTL	32.899N	80.040W	40	27.5	30.9	94.3	78.0	92.2	77.5	90.3	76.9	80.7	89.1	79.8	87.9	78.6	148.8	84.4	77.6	143.7	83.5	20.5	18.3	16.5	1845	2379
COLUMBIA METROPOLITAN	33.942N	81.118W	225	23.4	27.0	97.1	75.4	94.6	74.8	92.5	74.5	78.5	89.7	77.6	88.5	75.8	136.2	82.3	74.9	131.7	81.4	19.5	16.9	14.8	2432	2202
FLORENCE REGIONAL	34.185N	79.724W	146	24.1	27.3	96.1	76.6	93.5	75.9	91.4	75.4	79.3	90.6	78.3	88.8	76.5	138.8	83.5	75.4	133.8	82.3	19.4	17.6	15.6	2396	2114
FOLLY ISLAND	32.680N	79.890W	10	31.6	35.1	87.5	77.9	86.2	78.0	85.1	77.7	80.5	85.0	79.7	84.3	79.3	151.9	84.3	78.2	146.6	83.3	33.0	26.2	23.0	1878	2157
GREENVILLE-SPARTANBURG INTL	34.884N	82.221W	943	21.9	25.6	94.4	73.7	91.8	73.4	89.5	72.7	77.0	87.8	75.8	85.9	74.0	131.3	80.3	73.0	127.0	79.5	19.2	17.1	15.4	3016	1642
SHAW AFB	33.967N	80.467W	241	23.7	27.2	96.0	75.7	93.4	75.4	91.2	75.0	79.2	89.9	78.1	88.2	76.5	139.5	83.3	75.3	134.0	82.0	19.6	17.4	15.4	2435	2066
South Dakota																						*3 sites, 21 more on CD-ROM*				
ELLSWORTH AFB	44.150N	103.100W	3278	-8.2	-2.8	95.2	66.4	91.4	65.7	88.2	65.1	71.2	85.8	69.4	83.9	66.6	110.6	77.5	64.5	102.7	75.8	35.4	30.0	25.9	7028	678
RAPID CITY REGIONAL	44.043N	103.054W	3160	-8.8	-3.4	96.7	66.0	92.7	65.7	89.0	64.9	71.1	85.5	69.4	84.4	66.7	110.8	78.1	64.7	103.1	75.8	35.2	30.6	26.2	7069	659
SIOUX FALLS REGIONAL	43.578N	96.754W	1428	-11.3	-6.5	91.5	73.7	88.4	72.9	85.6	71.3	77.4	86.7	75.5	84.9	74.7	137.1	83.7	72.7	127.7	81.7	27.3	24.5	21.1	7510	739
Tennessee																						*7 sites, 6 more on CD-ROM*				
CHATTANOOGA AP	35.031N	85.201W	671	19.6	23.7	95.0	74.5	92.6	74.2	90.4	73.6	77.7	88.8	76.8	87.3	74.9	133.9	81.5	73.8	129.3	80.7	17.9	16.0	13.6	3093	1790
MCGHEE TYSON AP	35.818N	83.986W	962	17.1	21.6	92.7	73.9	90.4	73.5	88.3	72.9	77.1	87.5	76.0	85.8	74.0	131.5	81.4	73.0	127.1	80.4	20.6	17.8	15.3	3532	1525
MCKELLAR-SIPES REGIONAL	35.593N	88.917W	433	15.7	19.9	95.1	76.5	92.8	76.5	90.8	76.0	79.9	90.3	78.7	88.8	77.0	142.8	85.8	75.7	136.5	84.3	19.7	17.8	15.9	3436	1739
MEMPHIS INTL	35.056N	89.987W	254	19.5	23.8	96.6	76.9	94.4	76.4	92.5	76.0	79.9	91.5	78.8	90.0	76.7	140.6	85.8	75.6	135.3	85.0	20.5	18.4	16.7	2884	2266
MILLINGTON MUNICIPAL	35.350N	89.867W	320	18.6	23.1	99.2	80.3	96.9	78.7	93.5	77.4	82.9	95.1	81.2	92.4	79.4	154.6	89.3	78.7	150.9	88.6	18.2	16.1	13.6	3065	2125
NASHVILLE INTL	36.119N	86.689W	600	15.4	19.9	94.6	74.8	92.3	74.6	90.2	73.9	78.1	88.9	77.0	87.7	75.1	134.5	82.9	73.9	129.3	81.8	19.0	17.0	15.3	3491	1736
TRI-CITIES REGIONAL	36.473N	82.404W	1500	13.5	17.9	90.2	71.8	88.0	71.3	86.0	70.8	75.0	84.9	73.9	83.6	72.1	125.5	79.0	71.0	121.0	77.9	18.9	16.5	13.6	4177	1036
Texas																						*51 sites, 101 more on CD-ROM*				
ABILENE REGIONAL	32.411N	99.682W	1790	20.1	24.6	100.3	70.4	98.1	70.5	95.9	70.7	75.5	89.1	74.6	88.2	72.1	127.0	79.9	71.1	122.4	79.3	26.3	23.9	20.8	2444	2488
AL MANGHAM JR REGIONAL	31.578N	94.709W	355	25.0	27.6	99.1	75.7	96.8	75.8	93.2	75.6	79.0	90.1	78.3	89.5	76.5	140.1	82.3	75.2	134.1	81.7	18.5	16.3	14.0	2144	2460
AMARILLO RICK HUSBAND INTL	35.230N	101.704W	3604	10.1	15.8	98.6	65.4	95.8	65.8	93.2	65.8	71.2	86.4	70.1	85.6	67.2	114.3	75.2	66.0	109.6	74.3	30.1	26.7	24.4	4043	1471
ANGELINA COUNTY AP	31.236N	94.754W	288	26.9	29.7	99.0	76.1	96.5	76.4	94.3	76.3	79.9	90.4	79.1	89.4	77.3	143.8	82.8	76.7	140.8	82.5	17.8	16.1	14.2	1831	2709
AUSTIN-BERGSTROM INTL	30.183N	97.680W	480	26.7	30.2	100.2	74.1	98.2	74.4	96.2	74.5	78.5	88.9	78.0	88.3	76.4	140.0	81.1	75.5	135.9	80.5	20.9	18.8	17.0	1634	3005
BROWNSVILLE INTL	25.914N	97.423W	24	38.7	42.5	95.7	77.8	94.5	77.8	93.3	77.8	80.9	88.0	80.3	87.6	79.3	152.3	83.0	78.6	148.6	82.7	26.3	23.9	20.8	509	4073
CORPUS CHRISTI INTL	27.774N	97.512W	44	34.6	38.3	96.8	77.6	95.3	77.7	93.7	77.6	81.0	89.5	80.3	88.6	79.1	151.2	83.3	78.4	147.6	83.0	27.4	25.0	23.1	826	3625
CORPUS CHRISTI NAS	27.683N	97.283W	18	37.2	41.5	92.8	79.3	91.6	79.4	90.8	79.3	82.4	88.0	81.6	87.4	81.2	162.1	84.7	80.1	156.2	84.4	26.2	23.9	20.9	705	3752
DALLAS EXECUTIVE	32.681N	96.868W	658	24.9	28.0	101.7	74.1	99.4	74.6	97.2	74.4	78.1	91.7	77.3	90.8	74.8	133.8	82.3	73.5	127.5	81.4	22.7	19.4	17.5	2112	2854
DALLAS FORT WORTH INTL	32.898N	97.019W	560	23.3	27.6	101.4	74.2	99.1	74.5	96.9	74.6	78.5	91.4	77.7	90.7	75.4	135.7	83.7	74.3	131.0	82.8	26.1	23.7	20.6	2161	2878
DALLAS HENSLEY FIELD NAS	32.733N	96.967W	492	21.5	27.2	99.6	75.5	97.5	75.4	95.3	75.0	79.0	92.0	77.9	91.2	75.4	135.5	85.6	74.2	130.1	84.2	20.6	18.7	17.0	2171	2723
DALLAS LOVE FIELD	32.852N	96.856W	440	24.7	28.7	101.5	74.7	99.3	75.2	97.2	74.9	79.1	92.6	78.0	91.3	75.4	135.1	84.4	74.5	131.2	83.6	23.0	20.3	18.7	2031	3024
DEL RIO INTL	29.378N	100.927W	999	31.5	34.6	102.1	71.9	100.1	72.1	98.3	72.2	77.6	89.4	76.9	88.7	74.9	135.7	81.4	73.8	130.5	81.0	20.8	18.6	16.9	1269	3534
DRAUGHON-MILLER CENTRAL TEXAS	31.150N	97.417W	682	25.1	28.0	100.1	74.0	98.9	74.0	96.8	74.0	77.9	90.6	77.2	89.9	74.8	133.9	81.7	73.4	127.3	80.9	25.1	22.5	20.1	1978	2767
DYESS AFB ABILENE	32.417N	99.850W	1788	19.1	23.4	102.4	72.0	100.1	71.8	98.0	71.7	76.8	91.5	75.6	90.5	73.2	131.7	81.0	72.1	127.1	80.4	26.9	24.4	21.1	2509	2644
EASTERWOOD FIELD	30.589N	96.365W	305	28.4	31.7	100.0	75.4	98.0	75.6	95.9	75.5	79.7	90.6	78.6	88.9	77.2	143.2	83.1	76.3	139.0	82.2	20.5	18.5	16.8	1568	3100
EL PASO INTL	31.811N	106.376W	3918	25.1	28.4	101.1	64.1	98.8	63.7	96.6	63.6	69.9	86.3	69.1	85.4	66.3	112.3	73.2	65.1	107.4	73.4	27.2	23.5	19.5	2300	2521
FORT WORTH ALLIANCE AP	32.973N	97.318W	685	21.2	25.4	102.3	73.9	100.0	74.4	97.7	74.2	78.2	92.4	77.3	91.4	74.6	133.0	83.5	73.3	127.1	82.3	23.4	20.7	18.8	2397	2716
FORT WORTH NAS	32.767N	97.450W	608	24.1	28.1	102.5	73.1	100.2	73.6	98.0	73.7	78.2	92.2	77.3	91.2	74.6	132.6	83.3	73.4	127.0	82.5	24.7	21.7	19.5	2092	2978
FT WORTH MEACHAM INTL	32.819N	97.361W	687	22.5	26.8	101.5	73.9	99.3	74.5	97.1	74.3	78.4	91.6	77.5	90.8	75.1	135.3	83.5	74.0	130.1	82.4	23.3	20.3	18.6	2241	2778
GALVESTON SCHOLES INTL	29.273N	94.859W	5	36.5	39.6	92.0	79.1	90.9	79.2	90.0	79.1	81.9	87.7	81.2	87.1	80.9	160.3	84.5	79.4	152.5	84.5	25.5	22.7	20.1	990	3344
GEORGE BUSH INTERCONTINENTAL	29.980N	95.360W	95	31.1	34.1	97.5	76.4	95.5	76.5	93.8	76.5	80.1	88.8	79.4	88.1	78.0	146.1	82.8	77.3	142.5	82.4	20.0	18.1	16.4	1340	3123
GEORGETOWN MUNICIPAL	30.679N	97.679W	787	26.5	28.3	99.0	72.8	96.9	72.9	94.5	72.9	76.9	88.5	76.1	87.7	73.4	127.8	79.8	73.0	125.9	79.7	21.0	18.8	17.1	1948	2688
HOOKS MEMORIAL	30.068N	95.556W	152	29.7	32.9	98.7	75.7	96.2	76.0	93.5	76.2	80.2	88.1	79.4	87.6	78.8	150.4	82.5	77.4	143.1	82.1	17.7	15.9	13.6	1458	3016
HOUSTON ELLINGTON AP	29.617N	95.167W	32	34.1	37.0	96.8	78.1	94.9	78.5	93.0	78.5	82.1	88.8	81.3	88.0	81.1	161.8	84.1	79.4	152.9	83.3	19.8	18.1	16.4	1167	3197
JACK BROOKS REGIONAL	29.951N	94.021W	16	31.6	35.0	95.0	77.7	93.2	77.9	91.7	77.9	81.5	88.9	80.5	87.9	79.4	152.9	84.7	78.7	149.0	83.9	21.5	19.2	17.6	1328	2948
KILLEEN MUNICIPAL	31.083N	97.683W	841	26.3	28.9	100.0	74.0	98.9	74.1	96.8	74.1	77.5	91.8	76.9	91.0	73.4	127.9	82.3	72.9	126.1	82.1	22.1	19.8	18.2	1889	2861
LACKLAND AFB	29.383N	98.583W	690	29.7	33.1	100.4	74.1	99.0	74.1	96.9	74.1	80.0	88.1	78.8	87.0	78.5	151.8	82.1	77.0	144.5	81.6	20.8	18.7	16.9	1359	3278
LAREDO INTL	27.544N	99.463W	494	36.2	39.2	104.7	74.5	102.4	74.3	100.4	74.2	79.4	91.5	78.6	90.6	77.1	143.6	82.7	75.7	136.9	82.6	24.4	21.2	19.8	783	4454
LAUGHLIN AFB	29.367N	100.783W	1082	30.7	34.3	104.3	72.0	102.0	72.5	99.9	72.6	78.9	90.0	77.5	89.9	76.5	143.7	83.3	74.6	134.7	81.9	23.9	20.5	18.5	1179	3710
LONGVIEW EAST TEXAS REGIONAL	32.385N	94.712W	365	25.5	28.1	100.2	74.9	97.9	75.2	95.2	75.3	78.8	90.4	78.1	89.6	75.9	137.0	82.2	75.2	133.9	81.8	20.3	18.1	16.3	2121	2628
LUBBOCK INTL	33.666N	101.823W	3254	15.9	20.2	99.4	66.2	97.1	66.9	94.7	66.9	72.6	87.8	71.4	86.8	68.6	118.7	76.8	67.5	114.2	75.8	29.0	26.0	23.5	3256	1915
MCALLEN INTL	26.176N	98.240W	100	38.6	42.1	100.4	75.9	99.0	76.4	97.4	76.3	80.3	90.7	79.6	89.1	78.4	148.1	82.2	77.5	143.4	82.0	24.9	22.8	20.5	532	4502
MCGREGOR EXECUTIVE	31.485N	97.316W	592	24.8	27.7	101.7	74.4	99.5	74.6	97.4	74.5	78.4	91.9	77.7	91.6	75.1	134.7	83.1	73.4	127.2	82.2	23.6	20.6	18.7	2087	2794
MCKINNEY NATL	33.190N	96.591W	586	20.8	25.0	101.0	74.3	99.1	75.0	96.9	74.9	78.4	91.7	77.7	90.9	75.2	134.9	82.8	74.1	130.2	82.0	23.7	20.3	18.3	2511	2546
MIDLAND INTL	31.948N	102.209W	2862	20.5	24.4	101.1	66.7	98.6	67.1	96.5	67.3	73.0	87.9	72.0	86.9	69.4	120.4	76.0	68.4	116.0	75.9	26.9	24.2	20.9	2554	2392
NEW BRAUNFELS MUNICIPAL	29.709N	98.046W	645	28.3	31.4	100.3	73.7	98.6	73.9	96.6	73.9	78.1	88.9	77.4	88.0	75.5	136.9	80.3	75.0	134.6	80.1	23.7	20.5	18.6	1510	3061
PORT ARANSAS	27.830N	97.050W	0	37.5	41.6	86.2	78.0	85.4	78.2	85.0	78.1	80.5	83.8	80.2	83.7	79.5	152.9	83.1	79.0	150.4	82.9	38.5	32.3	26.8	795	3088

Table 4-8 Design Conditions for Selected Locations *(Continued)*

Meaning of acronyms:
DB: Dry bulb temperature, °F *WB: Wet bulb temperature, °F*
MCWB: Mean coincident wet bulb temperature, °F
Lat: Latitude, °
DP: Dew point temperature, °F
MCDB: Mean coincident dry bulb temperature, °F
Long: Longitude, °
HR: Humidity ratio, grains of moisture per lb of dry air
Elev: Elevation, ft
WS: Wind speed, mph
HDD and CDD 65: Annual heating and cooling degree-days, base 65°F, °F-day

Station	Lat	Long	Elev	Heating DB		Cooling DB/MCWB						Evaporation WB/MCDB				Dehumidification DP/HR/MCDB						Extreme Annual WS			Heat./Cool. Degree-Days	
						0.4%		1%		2%		0.4%		1%		0.4%			1%							
				99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		WB / MCDB		WB / MCDB		DP / HR / MCDB			DP / HR / MCDB			1%	2.5%	5%	HDD / CDD 65	
RANDOLPH AFB	29.533N	98.262W	728	28.5	31.9	100.2	73.7	98.6	73.7	96.5	73.8	78.4	88.2	77.7	87.5	76.4	141.4	80.6	75.4	136.6	80.1	21.5	19.2	17.6	1455	3103
REESE AFB LUBBOCK	33.600N	102.050W	3337	14.7	19.4	101.0	67.0	97.8	67.3	95.1	67.2	73.2	87.2	72.0	86.7	69.5	122.9	78.7	67.9	116.4	77.6	27.4	24.1	20.6	3182	1831
ROBERT GRAY AFF	31.067N	97.833W	1015	25.7	29.7	100.4	73.0	99.0	73.2	96.8	73.3	77.7	89.9	76.9	88.9	75.0	136.3	81.0	73.6	130.0	80.0	23.9	20.7	18.8	1802	2906
SABINE PASS	29.670N	94.050W	3	32.3	36.2	89.0	77.4	87.5	77.6	86.6	77.6	80.6	85.4	80.1	85.1	79.4	152.8	83.9	78.4	147.6	83.7	34.7	27.1	23.6	1428	2660
SAN ANGELO REGIONAL	31.352N	100.495W	1916	22.2	26.0	101.7	70.0	99.5	69.8	97.3	69.8	75.1	89.1	74.2	88.2	71.7	125.7	79.4	70.6	121.1	78.8	24.6	21.3	19.3	2185	2642
SAN ANTONIO INTL	29.544N	98.484W	789	29.9	33.0	99.3	73.5	97.6	73.5	95.8	73.8	78.1	88.0	77.4	87.1	76.0	139.8	80.2	75.2	136.3	80.0	20.6	18.7	16.9	1380	3222
SAN MARCOS REGIONAL	29.891N	97.864W	597	27.6	30.2	99.8	74.0	98.8	74.1	96.7	74.1	78.3	90.3	77.6	89.9	75.2	135.0	83.0	73.9	129.3	82.3	24.6	21.3	19.2	1622	3032
STINSON MUNICIPAL	29.339N	98.472W	571	30.6	33.8	101.1	73.6	99.3	73.9	97.3	73.6	78.7	88.9	78.0	88.2	76.3	140.3	81.7	75.3	135.6	80.8	18.9	17.2	15.8	1274	3399
VALLEY INTL	26.228N	97.654W	34	37.1	41.1	98.8	77.3	97.2	77.4	95.8	77.4	81.0	89.4	80.3	88.8	79.3	152.4	82.8	78.8	149.8	82.7	27.6	24.9	22.9	578	4104
VICTORIA REGIONAL	28.861N	96.930W	115	31.4	34.5	97.9	76.5	95.9	76.6	94.1	76.6	80.3	88.0	79.8	87.5	78.7	149.6	82.3	78.0	146.1	82.0	24.3	20.9	19.1	1164	3247
WACO REGIONAL	31.619N	97.228W	500	24.6	28.2	101.3	74.3	99.2	74.7	97.2	74.8	78.5	91.1	77.9	90.4	75.6	136.5	82.0	74.8	132.9	81.6	24.2	20.9	19.1	2008	2904
WICHITA FALL REGIONAL	33.979N	98.493W	1017	18.6	22.7	103.8	72.6	101.2	72.7	98.6	72.9	77.6	91.8	76.6	90.9	73.8	130.7	82.8	72.7	126.0	81.8	27.1	24.5	21.7	2793	2546
WILLIAM P HOBBY AP	29.638N	95.282W	44	33.6	36.7	96.0	77.1	94.1	77.1	92.6	77.0	80.4	89.0	79.7	88.2	78.5	148.2	83.3	77.5	143.1	82.8	20.6	18.6	16.9	1139	3222
Utah																										*5 sites, 10 more on CD-ROM*
HILL AFB	41.117N	111.967W	4789	8.7	12.2	93.9	60.9	91.3	60.2	89.2	59.8	64.5	84.7	63.3	84.2	58.1	86.3	71.6	55.3	77.9	72.5	23.7	20.3	18.5	5977	953
LOGAN-CACHE AP	41.787N	111.853W	4454	-5.8	0.3	94.8	61.6	92.2	61.0	89.9	60.4	65.1	85.2	63.9	84.3	59.5	89.6	69.9	56.8	81.4	69.5	20.0	16.7	13.2	7196	507
PROVO MUNICIPAL	40.219N	111.723W	4497	7.5	11.9	94.6	62.4	91.2	62.1	89.7	61.8	66.3	87.0	65.0	85.2	59.4	89.4	75.0	57.1	82.3	74.7	24.1	20.1	17.5	5972	778
SALT LAKE CITY INTL	40.778N	111.969W	4225	9.9	14.4	98.0	62.5	95.4	61.9	92.9	61.4	66.2	87.3	65.1	86.3	60.3	91.6	72.2	57.9	84.0	72.8	24.9	20.9	18.4	5466	1262
ST GEORGE MUNICIPAL	37.100N	113.600W	2936	25.2	27.8	106.2	66.2	103.4	65.2	100.4	64.3	69.1	93.5	68.1	92.7	63.2	96.9	75.8	61.0	89.6	77.5	26.7	23.1	19.4	2974	2721
Vermont																										*1 site, 6 more on CD-ROM*
BURLINGTON INTL	44.468N	73.150W	330	-7.3	-1.8	88.3	71.3	85.4	69.8	82.5	68.4	74.3	83.7	72.6	81.3	71.3	116.9	78.6	69.6	110.1	77.2	23.2	20.0	18.2	7227	525
Virginia																										*18 sites, 35 more on CD-ROM*
DANVILLE REGIONAL	36.573N	79.336W	571	18.3	21.7	93.9	74.5	91.3	74.1	89.6	73.4	77.7	89.1	76.5	87.1	74.5	131.7	82.3	73.2	126.2	80.9	18.6	16.4	14.0	3649	1440
DAVISON AAF	38.717N	77.183W	73	15.1	19.2	96.0	75.7	93.1	74.7	90.5	73.7	79.1	90.5	77.5	88.2	75.8	135.4	84.3	74.5	129.4	82.5	21.4	18.1	15.0	4229	1357
DINWIDDIE COUNTY	37.183N	77.500W	193	16.3	19.4	98.0	77.0	94.8	76.1	91.5	74.8	80.5	91.6	79.0	90.5	77.4	143.4	85.9	75.3	133.8	83.7	17.9	15.6	12.8	3682	1610
LANGLEY AFB	37.083N	76.360W	10	20.8	24.9	92.8	76.2	90.5	75.5	88.3	74.7	79.4	86.9	78.2	85.8	77.4	142.6	82.1	76.1	136.2	81.5	23.9	20.3	18.4	3415	1584
LESSBURG EXECUTIVE	39.078N	77.558W	389	14.2	18.2	95.0	76.3	92.2	75.0	90.3	74.2	79.1	90.4	77.8	88.3	75.5	135.4	82.9	74.8	132.4	82.3	22.5	18.8	16.2	4431	1347
LYNCHBURG REGIONAL	37.321N	79.207W	940	15.8	19.7	92.1	73.8	89.6	73.0	87.1	72.0	76.4	86.9	75.2	85.1	73.4	128.6	80.5	72.3	123.9	79.3	17.5	15.4	12.9	4192	1112
MANSAS REGIONAL	38.721N	77.515W	192	11.7	16.1	93.0	74.2	90.7	73.9	88.4	72.9	77.3	88.3	75.9	86.6	73.4	125.1	82.0	72.6	121.6	81.3	21.4	18.6	16.2	4826	1075
NEWPORT NEWS WILLIAMSBURG	37.132N	76.493W	42	20.4	24.3	94.5	77.1	92.0	76.1	89.9	75.2	79.4	90.3	78.2	88.2	76.7	139.4	84.0	75.3	133.0	82.8	20.3	18.4	16.8	3408	1638
NORFOLK INTL	36.903N	76.192W	30	22.9	26.6	93.7	76.7	91.3	76.0	88.9	75.0	79.1	88.5	78.0	87.0	76.7	139.1	83.0	75.5	133.8	81.7	24.4	20.7	18.7	3165	1715
NORFOLK NAS	36.937N	76.289W	17	24.0	27.6	94.4	77.1	91.9	76.3	89.9	75.6	79.9	89.7	78.5	87.9	77.2	141.7	84.1	75.8	134.8	83.0	25.3	21.8	19.1	2972	1889
OCEANA NAS	36.817N	76.033W	23	21.9	25.9	92.9	77.3	90.4	76.3	88.2	75.4	79.4	88.9	78.2	86.8	76.9	140.3	83.9	75.5	133.6	82.3	24.7	20.9	18.9	3262	1594
QUANTICO MCAF	38.504N	77.305W	10	17.0	20.5	92.6	76.3	90.2	75.6	87.9	74.6	79.5	88.9	77.9	87.1	76.8	139.5	85.2	75.1	131.9	83.2	20.1	17.6	15.7	4129	1376
RICHMOND INTL	37.505N	77.320W	164	18.5	22.1	95.1	75.7	92.7	75.1	90.1	74.0	78.4	89.3	77.4	87.8	75.7	135.2	82.8	74.5	129.8	81.5	20.9	18.7	16.7	3634	1564
ROANOKE-BLACKSBURG REGIONAL	37.317N	79.974W	1175	16.8	20.6	92.2	72.7	89.7	71.9	87.3	71.1	75.3	86.4	74.3	84.9	72.2	124.4	79.4	71.1	119.9	78.5	23.0	19.2	16.9	3961	1239
RONALD REAGAN WASHINGTON NATL	38.848N	77.034W	10	17.9	21.6	94.7	75.5	92.0	74.7	89.4	73.5	78.4	89.1	77.3	87.4	75.7	134.4	83.2	74.4	128.7	82.0	23.3	20.0	18.0	3901	1587
SHENANDOAH VALLEY REGIONAL	38.264N	78.896W	1201	11.8	16.4	93.2	73.8	90.9	73.4	88.7	72.7	78.2	87.1	76.7	85.8	75.3	138.7	82.4	73.5	130.1	80.3	17.6	15.3	12.6	4467	1129
VIRGINIA TECH MONTGOMERY EXEC	37.208N	80.408W	2132	11.0	15.9	89.7	72.8	86.5	71.2	83.9	70.5	75.9	83.6	74.4	81.9	73.3	133.9	78.7	72.3	129.7	78.0	20.1	17.7	15.4	4798	787
WASHINGTON DULLES INTL	38.941N	77.464W	290	13.3	17.3	93.5	74.6	90.9	73.7	88.3	72.7	77.5	88.8	76.3	86.4	74.3	129.7	81.7	73.2	124.8	80.5	21.4	18.7	16.7	4591	1188
Washington																										*20 sites, 28 more on CD-ROM*
ARLINGTON MUNICIPAL	48.161N	122.159W	137	19.5	24.5	82.0	66.0	79.3	64.4	75.3	63.0	67.4	80.3	65.6	77.4	62.7	85.5	74.0	60.9	80.2	71.3	20.8	18.0	15.6	5401	62
BELLINGHAM INTL	48.794N	122.537W	149	19.2	23.7	79.4	65.3	76.2	63.9	73.2	62.2	66.7	77.6	64.9	74.6	62.3	84.4	71.8	60.8	79.9	69.3	25.4	20.7	18.3	5339	56
BREMERTON NATL	47.483N	122.767W	444	22.6	26.6	85.6	65.1	81.7	63.5	78.8	62.2	66.5	82.6	64.8	79.6	60.8	80.8	71.9	57.9	72.7	68.7	19.4	17.0	15.0	5641	98
FAIRCHILD AFB	47.633N	117.650W	2461	5.2	11.2	91.8	62.0	89.0	61.3	85.6	60.5	64.9	84.5	63.3	83.0	59.0	81.7	68.0	56.6	74.7	66.2	24.4	20.5	18.2	6792	401
FELTS FIELD	47.683N	117.321W	1953	8.5	14.2	94.4	64.9	91.0	63.7	87.9	62.7	67.4	89.3	65.5	86.6	60.2	83.6	71.8	57.5	75.8	71.4	20.0	17.5	15.1	6122	464
GRAY AFF	47.083N	122.583W	300	20.0	24.5	87.5	65.5	83.1	64.2	79.6	62.8	67.2	83.5	65.5	80.1	61.5	82.6	68.8	60.6	79.8	68.6	18.5	16.0	13.3	5164	153
KING COUNTY INTL AP	47.530N	122.301W	18	25.8	29.4	85.6	65.3	81.9	64.0	79.0	62.7	66.8	82.7	65.2	79.2	61.2	80.8	69.5	59.6	76.2	69.5	18.4	16.3	13.9	4354	268
MCCHORD AFB	47.150N	122.483W	322	20.5	24.5	86.3	65.2	82.4	63.7	79.2	62.5	66.8	82.6	65.2	79.7	61.3	82.1	69.8	59.6	77.2	68.4	20.3	17.8	15.6	5161	134
OLYMPIA REGIONAL	46.973N	122.903W	188	20.4	24.3	87.3	66.0	83.3	64.7	79.7	63.4	67.9	84.5	66.0	80.9	61.7	82.7	71.3	60.2	78.4	69.7	18.8	16.5	14.3	5377	110
PAINE FIELD	47.908N	122.280W	606	25.0	28.9	80.1	63.5	76.3	62.2	73.1	61.0	65.3	76.9	63.5	73.8	61.0	81.8	68.5	59.1	76.4	67.6	24.5	20.2	17.5	5192	81
PEARSON FIELD	45.621N	122.657W	30	24.1	27.2	90.7	66.4	87.2	65.8	83.0	64.6	69.1	86.0	67.4	83.8	63.0	86.1	74.2	61.2	80.8	72.9	16.4	13.3	11.8	4416	382
SANDERSON FIELD	47.238N	123.141W	271	22.3	25.8	87.4	65.2	82.9	64.3	79.0	62.7	67.2	83.5	65.4	80.4	61.2	81.5	69.7	59.6	77.0	68.6	20.4	18.3	16.4	5491	100
SEATTLE-TACOMA INTL	47.444N	122.314W	370	25.4	29.5	85.3	65.1	81.6	63.8	78.2	62.7	66.9	82.4	65.1	78.9	61.1	81.6	70.2	59.7	77.4	68.6	20.4	18.2	16.3	4705	196
SOUTHWEST WASHINGTON REGIONAL	46.117N	122.894W	20	22.9	26.8	87.9	67.2	82.3	65.5	79.4	63.9	68.7	83.9	66.8	81.0	63.0	86.1	75.1	61.2	80.7	72.3	17.2	14.9	12.7	4832	182
SPOKANE INTL	47.622N	117.528W	2353	5.1	11.2	92.9	62.8	89.6	61.8	86.2	60.6	65.2	86.9	63.6	84.8	58.0	78.3	68.5	55.9	72.6	68.0	25.6	21.9	19.0	6627	461
TACOMA NARROWS AP	47.268N	122.576W	292	27.3	30.9	83.6	64.3	80.3	62.9	76.7	61.7	66.0	80.7	64.2	77.1	60.9	80.7	68.5	59.2	75.8	67.2	19.5	17.3	15.4	4792	145
TRI-CITIES AP	46.267N	119.117W	407	10.2	16.4	99.2	69.4	96.2	67.8	92.6	66.7	71.7	93.7	69.7	91.6	64.1	91.0	79.4	62.5	85.9	77.4	24.8	20.8	18.2	4945	815

Table 4-8 Design Conditions for Selected Locations *(Continued)*

Meaning of acronyms:
DB: Dry bulb temperature, °F
WB: Wet bulb temperature, °F
MCWB: Mean coincident wet bulb temperature, °F
Lat: Latitude, °
DP: Dew point temperature, °F
MCDB: Mean coincident dry bulb temperature, °F
Long: Longitude, °
HR: Humidity ratio, grains of moisture per lb of dry air
HDD and CDD 65: Annual heating and cooling degree-days, base 65°F, °F-day
Elev: Elevation, ft
WS: Wind speed, mph

Station	Lat	Long	Elev	Heating DB		Cooling DB/MCWB						Evaporation WB/MCDB				Dehumidification DP/HR/MCDB						Extreme Annual WS			Heat./Cool. Degree-Days	
						0.4%		1%		2%		0.4%		1%		0.4%			1%							
				99.6%	99%	DB /	MCWB	DB /	MCWB	DB /	MCWB	WB /	MCDB	WB /	MCDB	DP /	HR /	MCDB	DP /	HR /	MCDB	1%	2.5%	5%	HDD /	CDD 65
WALLA WALLA REGIONAL	46.095N	118.287W	1166	11.0	17.3	98.5	66.3	94.7	64.9	91.0	63.8	68.4	92.8	66.6	90.3	60.5	82.1	73.7	57.8	74.6	72.6	24.3	20.2	17.9	4810	934
WEST POINT	47.660N	122.440W	10	29.6	33.1	70.5	61.0	68.1	60.3	66.2	59.5	62.5	67.3	61.2	65.8	60.5	78.7	64.3	59.2	75.2	63.0	36.8	31.1	26.3	4927	9
YAKIMA AIR TERMINAL	46.568N	120.543W	1064	8.3	13.8	96.4	66.2	93.3	65.4	89.9	63.9	68.4	91.1	66.6	89.1	60.3	81.2	75.8	58.0	74.7	74.3	22.6	18.9	16.2	5845	556
West Virginia																										*3 sites, 12 more on CD-ROM*
HUNTINGTON TRI-STATE AP	38.365N	82.555W	824	10.8	15.8	91.7	73.8	89.2	73.3	86.9	72.4	77.3	86.6	75.9	84.7	74.6	133.5	81.6	73.2	127.3	80.1	16.8	14.7	12.6	4398	1143
MID-OHIO VALLEY REGIONAL	39.345N	81.439W	831	8.6	13.6	90.6	73.8	88.1	72.8	85.8	71.6	76.7	86.2	75.3	84.0	73.8	130.0	81.1	72.5	124.3	79.6	18.3	16.0	13.8	4886	943
YEAGER AP	38.379N	81.590W	910	10.7	15.8	91.2	73.1	88.8	72.6	86.5	71.8	76.6	85.9	75.3	84.1	73.9	130.8	80.5	72.7	125.2	79.0	17.3	14.7	12.2	4402	1078
Wisconsin																										*14 sites, 42 more on CD-ROM*
APPLETON INTL	44.267N	88.517W	917	-6.0	-0.3	88.4	75.2	84.9	72.8	82.3	70.9	77.8	85.3	75.5	82.6	75.2	136.7	81.6	73.0	127.0	79.6	24.7	21.7	19.0	7210	608
AUSTIN STRAUBEL INTL	44.479N	88.137W	687	-7.9	-2.8	88.0	73.7	85.0	71.7	82.4	70.1	76.3	84.8	74.3	82.0	73.4	127.6	81.4	71.6	119.5	79.6	23.8	20.2	18.2	7590	480
CENTRAL WISCONSIN AP	44.783N	89.667W	1277	-11.0	-7.3	86.5	72.3	83.6	70.5	81.2	68.1	74.3	83.2	72.0	80.6	71.7	122.7	81.2	69.6	114.2	78.8	23.1	19.8	17.7	8303	361
CHIPPEWA VALLEY REGIONAL	44.867N	91.488W	885	-13.1	-8.0	90.0	73.1	86.7	71.0	84.0	69.2	75.9	85.4	73.9	83.0	72.9	126.3	81.5	70.9	117.7	79.5	20.1	18.0	16.3	7843	587
DANE COUNTY REGIONAL	43.141N	89.345W	866	-6.3	-1.2	89.3	74.1	86.6	72.4	83.9	71.0	76.9	86.2	74.9	83.3	73.9	130.3	83.0	72.1	122.7	80.6	21.5	19.0	17.2	7083	640
FOND DU LAC COUNTY AP	43.769N	88.491W	807	-6.3	-1.0	88.9	73.5	85.8	71.5	82.8	70.1	76.3	85.2	74.3	82.6	73.1	126.6	82.1	71.9	121.5	80.4	23.6	20.2	18.3	7173	601
GENERAL MITCHELL INTL	42.955N	87.904W	670	-1.0	3.6	89.6	74.4	86.4	72.3	83.4	70.6	76.6	86.4	74.8	83.3	73.5	127.6	82.0	71.8	120.6	80.4	24.9	21.8	19.5	6676	693
KENOSHA REGIONAL	42.595N	87.938W	743	-2.4	2.2	90.3	74.3	87.5	73.2	84.0	71.2	77.0	86.9	75.1	83.8	73.4	127.7	82.0	72.3	123.0	80.8	24.8	21.5	19.2	6735	629
LA CROSSE MUNICIPAL	43.879N	91.253W	652	-9.2	-4.2	91.3	74.9	88.4	73.0	85.7	71.4	77.8	87.4	75.7	85.0	74.8	133.7	83.6	72.7	124.1	81.5	23.2	19.7	18.0	7026	820
MANITOWOC COUNTY AP	44.133N	87.667W	651	-4.3	0.5	84.7	71.7	81.9	70.4	79.5	68.4	74.9	82.7	72.7	79.7	72.4	122.7	80.3	70.3	114.3	77.6	24.1	20.7	18.8	7573	354
SHEBOYGAN	43.750N	87.690W	577	-1.8	2.8	83.0	71.5	79.5	70.6	76.6	69.8	76.4	79.3	74.2	76.9	75.5	136.3	77.3	73.4	126.8	75.8	41.1	33.8	28.4	7248	340
SHEBOYGAN COUNTY MEMORIAL	43.769N	87.851W	746	-4.5	0.1	88.4	73.9	84.5	71.4	81.9	70.0	76.0	85.0	74.0	82.1	73.0	125.9	81.5	71.6	120.0	79.2	24.1	20.5	18.5	7469	446
WAUSAU DOWNTOWN AP	44.929N	89.628W	1200	-12.0	-7.1	87.6	71.7	84.4	69.4	81.8	67.6	74.4	83.2	72.5	80.8	71.8	122.8	79.0	69.8	114.5	77.4	20.4	18.1	16.2	8025	455
WITTMAN REGIONAL	43.984N	88.557W	782	-6.2	-1.6	88.5	73.6	85.4	71.6	82.4	70.1	76.3	85.0	74.3	82.5	73.2	126.9	81.6	71.9	121.4	80.1	23.0	19.9	18.0	7343	571
Wyoming																										*2 sites, 21 more on CD-ROM*
CASPER-NATRONA COUNTY INTL	42.898N	106.474W	5313	-8.5	-1.0	94.0	59.6	91.2	59.0	88.4	58.5	63.2	82.8	61.8	81.9	57.6	86.3	66.5	55.4	79.7	66.0	32.3	28.2	25.6	7308	469
CHEYENNE REGIONAL	41.150N	104.817W	6130	-4.1	2.6	89.5	58.1	86.9	57.6	84.0	57.2	62.6	77.3	61.5	76.8	58.5	92.3	65.9	56.9	86.9	65.2	33.4	28.7	25.6	7056	355
Canada																										*100 sites, 665 more on CD-ROM*
Alberta																										*13 sites, 114 more on CD-ROM*
BOW ISLAND	49.730N	111.450W	2679	-20.7	-14.4	88.8	64.4	85.4	63.2	81.9	62.3	68.3	82.6	66.2	79.9	63.5	96.9	74.9	61.2	89.1	72.4	28.5	24.9	21.7	8604	200
CALGARY INTL	51.120N	114.010W	3606	-19.2	-12.8	83.4	60.9	79.8	59.9	76.5	58.8	64.1	77.6	62.0	75.9	59.1	85.4	70.0	56.7	78.3	67.1	27.0	23.1	20.2	9197	67
CANADIAN OLYMPIC PARK UPPER	51.080N	114.220W	4052	-17.2	-11.6	82.5	59.4	78.8	58.1	75.3	57.2	63.4	75.1	61.2	73.4	59.2	87.5	68.2	56.7	79.6	65.5	22.8	19.7	17.2	9071	74
EDMONTON CITY CENTRE AWOS	53.570N	113.520W	2203	-20.1	-14.5	83.2	64.4	79.9	62.8	76.8	61.0	66.5	79.6	64.6	76.7	61.9	89.8	72.6	59.7	82.9	70.3	22.4	19.3	16.9	9423	132
EDMONTON INTL	53.310N	113.580W	2373	-26.7	-20.6	82.1	64.3	78.7	62.9	75.8	61.0	67.1	78.4	65.0	75.8	62.8	93.5	73.3	60.5	86.1	70.8	22.9	19.8	17.3	10473	45
EDMONTON NAMAO AWOS	53.670N	113.470W	2257	-22.2	-16.6	82.1	64.3	78.8	62.5	75.6	60.6	66.3	78.5	64.4	75.9	61.8	89.7	72.2	59.7	83.1	69.7	22.9	19.9	17.3	10060	67
FORT MCMURRAY CS	56.650N	111.210W	1210	-30.5	-25.5	83.7	63.8	80.0	61.8	77.0	60.2	66.5	78.4	64.5	75.3	62.5	88.6	69.8	60.3	81.7	68.0	19.4	16.7	14.5	11244	75
GRANDE PRAIRIE	55.180N	118.880W	2195	-32.7	-24.4	81.8	62.2	78.3	60.6	75.2	59.0	64.5	77.8	62.5	74.8	59.8	83.2	69.0	57.6	76.9	66.6	25.3	22.0	18.9	10674	47
LACOMBE CDA 2	52.450N	113.760W	2822	-25.6	-19.0	82.6	64.8	79.0	63.1	75.9	61.3	67.3	78.7	65.0	76.1	63.0	95.8	73.8	60.6	87.8	71.1	21.0	18.1	15.5	10318	41
LETHBRIDGE CDA	49.700N	112.770W	2986	-18.1	-12.1	89.1	62.4	85.4	61.4	81.9	60.7	66.3	81.2	64.3	79.1	61.3	90.6	72.7	58.9	83.0	70.3	29.8	26.5	23.5	8108	205
MEDICINE HAT RCS	50.030N	110.720W	2346	-19.5	-13.4	91.5	62.7	88.0	61.6	84.5	61.0	66.0	82.9	64.2	81.2	60.4	85.4	71.6	58.2	79.1	69.8	24.1	20.8	18.2	8497	322
RED DEER	52.180N	113.890W	2968	-25.6	-18.8	82.3	63.1	78.8	61.4	75.7	60.0	65.6	78.4	63.5	75.5	60.6	88.1	72.0	58.6	81.9	69.5	20.0	18.0	16.1	10309	42
SPRINGBANK	51.100N	114.370W	3940	-24.5	-17.9	80.4	60.1	76.8	58.3	73.7	57.5	62.4	75.7	60.4	73.3	57.2	80.7	68.3	55.2	75.1	65.3	24.8	21.2	18.6	10296	8
British Columbia																										*27 sites, 71 more on CD-ROM*
ABBOTSFORD	49.030N	122.360W	194	18.1	22.9	85.6	67.4	81.9	66.0	78.5	64.5	69.0	83.1	67.0	79.7	63.3	87.8	75.9	61.6	82.5	73.3	20.3	17.1	14.5	5240	145
AGASSIZ RCS	49.240N	121.760W	63	18.7	23.1	86.2	68.5	82.8	67.3	79.6	66.1	70.9	82.6	68.9	80.1	66.5	97.7	77.6	64.6	91.3	74.9	22.7	17.7	13.9	5142	209
BALLENAS ISLAND	49.350N	124.160W	42	30.3	32.9	74.4	66.3	72.1	65.3	70.2	64.1	67.7	72.6	66.1	70.9	65.6	94.8	71.0	64.1	89.6	69.3	35.5	30.5	26.9	4772	102
COMOX	49.720N	124.900W	84	23.5	27.1	80.2	64.0	76.7	62.9	73.6	61.6	65.5	76.9	64.0	74.0	61.2	80.8	68.8	59.9	77.3	67.5	30.0	25.7	21.8	5561	102
DISCOVERY ISLAND	48.420N	123.230W	62	29.6	34.1	72.8	N/A	69.2	N/A	66.3	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	36.2	28.3	22.0	4995	18
ENTRANCE ISLAND	49.210N	123.810W	25	29.5	32.3	74.5	64.8	72.0	63.9	69.9	63.0	65.7	71.0	64.6	70.1	63.4	87.3	69.7	62.0	83.3	68.5	31.9	27.9	25.0	4848	104
ESQUIMALT HARBOUR	48.430N	123.440W	10	27.3	30.7	71.8	60.6	68.7	59.5	66.1	58.6	62.1	68.8	60.9	66.6	59.4	75.5	64.3	58.2	72.5	62.9	22.0	18.9	16.6	5475	11
HOWE SOUND PAM ROCKS	49.490N	123.300W	23	26.9	30.3	76.4	66.3	73.5	64.8	71.3	64.0	68.0	73.8	66.4	71.7	65.6	94.7	72.0	64.0	89.3	70.1	39.9	35.0	29.6	4769	139
KAMLOOPS	50.700N	120.450W	1133	-3.2	4.0	93.3	64.8	89.5	63.7	85.5	62.3	66.6	88.2	64.9	85.2	59.4	78.9	70.8	57.6	73.8	69.8	22.9	20.2	18.1	6359	501
KELOWNA	49.960N	119.380W	1421	0.8	7.3	91.6	64.7	88.1	63.5	84.4	62.1	66.7	86.0	64.8	83.4	60.1	81.6	71.0	58.2	76.4	69.6	17.7	14.6	12.2	7009	252
MALAHAT	48.570N	123.530W	1200	21.6	25.6	81.8	63.2	78.2	61.9	75.1	60.8	65.9	78.1	64.1	74.8	60.8	83.3	73.0	59.2	78.6	70.2	14.8	12.7	10.8	5857	173
PENTICTON	49.460N	119.600W	1130	7.8	12.7	91.2	65.7	87.8	64.5	84.5	63.2	67.4	86.8	65.7	84.1	60.3	81.4	73.4	58.6	76.5	72.3	23.3	20.4	18.2	6166	410
PITT MEADOWS CS	49.210N	122.690W	16	19.2	23.5	86.4	67.7	82.7	66.3	79.2	65.0	69.2	82.8	67.3	79.9	64.1	89.5	74.9	62.3	84.1	71.9	12.2	10.2	8.9	5382	141
POINT ATKINSON	49.330N	123.260W	46	28.7	31.9	76.2	N/A	73.8	N/A	71.9	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	30.1	25.7	21.9	4380	185
PRINCE GEORGE	53.880N	122.680W	2267	-22.4	-13.7	82.5	61.6	78.7	60.0	75.1	58.3	63.2	78.7	61.2	75.3	57.4	76.3	66.5	55.7	71.8	64.9	21.9	18.8	16.7	9244	40
SANDHEADS CS	49.110N	123.300W	36	25.9	29.8	72.0	N/A	70.1	N/A	68.3	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	30.4	26.8	23.8	4982	53
SUMMERLAND CS	49.560N	119.650W	1490	6.8	12.6	91.2	63.8	87.9	63.0	84.4	61.9	66.7	84.8	64.9	82.4	60.6	83.3	72.1	58.5	77.3	70.7	18.3	14.5	11.9	6306	458

Table 4-8 Design Conditions for Selected Locations *(Continued)*

Meaning of acronyms:
DB: Dry bulb temperature, °F *WB: Wet bulb temperature, °F*
MCWB: Mean coincident wet bulb temperature, °F
Lat: Latitude, °
DP: Dew point temperature, °F
MCDB: Mean coincident dry bulb temperature, °F
Long: Longitude, °
HR: Humidity ratio, grains of moisture per lb of dry air
HDD and CDD 65: Annual heating and cooling degree-days, base 65°F, °F-day
Elev: Elevation, ft
WS: Wind speed, mph

Station	Lat	Long	Elev	Heating DB		Cooling DB/MCWB						Evaporation WB/MCDB				Dehumidification DP/HR/MCDB						Extreme Annual WS			Heat./Cool. Degree-Days	
						0.4%		1%		2%		0.4%		1%		0.4%			1%							
				99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	DP	HR	MCDB	DP	HR	MCDB	1%	2.5%	5%	HDD	CDD 65
VANCOUVER HARBOUR CS	49.300N	123.120W	8	26.8	30.5	78.5	64.2	75.9	63.0	73.5	62.0	65.7	75.6	64.4	74.3	61.3	81.1	70.8	60.1	77.5	69.0	N/A	N/A	N/A	4829	127
VANCOUVER INTL	49.190N	123.180W	14	21.0	25.8	77.2	65.2	74.5	64.2	72.3	63.0	66.5	75.0	65.1	73.0	62.9	85.9	71.4	61.6	81.9	69.8	24.1	20.5	17.8	5240	83
VERNON	50.220N	119.190W	1581	2.3	8.2	91.2	65.2	87.5	64.1	83.6	62.7	67.4	84.8	65.6	82.9	62.0	88.1	70.6	60.0	82.0	69.1	17.3	13.1	10.9	6787	369
VICTORIA GONZALES CS	48.410N	123.320W	200	26.6	30.7	76.3	62.4	72.0	60.8	68.9	59.4	63.8	73.4	62.0	69.8	60.0	78.0	66.2	58.8	74.5	64.6	27.7	23.8	20.8	5170	40
VICTORIA HARTLAND CS	48.530N	123.460W	506	25.3	29.1	83.2	65.7	79.9	64.2	76.7	63.1	67.9	79.6	66.2	76.7	63.7	89.9	72.3	62.1	84.9	70.2	20.7	17.8	15.3	5142	171
VICTORIA INTL	48.650N	123.430W	64	24.7	27.9	80.1	63.8	76.3	62.5	73.3	61.1	64.7	77.8	63.1	74.6	59.1	75.1	68.4	57.9	71.8	67.3	20.3	17.0	14.5	5413	46
VICTORIA UNIVERSITY CS	48.460N	123.300W	197	27.2	30.9	80.5	64.6	77.0	63.4	73.9	62.3	66.7	77.2	65.0	74.1	62.9	86.4	70.0	61.3	81.5	68.1	12.6	11.0	9.6	4990	64
WEST VANCOUVER	49.350N	123.190W	558	21.6	26.2	80.7	65.1	77.3	64.3	74.4	63.2	67.3	77.7	65.6	75.0	63.1	88.3	72.8	61.6	83.7	70.9	10.9	9.2	7.7	5398	136
WHITE ROCK CAMPBELL SCI	49.020N	122.780W	43	22.4	26.7	76.7	65.9	73.8	64.5	71.6	63.4	67.4	74.5	65.7	72.1	64.6	91.4	71.5	62.9	86.0	69.2	13.9	11.3	9.2	5018	55
YOHO PARK	51.440N	116.340W	5256	-21.8	-15.8	77.9	56.7	74.1	55.3	70.3	53.9	58.5	72.9	56.7	70.5	53.4	73.9	60.4	51.6	69.1	58.9	23.6	21.2	19.5	11693	2
Manitoba																										*1 site, 41 more on CD-ROM*
WINNIPEG INTL	49.910N	97.240W	783	-25.7	-21.4	86.5	70.3	83.5	68.6	80.7	66.9	73.4	83.1	71.0	80.4	70.2	114.3	79.6	67.6	104.6	76.6	28.4	25.1	22.4	10365	286
New Brunswick																										*3 sites, 19 more on CD-ROM*
FREDERICTON INTL	45.870N	66.540W	68	-9.8	-5.1	85.9	69.8	82.5	67.9	79.6	66.2	72.1	82.1	70.3	79.1	68.8	106.1	76.6	67.2	100.4	74.7	22.2	19.3	17.3	8253	257
MONCTON INTL	46.110N	64.680W	232	-8.0	-3.5	83.6	69.7	80.5	67.6	77.8	66.1	71.9	80.0	70.2	77.3	69.2	108.2	75.9	67.6	102.4	74.2	28.3	24.6	21.6	8397	199
SAINT JOHN	45.320N	65.890W	357	-8.1	-3.1	79.2	65.8	76.3	64.2	73.5	62.6	68.4	75.4	66.6	72.8	65.9	96.8	71.3	64.3	91.5	69.1	27.7	23.9	21.1	8445	60
Newfoundland and Labrador																										*1 site, 41 more on CD-ROM*
ST JOHN'S INTL	47.620N	52.750W	461	4.8	8.6	76.6	66.4	73.9	65.0	71.4	63.8	69.1	73.7	67.3	71.7	67.5	102.7	71.9	65.5	95.7	69.9	35.3	30.1	27.1	8629	60
Northwest Territories																										*1 site, 45 more on CD-ROM*
YELLOWKNIFE	62.460N	114.440W	675	-40.8	-36.9	77.7	60.9	74.7	59.4	71.9	58.3	63.0	72.9	61.3	71.2	59.1	76.8	66.5	57.2	71.6	65.2	20.8	18.5	16.6	14687	66
Nova Scotia																										*3 sites, 34 more on CD-ROM*
HALIFAX STANFIELD INTL	44.880N	63.500W	477	-0.8	3.0	82.2	68.7	79.1	66.9	76.3	65.4	71.2	78.2	69.5	75.6	69.0	108.6	74.1	67.5	102.7	72.1	28.1	24.3	21.2	7655	199
SHEARWATER	44.630N	63.500W	144	1.0	4.9	79.2	67.1	76.3	65.4	73.4	64.3	69.9	75.0	68.2	72.9	68.3	104.4	71.8	66.6	98.3	70.1	26.7	23.2	20.6	7529	130
SYDNEY	46.170N	60.050W	203	0.2	4.7	81.6	68.5	78.6	67.0	75.5	65.4	70.8	78.5	69.0	75.6	68.2	104.2	74.6	66.5	98.3	72.4	27.9	24.5	21.7	8092	158
Nunavut																										*1 site, 60 more on CD-ROM*
IQALUIT CLIMATE	63.750N	68.540W	110	-32.0	-29.4	64.3	54.6	58.7	52.1	54.7	49.6	55.6	63.3	52.6	58.0	50.9	55.5	58.0	48.8	51.3	55.8	29.5	25.6	22.4	17043	1
Ontario																										*21 sites, 75 more on CD-ROM*
BEAUSOLEIL	44.850N	79.870W	600	-12.4	-5.9	85.9	74.0	82.6	71.7	79.7	70.3	75.9	82.6	73.9	79.9	73.9	129.2	79.5	71.9	120.8	77.2	14.0	12.1	10.7	7909	382
BELLE RIVER	42.300N	82.700W	602	6.5	10.6	88.6	75.3	85.7	74.0	82.8	72.4	79.3	84.3	77.0	82.4	78.2	150.0	82.3	75.4	136.1	80.1	28.8	25.3	22.2	6023	762
ERIEAU	42.250N	81.900W	584	5.3	9.3	80.3	73.2	78.5	72.0	77.0	71.1	76.1	78.1	74.5	76.8	75.5	136.5	77.5	73.7	128.4	76.1	28.4	25.0	22.0	6502	507
GUELPH TURFGRASS INSTITUTE	43.547N	80.215W	1066	-6.6	-1.1	85.6	71.1	82.5	69.6	79.9	67.9	73.6	81.7	71.8	79.3	71.3	120.1	77.7	69.5	112.7	75.9	19.0	16.9	15.1	7981	275
HAMILTON INTL	43.170N	79.940W	780	-0.7	4.1	87.1	72.4	84.2	71.1	81.3	69.5	74.9	83.4	73.1	81.1	72.3	123.0	79.2	70.6	115.9	77.4	27.2	23.2	20.6	7066	449
LONDON INTL	43.030N	81.150W	912	-0.8	3.9	86.4	72.4	83.5	71.0	80.9	69.3	74.7	82.8	73.0	80.7	72.2	123.1	79.3	70.4	115.8	77.3	23.8	21.0	18.7	7097	430
NORTH BAY	46.360N	79.420W	1215	-17.4	-12.0	82.3	68.1	79.4	66.5	76.8	65.1	71.2	78.5	69.3	75.3	68.9	111.0	74.4	67.1	104.4	72.8	22.3	19.5	17.5	9272	227
OTTAWA INTL	45.320N	75.670W	374	-11.5	-6.3	87.1	71.3	84.2	69.5	81.2	68.0	73.9	83.1	72.0	80.3	71.0	115.8	78.7	69.2	108.9	76.7	22.6	19.9	17.8	8070	434
PETERBOROUGH TRENT U	44.350N	78.300W	709	-8.3	-2.5	87.4	70.4	84.0	68.8	81.2	67.1	72.8	83.2	71.1	80.5	69.6	111.6	77.6	68.0	105.5	76.2	13.8	12.0	10.7	7782	319
PORT WELLER	43.250N	79.220W	259	8.9	12.6	84.4	73.0	81.7	71.7	79.2	70.5	75.8	80.9	74.0	78.7	74.3	129.5	78.9	72.6	121.9	77.0	31.6	27.7	24.3	6291	558
REGION OF WATERLOO INTL	43.460N	80.380W	1055	-5.0	0.7	87.0	71.5	84.0	70.1	81.0	68.5	73.9	83.1	72.3	80.8	71.2	119.8	78.1	69.6	113.1	76.6	25.7	22.5	19.9	7634	332
SAULT STE MARIE	46.480N	84.510W	630	-12.2	-6.8	83.1	70.0	79.9	68.0	77.1	66.2	72.1	79.8	70.0	77.1	69.4	110.7	76.5	67.6	103.7	74.1	22.9	19.7	17.5	8893	162
SUDBURY	46.630N	80.800W	1143	-17.8	-12.3	84.2	68.3	80.9	66.3	78.0	64.5	70.8	80.1	68.9	76.9	67.8	106.5	74.4	65.9	99.8	72.8	22.6	20.0	18.0	9385	223
THUNDER BAY CS	48.370N	89.330W	654	-19.9	-15.2	84.5	68.4	81.3	66.2	78.1	64.7	70.8	80.3	68.5	77.2	67.3	102.8	76.7	65.2	95.5	73.7	21.1	18.8	16.7	9945	145
TIMMINS	48.570N	81.380W	967	-27.7	-21.6	85.3	67.9	81.6	65.3	78.4	64.0	70.6	81.0	68.3	77.8	66.9	102.7	75.5	65.0	95.8	73.3	18.7	17.4	14.5	10768	158
TORONTO BILLY BISHOP	43.630N	79.400W	252	3.8	8.4	83.1	70.9	80.3	69.9	77.5	68.9	74.0	79.4	72.4	77.5	72.3	120.6	77.1	70.7	114.0	75.6	29.7	26.2	23.2	6627	432
TORONTO BUTTONVILLE	43.860N	79.370W	650	-3.4	1.8	88.9	72.2	85.5	70.3	82.4	68.8	74.4	85.0	72.5	82.2	71.0	117.1	79.8	69.3	110.2	77.8	21.1	18.7	16.8	7232	475
TORONTO PEARSON INTL	43.680N	79.630W	569	-0.5	4.1	88.4	72.4	85.2	70.6	82.1	69.1	74.7	84.6	72.9	82.2	71.6	119.3	80.1	69.8	111.9	78.2	27.3	23.6	20.9	6907	547
TRENTON	44.120N	77.530W	283	-7.0	-1.6	84.5	71.8	81.8	70.3	79.4	68.9	74.3	81.3	72.6	79.2	71.9	119.1	78.6	70.2	112.5	76.8	23.4	20.4	17.9	7450	375
WINDSOR	42.280N	82.960W	622	4.0	8.6	89.5	73.6	86.7	72.2	84.2	70.7	76.2	85.7	74.4	83.1	73.3	126.6	81.6	71.6	119.6	79.3	25.5	22.5	20.1	6158	788
Prince Edward Island																										*1 site, 6 more on CD-ROM*
CHARLOTTETOWN	46.290N	63.120W	159	-4.3	0.1	80.4	69.4	77.8	67.4	75.3	65.9	71.2	77.7	69.5	75.2	69.0	107.1	75.0	67.3	100.9	73.2	26.5	22.9	20.3	8230	194
Québec																										*22 sites, 81 more on CD-ROM*
BAGOTVILLE	48.330N	71.000W	522	-21.6	-16.8	84.8	67.0	81.3	65.3	78.0	64.0	70.0	79.6	68.0	77.2	66.6	100.0	73.7	64.9	93.9	72.0	26.9	23.6	21.0	10092	183
BIG TROUT LAKE	53.820N	89.900W	730	-34.0	-29.8	80.6	65.7	77.5	63.8	74.4	62.5	68.5	76.4	66.3	74.1	65.7	97.4	72.2	63.3	89.3	69.8	20.3	17.9	15.9	13229	94
JONQUIERE	48.430N	71.140W	445	-22.4	-16.9	84.2	67.6	80.8	66.1	77.6	64.8	71.1	79.4	69.2	76.7	68.4	106.0	75.2	66.5	99.1	72.8	23.3	20.8	18.7	9914	177
LA BAIE	48.300N	70.920W	498	-22.4	-17.8	84.4	67.4	80.7	66.2	77.4	64.9	71.0	79.5	69.0	76.7	68.2	105.7	74.8	66.3	98.7	72.6	22.8	20.1	17.8	10224	128
LAC SAINT-PIERRE	46.180N	72.920W	53	-11.6	-6.4	82.3	70.0	79.6	68.6	77.3	67.3	72.7	78.8	71.1	77.0	70.6	112.7	76.5	69.0	106.8	75.1	30.1	26.8	23.9	8230	359
L'ACADIE	45.290N	73.350W	144	-11.8	-7.1	86.2	70.8	83.4	69.7	80.8	68.3	74.4	82.0	72.4	79.4	71.8	118.2	78.9	70.1	111.5	76.7	23.0	19.8	17.0	7936	412
L'ASSOMPTION	45.810N	73.430W	69	-14.1	-8.6	86.8	71.4	83.8	69.6	81.0	68.1	74.2	82.9	72.2	80.0	71.4	116.2	78.6	69.6	109.3	76.7	18.8	16.3	14.3	8288	380
LENNOXVILLE	45.370N	71.820W	594	-13.7	-7.9	84.9	70.7	82.2	69.2	79.7	67.7	73.6	81.2	71.7	79.0	71.2	117.6	77.7	69.2	109.8	75.7	20.0	17.6	15.5	8253	272

Table 4-8 Design Conditions for Selected Locations *(Continued)*

Meaning of acronyms:
DB: Dry bulb temperature, °F *WB: Wet bulb temperature, °F* *Lat: Latitude, °* *Long: Longitude, °* *Elev: Elevation, ft*
DP: Dew point temperature, °F *HR: Humidity ratio, grains of moisture per lb of dry air* *WS: Wind speed, mph*
MCWB: Mean coincident wet bulb temperature, °F *MCDB: Mean coincident dry bulb temperature, °F* *HDD and CDD 65: Annual heating and cooling degree-days, base 65°F, °F-day*

Station	Lat	Long	Elev	Heating DB		Cooling DB/MCWB						Evaporation WB/MCDB				Dehumidification DP/HR/MCDB						Extreme Annual WS			Heat./Cool. Degree-Days	
						0.4%		1%		2%		0.4%		1%		0.4%			1%							
				99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		WB / MCDB		WB / MCDB		DP / HR / MCDB			DP / HR / MCDB			1%	2.5%	5%	HDD / CDD 65	
MONT-JOLI	48.610N	68.210W	172	-10.7	-6.6	80.3	67.6	77.2	65.7	74.5	64.1	69.3	77.5	67.3	75.0	66.0	96.5	74.6	64.1	90.2	72.3	28.4	25.1	22.4	9511	130
MONT-ORFORD	45.310N	72.240W	2776	-19.4	-13.4	77.2	65.4	74.3	64.0	71.6	62.9	69.0	73.6	66.8	70.7	67.5	112.1	71.3	65.6	104.7	69.0	35.1	30.3	27.2	10186	96
MONTREAL MCTAVISH	45.510N	73.580W	238	-7.5	-2.5	86.2	71.4	83.4	69.6	81.0	68.1	73.9	82.7	72.0	79.9	71.0	115.1	79.0	69.3	108.7	77.3	11.2	9.8	8.8	7463	547
MONTREAL MIRABEL INTL	45.680N	74.040W	270	-14.8	-9.3	85.2	71.4	82.4	69.4	79.8	67.9	73.5	82.2	71.6	79.5	70.6	113.7	78.9	68.7	106.6	76.4	18.6	16.0	13.9	8509	323
MONTREAL ST-HUBERT	45.520N	73.420W	90	-10.2	-5.4	86.1	71.8	83.4	70.0	80.9	68.6	74.3	82.6	72.4	80.1	71.6	116.9	78.9	69.8	109.9	77.2	25.2	22.2	19.8	7945	423
MONTREAL TRUDEAU INTL	45.470N	73.740W	118	-9.5	-4.6	86.0	71.7	83.3	69.9	80.9	68.4	73.8	82.7	72.0	80.0	70.8	114.2	78.8	69.3	108.1	77.2	25.7	22.4	19.9	7771	490
NICOLET	46.230N	72.660W	26	-14.1	-9.0	83.8	72.2	81.0	70.4	78.5	68.8	74.3	81.1	72.3	78.6	72.0	118.3	78.6	70.1	111.0	76.3	21.1	18.1	15.7	8436	302
POINTE-AU-PERE INRS	48.510N	68.470W	16	-7.2	-2.6	73.5	65.4	70.7	63.7	68.3	61.9	67.3	71.8	64.9	69.4	65.3	93.4	70.6	62.9	85.8	68.2	28.7	25.2	22.4	9554	20
QUEBEC CITY JEAN LESAGE INTL	46.800N	71.380W	197	-12.0	-7.4	83.7	69.5	81.2	68.3	78.4	66.5	72.8	80.6	70.8	77.9	70.1	111.5	77.5	68.1	104.1	75.2	22.3	19.8	17.8	8783	247
SAINTE-FOY U LAVAL	46.780N	71.290W	300	-11.6	-6.9	84.3	69.1	81.5	67.3	78.7	65.7	72.4	80.3	70.4	77.5	69.8	110.8	76.3	68.0	104.1	74.3	20.8	17.9	15.2	8675	270
SHERBROOKE	45.440N	71.690W	792	-17.2	-11.5	83.8	70.0	81.1	68.3	78.7	66.9	72.6	80.4	70.7	78.2	69.9	113.3	77.5	68.1	106.1	75.0	20.4	17.8	15.5	8843	185
ST-ANICET 1	45.120N	74.290W	161	-12.3	-7.0	87.0	72.8	84.3	71.2	81.7	69.6	75.5	83.7	73.7	81.4	72.9	122.7	80.4	71.1	115.4	78.2	20.7	18.0	15.9	7950	393
STE-ANNE-DE-BELLEVUE 1	45.430N	73.930W	128	-10.8	-5.5	86.0	71.2	83.3	69.8	80.7	68.3	74.2	82.4	72.3	79.7	71.5	116.9	78.6	69.9	110.4	76.3	19.8	17.5	15.6	7928	416
TROIS-RIVIERES	46.350N	72.520W	20	-11.6	-6.8	81.2	70.3	79.0	69.4	76.9	68.3	73.2	78.2	71.6	76.7	71.4	115.8	76.5	69.8	109.8	75.2	24.1	20.9	18.5	8298	332
VARENNES	45.720N	73.380W	59	-10.3	-5.7	86.5	71.2	83.5	69.6	80.9	68.1	74.2	82.5	72.3	80.0	71.5	116.5	78.8	69.8	109.7	76.8	24.5	21.2	18.9	8059	373
Saskatchewan																									*5 sites, 47 more on CD-ROM*	
MOOSE JAW CS	50.330N	105.540W	1893	-22.2	-17.3	89.2	65.5	85.4	64.6	81.8	63.4	71.0	80.8	68.4	78.8	67.9	109.9	76.5	65.2	99.8	72.9	27.5	24.6	22.0	9705	209
PRINCE ALBERT	53.210N	105.670W	1405	-32.2	-26.3	84.0	66.1	80.6	64.4	77.7	62.6	68.5	79.9	66.4	77.3	64.4	95.5	73.7	62.4	88.9	71.3	20.8	18.5	16.7	11167	120
REGINA RCS	50.430N	104.670W	1894	-26.1	-21.2	87.9	67.2	83.9	65.3	80.7	63.8	71.1	82.1	68.3	79.1	67.5	108.4	76.8	64.6	97.9	73.9	28.1	25.1	22.5	10393	199
SASKATOON INTL	52.170N	106.700W	1654	-29.5	-24.0	86.4	66.1	82.8	64.6	79.5	63.4	69.3	81.2	67.2	78.6	65.3	99.6	75.0	63.2	92.1	72.4	24.9	21.9	19.5	10628	166
SASKATOON KERNEN FARM	52.150N	106.550W	1673	-28.3	-23.0	87.2	63.8	83.4	62.4	80.2	61.0	68.9	80.5	66.5	76.9	65.1	98.8	74.6	62.6	90.3	71.6	24.0	21.2	19.0	10626	182
Yukon Territory																									*1 site, 17 more on CD-ROM*	
WHITEHORSE	60.710N	135.070W	2317	-39.4	-30.2	78.3	57.5	74.1	56.0	70.4	54.4	58.8	74.7	57.0	71.6	52.3	63.3	61.2	50.5	59.4	60.3	23.0	20.7	18.6	12188	13

- Wet-bulb temperature corresponding to 0.4, 2.0, 5.0, and 10.0% cumulative frequency of occurrence for indicated month, °F; mean coincident dry-bulb temperature, °F.

For a 30-day month, the 0.4, 2.0, 5.0 and 10.0% values of occurrence represent the value that occurs or is exceeded for a total of 3, 14, 36, or 72 h, respectively, per month on average over the period of record. Monthly percentile values of dry- or wet-bulb temperature may be higher or lower than the annual design conditions corresponding to the same nominal percentile, depending on the month and the seasonal distribution of the parameter at that location. Generally, for the hottest or most humid months of the year, the monthly percentile value exceeds the design condition for the same element corresponding to the same nominal percentile. For example, Table 4-4 shows that the annual 0.4% design dry-bulb temperature at Atlanta, GA, is 93.9°F; the 0.4% monthly dry-bulb temperature exceeds 93.4°F for June, July, and August, with values of 94.5°F, 97.8°F, and 97.4°F, respectively. Two new percentiles were added to this chapter (5.0 and 10.0% values) to give a greater range in the frequency of occurrence, in particular providing less extreme options to select for design calculations.

A general, very approximate rule of thumb is that the *n*% annual cooling design condition is roughly equivalent to the *5n*% monthly cooling condition for the hottest month; that is, the 0.4% annual design dry-bulb temperature is roughly equivalent to the 2% monthly design dry-bulb temperature for the hottest month; the 1% annual value is roughly equivalent to the 5% monthly value for the hottest month, and the 2% annual value is roughly equivalent to the 10% monthly value for the hottest month.

Mean Daily Temperature Range. These values are useful in calculating daily dry- and wet-bulb temperature profiles, as explained in the section on Generating Design-Day Data. Three kinds of profiles are defined:

- Mean daily temperature range for month indicated, °F (defined as mean of difference between daily maximum and minimum dry-bulb temperatures).
- Mean daily dry- and wet-bulb temperature ranges coincident with the 5% monthly design dry-bulb temperature. This is the difference between daily maximum and minimum dry- or wet-bulb temperatures, respectively, averaged over all days where the maximum daily dry-bulb temperature exceeds the 5% monthly design dry-bulb temperature.
- Mean daily dry- and wet-bulb temperature ranges coincident with the 5% monthly design wet-bulb temperature. This is the difference between daily maximum and minimum dry- or wet-bulb temperatures, respectively, averaged over all days where the maximum daily wet-bulb temperature exceeds the 5% monthly design wet-bulb temperature.

Clear-Sky Solar Irradiance. Clear-sky irradiance parameters are useful in calculating solar-related air conditioning loads for any time of any day of the year. Parameters are provided for the 21st day of each month. The 21st of the month is usually a convenient day for solar calculations because June 21 and December 21 represent the solstices (longest and shortest days) and March 21 and September 21 are close to the equinox (days and nights have the same length). Parameters listed in the tables are

- Clear-sky optical depths for beam and diffuse irradiances, which are used to calculate beam and diffuse irradiance as explained in the section on Calculating Clear-Sky Solar Radiation.
- Clear-sky beam normal and diffuse horizontal irradiances at solar noon. These two values can be calculated from the clear-sky optical depths but are listed here for convenience.

All-Sky Solar Radiation. All-sky solar radiation parameters are useful for evaluating the potential of solar technologies (e.g., solar heating, photovoltaics), which are valuable in the design of net zero energy buildings. Parameters listed in the tables are

- Monthly average daily global radiation on a horizontal surface. This is a traditional way to characterize the solar resource at a site.
- Standard deviation of monthly average daily radiation on a horizontal surface. This parameter gives an idea of the year-to-year variability of the solar resource at the site.

Differences from Previously Published Design Conditions

- Climatic design conditions in this chapter are generally similar to those in previous editions, because similar if not identical analysis procedures were used. There are some differences, however, owing to a more recent period of record (generally 1990–2014 versus 1982–2006). For example, compared to the 2009 edition, 99.6% heating dry-bulb temperatures have increased by 0.09°F on average, and 0.4% cooling dry-bulb temperatures have increased by 0.15°F on average. Similar trends are observed for other design temperatures. The root mean square differences are 1.13°F for the 99.6% heating dry-bulb values and 0.65°F for 0.4% cooling dry-bulb. The increases noted here are generally consistent with other observations of climate change.

Because of the changes noted above, the weather data used in many of the example problems and the end of chapter problems solved in the Solutions Manual have not been changed from what was used in the previous editions of this book. This allows the user to substitute current weather data (given in this chapter) and observe the effect of climate change on the design and operation of air-conditioning systems.

Applicability and Characteristics of Design Conditions

Climatic design values in this chapter represent different psychrometric conditions. Design data based on dry-bulb temperature represent peak occurrences of the sensible component of ambient outdoor conditions. Design values based on wet-bulb temperature are related to the enthalpy of the outdoor air. Conditions based on dew point relate to the peaks of the humidity ratio. The designer, engineer, or other user must decide which set(s) of

conditions and probability of occurrence apply to the design situation under consideration.

Annual Heating and Humidification Design Conditions. The month with the lowest mean dry-bulb temperature is used, for example, to determine the time of year where the maximum heating load occurs.

The 99.6 and 99.0% design conditions are often used in sizing heating equipment.

The humidification dew point and mean coincident dry-bulb temperatures and humidity ratio provide information for cold-season humidification applications.

Wind design data provide information for estimating peak loads accounting for infiltration: extreme wind speeds for the coldest month, with the mean coincident dry-bulb temperature; and mean wind speed and direction coincident to the 99.6% design dry-bulb temperature.

Annual Cooling, Dehumidification, and Enthalpy Design Conditions. The month with the highest mean dry-bulb temperature is used, for example, to determine the time of year where the maximum sensible cooling load occurs, not taking into account solar loads.

The mean daily dry-bulb temperature range for the hottest month is the mean difference between the daily maximum and minimum temperatures during the hottest month and is calculated from the extremes of the hourly temperature observations. The true maximum and minimum temperatures for any day generally occur between hourly readings. Thus, the mean maximum and minimum temperatures calculated in this way are about 1°F less extreme than the mean daily extreme temperatures observed with maximum and minimum thermometers. This results in the true daily temperature range generally about 2°F greater than that calculated from hourly data. The mean daily dry-bulb temperature range is used in cooling load calculations.

The 0.4, 1.0, and 2.0% dry-bulb temperatures and mean coincident wet-bulb temperatures often represent conditions on hot, mostly sunny days. These are often used in sizing cooling equipment such as chillers or air-conditioning units.

Design conditions based on wet-bulb temperature represent extremes of the total sensible plus latent heat of outdoor air. This information is useful for design of cooling towers, evaporative coolers, and outdoor air ventilation systems.

The mean wind speed and direction coincident with the 0.4% design dry-bulb temperature is used for estimating peak loads accounting for infiltration.

Design conditions based on dew-point temperatures are directly related to extremes of humidity ratio, which represent peak moisture loads from the weather. Extreme dew-point conditions may occur on days with moderate dry-bulb temperatures, resulting in high relative humidity. These values are especially useful for humidity control applications, such as desiccant cooling and dehumidification, cooling-based dehumidification, and fresh-air ventilation systems. The values are also used as a check point when analyzing the behavior of cooling systems at part-load conditions, particularly when such systems are used for humidity control as a secondary function. Humidity ratio values are calculated from the corresponding dew-point temperature and the standard pressure at the location's elevation.

Annual enthalpy design conditions give the annual enthalpy for the cooling season; this is used for calculating cooling loads caused by infiltration and/or ventilation into buildings. Enthalpy represents the total heat content of air (the sum of its sensible and latent energies). Cooling loads can be calculated knowing the conditions of both the outdoor ambient and the building's interior air.

Extreme Annual Design Conditions. Extreme annual design wind speeds are used in designing smoke management systems.

General Design Conditions Discussion.

Minimum temperatures usually occur between 6:00 A.M. and 8:00 A.M. suntime on clear days when the daily range is greatest. For residential or other applications where the occupancy is continuous throughout the day, the recommended design temperatures apply. With commercial applications or other applications where occupancy is only during hours near the middle of the day, design temperatures above the recommended minimum may apply.

Maximum temperatures usually occur between 2:00 P.M. and 4:00 P.M. suntime with deviations on cloudy days when the daily range is less. Typically, the design dry-bulb temperatures should be used with the coincident wet-bulb temperatures in computing building cooling loads. For residential or other applications where the occupancy is continuous throughout the day, the recommended design temperatures apply. For commercial applications or other applications where occupancy is only during hours near the middle of the day, design temperatures below the recommended maximum might apply. In some cases, the peak occupancy load occurs before the effect of the outdoor maximum temperature has reached the space by conduction through the building mass. In other cases, the peak occupancy loads may be in months other than the three or four summer months when the maximum outdoor temperature is expected; here design temperatures from other months will apply.

When determining the heat loss for below grade components of a building (e.g., basement walls and floor), the average winter outdoor air temperature needed for the current ASHRAE loads methodology. Thus, Table 4-8 is provided. The heating load for the below grade structure also uses the amplitude of the ground temperature to determine the design ground surface temperature (see Chapter 6).

Although not a direct design criterion, another environmental index of general interest is the wind chill index. The wind chill index reliably expresses combined effects of temperature and wind on subjective discomfort. However, rather than using the numerical value of the wind chill index, meteorologists use an index derived from the WCI called the equivalent wind chill temperature. Table 4-9 shows a typical wind chill chart, expressed in equivalent wind chill temperature.

Table 4-8 Average Winter Temperature and Yearly Degree Days[a,b,c]

State	Station		Avg. Winter Temp,[d] °F	Degree-Days Yearly Total
Ala.	Birmingham	A	54.2	2551
	Huntsville	A	51.3	3070
	Mobile	A	59.9	1560
	Montgomery	A	55.4	2291
Alaska	Anchorage	A	23.0	10,864
	Fairbanks	A	6.7	14,276
	Juneau	A	32.1	9075
	Nome	A	13.1	14,171
Ariz.	Flagstaff	A	35.6	7152
	Phoenix	A	58.5	1765
	Tucson	A	58.1	1800
	Winslow	A	43.0	4782
	Yuma	A	64.2	974
Ark.	Fort Smith	A	50.3	3292
	Little Rock	A	50.5	3219
	Texarkana	A	54.2	2533
Calif.	Bakersfield	A	55.4	2122
	Bishop	A	46.0	4275
	Blue Canyon	A	42.2	5596
	Burbank	A	58.6	1646
	Eureka	C	49.9	4643
	Fresno	A	53.3	2611
	Long Beach	A	57.8	1803
	Los Angeles	A	57.4	2061
	Los Angeles	C	60.3	1349
	Mt. Shasta	C	41.2	5722
	Oakland	A	53.5	2870
	Red Bluff	A	53.8	2515
	Sacramento	A	53.9	2502
	Sacramento	C	54.4	2419
	Sandberg	C	46.8	4209
	San Diego	A	59.5	1458
	San Francisco	A	53.4	3015
	San Francisco	C	55.1	3001
	Santa Maria	A	54.3	2967
Colo.	Alamosa	A	29.7	8529
	Colorado Springs	A	37.3	6423
	Denver	A	37.6	6283
	Denver	C	40.8	5524
	Grand Junction	A	39.3	5641
	Pueblo	A	40.4	5462
Conn.	Bridgeport	A	39.9	5617
	Hartford	A	37.3	6235
	New Haven	A	39.0	5897
Del.	Wilmington	A	42.5	4930
D.C.	Washington	A	45.7	4224
Fla.	Apalachicola	C	61.2	1308
	Daytona Beach	A	64.5	879
	Fort Myers	A	68.6	442
	Jacksonville	A	61.9	1239
	Key West	A	73.1	108
	Lakeland	C	66.7	661
	Miami	A	71.1	214
	Miami Beach	C	72.5	141
Fla. (Cont'd	Orlando	A	65.7	766
	Pensacola	A	60.4	1463
	Tallahassee	A	60.1	1485
	Tampa	A	66.4	683
	West Palm Beach	A	68.4	253
Ga.	Athens	A	51.8	2929
	Atlanta	A	51.7	2961
	Augusta	A	54.5	2397
	Columbus	A	54.8	2383
	Macon	A	56.2	2136
	Rome	A	49.9	3326
	Savannah	A	57.8	1819
	Thomasville	C	60.0	1529
Hawaii	Lihue	A	72.7	0
	Honolulu	A	74.2	0
	Hilo	A	71.9	0
Idaho	Boise	A	39.7	5809
	Lewiston	A	41.0	5542
	Pocatello	A	34.8	7033
Ill.	Cairo	C	47.9	3821
	Chicago (O'Hare)	A	35.8	6639
	Chicago (Midway)	A	37.5	6155
	Chicago	C	38.9	5882
	Moline	A	36.4	6408
	Peoria	A	38.1	6025
	Rockford	A	34.8	6830
	Springfield	A	40.6	5429
Ind.	Evansville	A	45.0	4435
	Fort Wayne	A	37.3	6205
	Indianapolis	A	39.6	5699
	South Bend	A	36.6	6439
Iowa	Burlington	A	37.6	6114
	Des Moines	A	35.5	6588
	Dubuque	A	32.7	7376
	Sioux City	A	34.0	6951
	Waterloo	A	32.6	7320
Kans.	Concordia	A	40.4	5479
	Dodge City	A	42.5	4986
	Goodland	A	37.8	6141
	Topeka	A	41.7	5182
	Wichita	A	44.2	4620
Ky.	Covington	A	41.1	5265
	Lexington	A	43.8	4683
	Louisville	A	44.0	4660
La.	Alexandria	A	57.5	1921
	Baton Rouge	A	59.8	1560
	Lake Charles	A	60.5	1459
	New Orleans	A	61.0	1385
	New Orleans	C	61.8	1254
	Shreveport	A	56.2	2184
Mass.	Boston	A	40.0	5634
	Nantucket	A	40.2	5891

[a] Data for US cities from a publication of the US Weather Bureau, *Monthly Normals of Temperature, Precipitation and Heating Degree Days,* 1962, are for the period 1931 to 1960 inclusive. These data also include information from the 1963 revisions to this publication, where available.

[b] Data for airport station, A, and city stations, C, are both given where available.

[c] Data for Canadian cities were computed by the Climatology Division, Department of Transport, from normal monthly mean temperatures, and the monthly values of heating days data were obtained using the National Research Council computer and a method devised by H.C.S. Thom of the US Weather Bureau. The heating days are based on the period from 1931 to 1960.

[d] For period October to April, inclusive.

Table 4-8 Average Winter Temperature and Yearly Degree Days[a,b,c] *(Continued)*

State	Station		Avg. Winter Temp,[d] °F	Degree-Days Yearly Total
Mass. (Cont'd	Pittsfield	A	32.6	7578
	Worcester	A	34.7	6969
Md.	Baltimore	A	43.7	4654
	Baltimore	C	46.2	4111
	Frederick	A	42.0	5087
Me.	Caribou	A	24.2	9767
	Portland	A	33.0	7511
Mich.	Alpena	A	29.7	8506
	Detroit (City)	A	37.2	6232
	Detroit (Wayne)	A	37.1	6293
	Detroit (Willow Run)	A	37.2	6258
	Escanaba	C	29.6	8481
	Flint	A	33.1	7377
	Grand Rapids	A	34.9	6894
	Lansing	A	34.8	6909
	Marquette	C	30.2	8393
	Muskegon	A	36.0	6696
	Sault Ste. Marie	A	27.7	9048
Minn.	Duluth	A	23.4	10,000
	Minneapolis	A	28.3	8382
	Rochester	A	28.8	8295
Miss.	Jackson	A	55.7	2239
	Meridian	A	55.4	2289
	Vicksburg	C	56.9	2041
Mo.	Columbia	A	42.3	5046
	Kansas City	A	43.9	4711
	St. Joseph	A	40.3	5484
	St. Louis	A	43.1	4900
	St. Louis	C	44.8	4484
	Springfield	A	44.5	4900
Mont.	Billings	A	34.5	7049
	Glasgow	A	26.4	8996
	Great Falls	A	32.8	7750
	Havre	A	28.1	8700
	Havre	C	29.8	8182
	Helena	A	31.1	8129
	Kalispell	A	31.4	8191
	Miles City	A	31.2	7723
	Missoula	A	31.5	8125
Neb.	Grand Island	A	36.0	6530
	Lincoln	C	38.8	5864
	Norfolk	A	34.0	6979
	North Platte	A	35.5	6684
	Omaha	A	35.6	6612
	Scottsbluff	A	35.9	6673
	Valentine	A	32.6	7425
Nev.	Elko	A	34.0	7433
	Ely	A	33.1	7733
	Las Vegas	A	53.3	2709
	Reno	A	39.3	6332
	Winnemucca	A	36.7	6761
N.H.	Concord	A	33.0	7383
	Mt. Washington Obsv.		15.2	13,817
N.J.	Atlantic City	A	43.2	4812
	Newark	A	42.8	4589
	Trenton	C	42.4	4980
N.M.	Albuquerque	A	45.0	4348
	Clayton	A	42.0	5158
N.M. (Cont'd	Raton	A	38.1	6228
	Roswell	A	47.5	3793
	Silver City	A	48.0	3705
N.Y.	Albany	A	34.6	6875
	Albany	C	37.2	6201
	Binghamton	A	33.9	7286
	Binghamton	C	36.6	6451
	Buffalo	A	34.5	7062
	New York (Cent. Park)	C	42.8	4871
	New York (La Guardia)	A	43.1	4811
	New York (Kennedy)	A	41.4	5219
	Rochester	A	35.4	6748
	Schenectady	C	35.4	6650
	Syracuse	A	35.2	6756
N.C.	Asheville	C	46.7	4042
	Cape Hatteras		53.3	2612
	Charlotte	A	50.4	3191
	Greensboro	A	47.5	3805
	Raleigh	A	49.4	3393
	Wilmington	A	54.6	2347
	Winston-Salem	A	48.4	3595
N.D.	Bismarck	A	26.6	8851
	Devils Lake	C	22.4	9901
	Fargo	A	24.8	9226
	Williston	A	25.2	9243
Ohio	Akron-Canton	A	38.1	6037
	Cincinnati	C	45.1	4410
	Cleveland	A	37.2	6351
	Columbus	A	39.7	5660
	Columbus	C	41.5	5211
	Dayton	A	39.8	5622
	Mansfield	A	36.9	6403
	Sandusky	C	39.1	5796
	Toledo	A	36.4	6494
	Youngstown	A	36.8	6417
Okla.	Oklahoma City	A	48.3	3725
	Tulsa	A	47.7	3860
Ore.	Astoria	A	45.6	5186
	Burns	C	35.9	6957
	Eugene	A	45.6	4726
	Meacham	A	34.2	7874
	Medford	A	43.2	5008
	Pendleton	A	42.6	5127
	Portland	A	45.6	4635
	Portland	C	47.4	4109
	Roseburg	A	46.3	4491
	Salem	A	45.4	4486
Pa.	Allentown	A	38.9	5810
	Erie	A	36.8	6451
	Harrisburg	A	41.2	5251
	Philadelphia	A	41.8	5144
	Philadelphia	C	44.5	4486
	Pittsburgh	A	38.4	5987
	Pittsburgh	C	42.2	5053
	Reading	C	42.4	4945
	Scranton	A	37.2	6254
	Williamsport	A	38.5	5934
R.I.	Block Island	A	40.1	5804
	Providence	A	38.8	5954
S.C.	Charleston	A	56.4	2033
	Charleston	C	57.9	1794

Table 4-8 Average Winter Temperature and Yearly Degree Days[a,b,c] *(Continued)*

State	Station		Avg. Winter Temp,[d] °F	Degree-Days Yearly Total
S.C. (Cont'd	Columbia	A	54.0	2484
	Florence	A	54.5	2387
	Greenville-Spartenburg	A	51.6	2980
S.D.	Huron	A	28.8	8223
	Rapid City	A	33.4	7345
	Sioux Falls	A	30.6	7839
Tenn.	Bristol	A	46.2	4143
	Chattanooga	A	50.3	3254
	Knoxville	A	49.2	3494
	Memphis	A	50.5	3232
	Memphis	C	51.6	3015
	Nashville	A	48.9	3578
	Oak Ridge	C	47.7	3817
Tex.	Abilene	A	53.9	2624
	Amarillo	A	47.0	3985
	Austin	A	59.1	1711
	Brownsville	A	67.7	600
	Corpus Christi	A	64.6	914
	Dallas	A	55.3	2363
	El Paso	A	52.9	2700
	Fort Worth	A	55.1	2405
	Galveston	A	62.2	1274
	Galveston	C	62.0	1235
	Houston	A	61.0	1396
	Houston	C	62.0	1278
	Laredo	A	66.0	797
	Lubbock	A	48.8	3578
	Midland	A	53.8	2591
	Port Arthur	A	60.5	1447
	San Angelo	A	56.0	2255
	San Antonio	A	60.1	1546
	Victoria	A	62.7	1173
	Waco	A	57.2	2030
	Wichita Falls	A	53.0	2832
Utah	Milford	A	36.5	6497
	Salt Lake City	A	38.4	6052
	Wendover	A	39.1	5778
Vt.	Burlington	A	29.4	8269
Va.	Cape Henry	C	50.0	3279
	Lynchburg	A	46.0	4166
	Norfolk	A	49.2	3421
	Richmond	A	47.3	3865
	Roanoke	A	46.1	4150
Wash.	Olympia	A	44.2	5236
	Seattle-Tacoma	A	44.2	5145
	Seattle	C	46.9	4424
	Spokane	A	36.5	6655
	Walla Walla	C	43.8	4805
	Yakima	A	39.1	5941
W.Va.	Charleston	A	44.8	4476
	Elkins	A	40.1	5675
	Huntington	A	45.0	4446
	Parkersburg	C	43.5	4754
Wisc.	Green Bay	A	30.3	8029
	La Crosse	A	31.5	7589
	Madison	A	30.9	7893
	Milwaukee	A	32.6	7635
Wyo.	Casper	A	33.4	7410
	Cheyenne	A	34.2	7381
	Lander	A	31.4	7870
	Sheridan	A	32.5	7680
Alta.	Banff	C	—	10,551
	Calgary	A	—	9703
	Edmonton	A	—	10,268
	Lethbridge	A	—	8644
B.C.	Kamloops	A	—	6799
	Prince George*	A	—	9755
	Prince Rupert	C	—	7029
	Vancouver*	A	—	5515
	Victoria*	A	—	5699
	Victoria	C	—	5579
Man.	Brandon*	A	—	11,036
	Churchill	A	—	16,728
	The Pas	C	—	12,281
	Winnipeg	A	—	10,679
N.B.	Fredericton*	A	—	8671
	Moncton	C	—	8727
	St. John	C	—	8219
Nfld.	Argentia	A	—	8440
	Corner Brook	C	—	8978
	Gander	A	—	9254
	Goose*	A	—	11,887
	St. John's*	A	—	8991
N.W.T.	Aklavik	C	—	18,017
	Fort Norman	C	—	16,109
	Resolution Island	C	—	16,021
N.S.	Halifax	C	—	7361
	Sydney	A	—	8049
	Yarmouth	A	—	7340
Ont.	Cochrane	C	—	11,412
	Fort William	A	—	10,405
	Kapuskasing	C	—	11,572
	Kitchner	C	—	7566
	London	A	—	7349
	North Bay	C	—	9219
	Ottawa	C	—	8735
	Toronto	C	—	6827
P.E.I.	Charlottetown	C	—	8164
	Summerside	C	—	8488
Que.	Arvida	C	—	10,528
	Montreal*	A	—	8203
	Montreal	C	—	7899
	Quebec*	A	—	9372
	Quebec	C	—	8937
Sask.	Prince Albert	A	—	11,630
	Regina	A	—	10,806
	Saskatoon	C	—	10,870
Y.T.	Dawson	C	—	15,067
	Mayo Landing	C	—	14,454

*The data for these normals were from the full 10-year period 1951-1960, adjusted for the standard journal period 1931-1960.

Table 4-9 Equivalent Wind Chill Temperatures

(Table 13, Chapter 9, 2017 *ASHRAE Handbook—Fundamentals)*

Wind Speed, mph	Actual Thermometer Reading, °F											
	50	**40**	**30**	**20**	**10**	**0**	**–10**	**–20**	**–30**	**–40**	**–50**	**–60**
	Equivalent Chill Temperature, °F											
0	50	40	30	20	10	0	–10	–20	–30	–40	–50	–60
5	48	37	27	16	6	–5	–15	–26	–36	–47	–57	–68
10	40	28	16	3	–9	–21	–34	–46	–58	–71	–83	–95
15	36	22	9	–5	–18	–32	–45	–59	–72	–86	–99	–113
20	32	18	4	–11	–25	–39	–53	–68	–82	–96	–110	–125
25	30	15	0	–15	–30	–44	–59	–74	–89	–104	–119	–134
30	28	13	–3	–18	–33	–48	–64	–79	–94	–110	–125	–140
35	27	11	–4	–20	–36	–51	–67	–83	–98	–114	–129	–145
40	26	10	–6	–22	–38	–53	–69	–85	–101	–117	–133	–148
	Little danger: In less than 5 h, with dry skin. Maximum danger from false sense of security. (WCI less than 1400)				**Increasing danger:** Danger of freezing exposed flesh within one minute. (WCI between 1400 and 2000)			**Great danger:** Flesh may freeze within 30 seconds. (WCI greater than 2000)				

Notes: Cooling power of environment expressed as an equivalent temperature under calm conditions [Equation (79)].

Winds greater than 43 mph have little added chilling effect.

Source: US Army Research Institute of Environmental Medicine.

4.3 Other Factors Affecting Design

In the interest of energy conservation, new buildings used primarily for human occupancy must meet certain minimum design requirements that enable the efficient use of energy in such new buildings. ASHRAE/IES Standard 90.1, “Energy Efficient Design of New Buildings Except Low-Rise Residential Buildings,” has been widely adopted. In fact, the U. S. Energy Policy Act of 1992 requires that each state have energy policies in place that require buildings to conform, on the minimum, to Standard 90.1.

Although a heating or cooling system is sized to meet design conditions, it usually functions at only partial capacity. The proper use and application of controls should receive primary consideration at the time the heating or cooling system is being designed. Control devices that produce almost any degree of control can be used, but it is useless to provide such controls unless the air-conditioning system is capable of properly responding to the demands of the controllers. For example, it is impossible to maintain close control of temperature and humidity by starting and stopping a refrigeration compressor or by opening and closing a refrigerant valve. Instead, refrigeration equipment that permits proportional control, or a chilled-water system with a cooling coil or spray dehumidifier that permits proportional control, must be used. It is neither economical nor good practice to select equipment capable of producing far more precise control than the application requires, or to complicate the system to obtain special sequences or cycles of operation when they are not necessary. Because the system must be adjusted and maintained in operation for many years, the simplest system which produces the necessary results is usually the best.

Another factor at least as important as comfort, first cost, and owning and operating cost, is the application, which includes such factors as

- Flexibility for change
- Suitability for all spaces
- Appearance of completed building
- Special requirements of project income potential
- Durability, reliability, and serviceability
- Fire and smoke control
- Pollution control

Example 4-3: Select appropriate summer and winter, indoor and outdoor, design temperatures and humidities for an office building near the Philadelphia International Airport, Pennsylvania.

Solution:

Based upon the climatic data of Table 4.7, the *outdoor design values* are

SUMMER: 90.8°F db and 74.4°F wb (1% values)

WINTER: 13.8°F (0.4% value) and 100% rh

Based upon the comfort envelope of Standard 55, the selected *indoor design values* are

SUMMER: 78°F db, humidity ratio of 0.012

WINTER: 72°F db, 30% rh

4.4 Temperatures in Adjacent Unconditioned Spaces

The heat loss or gain between conditioned rooms and unconditioned rooms or spaces must be based on the estimated or assumed temperature in such unconditioned spaces. This temperature normally lies in the range between the indoor and outdoor temperatures. The temperature in the unconditioned space may be estimated by

$$\begin{aligned} t_u = {} & [\, t_i(A_1U_1 + A_2U_2 + \ldots + \text{etc.}) \\ & + t_o(KV_o + A_aU_a + A_bU_b + \ldots + \text{etc.})] \\ & \div ([\, A_1U_1 + A_2U_2 + \ldots + \text{etc.}]) \\ & + (KV_o + A_aU_a + A_bU_b + \ldots + \text{etc.})] \end{aligned} \tag{4-6}$$

where

t_u =temperature in unheated space, °F (°C)

t_i = indoor design temperature of heated room, °F (°C)

t_o = outdoor design temperature, °F (°C)

A_1, A_2, etc. = areas of surface of unheated space adjacent to heated space, ft^2 (m^2)

A_a, A_b, etc. = areas of surface of unheated space exposed to outdoors, ft^2 (m^2)

U_1, U_2, etc. = heat transfer coefficients of surfaces of A_1, A_2, etc., $\text{Btu/h}\cdot\text{ft}^2\cdot°\text{F}$ ($\text{W/[m}^2\cdot°\text{C]}$)

U_a, U_b, etc. = heat transfer coefficients of surfaces of A_a, A_b, etc. $\text{Btu/h}\cdot\text{ft}^2\cdot°\text{F}$ ($\text{W/[m}^2\cdot°\text{C]}$)

V_o = rate of introduction of outdoor air into the unheated space by infiltration and/or ventilation, cfm (L/s)

K = 1.10 (1200)

Reasonable accuracy for ordinary unconditioned spaces may be attained if the following approximations for adjacent rooms are used:

1. Cooling with adjacent unconditioned room. Select for computation a temperature equal to $t_i + 0.667(t_o - t_i)$ in the unconditioned space.
2. Heating with adjacent room unheated. Select for computation a temperature equal to $t_i - 0.50(t_i - t_o)$ in the unconditioned space.

Temperatures in unconditioned spaces having large glass areas and two or more surfaces exposed to the outdoors (such as sleeping porches or sun parlors) and/or large amounts of infiltration (such as garages with poor fitting doors) are generally assumed to be that of the outdoors.

Example 4-4 Calculate the temperature in an unheated space adjacent to a conditioned room with three common surface areas of 100, 120, and 140 ft^2 and overall heat transfer coefficients of 0.15, 0.20, and 0.25 $\text{Btu/h}\cdot\text{ft}^2\cdot°\text{F}$, respectively. The surface areas of the unheated space exposed to the outdoors are 100 and 140 ft^2 with corresponding overall heat transfer coefficients are 0.10 and 0.30 $\text{Btu/h}\cdot\text{ft}^2\cdot°\text{F}$. The sixth surface is on the ground and can be neglected for this example as can be the effect of any outdoor air entering the space. Inside and outside design temperatures are 70 °F and –10 °F, respectively.

Solution:

Substituting into Equation 4-6,

$$\begin{aligned} t_u = {} & [70(100 \times 0.15 + 120 \times 0.20 + 140 \times 0.25) \\ & + (-10)(100 \times 0.10 + 140 \times 0.30)]/ \\ & [100 \times 0.15 + 120 \times 0.20 + 140 \times 0.25 \\ & + 100 \times 0.10 + 140 \times 0.30] \\ = {} & 4660/126 = 37°\text{F} \end{aligned}$$

4.5 Problems

4.1 Describe the various processes of heat transfer involved in maintaining heat balance between the human body and the surrounding space.

4.2 Discuss the following:

(a) What single factor governs most specifications of outdoor air ratio?

(b) How does the fresh air rate vary with the allowable air space per person?

(c) What are typical outdoor air rates for large air spaces? for small (<100 ft^3 per person) air spaces?

4.3 Discuss the following:

(a) What factors affect the temperature and humidity required for comfort?

(b) Of what is "effective temperature" (ET*) a function?

(c) What are the criteria on which the values given on the ET* chart are based?

4.4 How must the ASHRAE Comfort Chart Data be altered for (a) women, (b) older people, (c) people in warm climates, and (d) hot or cold radiative walls or windows?

4.5 Discuss the disadvantages of specifying a high humidity? A low humidity? Answer with respect to both equipment and room occupancy consideration. What are recommended limits of humidity?

4.6 For an office building in St. Louis, Missouri, the inside dry-bulb temperature is maintained at 24°C (75°F) during the summer. What is the maximum allowable humidity ratio consistent with the ASHRAE Comfort Standard? [Ans: 0.012]

4.7 A person reclining nude would probably consider which of the following interior environments comfortable?

(a) $t = 78°\text{F}$; $\phi = 50\%$; 25 fpm; MRT = 78°F

(b) $t = 75°\text{F}$; $\phi = 10\%$; 25 fpm; MRT = 75°F

(c) $t = 75°\text{F}$; $\phi = 70\%$; 30 fpm; MRT = 82°F

(d) $t = 80°\text{F}$; $\phi = 50\%$; 20 fpm; MRT = 80°F

(e) $t = 90°\text{F}$; $\phi = 20\%$; 30 fpm; MRT = 80°F

(f) $t = 70°\text{F}$; $\phi = 70\%$; 20 fpm; MRT = 70°F

4.8 For the person in Problem 4.7 (5 ft, 5 in., 120 lb), compute the body surface area (ft^2). [Ans: 17.1 ft^2]

4.9 Compute your body surface area based on the DuBois formula and estimate your current rate of dry heat exchange with the surroundings.

4.10 From weather data, estimate the lowest equivalent wind chill temperature which will be exceeded 22 h per year for the following locations:
(a) Fairbanks, Alaska
(b) Los Angeles, California
(c) Denver, Colorado
(d) Hartford, Connecticut
(e) Minneapolis, Minnesota
(f) Wichita, Kansas

4.11 Specify comfortable environments for the following space usage:
(a) senior citizens retirement home in Florida
(b) gymnasium in Missouri
(c) office building in Colorado
(d) residence in Nevada

4.12 The living room in a home is occupied by adults at rest wearing medium clothing. The mean radiant temperature is 18°C (64°F). Determine the air temperature necessary for comfort. [Ans: 30°C (86°F)]

4.13 A room has a net outside wall area of 275 ft^2 with a surface temperature of 54°F, 45 ft^2 of glass with a surface temperature of 20°F, 540 ft^2 of ceiling with a surface temperature of 60°F, 670 ft^2 of partitions with a surface temperature of 70°F, and 540 ft^2 of floor with a surface temperature of 70°F. If the air movement is 20 fpm and light clothing is being worn, determine the air temperature necessary for comfort.

4.14 Workers on an assembly line making electronic equipment dissipate 700 Btu/h, of which 310 Btu/h is latent heat. When the MRT for the area is 69°F, what air temperature must the heating system maintain for comfort of the workers if the air movement is 40 fpm?

4.15 A room has 1000 ft^2 of surface, of which 120 ft^2 is to be heated, and the balance has an average surface temperature of 60°F. The air temperature in the room is 68°F. The room is occupied by light clothed adults at rest. Determine the surface temperature of the heated panel necessary to produce comfort if the air velocity is 20 fpm.
[Ans: 243°F ? unfeasible]

4.16 Assume that in Problem 4.15, the maximum allowable panel temperature is 120°F. The average temperature of other surfaces in the room remains at 60°F and the air temperature is still 68°F. What panel area will be required if the room is occupied by adults at rest?

4.17 In an auditorium, 17,150 students are watching slides. The MRT is 80°F and the average room air temperature is 72°F. Air enters the room at 57°F. Assume the lights are out and no heat gain or loss occurs through the walls, floor, and ceiling.
(a) How much air (CFM) should be supplied to remove the sensible heat?
(b) Explain what must be done to remove the latent heat.

4.18 Two hundred people attend a theater matinee. Air is supplied at 60°F. Determine the required flow rate (lb/h) to handle the heat gain from the occupants if the return air temperature is not to exceed 75°F. [Ans: 11,475 lb/h]

4.19 Determine the increase in humidity ratio due to 80 people in a dance hall if air is circulated at the rate of 0.64 m^3/s (1350 cfm).

4.20 Specify the MRT for comfort in a space where the air temperature is 68°F and the relative velocity is 20 fpm, for sedentary activity and light clothing. [Ans: 92°F]

4.21 Specify completely a suitable set of indoor and outdoor design conditions for each of the following cases:
(a) winter; apartment building; St. Louis, Missouri
(b) summer; apartment building; St. Louis, Missouri
(c) winter; factory (medium activity); Rochester, Minnesota

4.22 The mean radiant temperature in a bus is 6°C lower in winter than the air temperature. For passengers seated without coats, determine the desired air temperature if the relative air velocity is 0.2 m/s. [Ans: 28°C]

4.23 For Atlanta, Georgia, specify the normal indoor design conditions listed below for

(a) Winter: Dry bulb = _____ °C; W = _______ kg/kg

(b) Summer: Dry bulb = _____ °C; W = _______ kg/kg

4.24 Specify completely indoor and outdoor design conditions for winter for a clean room in Kansas City, Missouri, having a 1.2 m by 1.2 m radiant panel at 49°C on each of the four walls. The room is 6 m by 4 m by 3 m high and the other surfaces are all at 22°C. Assume very little activity and light clothing.

4.6 Bibliography

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SI Tables and Figures

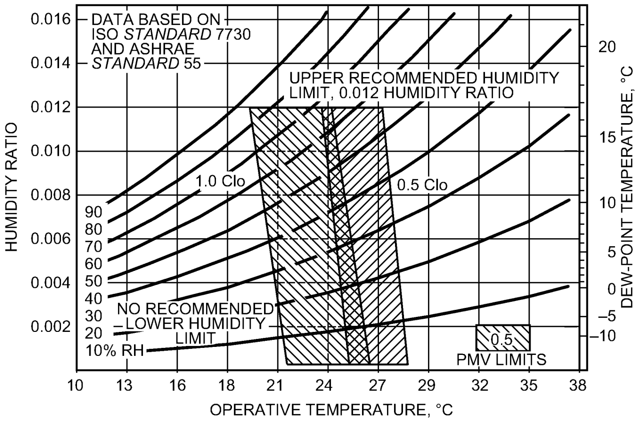
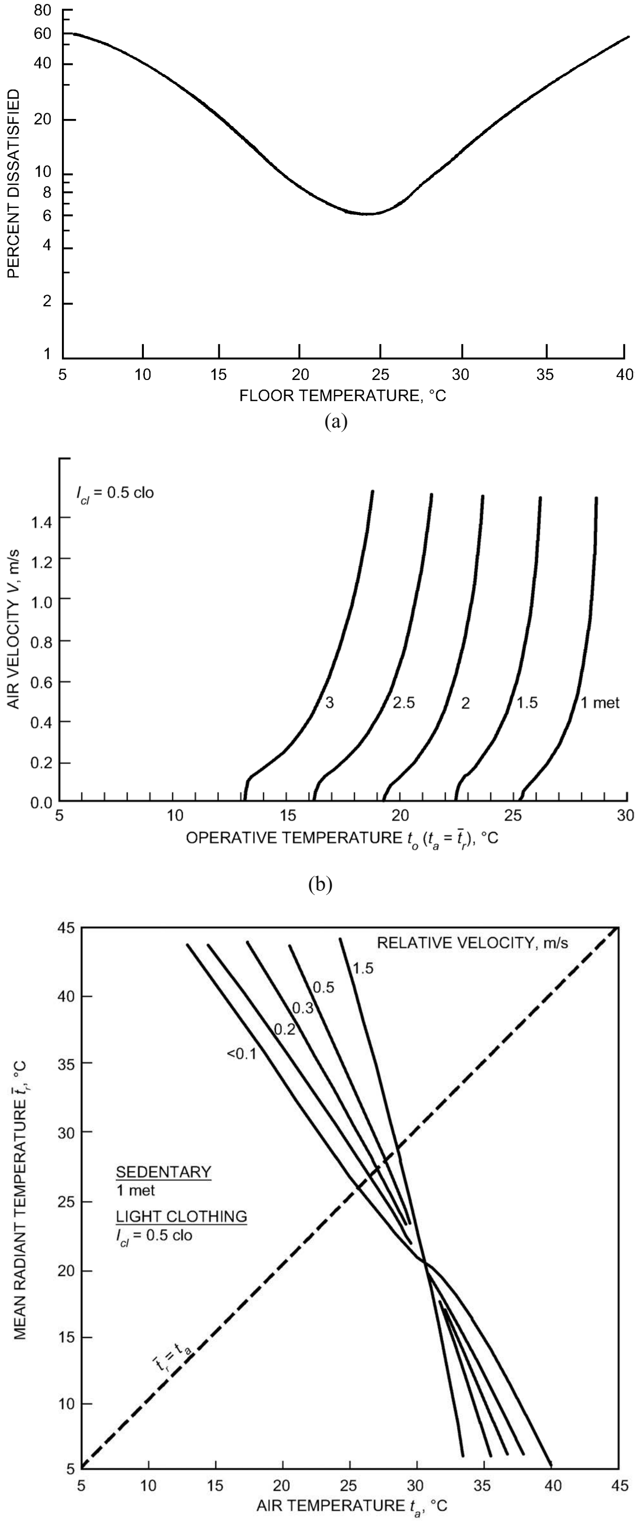


Fig. 4-2 SI ASHRAE Summer and Winter Comfort Zones

(Acceptable ranges of operative temperature and humidity for people in typical summer and winter clothing during primarily sedentary activity.)

(Figure 5, Chapter 9, 2017 ASHRAE Handbook—Fundamentals)



(a)

(b)

(c)

Fig. 4-3 SI Examples of Fanger's Charts

(Figures 13, 14, and 15, Chapter 9, 2017 ASHRAE Handbook—Fundamentals)

Table 4-9 SI Equivalent Wind Chill Temperatures[a]

(Table 13, Chapter 9, 2017 ASHRAE Handbook—Fundamentals)

Wind Speed, km/h	Actual Thermometer Reading, °C												
	10	**5**	**0**	**–5**	**–10**	**–15**	**–20**	**–25**	**–30**	**–35**	**–40**	**–45**	**–50**
	Equivalent Chill Temperature, °C												
Calm	10	5	0	–5	–10	–15	–20	–25	–30	–35	–40	–45	–50
10	8	2	–3	–9	–14	–20	–25	–31	–37	–42	–48	–53	–59
20	3	–3	–10	–16	–23	–29	–35	–42	–48	–55	–61	–68	–74
30	1	–6	–13	–20	–27	–34	–42	–49	–56	–63	–70	–77	–84
40	–1	–8	–16	–23	–31	–38	–46	–53	–60	–68	–75	–83	–90
50	–2	–10	–18	–25	–33	–41	–48	–56	–64	–71	–79	–87	–94
60	–3	–11	–19	–27	–35	–42	–50	–58	–66	–74	–82	–90	–97
70	–4	–12	–20	–28	–35	–43	–51	–59	–67	–75	–83	–91	–99
	Little danger: In less than 5 h, with dry skin. Maximum danger from false sense of security. (WCI less than 1400)				**Increasing danger:** Danger of freezing exposed flesh within one minute. (WCI between 1400 and 2000)			**Great danger:** Flesh may freeze within 30 seconds. (WCI greater than 2000)					

Note: Cooling power of environment expressed as an equivalent temperature under calm conditions [Equation (79)].

Winds greater than 70 km/h have little added chilling effect.

Source: US Army Research Institute of Environmental Medicine.

Chapter 5

LOAD ESTIMATING FUNDAMENTALS

In this chapter, the fundamental elements that accompany the load calculations for sizing heating and cooling systems are presented. The material includes the estimation of outdoor air quantities and the evaluation of the overall coefficient of heat transfer for building components. Chapters 15, 16, 23, 25, 26, and 27 of the 2017 *ASHRAE Handbook—Fundamentals* and ASHRAE's *Load Calculations Applications Manual* (2014) are the major sources of information.

5.1 General Considerations

The basic components of heating and cooling loads are illustrated in Figure 5-1.

Proper design of space heating, air-conditioning, or refrigeration systems, and other industrial applications requires a knowledge of thermal insulations and the thermal behavior of building structures. Chapters 25, 26, and 27 of the 2017 *ASHRAE Handbook—Fundamentals*, dealing with thermal insulation and vapor retarders, provide the fundamentals and properties of thermal insulating materials, water vapor barriers, economic thickness of insulation, general practices for building and industrial insulation, and the insulating of mobile equipment and environmental spaces.

Flowing fluids such as air, water, and refrigerants are used in heating, ventilating, air-conditioning, and refrigeration systems to carry heat or mass. An understanding of fluid flow and the nature of its mechanisms is vital to engineers working in these fields. Chapter 3, "Fluid Flow," of the 2017 *ASHRAE Handbook—Fundamentals* introduces the principles of fluid mechanics relevant to these processes.

Heating or cooling air involves only heat transfer, resulting in a temperature change of the air. However, in a true air-conditioning process, there is a simultaneous transfer of heat and mass (water vapor). Chapter 6 of the 2017 *ASHRAE Handbook—Fundamentals* presents the elementary principles of mass transfer to provide a basic understanding of the air-conditioning processes involving mass transfer.

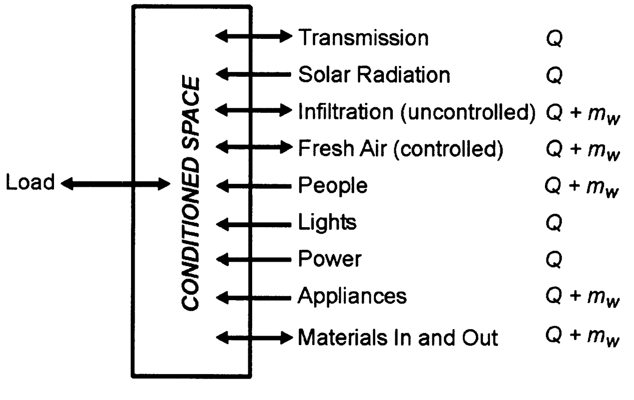


Fig. 5-1 Components of Heating and Cooling Loads

The humidification or dehumidification load depends primarily on the ventilation rate of the space to be conditioned, but other sources of moisture gain or loss should be considered. Chapter 1, "Psychrometrics," of the 2017 *ASHRAE Handbook—Fundamentals* contains equations and examples for determining loads and energy requirements associated with humidification or dehumidification.

5.2 Outdoor Air Load Components

5.2.1 Basic Concepts

Outdoor air that flows through a building is often used to dilute and remove indoor air contaminants. However, the energy required to condition this outdoor air can be a significant portion of the total space-conditioning load. The magnitude of the outdoor airflow into the building must be known for proper sizing of the HVAC equipment and evaluation of energy consumption. For buildings without mechanical cooling and dehumidification, proper ventilation and infiltration airflows are important for providing comfort for occupants.

A conditioned space may be ventilated by natural infiltration, alone or in combination with intentional mechanical ventilation. Natural infiltration varies with indoor-outdoor temperature difference, wind velocity, and the tightness of the construction, as discussed in Chapter 16, "Ventilation and Infiltration," of the 2017 *ASHRAE Handbook—Fundamentals*. Related information is presented in Chapter 24, "Airflow Around Buildings," of the 2017 *ASHRAE Handbook—Fundamentals*.

Air exchange of outdoor air with the air already in a building can be divided into two broad classifications: ventilation and infiltration. **Ventilation** is the intentional introduction of air from the outside into a building; it is further subdivided into natural ventilation and forced ventilation. **Natural ventilation** is the intentional flow of air through open windows, doors, grilles, and other planned building envelope penetrations, and it is driven by natural and/or artificially produced pressure differentials. **Forced ventilation** is the intentional movement of air into and out of a building using fans and intake and exhaust vents; it is also called **mechanical ventilation**.

Infiltration is the uncontrolled flow of outdoor air into a building through cracks and other unintentional openings and through the normal use of exterior doors for entrance and egress. Infiltration is also known as **air leakage** into a building. **Exfiltration** is the leakage of indoor air out of a building. Like natural ventilation, infiltration and exfiltration are driven by natural and/or artificial pressure differences. **Transfer air** (ta) is air that moves from one interior space to another, either intentionally or not.

These modes of air exchange differ significantly in how they affect energy, air quality, and thermal comfort, and they can each vary with weather conditions, building operation, and use. Although one mode may be expected to dominate in a particular building, all must be considered for the proper design and operation of an HVAC system.

Modern commercial and institutional buildings are normally required to have forced ventilation and are usually pressurized somewhat (approximately 0.05 in H_20) to reduce or eliminate infiltration. Forced ventilation has the greatest potential for control of air exchange when the system is properly designed, installed, and operated; it should provide acceptable indoor air quality when ASHRAE Standard 62.1 requirements are followed. Forced ventilation equipment and systems are described in Chapters 1, 2, and 4 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

In commercial and institutional buildings, uncontrolled natural ventilation, such as through operable windows, may not be desirable from the point of view of energy conservation and comfort. In commercial and institutional buildings with mechanical cooling and forced ventilation, an air- or water-side economizer cycle may be preferable to operable windows for taking advantage of cool outdoor conditions when interior cooling is required. Infiltration may be significant in commercial and institutional buildings, especially in tall, leaky, or underpressurized buildings.

In most of the United States, residential buildings typically rely on infiltration and natural ventilation to meet their ventilation needs. Infiltration is not reliable for ventilation purposes because it depends on weather conditions, building construction, and maintenance. Natural ventilation, usually through operable windows, is dependent on weather and building design but allows occupants to control airborne contaminants and interior air temperature. However, natural ventilation can have a substantial energy cost if used while the residence's heating or cooling equipment is operating.

In place of operable windows, small exhaust fans may be provided for localized venting in residential spaces such as kitchens and bathrooms. Not all local buildings codes require that the exhaust be vented to the outside. Instead, the code may allow the air to be treated and returned to the space or to be discharged to an attic space. Poor maintenance of these treatment devices can make nonducted vents ineffective for ventilation purposes. Condensation in attics should be avoided. In northern Europe and in Canada, some building codes require general forced ventilation in residences, and heat recovery heat exchangers are popular for reducing the energy impact. Residential buildings with low rates of infiltration and natural ventilation require forced ventilation at rates given in ASHRAE Standard 62.2.

Regardless of these complexities and uncertainties, designers and operators need guidance on ventilation and indoor air quality.

ASHRAE Standard 62.1 provides guidance on ventilation and indoor air quality in the form of several alternative procedures. The **Ventilation Rate Procedure** (VRP), the **Indoor Air Quality Procedure** (IAQP), and/or the **Natural Ventilation Procedure** (NVP) are required to satisfy the requirements of Standard 62. In the **Ventilation Rate Procedure**, indoor air quality is assumed to be acceptable if (1) the concentrations of six pollutants in the incoming outdoor air meet the United States national ambient air quality standards, and (2) the outdoor air supply rates meet or exceed values (which vary depending on the type of space) provided in a table. The minimum outdoor air supply for most types of space is 5 cfm (2.5 L/s) per person plus 0.06 cfm/ft^2 (0.3 L/s·m^2). This minimum rate will maintain an indoor CO_2 concentration below 0.1% (1000 parts per million) assuming a typical CO_2 generation rate per occupant.

The second alternative in ASHRAE Standard 62 is the **Indoor Air Quality Procedure**. In this procedure, any outdoor air supply rate is acceptable if (1) the indoor concentrations of nine pollutants are maintained below specified values, and (2) the air is deemed acceptable via subjective evaluations of odor. If users of the IAQ Procedure control pollutant source strengths or use air cleaning of local exhaust ventilation, they may be able to reduce the outdoor air supply rates to below those specified in the ventilation rate procedure.

The **Natural Ventilation Procedure** (NVP) allows some exceptions using natural ventilation in conjunction with the VRP on the IAQP.

As a rough guideline, the minimum infiltration outdoor air allowance may be taken as 0.5 air changes per hour (actually to produce a slight positive pressure within the structure producing exfiltration from the conditioned spaces). The minimum ventilation air allowance based on ASHRAE Standard 62.1 is 5 cfm (2.5 L/s) per person plus 0.06 cfm/ft^2 [0.3 L/(s· m^2)]. However, local ventilation ordinances must be checked as they may require greater quantities of outdoor air.

5.2.2 Terminology

Figure 5-2 shows a simple **air-handling unit** (AHU) that conditions air for a building. Air brought back to the air handler from the conditioned space is *return air* (ra). The portion of the return air that is discharged to the environment is *exhaust air* (ea), and the part of the return air that is reused is *recirculated air* (ca). Air brought in intentionally from the environment is *outdoor* or *outdoor air* (oa). Because outdoor air may need treatment to be acceptable for use in a building, it should not be called "fresh air." The outdoor air and the recirculated air are combined to form *mixed air* (ma), which is then conditioned and delivered to the thermal zone as *supply air* (sa). Any portion of the mixed air that intentionally or unintentionally circumvents conditioning is *bypass air* (ba). *Ventilation air* is

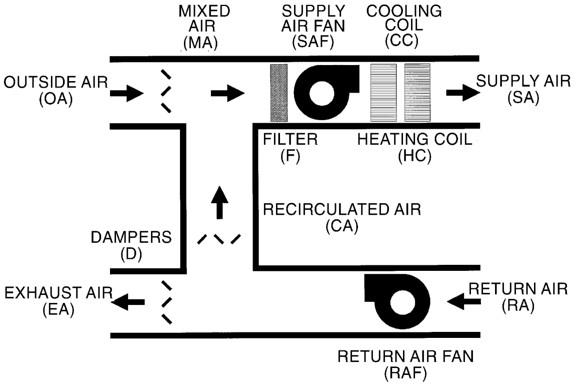


Fig. 5-2 Simple All-Air Air-Handling Unit with Associated Airflows

(Figure 2, Chapter 16, 2017 ASHRAE Handbook—Fundamentals)

used to provide acceptable indoor air quality. It may be composed of forced or natural ventilation air, infiltration air, suitable treated recirculated air, transfer air, or an appropriate combination. Due to the wide variety of air-handling systems, these airflows may not all be present in a particular system as defined here. Also, more complex systems may have additional airflows.

The outdoor airflow being introduced to a building or zone by an air-handling unit can also be described by the *outdoor air fraction* X_{oa}, which is the ratio of the flow rate of outdoor air brought in by the air handler to the total supply airflow rate. When expressed as a percentage, the outdoor air fraction is called the *percent outdoor air*. The design outdoor airflow rate for a building's ventilation system is found through evaluating the requirements of ASHRAE Standard 62. The supply airflow rate is that required to meet the thermal load. The outdoor air fraction and percent outdoor air then describe the degree of recirculation, where a low value indicates a high rate of recirculation, and a high value shows little recirculation. Conventional all-air-handling systems for commercial and institutional buildings have approximately 10% to 40% outdoor air. *100% outdoor air* means no recirculation of return air through the air-handling system. Instead, all the supply air is treated outdoor air, also known as *makeup air*, and all return air is discharged directly to the outside as *relief air*. An air-handling unit that provides 100% outdoor air is typically called a **makeup air unit** (MAU).

Outdoor air introduced into a building constitutes a large part of the total space-conditioning (heating, cooling, humidification, and dehumidification) load, which is one reason to limit air exchange rates in buildings to the minimum required. Air exchange typically represents 20% to 40% of a shell-dominated building's thermal load. First, the incoming air must be heated or cooled from the outdoor air temperature to the indoor air temperature. The rate of energy consumption due to this sensible heating or cooling is given by

$$q_s = Q\rho c_p \Delta t \tag{5-1}$$

where

q_s = sensible heat load
Q = airflow rate
ρ = air density
c_p = specific heat of air
Δt = indoor-outdoor temperature difference

Second, air exchange modifies the moisture content of the air in a building. This is particularly important in some locations in the summer when the outdoor air must be dehumidified. In the winter, when the relative humidity of the indoor air is below 30%, humidification may be needed. The rate of energy consumption associated with these latent loads is given by

$$q_l = Q\rho h_{fg} \Delta W \tag{5-2}$$

where

q_l = latent heat load
h_{fg} = latent heat of water
ΔW = humidity ratio of indoor air minus humidity ratio of outdoor air

5.2.3 Infiltration

Infiltration or **exfiltration** is air leakage through cracks and interstices, around windows and doors, and through floors and walls of any type of building. The magnitude of infiltration/exfiltration depends on the type of construction, the workmanship, and the condition of the building.

Outdoor air infiltration/exfiltration may account for a significant proportion of the heating or cooling requirements for buildings. Thus, it is important to make an adequate estimate of its contribution with respect to both design loads and seasonal energy requirements. Air infiltration is also an important factor in determining the relative humidity that occurs in buildings or, conversely, the amount of humidification or dehumidification required to maintain given humidities.

The rate of airflow into and out of a building due to either infiltration, exfiltration, or natural ventilation depends on the magnitude of the pressure difference between the inside and outside of the structure and on the resistances to airflow offered by openings and interstices in the building. The pressure difference exerted on the building enclosure by the air may be caused either by wind or by a difference in density of the inside and outdoor air. The effect of the difference in density is often called the **chimney** or **stack effect**, and is often the major factor contributing to air leakage. The pattern of airflow through any part of the structure depends on both the pressure difference and the area of openings.

When the pressure difference is the result of wind pressure, air enters the building through openings in the windward walls and leaves through openings in the leeward walls or, as may be the case in one-story commercial buildings, through ventilating ducts in the roof. When the pressure difference is caused by the indoor-outdoor temperature difference, the flow is along the path of least resistance from inlets at lower levels to outlets at higher levels in a heated building

or in the opposite direction for an air-conditioned building, as may be the case in multistory skyscrapers.

In most instances, the pressure difference between inside and outdoor air results from temperature difference forces. Depending on the design, mechanical ventilation and exhaust systems can affect pressure differences across the building enclosure.

The principles of infiltration calculations are discussed in Chapter 16 of the 2017 *ASHRAE Handbook—Fundamentals*, with emphasis placed on the heating season. For the cooling season, infiltration calculations are usually limited to doors and windows. However, in multistory commercial buildings, a reversed chimney effect may exist.

Heat gain due to infiltration must be included whenever the outdoor air mechanically introduced by the system is unable to maintain a positive pressure within the enclosure to prevent infiltration. Most buildings require input of ventilation air for occupants or processes and by properly balancing or controlling the input air versus exhaust/relief air, a positive pressure can be maintained to minimize infiltration. This excess outdoor air introduced through the air-conditioning equipment will maintain a constant outward escape of air, thereby eliminating the infiltration portion of the gain. The positive pressure maintained must be sufficient to overcome wind pressure through cracks and door openings. When this condition prevails, it is unnecessary to include any infiltration component of heat gain. However, on a typical building, infiltration is usually not eliminated due to the pressure that a 25–30 mph wind can exert and, as buildings age, they develop cracks and leak. When the quantity of outdoor air introduced through the cooling equipment is unable to build up the pressure needed to eliminate infiltration, the entire infiltration load should be included in the space heat gain calculations.

Two methods are used to estimate air infiltration in buildings. In one case, the estimate is based on measured leakage characteristics of the building components and selected pressure differences. This is known as the **crack method**, since cracks around windows and doors are usually the major source of air leakage. The other method is known as the air change method and consists of estimating a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use, and location of the room.

The air change method is used often by engineers and designers both for its simplicity and because either method requires the estimation of at least one appropriate numerical value. The method requires the assumption, based on the performance of similar construction, of the number of air changes per hour (ACH) that a space will experience. The infiltration rate is then obtained as follows:

$$Q = \text{ACH} \times \text{VOL}/60 \qquad (5\text{-}2b)$$

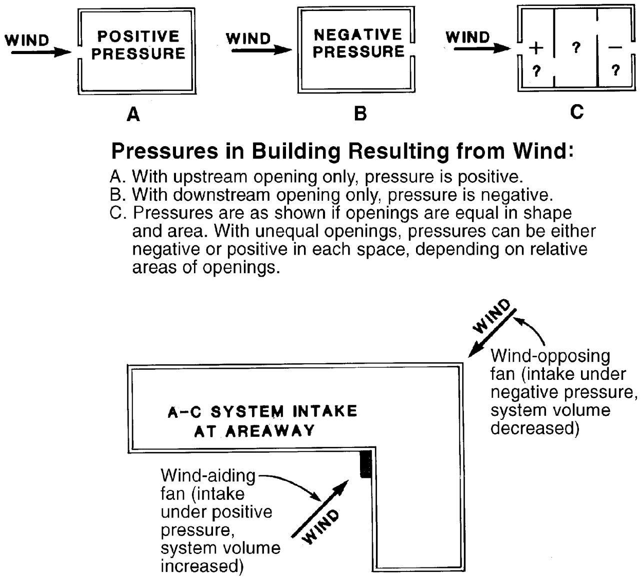
where

Q = infiltration rate, cfm
VOL = gross space volume, ft^3

Table 5-1 Change Rates as a Function of Airtightness

	Outdoor Design Temperature, °F									
Class	**50**	**40**	**30**	**20**	**10**	**0**	**–10**	**–20**	**–30**	**–40**
Tight	0.41	0.43	0.45	0.47	0.49	0.51	0.53	0.55	0.57	0.59
Medium	0.69	0.73	0.77	0.81	0.85	0.89	0.93	0.97	1.00	1.05
Loose	1.11	1.15	1.20	1.23	1.27	1.30	1.35	1.40	1.43	1.47

Note: Values are for 15 mph wind and indoor temperature of 68°F.



Pressures in Building Resulting from Wind:

A. With upstream opening only, pressure is positive.
B. With downstream opening only, pressure is negative.
C. Pressures are as shown if openings are equal in shape and area. With unequal openings, pressures can be either negative or positive in each space, depending on relative areas of openings.

Fig. 5-3 Building Pressure due to Wind Effect
(Figure 15, Chapter 24, 2017 ASHRAE Handbook—Fundamentals)

The crack method is more firmly based on scientific principles and is generally regarded as being more accurate, *provided that leakage characteristics and pressure differences can be properly evaluated*. Otherwise, the air change method may be justified. The accuracy of estimating infiltration for design load calculations by the crack or component method is restricted by limitations in information on air leakage characteristics of components and by the difficulty of estimating pressure differences under appropriate design conditions of temperature and wind.

A building with only upwind openings is under a *positive pressure* (Figure 5-3). Building pressures are *negative* when there are only downwind openings. A building with internal partitions and openings is under various pressures depending on the relative sizes of the openings and the wind direction. With larger openings on the windward face, the building interior tends to remain under positive pressure; the reverse is also true. Airflow through a wall opening results from positive or negative external and internal pressures. Such differential pressures may exceed 0.5 in. of water during high winds. Supply and exhaust systems, openings, dampers, louvers, doors, and windows make the building flow conditions too complex for most calculations. The opening and closing of doors and windows by building occupants adds further complications.

It is impossible to accurately predict infiltration from theory alone because of the many unknowns. However, it is possible to

develop relationships describing the general nature of the problem based on theory and add numerical constants and exponents from experience and experimentation. These semi-empirical expressions are then useful in estimating infiltration rates.

Infiltration is caused by a greater air pressure on the outside of the building than on the inside. The amount of infiltrated air depends on the pressure difference, the nature of the flow through gaps and cracks (laminar versus turbulent), and the size and shape of the cracks. The relationship between the airflow Q through an opening in the building envelope and the pressure difference Δp across it is called the **leakage function** of the opening. The fundamental equation for the airflow rate through an opening is

$$Q = C_D A(2\Delta p/\rho)^n \tag{5-3a}$$

where

Q = airflow rate
C_D = discharge coefficient for the opening
A = cross-section area of the opening
ρ = air density
Δp = pressure difference across opening
n = flow exponent (1.0 if laminar; 0.5 if turbulent)

The discharge coefficient C_D is a dimensionless number that depends on the opening geometry and the Reynolds number of the flow.

The above equation is often simplified when evaluating infiltration by combining the discharge coefficient and the crack area into a flow coefficient C yielding

$$Q = C(\Delta p)^n \tag{5-3b}$$

The pressure difference is given by

$$\Delta p = \Delta p_s + \Delta p_w + \Delta p_p$$

where

Δp_s = pressure difference caused by stack effect
Δp_w = pressure difference caused by wind
Δp_p = pressure difference due to building pressurization

Stack effect occurs when air densities are different on the inside and outside of a building. The air density decreases with increasing temperature and decreases slightly with increasing humidity. Because the pressure of the air is due to the weight of a column of air, on winter days the air pressure at ground level will be less inside the building due to warmer inside air than outdoor air. As a result of this pressure difference, air will infiltrate at ground level and flow upward inside the building. Under summer conditions when the air is cooler inside, outdoor air enters the top of the building and flows downward on the inside. With the stack effect there will be a vertical location in the building where the inside pressure equals the outside pressure, called the neutral pressure level of the building. Unless there is detailed information on the vertical distribution of cracks and other openings, it is assumed that the neutral pressure will be at the building mid-height when under the influence of the stack effect alone.

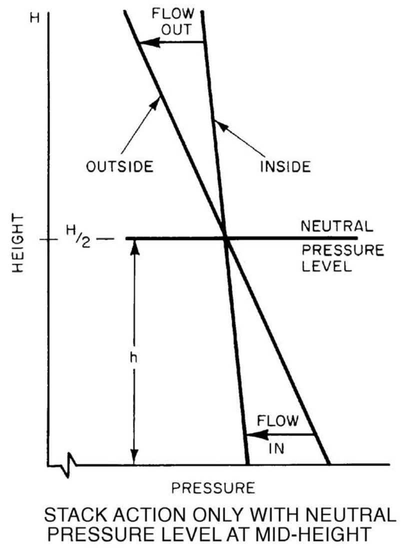


Fig. 5-4 Winter Stack Effect Showing Theoretical Pressure Difference

(Figure 6.A, Chapter 16, 2017ASHRAE Handbook—Fundamentals)

The theoretical pressure difference due to the stack effect can be found from:

$$\Delta p_s = 0.52\, p_b h[(1/T_o) - (1/T_i)] \tag{5-4}$$

where

Δp_s = theoretical pressure difference, in. water
p_b = outside absolute (barometric) pressure, psi
h = vertical distance from neutral pressure level, ft
T_o = outside absolute temperature, °R
T_i = inside absolute temperature, °R

Figure 5-4 shows the stack effect pressure variations under winter conditions.

The pressure associated with the wind velocity, called velocity pressure, is

$$p_v = \rho V_w^2/2$$

Even on the windward side of a building the wind velocity does not go to zero as the air comes in contact with the building. To account for this, a wind pressure coefficient C_p is used when determining the static pressure obtained from the velocity pressure of the wind:

$$\Delta p_w = 0.5 C_p \rho V_w^2 \tag{5-5}$$

The pressure coefficient will always have a value less than 1.0 and can be negative when the wind causes outdoor pressures below atmospheric such as on the leeward side of buildings. Figure 5-5 provides average pressure coefficients for tall buildings.

The building pressure and the corresponding pressure difference Δp_p depend on the design and operation of the HVAC

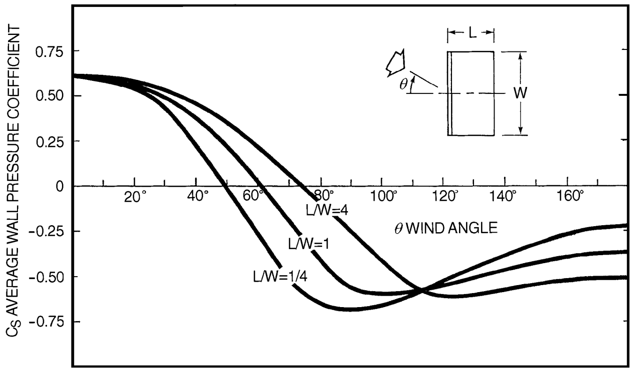


Fig. 5-5 Wind Pressure Coefficients for Tall Buildings

system. A building can be operated either at a positive or negative pressure depending upon the relative flow resistances of the supply and return duct systems. A positive building pressure results in a negative pressure difference Δp_p and a reduction in infiltration from wind and stack effects. While building pressurization is usually desired and assumed to occur, the air circulation system must be carefully designed and balanced to achieve this effect. Care must be taken to estimate a realistic value that the system can actually achieve.

The flow coefficient C in Equation (5-3b) has a particular value for each crack and each window and door perimeter gap. Although values of C are determined experimentally for window and door gaps, this procedure does not work for cracks. Cracks occur at random in fractures of building materials and at the interface of materials. The number and size of cracks depend on the type of construction, the workmanship during construction, and the maintenance of the building after construction. To determine a value of C for each crack is impractical; however, an overall leakage coefficient can used by modifying Equation (5-3b) into the following form:

$$Q = KA(\Delta p)^n \tag{5-6}$$

where

A = wall area

K = leakage coefficient

Table 5-2 provides typical values of the leakage coefficient for various types of curtain wall construction. The associated infiltration can be determined from Figure 5-6.

Although the terms infiltration and air leakage are sometimes used synonymously, they are different, though related, quantities. Infiltration is the rate of uncontrolled air exchange through unintentional openings that occur under given conditions, while air leakage is a measure of the airtightness of the building shell. The greater the air leakage of a building, the greater its infiltration rate. The infiltration rate of an individual building depends on weather conditions, equipment operation, and occupant activities. The rate can vary by a factor of five from weather effects alone.

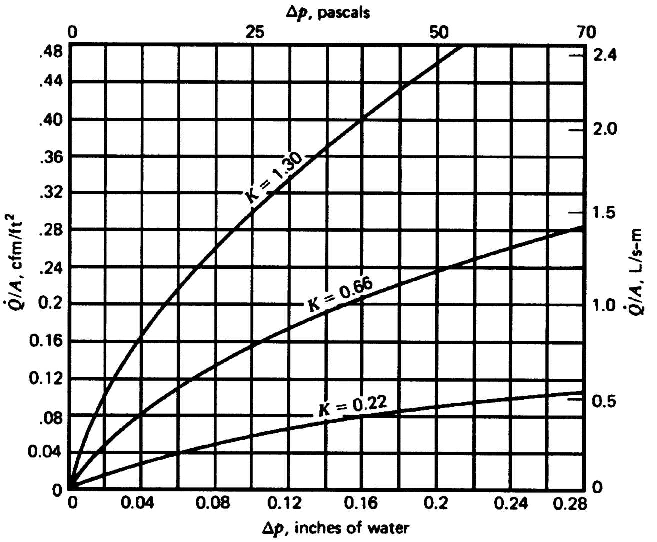


Fig. 5-6 Curtain Wall Infiltration Rates

Table 5-2 Curtain Wall Leakage Coefficients

Leakage Coefficient	Description	Curtain Wall Construction
$K = 0.22$	Tight-fitting wall	Constructed under close supervision of workmanship on wall joints. When joints seals appear inadequate they must be redone
$K = 0.66$	Average-fitting wall	Conventional construction procedures are used
$K = 1.30$	Loose-fitting wall	Poor construction quality control or an older building having separated joints

Typical infiltration values in housing in North America vary by about a factor of ten, from tight housing with seasonal air change rates of 0.2 per hour (ACH), to housing with infiltration rates as great as 2.0 per hour. Histograms of infiltration rates measured in two different samples of North American housing are shown in Figures 5-7 and 5-8. The average seasonal infiltration of 312 houses located in many different areas in North America is shown in Figure 5-7. The median infiltration value of this sample is 0.50 ach. Measurements in 266 houses located in 16 cities in the United States are represented in Figure 5-8. The median value of this sample is 0.90 ach. The group of houses in the Figure 5-7 sample is biased toward new energy-efficient houses, while the group in Figure 5-8 represents older, low-income housing in the United States. While these two samples are not a valid random sample of North American housing, they indicate the distribution expected in an appropriate building sample. Note that infiltration values listed are appropriate for unoccupied structures. *Although occupancy influences have not been measured directly, estimates add an average of 0.10 to 0.15 ach to unoccupied values.*

When estimating infiltration rates for a large group of houses to determine gross energy loads, an adequate assumption for infiltration rates is obtained by using Figures 5-7 and 5-8. A combination of the median values in these figures, with an additional 0.10 to 0.15 ach to account for occupancy

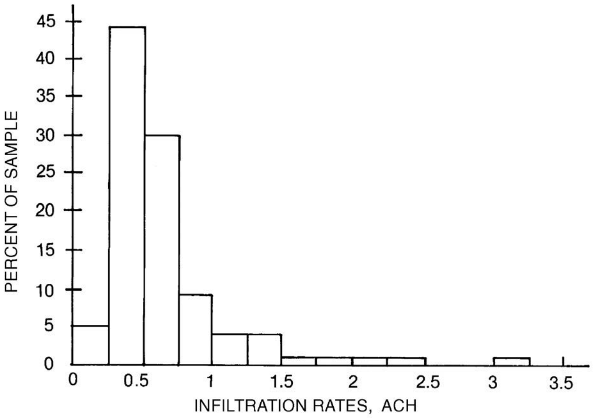


Fig. 5-7 Histogram of Infiltration Values—New Construction

(Figure 11, Chapter 16, 2017ASHRAE Handbook—Fundamentals)

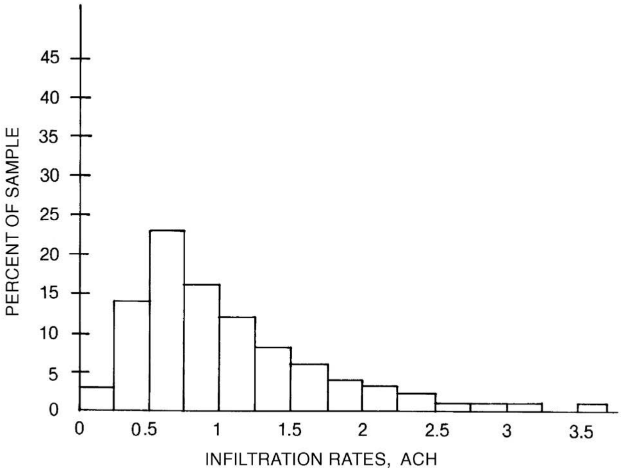


Fig. 5-8 Histogram of Infiltration Values—Low-Income Housing

(Figure 12, Chapter 16, 2017ASHRAE Handbook—Fundamentals)

Table 5-3 Total Ventilation Air Requirements

Area Based	Occupancy Based
1 cfm/100 ft^2 of floor space	7.5 cfm per person, based on normal occupancy

effects, represents average infiltration for a large group of houses.

The building envelopes of large commercial buildings are often thought to be nearly airtight. The National Association of Architectural Metal Manufacturers specifies a maximum leakage per unit of exterior wall area of 0.060 cfm/ft^2 at a pressure difference of 0.30 in. of water, exclusive of leakage through operable windows. Recent measurements on eight US office buildings ranged from 0.213 to 1.028 cfm/ft^2 at 0.30 in. of water (0.1 to 0.6 air changes per hour) with no outdoor air intake. Therefore, office building envelopes may leak more than expected. Typical air leakage values per unit wall area at 0.30 in. of water are 0.10, 0.30, and 0.60 for tight, average, and leaky walls, respectively.

Infiltration may also be estimated by statistical analysis of long-term data of infiltration measurements for specific sites. Weather-related pressures that drive infiltration are estimated by finding the regression constants in a function of the form:

$$I = K_1 + K_2\Delta t + K_3 V \tag{5-7}$$

where

I = air change rate, h^{-1}
Δt = indoor-outdoor temperature difference, °F
V = wind speed, mph
K_1, K_2, K_3 = empirical regression constants derived from measurements at the site

The analysis reveals that correlation with weather variables are only moderately successful. Because the regression coefficients reflect structural characteristics, as well as shielding effects and occupants' behavior, the values of K_1, K_2, and K_3 have varied by 20:1 between similar residences, and the model may be inappropriate as a design tool or for inclusion in computer simulations for building energy analysis.

Several procedures have been developed that treat the building as a well-mixed zone. But a study of a large variety of single-cell models found that these models can only be used to calculate air change rates in structures that can be assumed to have a uniform internal pressure.

Multicell models treat the actual complexity of flows in a building by recognizing effects of internal flow restrictions. They require extensive information on flow characteristics and pressure distributions, and, in many cases, are too complex to justify their use to predict flow for simple structures such as single-family residences.

A simple, single-zone approach to calculating air infiltration rates in houses requires an effective leakage area at 0.016 in. of water. Although control of significant sources of pollution in a dwelling is important, whole house ventilation may still be needed. Each dwelling should be provided with outdoor air according to Table 5-3. The rate is the sum of the "Area Based" and "Occupancy Based" columns. Design occupancy can be based on the number of bedrooms as follows: first bedroom, two persons; each additional bedroom, one person. Additional ventilation should be considered when occupancy densities exceed 1/250 ft^2. ASHRAE Standard 62.2 provides guidance for ventilation and acceptable indoor air quality in low-rise residential buildings.

Using the effective leakage area, the airflow rate due to infiltration is calculated according to the following equation:

$$Q = A_L(C_S\Delta t + C_W V^2)^{0.5} \tag{5-8}$$

where

Q = airflow rate, cfm
A_L = effective leakage area, in^2
C_S = stack coefficient, cfm^2/(in^4·°F)
Δt = average indoor-outdoor temperature difference, °F
C_W = wind coefficient, cfm^2/(in^4·mph^2)
V = average wind speed, mph

Table 5-4 Local Shelter Classes

(Table 5, Chapter 16, 2017 ASHRAE Handbook—Fundamentals)

Shelter Class	Description
1	No obstructions or local shielding
2	Typical shelter for an isolated rural house
3	Typical shelter caused by other buildings across street from building under study
4	Typical shelter for urban buildings on larger lots where sheltering obstacles are more than one building height away
5	Typical shelter produced by buildings or other structures immediately adjacent (closer than one house height): e.g., neighboring houses on same side of street, trees, bushes, etc.

Table 5-5 Stack Coefficient C_S

(Table 4, Chapter 16, 2017 ASHRAE Handbook—Fundamentals)

	House Height (Stories)		
	One	Two	Three
Stack coefficient	0.0150	0.0299	0.0449

Table 5-6 Wind Coefficient C_W

(Table 6, Chapter 16, 2017 ASHRAE Handbook—Fundamentals)

Shielding Class	House Height (Stories)		
	One	Two	Three
1	0.0119	0.0157	0.0184
2	0.0092	0.0121	0.0143
3	0.0065	0.0086	0.0101
4	0.0039	0.0051	0.0060
5	0.0012	0.0016	0.0018

The infiltration rate of the building is obtained by dividing Q by the building volume. The value of C_W depends on the local shielding class of the building. Five different shielding classes are listed in Table 5-4. Values for C_S for one-, two-, and three-story houses are presented in Table 5-5. Values of C_W for one-, two-, and three-story houses in shielding classes one through five are found in Table 5-6. The heights of the one-, two-, and three-story buildings are 8, 16, and 24 ft, respectively.

Example 5-1 Calculate the average infiltration during a one-week period in January for a one-story house in Portland, Oregon. During this period, the average indoor-outdoor temperature difference is 30°F, and the average wind speed is 6 mph. The house has a volume of 9000 ft^3 and an effective air leakage area of 107 in.^2, and it is located in an area with buildings and trees within 30 ft in most directions (shelter class 4).

Solution: From Equation (5-8), the airflow rate due to infiltration is

$$Q = 107\sqrt{(0.0150)(30) + (0.0039)(6^2)} = 82.2 \text{ cfm} = 4930 \text{ ft}^3/\text{h}$$

The air exchange rate is therefore

$$I = 4930 / 9000 = 0.55 \text{ h}^{-1} = 0.55 \text{ ach}$$

Example 5-2 Estimate the infiltration at design conditions for a two-story house in Nebraska. The house has an effective leakage area of 77 in.^2, and is surrounded by a thick hedge. The design temperature is –2°F.

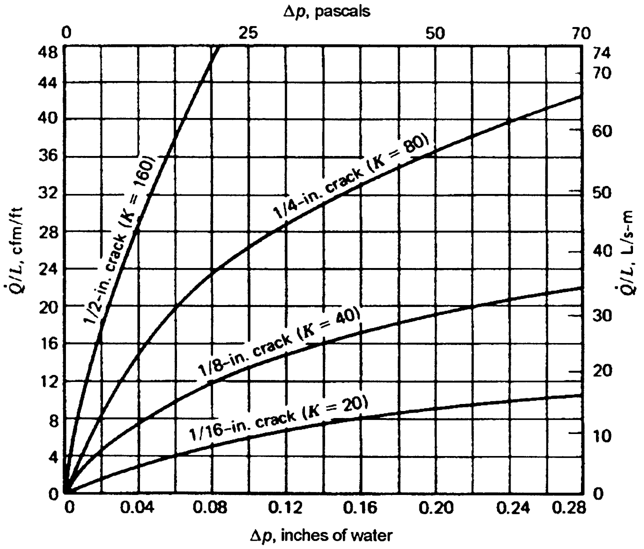


Fig. 5-9 Infiltration through Closed Swinging Door Cracks

Solution: Assume indoor temperature is 75°F. Design wind speed is 15 mph. Shielding Class 3 is due to hedge. From Table 5-5, CS = 0.0299, and from Table 5-6, C_W = 0.0086. The airflow rate due to infiltration is thus:

$$Q = 77[(0.0299 \times 77) + (0.0086 \times 15^2)]^{0.5}$$
$$= 158 \text{ cfm} = 9510 \text{ ft}^3/\text{h}$$

Infiltration rate I equals Q divided by the building volume:

$$I = 9510/12000 = 0.79 \text{ ach}$$

An example of data for the infiltration through cracks in swinging doors is given in Figure 5-9. Commercial buildings often have a large number of people entering and leaving with associated infiltration. Figures 5-10 and 5-11 provide examples of data for estimating the infiltration due to traffic in and out of the building. The figures are based on a standard sized (3 by 7 ft) door. Automatic doors stay open two to four times longer than manually operated doors. Thus, doubling of the infiltration obtained for manually operated doors would be a reasonable estimate for automatic doors.

The total infiltration is the infiltration through the cracks when the door is closed added to the infiltration due to traffic.

Figure 5-12 shows the infiltration due to a pressure difference across the seals of a standard sized revolving door. Figure 5-13 and 5-14 provide data for estimating the infiltration caused by the rotation of the standard-sized door. The amount of infiltration depends upon the inside-outside temperature difference and the rotational speed of the door. The total infiltration is the infiltration due to leakage through the door seals and the infiltration due to the mechanical interchange of air due to the rotation of the door.

5.2.4 Ventilation Air

Outdoor air requirements for acceptable indoor air quality (IAQ) have long been debated and different rationales have produced quite different ventilation standards. Historically,

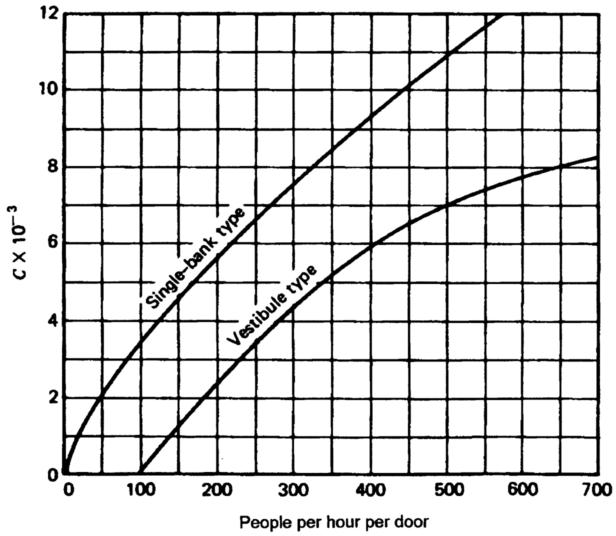


Fig. 5-10 Flow Coefficients for Swinging Doors

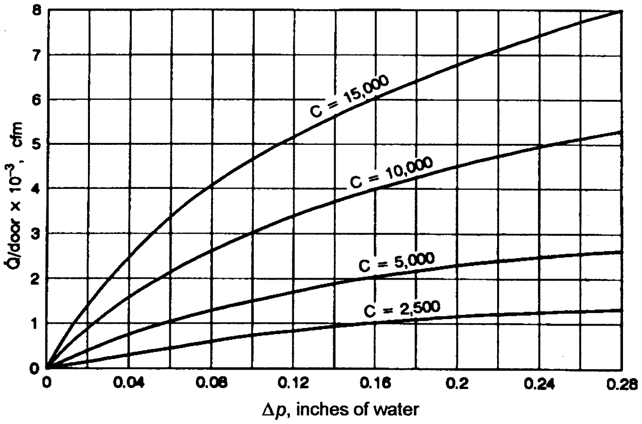


Fig. 5-11 Infiltration for Swinging Doors with Traffic

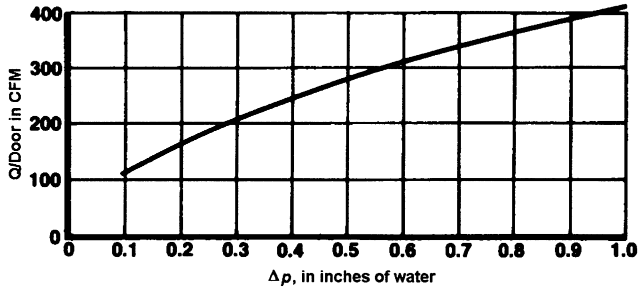


Fig. 5-12 Infiltration Through Revolving Door Seals while Stationary

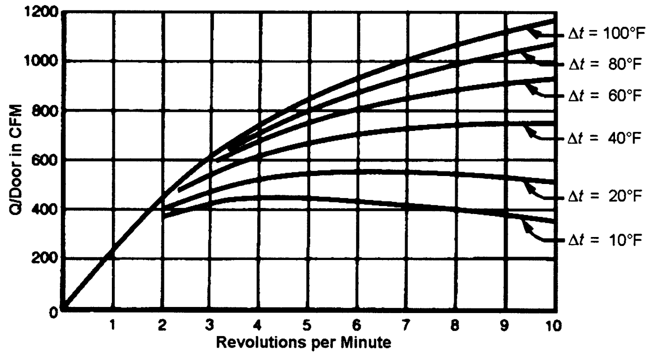


Fig. 5-13 Infiltration for Motor-Operated Revolving Door

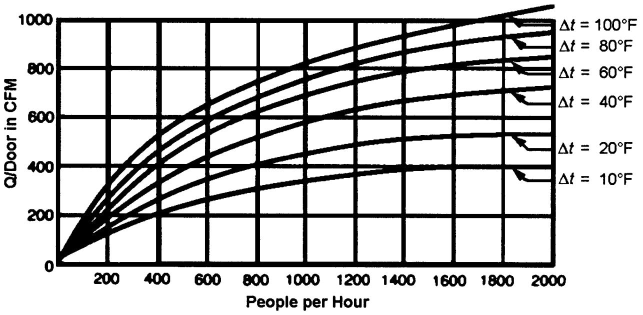


Fig. 5-14 Infiltration for Manual Revolving Door

the major considerations have included the amount of outdoor air required to control moisture, carbon dioxide, odors, and tobacco smoke generated by occupants. These considerations have led to the minimum rate of outdoor air supply per occupant. More recently, the maintenance of acceptable indoor concentrations of a variety of additional pollutants that are not generated primarily by occupants has been a major concern. The most common pollutants of concern and their sources are given in Table 5-7.

Regardless of the complexities and uncertainties that exist regarding proper ventilation, building designers and operators require guidance on minimum outdoor air quantities and indoor air quality. ASHRAE Standard 62.1 provides guidance on ventilation and indoor air quality in the form of alternative procedures. In the **ventilation rate procedure**, indoor air quality is assumed to be acceptable if the concentrations of pollutants in the incoming outdoor air meet the US national ambient air quality standards (Table 5-8) and the outdoor air supply rates meet or exceed values provided in Table 5-9. ANSI/ASHRAE Standard 62.1-2016 is the latest edition of Standard 62, which has been given the new designation of 62.1 to distinguish it from ANSI/ASHRAE Standard 62.2-2016, *Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings*. The purpose of this standard is to specify minimum ventilation rates and indoor air quality that will be acceptable to human occupants and are intended to minimize the potential for adverse health effects. It is intended for regulatory application to new buildings and additions and some changes to existing buildings. It is also intended to be used to guide the improvement of indoor air quality in existing buildings.

Table 5-7 Indoor Air Pollutants and Sources

Sources	Contaminants
OUTDOOR	
Ambient air	SO_2, NO, NO_2, O_3, hydrocarbons, CO, particulates, bioaerosols
Motor vehicles	CO, Pb, hydrocarbons, particulates
Soil	Radon organics
INDOOR	
Building construction materials	
Concrete, stone	Radon
Particleboard, plywood	Formaldehyde
Insulation	Formaldehyde, fiberglass
Fire retardant	Asbestos
Adhesives	Organics
Paint	Mercury, organics
Building Contents	
Heating and cooking combustion appliances	CO, NO, NO_2, formaldehyde, particulates, organics
Furnishings	Organics
Water service, natural gas	Radon
Human Occupants	
Tobacco smoke	CO, NO_2, organics, particulates, odors
Aerosol sprays	Fluorocarbons, vinyl chloride, organics
Cleaning and cooking products	Organics, NH_2, odors
Hobbies and crafts	Organics
Damp organic materials, stagnant water	
Coil drain pans	Bioaerosols
Humidifiers	

Table 5-8 United States Ambient Air Quality Standards
(Table I-1, ANSI/ASHRAE Standard 62.1-2016)

Pollutant	Primary Standard Levels	Averaging Times	Secondary Standard Levels
Carbon monoxide	9 ppm (10 mg/m^3)	8-hour[a]	None
	35 ppm (40 mg/m^3)	1-hour[a]	None
Lead	0.15 μg/m^3	Rolling three-month average	Same as primary
Nitrogen dioxide	100 ppb	1-hour[b]	—
	0.053 ppm (100 μg/m^3)	Annual (arithmetic mean)	Same as primary
Particulate matter (PM10)	150 μg/m^3	24-hour[c]	Same as primary
Particulate matter (PM2.5)	12 μg/m^3	Annual[d] (arithmetic mean)	15 μg/m^3
Ozone	0.075 ppm	8-hour[e]	Same as primary
Sulfur dioxide	75 ppb	1-hour[f]	—
	—	3-hour(1)	0.5 ppm

a. Not to be exceeded more than once per year.
b. 98th percentile, averaged over 3 years
c. Not to be exceeded more than once per year on average over 3 years.
d. Average over three years.
e. 3-year average of the fourth-highest daily maximum 8-hour average ozone concentration.
f. 99th percentile of 1-hour daily maximum concentrations, averaged over 3 years.

Air movement within spaces affects the diffusion of ventilation air and, therefore, indoor air quality and comfort. **Ventilation effectiveness** is a description of an air distribution system's ability to remove internally generated pollutants from a building, zone, or space. **Air change effectiveness** is a description of an air distribution system's ability to deliver ventilation air to a building, zone, or space. The HVAC design engineer does not have knowledge or control of actual pollutant sources within buildings, so Table 5-9 (Table 6.2.2.1 from ASHRAE Standard 62.1-2016) defines outdoor air requirements for typical, expected building uses. For most projects, therefore, the air change effectiveness is of more relevance to HVAC system design than the ventilation effectiveness.

Currently, the HVAC design engineer must assume that a properly designed, installed, operated, and maintained air distribution system provides an air change effectiveness of about 1. Therefore, the Table 5-9 values from ASHRAE Standard 62.1 are appropriate for design of commercial and institutional buildings when the ventilation rate procedure is used. If the indoor air quality procedure of Standard 62.1 is used, then actual pollutant sources and the air change effectiveness must be known for the successful design of HVAC systems that have fixed ventilation airflow rates. Where appropriate, the table lists the estimated density of people for design purposes. The requirements for ventilation air quantities given in Table 5-9 are for 100% outdoor air when the outdoor air quality meets the national specifications for acceptable outdoor air quality.

Properly cleaned air may be recirculated. Under the ventilation rate procedure, for other than intermittent variable occupancy, outdoor air flow rates may not be reduced below the requirements in Table 5-9. If cleaned, recirculated air is used to reduce the outdoor air flow rate below the values of Table 5-9, the indoor air quality procedure must be used. The ***indoor air quality procedure*** provides an alternative performance method to the ***ventilation rate procedure*** for achieving acceptable air quality. The ventilation rate procedure is deemed to provide acceptable indoor air quality. Nevertheless that procedure, through prescription of required ventilation rates, provides only an indirect solution to the control of indoor contaminants. The indoor air quality procedure provides a direct solution by restricting contaminant concentration concern to acceptable levels.

The amount of outdoor air specified in Table 5-9 may be reduced by recirculating air from which offending contaminants have been removed or converted to less objectionable forms. The amount of outdoor air required depends on the contaminant concentrations in the indoor and outdoor air, the contaminant generation in the space, the filter location, the filter efficiency for the contaminants in question, the ventilation effectiveness, the supply air circulation rate, and the fraction recirculated.

In ASHRAE Standard 62.1 a third procedure is defined. This is the Natural Ventilation Procedure and allows some exceptions using natural ventilation in conjunction with the Ventilation Rate Procedure or the Indoor Air Quality Procedure.

Figure 5-15 shows a representative HVAC ventilation system. A filter may be located in the recirculated airstream (location A) or in the supply (mixed) airstream (location B). Variable air volume (VAV) systems reduce the circulation rate when the thermal load is satisfied. This is accounted for by a flow reduction factor F_r. A mass balance for the contaminant may be written to determine the space contaminant concen-

Table 5-9 Minimum Ventilation Rates in Breathing Zone
(This table is not valid in isolation; it must be used in conjunction with the accompanying notes.)

(Table 6.2.2.1, ASHRAE Standard 62.1-2016)

	People Outdoor Air Rate R_p		Area Outdoor Air Rate R_a			Default Values: Occupant Density (see Note 4)	Default Values: Combined Outdoor Air Rate (see Note 5)		
Occupancy Category	**cfm/person**	**L/s·person**	**cfm/ft²**	**L/s·m²**	**Notes**	**#/1000 ft² or #/100 m²**	**cfm/person**	**L/s·person**	**Air Class**
Correctional Facilities									
Cell	5	2.5	0.12	0.6		25	10	4.9	2
Dayroom	5	2.5	0.06	0.3		30	7	3.5	1
Guard stations	5	2.5	0.06	0.3		15	9	4.5	1
Booking/waiting	7.5	3.8	0.06	0.3		50	9	4.4	2
Educational Facilities									
Daycare (through age 4)	10	5	0.18	0.9		25	17	8.6	2
Daycare sickroom	10	5	0.18	0.9		25	17	8.6	3
Classrooms (ages 5–8)	10	5	0.12	0.6		25	15	7.4	1
Classrooms (age 9 plus)	10	5	0.12	0.6		35	13	6.7	1
Lecture classroom	7.5	3.8	0.06	0.3	H	65	8	4.3	1
Lecture hall (fixed seats)	7.5	3.8	0.06	0.3	H	150	8	4.0	1
Art classroom	10	5	0.18	0.9		20	19	9.5	2
Science laboratories	10	5	0.18	0.9		25	17	8.6	2
University/college laboratories	10	5	0.18	0.9		25	17	8.6	2
Wood/metal shop	10	5	0.18	0.9		20	19	9.5	2
Computer lab	10	5	0.12	0.6		25	15	7.4	1
Media center	10	5	0.12	0.6	A	25	15	7.4	1
Music/theater/dance	10	5	0.06	0.3	H	35	12	5.9	1
Multiuse assembly	7.5	3.8	0.06	0.3	H	100	8	4.1	1
Food and Beverage Service									
Restaurant dining rooms	7.5	3.8	0.18	0.9		70	10	5.1	2
Cafeteria/fast-food dining	7.5	3.8	0.18	0.9		100	9	4.7	2
Bars, cocktail lounges	7.5	3.8	0.18	0.9		100	9	4.7	2
Kitchen (cooking)	7.5	3.8	0.12	0.6		20	14	7.0	2
General									
Break rooms	5	2.5	0.06	0.3	H	25	7	3.5	1
Coffee stations	5	2.5	0.06	0.3	H	20	8	4	1
Conference/meeting	5	2.5	0.06	0.3	H	50	6	3.1	1
Corridors	—	—	0.06	0.3	H	—			1
Occupiable storage rooms for liquids or gels	5	2.5	0.12	0.6	B	2	65	32.5	2
Hotels, Motels, Resorts, Dormitories									
Bedroom/living room	5	2.5	0.06	0.3	H	10	11	5.5	1
Barracks sleeping areas	5	2.5	0.06	0.3	H	20	8	4.0	1
Laundry rooms, central	5	2.5	0.12	0.6		10	17	8.5	2
Laundry rooms within dwelling units	5	2.5	0.12	0.6		10	17	8.5	1
Lobbies/prefunction	7.5	3.8	0.06	0.3	H	30	10	4.8	1
Multipurpose assembly	5	2.5	0.06	0.3	H	120	6	2.8	1
Office Buildings									
Breakrooms	5	2.5	0.12	0.6		50	7	3.5	1
Main entry lobbies	5	2.5	0.06	0.3	H	10	11	5.5	1
Occupiable storage rooms for dry materials	5	2.5	0.06	0.3		2	35	17.5	1
Office space	5	2.5	0.06	0.3	H	5	17	8.5	1
Reception areas	5	2.5	0.06	0.3	H	30	7	3.5	1
Telephone/data entry	5	2.5	0.06	0.3	H	60	6	3.0	1
Miscellaneous Spaces									
Bank vaults/safe deposit	5	2.5	0.06	0.3	H	5	17	8.5	2
Banks or bank lobbies	7.5	3.8	0.06	0.3	H	15	12	6.0	1
Computer (not printing)	5	2.5	0.06	0.3	H	4	20	10.0	1
Freezer and refrigerated spaces (<50°F)	10	5	0	0	E	0	0	0	2
General manufacturing (excludes heavy industrial and processes using chemicals)	10	5.0	0.18	0.9		7	36	18	3
Pharmacy (prep. area)	5	2.5	0.18	0.9		10	23	11.5	2
Photo studios	5	2.5	0.12	0.6		10	17	8.5	1
Shipping/receiving	10	5	0.12	0.6	B	2	70	35	2
Sorting, packing, light assembly	7.5	3.8	0.12	0.6		7	25	12.5	2
Telephone closets	—	—	0.00	0.0		—			1
Transportation waiting	7.5	3.8	0.06	0.3	H	100	8	4.1	1
Warehouses	10	5	0.06	0.3	B	—			2
Public Assembly Spaces									
Auditorium seating area	5	2.5	0.06	0.3	H	150	5	2.7	1
Places of religious worship	5	2.5	0.06	0.3	H	120	6	2.8	1
Courtrooms	5	2.5	0.06	0.3	H	70	6	2.9	1
Legislative chambers	5	2.5	0.06	0.3	H	50	6	3.1	1
Libraries	5	2.5	0.12	0.6		10	17	8.5	1
Lobbies	5	2.5	0.06	0.3	H	150	5	2.7	1
Museums (children’s)	7.5	3.8	0.12	0.6		40	11	5.3	1
Museums/galleries	7.5	3.8	0.06	0.3	H	40	9	4.6	1
Residential									
Dwelling unit	5	2.5	0.06	0.3	F,G, H	F			1
Common corridors	—	—	0.06	0.3	H				1
Retail									
Sales (except as below)	7.5	3.8	0.12	0.6		15	16	7.8	2

Table 5-9 Minimum Ventilation Rates in Breathing Zone ***(Continued)***
(This table is not valid in isolation; it must be used in conjunction with the accompanying notes.)
(Table 6.2.2.1, ASHRAE Standard 62.1-2016)

	People Outdoor Air Rate R_p		Area Outdoor Air Rate R_a			Default Values			
						Occupant Density (see Note 4)	Combined Outdoor Air Rate (see Note 5)		
Occupancy Category	**cfm/person**	**L/s·person**	**cfm/ft²**	**L/s·m²**	**Notes**	**#/1000 ft² or #/100 m²**	**cfm/person**	**L/s·person**	**Air Class**
Mall common areas	7.5	3.8	0.06	0.3	H	40	9	4.6	1
Barbershop	7.5	3.8	0.06	0.3	H	25	10	5.0	2
Beauty and nail salons	20	10	0.12	0.6		25	25	12.4	2
Pet shops (animal areas)	7.5	3.8	0.18	0.9		10	26	12.8	2
Supermarket	7.5	3.8	0.06	0.3	H	8	15	7.6	1
Coin-operated laundries	7.5	3.8	0.12	0.6		20	14	7.0	2
Sports and Entertainment									
Gym, sports arena (play area)	20	10	0.18	0.9	E	7	45	23	2
Spectator areas	7.5	3.8	0.06	0.3	H	150	8	4.0	1
Swimming (pool & deck)	—	—	0.48	2.4	C	—			2
Disco/dance floors	20	10	0.06	0.3	H	100	21	10.3	2
Health club/aerobics room	20	10	0.06	0.3		40	22	10.8	2
Health club/weight rooms	20	10	0.06	0.3		10	26	13.0	2
Bowling alley (seating)	10	5	0.12	0.6		40	13	6.5	1
Gambling casinos	7.5	3.8	0.18	0.9		120	9	4.6	1
Game arcades	7.5	3.8	0.18	0.9		20	17	8.3	1
Stages, studios	10	5	0.06	0.3	D, H	70	11	5.4	1

GENERAL NOTES FOR TABLE 5.9

1 **Related requirements:** The rates in this table are based on all other applicable requirements of this standard being met.
2 **Environmental Tobacco Smoke:** This table applies to ETS-free areas. Refer to Section 5.17 for requirements for buildings containing ETS areas and ETS-free areas.
3 **Air density:** Volumetric airflow rates are based on dry air density of 0.075 $\mathrm{lb_{da}/ft^3}$ (1.2 $\mathrm{kg_{da}/m^3}$) at a barometric pressure of 1 atm (101.3 kPa) and an air temperature of 70°F (21°C). Rates shall be permitted to be adjusted for actual density.
4 **Default occupant density:** The default occupant density shall be used where the actual occupant density is not known.
5 **Default combined outdoor air rate (per person):** Rate is based on the default occupant density.
6 **Unlisted occupancies:** Where the occupancy category for a proposed space or zone is not listed, the requirements for the listed occupancy category that is most similar in terms of occupant density, activities, and building construction shall be used.

ITEM-SPECIFIC NOTES FOR TABLE 5.9

A For high-school and college libraries, the values shown for "Public Assembly Spaces—Libraries" shall be used.
B Rate may not be sufficient where stored materials include those having potentially harmful emissions.
C Rate does not allow for humidity control. "Deck area" refers to the area surrounding the pool that is capable of being wetted during pool use or when the pool is occupied. Deck area that is not expected to be wetted shall be designated as an occupancy category.
D Rate does not include special exhaust for stage effects such as dry ice vapors and smoke.
E Where combustion equipment is intended to be used on the playing surface or in the space, additional dilution ventilation, source control, or both shall be provided.
F Default occupancy for dwelling units shall be two persons for studio and one-bedroom units, with one additional person for each additional bedroom.
G Air from one residential dwelling shall not be recirculated or transferred to any other space outside of that dwelling.
H Ventilation air for this occupancy category shall be permitted to be reduced to zero when the space is in occupied-standby mode.

tration for each of the system arrangements. The various permutations for the air-handling and distribution systems are described in Table 5-10. The mass balance equations for computing the space contaminant concentration for each system are included in Table 5-10.

If the allowable space contamination is specified, the equations in Table 5-10 may be solved for the outdoor flow rate. When the outdoor flow rate is specified, the equations may be used to determine the resulting contaminant concentration.

Local codes and ordinances frequently specify outdoor air ventilation requirements for public places and for industrial installations and must be checked and complied with if their requirements are for greater quantities of outdoor air than provided above.

Example 5-3 A small fast-food cafeteria building, 30 × 100 × 9 ft, located in downtown Chicago, IL, has windows and doors on the east and north sides, but none on the south and west. The HVAC system is to include a humidifier. Estimate the winter design heat losses due to ventilation and/or infiltration. Indoor and outdoor design conditions are specified as 72°F, 30% rh and –3°F, respectively.

Based on ASHRAE Standard 62, minimum ventilation is to be at the rate of 7.5 cfm/person with a density of 100 people per 1000 ft² plus 0.18 cfm/ft². Thus,

$$\text{Ventilation} = [(30 \times 100)/1000] \times 100 \times 7.5 + (30 \times 100)(0.18) = 2790 \text{ cfm}$$

With the limited number of openings, infiltration is estimated as 1/2 ach (see Table 5-1) plus that due to traffic through swinging doors, approximated as 2000 cfm from Figures 5-10 and 5-11. Thus,

$$\text{Infiltration} = (1/2)(30\ ´\ 100\ ´\ 9)/60 + 2000 = 2225 \text{ cfm}$$

The ventilation airflow of 2790 cfm is larger than the estimated 2225 cfm of infiltration and should be sufficient to pressurize the building, actually producing exfiltration, rather than having infiltration.

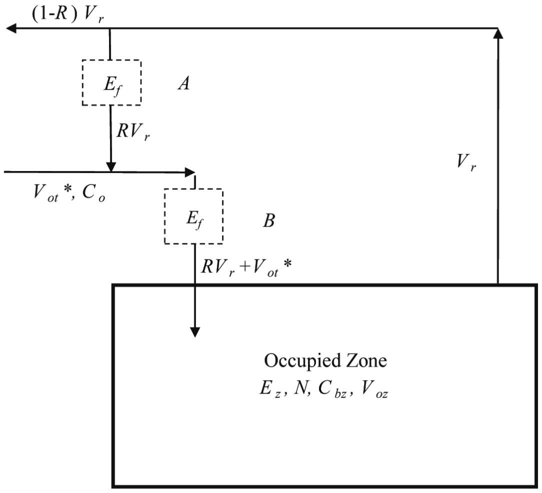
The design heat losses due to outdoor air entering the building due to ventilation are determined as

$$q_s = 1.10(2790)[72 - (-3)] = 230{,}000 \text{ Btu/h}$$

$$q_l = 4840(2790)(0.005 - 0.0007) = 58{,}000 \text{ Btu/h}$$

$$q_{\text{total}} = 230{,}000 + 58{,}000 = 288{,}000 \text{ Btu/h}$$

It is necessary for the designer to read and be familiar with ASHRAE Standard 62.1-2016 before designing ventilation and air distribution systems.



Symbol or Subscript	Definition
A, B	filter location
V	volumetric flow
C	contaminant concentration
E_z	zone air distribution effectiveness
E_f	filter efficiency
F_r	design flow reduction fraction factor
N	contaminant generation rate
R	recirculation flow factor
Subscript: *o*	outdoor
Subscript: *r*	return
Subscript: *b*	breathing
Subscript: *z*	zone

Fig. 5-15 HVAC Ventilation System Schematic
(Figure E-1, ASHRAE Standard 62.1-2016)

Table 5-10 Required Zone Outdoor Airflow or Space Breathing Zone Contaminant Concentration with Recirculation and Filtration for Single-Zone Systems
(Table E-1, ASHRAE Standard 62.1-2016)

Required Recirculation Rate				
Filter Location	**Flow**	**Outdoor Airflow**	**Required Zone Outdoor Airflow (V_{oz} in Section 6)**	**Space Breathing Zone Contaminant Concentration**
None	VAV	100%	$V_{oz} = \frac{N}{E_z F_r (C_{bz} - C_o)}$	$C_{bz} = C_o + \frac{N}{E_z F_r V_{oz}}$
A	Constant	Constant	$V_{oz} = \frac{1 - E_z R V_r E_f C_{bz}}{E_z (C_{bz} - C_o)}$	$C_{bz} = \frac{N + E_z V_{oz} C_o}{E_z (V_{oz} + R V_r E_f)}$
A	VAV	Constant	$V_{oz} = \frac{N - E_z F_r R V_r E_f C_{bz}}{E_z (C_{bz} - C_o)}$	$C_{bz} = \frac{N + E_z V_{oz} C_o}{E_z (V_{oz} + F_r R V_r E_f)}$
B	Constant	Constant	$V_{oz} = \frac{N - E_z R V_r E_f C_{bz}}{E_z [C_{bz} - (1 - E_f)(C_o)]}$	$C_{bz} = \frac{N + E_z V_{oz} (1 - E_f) C_o}{E_z (V_{oz} + R V_r E_f)}$
B	VAV	100%	$V_{oz} = \frac{N}{E_z F_r [C_{bz} - (1 - E_f)(C_o)]}$	$C_{bz} = \frac{N + E_z F_r V_{oz} (1 - E_f) C_o}{E_z F_r V_{oz}}$
B	VAV	Constant	$V_{oz} = \frac{N - E_z F_r R V_r E_f C_{bz}}{E_z [C_{bz} - (1 - E_f)(C_o)]}$	$C_{bz} = \frac{N + E_z V_{oz} (1 - E_f) C_o}{E_z (V_{oz} + F_r R V_r E_f)}$

5.3 Heat Transfer Coefficients

5.3.1 Modes of Heat Transfer

The design of a heating, refrigerating, or air-conditioning system, including selection of building insulation or sizing of piping and ducts, or the evaluation of the thermal performance of system parts such as chillers, heat exchangers, and fans, is based on the principles of heat transfer given in Chapter 4, "Heat Transfer," of the 2017 *ASHRAE Handbook—Fundamentals*.

Whenever a temperature difference between two areas (indoor-outdoor) exists, heat flows from the warmer area to the cooler area. The flow or transfer of heat takes place by one or more of three modes—conduction, convection, or radiation.

Conduction is the transfer of heat through a solid. When a poker is left in a fire, the handle is also warmed even though it is not in direct contact with the flame. The flow of heat along the poker is by conduction. The rate of flow is influenced by the temperature difference, the area of the material, the distance through the material from warm side to cool side, and the thermal conductivity of the material. Insulating materials have low thermal conductivity, which, combined with their thickness, provide a barrier that slows conductive heat transfer.

Convection transfers heat by movement through liquids or gases. As a gas (e.g., air) is heated, it expands, becomes lighter, and rises. It is then displaced with cooler air which follows the same cycle, carrying heat with it. The continuous cycle of rising warm air and descending cool air is a convection current. By dividing a large space into many small spaces and providing barriers that restrict convection, the flow of heat can be slowed. The mat of random fibers in batt-type insulation, such as mineral wool or wood fiber insulation, provides such barriers.

Radiation is a method of heat transfer whereby a warm object can heat a cool object without the need of a solid, liquid, or gas between them. An example of radiation is heat from the sun passing through the vacuum of space and warming the earth. In another example, standing in front of a campfire results in the warming of those parts of the body that face the fire. For other body parts to be warmed, the body position must be reversed. The body can also be the radiator and radiate heat to cooler objects.

The **U-factor** and the **R-value** are used to indicate the relative insulating value of materials and sections of walls, floors, and ceilings. The U-factor indicates the rate at which heat flows through a specific material or a building section (Figure 5-16). The smaller the U-factor, the better the insulating value of the material or group of materials making up the wall, ceiling, or floor.

The R-value indicates the ability of one specific material, or a group of materials in a building section, to resist heat flow through them. Many insulating materials now have their R-value stamped on the outside of the package, batt, or blanket. The R-value and relative heat-resisting values of several of these materials are listed in Table 5-11. The R-value for batt, blanket, and loose-fill insulation as listed in this figure is for a thickness of 1 in. The R-value for greater thicknesses can be determined by multiplying the thickness desired, in inches, by the R-value listed.

The greater the R-value, the greater the insulating value of the material, and the lower the heat loss. Thus, a high R-value means lower heating and cooling costs and less energy used to maintain a comfortable temperature.

Thermal conductivity *k* is a property of a homogeneous material. Building materials, such as lumber, brick, and stone, are usually considered homogeneous. Most thermal insulation and many other building materials are porous and consist of combinations of solid matter with small voids. For most insulating materials, conduction is not the only mode of heat transfer. Consequently, the term **apparent thermal conductivity** describes the heat flow properties of most materials. Some materials with low thermal conductivities are almost purely conductive (silica opacified aerogel, corkboard, etc.). The apparent thermal conductivity of insulation varies with form and physical structure, environment, and application conditions. Form and physical structure vary with the basic material and manufacturing process. Typical variations include density, cell size, diameter and arrangement of fibers or particles, degree and extent of bonding materials, transparency to thermal radiation, and the type and pressure of gas within the insulation.

The method of calculating an overall coefficient of heat transmission requires knowledge of (1) the apparent thermal conductivity and thickness of homogeneous components, (2) thermal conductance of nonhomogeneous components, (3) surface conductances of both sides of the construction, and (4) conductance of air spaces in the construction.

Surface conductance is the heat transfer to or from the surface by the combined effects of radiation, convection, and conduction. Each of these transport modes can vary independently. Heat transfer by radiation between two surfaces is controlled by the character of the surfaces (emittance and reflectivity), the temperature difference between them, and the solid angle through which they see each other. Heat transfer by convection and conduction is controlled by surface roughness, air movement, and temperature difference between the air and surface.

Heat transfer across an air space is affected by the nature of the boundary surfaces, as well as the intervening air space, the orientation of the air space, the distance between boundary sur-

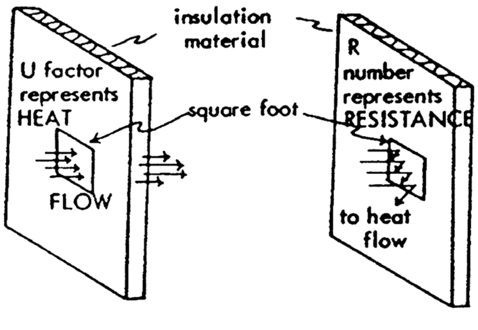


Fig. 5-16 U-Factor and R-value.

faces, and the direction of heat flow. Air space conductance coefficients represent the total conductance from one surface bounding the air space to the other. The total conductance is the sum of a radiation component and a convection and conduction component. In all cases, the spaces are considered airtight with no through air leakage.

The combined effect of the emittances of the boundary surfaces of an air space is expressed by the effective emittance *E* of the air space. The radiation component is affected only slightly by the thickness of the space, its orientation, the direction of heat flow, or the order of emittance (hot or cold surface). The heat transfer by convection and conduction combined is affected markedly by orientation of the air space and the direction of heat flow, by the temperature difference across the space, and, in some cases, by the thickness of the space. It is also slightly affected by the mean temperature of its surfaces.

The steady-state thermal resistances (R-values) of building components (walls, floors, windows, roof systems, etc.) can be calculated from the thermal properties of the materials in the component; or the heat flow through the assembled component can be measured directly with laboratory equipment such as the guarded hot box (ASTM Standard C236) or the calibrated hot box (ASTM Standard C976).

Tables 5-12 through 5-15 list thermal values which may be used to calculate thermal resistances of building wall, floors, and ceilings. The values shown in these tables were developed under ideal conditions. In practice, overall thermal performance can be reduced significantly by such factors as improper installation and shrinkage, settling, or compression of the insulation.

The performance of materials fabricated in the field is especially subject to the quality of workmanship during construction and installation. Good workmanship becomes increasingly important as the insulation requirement becomes greater. Therefore, some engineers include additional insulation or other safety factors based on experience in their design.

When installing insulation, irregular areas must be given careful attention. Blanket-type insulation should be sealed by stapling or taping it to the floor, ceiling plates, and studs. Insulating material should be carefully fitted around all plumbing, wiring, and other projections. The proper thickness should be maintained throughout the walls, ceilings, and floors. To get the most value from reflective materials, such as aluminum foil facing on batts or blankets, allow a 0.6 in. (15 mm) airspace between the foil and the wallboard.

In order to obtain the proper density of insulation, competent operators using special equipment blow loose fill insulation (be it for new or remodeled homes) into the walls, floors, and ceilings. Insulation should be blown into each wall cavity at both top and bottom so that all spaces are filled and variations in density are minimized. Variations in insulation density affect the resistance value and the R-value of material. Loose fill insulation tends to settle, resulting in high density insulation below and none above. Therefore, batt or blanket insulation is preferable for vertical spaces in new construction.

Insulation resists heat flow in proportion to its R-value *only* if it is installed according to these general recommendations and the manufacturer's instructions. Wall voids are frequently left open at the top of the stud space, thereby allowing outdoor air to enter and greatly affect the U-factor of the wall.

Common variations in conditions, materials, workmanship, and so forth, can introduce much greater variations in U-values than the variations resulting from assumed mean temperatures and temperature differences. *Therefore, stating a U-factor of more than two significant figures may assume more precision than can possibly exist.*

Shading devices, such as venetian blinds, draperies, and roller shades, substantially reduce the U-factor for windows and/or glass doors if they fit tightly to the window jambs, head, and sill, and are made of nonporous material. As a rough approximation, tight-fitting shading devices may be considered to reduce the U-factor of vertical exterior single glazing by 25% and of vertical exterior double glazing and glass block by 15%. These adjustments should not be considered in choosing heating equipment, but may be used for calculating design cooling loads.

5.3.2 Determining U-Factors

The total resistance to heat flow through building construction such as a flat ceiling, floor, or wall (or curved surface if the curvature is small) is the numerical sum of the resistances (R-values) of all parts of the construction in series:

$$R = R_1 + R_2 + R_3 + R_4 + ... + R_n$$

where

R_1, R_2, ..., R_n = individual resistances of the parts
R = resistance of the construction from inside surface to outside surface

However, in buildings, to obtain the overall resistance R_T, the air film resistances R_i and R_o must be added to R.

$$R_T = \mathrm{R}_i + R + \mathrm{R}_o \qquad (5\text{-}9)$$

The U-factor (thermal transmittance) is the reciprocal of R_T:

$$U = \frac{1}{R_T}$$

Thus, U is computed by adding up all of the R-values, including those of inside- and outside-air films, the air gap, and all building materials.

With the use of higher values of R_T, the corresponding values of U become very small. This is one reason why it is sometimes preferable to specify resistance rather than transmittance. Also, a whole number is more understandable to an insulation buyer than is a decimal or fraction.

For a wall with air space construction, consisting of two homogeneous materials of conductivities k_1 and k_2 and thickness x_1 and x_2, respectively, and separated by an air space of conductance C, the overall resistance would be determined from

$$R_T = \frac{1}{h_i} + \frac{x_1}{k_1} + \ldots + \frac{x_n}{k_n} + \frac{1}{h_0}$$

where h_i and h_0 are the heat transfer film coefficients.

Table 5-11 Relative Thermal Resistances of Building Material

Material Description	Material Density, lb/ft^3	Material Thickness, in.	Resistance for Thickness Listed, °F·ft^2·h/Btu
Building paper	–	–	0.06
Gypsum plaster, sand aggregate	105	1/2	0.09
Structural glass	–	–	0.10
Air surface, 15 mph wind, outside surface	–	–	0.17
Gypsum or plaster board	50	3/8	0.32
Stone, lime, or sand	–	4	0.32
Concrete, sand-gravel aggregate	140	4	0.32
Built-up roofing	70	3/8	0.33
Brick, face	130	4	0.44
Still air surface, horiz., ordinary materials, heat flow up	–	–	0.61
Aluminum, steel, or vinyl over sheathing, hollow backed	–	–	0.61
Plywood	34	1/2	0.63
Still air surface, vertical, ordinary mtrls, horiz. heat flow	–	–	0.68
Wood siding, bevel, 1/2 in 8 in. lapped	–	–	0.81
Wood shingle siding, 16 in., 7 1/2 in. exposure	–	–	0.87
Oak, maple, and similar hardwoods	45	1	0.91
Air space, vertical, ordinary materials, horiz. heat flow	–	3/4 to 4	0.97
Clay tile, one cell deep	–	4	1.11
Concrete block, 3 core, sand-gravel aggregate	–	8	1.11
Acoustical tile, wood or cane fiber	–	1/2	1.19
Fir, pine, and similar softwoods	32	1	1.25
Insulation board, impregnated	20	1/2	1.32
Concrete, lightweight aggregate	80	4	1.50
Air space, vertical, bounded by reflective material	–	3/4 to 4	1.70
Concrete block, 3 core, cinder aggregate	–	8	1.72
Concrete block, 3 core, lightweight aggregate	–	8	2.00
Vermiculite, expanded	7	1	2.08
Carpet and fibrous pad	–	–	2.08
Cellular glass insulation board	9	1	2.50
Roof insulation, preformed for above deck	–	1	2.78
Mineral wool, loose fill, from slag glass or rock	2–5	1	3.33
Wood fiber, loose fill, hemlock, fir or redwood	2–3.5	1	3.33
Plastic, foamed	1.62	1	3.45
Macerated paper or pulp	2–3.5	1	3.57
Corkboard, without added binder	6.5–8	1	3.70
Batt and Blankets Bounded by Nonreflective Materials			
Mineral wool, fibrous form, rock, slag, or glass	1.5–4	1	3.70
Wood fiber, multilayer, stitched expanded	1.5–2	1	3.70
Cotton fiber	0.8–2	1	3.85
Wood fiber	3.2–3.6	1	4.00

Relative Value of Resistance to Heat flow

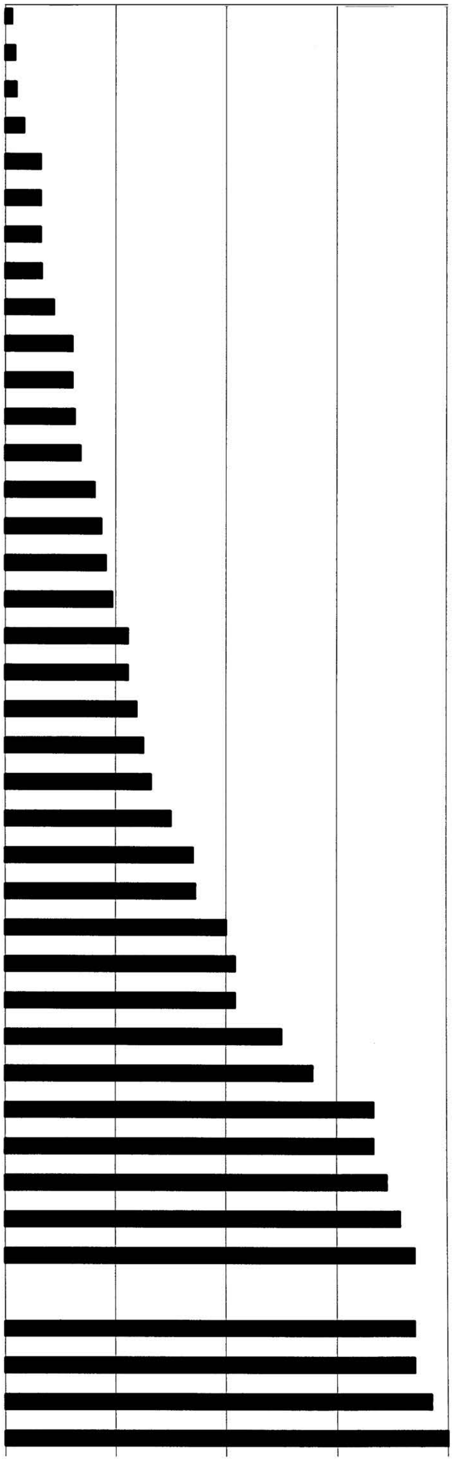


Table 5-12 Surface Film Coefficients/Resistances

(Table 10, Chapter 26, 2017 ASHRAE Handbook—Fundamentals)

Position of Surface	Direction of Heat Flow	Surface Emittance, ε Non-reflective ε = 0.90 h_i	 R	 Reflective ε = 0.20 h_i	 R	 ε = 0.05 h_i	 R
STILL AIR							
Horizontal	Upward	1.63	0.61	0.91	1.10	0.76	1.32
Sloping—45°	Upward	1.60	0.62	0.88	1.14	0.73	1.37
Vertical	Horizontal	1.46	0.68	0.74	1.35	0.59	1.70
Sloping—45°	Downward	1.32	0.76	0.60	1.67	0.45	2.22
Horizontal	Downward	1.08	0.92	0.37	2.70	0.22	4.55
MOVING AIR (Any position)		h_o	R				
15 mph Wind (for winter)	Any	6.00	0.17	—	—	—	—
7.5 mph Wind (for summer)	Any	4.00	0.25	—	—	—	—

Notes: (References are to Chapter 26 in the 2013 *ASHRAE Handbook—Fundamentals*)

1. Surface conductance h_i and h_o measured in Btu/h·ft^2·°F; resistance R in °F·ft^2·h/Btu.
2. No surface has both an air space resistance value and a surface resistance value.
3. Conductances are for surfaces of the stated emittance facing virtual blackbody surroundings at the same temperature as the ambient air. Values are based on a surface-air temperature difference of 10°F and for surface temperatures of 70°F.
4. See Chapter 4 in the 2013 *ASHRAE Handbook—Fundamentals* for more detailed information.
5. Condensate can have a significant impact on surface emittance.

Table 5-13 Emissivity of Various Surfaces and Effective Emittances of Facing Air Spaces[a]

(Table 2, Chapter 26, 2017 ASHRAE Handbook—Fundamentals)

Surface	Average Emissivity ε	Effective Emittance ε_{eff} of Air Space One Surface's Emittance ε; Other, 0.9	 Both Surfaces' Emittance ε
Aluminum foil, bright	0.05	0.05	0.03
Aluminum foil, with condensate just visible (>0.7 g/ft^2)(>0.5 g/m^2)	0.30[b]	0.29	—
Aluminum foil, with condensate clearly visible (>2.9 g/ft^2)(>2.0 g/m^2)	0.70[b]	0.65	—
Aluminum sheet	0.12	0.12	0.06
Aluminum-coated paper, polished	0.20	0.20	0.11
Brass, nonoxidized	0.04	0.038	0.02
Copper, black oxidized	0.74	0.41	0.59
Copper, polished	0.04	0.038	0.02
Iron and steel, polished	0.2	0.16	0.11
Iron and steel, oxidized	0.58	0.35	0.41
Lead, oxidized	0.27	0.21	0.16
Nickel, nonoxidized	0.06	0.056	0.03
Silver, polished	0.03	0.029	0.015
Steel, galvanized, bright	0.25	0.24	0.15
Tin, nonoxidized	0.05	0.047	0.026
Aluminum paint	0.50	0.47	0.35
Building materials: wood, paper, masonry, nonmetallic paints	0.90	0.82	0.82
Regular glass	0.84	0.77	0.72

[a] Values apply in 4 to 40 μm range of electromagnetic spectrum. Also, oxidation, corrosion, and accumulation of dust and dirt can dramatically increase surface emittance. Emittance values of 0.05 should only be used where the highly reflective surface can be maintained over the service life of the assembly. Except as noted, data from VDI (1999).

[b] Values based on data in Bassett and Trethowen (1984).

Series and Parallel Heat Flow Paths. In many installations, components are arranged so that heat flows in parallel paths of different conductances. If no heat flows between lateral paths, the U-factor for each path is calculated. The average transmittance is then

$$U_{av} = aU_a + bU_b + ... + nU_n \tag{5-10}$$

where a, b, ..., n are respective fractions of a typical basic area composed of several different paths with transmittances U_a, U_b, ..., U_n.

If heat can flow laterally with little resistance in any continuous layer so that transverse isothermal planes result, total average resistance $R_{T(av)}$ is the sum of the resistance of the layers between such planes. This is a series combination of layers, of which one (or more) provides parallel paths.

The average overall R-values and U-factors of wood frame walls can be calculated by assuming parallel heat flow paths through areas with different thermal resistances.

The following equation is recommended to correct for the effect of framing members.

$$U_{av} = (S/100)U_s + (1 - S/100)U_i \tag{5-11}$$

where

U_{av} = average U-factor for building section
U_i = U-factor for area between framing members
U_s = U-factor for area backed by framing members
S = percentage of area backed by framing members

The framing factor or fraction of the building component that is framing depends on the specific type of construction, and it may vary based on local construction practices even for the same type of construction. For stud walls 16 in. on center (OC) the fraction of insulated cavity may be as low as 0.75, where the fraction of studs, plates, and sills is 0.21 and the fraction of headers is 0.04, For studs 24 in. OC, the respective values are 0.78, 0.18, and 0.04. These fractions contain an allowance for multiple studs, plates, sills, extra framing around windows, headers, and band joists.

Unequal Areas. A construction may be made up of two or more layers of unequal area, separated by an airspace, and arranged so that heat flows through the layers in series. The most common such construction is a ceiling and roof combination where the attic space is unheated and unventilated. A combined coefficient based on the most convenient area, say the ceiling area, from air inside to air outside can be calculated from

$$R_{T,c} = R_c + R_r/n \text{ and } U_c = 1/R_{T,c} \tag{5-12}$$

where

U_c = combined coefficient based on ceiling area
n = ratio of roof to ceiling area, A_r/A_c

Windows and Doors. Table 5-16 lists U-factors for various fenestration products. Tables 5-17 through 5-20 provide U-factors for various exterior doors. All U-factors are approximate, because a significant portion of the resistance of a window or door is contained in the air film resistances, and some parameters that may have important effects are not considered. For

Table 5-14 Effective Thermal Resistance of Plane Air Spaces,[a,b,c] $h\cdot ft^2\cdot{}^\circ F/Btu$

(Table 3, Chapter 26, 2017 ASHR$AE Handbook—Fundamentals)

		Air Space		Effective Emittance ε_{eff}[d,e]									
Position of Air Space	Direction of Heat Flow	Mean Temp.[d], °F	Temp. Diff.,[d] °F	0.5 in. Air Space[c]					0.75 in. Air Space[c]				
				0.03	0.05	0.2	0.5	0.82	0.03	0.05	0.2	0.5	0.82
Horiz.	Up ↑	90	10	2.13	2.03	1.51	0.99	0.73	2.34	2.22	1.61	1.04	0.75
		50	30	1.62	1.57	1.29	0.96	0.75	1.71	1.66	1.35	0.99	0.77
		50	10	2.13	2.05	1.60	1.11	0.84	2.30	2.21	1.70	1.16	0.87
		0	20	1.73	1.70	1.45	1.12	0.91	1.83	1.79	1.52	1.16	0.93
		0	10	2.10	2.04	1.70	1.27	1.00	2.23	2.16	1.78	1.31	1.02
		–50	20	1.69	1.66	1.49	1.23	1.04	1.77	1.74	1.55	1.27	1.07
		–50	10	2.04	2.00	1.75	1.40	1.16	2.16	2.11	1.84	1.46	1.20
45° Slope	Up ↗	90	10	2.44	2.31	1.65	1.06	0.76	2.96	2.78	1.88	1.15	0.81
		50	30	2.06	1.98	1.56	1.10	0.83	1.99	1.92	1.52	1.08	0.82
		50	10	2.55	2.44	1.83	1.22	0.90	2.90	2.75	2.00	1.29	0.94
		0	20	2.20	2.14	1.76	1.30	1.02	2.13	2.07	1.72	1.28	1.00
		0	10	2.63	2.54	2.03	1.44	1.10	2.72	2.62	2.08	1.47	1.12
		–50	20	2.08	2.04	1.78	1.42	1.17	2.05	2.01	1.76	1.41	1.16
		–50	10	2.62	2.56	2.17	1.66	1.33	2.53	2.47	2.10	1.62	1.30
Vertical	Horiz. →	90	10	2.47	2.34	1.67	1.06	0.77	3.50	3.24	2.08	1.22	0.84
		50	30	2.57	2.46	1.84	1.23	0.90	2.91	2.77	2.01	1.30	0.94
		50	10	2.66	2.54	1.88	1.24	0.91	3.70	3.46	2.35	1.43	1.01
		0	20	2.82	2.72	2.14	1.50	1.13	3.14	3.02	2.32	1.58	1.18
		0	10	2.93	2.82	2.20	1.53	1.15	3.77	3.59	2.64	1.73	1.26
		–50	20	2.90	2.82	2.35	1.76	1.39	2.90	2.83	2.36	1.77	1.39
		–50	10	3.20	3.10	2.54	1.87	1.46	3.72	3.60	2.87	2.04	1.56
45° Slope	Down ↘	90	10	2.48	2.34	1.67	1.06	0.77	3.53	3.27	2.10	1.22	0.84
		50	30	2.64	2.52	1.87	1.24	0.91	3.43	3.23	2.24	1.39	0.99
		50	10	2.67	2.55	1.89	1.25	0.92	3.81	3.57	2.40	1.45	1.02
		0	20	2.91	2.80	2.19	1.52	1.15	3.75	3.57	2.63	1.72	1.26
		0	10	2.94	2.83	2.21	1.53	1.15	4.12	3.91	2.81	1.80	1.30
		–50	20	3.16	3.07	2.52	1.86	1.45	3.78	3.65	2.90	2.05	1.57
		–50	10	3.26	3.16	2.58	1.89	1.47	4.35	4.18	3.22	2.21	1.66
Horiz.	Down ↓	90	10	2.48	2.34	1.67	1.06	0.77	3.55	3.29	2.10	1.22	0.85
		50	30	2.66	2.54	1.88	1.24	0.91	3.77	3.52	2.38	1.44	1.02
		50	10	2.67	2.55	1.89	1.25	0.92	3.84	3.59	2.41	1.45	1.02
		0	20	2.94	2.83	2.20	1.53	1.15	4.18	3.96	2.83	1.81	1.30
		0	10	2.96	2.85	2.22	1.53	1.16	4.25	4.02	2.87	1.82	1.31
		–50	20	3.25	3.15	2.58	1.89	1.47	4.60	4.41	3.36	2.28	1.69
		–50	10	3.28	3.18	2.60	1.90	1.47	4.71	4.51	3.42	2.30	1.71
		Air Space		1.5 in. Air Space[c]					3.5 in. Air Space[c]				
Horiz.	Up ↑	90	10	2.55	2.41	1.71	1.08	0.77	2.84	2.66	1.83	1.13	0.80
		50	30	1.87	1.81	1.45	1.04	0.80	2.09	2.01	1.58	1.10	0.84
		50	10	2.50	2.40	1.81	1.21	0.89	2.80	2.66	1.95	1.28	0.93
		0	20	2.01	1.95	1.63	1.23	0.97	2.25	2.18	1.79	1.32	1.03
		0	10	2.43	2.35	1.90	1.38	1.06	2.71	2.62	2.07	1.47	1.12
		–50	20	1.94	1.91	1.68	1.36	1.13	2.19	2.14	1.86	1.47	1.20
		–50	10	2.37	2.31	1.99	1.55	1.26	2.65	2.58	2.18	1.67	1.33
45° Slope	**Up** ↗	90	10	2.92	2.73	1.86	1.14	0.80	3.18	2.96	1.97	1.18	0.82
		50	30	2.14	2.06	1.61	1.12	0.84	2.26	2.17	1.67	1.15	0.86
		50	10	2.88	2.74	1.99	1.29	0.94	3.12	2.95	2.10	1.34	0.96
		0	20	2.30	2.23	1.82	1.34	1.04	2.42	2.35	1.90	1.38	1.06
		0	10	2.79	2.69	2.12	1.49	1.13	2.98	2.87	2.23	1.54	1.16
		–50	20	2.22	2.17	1.88	1.49	1.21	2.34	2.29	1.97	1.54	1.25
		–50	10	2.71	2.64	2.23	1.69	1.35	2.87	2.79	2.33	1.75	1.39
Vertical	**Horiz.** →	90	10	3.99	3.66	2.25	1.27	0.87	3.69	3.40	2.15	1.24	0.85
		50	30	2.58	2.46	1.84	1.23	0.90	2.67	2.55	1.89	1.25	0.91
		50	10	3.79	3.55	2.39	1.45	1.02	3.63	3.40	2.32	1.42	1.01
		0	20	2.76	2.66	2.10	1.48	1.12	2.88	2.78	2.17	1.51	1.14
		0	10	3.51	3.35	2.51	1.67	1.23	3.49	3.33	2.50	1.67	1.23
		–50	20	2.64	2.58	2.18	1.66	1.33	2.82	2.75	2.30	1.73	1.37
		–50	10	3.31	3.21	2.62	1.91	1.48	3.40	3.30	2.67	1.94	1.50

Table 5-14 Effective Thermal Resistance of Plane Air Spaces,[a,b,c] h·ft^2·°F/Btu *(Continued)*

(Table 3, Chapter 26, 2017 ASHR$AE Handbook—Fundamentals)

Air Space				Effective Emittance ε_{eff}[d,e]									
				1.5 in. Air Space[c]					3.5 in. Air Space[c]				
Position of Air Space	Direction of Heat Flow	Mean Temp.[d], °F	Temp. Diff.,[d] °F	0.03	0.05	0.2	0.5	0.82	0.03	0.05	0.2	0.5	0.82
45° Slope	**Down** ↘	90	10	5.07	4.55	2.56	1.36	0.91	4.81	4.33	2.49	1.34	0.90
		50	30	3.58	3.36	2.31	1.42	1.00	3.51	3.30	2.28	1.40	1.00
		50	10	5.10	4.66	2.85	1.60	1.09	4.74	4.36	2.73	1.57	1.08
		0	20	3.85	3.66	2.68	1.74	1.27	3.81	3.63	2.66	1.74	1.27
		0	10	4.92	4.62	3.16	1.94	1.37	4.59	4.32	3.02	1.88	1.34
		–50	20	3.62	3.50	2.80	2.01	1.54	3.77	3.64	2.90	2.05	1.57
		–50	10	4.67	4.47	3.40	2.29	1.70	4.50	4.32	3.31	2.25	1.68
Horiz.	**Down** ↓	90	10	6.09	5.35	2.79	1.43	0.94	10.07	8.19	3.41	1.57	1.00
		50	30	6.27	5.63	3.18	1.70	1.14	9.60	8.17	3.86	1.88	1.22
		50	10	6.61	5.90	3.27	1.73	1.15	11.15	9.27	4.09	1.93	1.24
		0	20	7.03	6.43	3.91	2.19	1.49	10.90	9.52	4.87	2.47	1.62
		0	10	7.31	6.66	4.00	2.22	1.51	11.97	10.32	5.08	2.52	1.64
		–50	20	7.73	7.20	4.77	2.85	1.99	11.64	10.49	6.02	3.25	2.18
		–50	10	8.09	7.52	4.91	2.89	2.01	12.98	11.56	6.36	3.34	2.22

Air Space				5.5 in. Air Space[c]				
Horiz.	Up ↑	90	10	3.01	2.82	1.90	1.15	0.81
		50	30	2.22	2.13	1.65	1.14	0.86
		50	10	2.97	2.82	2.04	1.31	0.95
		0	20	2.40	2.33	1.89	1.37	1.06
		0	10	2,90	2.79	2.18	1.52	1.15
		–50	20	2.31	2.27	1.95	1.53	1.24
		–50	10	2.80	2.73	2.29	1.73	1.37
45° Slope	Up ↗	90	10	3.26	3.04	2.00	1.19	0.83
		50	30	2.19	2.10	1.64	1.13	0.85
		50	10	3.16	2.99	2.12	1.35	0.97
		0	20	2.35	2.28	1.86	1.35	1.05
		0	10	3.00	2.88	2.24	1.54	1.16
		–50	20	2.16	2.12	1.84	1.46	1.20
		–50	10	2.78	2.71	2.27	1.72	1.37
Vertical	Horiz. →	90	10	3.76	3.46	2.17	1.25	0.86
		50	30	2.83	2.69	1.97	1.28	0.93
		50	10	3.72	3.49	2.36	1.44	1.01
		0	20	3.08	2.95	2.28	1.57	1.17
		0	10	3.66	3.49	2.59	1.70	1.25
		–50	20	3.03	2.95	2.44	1.81	1.42
		–50	10	3.59	3.47	2.78	2.00	1.53
45° Slope	Down ↘	90	10	4.90	4.41	2.51	1.35	0.91
		50	30	3.86	3.61	2.42	1.46	1.02
		50	10	4.93	4.52	2.80	1.59	1.09
		0	20	4.24	4/-1	2.86	1.82	1.31
		0	10	4.93	4.63	3.16	1.94	1.37
		–50	20	4.28	4.12	3.19	2.19	1.65
		–50	10	4.93	4.71	3.53	2.35	1.74
Horiz.	Down ↓	90	10	11.72	9.24	3.58	1.61	1.01
		50	30	10.61	8.89	4.02	1.92	1.23
		50	10	12.70	10.32	4.28	1.98	1.25
		0	20	12.10	10.42	5.10	2.52	1.64
		0	10	13.80	11.65	5.38	2.59	1.67
		–50	20	12.45	11.14	6.22	3.31	2.20
		–50	10	14.60	12.83	6.72	3.44	2.26

[a] See Chapter 25 in the 2013 *ASHRAE Handbook—Fundamentals*. Thermal resistance values were determined from $R = 1/C$, where $C = h_c + \varepsilon_{eff} h_r$, h_c is conduction/convection coefficient, $\varepsilon_{eff} h_r$ is radiation coefficient $\approx 0.0068\varepsilon_{eff}[(t_m + 460)/100]^3$, and t_m is mean temperature of air space. Values for h_c were determined from data developed by Robinson et al. (1954). Equations (5) to (7) in Yarbrough (1983) show data in this table in analytic form. For extrapolation from this table to air spaces less than 0.5 in. (e.g., insulating window glass), assume $h_c = 0.159(1 + 0.0016t_m)/l$, where l is air space thickness in in., and h_c is heat transfer through air space only.

[b] Values based on data presented by Robinson et al. (1954). (Also see Chapter 4, Tables 5 and 6, and Chapter 33). Values apply for ideal conditions (i.e., air spaces of uniform thickness bounded by plane, smooth, parallel surfaces with no air leakage to or from the space). **This table should not be used for hollow siding or profiled cladding: see Table 1.** For greater accuracy, use overall U-factors determined through guarded hot box (ASTM Standard C1363) testing. Thermal resistance values for multiple air spaces must be based on careful estimates of mean temperature differences for each air space.

[c] A single resistance value cannot account for multiple air spaces; each air space requires a separate resistance calculation that applies only for established boundary conditions. Resistances of horizontal spaces with heat flow downward are substantially independent of temperature difference.

[d] Interpolation is permissible for other values of mean temperature, temperature difference, and effective emittance ε_{eff}. Interpolation and moderate extrapolation for air spaces greater than 3.5 in. are also permissible.

[e] Effective emittance ε_{eff} of air space is given by $1/\varepsilon_{eff} = 1/\varepsilon_1 + 1/\varepsilon_2 - 1$, where ε_1 and ε_2 are emittancess of air space (see Table 2). **Also, oxidation, corrosion, and accumulation of dust and dirt can dramatically increase surface emittance. Emittance values of 0.05 should only be used where the highly reflective surface can be maintained over the service life of the assembly.**

Table 5-15 Building and Insulating Materials: Design Values[a]

(Table 1, Chapter 26, 2013 ASHRAE Handbook—Fundamentals)

Description	Density, lb/ft^3	Conductivity[b] k, $\text{Btu·in/h·ft}^2\text{·°F}$	Resistance R, $\text{h·ft}^2\text{·°F/Btu}$	Specific Heat, Btu/lb·°F	Reference[l]
Insulating Materials					
Blanket and batt[c,d]					
Glass-fiber batts				0.2	Kumaran (2002)
	0.47 to 0.51	0.32 to 0.33	—	—	Four manufacturers (2011)
	0.61 to 0.75	0.28 to 0.30	—	—	Four manufacturers (2011)
	0.79 to 0.85	0.26 to 0.27	—	—	Four manufacturers (2011)
	1.4	0.23	—	—	Four manufacturers (2011)
Rock and slag wool batts.	—	—	—	0.2	Kumaran (1996)
	2 to 2.3	0.25 to 0.26	—	—	One manufacturer (2011)
	2.8	0.23 to 0.24	—	—	One manufacturer (2011)
Mineral wool, felted	1 to 3	0.28	—	—	CIBSE (2006), NIST (2000)
	1 to 8	0.24	—	—	NIST (2000)
Board and slabs					
Cellular glass	7.5	0.29	—	0.20	One manufacturer (2011)
Cement fiber slabs, shredded wood with Portland cement binder	25 to 27	0.50 to 0.53	—	—	
with magnesia oxysulfide binder	22	0.57	—	0.31	
Glass fiber board	—	—	—	0.2	Kumaran (1996)
	1.5 to 6.0	0.23 to 0.24	—	—	One manufacturer (2011)
Expanded rubber (rigid)	4	0.2	—	0.4	Nottage (1947)
Extruded polystyrene, smooth skin	—	—	—	0.35	Kumaran (1996)
aged per Can/ULC Standard S770-2003	1.4 to 3.6	0.18 to 0.20	—	—	Four manufacturers (2011)
aged 180 days	1.4 to 3.6	0.20			One manufacturer (2011)
European product	1.9	0.21			One manufacturer (2011)
aged 5 years at 75°F	2 to 2.2	0.21	—	—	One manufacturer (2011)
blown with low global warming potential (GWP) (<5) blowing agent		0.24 to 0.25	—	—	One manufacturer (2011)
Expanded polystyrene, molded beads	—	—	—	0.35	Kumaran (1996)
	1.0 to 1.5	0.24 to 0.26	—	—	Independent test reports (2008)
	1.8	0.23	—	—	Independent test reports (2008)
Mineral fiberboard, wet felted	10	0.26	—	0.2	Kumaran (1996)
Rock wool board	—	—	—	0.2	Kumaran (1996)
floors and walls	4.0 to 8.0	0.23 to 0.25	—	—	Five manufacturers (2011)
roofing	10. to 11.	0.27 to 0.29	—	0.2	Five manufacturers (2011)
Acoustical tile[e]	21 to 23	0.36 to 0.37	—	0.14 to 0.19	
Perlite board	9	0.36	—	—	One manufacturer (2010)
Polyisocyanurate	—	—	—	0.35	Kumaran (1996)
unfaced, aged per Can/ULC Standard S770-2003	1.6 to 2.3	0.16 to 0.17	—	—	Seven manufacturers (2011)
with foil facers, aged 180 days	—	0.15 to 0.16	—	—	Two manufacturers (2011)
Phenolic foam board with facers, aged	—	0.14 to 0.16	—	—	One manufacturer (2011)
Loose fill					
Cellulose fiber, loose fill	—	—	—	0.33	NIST (2000), Kumaran (1996)
attic application up to 4 in.	1.0 to 1.2	0.31 to 0.32	—	—	Four manufacturers (2011)
attic application > 4 in.	1.2 to 1.6	0.27 to 0.28	—	—	Four manufacturers (2011)
wall application, densely packed	3.5	0.27 – 0.28	—	—	One manufacturer (2011)
Perlite, expanded	2 to 4	0.27 to 0.31	—	0.26	(Manufacturer, pre-2001)
	4 to 7.5	0.31 to 0.36	—	—	(Manufacturer, pre-2001)
	7.5 to 11	0.36 to 0.42	—	—	(Manufacturer, pre-2001)
Glass fiber[d]					
attics, ~4 to 12 in.	0.4 to 0.5	0.36 to 0.38	—	—	Four manufacturers (2011)
attics, ~12 to 22 in.	0.5 to 0.6	0.34 to 0.36	—	—	Four manufacturers (2011)
closed attic or wall cavities	1.8 to 2.3	0.24 to 0.25	—	—	Four manufacturers (2011)
Rock and slag wool[d]					
attics, ~3.5 to 4.5 in.	1.5 to 1.6	0.34	—	—	Three manufacturers (2011)
attics, ~5 to 17 in.	1.5 to 1.8	0.32 to 0.33	—	—	Three manufacturers (2011)
closed attic or wall cavities	4.0	0.27 to 0.29	—	—	Three manufacturers (2011)
Vermiculite, exfoliated	7.0 to 8.2	0.47	—	0.32	Sabine et al. (1975)
	4.0 to 6.0	0.44	—	—	Manufacturer (pre-2001)
Spray applied					
Cellulose, sprayed into open wall cavities	1.6 to 2.6	0.27 to 0.28	—	—	Two manufacturers (2011)
Glass fiber, sprayed into open wall or attic cavities	1.0	0.27 to 0.29	—	—	Manufacturers' association (2011)
	1.8 to 2.3	0.23 to 0.26	—	—	Four manufacturers (2011)
Polyurethane foam	—	—	—	0.35	Kumaran (2002)
low density, open cell	0.45 to 0.65	0.26 to 0.29	—	—	Three manufacturers (2011)
medium density, closed cell, aged 180 days	1.9 to 3.2	0.14 to 0.20	—	—	Five manufacturers (2011)

Table 5-15 Building and Insulating Materials: Design Values[a] *(Continued)*

(Table 1, Chapter 26, 2013 ASHRAE Handbook—Fundamentals)

Description	Density, lb/ft^3	Conductivity[b] k, Btu·in/h·ft^2·°F	Resistance R, h·ft^2·°F/Btu	Specific Heat, Btu/lb·°F	Reference[l]
Building Board and Siding					
Board					
Asbestos/cement board	120	4	—	0.24	Nottage (1947)
Cement board	71	1.7	—	0.2	Kumaran (2002)
Fiber/cement board	88	1.7	—	0.2	Kumaran (2002)
	61	1.3	—	0.2	Kumaran (1996)
	26	0.5	—	0.45	Kumaran (1996)
	20	0.4	—	0.45	Kumaran (1996)
Gypsum or plaster board	40	1.1	—	0.21	Kumaran (2002)
Oriented strand board (OSB) 7/16 in.	41	—	0.62	0.45	Kumaran (2002)
1/2 in.	41	—	0.68	0.45	Kumaran (2002)
Plywood (douglas fir) 1/2 in.	29	—	0.79	0.45	Kumaran (2002)
5/8 in.	34	—	0.85	0.45	Kumaran (2002)
Plywood/wood panels 3/4 in.	28	—	1.08	0.45	Kumaran (2002)
Vegetable fiber board					
sheathing, regular density 1/2 in.	18	—	1.32	0.31	Lewis (1967)
intermediate density 1/2 in.	22	—	1.09	0.31	Lewis (1967)
nail-based sheathing 1/2 in.	25	—	1.06	0.31	
shingle backer 3/8 in.	18	—	0.94	0.3	
sound-deadening board 1/2 in.	15	—	1.35	0.3	
tile and lay-in panels, plain or acoustic	18	0.4	—	0.14	
laminated paperboard	30	0.5	—	0.33	Lewis (1967)
homogeneous board from repulped paper	30	0.5	—	0.28	
Hardboard					
medium density	50	0.73	—	0.31	Lewis (1967)
high density, service-tempered and service grades	55	0.82	—	0.32	Lewis (1967)
high density, standard-tempered grade	63	1.0	—	0.32	Lewis (1967)
Particleboard					
low density	37	0.71	—	0.31	Lewis (1967)
medium density	50	0.94	—	0.31	Lewis (1967)
high density	62	1.18	0.85	—	Lewis (1967)
underlayment 5/8 in.	44	0.73	0.82	0.29	Lewis (1967)
Waferboard	37	0.63	0.21	0.45	Kumaran (1996)
Shingles					
Asbestos/cement	120	—	0.21	—	
Wood, 16 in., 7 1/2 in. exposure	—	—	0.87	0.31	
Wood, double, 16 in., 12 in. exposure	—	—	1.19	0.28	
Wood, plus ins. backer board 5/16 in.	—	—	1.4	0.31	
Siding					
Asbestos/cement, lapped 1/4 in.	—	—	0.21	0.24	
Asphalt roll siding	—	—	0.15	0.35	
Asphalt insulating siding (1/2 in. bed)	—	—	0.21	0.24	
Hardboard siding 7/16 in.	—	—	0.15	0.35	
Wood, drop, 8 in. 1 in.	—	—	0.79	0.28	
Wood, bevel					
8 in., lapped 1/2 in.	—	—	0.81	0.28	
10 in., lapped 3/4 in.	—	—	1.05	0.28	
Wood, plywood, 3/8 in., lapped	—	—	0.59	0.29	
Aluminum, steel, or vinyl,[h, i] over sheathing				—	
hollow-backed	—	—	0.62	0.29[i]	
insulating-board-backed 3/8 in.	—	—	1.82	0.32	
foil-backed 3/8 in.	—	—	2.96	—	
Architectural (soda-lime float) glass	158	6.9	—	0.21	
Building Membrane					
Vapor-permeable felt	—	—	0.06	—	
Vapor: seal, 2 layers of mopped 15 lb felt	—	—	0.12	—	
Vapor: seal, plastic film	—	—	Negligible	—	
Finish Flooring Materials					
Carpet and rebounded urethane pad 3/4 in.	7	—	2.38	—	NIST (2000)
Carpet and rubber pad (one-piece) 3/8 in.	20	—	0.68	—	NIST (2000)
Pile carpet with rubber pad 3/8 to 1/2 in.	18	—	1.59	—	NIST (2000)
Linoleum/cork tile 1/4 in.	29	—	0.51	—	NIST (2000)
PVC/rubber floor covering	—	2.8	—	—	CIBSE (2006)
rubber tile 1.0 in.	119	—	0.34	—	NIST (2000)
terrazzo 1.0 in.	—	—	0.08	0.19	
Metals (See Chapter 33, Table 3)					

Table 5-15 Building and Insulating Materials: Design Values[a] *(Continued)*

(Table 1, Chapter 26, 2013 ASHRAE Handbook—Fundamentals)

Description	Density, lb/ft^3	Conductivity[b] k, Btu·in/h·ft^2·°F	Resistance R, h·ft^2·°F/Btu	Specific Heat, Btu/lb·°F	Reference[l]
Roofing					
Asbestos/cement shingles	120	—	0.21	0.24	
Asphalt (bitumen with inert fill)	100	2.98	—	—	CIBSE (2006)
	119	4.0	—	—	CIBSE (2006)
	144	7.97	—	—	CIBSE (2006)
Asphalt roll roofing	70	—	0.15	0.36	
Asphalt shingles	70	—	0.44	0.3	
Built-up roofing 3/8 in.	70	—	0.33	0.35	
Mastic asphalt (heavy, 20% grit)	59	1.32	—	—	CIBSE (2006)
Reed thatch	17	0.62	—	—	CIBSE (2006)
Roofing felt	141	8.32	—	—	CIBSE (2006)
Slate 1/2 in.	—	—	0.05	0.3	
Straw thatch	15	0.49	—	—	CIBSE (2006)
Wood shingles, plain and plastic-film-faced	—	—	0.94	0.31	
Plastering Materials					
Cement plaster, sand aggregate	116	5.0	—	0.2	
Sand aggregate 3/8 in.	—	—	0.08	0.2	
3/4 in.	—	—	0.15	0.2	
Gypsum plaster	70	2.63	—	—	CIBSE (2006)
	80	3.19	—	—	CIBSE (2006)
Lightweight aggregate 1/2 in.	45	—	0.32	—	
5/8 in.	45	—	0.39	—	
on metal lath 3/4 in.	—	—	0.47	—	
Perlite aggregate	45	1.5	—	0.32	
Sand aggregate	105	5.6	—	0.2	
on metal lath 3/4 in.	—	—	0.13	—	
Vermiculite aggregate	30	1.0	—	—	CIBSE (2006)
	40	1.39	—	—	CIBSE (2006)
	45	1.7	—	—	CIBSE (2006)
	50	1.8	—	—	CIBSE (2006)
	60	2.08	—	—	CIBSE (2006)
Perlite plaster	25	0.55	—	—	CIBSE (2006)
	38	1.32	—	—	CIBSE (2006)
Pulpboard or paper plaster	38	0.48	—	—	CIBSE (2006)
Sand/cement plaster, conditioned	98	4.4	—	—	CIBSE (2006)
Sand/cement/lime plaster, conditioned	90	3.33	—	—	CIBSE (2006)
Sand/gypsum (3:1) plaster, conditioned	97	4.5	—	—	CIBSE (2006)
Masonry Materials					
Masonry units					
Brick, fired clay	150	8.4 to 10.2	—	—	Valore (1988)
	140	7.4 to 9.0	—	—	Valore (1988)
	130	6.4 to 7.8	—	—	Valore (1988)
	120	5.6 to 6.8	—	0.19	Valore (1988)
	110	4.9 to 5.9	—	—	Valore (1988)
	100	4.2 to 5.1	—	—	Valore (1988)
	90	3.6 to 4.3	—	—	Valore (1988)
	80	3.0 to 3.7	—	—	Valore (1988)
	70	2.5 to 3.1	—	—	Valore (1988)
Clay tile, hollow					
1 cell deep 3 in.	—	—	0.80	0.21	Rowley and Algren (1937)
4 in.	—	—	1.11	—	Rowley and Algren (1937)
2 cells deep 6 in.	—	—	1.52	—	Rowley and Algren (1937)
8 in.	—	—	1.85	—	Rowley and Algren (1937)
10 in.	—	—	2.22	—	Rowley and Algren (1937)
3 cells deep 12 in.	—	—	2.50	—	Rowley and Algren (1937)
Lightweight brick	50	1.39	—	—	Kumaran (1996)
	48	1.51	—	—	Kumaran (1996)
Concrete blocks[f, g]					
Limestone aggregate					
8 in., 36 lb, 138 lb/ft^3 concrete, 2 cores	—	—	—	—	
with perlite-filled cores	—	—	2.1	—	Valore (1988)
12 in., 55 lb, 138 lb/ft^3 concrete, 2 cores	—		—	—	
with perlite-filled cores	—	—	3.7	—	Valore (1988)
Normal-weight aggregate (sand and gravel)					
8 in., 33 to 36 lb, 126 to 136 lb/ft^3 concrete, 2 or 3 cores	—	—	1.11 to 0.97	0.22	Van Geem (1985)
with perlite-filled cores	—	—	2.0	—	Van Geem (1985)
with vermiculite-filled cores	—	—	1.92 to 1.37	—	Valore (1988)

Table 5-15 Building and Insulating Materials: Design Values[a] *(Continued)*

(Table 1, Chapter 26, 2013 ASHRAE Handbook—Fundamentals)

Description	Density, lb/ft^3	Conductivity[b] k, Btu·in/h·ft^2·°F	Resistance R, h·ft^2·°F/Btu	Specific Heat, Btu/lb·°F	Reference[l]
12 in., 50 lb, 125 lb/ft^3 concrete, 2 cores	—	—	1.23	0.22	Valore (1988)
Medium-weight aggregate (combinations of normal and lightweight aggregate)					
8 in., 26 to 29 lb, 97 to 112 lb/ft^3 concrete, 2 or 3 cores	—	—	1.71 to 1.28	—	Van Geem (1985)
with perlite-filled cores	—	—	3.7 to 2.3	—	Van Geem (1985)
with vermiculite-filled cores	—	—	3.3	—	Van Geem (1985)
with molded-EPS-filled (beads) cores	—	—	3.2	—	Van Geem (1985)
with molded EPS inserts in cores	—	—	2.7	—	Van Geem (1985)
Lightweight aggregate (expanded shale, clay, slate or slag, pumice)					
6 in., 16 to 17 lb, 85 to 87 lb/ft^3 concrete, 2 or 3 cores	—	—	1.93 to 1.65	—	Van Geem (1985)
with perlite-filled cores	—	—	4.2	—	Van Geem (1985)
with vermiculite-filled cores	—	—	3.0	—	Van Geem (1985)
8 in., 19 to 22 lb, 72 to 86 lb/ft^3 concrete	—	—	3.2 to 1.90	0.21	Van Geem (1985)
with perlite-filled cores	—	—	6.8 to 4.4	—	Van Geem (1985)
with vermiculite-filled cores	—	—	5.3 to 3.9	—	Shu et al. (1979)
with molded-EPS-filled (beads) cores	—	—	4.8	—	Shu et al. (1979)
with UF foam-filled cores	—	—	4.5	—	Shu et al. (1979)
with molded EPS inserts in cores	—	—	3.5	—	Shu et al. (1979)
12 in., 32 to 36 lb, 80 to 90 lb/ft^3, concrete, 2 or 3 cores	—	—	2.6 to 2.3	—	Van Geem (1985)
with perlite-filled cores	—	—	9.2 to 6.3	—	Van Geem (1985)
with vermiculite-filled cores	—	—	5.8	—	Valore (1988)
Stone, lime, or sand	180	72	—	—	Valore (1988)
Quartzitic and sandstone	160	43	—	—	Valore (1988)
	140	24	—	—	Valore (1988)
	120	13	—	0.19	Valore (1988)
Calcitic, dolomitic, limestone, marble, and granite	180	30	—	—	Valore (1988)
	160	22	—	—	Valore (1988)
	140	16	—	—	Valore (1988)
	120	11	—	0.19	Valore (1988)
	100	8	—	—	Valore (1988)
Gypsum partition tile					
3 by 12 by 30 in., solid	—	—	1.26	0.19	Rowley and Algren (1937)
4 cells	—	—	1.35	—	Rowley and Algren (1937)
4 by 12 by 30 in., 3 cells	—	—	1.67	—	Rowley and Algren (1937)
Limestone	150	3.95	—	0.2	Kumaran (2002)
	163	6.45	—	0.2	Kumaran (2002)
Concretes[i]					
Sand and gravel or stone aggregate concretes	150	10.0 to 20.0	—	—	Valore (1988)
(concretes with >50% quartz or quartzite sand have	140	9.0 to 18.0	—	0.19 to 0.24	Valore (1988)
conductivities in higher end of range)	130	7.0 to 13.0	—	—	Valore (1988)
Lightweight aggregate or limestone concretes	120	6.4 to 9.1	—	—	Valore (1988)
expanded shale, clay, or slate; expanded slags; cinders;	100	4.7 to 6.2	—	0.2	Valore (1988)
pumice (with density up to 100 lb/ft^3); scoria (sanded	80	3.3 to 4.1	—	0.2	Valore (1988)
concretes have conductivities in higher end of range)	60	2.1 to 2.5	—	—	Valore (1988)
	40	1.3	—	—	Valore (1988)
Gypsum/fiber concrete (87.5% gypsum, 12.5% wood chips)	51	1.66	—	0.2	Rowley and Algren (1937)
Cement/lime, mortar, and stucco	120	9.7	—	—	Valore (1988)
	100	6.7	—	—	Valore (1988)
	80	4.5	—	—	Valore (1988)
Perlite, vermiculite, and polystyrene beads	50	1.8 to 1.9	—	—	Valore (1988)
	40	1.4 to 1.5	—	0.15 to 0.23	Valore (1988)
	30	1.1	—	—	Valore (1988)
	20	0.8	—	—	Valore (1988)
Foam concretes	120	5.4	—	—	Valore (1988)
	100	4.1	—	—	Valore (1988)
	80	3.0	—	—	Valore (1988)
	70	2.5	—	—	Valore (1988)
Foam concretes and cellular concretes	60	2.1	—	—	Valore (1988)
	40	1.4	—	—	Valore (1988)
	20	0.8	—	—	Valore (1988)
Aerated concrete (oven-dried)	27 to 50	1.4	—	0.2	Kumaran (1996)
Polystyrene concrete (oven-dried)	16 to 50	2.54	—	0.2	Kumaran (1996)
Polymer concrete	122	11.4	—	—	Kumaran (1996)
	138	7.14	—	—	Kumaran (1996)
Polymer cement	117	5.39	—	—	Kumaran (1996)
Slag concrete	60	1.5	—	—	Touloukian et al (1970)
	80	2.25	—	—	Touloukian et al. (1970)
	100	3	—	—	Touloukian et al. (1970)

Table 5-15 Building and Insulating Materials: Design Values[a] *(Continued)*

(Table 1, Chapter 26, 2013 ASHRAE Handbook—Fundamentals)

Description	Density, lb/ft^3	Conductivity[b] k, Btu·in/h·ft^2·°F	Resistance R, h·ft^2·°F/Btu	Specific Heat, Btu/lb·°F	Reference[l]
	125	8.53	—	—	Touloukian et al. (1970)
Woods (12% moisture content)[j]					
Hardwoods	—	—	—	0.39[k]	Wilkes (1979)
Oak	41 to 47	1.12 to 1.25	—	—	Cardenas and Bible (1987)
Birch	43 to 45	1.16 to 1.22	—	—	Cardenas and Bible (1987)
Maple	40 to 44	1.09 to 1.19	—	—	Cardenas and Bible (1987)
Ash	38 to 42	1.06 to 1.14	—	—	Cardenas and Bible (1987)
Softwoods	—	—	—	0.39[k]	Wilkes (1979)
Southern pine	36 to 41	1.00 to 1.12	—	—	Cardenas and Bible (1987)
Southern yellow pine	31	1.06 to 1.16	—	—	Kumaran (2002)
Eastern white pine	25	0.85 to 0.94	—	—	Kumaran (2002)
Douglas fir/larch	34 to 36	0.95 to 1.01	—	—	Cardenas and Bible (1987)
Southern cypress	31 to 32	0.90 to 0.92	—	—	Cardenas and Bible (1987)
Hem/fir, spruce/pine/fir	24 to 31	0.74 to 0.90	—	—	Cardenas and Bible (1987)
Spruce	25	0.74 to 0.85	—	—	Kumaran (2002)
Western red cedar	22	0.83 to 0.86	—	—	Kumaran (2002)
West coast woods, cedars	22 to 31	0.68 to 0.90	—	—	Cardenas and Bible (1987)
Eastern white cedar	23	0.82 to 0.89	—	—	Kumaran (2002)
California redwood	24 to 28	0.74 to 0.82	—	—	Cardenas and Bible (1987)
Pine (oven-dried)	23	0.64	—	0.45	Kumaran (1996)
Spruce (oven-dried)	25	0.69	—	0.45	Kumaran (1996)

Notes for Table 5-15

[a] Values are for mean temperature of 75°F (24°C). Representative values for dry materials are intended as design (not specification) values for materials in normal use. Thermal values of insulating materials may differ from design values depending on in-situ properties (e.g., density and moisture content, orientation, etc.) and manufacturing variability. For properties of specific product, use values supplied by manufacturer or unbiased tests.

[b] Symbol λ also used to represent thermal conductivity.

[c] Does not include paper backing and facing, if any. Where insulation forms boundary (reflective or otherwise) of airspace, see Tables 2 and 3 in Chapter 26 of 2013 *ASHRAE Handbook—Fundamentals* for insulating value of airspace with appropriate effective emittance and temperature conditions of space.

[d] Conductivity varies with fiber diameter (see Chapter 25). Batt, blanket, and loose-fill mineral fiber insulations are manufactured to achieve specified R-values, the most common of which are listed in the table. Because of differences in manufacturing processes and materials, the product thicknesses, densities, and thermal conductivities vary over considerable ranges for a specified R-value.

[e] Insulating values of acoustical tile vary, depending on density of board and on type, size, and depth of perforations.

[f] Values for fully grouted block may be approximated using values for concrete with similar unit density.

[g] Values for concrete block and concrete are at moisture contents representative of normal use.

[h] Values for metal or vinyl siding applied over flat surfaces vary widely, depending on ventilation of the airspace beneath the siding; whether airspace is reflective or nonreflective; and on thickness, type, and application of insulating backing-board used. Values are averages for use as design guides, and were obtained from several guarded hot box tests (ASTM Standard C1363) on hollow-backed types and types made using backing of wood fiber, foamed plastic, and glass fiber. Departures of ±50% or more from these values may occur.

[i] Vinyl specific heat = 0.25 Btu/lb·°F1.0 kJ/(kg·K)

[j] See Adams (1971), MacLean (1941), and Wilkes (1979). Conductivity values listed are for heat transfer across the grain. Thermal conductivity of wood varies linearly with density, and density ranges listed are those normally found for wood species given. If density of wood species is not known, use mean conductivity value. For extrapolation to other moisture contents, the following empirical equation developed by Wilkes (1979) may be used:

$$k = 0.1791 + \frac{(1.874 \times 10^{-2} + 5.753 \times 10^{-4} M)\rho}{1 + 0.01 M}$$

where ρ is density of moist wood in lb/ft^{3}kg/m^3, and M is moisture content in percent.

[k] From Wilkes (1979), an empirical equation for specific heat of moist wood at 75°F (24°C) is as follows:

$$c_p = \frac{(0.299 + 0.01M)}{(1 + 0.01M)} + \Delta c_p$$

where Δc_p accounts for heat of sorption and is denoted by

$$\Delta c_p = M(1.921 \times 10^{-3} - 3.168 \times 10^{-5} M)$$

where M is moisture content in percent by mass.

[l] Blank space in reference column indicates historical values from previous volumes of *ASHRAE Handbook*. Source of information could not be determined.

example, the listed U-factors assume the surface temperatures of surrounding bodies are equal to the ambient air temperature.

Most fenestration products consist of transparent multi-pane glazing units and opaque elements comprising the sash and frame (hereafter called frame). The glazing unit's heat transfer paths include a one-dimensional center-of-glass contribution and a two-dimensional edge contribution. The frame contribution is primarily two-dimensional.

Consequently, the total heat transfer can be determined by calculating the separate heat transfer contributions of the center glass, edge glass, and frame. (When present, glazing dividers, such as decorative grilles and muntins, also affect heat transfer, and their contribution must be considered.) The overall U-factor may be estimated by adding the area-weighted U-factors for each contribution.

Table 5-16 lists computed U-factors for a variety of generic fenestration products, *which should only be used as an estimating tool for the early phases of design*. The table is based on ASHRAE-sponsored research involving laboratory testing and computer simulation of various fenestration products. Consequently, computer simulations (with high/low validation by testing) are now accepted as a standard method for determining accurate, product-specific U-factors.

While these U-factors have been determined for winter conditions, they can also be used to estimate heat gain during peak cooling conditions, because conductive gain is usually a small portion of the total heat gain for fenestration in direct sunlight. Glazing designs and framing materials may be compared in choosing a product that needs a specific winter design U-factor.

Table 5-16 lists 48 types of glazing. The multiple glazing categories are appropriate for sealed glass units and the combinations of storm sash and other glazing units. Unless otherwise noted, all multiple-glazed units are filled with dry air.

Several frame types are listed (through not all for any one category), in order of improving thermal performance. The most conservative frame to assume is the aluminum frame **without thermal break** (although some products on the market have higher U-factors). The aluminum frame with thermal break has at least 3/8 in. (10 mm) thermal break between the inside and outside for all members including both the frame and the operable sash, if applicable. (Products are available with significantly wider thermal breaks, which reduce heat flow considerably.)

The **reinforced vinyl/aluminum clad wood** category represents vinyl-frame products, such as sliding glass doors or large windows. These units have extensive metal reinforcing in the frame and wood products with extensive metal, usually on the exterior surface of the frame. The metal, of course, degrades the thermal performance of the frame material.

The **wood/vinyl frame** represents improved thermal performance over reinforced vinyl/aluminum clad wood. **Insulated fiberglass/vinyl frames** do not have metal reinforcing and the frame cavities are filled with insulation.

Shading devices, such as venetian blinds, draperies, and roller shades, substantially reduce the U-factor for windows and/or glass doors if they fit tightly to the window jambs, head, and sill, and are made of nonporous material. As a rough approximation, tight-fitting shading devices may be considered to reduce the U-factor of vertical exterior single glazing by 25% and of vertical exterior double glazing and glass block by 15%. These adjustments should not be considered in choosing heating equipment, but may be used for calculating design cooling loads.

5.3.3 The Overall Thermal Transmittance

U_o is the combined thermal transmittance of the respective areas of gross exterior wall, roof or ceiling or both, and floor assemblies. The overall thermal transmittance of the building envelope assembly shall be calculated from

$$\begin{aligned} U_o &= \Sigma U_i A_i / A_o \\ &= (U_1 A_1 + U_2 A_2 + \ldots + U_n A_n)/A_o \end{aligned} \tag{5-13}$$

where

U_o = the area-weighted average thermal transmittance of the gross area of an envelope assembly; i.e., the exterior wall assembly including fenestration and doors, the roof and ceiling assembly, and the floor assembly, Btu/(h·ft^2·°F)

A_o = The gross area of the envelope assembly, ft^2

U_i = the thermal transmittance of each individual path of the envelope assembly, i.e., the opaque portion or the fenestration, Btu/(h·ft^2·°F)

U_i = $1/R_i$ (where R_i is the total resistance to heat flow of an individual path through an envelope assembly).

A_i = the area of each individual element of the envelope assembly, ft^2

5.4 Calculating Surface Temperatures

The temperature at any interface can be calculated, since the temperature drop through any component of the wall is proportional to its resistance. Thus, the temperature drop Δt through R_1 is

$$\Delta t_1 = \frac{R_1(t_i - t_0)}{R_T} \tag{5-14}$$

where t_1 and t_0 are the indoor and outdoor temperatures, respectively. Hence, the temperature at the interface between R_1 and R_2 is

$$t_{1-2} = t_i - \Delta t_1$$

If the resistances of materials in a wall are highly dependent on temperature, the mean temperature must be known to assign the correct value. In such cases, it is perhaps most convenient to use a trial-and-error procedure for the calculation of the total resistance R_T. First the mean operating temperature for each layer is estimated, and R-values for the particular materials are selected. The total resistance R_T is then calculated and the temperature at each interface is calculated.

The mean temperature of each component (arithmetic mean of its surface temperatures) can then be used to obtain second generation R-values. This procedure can then be repeated until the R-values have been correctly selected for the resulting mean temperatures. Generally, this can be done in two or three trial calculations.

Figure 5-17 illustrates the procedure for determining the temperatures throughout the structure. Tables 5-17, 5-18, and 5-19

	Item	R	R_T	U
①	Outdoor air film	0.17	0.17	
②	8 in.concrete block, light aggregate vermiculite filled core—specified	4.6	4.77	
③	3 1/2 in. mineral fiber insulation—specified	11.0	15.77	
④	1/2 in. mineral fiber insulation—specified	0.45	16.22	
⑤	Inside air film	0.68	16.9	0.059

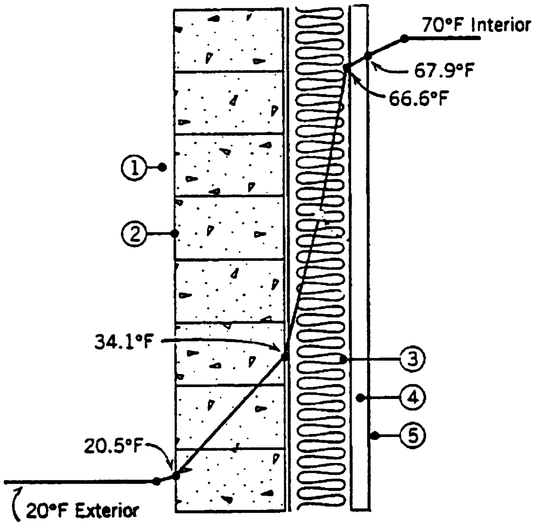


Fig. 5-17 Temperatures Throughout Wall

Table 5-16 U-Factors for Various Fenestration Products in Btu/h·ft^2·°F

Product Type		Glass Only		Vertical Installation									
				Operable (including sliding and swinging glass doors)					Fixed				
Frame Type		Center of Glass	Edge of Glass	Aluminum Without Thermal Break	Aluminum with Thermal Break	Reinforced Vinyl/ Aluminum Clad Wood	Wood/ Vinyl	Insulated Fiberglass/ Vinyl	Aluminum Without Thermal Break	Aluminum with Thermal Break	Reinforced Vinyl/ Aluminum Clad Wood	Wood/ Vinyl	Insulated Fiberglass/ Vinyl
ID	Glazing Type												
	Single Glazing												
1	1/8 in. glass	1.04	1.04	1.23	1.07	0.93	0.91	0.85	1.12	1.07	0.98	0.98	1.04
2	1/4 in. acrylic/polycarbonate	0.88	0.88	1.10	0.94	0.81	0.80	0.74	0.98	0.92	0.84	0.84	0.88
3	1/8 in. acrylic/polycarbonate	0.96	0.96	1.17	1.01	0.87	0.86	0.79	1.05	0.99	0.91	0.91	0.96
	Double Glazing												
4	1/4 in. air space	0.55	0.64	0.81	0.64	0.57	0.55	0.50	0.68	0.62	0.56	0.56	0.55
5	1/2 in. air space	0.48	0.59	0.76	0.58	0.52	0.50	0.45	0.62	0.56	0.50	0.50	0.48
6	1/4 in. argon space	0.51	0.61	0.78	0.61	0.54	0.52	0.47	0.65	0.59	0.53	0.52	0.51
7	1/2 in. argon space	0.45	0.57	0.73	0.56	0.50	0.48	0.43	0.60	0.53	0.48	0.47	0.45
	Double Glazing, *e* = 0.60 on surface 2 or 3												
8	1/4 in. air space	0.52	0.62	0.79	0.61	0.55	0.53	0.48	0.66	0.59	0.54	0.53	0.52
9	1/2 in. air space	0.44	0.56	0.72	0.55	0.49	0.48	0.43	0.59	0.53	0.47	0.47	0.44
10	1/4 in. argon space	0.47	0.58	0.75	0.57	0.51	0.50	0.45	0.61	0.55	0.49	0.49	0.47
11	1/2 in. argon space	0.41	0.54	0.70	0.53	0.47	0.45	0.41	0.56	0.50	0.44	0.44	0.41
	Double Glazing, *e* = 0.40 on surface 2 or 3												
12	1/4 in. air space	0.49	0.60	0.76	0.59	0.53	0.51	0.46	0.63	0.57	0.51	0.51	0.49
13	1/2 in. air space	0.40	0.54	0.69	0.52	0.47	0.45	0.40	0.55	0.49	0.44	0.43	0.40
14	1/4 in. argon space	0.43	0.56	0.72	0.54	0.49	0.47	0.42	0.58	0.52	0.46	0.46	0.43
15	1/2 in. argon space	0.36	0.51	0.66	0.49	0.44	0.42	0.37	0.52	0.46	0.40	0.40	0.36
	Double Glazing, *e* = 0.20 on surface 2 or 3												
16	1/4 in. air space	0.45	0.57	0.73	0.56	0.50	0.48	0.43	0.60	0.53	0.48	0.47	0.45
17	1/2 in. air space	0.35	0.50	0.65	0.48	0.43	0.41	0.37	0.51	0.45	0.39	0.39	0.35
18	1/4 in. argon space	0.38	0.52	0.68	0.51	0.45	0.43	0.39	0.54	0.47	0.42	0.42	0.38
19	1/2 in. argon space	0.30	0.46	0.61	0.45	0.39	0.38	0.33	0.47	0.41	0.35	0.35	0.30
	Double Glazing, *e* = 0.10 on surface 2 or 3												
20	1/4 in. air space	0.42	0.55	0.71	0.54	0.48	0.46	0.41	0.57	0.51	0.45	0.45	0.42
21	1/2 in. air space	0.32	0.48	0.63	0.46	0.41	0.39	0.34	0.49	0.42	0.37	0.37	0.32
22	1/4 in. argon space	0.35	0.50	0.65	0.48	0.43	0.41	0.37	0.51	0.45	0.39	0.39	0.35
23	1/2 in. argon space	0.27	0.44	0.59	0.42	0.37	0.36	0.31	0.44	0.38	0.33	0.32	0.27
	Double Glazing, *e* = 0.05 on surface 2 or 3												
24	1/4 in. air space	0.41	0.54	0.70	0.53	0.47	0.45	0.41	0.56	0.50	0.44	0.44	0.41
25	1/2 in. air space	0.30	0.46	0.61	0.45	0.39	0.38	0.33	0.47	0.41	0.35	0.35	0.30
26	1/4 in. argon space	0.33	0.48	0.64	0.47	0.42	0.40	0.35	0.49	0.43	0.38	0.37	0.33
27	1/2 in. argon space	0.25	0.42	0.57	0.41	0.36	0.34	0.30	0.43	0.36	0.31	0.31	0.25
	Triple Glazing												
28	1/4 in. air spaces	0.38	0.52	0.67	0.49	0.43	0.43	0.38	0.53	0.47	0.42	0.42	0.38
29	1/2 in. air spaces	0.31	0.47	0.61	0.44	0.38	0.38	0.34	0.47	0.41	0.36	0.36	0.31
30	1/4 in. argon spaces	0.34	0.49	0.63	0.46	0.41	0.40	0.36	0.50	0.44	0.38	0.38	0.34
31	1/2 in. argon spaces	0.29	0.45	0.59	0.42	0.37	0.36	0.32	0.45	0.40	0.34	0.34	0.29
	Triple Glazing, *e* = 0.20 on surface 2, 3, 4, or 5												
32	1/4 in. air spaces	0.33	0.48	0.62	0.45	0.40	0.39	0.35	0.49	0.43	0.37	0.37	0.33
33	1/2 in. air spaces	0.25	0.42	0.56	0.39	0.34	0.33	0.29	0.42	0.36	0.31	0.31	0.25
34	1/4 in. argon spaces	0.28	0.45	0.58	0.41	0.36	0.36	0.31	0.45	0.39	0.33	0.33	0.28
35	1/2 in. argon spaces	0.22	0.40	0.54	0.37	0.32	0.31	0.27	0.39	0.33	0.28	0.28	0.22
	Triple Glazing, *e* = 0.20 on surfaces 2 or 3 and 4 or 5												
36	1/4 in. air spaces	0.29	0.45	0.59	0.42	0.37	0.36	0.32	0.45	0.40	0.34	0.34	0.29
37	1/2 in. air spaces	0.20	0.39	0.52	0.35	0.31	0.30	0.26	0.38	0.32	0.26	0.26	0.20
38	1/4 in. argon spaces	0.23	0.41	0.54	0.37	0.33	0.32	0.28	0.40	0.34	0.29	0.29	0.23
39	1/2 in. argon spaces	0.17	0.36	0.49	0.33	0.28	0.28	0.24	0.35	0.29	0.24	0.24	0.17
	Triple Glazing, *e* = 0.10 on surfaces 2 or 3 and 4 or 5												
40	1/4 in. air spaces	0.27	0.44	0.58	0.40	0.36	0.35	0.31	0.44	0.38	0.32	0.32	0.27
41	1/2 in. air spaces	0.18	0.37	0.50	0.34	0.29	0.28	0.25	0.36	0.30	0.25	0.25	0.18
42	1/4 in. argon spaces	0.21	0.39	0.53	0.36	0.31	0.31	0.27	0.38	0.33	0.27	0.27	0.21
43	1/2 in. argon spaces	0.14	0.34	0.47	0.30	0.26	0.26	0.22	0.32	0.27	0.21	0.21	0.14
	Quadruple Glazing, *e* = 0.10 on surfaces 2 or 3 and 4 or 5												
44	1/4 in. air spaces	0.22	0.40	0.54	0.37	0.32	0.31	0.27	0.39	0.33	0.28	0.28	0.22
45	1/2 in. air spaces	0.15	0.35	0.48	0.31	0.27	0.26	0.23	0.33	0.27	0.22	0.22	0.15
46	1/4 in. argon spaces	0.17	0.36	0.49	0.33	0.28	0.28	0.24	0.35	0.29	0.24	0.24	0.17
47	1/2 in. argon spaces	0.12	0.32	0.45	0.29	0.25	0.24	0.20	0.31	0.25	0.20	0.20	0.12
48	1/4 in. krypton spaces	0.12	0.32	0.45	0.29	0.25	0.24	0.20	0.31	0.25	0.20	0.20	0.12

Notes:

1. All heat transmission coefficients in this table include film resistances and are based on winter conditions of 0°F outdoor air temperature and 70°F indoor air temperature, with 15 mph outdoor air velocity and zero solar flux. With the exception of single glazing, small changes in indoor and outdoor temperatures will not significantly affect overall U-factors. Coefficients are for vertical position except skylight values, which are for 20° from horizontal with heat flow up.
2. Glazing layer surfaces are numbered from outdoor to indoor. Double, triple, and quadruple refer to the number of glazing panels. All data are based on 1/8 in. glass, unless otherwise noted. Thermal conductivities are: 0.53 Btu/h·ft·°F for glass, and 0.11 Btu/h·ft·°F for acrylic and polycarbonate.
3. Standard spacers are metal. Edge-of-glass effects are assumed to extend over the 2 1/2 in. band around perimeter of each glazing unit.

Table 5-16 U-Factors for Various Fenestration Products in Btu/h·ft^2·°F *(Continued)*

Vertical Installation					Sloped Installation										
Garden Windows		Curtain Wall			Glass Only (Skylights)		Manufactured Skylight				Site-Assembled Sloped/Overhead Glazing				
Aluminum Without Thermal Break	Wood/ Vinyl	Aluminum Without Thermal Break	Aluminum with Thermal Break	Structural Glazing	Center of Glass	Edge of Glass	Aluminum Without Thermal Break	Aluminum with Thermal Break	Reinforced Vinyl/ Aluminum Clad Wood	Wood/ Vinyl	Aluminum Without Thermal Break	Aluminum with Thermal Break	Structural Glazing	ID
2.50	2.10	1.21	1.10	1.10	1.19	1.19	1.77	1.70	1.61	1.42	1.35	1.34	1.25	1
2.24	1.84	1.06	0.96	0.96	1.03	1.03	1.60	1.54	1.45	1.31	1.20	1.20	1.10	2
2.37	1.97	1.13	1.03	1.03	1.11	1.11	1.68	1.62	1.53	1.39	1.27	1.27	1.18	3
1.72	1.32	0.77	0.67	0.63	0.58	0.66	1.10	0.96	0.92	0.84	0.80	0.83	0.66	4
1.62	1.22	0.71	0.61	0.57	0.57	0.65	1.09	0.95	0.91	0.84	0.79	0.82	0.65	5
1.66	1.26	0.74	0.63	0.59	0.53	0.63	1.05	0.91	0.87	0.80	0.76	0.80	0.62	6
1.57	1.17	0.68	0.58	0.54	0.53	0.63	1.05	0.91	0.87	0.80	0.76	0.80	0.62	7
1.68	1.28	0.74	0.64	0.60	0.54	0.63	1.06	0.92	0.88	0.81	0.77	0.80	0.63	8
1.56	1.16	0.68	0.57	0.53	0.53	0.63	1.05	0.91	0.87	0.80	0.76	0.80	0.62	9
1.60	1.20	0.70	0.60	0.56	0.49	0.60	1.01	0.87	0.83	0.76	0.72	0.77	0.58	10
1.51	1.11	0.65	0.55	0.51	0.49	0.60	1.01	0.87	0.83	0.76	0.72	0.77	0.58	11
1.63	1.23	0.72	0.62	0.58	0.51	0.61	1.03	0.89	0.85	0.78	0.74	0.78	0.60	12
1.50	1.10	0.64	0.54	0.50	0.50	0.61	1.02	0.88	0.84	0.77	0.73	0.78	0.59	13
1.54	1.14	0.67	0.56	0.52	0.44	0.56	0.96	0.83	0.78	0.72	0.68	0.74	0.54	14
1.44	1.04	0.61	0.50	0.46	0.46	0.58	0.98	0.85	0.80	0.74	0.70	0.75	0.56	15
1.57	1.17	0.68	0.58	0.54	0.46	0.58	0.98	0.85	0.80	0.74	0.70	0.75	0.56	16
1.43	1.03	0.60	0.50	0.45	0.46	0.58	0.98	0.85	0.80	0.74	0.70	0.75	0.56	17
1.47	1.07	0.62	0.52	0.48	0.39	0.53	0.91	0.78	0.74	0.68	0.64	0.70	0.50	18
1.35	0.95	0.55	0.45	0.41	0.40	0.54	0.92	0.79	0.75	0.68	0.64	0.71	0.51	19
1.53	1.13	0.66	0.56	0.51	0.44	0.56	0.96	0.83	0.78	0.72	0.68	0.74	0.54	20
1.38	0.98	0.57	0.47	0.43	0.44	0.56	0.96	0.83	0.78	0.72	0.68	0.74	0.54	21
1.43	1.03	0.60	0.50	0.45	0.36	0.51	0.88	0.75	0.71	0.65	0.61	0.68	0.47	22
1.30	0.90	0.53	0.43	0.38	0.38	0.52	0.90	0.77	0.73	0.67	0.63	0.69	0.49	23
1.51	1.11	0.65	0.55	0.51	0.42	0.55	0.94	0.81	0.76	0.70	0.66	0.72	0.52	24
1.35	0.95	0.55	0.45	0.41	0.43	0.56	0.95	0.82	0.77	0.71	0.67	0.73	0.53	25
1.40	1.00	0.58	0.48	0.44	0.34	0.49	0.86	0.73	0.69	0.63	0.59	0.66	0.45	26
1.27	0.87	0.51	0.41	0.37	0.36	0.51	0.88	0.75	0.71	0.65	0.61	0.68	0.47	27
see note 7	see note 7	0.61	0.51	0.46	0.39	0.53	0.90	0.75	0.71	0.64	0.62	0.69	0.48	28
		0.55	0.45	0.40	0.36	0.51	0.87	0.72	0.68	0.61	0.60	0.67	0.45	29
		0.58	0.48	0.43	0.35	0.50	0.86	0.71	0.67	0.60	0.59	0.66	0.44	30
		0.53	0.43	0.38	0.33	0.48	0.84	0.69	0.65	0.59	0.57	0.65	0.42	31
see note 7	see note 7	0.57	0.47	0.42	0.34	0.49	0.85	0.70	0.66	0.59	0.58	0.65	0.43	32
		0.50	0.40	0.35	0.31	0.47	0.82	0.67	0.63	0.57	0.56	0.63	0.41	33
		0.53	0.43	0.37	0.28	0.45	0.80	0.64	0.60	0.54	0.53	0.61	0.38	34
		0.47	0.37	0.32	0.27	0.44	0.79	0.63	0.59	0.53	0.52	0.60	0.37	35
see note 7	see note 7	0.53	0.43	0.38	0.29	0.45	0.81	0.65	0.61	0.55	0.54	0.62	0.39	36
		0.46	0.36	0.30	0.27	0.44	0.79	0.63	0.59	0.53	0.52	0.60	0.37	37
		0.48	0.38	0.33	0.24	0.42	0.76	0.60	0.57	0.50	0.49	0.58	0.35	38
		0.43	0.33	0.28	0.22	0.40	0.74	0.58	0.55	0.49	0.48	0.57	0.33	39
see note 7	see note 7	0.52	0.42	0.37	0.27	0.44	0.79	0.63	0.59	0.53	0.52	0.60	0.37	40
		0.44	0.34	0.29	0.25	0.42	0.77	0.61	0.57	0.51	0.50	0.59	0.36	41
		0.46	0.36	0.31	0.21	0.39	0.73	0.57	0.54	0.48	0.47	0.56	0.32	42
		0.40	0.30	0.25	0.20	0.39	0.72	0.56	0.53	0.47	0.46	0.55	0.31	43
		0.47	0.37	0.32	0.22	0.40	0.74	0.58	0.55	0.49	0.48	0.57	0.33	44
see note 7	see note 7	0.41	0.31	0.26	0.19	0.38	0.71	0.55	0.52	0.46	0.45	0.54	0.30	45
		0.43	0.33	0.28	0.18	0.37	0.70	0.54	0.51	0.45	0.44	0.54	0.29	46
		0.39	0.29	0.23	0.16	0.35	0.68	0.52	0.49	0.43	0.42	0.52	0.28	47
		0.39	0.29	0.23	0.13	0.33	0.65	0.49	0.46	0.40	0.40	0.50	0.25	48

4. Product sizes are described in Figure 4, and frame U-factors are from Table 1.
5. Use U = 0.6 Btu/(h·ft^2·°F) for glass block with mortar but without reinforcing or framing.
6. Use of this table should be limited to that of an estimating tool for the early phases of design.
7. Values for triple- and quadruple-glazed garden windows are not listed, because these are not common products.
8. U-factors in this table were determined using NFRC 100-91. They have not been updated to the current rating methodology in NFRC 100-2004.

Table 5-17 Design U-Factors of Swinging Doors in Btu/h·ft^2·°F

(Table 6, Chapter 15, 2017 ASHRAE Handbook—Fundamentals)

Door Type (Rough Opening = 38 × 82 in.)	No Glazing	Single Glazing	Double Glazing with 1/2 in. Air Space	Double Glazing with $e = 0.10$, 1/2 in. Argon
Slab Doors				
Wood slab in wood frame[a]	0.46			
6% glazing (22 × 8 in. lite)	—	0.48	0.46	0.44
25% glazing (22 × 36 in. lite)	—	0.58	0.46	0.42
45% glazing (22 × 64 in. lite)	—	0.69	0.46	0.39
More than 50% glazing	Use Table 5-16 (operable)			
Insulated steel slab with wood edge in wood frame[b]	0.16			
6% glazing (22 × 8 in. lite)	—	0.21	0.19	0.18
25% glazing (22 × 36 in. lite)	—	0.39	0.26	0.23
45% glazing (22 × 64 in. lite)	—	0.58	0.35	0.26
More than 50% glazing	Use Table 5-16 (operable)			
Foam insulated steel slab with metal edge in steel frame[c]	0.37			
6% glazing (22 × 8 in. lite)	—	0.44	0.41	0.39
25% glazing (22 × 36 in. lite)	—	0.55	0.48	0.44
45% glazing (22 × 64 in. lite)	—	0.71	0.56	0.48
More than 50% glazing	Use Table 5-16 (operable)			
Cardboard honeycomb slab with metal edge in steel frame	0.61			
Stile-and-Rail Doors				
Sliding glass doors/French doors	Use Table 5-16 (operable)			
Site-Assembled Stile-and-Rail Doors				
Aluminum in aluminum frame	—	1.32	0.93	0.79
Aluminum in aluminum frame with thermal break	—	1.13	0.74	0.63

Notes:

[a] Thermally broken sill [add 0.03 Btu/h·ft^2·°F for non-thermally broken sill]

[b] Non-thermally broken sill

[c] Nominal U-factors are through center of insulated panel before consideration of thermal bridges around edges of door sections and because of frame.

Table 5-18 Design U-Factors for Revolving Doors in Btu/h·ft^2·°F

(Table 7, Chapter 15, 2017 ASHRAE Handbook—Fundamentals)

Type	Size (Width × Height)	U-Factor
3-wing	8 × 7 ft	0.79
	10 × 8 ft	0.80
4-wing	7 × 6.5 ft	0.63
	7 × 7.5 ft	0.64
Open*	82 × 84 in.	1.32

Notes:

*U-factor of Open door determined using NFRC *Technical Document* 100-91. It has not been updated to current rating methodology in NFRC *Technical Document* 100-2004.

contain representative values for design U-factors for various types of doors. These can be used as estimates when the manufacturer's data is not available.

Example 5-4 Determine the winter U-factor for a cavity wall consisting of face brick, 8 in. concrete block (three oval core, lightweight aggregate), a 3/4 in. airspace as the cavity, another layer of the same type of concrete block, 2 in. of rigid organic bonded glass fiber insulation, and 1/2 in. plasterboard.

Solution:

Component	**Resistance**	
Outdoor air	0.17	(Table 5-12)
Concrete block, 8 in.	2.00	(Table 5-11)
Airspace, 3/4 in. (estimated)	1.00	(Table 5-14)
Concrete block, 8 in.	2.00	(Table 5-15)
Glass fiber insulation, 2 in.	8.00	(Table 5-14, 5-15)
Plasterboard, 1/2 in.	0.45	(Table 5-15)
Inside air	0.68	(Table 5-12)
Total R =	14.74	

$$U = 1/R_T = 1/14.74 = 0.068 \text{ Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$

Table 5-19 Design U-Factors for Double-Skin Steel Emergency Exit Doors in Btu/h·ft^2·°F

(Table 8, Chapter 15, 2017 ASHRAE Handbook—Fundamentals)

Core Insulation		Rough Opening Size	
Thickness, in.	Type	3 ft × 6 ft 8 in.	6 ft × 6 ft 8 in.
1 3/8	Honeycomb kraft paper	0.57	0.52
	Mineral wool, steel ribs	0.44	0.36
	Polyurethane foam	0.34	0.28
1 3/4	Honeycomb kraft paper	0.57	0.54
	Mineral wool, steel ribs	0.41	0.33
	Polyurethane foam	0.31	0.26
1 3/8	Honeycomb kraft paper	0.60	0.55
	Mineral wool, steel ribs	0.47	0.39
	Polyurethane foam	0.37	0.31
1 3/4	Honeycomb kraft paper	0.60	0.57
	Mineral wool, steel ribs	0.44	0.37
	Polyurethane foam	0.34	0.30

*With thermal break

Example 5-5 Determine the overall coefficients (U-factors) for winter for (a) a solid wood door, (b) the flat ceiling/roof of an industrial building, which consists of a painted sheet metal exterior, 1/2 in. nail-base fiberboard sheathing, and wood rafters.

Solution:

(a) Use Table 5-17 to find the U-factor.

$$U = 0.46 \text{ Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$

(b) To calculate the U-factor of the ceiling, add the resistance values of each element.

Component	**Resistance**	
Inside air	0.61	(Table 5-12)
Sheathing	1.06	(Table 5-15)
Outdoor air	0.17	(Table 5-12)
Total R =	1.84	

$$U = 1/1.84 = 0.54 \text{ Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$

Example 5-6 Calculate the U-factor of the 2 by 4 stud wall shown in Figure 5-18. The studs are at 16 in. OC. There is 3.5 in. mineral fiber batt insulation (R-13) in the stud space. The inside finish is 0.5 in. gypsum wallboard; the outside is finished with rigid foam insulating sheathing (R-4) and 0.5 in. by 8 in. wood bevel lapped siding. The insulated cavity occupies approximately 75% of the transmission area; the studs, plates, and sills occupy 21%; and the headers occupy 4%.

Solution. Obtain the R-values of the various building elements from Tables 5-12 and 5-15. Assume the $R = 1.25$ per inch for the wood framing. Also, assume the headers are solid wood, in this case, and group them with the studs, plates, and sills.

Element	*R* (Insulated Cavity)	*R* (Studs, Plates, and Headers)
1. Outside surface, 15 mph wind	0.17	0.17
2. Wood bevel lapped siding	0.81	0.81
3. Rigid foam insulating sheathing	4.0	4.0
4. Mineral fiber batt insulation, 3.5 in.	13.0	—
5. Wood stud, nominal 2 × 4 (est.)	—	4.38
6. Gypsum wallboard, 0.5 in.	0.45	0.45
7. Inside surface, still air	0.68	0.68
	$R_1 = 19.11$	$R_2 = 10.49$

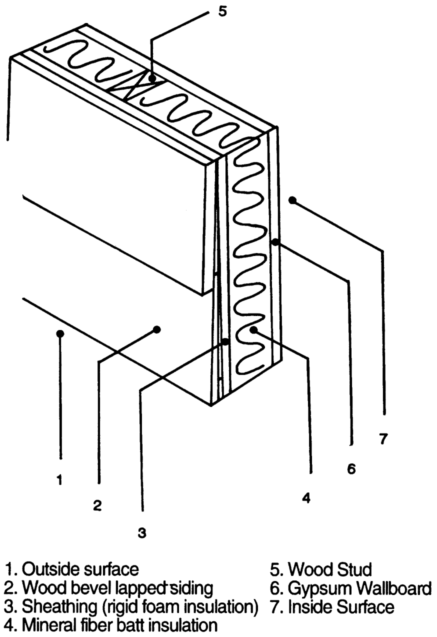


Fig. 5-18 Insulated Wood Frame Wall

Since the U-factor is the reciprocal of R-value, $U_1 = 0.052$ and $U_2 = 0.095$ Btu/h·ft^2·°F.

If the wood framing is accounted for using the parallel-path flow method, the U-factor of the wall is determined using the following equation

$$U_{av} = (0.75 \times 0.052) + (0.25 \times 0.095) = 0.063 \text{ Btu/h·ft2·°F}$$

If the wood framing (thermal bridging) is not included, Equation (3) from Chapter 22 may be used to calculate the U-factor of the wall as follows:

$$U_{av} = U_1 = \frac{1}{R_1} = 0.052 \text{ Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

5.5 Problems

5.1 With an 11.2 m/s wind blowing uniformly against one face of a building, what pressure differential would be used to calculate the air leakage into the building?

5.2 Can wind forces and Δt forces cancel each other when predicting infiltration? Reinforce? How can infiltration be prevented entirely?

5.3 A double door has a 1/8 in. crack on all sides except between the two doors, which has a 1/4 in. crack. What would be the leakage rate for the building of Problem 5.1?

[Ans: 453 cfm]

5.4 Determine the heat loss due to infiltration of 236 L/s of outdoor air at 9°C when the indoor air is 24°C.

5.5 Give an expression for (a) the sensible load due to infiltration, and (b) the latent load due to infiltration.

5.6 A building is 75 ft wide by 100 ft long and 10 ft high. The indoor conditions are 75°F and 24% rh, and the outside conditions are 0°F and saturated. The infiltration rate is estimated to be 0.75 ach. Calculate the sensible and latent heat loss. [Ans: 77,300 Btu/h; 17,700 Btu/h]

5.7 How can infiltration rates be reduced? Does a reduction in the infiltration rate always result in a reduction of the air-conditioning load? In air-conditioning equipment size? Explain.

5.8 A 3 ft by 3 ft ventilation opening is in a wall facing in the prevalent wind direction. There are adequate openings in the roof for the passage of exhaust air. Estimate the ventilation rate for a 25 mph wind.

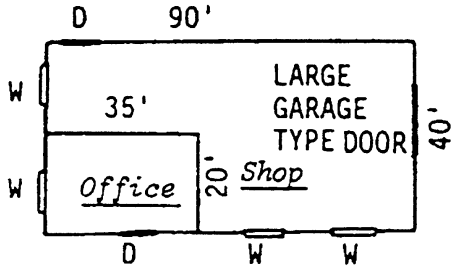
5.9 A building 20 ft by 40 ft by 9 ft has an anticipated infiltration rate of 0.75 air changes per hour. Indoor design conditions are 75°F, 30% rh minimum. Outdoor design temperature is 5°F.

(a) Determine sensible, latent, and total heat loads (Btu/h) due to infiltration.
(b) Specify the necessary humidifier size (lb/h).

[Ans: 8890 Btu/h, 1.8 lb/h]

5.10 A small factory with a 10 ft high ceiling is shown. There are 22 employees normally in the shop area and 4 employees in the office area. On a winter day when the outside temperature is 0°F, the office is maintained at 75°F, 25% rh, and the shop is kept at 68°F with no humidity control. Determine for each area

(a) infiltration, CFM
(b) minimum required outdoor air, CFM
(c) sensible heat loss due to infiltration, Btu/h
(d) latent heat loss due to infiltration, Btu/h



5.11 Specify an acceptable amount of outdoor air for ventilation of the following:

(a) 12 ft by 12 ft private office with 8 ft high walls
(b) department store with 20,000 ft^2 of floor area

5.12 A wall consists of 4 in. of face brick, 1/2 in. of cement mortar, 8 in. hollow clay tile, an airspace 1 5/8 in. wide, and

Table 5-20 Design U-Factors for Double-Skin Steel Garage and Aircraft Hangar Doors in Btu/h·ft^2·°F

(Table 9, Chapter 15, 2009 ASHRAE Handbook—Fundamentals)

Insulation		One-Piece Tilt-Up[a]		Sectional Tilt-Up[b]	Aircraft Hangar	
Thickness, in.	Type	8 × 7 ft	16 × 7 ft	9 × 7 ft	72 × 12 ft[c]	240 × 50 ft[d]
1 3/8	EPS, steel ribs[e]	0.36	0.33	0.34 – 0.39		
	XPS, steel ribs[f]	0.33	0.31	0.31 – 0.36		
2	EPS, steel ribs[e]	0.31	0.28	0.29 – 0.33		
	XPS, steel ribs[f]	0.29	0.26	0.27 – 0.31		
3	EPS, steel ribs[e]	0.26	0.23	0.25 – 0.28		
	XPS, steel ribs[f]	0.24	0.21	0.24 – 0.27		
4	EPS, steel ribs[e]	0.23	0.20	0.23 – 0.25		
	XPS, steel ribs[f]	0.21	0.19	0.21 – 0.24		
6	EPS, steel ribs[e]	0.20	0.16	0.20 – 0.21		
	XPS, steel ribs[f]	0.19	0.15	0.19 – 0.21		
4	XPS				0.25	0.16
	Mineral wool, steel ribs				0.25	0.16
	EPS				0.23	0.15
6	XPS				0.21	0.13
	Mineral wool, steel ribs				0.23	0.13
	EPS				0.20	0.12
—	Uninsulated[g]				1.10	1.23
	All products[f]	1.15[g]				

Notes:

[a] Values are for thermally broken or thermally unbroken doors.

[b] Lower values are for thermally broken doors; upper values are for doors with no thermal break.

[c] Typical size for a small private airplane (single- or twin-engine.)

[d] Typical hangar door for a midsize commercial jet airliner.

[e] EPS = extruded polystyrene; XPS = expanded polystyrene.

[f] U-factor determined using NFRC *Technical Document* 100-91. Not updated to current rating methodology in NFRC *Technical Document* 100-2004.

[g] U-factor determined for 10 × 10 ft sectional door, but is representative of similar products of different size.

wood lath and plaster totaling 3/4 in. Find the U-factors for both winter and summer.

5.13 The ceiling of a house is 3/4 in. (19 mm) acoustical tile on furring strips with highly reflective aluminum foil across the top of the ceiling joists. Determine the U-factor for cooling load calculations.

5.14 Calculate the winter U-factor for a wall consisting of 4 in. (100 mm) face brick, 4 in. (100 mm) common brick, and 1/2 in. (13 mm) of gypsum plaster (sand aggregate).

[Ans: 0.459 Btu/h· ft^2·°F [2.61 W/(m^2· K)]]

5.15 Find the overall coefficient of heat transmission U for a wall consisting of 4 in. of face brick, 1/2 in. of cement mortar, 8 in. of stone, and 3/4 in. of gypsum plaster. The outdoor air velocity is 15 mph and the inside air is still.

5.16 A wall has an overall coefficient $U = 1.31$ W/(m^2·K). What is the conductance of the wall when its outside surface is exposed to a wind velocity of 6.7 m/s and the inside air is still?

5.17 Compute the U-factor for a wall of frame construction consisting of 1/2 by 8 bevel siding, permeable felt building paper, 25/32 in. wood fiber sheathing, 2 by 4 studding on 16 in. centers, and 3/4 in. metal lath and sand plaster. Out-side wind velocity is 15 mph. [Ans: 0.206 Btu/ h·ft^2·°F]

5.18 For the wall of Problem 5.17, determine U if the space between the studs is filled with fiberglass blanket insulation. Neglect the effect of the studs.

5.19 Rework Problem 5.18 including the effect of the studs.

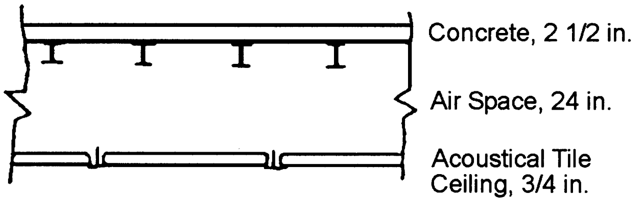
5.20 A concrete wall 250 mm thick is exposed to outdoor air at –15°C with a velocity of 6.7 m/s. Inside air temperature is 15.6°C. Determine the heat flow through 14.9 m^2 of this wall. [Ans: 1548 W]

5.21 Find the overall coefficient of heat transfer and the total thermal resistance for the following exterior wall exposed to a 25 mph wind: face brick veneer, 25/32 in. insulating board sheathing, 3 in. fiberglass insulation in stud space, and 1/4 in. walnut veneer plywood panels for the interior.

5.22 What is the thermal resistance of 12.1 cm (4 3/4 in.) thick precast concrete (stone aggregate, oven dried)?

5.23 A composite wall structure experiences a –10°F air temperature on the outside and a 75°F air temperature on the inside. The wall consists of a 4 in. thick outer facebrick, a 2 in. batt of fiberglass insulation, and a 3/8 in. sheet of gypsum board. Determine the U-factor and the heat flow rate per ft^2. Plot the steady-state temperature profile across the wall. [Ans: 0.118 Btu/h·ft^2·°F, 10.07 Btu/h·ft^2]

5.24 Find the overall heat transmission coefficient for a floor-ceiling sandwich (heat flow up) having the following construction. [Ans: 0.22 Btu/h· ft·°F]

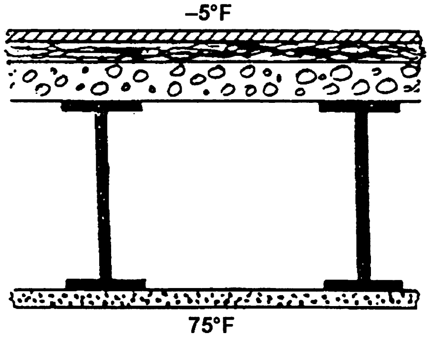


5.25 The exterior windows of a house are of double insulating glass with 1/4 in. airspace and have metal sashes. Determine the design U-factor for heating.

5.26 In designing a house, the total heat loss is calculated as 17.9 kW. The heat loss through the outside walls is 28% of this total when the overall coefficient for the outside walls is 1.4 W/(m^2·K). If 50 mm organic bonded fiberglass is added to the wall in the stud space, determine the new total heat loss for the house. [Ans: 14.6 kW]

5.27 The top floor ceiling of a building 30 ft by 36 ft is constructed of 2 in. by 4 in. joists on 18 in. centers. On the underside is metal lath with plaster, 3/4 in. thick. On top of the joists there are only scattered walking planks, but the space between the joists is filled with rock wool. The air temperature at the ceiling in the room is 78°F and the attic temperature is 25°F. Find the overall coefficient of heat transfer for the ceiling.

5.28 Determine the U-factor and the temperature at each point of change of material for the flat roof shown below. The roof has 3/8 in. built-up roofing, 1 1/2 in. roof insulation, 2 in. thick, 80 lb/ft^3 lightweight aggregate concrete on corrugated metal over steel joists, with a metal lath and 3/4 in. (sand) plaster ceiling. Omit correction for framing.



5.29 Calculate the heat loss through a roof of 100 ft^2 area where the inside air temperature is to be 70°F, outdoor air 10°F, and the composition from outside to inside: 3/8 in. built-up roofing, 1 in. cellular glass insulation, 4 in. concrete slab, and 3/4 in. acoustical tile. [Ans: 1030 Btu/h (0.30 kW)]

5.30 Calculate the heat loss through 100 ft^2 (9.29 m^2) of 1/4 in. (6.5 mm) plate glass with inside and outdoor air temperatures of 70°F and 10°F (21.1°C and –12.2°C), respectively.

5.31 A building has single-glass windows and an indoor temperature of 75°F. The outdoor air temperature is 40°F. With a 15 mph outside wind, still air inside, and after sundown, what can the maximum relative humidity of the inside air be without condensation forming on the glass?

5.32 Repeat Problem 5.31 for a double-glass window with a 1/2 in. airspace. [Ans: 67% rh]

5.33 A wall is constructed of 4 in. face brick, pressed fiber board sheathing ($k = 0.44$), 3 1/2 in. airspace, and 1/2 in. lightweight gypsum plaster on 1/2 in. plasterboard. When the inside air temperature is 70°F and the outside temperature is –15°F, how thick must the sheathing be to prevent water pipes in the stud space from freezing?

5.34 The roof of a rapid transit car is constructed of 3/8 in. plywood ($C = 2.12$), a vapor seal having negligible thermal resistance, expanded polystyrene insulation ($k = 0.24$), 3/4 in. airspace, 1/16 in. steel with welded joints and aluminum paint. If the car is traveling at 60 mph (film coefficient is 20.0 Btu/h·ft^2·°F) when the ambient temperature is –20°F, what thickness of insulation is necessary to prevent condensation when the inside conditions in the car are 72°F dry bulb and 55% rh?

5.35 A roof is constructed of 2 in. wood decking, insulation on top of deck, and 3/8 in. built-up roofing. It has no ceiling. Assuming that the insulation forms a perfect vapor barrier, determine the required resistance of the insulation to prevent condensation from occurring at the deck-insulation interface when indoor conditions are 70°F and 40% rh, and the outside temperature is 20°F. [Ans: 2.48 ft^2·°F· h/Btu]

5.36 Determine the summer U-factor for each of the following:

(a) Building wall consisting of face brick veneer, 3/4 in. plywood sheathing, 2 by 6 studs on 24 in. centers, no insulation, and 5/32 in. plywood paneling
(b) Ceiling/roof where the ceiling is composed of 1/2 in. plasterboard nailed to 2 by 6 joists on 16 in. centers and the roof consists of asphalt shingles on 3/4 in. plywood on 2 by 4 rafters on 16 in. centers. The roof area is 2717 ft^2 while the ceiling area is 1980 ft^2.
(c) Sliding patio door with insulating glass (double) having a 0.50 in. airspace in a metal frame
(d) A 2 in. solid wood door

5.37 A prefabricated commercial building has exterior walls constructed of 2 in. expanded polyurethane bonded between 1/8 in. aluminum sheet and 1/4 in. veneer plywood. Design conditions include 105°F outdoor air temperature, 72°F indoor air, and 7.5 mph wind. Determine

(a) overall thermal resistance
(b) value of U
(c) heat gain per ft^2

[Ans: (a) 13.74 ft^2·°F·h/Btu; (b) 0.073 Btu/h·ft^2·°F; (c) 2.40 Btu/h·ft^2]

5.38 An outside wall consists of 4 in. face brick, 25/32 in. insulating board sheathing 2 in. mineral fiber batt between the 2 by 4 studs, and 1/2 in. plasterboard. Determine the winter U-factor.

5.39 Solve the following:

(a) Compute the winter U-factor for the wall of Problem 5.38 if the wind velocity is 30 mph.
(b) Compute the summer U-factor for the wall of Problem 5.38.
(c) If full wall insulation is used, compute the summer U-factor for the wall of Problem 5.38.

5.40 An exterior wall contains a 3 ft by 7 ft solid wood door, 1 3/8 in. thick, and a 6 ft by 7 ft sliding patio door with double glass having a 1/2 in. airspace and metal frame. Determine the summer U-factor for each door.

[Ans: 0.46 Btu/h·ft^2·°F; 0.76 Btu/h·ft^2·°F]

5.41 If the doors of Problem 5.40 are between the residence and a completely enclosed swimming pool area, determine the U-factor for each door.

5.42 Determine the winter U-factor in W/(m^2·K) for the wall of a building that has the following construction: face brick, 4 in.; airspace, 3/4 in.; concrete, 9 in.; cellular glass board insulation, 1 in.; plywood paneling, 1/4 in.

5.43 Determine the summer U-factor for the following building components

(a) Wall: wood drop siding, 1 by 8 in.; 1/2 in. nail-base insulating board sheathing; 2 by 4 studs (16 in. oc) with full wall fiberglass insulation; 1/4 in. paneling
(b) Door: wood, 1 1/2 in. thick, with 25% single-pane glass

5.44 Determine the combined ceiling and roof winter U-factor for the following construction: The ceiling consists of 3/8 in. gypsum board on 2 by 6 in. ceiling joists. Six inches of fiberglass (mineral/glass wool) insulation fills the space between the joists. The pitched roof has asphalt shingles on 25/32 in. solid wood sheathing with no insulation between the rafters. The ratio of roof area to ceiling area is 1.3. The attic is unvented in winter.

5.45 The west wall of a residence is 70 ft long by 8 ft high. The wall contains four 3 ft by 5 ft wood sash 80% glass single-pane windows each with a storm window; one double-glazed (1/2 in. airspace) picture window, 5 1/2 ft by 10 ft; and one 1 3/4 in. thick solid wood door, 3 ft by 7 ft. The wall itself has the construction of Problem 5.21. Specify the U-factor and corresponding area for each of the various parts of the wall with normal winter air velocities.

5.46 A wall is 20 m by 3 m, which includes 14% double glazed glass windows with a 6 mm airspace. The wall proper consists of one layer of face brick backed by 250 mm of concrete with 12 mm of gypsum plaster on the inside. For indoor and outdoor design temperatures of 22°C and –15°C, respectively, determine the heat loss through this wall, kW.

5.6 Bibliography

ASHRAE Standard 55. Thermal Environmental Conditions for Human Occupancy.

ASHRAE Standard 62.1. Ventilation for Acceptable Indoor Air Quality.

ASHRAE Standard 62.2. Acceptable Indoor Air Quality in Low-Rise Residential Buildings.

ASHRAE Standard 90.1. Energy Efficient Design of New Buildings Except New Low-Rise Residential Buildings.

ASHRAE. 2014. *Load Calculation Applications Manual, second ed.* Jeffrey D. Spitler.

Persily, A., J. Gordfain, and G. Brunner. 2015. Ventilation design and performance in U.S. office buildings. *ASHRAE Journal*, April.

SI Figures and Tables

Table 5-1 SI Change Rates as a Function of Airtightness

	Outdoor Design Temperature, °C									
Class	**10**	**4**	**–1**	**–7**	**–12**	**–18**	**–23**	**–29**	**–34**	**–40**
Tight	0.41	0.43	0.45	0.47	0.49	0.51	0.53	0.55	0.57	0.59
Medium	0.69	0.73	0.77	0.81	0.85	0.89	0.93	0.97	1.00	1.05
Loose	1.11	1.15	1.20	1.23	1.27	1.30	1.35	1.40	1.43	1.47

Note: Values are for 6.7 m/s (24 km/h) wind and indoor temperature of 20 °C.

Table 5-5 SI Basic Model Stack Coefficient C_s

(Table 4, Chapter 16, 2017 ASHRAE Handbook—Fundamentals, SI Version)

	House Height (Stories)		
	One	**Two**	**Three**
Stack coefficient	0.000 145	0.000 290	0.000 435

Table 5-6 SI Basic Model Wind Coefficient C_w

(Table 6, Chapter 16, 2017 ASHRAE Handbook—Fundamentals, SI Version)

Shelter Class	House Height (Stories)		
	One	**Two**	**Three**
1	0.000 319	0.000 420	0.000 494
2	0.000 246	0.000 325	0.000 382
3	0.000 174	0.000 231	0.000 271
4	0.000 104	0.000 137	0.000 161
5	0.000 032	0.000 042	0.000 049

Table 5-12 SI Surface Film Coefficients/Resistances

(Table 10, Chapter 26, 2017 ASHRAE Handbook—Fundamentals, SI Version)

		Surface Emittance, ε					
		Nonreflective		Reflective			
		ε = 0.90		ε = 0.20		ε = 0.05	
Position of Surface	**Direction of Heat Flow**	h_i	R_i	h_i	R_i	h_i	R_i
Indoor							
Horizontal	Upward	9.26	0.11	5.17	0.19	4.32	0.23
Sloping at 45°	Upward	9.09	0.11	5.00	0.20	4.15	0.24
Vertical	Horizontal	8.29	0.12	4.20	0.24	3.35	0.30
Sloping at 45°	Downward	7.50	0.13	3.41	0.29	2.56	0.39
Horizontal	Downward	6.13	0.16	2.10	0.48	1.25	0.80
Outdoor (any position)		h_o	R_o				
Wind (for winter) at 6.7 m/s	Any	34.0	0.030	—	—	—	—
Wind (for summer) at 3.4 m/s	Any	22.7	0.044	—	—	—	—

Notes:

1. Surface conductance h_i and h_o measured in W/(m^2·K); resistance R_i and R_o in (m^2·K)/W.
2. No surface has both an air space resistance value and a surface resistance value.
3. Conductances are for surfaces of the stated emittance facing virtual blackbody surroundings at same temperature as ambient air. Values based on surface/air temperature difference of 5.6 K and surface temperatures of 21°C.
4. See Chapter 4 for more detailed information.
5. Condensate can have significant effect on surface emittance. Also, oxidation, corrosion, and accumulation of dust and dirt can dramatically increase surface emittance. Emittance values of 0.05 should only be used where highly reflective surface can be maintained over the service life of the assembly.

Table 5-13 SI Emissivity of Various Surfaces and Effective Emittances of Facing Air Spaces[a]

(Table 2, Chapter 26, 2017 ASHRAE Handbook—Fundamentals, SI Version)

		Effective Emittance ε_{eff} of Air Space	
Surface	**Average Emissivity ε**	**One Surface's Emittance ε; Other, 0.9**	**Both Surfaces' Emittance ε**
Aluminum foil, bright	0.05	0.05	0.03
Aluminum foil, with condensate just visible (>0.7 g/ft^2)(>0.5 g/m^2)	0.30[b]	0.29	—
Aluminum foil, with condensate clearly visible (>2.9 g/ft^2)(>2.0 g/m^2)	0.70[b]	0.65	—
Aluminum sheet	0.12	0.12	0.06
Aluminum-coated paper, polished	0.20	0.20	0.11
Brass, nonoxidized	0.04	0.038	0.02
Copper, black oxidized	0.74	0.41	0.59
Copper, polished	0.04	0.038	0.02
Iron and steel, polished	0.2	0.16	0.11
Iron and steel, oxidized	0.58	0.35	0.41
Lead, oxidized	0.27	0.21	0.16
Nickel, nonoxidized	0.06	0.056	0.03
Silver, polished	0.03	0.029	0.015
Steel, galvanized, bright	0.25	0.24	0.15
Tin, nonoxidized	0.05	0.047	0.026
Aluminum paint	0.50	0.47	0.35
Building materials: wood, paper, masonry, nonmetallic paints	0.90	0.82	0.82
Regular glass	0.84	0.77	0.72

[a] Values apply in 4 to 40 μm range of electromagnetic spectrum. Also, oxidation, corrosion, and accumulation of dust and dirt can dramatically increase surface emittance. Emittance values of 0.05 should only be used where the highly reflective surface can be maintained over the service life of the assembly. Except as noted, data from VDI (1999).

[b] Values based on data in Bassett and Trethowen (1984).

Table 5-14 SI Effective Thermal Resistances of Plane Air Spaces,^{a,b,c} (m^2·K)/W

(Table 3, Chapter 26, 2017 ASHRAE Handbook—Fundamentals, SI Version)

		Air Space		Effective Emittance ε_{eff}^{d,e}									
Position of Air Space	**Direction of Heat Flow**	**Mean Temp.^{d}, °C**	**Temp. Diff.,^{d} K**	**13 mm Air Space^{c}**					**20 mm Air Space^{c}**				
				0.03	**0.05**	**0.2**	**0.5**	**0.82**	**0.03**	**0.05**	**0.2**	**0.5**	**0.82**
Horiz.	Up ↑	32.2	5.6	0.37	0.36	0.27	0.17	0.13	0.41	0.39	0.28	0.18	0.13
		10.0	16.7	0.29	0.28	0.23	0.17	0.13	0.30	0.29	0.24	0.17	0.14
		10.0	5.6	0.37	0.36	0.28	0.20	0.15	0.40	0.39	0.30	0.20	0.15
		–17.8	11.1	0.30	0.30	0.26	0.20	0.16	0.32	0.32	0.27	0.20	0.16
		–17.8	5.6	0.37	0.36	0.30	0.22	0.18	0.39	0.38	0.31	0.23	0.18
		–45.6	11.1	0.30	0.29	0.26	0.22	0.18	0.31	0.31	0.27	0.22	0.19
		–45.6	5.6	0.36	0.35	0.31	0.25	0.20	0.38	0.37	0.32	0.26	0.21
45° Slope	Up ↗	32.2	5.6	0.43	0.41	0.29	0.19	0.13	0.52	0.49	0.33	0.20	0.14
		10.0	16.7	0.36	0.35	0.27	0.19	0.15	0.35	0.34	0.27	0.19	0.14
		10.0	5.6	0.45	0.43	0.32	0.21	0.16	0.51	0.48	0.35	0.23	0.17
		–17.8	11.1	0.39	0.38	0.31	0.23	0.18	0.37	0.36	0.30	0.23	0.18
		–17.8	5.6	0.46	0.45	0.36	0.25	0.19	0.48	0.46	0.37	0.26	0.20
		–45.6	11.1	0.37	0.36	0.31	0.25	0.21	0.36	0.35	0.31	0.25	0.20
		–45.6	5.6	0.46	0.45	0.38	0.29	0.23	0.45	0.43	0.37	0.29	0.23
Vertical	Horiz. →	32.2	5.6	0.43	0.41	0.29	0.19	0.14	0.62	0.57	0.37	0.21	0.15
		10.0	16.7	0.45	0.43	0.32	0.22	0.16	0.51	0.49	0.35	0.23	0.17
		10.0	5.6	0.47	0.45	0.33	0.22	0.16	0.65	0.61	0.41	0.25	0.18
		–17.8	11.1	0.50	0.48	0.38	0.26	0.20	0.55	0.53	0.41	0.28	0.21
		–17.8	5.6	0.52	0.50	0.39	0.27	0.20	0.66	0.63	0.46	0.30	0.22
		–45.6	11.1	0.51	0.50	0.41	0.31	0.24	0.51	0.50	0.42	0.31	0.24
		–45.6	5.6	0.56	0.55	0.45	0.33	0.26	0.65	0.63	0.51	0.36	0.27
45° Slope	Down ↘	32.2	5.6	0.44	0.41	0.29	0.19	0.14	0.62	0.58	0.37	0.21	0.15
		10.0	16.7	0.46	0.44	0.33	0.22	0.16	0.60	0.57	0.39	0.24	0.17
		10.0	5.6	0.47	0.45	0.33	0.22	0.16	0.67	0.63	0.42	0.26	0.18
		–17.8	11.1	0.51	0.49	0.39	0.27	0.20	0.66	0.63	0.46	0.30	0.22
		–17.8	5.6	0.52	0.50	0.39	0.27	0.20	0.73	0.69	0.49	0.32	0.23
		–45.6	11.1	0.56	0.54	0.44	0.33	0.25	0.67	0.64	0.51	0.36	0.28
		–45.6	5.6	0.57	0.56	0.45	0.33	0.26	0.77	0.74	0.57	0.39	0.29
Horiz.	Down ↓	32.2	5.6	0.44	0.41	0.29	0.19	0.14	0.62	0.58	0.37	0.21	0.15
		10.0	16.7	0.47	0.45	0.33	0.22	0.16	0.66	0.62	0.42	0.25	0.18
		10.0	5.6	0.47	0.45	0.33	0.22	0.16	0.68	0.63	0.42	0.26	0.18
		–17.8	11.1	0.52	0.50	0.39	0.27	0.20	0.74	0.70	0.50	0.32	0.23
		–17.8	5.6	0.52	0.50	0.39	0.27	0.20	0.75	0.71	0.51	0.32	0.23
		–45.6	11.1	0.57	0.55	0.45	0.33	0.26	0.81	0.78	0.59	0.40	0.30
		–45.6	5.6	0.58	0.56	0.46	0.33	0.26	0.83	0.79	0.60	0.40	0.30
		Air Space		**40 mm Air Space^{c}**					**90 mm Air Space^{c}**				
Horiz.	Up ↑	32.2	5.6	0.45	0.42	0.30	0.19	0.14	0.50	0.47	0.32	0.20	0.14
		10.0	16.7	0.33	0.32	0.26	0.18	0.14	0.27	0.35	0.28	0.19	0.15
		10.0	5.6	0.44	0.42	0.32	0.21	0.16	0.49	0.47	0.34	0.23	0.16
		–17.8	11.1	0.35	0.34	0.29	0.22	0.17	0.40	0.38	0.32	0.23	0.18
		–17.8	5.6	0.43	0.41	0.33	0.24	0.19	0.48	0.46	0.36	0.26	0.20
		–45.6	11.1	0.34	0.34	0.30	0.24	0.20	0.39	0.38	0.33	0.26	0.21
		–45.6	5.6	0.42	0.41	0.35	0.27	0.22	0.47	0.45	0.38	0.29	0.23
45° Slope	Up ↗	32.2	5.6	0.51	0.48	0.33	0.20	0.14	0.56	0.52	0.35	0.21	0.14
		10.0	16.7	0.38	0.36	0.28	0.20	0.15	0.40	0.38	0.29	0.20	0.15
		10.0	5.6	0.51	0.48	0.35	0.23	0.17	0.55	0.52	0.37	0.24	0.17
		–17.8	11.1	0.40	0.39	0.32	0.24	0.18	0.43	0.41	0.33	0.24	0.19
		–17.8	5.6	0.49	0.47	0.37	0.26	0.20	0.52	0.51	0.39	0.27	0.20
		–45.6	11.1	0.39	0.38	0.33	0.26	0.21	0.41	0.40	0.35	0.27	0.22
		–45.6	5.6	0.48	0.46	0.39	0.30	0.24	0.51	0.49	0.41	0.31	0.24
Vertical	Horiz. →	32.2	5.6	0.70	0.64	0.40	0.22	0.15	0.65	0.60	0.38	0.22	0.15
		10.0	16.7	0.45	0.43	0.32	0.22	0.16	0.47	0.45	0.33	0.22	0.16
		10.0	5.6	0.67	0.62	0.42	0.26	0.18	0.64	0.60	0.41	0.25	0.18
		–17.8	11.1	0.49	0.47	0.37	0.26	0.20	0.51	0.49	0.38	0.27	0.20
		–17.8	5.6	0.62	0.59	0.44	0.29	0.22	0.61	0.59	0.44	0.29	0.22
		–45.6	11.1	0.46	0.45	0.38	0.29	0.23	0.50	0.48	0.40	0.30	0.24
		–45.6	5.6	0.58	0.56	0.46	0.34	0.26	0.60	0.58	0.47	0.34	0.26

Table 5-14 SI Effective Thermal Resistances of Plane Air Spaces,[a,b,c] (m^2·K)/W *(Continued)*

(Table 3, Chapter 26, 2017 ASHRAE Handbook—Fundamentals, SI Version)

Air Space				Effective Emittance ε_{eff}[d,e]									
Position of Air Space	Direction of Heat Flow	Mean Temp.[d], °C	Temp. Diff.,[d] K	40 mm Air Space[c]					90 mm Air Space[c]				
				0.03	0.05	0.2	0.5	0.82	0.03	0.05	0.2	0.5	0.82
45° Slope	Down ↘	32.2	5.6	0.89	0.80	0.45	0.24	0.16	0.85	0.76	0.44	0.24	0.16
		10.0	16.7	0.63	0.59	0.41	0.25	0.18	0.62	0.58	0.40	0.25	0.18
		10.0	5.6	0.90	0.82	0.50	0.28	0.19	0.83	0.77	0.48	0.28	0.19
		−17.8	11.1	0.68	0.64	0.47	0.31	0.22	0.67	0.64	0.47	0.31	0.22
		−17.8	5.6	0.87	0.81	0.56	0.34	0.24	0.81	0.76	0.53	0.33	0.24
		–45.6	11.1	0.64	0.62	0.49	0.35	0.27	0.66	0.64	0.51	0.36	0.28
		–45.6	5.6	0.82	0.79	0.60	0.40	0.30	0.79	0.76	0.58	0.40	0.30
Horiz.	Down ↓	32.2	5.6	1.07	0.94	0.49	0.25	0.17	1.77	1.44	0.60	0.28	0.18
		10.0	16.7	1.10	0.99	0.56	0.30	0.20	1.69	1.44	0.68	0.33	0.21
		10.0	5.6	1.16	1.04	0.58	0.30	0.20	1.96	1.63	0.72	0.34	0.22
		−17.8	11.1	1.24	1.13	0.69	0.39	0.26	1.92	1.68	0.86	0.43	0.29
		−17.8	5.6	1.29	1.17	0.70	0.39	0.27	2.11	1.82	0.89	0.44	0.29
		–45.6	11.1	1.36	1.27	0.84	0.50	0.35	2.05	1.85	1.06	0.57	0.38
		–45.6	5.6	1.42	1.32	0.86	0.51	0.35	2.28	2.03	1.12	0.59	0.39
		Air Space		143 mm Air Space[c]									
Horiz.	Up ↑	32.2	5.6	0.53	0.50	0.33	0.20	0.14					
		10.0	16.7	0.39	0.38	0.29	0.20	0.15					
		10.0	5.6	0.52	0.50	0.36	0.23	0.17					
		17.8	11.1	0.42	0.41	0.33	0.24	0.19					
		17.8	5.6	0.51	0.49	0.38	0.27	0.20					
		45.6	11.1	0.41	0.40	0.34	0.27	0.22					
		45.6	5.6	0.49	0.48	0.40	0.30	0.24					
45° Slope	Up ↗	32.2	5.6	0.57	0.54	0.35	0.21	0.15					
		10.0	16.7	0.39	0.37	0.29	0.20	0.15					
		10.0	5.6	0.56	053	0.37	0.24	0.17					
		17.8	11.1	0.41	0.40	0.33	0.24	0.18					
		17.8	5.6	0.53	0.51	0.39	0.27	0.20					
		45.6	11.1	0.38	0.37	0.32	0.26	0.21					
		45.6	11.1	0.49	0.48	0.40	0.30	0.24					
Vertical	Horiz. →	32.2	5.6	0.66	0.61	0.38	0.22	0.15					
		10.0	16.7	0.50	0.47	0.35	0.23	0.16					
		10.0	5.6	0.66	0.61	0.42	0.25	0.18					
		17.8	11.1	0.54	0.52	0.40	0.28	0.21					
		17.8	5.6	0.64	0.61	0.46	0.30	0.22					
		45.6	11.1	0.53	0.52	0.43	0.32	0.25					
		45.6	11.1	0.63	0.61	0.49	0.35	0.27					
45° Slope	Down ↘	32.2	5.6	0.865	0.78	0.44	0.24	0.16					
		10.0	16.7	0.68	0.64	0.43	0.26	0.18					
		10.0	5.6	0.87	0.80	0.49	0.28	0.19					
		17.8	11.1	0.75	0.71	0.50	0.32	0.23					
		17.8	5.6	0.87	0.82	0.56	0.34	0.24					
		45.6	11.1	0.75	0.73	0.56	0.39	0.29					
		45.6	11.1	0.87	0.83	0.62	0.41	0.31					
Horiz.	Down ↓	32.2	5.6	2.06	1.63	0.63	0.28	0.18					
		10.0	16.7	1.87	1.57	0.71	0.34	0.22					
		10.0	5.6	2.24	1.82	0.75	0.35	0.22					
		17.8	11.1	2.13	1.84	0.90	0.44	0.29					
		17.8	5.6	2.43	2.05	0.95	0.46	0.29					
		45.6	11.1	2.19	1.96	1.10	0.58	0.39					
		45.6	11.1	2.57	2.26	1.18	0.61	0.40					

[a] See Chapter 25. Thermal resistance values were determined from $R = 1/C$, where $C = h_c + \varepsilon_{eff} h_r$, h_c is conduction/convection coefficient, $\varepsilon_{eff} h_r$ is radiation coefficient $\approx 0.227\varepsilon_{eff}[(t_m + 273)/100]^3$, and t_m is mean temperature of air space. Values for h_c were determined from data developed by Robinson et al. (1954). Equations (5) to (7) in Yarbrough (1983) show data in this table in analytic form. For extrapolation from this table to air spaces less than 12.5 mm (e.g., insulating window glass), assume $h_c = 21.8(1 + 0.00274t_m)/l$, where l is air space thickness in mm, and h_c is heat transfer in W/(m^2·K) through air space only.

[b] Values based on data presented by Robinson et al. (1954). (Also see Chapter 4, Tables 5 and 6, and Chapter 33). Values apply for ideal conditions (i.e., air spaces of uniform thickness bounded by plane, smooth, parallel surfaces with no air leakage to or from the space). **This table should not be used for hollow siding or profiled cladding: see Table 1.** For greater accuracy, use overall U-factors determined through guarded hot box (ASTM Standard C1363) testing. Thermal resistance values for multiple air spaces must be based on careful estimates of mean temperature differences for each air space.

[c] A single resistance value cannot account for multiple air spaces; each air space requires a separate resistance calculation that applies only for established boundary conditions. Resistances of horizontal spaces with heat flow downward are substantially independent of temperature difference.

[d] Interpolation is permissible for other values of mean temperature, temperature difference, and effective emittance $\square_{eff}$. Interpolation and moderate extrapolation for air spaces greater than 90 mm are also permissible.

[e] Effective emittance $\square_{eff}$ of air space is given by $1/\varepsilon_{eff} = 1/\varepsilon_1 + 1/\varepsilon_2 - 1$, where $\square_1$ and $\square_2$ are emittances of surfaces of air space (see Table 2). **Also, oxidation, corrosion, and accumulation of dust and dirt can dramatically increase surface emittance. Emittance values of 0.05 should only be used where the highly reflective surface can be maintained over the service life of the assembly.**

Table 5-15 SI Building and Insulating Materials: Design Values[a]

(Table 1, Chapter 26, 2017 ASHRAE Handbook—Fundamentals)

Description	Density, kg/m^3	Conductivity[b] k, W/(m·K)	Resistance R, (m^2·K)/W	Specific Heat, kJ/(kg·K)	Reference[l]
Insulating Materials					
Blanket and batt[c,d]					
Glass-fiber batts				0.8	Kumaran (2002)
	7.5 to 8.2	0.046 to 0.048	—	—	Four manufacturers (2011)
	9.8 to 12	0.040 to 0.043	—	—	Four manufacturers (2011)
	13 to 14	0.037 to 0.039	—	—	Four manufacturers (2011)
	22	0.033	—	—	Four manufacturers (2011)
Rock and slag wool batts	—	—	—	0.8	Kumaran (1996)
	32 to 37	0.036 to 0.037	—	—	One manufacturer (2011)
	45	0.033 to 0.035	—	—	One manufacturer (2011)
Mineral wool, felted	16 to 48	0.040	—	—	CIBSE (2006), NIST (2000)
	16 to 130	0.035	—	—	NIST (2000)
Board and slabs					
Cellular glass	120	0.042	—	0.8	One manufacturer (2011)
Cement fiber slabs, shredded wood with Portland cement binder	400 to 430	0.072 to 0.076	—	—	
with magnesia oxysulfide binder	350	0.082	—	1.3	
Glass fiber board	—	—	—	0.8	Kumaran (1996)
	24 to 96	0.033 to 0.035	—	—	One manufacturer (2011)
Expanded rubber (rigid)	64	0.029	—	1.7	Nottage (1947)
Extruded polystyrene, smooth skin	—	—	—	1.5	Kumaran (1996)
aged per Can/ULC Standard S770-2003	22 to 58	0.026 to 0.029	—	—	Four manufacturers (2011)
aged 180 days	22 to 58	0.029			One manufacturer (2011)
European product	30	0.030			One manufacturer (2011)
aged 5 years at 24°C	32 to 35	0030	—	—	One manufacturer (2011)
blown with low global warming potential (GWP) (<5) blowing agent		0.035 to 0.036	—	—	One manufacturer (2011)
Expanded polystyrene, molded beads	—	—	—	1.5	Kumaran (1996)
	16 to 24	0.035 to 0.037	—	—	Independent test reports (2008)
	29	0.033	—	—	Independent test reports (2008)
Mineral fiberboard, wet felted	160	0.037	—	0.8	Kumaran (1996)
Rock wool board	—	—	—	0.8	Kumaran (1996)
floors and walls	64 to 130	0.033 to 0.036	—	—	Five manufacturers (2011)
roofing	160 to 180.	0.039 to 0.042	—	0.8	Five manufacturers (2011)
Acoustical tile[e]	340 to 370	0.052 to 0.053	—	0.6 to 0.8	
Perlite board	140	0.052	—	—	One manufacturer (2010)
Polyisocyanurate	—	—	—	1.5	Kumaran (1996)
unfaced, aged per Can/ULC Standard S770-2003	26 to 37	0.023 to 0.025	—	—	Seven manufacturers (2011)
with foil facers, aged 180 days	—	0.022 to 0.023	—	—	Two manufacturers (2011)
Phenolic foam board with facers, aged	—	0.020 to 0.023	—	—	One manufacturer (2011)
Loose fill					
Cellulose fiber, loose fill	—	—	—	1.4	NIST (2000), Kumaran (1996)
attic application up to 100 mm	16 to 19	0.045 to 0.046	—	—	Four manufacturers (2011)
attic application > 100 mm	19 to 26	0.039 to 0.040	—	—	Four manufacturers (2011)
wall application, dense packed	56	0.039 to 0.040	—	—	One manufacturer (2011)
Perlite, expanded	32 to 64	0.039 to 0.045	—	1.1	(Manufacturer, pre 2001)
	64 to 120	0.045 to 0.052	—	—	(Manufacturer, pre 2001)
	120 to 180	0.052 to 0.061	—	—	(Manufacturer, pre 2001)
Glass fiber[d]					
attics, ~100 to 600 mm	6.4 to 8.0	0.052 to 0.055	—	—	Four manufacturers (2011)
attics, ~600 to 1100 mm	8 to 9.6	0.049 to 0.052	—	—	Four manufacturers (2011)
closed attic or wall cavities	29 to 37	0.035 to 0.036	—	—	Four manufacturers (2011)
Rock and slag wool[d]					
attics, ~90 to 115 mm	24 to 26	0.049	—	—	Three manufacturers (2011)
attics, ~125 to 430 mm	24 to 29	0.046 to 0.048	—	—	Three manufacturers (2011)
closed attic or wall cavities	64	0.039 to 0.042	—	—	Three manufacturers (2011)
Vermiculite, exfoliated	112 to 131	0.068	—	1.3	Sabine et al. (1975)
	64 to 96	0.063	—	—	Manufacturer (pre 2001)
Spray-applied					
Cellulose, sprayed into open wall cavities	26 to 42	0.039 to 0.040	—	—	Two manufacturers (2011)
Glass fiber, sprayed into open wall or attic cavities	16	0.039 to 0.042	—	—	Manufacturers' association (2011)
	29 to 37	0.033 to 0.037	—	—	Four manufacturers (2011)
Polyurethane foam	—	—	—	1.5	Kumaran (2002)
low density, open cell	7.2 to 10	0.037 to 0.042	—	—	Three manufacturers (2011)
medium density, closed cell, aged 180 days	30 to 51	0.020 to 0.029	—	—	Five manufacturers (2011)

Table 5-15 SI Building and Insulating Materials: Design Values[a] *(Continued)*

(Table 1, Chapter 26, 2017 ASHRAE Handbook—Fundamentals)

Description	Density, kg/m^3	Conductivity[b] k, $W/(m \cdot K)$	Resistance R, $(m^2 \cdot K)/W$	Specific Heat, $kJ/(kg \cdot K)$	Reference[l]
Building Board and Siding					
Board					
Asbestos/cement board	1900	0.57	—	1.00	Nottage (1947)
Cement board	1150	0.25	—	0.84	Kumaran (2002)
Fiber/cement board	1400	0.25	—	0.84	Kumaran (2002)
	1000	0.19	—	0.84	Kumaran (1996)
	400	0.07	—	1.88	Kumaran (1996)
	300	0.06	—	1.88	Kumaran (1996)
Gypsum or plaster board	640	0.16	—	1.15	Kumaran (2002)
Oriented strand board (OSB) 9 to 11 mm	650	—	0.11	1.88	Kumaran (2002)
12.7 mm	650	—	0.12	1.88	Kumaran (2002)
Plywood (douglas fir) 12.7 mm	460	—	0.14	1.88	Kumaran (2002)
15.9 mm	540	—	0.15	1.88	Kumaran (2002)
Plywood/wood panels 19.0 mm	450	—	0.19	1.88	Kumaran (2002)
Vegetable fiber board	650	—	0.11	1.88	Kumaran (2002)
sheathing, regular density 12.7 mm	290	—	0.23	1.30	Lewis (1967)
intermediate density 12.7 mm	350	—	0.19	1.30	Lewis (1967)
nail-based sheathing 12.7 mm	400	—	0.19	1.30	
shingle backer 9.5 mm	290	—	0.17	1.30	
sound deadening board 12.7 mm	240	—	0.24	1.26	
tile and lay-in panels, plain or acoustic	290	0.058	—	0.59	
laminated paperboard	480	0.072	—	1.38	Lewis (1967)
homogeneous board from repulped paper	480	0.072	—	1.17	
Hardboard					
medium density	800	0.105	—	1.30	Lewis (1967)
high density, service-tempered grade and service grade	880	0.12	—	1.34	Lewis (1967)
high density, standard-tempered grade	1010	0.144	—	1.34	Lewis (1967)
Particleboard					
low density	590	0.102	—	1.30	Lewis (1967)
medium density	800	0.135	—	1.30	Lewis (1967)
high density	1000	1.18	—	—	Lewis (1967)
underlayment 15.9 mm	640	—	1.22	1.21	Lewis (1967)
Waferboard	700	0.072	—	1.88	Kumaran (1996)
Shingles					
Asbestos/cement	1900	—	0.037	—	
Wood, 400 mm, 190 mm exposure	—	—	0.15	1.30	
Wood, double, 400 mm, 300 mm exposure	—	—	0.21	1.17	
Wood, plus ins. backer board 8 mm	—	—	0.25	1.30	
Siding					
Asbestos/cement, lapped 6.4 mm	—	—	0.037	1.01	
Asphalt roll siding	—	—	0.026	1.47	
Asphalt insulating siding (12.7 mm bed)	—	—	0.26	1.47	
Hardboard siding 11 mm		—	—	0.12	1.17
Wood, drop, 200 mm 25 mm	—	—	0.14	1.17	
Wood, bevel					
200 mm, lapped 13 mm	—	—	0.14	1.17	
250 mm, lapped 19 mm	—	—	0.18	1.17	
Wood, plywood, lapped 9.5 mm	—	—	0.10	1.22	
Aluminum, steel, or vinyl,[h, i] over sheathing			—		
hollow-backed	—	—	0.11	1.22[i]	
insulating-board-backed 9.5 mm	—	—	0.32	1.34	
foil-backed 9.5 mm	—	—	0.52	—	
Architectural (soda-lime float) glass	2500	1.0	—	0.84	
Building Membrane					
Vapor-permeable felt	—	—	0.011	—	
Vapor: seal, 2 layers of mopped 0.73 kg/m^2 felt	—	—	0.21	—	
Vapor: seal, plastic film	—	—	Negligible	—	
Finish Flooring Materials					
Carpet and rebounded urethane pad 19 mm	110	—	0.42	—	NIST (2000)
Carpet and rubber pad (one-piece) 9.5 mm	320	—	0.12	—	NIST (2000)
Pile carpet with rubber pad 9.5 to 12.7 mm	290	—	0.28	—	NIST (2000)
Linoleum/cork tile 6.4 mm	465	—	0.09	—	NIST (2000)
PVC/rubber floor covering	—	0.40	—	—	CIBSE (2006)
rubber tile 25 mm	1900	—	0.06	—	NIST (2000)
terrazzo 25 mm	—	—	0.014	0.80	
Metals (See Chapter 33, Table 3)					

Table 5-15 SI Building and Insulating Materials: Design Values[a] *(Continued)*

(Table 1, Chapter 26, 2017 ASHRAE Handbook—Fundamentals)

Description	Density, kg/m^3	Conductivity[b] k, W/(m·K)	Resistance R, (m^2·K)/W	Specific Heat, kJ/(kg·K)	Reference[l]
Roofing					
Asbestos/cement shingles	1920	—	0.037	1.00	
Asphalt (bitumen with inert fill)	1600	0.43	—	—	CIBSE (2006)
	1900	0.58	—	—	CIBSE (2006)
	2300	1.15	—	—	CIBSE (2006)
Asphalt roll roofing	920	—	0.027	1.51	
Asphalt shingles	920	—	0.078	1.26	
Built-up roofing 10 mm	920	—	0.059	1.47	
Mastic asphalt (heavy, 20% grit)	950	0.19	—	—	CIBSE (2006)
Reed thatch	270	0.09	—	—	CIBSE (2006)
Roofing felt	2250	1.20	—	—	CIBSE (2006)
Slate 13 mm	—	—	0.009	1.26	
Straw thatch	240	0.07	—	—	CIBSE (2006)
Wood shingles, plain and plastic-film-faced	—	—	0.166	1.30	
Plastering Materials					
Cement plaster, sand aggregate	1860	0.72	—	0.84	
Sand aggregate 10 mm	—	—	0.013	0.84	
20 mm	—	—	0.026	0.84	
Gypsum plaster	1120	0.38	—	—	CIBSE (2006)
	1280	0.46	—	—	CIBSE (2006)
Lightweight aggregate 13 mm		720	—	0.056	—
16 mm		720	—	0.066	—
on metal lath 19 mm	—	—	0.083	—	
Perlite aggregate	720	0.22	—	1.34	
Sand aggregate	1680	0.81	—	0.84	
on metal lath 19 mm	—	—	0.023	—	
Vermiculite aggregate	480	0.14	—	—	CIBSE (2006)
	600	0.20	—	—	CIBSE (2006)
	720	0.25	—	—	CIBSE (2006)
	840	0.26	—	—	CIBSE (2006)
	960	0.30	—	—	CIBSE (2006)
Perlite plaster	400	0.08	—	—	CIBSE (2006)
	600	0.19	—	—	CIBSE (2006)
Pulpboard or paper plaster	600	0.07	—	—	CIBSE (2006)
Sand/cement plaster, conditioned	1560	0.63	—	—	CIBSE (2006)
Sand/cement/lime plaster, conditioned	1440	0.48	—	—	CIBSE (2006)
Sand/gypsum (3:1) plaster, conditioned	1550	0.65	—	—	CIBSE (2006)
Masonry Materials					
Masonry units					
Brick, fired clay	2400	1.21 to 1.47	—	—	Valore (1988)
	2240	1.07 to 1.30	—	—	Valore (1988)
	2080	0.92 to 1.12	—	—	Valore (1988)
	1920	0.81 to 0.98	—	0.80	Valore (1988)
	1760	0.71 to 0.85	—	—	Valore (1988)
	1600	0.61 to 0.74	—	—	Valore (1988)
	1440	0.52 to 0.62	—	—	Valore (1988)
	1280	0.43 to 0.53	—	—	Valore (1988)
	1120	0.36 to 0.45	—	—	Valore (1988)
Clay tile, hollow					
1 cell deep 75 mm	—	—	0.14	0.88	Rowley and Algren (1937)
100 mm	—	—	0.20	—	Rowley and Algren (1937)
2 cells deep 150 mm	—	—	0.27	—	Rowley and Algren (1937)
200 mm	—	—	0.33	—	Rowley and Algren (1937)
250 mm	—	—	0.39	—	Rowley and Algren (1937)
3 cells deep 300 mm	—	—	0.44	—	Rowley and Algren (1937)
Lightweight brick	800	0.20	—	—	Kumaran (1996)
	770	0.22	—	—	Kumaran (1996)
Concrete blocks[f, g]					
Limestone aggregate					
~200 mm, 16.3 kg, 2200 kg/m^3 concrete, 2 cores	—	—	—	—	
with perlite-filled cores	—	—	0.37	—	Valore (1988)
~300 mm, 25 kg, 2200 kg/m^3 concrete, 2 cores	—		—	—	
with perlite-filled cores	—	—	0.65	—	Valore (1988)
Normal-weight aggregate (sand and gravel)					
~200 mm, 16 kg, 2100 kg/m^3 concrete, 2 or 3 cores…	—	—	0.20 to 0.17	0.92	Van Geem (1985)
with perlite-filled cores	—	—	0.35	—	Van Geem (1985)
with vermiculite-filled cores	—	—	0.34 to 0.24	—	Valore (1988)

Table 5-15 SI Building and Insulating Materials: Design Values[a] *(Continued)*

(Table 1, Chapter 26, 2017 ASHRAE Handbook—Fundamentals)

Description	Density, kg/m^3	Conductivity[b] k, W/(m·K)	Resistance R, (m^2·K)/W	Specific Heat, kJ/(kg·K)	Reference[l]
~300 mm, 22.7 kg, 2000 kg/m^3 concrete, 2 cores	—	—	0.217	0.92	Valore (1988)
Medium-weight aggregate (combinations of normal and lightweight aggregate)					
~200 mm, 13 kg, 1550 to 1800 kg/m^3 concrete, 2 or 3 cores	—	—	0.30 to 0.22	—	Van Geem (1985)
with perlite-filled cores	—	—	0.65 to 0.41	—	Van Geem (1985)
with vermiculite-filled cores	—	—	0.58	—	Van Geem (1985)
with molded-EPS-filled (beads) cores	—	—	0.56	—	Van Geem (1985)
with molded EPS inserts in cores	—	—	0.47	—	Van Geem (1985)
Low-mass aggregate (expanded shale, clay, slate or slag, pumice)					
~150 mm, 7 1/2 kg, 1400 kg/m^2 concrete, 2 or 3 cores	—	—	0.34 to 0.29	—	Van Geem (1985)
with perlite-filled cores	—	—	0.74	—	Van Geem (1985)
with vermiculite-filled cores	—	—	0.53	—	Van Geem (1985)
200 mm, 8 to 10 kg, 1150 to 1380 kg/m^2 concrete	—	—	0.56 to 0.33	0.88	Van Geem (1985)
with perlite-filled cores	—	—	1.20 to 0.77	—	Van Geem (1985)
with vermiculite-filled cores	—	—	0.93 to 0.69	—	Shu et al. (1979)
with molded-EPS-filled (beads) cores	—	—	0.85	—	Shu et al. (1979)
with UF foam-filled cores	—	—	0.79	—	Shu et al. (1979)
with molded EPS inserts in cores	—	—	0.62	—	Shu et al. (1979)
300 mm, 16 kg, 1400 kg/m^3, concrete, 2 or 3 cores	—	—	0.46 to 0.40	—	Van Geem (1985)
with perlite-filled cores	—	—	1.6 to 1.1	—	Van Geem (1985)
with vermiculite-filled cores	—	—	1.0	—	Valore (1988)
Stone, lime, or sand	2880	10.4	—	—	Valore (1988)
Quartzitic and sandstone	2560	6.2	—	—	Valore (1988)
	2240	3.46	—	—	Valore (1988)
	1920	1.88	—	0.88	Valore (1988)
Calcitic, dolomitic, limestone, marble, and granite	2880	4.33	—	—	Valore (1988)
	2560	3.17	—	—	Valore (1988)
	2240	2.31	—	—	Valore (1988)
	1920	1.59	—	0.88	Valore (1988)
	1600	1.15	—	—	Valore (1988)
Gypsum partition tile					
75 by 300 by 760 mm, solid	—	—	0.222	0.79	Rowley and Algren (1937)
4 cells	—	—	0.238	—	Rowley and Algren (1937)
100 by 300 by 760 mm, 3 cells	—	—	0.294	—	Rowley and Algren (1937)
Limestone	2400	0.57	—	0.84	Kumaran (2002)
	2600	0.93	—	0.84	Kumaran (2002)
Concretes[i]					
Sand and gravel or stone aggregate concretes	2400	1.4 to 2.9	—	—	Valore (1988)
(concretes with >50% quartz or quartzite sand have conductivities in higher end of range)	2240	1.3 to 2.6	—	0.80 to 1.00	Valore (1988)
	2080	1.0 to 1.9	—	—	Valore (1988)
Low-mass aggregate or limestone concretes	1920	0.9 to 1.3	—	—	Valore (1988)
expanded shale, clay, or slate; expanded slags; cinders; pumice (with density up to 1600 kg/m^3); scoria (sanded concretes have conductivities in higher end of range)	1600	0.68 to 0.89	—	0.84	Valore (1988)
	1280	0.48 to 0.59	—	0.84	Valore (1988)
	960	0.30 to 0.36	—	—	Valore (1988)
	640	0.18	—	—	Valore (1988)
Gypsum/fiber concrete (87.5% gypsum, 12.5% wood chips)	800	0.24	—	0.84	Rowley and Algren (1937)
Cement/lime, mortar, and stucco	1920	1.40	—	—	Valore (1988)
	1600	0.97	—	—	Valore (1988)
	1280	0.65	—	—	Valore (1988)
Perlite, vermiculite, and polystyrene beads	800	0.26 to 0.27	—	—	Valore (1988)
	640	0.20 to 0.22	—	0.63 to 0.96	Valore (1988)
	480	0.16	—	—	Valore (1988)
	320	0.12	—	—	Valore (1988)
Foam concretes	1920	0.75	—	—	Valore (1988)
	1600	0.60	—	—	Valore (1988)
	1280	0.44	—	—	Valore (1988)
	1120	0.36	—	—	Valore (1988)
Foam concretes and cellular concretes	960	0.30	—	—	Valore (1988)
	640	0.20	—	—	Valore (1988)
	320	0.12	—	—	Valore (1988)
Aerated concrete (oven-dried)	430 to 800	0.20	—	0.84	Kumaran (1996)
Polystyrene concrete (oven-dried)	255 to 800	0.37	—	0.84	Kumaran (1996)
Polymer concrete	1950	1.64	—	—	Kumaran (1996)
	2200	1.03	—	—	Kumaran (1996)
Polymer cement	1870	0.78	—	—	Kumaran (1996)
Slag concrete	960	0.22	—	—	Touloukian et al (1970)
	1280	0.32	—	—	Touloukian et al. (1970)
	1600	0.43	—	—	Touloukian et al. (1970)

Table 5-15 SI Building and Insulating Materials: Design Values[a] *(Continued)*

(Table 1, Chapter 26, 2017 ASHRAE Handbook—Fundamentals)

Description	Density, kg/m³	Conductivity[b] *k*, W/(m·K)	Resistance *R*, (m²·K)/W	Specific Heat, kJ/(kg·K)	Reference[l]
	2000	1.23	—	—	Touloukian et al. (1970)
Woods (12% moisture content)[j]					
Hardwoods	—	—	—	1.63[k]	Wilkes (1979)
Oak	660 to 750	0.16 to 0.18	—	—	Cardenas and Bible (1987)
Birch	680 to 725	0.17 to 0.18	—	—	Cardenas and Bible (1987)
Maple	635 to 700	0.16 to 0.17	—	—	Cardenas and Bible (1987)
Ash	615 to 670	0.15 to 0.16	—	—	Cardenas and Bible (1987)
Softwoods	—	—	—	1.63[k]	Wilkes (1979)
Southern pine	570 to 660	0.14 to 0.16	—	—	Cardenas and Bible (1987)
Southern yellow pine	500	0.13	—	—	Kumaran (2002)
Eastern white pine	400	0.10	—	—	Kumaran (2002)
Douglas fir/larch	535 to 580	0.14 to 0.15	—	—	Cardenas and Bible (1987)
Southern cypress	500 to 515	0.13	—	—	Cardenas and Bible (1987)
Hem/fir, spruce/pine/fir	390 to 500	0.11 to 0.13	—	—	Cardenas and Bible (1987)
Spruce	400	0.09	—	—	Kumaran (2002)
Western red cedar	350	0.09	—	—	Kumaran (2002)
West coast woods, cedars	350 to 500	0.10 to 0.13	—	—	Cardenas and Bible (1987)
Eastern white cedar	360	0.10	—	—	Kumaran (2002)
California redwood	390 to 450	0.11 to 0.12	—	—	Cardenas and Bible (1987)
Pine (oven-dried)	370	0.092	—	1.88	Kumaran (1996)
Spruce (oven-dried)	395	0.10	—	1.88	Kumaran (1996)

Notes for Table 5-15 SI

(Chapter references for 2013 ASHRAE Handbook—Fundamentals)

[a] Values are for mean temperature of 75°F (24°C). Representative values for dry materials are intended as design (not specification) values for materials in normal use. Thermal values of insulating materials may differ from design values depending on in-situ properties (e.g., density and moisture content, orientation, etc.) and manufacturing variability. For properties of specific product, use values supplied by manufacturer or unbiased tests.

[b] Symbol λ also used to represent thermal conductivity.

[c] Does not include paper backing and facing, if any. Where insulation forms boundary (reflective or otherwise) of airspace, see Tables 2 and 3 for insulating value of airspace with appropriate effective emittance and temperature conditions of space.

[d] Conductivity varies with fiber diameter (see Chapter 25). Batt, blanket, and loose-fill mineral fiber insulations are manufactured to achieve specified R-values, the most common of which are listed in the table. Because of differences in manufacturing processes and materials, the product thicknesses, densities, and thermal conductivities vary over considerable ranges for a specified R-value.

[e] Insulating values of acoustical tile vary, depending on density of board and on type, size, and depth of perforations.

[f] Values for fully grouted block may be approximated using values for concrete with similar unit density.

[g] Values for concrete block and concrete are at moisture contents representative of normal use.

[h] Values for metal or vinyl siding applied over flat surfaces vary widely, depending on ventilation of the airspace beneath the siding; whether airspace is reflective or nonreflective; and on thickness, type, and application of insulating backing-board used. Values are averages for use as design guides, and were obtained from several guarded hot box tests (ASTM Standard C1363) on hollow-backed types and types made using backing of wood fiber, foamed plastic, and glass fiber. Departures of ±50% or more from these values may occur.

[i] Vinyl specific heat = 0.25 Btu/lb·°F1.0 kJ/(kg·K)

[j] See Adams (1971), MacLean (1941), and Wilkes (1979). Conductivity values listed are for heat transfer across the grain. Thermal conductivity of wood varies linearly with density, and density ranges listed are those normally found for wood species given. If density of wood species is not known, use mean conductivity value. For extrapolation to other moisture contents, the following empirical equation developed by Wilkes (1979) may be used:

$$k = 0.1791 + \frac{(1.874 \times 10^{-2} + 5.753 \times 10^{-4} M)\rho}{1 + 0.01M}$$

where ρ is density of moist wood in lb/ft³kg/m³, and M is moisture content in percent.

[k] From Wilkes (1979), an empirical equation for specific heat of moist wood at 75°F (24°C) is as follows:

$$c_p = \frac{(0.299 + 0.01M)}{(1 + 0.01M)} + \Delta c_p$$

where Δc_p accounts for heat of sorption and is denoted by

$$\Delta c_p = M(1.921 \times 10^{-3} - 3.168 \times 10^{-5} M)$$

where M is moisture content in percent by mass.

[l] Blank space in reference column indicates historical values from previous volumes of *ASHRAE Handbook*. Source of information could not be determined.

Table 5-16 SI U-Factors for Various Fenestration Products in W/($m^2 \cdot K$)

(Table 4, Chapter 26, 2017 ASHRAE Handbook—Fundamentals)

Product Type		Glass Only		Vertical Installation									
				Operable (including sliding and swinging glass doors)					Fixed				
Frame Type ID	Glazing Type	Center of Glass	Edge of Glass	Aluminum Without Thermal Break	Aluminum With Thermal Break	Reinforced Vinyl/ Aluminum Clad Wood	Wood/ Vinyl	Insulated Fiberglass/ Vinyl	Aluminum Without Thermal Break	Aluminum With Thermal Break	Reinforced Vinyl/ Aluminum Clad Wood	Wood/ Vinyl	Insulated Fiberglass/ Vinyl
	Single Glazing												
1	3 mm glass	*5.91*	*5.91*	7.01	6.08	5.27	5.20	4.83	6.38	6.06	5.58	5.58	5.40
2	6 mm acrylic/polycarb	*5.00*	*5.00*	6.23	5.35	4.59	4.52	4.18	5.55	5.23	4.77	4.77	4.61
3	3.2 mm acrylic/polycarb	*5.45*	*5.45*	6.62	5.72	4.93	4.86	4.51	5.96	5.64	5.18	5.18	5.01
	Double Glazing												
4	6 mm airspace	*3.12*	*3.63*	4.62	3.61	3.24	3.14	2.84	3.88	3.52	3.18	3.16	3.04
5	13 mm airspace	*2.73*	*3.36*	4.30	3.31	2.96	2.86	2.58	3.54	3.18	2.85	2.83	2.72
6	6 mm argon space	*2.90*	*3.48*	4.43	3.44	3.08	2.98	2.69	3.68	3.33	3.00	2.98	2.86
7	13 mm argon space	*2.56*	*3.24*	4.16	3.18	2.84	2.74	2.46	3.39	3.04	2.71	2.69	2.58
	Double Glazing, e = 0.60 on surface 2 or 3												
8	6 mm airspace	*2.95*	*3.52*	4.48	3.48	3.12	3.02	2.73	3.73	3.38	3.04	3.02	2.90
9	13 mm airspace	*2.50*	*3.20*	4.11	3.14	2.80	2.70	2.42	3.34	2.99	2.67	2.65	2.53
10	6 mm argon space	*2.67*	*3.32*	4.25	3.27	2.92	2.82	2.54	3.49	3.13	2.81	2.79	2.67
11	13 mm argon space	*2.33*	*3.08*	3.98	3.01	2.68	2.58	2.31	3.20	2.84	2.52	2.50	2.39
	Double Glazing, e = 0.40 on surface 2 or 3												
12	6 mm airspace	*2.78*	*3.40*	4.34	3.35	3.00	2.90	2.61	3.59	3.23	2.90	2.88	2.77
13	13 mm airspace	*2.27*	*3.04*	3.93	2.96	2.64	2.54	2.27	3.15	2.79	2.48	2.46	2.35
14	6 mm argon space	*2.44*	*3.16*	4.07	3.09	2.76	2.66	2.38	3.30	2.94	2.62	2.60	2.49
15	13 mm argon space	*2.04*	*2.88*	3.75	2.79	2.48	2.38	2.11	2.95	2.60	2.29	2.27	2.16
	Double Glazing, e = 0.20 on surface 2 or 3												
16	6 mm airspace	*2.56*	*3.24*	4.16	3.18	2.84	2.74	2.46	3.39	3.04	2.71	2.69	2.58
17	13 mm airspace	*1.99*	*2.83*	3.70	2.75	2.44	2.34	2.07	2.91	2.55	2.24	2.22	2.12
18	6 mm argon space	*2.16*	*2.96*	3.84	2.88	2.56	2.46	2.19	3.05	2.70	2.38	2.36	2.26
19	13 mm argon space	*1.70*	*2.62*	3.47	2.53	2.24	2.14	1.88	2.66	2.30	2.00	1.98	1.88
	Double Glazing, e = 0.10 on surface 2 or 3												
20	6 mm airspace	*2.39*	*3.12*	4.02	3.05	2.72	2.62	2.34	3.25	2.89	2.57	2.55	2.44
21	13 mm airspace	*1.82*	*2.71*	3.56	2.62	2.32	2.22	1.96	2.76	2.40	2.10	2.08	1.98
22	6 mm argon space	*1.99*	*2.83*	3.70	2.75	2.44	2.34	2.07	2.91	2.55	2.24	2.22	2.12
23	13 mm argon space	*1.53*	*2.49*	3.33	2.40	2.12	2.02	1.76	2.51	2.16	1.86	1.84	1.74
	Double Glazing, e = 0.05 on surface 2 or 3												
24	6 mm airspace	*2.33*	*3.08*	3.98	3.01	2.68	2.58	2.31	3.20	2.84	2.52	2.50	2.39
25	13 mm airspace	*1.70*	*2.62*	3.47	2.53	2.24	2.14	1.88	2.66	2.30	2.00	1.98	1.88
26	6 mm argon space	*1.87*	*2.75*	3.61	2.66	2.36	2.26	2.00	2.81	2.45	2.15	2.12	2.02
27	13 mm argon space	*1.42*	*2.41*	3.24	2.31	2.04	1.94	1.69	2.42	2.06	1.76	1.74	1.65
	Triple Glazing												
28	6 mm airspace	*2.16*	*2.96*	3.78	2.78	2.46	2.42	2.17	3.02	2.68	2.36	2.36	2.25
29	13 mm airspace	*1.76*	*2.67*	3.46	2.47	2.18	2.14	1.90	2.68	2.34	2.03	2.03	1.92
30	6 mm argon space	*1.93*	*2.79*	3.60	2.60	2.30	2.26	2.02	2.82	2.49	2.17	2.17	2.06
31	13 mm argon space	*1.65*	*2.58*	3.36	2.39	2.10	2.06	1.83	2.58	2.24	1.93	1.93	1.83
	Triple Glazing, e = 0.20 on surface 2, 3, 4, or 5												
32	6 mm airspace	*1.87*	*2.75*	3.55	2.56	2.26	2.22	1.98	2.78	2.44	2.12	2.12	2.01
33	13 mm airspace	*1.42*	*2.41*	3.18	2.21	1.94	1.90	1.67	2.38	2.05	1.74	1.74	1.64
34	6 mm argon space	*1.59*	*2.54*	3.32	2.34	2.06	2.02	1.79	2.53	2.20	1.89	1.89	1.78
35	13 mm argon space	*1.25*	*2.28*	3.04	2.08	1.82	1.78	1.55	2.24	1.90	1.60	1.60	1.50
	Triple Glazing, e = 0.20 on surfaces 2 or 3 and 4 or 5												
36	6 mm airspace	*1.65*	*2.58*	3.36	2.39	2.10	2.06	1.83	2.58	2.24	1.93	1.93	1.83
37	13 mm airspace	*1.14*	*2.19*	2.95	1.99	1.74	1.69	1.48	2.14	1.80	1.50	1.50	1.40
38	6 mm argon space	*1.31*	*2.32*	3.09	2.12	1.86	1.82	1.59	2.29	1.95	1.65	1.65	1.55
39	13 mm argon space	*0.97*	*2.05*	2.81	1.86	1.62	1.57	1.36	1.99	1.65	1.36	1.36	1.26
	Triple Glazing, e = 0.10 on surfaces 2 or 3 and 4 or 5												
40	6 mm airspace	*1.53*	*2.49*	3.27	2.30	2.02	1.98	1.75	2.48	2.15	1.84	1.84	1.73
41	13 mm airspace	*1.02*	*2.10*	2.85	1.90	1.66	1.61	1.40	2.04	1.70	1.41	1.41	1.31
42	6 mm argon space	*1.19*	*2.23*	2.99	2.04	1.78	1.73	1.52	2.19	1.85	1.55	1.55	1.45
43	13 mm argon space	*0.80*	*1.92*	2.67	1.73	1.49	1.45	1.24	1.84	1.51	1.22	1.22	1.12
	Quadruple Glazing, e = 0.10 on surfaces 2 or 3 and 4 or 5												
44	6 mm airspaces	*1.25*	*2.28*	3.04	2.08	1.82	1.78	1.55	2.24	1.90	1.60	1.60	1.50
45	13 mm airspaces	*0.85*	*1.96*	2.71	1.77	1.54	1.49	1.28	1.89	1.55	1.26	1.26	1.17
46	6 mm argon spaces	*0.97*	*2.05*	2.81	1.86	1.62	1.57	1.36	1.99	1.65	1.36	1.36	1.26
47	13 mm argon spaces	*0.68*	*1.83*	2.57	1.64	1.41	1.37	1.16	1.74	1.41	1.12	1.12	1.03
48	6 mm krypton spaces	*0.68*	*1.83*	2.57	1.64	1.41	1.37	1.16	1.74	1.41	1.12	1.12	1.03

Notes:

1. All heat transmission coefficients in this table include film resistances and are based on winter conditions of –18°C outdoor air temperature and 21°C indoor air temperature, with 6.7 m/s outdoor air velocity and zero solar flux. Except for single glazing, small changes in the indoor and outdoor temperatures do not significantly affect overall U-factors. Coefficients are for vertical position except skylight values, which are for 20° from horizontal with heat flow up.
2. Glazing layer surfaces are numbered from outdoor to indoor. Double, triple, and quadruple refer to number of glazing panels. All data are based on 3 mm glass, unless otherwise noted. Thermal conductivities are: 0.917 W/(m·K) for glass, and 0.19 W/(m·K) for acrylic and polycarbonate.
3. Standard spacers are metal. Edge-of-glass effects are assumed to extend over the 63.5 mm band around perimeter of each glazing unit.

Table 5-16 SI U-Factors for Various Fenestration Products in W/(m^2·K) (*Continued*)

Vertical Installation					Sloped Installation									
Garden Windows		Curtainwall			Glass Only (Skylights)		Manufactured Skylight				Site-Assembled Sloped/Overhead Glazing			
Aluminum Without Thermal Break	Wood/ Vinyl	Aluminum Without Thermal Break	Aluminum With Thermal Break	Structural Glazing	Center of Glass	Edge of Glass	Aluminum Without Thermal Break	Aluminum With Thermal Break	Reinforced Vinyl/ Aluminum Clad Wood	Wood/ Vinyl	Aluminum Without Thermal Break	Aluminum With Thermal Break	Structural Glazing	ID
14.21	11.94	6.86	6.27	6.27	*6.76*	*6.76*	10.03	9.68	9.16	8.05	7.66	7.64	7.10	1
12.70	10.42	6.03	5.44	5.44	*5.85*	*5.85*	9.09	8.74	8.23	7.45	6.83	6.80	6.27	2
13.45	11.18	6.44	5.86	5.86	*6.30*	*6.30*	9.56	9.21	8.70	7.89	7.24	7.22	6.68	3
9.78	7.50	4.38	3.79	3.56	*3.29*	*3.75*	6.23	5.46	5.21	4.79	4.54	4.71	3.75	4
9.19	6.92	4.03	3.45	3.22	*3.24*	*3.71*	6.17	5.41	5.16	4.74	4.49	4.68	3.70	5
9.44	7.17	4.18	3.60	3.37	*3.01*	*3.56*	5.96	5.19	4.94	4.54	4.30	4.52	3.51	6
8.94	6.67	3.89	3.30	3.07	*3.01*	*3.56*	5.96	5.19	4.94	4.54	4.30	4.52	3.51	7
9.53	7.25	4.23	3.65	3.41	*3.07*	*3.60*	6.01	5.24	4.99	4.59	4.35	4.56	3.55	8
8.86	6.58	3.84	3.25	3.02	*3.01*	*3.56*	5.96	5.19	4.94	4.54	4.30	4.52	3.51	9
9.11	6.84	3.99	3.40	3.17	*2.78*	*3.40*	5.74	4.97	4.72	4.34	4.10	4.37	3.31	10
8.61	6.33	3.69	3.11	2.88	*2.78*	*3.40*	5.74	4.97	4.72	4.34	4.10	4.37	3.31	11
9.28	7.00	4.08	3.50	3.27	*2.90*	*3.48*	5.85	5.08	4.83	4.44	4.20	4.45	3.41	12
8.52	6.25	3.64	3.06	2.83	*2.84*	*3.44*	5.79	5.02	4.78	4.39	4.15	4.41	3.36	13
8.77	6.50	3.79	3.21	2.97	*2.50*	*3.20*	5.46	4.69	4.45	4.09	3.86	4.18	3.07	14
8.18	5.91	3.45	2.86	2.63	*2.61*	*3.28*	5.57	4.80	4.56	4.19	3.96	4.25	3.17	15
8.94	6.67	3.89	3.30	3.07	*2.61*	*3.28*	5.57	4.80	4.56	4.19	3.96	4.25	3.17	16
8.10	5.82	3.40	2.81	2.58	*2.61*	*3.28*	5.57	4.80	4.56	4.19	3.96	4.25	3.17	17
8.35	6.08	3.54	2.96	2.73	*2.22*	*3.00*	5.19	4.42	4.18	3.84	3.61	3.98	2.83	18
7.67	5.39	3.15	2.56	2.33	*2.27*	*3.04*	5.24	4.47	4.24	3.89	3.66	4.02	2.88	19
8.69	6.42	3.74	3.16	2.92	*2.50*	*3.20*	5.46	4.69	4.45	4.09	3.86	4.18	3.07	20
7.84	5.57	3.25	2.66	2.43	*2.50*	*3.20*	5.46	4.69	4.45	4.09	3.86	4.18	3.07	21
8.10	5.82	3.40	2.81	2.58	*2.04*	*2.88*	5.02	4.25	4.02	3.69	3.46	3.86	2.68	22
7.41	5.14	3.00	2.42	2.18	*2.16*	*2.96*	5.13	4.36	4.13	3.79	3.56	3.94	2.78	23
8.61	6.33	3.69	3.11	2.88	*2.39*	*3.12*	5.35	4.58	4.34	3.99	3.76	4.10	2.97	24
7.67	5.39	3.15	2.56	2.33	*2.44*	*3.16*	5.41	4.64	4.40	4.04	3.81	4.14	3.02	25
7.93	5.65	3.30	2.71	2.48	*1.93*	*2.79*	4.91	4.14	3.91	3.58	3.37	3.77	2.58	26
7.24	4.96	2.90	2.32	2.09	*2.04*	*2.88*	5.02	4.25	4.02	3.69	3.46	3.86	2.68	27
see	see	3.48	2.91	2.62	*2.22*	*3.00*	5.13	4.24	4.03	3.63	3.55	3.92	2.70	28
note	note	3.14	2.57	2.27	*2.04*	*2.88*	4.96	4.07	3.87	3.48	3.40	3.80	2.56	29
7	7	3.28	2.71	2.42	*1.99*	*2.83*	4.91	4.01	3.81	3.43	3.35	3.76	2.51	30
		3.04	2.47	2.17	*1.87*	*2.75*	4.80	3.90	3.70	3.33	3.25	3.68	2.41	31
see	see	3.23	2.66	2.37	*1.93*	*2.79*	4.85	3.96	3.76	3.38	3.30	3.72	2.46	32
note	note	2.84	2.27	1.97	*1.76*	*2.67*	4.68	3.79	3.59	3.22	3.16	3.59	2.31	33
7	7	2.99	2.42	2.12	*1.59*	*2.54*	4.52	3.63	3.43	3.07	3.01	3.47	2.17	34
		2.69	2.12	1.83	*1.53*	*2.49*	4.46	3.57	3.37	3.02	2.96	3.43	2.12	35
see	see	3.04	2.47	2.17	*1.65*	*2.58*	4.57	3.68	3.48	3.12	3.06	3.51	2.22	36
note	note	2.59	2.02	1.73	*1.53*	*2.49*	4.46	3.57	3.37	3.02	2.96	3.43	2.12	37
7	7	2.74	2.17	1.87	*1.36*	*2.36*	4.29	3.40	3.21	2.86	2.81	3.30	1.97	38
		2.44	1.87	1.58	*1.25*	*2.28*	4.18	3.29	3.10	2.76	2.71	3.22	1.87	39
see	see	2.94	2.37	2.07	*1.53*	*2.49*	4.46	3.57	3.37	3.02	2.96	3.43	2.12	40
note	note	2.49	1.92	1.63	*1.42*	*2.41*	4.35	3.46	3.27	2.91	2.86	3.34	2.02	41
7	7	2.64	2.07	1.78	*1.19*	*2.23*	4.13	3.24	3.04	2.71	2.66	3.18	1.82	42
		2.29	1.72	1.43	*1.14*	*2.19*	4.07	3.18	2.99	2.66	2.61	3.13	1.77	43
		2.69	2.12	1.83	*1.25*	*2.28*	4.18	3.29	3.10	2.76	2.71	3.22	1.87	44
see	see	2.34	1.77	1.48	*1.08*	*2.14*	4.02	3.12	2.93	2.60	2.56	3.09	1.72	45
note	note	2.44	1.87	1.58	*1.02*	*2.10*	3.96	3.07	2.88	2.55	2.51	3.05	1.67	46
7	7	2.19	1.62	1.33	*0.91*	*2.01*	3.85	2.96	2.77	2.45	2.41	2.96	1.58	47
		2.19	1.62	1.33	*0.74*	*1.87*	3.68	2.79	2.60	2.29	2.26	2.83	1.43	48

4. Product sizes are described in Figure 4, Chapter 15, 2013 *ASHRAE Handbook—Fundamentals* and frame U-factors are from Table 1.
5. Use $U = 3.40$ W/(m^2·K) for glass block with mortar but without reinforcing or framing.
6. Use of this table should be limited to that of an estimating tool for early phases of design.
7. Values for triple- and quadruple-glazed garden windows are not listed, because these are not common products.
8. U-factors in this table were determined using NFRC 100-91. They have not been updated to the current rating methodology in NFRC 100-2010.

Table 5-17 SI Design U-Factors of Swinging Doors in W/(m²·K)

(Table 6, Chapter 15, 2017 ASHRAE Handbook—Fundamentals)

Door Type (Rough Opening = 970 × 2080 mm)	No Glazing	Single Glazing	Double Glazing with 12.7 mm Air Space	Double Glazing with $e = 0.10$, 12.7 mm Argon
Slab Doors				
Wood slab in wood frame[a]	2.61			
6% glazing (560 × 200 lite)	—	2.73	2.61	2.50
25% glazing (560 × 910 lite)	—	3.29	2.61	2.38
45% glazing (560 × 1620 lite)	—	3.92	2.61	2.21
More than 50% glazing		Use Table 4 (operable)		
Insulated steel slab with wood edge in wood frame[b]	0.91			
6% glazing (560 × 200 lite)	—	1.19	1.08	1.02
25% glazing (560 × 910 lite)	—	2.21	1.48	1.31
45% glazing (560 × 1630 lite)	—	3.29	1.99	1.48
More than 50% glazing		Use Table 4 (operable)		
Foam-insulated steel slab with metal edge in steel frame[c]	2.10			
6% glazing (560 × 200 lite)	—	2.50	2.33	2.21
25% glazing (560 × 910 lite)	—	3.12	2.73	2.50
45% glazing (560 × 1630 lite)	—	4.03	3.18	2.73
More than 50% glazing		Use Table 4 (operable)		
Cardboard honeycomb slab with metal edge in steel frame	3.46			
Stile-and-Rail Doors				
Sliding glass doors/French doors		Use Table 4 (operable)		
Site-Assembled Stile-and-Rail Doors				
Aluminum in aluminum frame	—	7.49	5.28	4.49
Aluminum in aluminum frame with thermal break	—	6.42	4.20	3.58

Notes:

[a] Thermally broken sill [add 0.17 W/(m²·K) for non-thermally broken sill]

[b] Non-thermally broken sill

[c] Nominal U-factors are through center of insulated panel before consideration of thermal bridges around edges of door sections and because of frame.

Table 5-18 SI Design U-factors for Revolving Doors in W/(m²·K)

(Table 7, Chapter 15, 2017 ASHRAE Handbook—Fundamentals)

Type	Size (Width × Height)	U-Factor
3-wing	2.44 × 2.13 m	4.46
	3.28 × 2.44 m	4.53
4-wing	2.13 × 1.98 m	3.56
	2.13 × 2.29 m	3.63
Open*	2.08 × 2.13 m	7.49

*U-factor of Open door determined using NFRC *Technical Document* 100-91. It has not been updated to current rating methodology in NFRC *Technical Document* 100-2004.

Example 5-5 SI Calculate the U-factor of the 38 mm by 90 mm stud wall shown in Figure 5-18. The studs are at 400 mm OC. There is 90 mm mineral fiber batt insulation (R = 2.3 (m²·K)/W) in the stud space. The inside finish is 13 mm gypsum wallboard; the outside is finished with rigid foam insulating sheathing (R = 0.7 (m²·K)/W) and 13 mm by 200 mm wood bevel lapped siding. The insulated cavity occupies approximately 75% of the transmission area; the studs, plates, and sills occupy 21%; and the headers occupy 4%.

Solution. Obtain the R-values of the various building elements from Tables 5-12 and 5-15. Assume the R = 7.0 (m²·K)/W for the wood framing. Also, assume the headers are solid wood, in this case, and group them with the studs, plates, and sills.

Element	*R* (Insulated Cavity)	*R* (Studs, Plates, and Headers)
1. Outside surface, 24 km/h wind	0.03	0.03
2. Wood bevel lapped siding	0.14	0.14
3. Rigid foam insulating sheathing	0.70	0.70
4. Mineral fiber batt insulation	2.30	—
5. Wood stud	—	0.63
6. Gypsum wallboard	0.08	0.08
7. Inside surface, still air	0.12	0.12
	3.37	1.70

Since the U-factor is the reciprocal of R-value, U_1 = 0.297 W/(m²·K) and U_2 = 0.588 W/(m²·K).

If the wood framing (thermal bridging) is not included, Equation (3) from Chapter 22 may be used to calculate the U-factor of the wall as follows:

$$U_{av} = U_1 = \frac{1}{R_1} = 0.30\ \text{W/(m}^2\cdot\text{K)}$$

If the wood framing is accounted for using the parallel-path flow method, the U-factor of the wall is determined using Equation (5) from Chapter 22 as follows:

$$U_{av} = (0.75 \times 0.297) + (0.25 \times 0.588) = 0.37\ \text{W/(m}^2\cdot\text{K)}$$

If the wood framing is included using the isothermal planes method, the U-factor of the wall is determined using Equations (2) and (3) from Chapter 22 as follows:

$$R_{T(av)} = 4.98 + 1/[(0.75/2.30) + (0.25/0.63)] + 0.22$$

$$= 2.47\ \text{K}\cdot\text{m}^2/\text{W}$$

$$U_{av} = 0.40\ \text{W/(m}^2\cdot\text{K)}$$

For a frame wall with a 600 mm OC stud space, the average overall R-value is 0.25 m²·K/W. Similar calculation procedures may be used to evaluate other wall designs, except those with thermal bridges.

Table 5-19 SI Design U-factors for Double-Skin Steel Emergency Exit Doors in W/(m^2·K)

(Table 8, Chapter 15, 2017 ASHRAE Handbook—Fundamentals)

Core Insulation		Rough Opening Size	
Thickness, mm	**Type**	**0.9 × 2 m**	**1.8 × 2 m**
35*	Honeycomb kraft paper	3.23	2.97
	Mineral wool, steel ribs	2.50	2.05
	Polyurethane foam	1.92	1.60
44*	Honeycomb kraft paper	3.25	3.06
	Mineral wool, steel ribs	2.30	1.90
	Polyurethane foam	1.77	1.50
35	Honeycomb kraft paper	3.38	3.11
	Mineral wool, steel ribs	2.67	2.21
	Polyurethane foam	2.10	1.77
44	Honeycomb kraft paper	3.38	3.22
	Mineral wool, steel ribs	2.47	2.08
	Polyurethane foam	1.95	1.69

*With thermal break

Chapter 6

RESIDENTIAL COOLING AND HEATING LOAD CALCULATIONS

The procedures described in this chapter are for calculating the design cooling and heating loads for residential buildings. Additional details can be found in Chapter 17 of the 2017 *ASHRAE Handbook—Fundamentals*, Chapter 1 of the 2015 *ASHRAE Handbook—HVAC Applications*, and Chapter 10 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment.*

6.1 Background

The primary function of heat loss and heat gain calculations is to estimate the capacity that will be required for the heating and cooling components of the air-conditioning systems in order to maintain comfort in the building. These calculations are therefore based on peak load conditions for heating and cooling and correspond to weather conditions that are near the extremes normally encountered (see Chapter 4). A number of load calculation procedures have been developed over the years. Most procedures, even though they differ in some respects, are based on a systematic evaluation of the components of heat loss and heat gain. The most recent procedures developed by ASHRAE, as included in Chapter 17 of the 2017 *ASHRAE Handbook—Fundamentals*, are used here.

Residences and small commercial buildings have heat gains and cooling loads that are dominated by the building envelope (walls, roof, windows, and doors) whereas internal gains from occupants, lights, equipment, and appliances play a significant role and often dominate in commercial buildings. For all buildings, the energy required to heat/cool and/or to humidify/dehumidify any outdoor air entering the space, whether intentional for ventilation or leaking in through cracks and openings (infiltration) is often significant and must be included. ASHRAE Standard 62.2-2016 should be consulted for recommended residential ventilation rates.

A key common element of all cooling methods is attention to temperature swing via empirical data or suitable models. Throughout the literature, it is repeatedly emphasized that direct application of nonresidential methods (based on a fixed setpoint) results in unrealistically high cooling loads for residential applications.

The procedures in this chapter are based on the same fundamentals as the nonresidential methods presented in Chapter 7. However, many characteristics distinguish residential loads, and the Chapter 7 procedures should be applied with care to residential applications.

With respect to heating and cooling load calculation and equipment sizing, the unique features distinguishing residences from other types of buildings are the following:

- **Smaller Internal Heat Gains.** Residential system loads are primarily imposed by heat gain or loss through structural components and by air leakage or ventilation. Internal heat gains, particularly those from occupants and lights, are small compared to those in commercial or industrial structures.
- **Varied Use of Spaces.** Use of spaces in residences is more flexible than in commercial buildings. Localized or temporary temperature excursions are often tolerable.
- **Fewer Zones.** Residences are generally conditioned as a single zone or, at most, a few zones. Typically, a thermostat located in one room controls unit output for multiple rooms, and capacity cannot be redistributed from one area to another as loads change over the day. This results in some hour-to-hour temperature variation or “swing” that has a significant moderating effect on peak loads because of heat storage in building components.
- **Greater Distribution Losses.** Residential ducts are frequently installed in attics or other unconditioned buffer spaces. Duct leakage and heat gain or loss can require significant increases in unit capacity. Residential distribution gains and losses cannot be neglected or estimated with simple rules of thumb.
- **Partial Loads.** Most residential cooling systems use units of relatively small capacity (about 12,000 to 60,000 Btu/h cooling, 40,000 to 120,000 Btu/h heating). Because loads are largely determined by outside conditions, and few days each season are design days, the unit operates at partial load during most of the season; thus, an oversized unit is detrimental to good system performance, especially for cooling in areas of high wet-bulb temperature.
- **Dehumidification Issues.** Dehumidification occurs during cooling unit operation only, and space condition control is usually limited to use of room thermostats (sensible heat-actuated devices). Excessive sensible capacity results in short-cycling and severely degraded dehumidification performance.

In addition to these general features, residential buildings can be categorized according to their exposure:

- **Single-Family Detached.** A house in this category usually has exposed walls in four directions, often more than one story, and a roof. The cooling system is a single-zone, unitary system with a single thermostat. Two-story houses

may have a separate cooling system for each floor. Rooms are reasonably open and generally have a centralized air return. In this configuration, both air and load from rooms are mixed, and a load-leveling effect, which requires a distribution of air to each room that is different from a pure commercial system, results. Because the amount of air supplied to each room is based on the load for that room, proper load calculation procedures must be used.

- **Multifamily.** Unlike single-family detached units, multifamily units generally do not have exposed surfaces facing in all directions. Rather, each unit typically has a maximum of three exposed walls and possibly a roof. Both east and west walls might not be exposed in a given living unit. Each living unit has a single unitary cooling system or a single fan-coil unit and the rooms are relatively open to one another. This configuration does not have the same load-leveling effect as a single-family detached house.
- **Other.** Many buildings do not fall into either of the preceding categories. Critical to the designation of a single-family detached building is well-distributed exposure so there is not a short-duration peak; however, if fenestration exposure is predominantly east or west, the cooling load profile resembles that of a multifamily unit. On the other hand, multifamily units with both east and west exposures or neither east nor west exposure exhibit load profiles similar to single-family detached.

Variations in the characteristics of residences can lead to surprisingly complex load calculations. Time-varying heat flows combine to produce a time-varying load. The relative magnitude and pattern of the heat flows depends on the building characteristics and exposure, resulting in a building-specific load profile. In general, an hour-by-hour analysis is required to determine that profile and find its peak.

In theory, cooling and heating processes are identical; a common analysis procedure should apply to either. Acceptable simplifications are possible for heating; however, for cooling, different approaches are used.

Heating calculations use simple worst-case assumptions: no solar or internal gains and no heat storage (with all heat losses evaluated instantaneously). With these simplifications, the heating problem is reduced to a basic $UA\Delta t$ calculation. The heating procedures in this chapter use this long-accepted approach and thus differ only in details from prior methods put forth by ASHRAE and others.

In contrast, the cooling procedures in this edition are extensively revised, based on the results of ASHRAE research project RP-1199 and supported by the Air-Conditioning Contractors of America (ACCA).

A key common element of all cooling methods is attention to temperature swing via empirical data or suitable models. Throughout the literature, it is repeatedly emphasized that direct application of nonresidential methods (based on a fixed setpoint) results in unrealistically high cooling loads for residential applications. The procedure presented in this chapter is the Residential Load Factor (RLF) method. RLF is a simplified procedure derived from a detailed residential heat balance (RHB) analysis of prototypical buildings across a range of climates. The method is tractable by hand but is best applied using a spreadsheet. The RLF method should not be applied to situations outside the range of underlying cases, as shown in Table 6-1.

6.2 General Guidelines

6.2.1 Overview

The following guidelines, data requirements, and procedures apply to all load calculation approaches, whether heating or cooling, hand-tractable or computerized.

Table 6-1 RLF Limitations

(Table 1, Chapter 17, 2017 ASHRAE Handbook—Fundamentals)

Item	Valid Range	Notes
Latitude	20°N to 60°N	Also approximately valid for 20°S to 60°S with N and S orientations reversed for southern hemisphere.
Date	July 21	Application must be summer peaking. Buildings in mild climates with significant SE/S/SW glazing may experience maximum cooling load in fall or even winter. Use RHB if local experience indicates this is a possibility.
Elevation	Less than 6500 ft	RLF factors assume 164 ft elevation. With elevation-corrected C_s, method is acceptably accurate except at very high elevations.
Climate	Warm/hot	Design-day average outdoor temperature assumed to be above indoor design temperature.
Construction	Lightweight residential construction (wood or metal framing, wood or stucco siding)	May be applied to masonry veneer over frame construction; results are conservative. Use RHB for structural masonry or unconventional construction.
Fenestration area	0% to 15% of floor area on any façade, 0% to 30% of floor area total	Spaces with high fenestration fraction should be analyzed with RHB.
Fenestration tilt	Vertical or horizontal	Skylights with tilt less than 30° can be treated as horizontal. Buildings with significant sloped glazing areas should be analyzed with RHB.
Occupancy	Residential	Applications with high internal gains and/or high occupant density should be analyzed with RHB or nonresidential procedures.
Temperature swing	3°F	
Distribution losses	Typical	Applications with extensive duct runs in unconditioned spaces should be analyzed with RHB.

Design for Typical Building Use. In general, residential systems should be designed to meet representative maximum-load conditions, not extreme conditions. Normal occupancy should be assumed, not the maximum that might occur during an occasional social function. Intermittently operated ventilation fans should be assumed to be off. These considerations are especially important for cooling-system sizing.

Building Codes and Standards. This chapter presentation is necessarily general. Codes and regulations take precedence; consult local authorities to determine applicable requirements.

Designer Judgment. Designer experience with local conditions, building practices, and prior projects should be considered when applying the procedures in this chapter. For equipment-replacement projects, occupant knowledge concerning performance of the existing system can often provide useful guidance in achieving a successful design.

Verification. Postconstruction commissioning and verification are important steps in achieving design performance. Designers should encourage pressurization testing and other procedures that allow identification and repair of construction shortcomings.

Uncertainty and Safety Allowances. Residential load calculations are inherently approximate. Many building characteristics are estimated during design and ultimately determined by construction quality and occupant behavior. These uncertainties apply to all calculation methods, including first-principles procedures such as RHB. It is therefore tempting to include safety allowances for each aspect of a calculation. However, this practice has a compounding effect and often produces oversized results. Typical conditions should be assumed; safety allowances, if applied at all, should be added to the final calculated loads rather than to intermediate components. In addition, temperature swing provides a built-in safety factor for sensible cooling: a 20% capacity shortfall typically results in a temperature excursion of at most about one or two degrees.

6.2.2 Basic Relationships

Common air-conditioning processes involve transferring heat via air transport or leakage. The sensible. latent, and total heat conveyed by air on a volumetric basis is:

$$q_s = C_s Q \Delta t \qquad (6\text{-}1)$$

$$q_l = C_l Q \Delta W \qquad (6\text{-}2)$$

$$q_t = C_t Q \Delta h \qquad (6\text{-}3)$$

$$q_t = q_s + q_l \qquad (6\text{-}4)$$

where

q_s, q_l, q_t = sensible, latent, total heat transfer rates, Btu/h
C_s = air sensible heat factor, Btu/ h·°F·cfm (1.1 at sea level)
C_l = air latent heat factor, Btu/h·cfm (4840 at sea level)
C_t = air total heat factor, Btu/h·cfm per Btu/lb enthalpy, h (4.5 at sea level)
Q = air volumetric flow rate, cfm
Δt = air temperature difference across process, °F
ΔW = air humidity ratio difference across process, lb/lb
Δh = air enthalpy difference across process, Btu/lb

The heat factors C_s, C_l, and C_t are elevation dependent. The sea level values given in the preceding definitions are appropriate for elevations up to about 1000 ft. Procedures are provided in Chapter 7 for calculating adjusted values for higher elevations.

6.2.3 Design Conditions

The initial step in the load calculation is selecting indoor and outdoor design conditions.

Indoor Conditions. Indoor conditions assumed for design purposes depend on building use, type of occupancy, and/or code requirements. Chapter 4 and ASHRAE Standard 55 define the relationship between indoor conditions and comfort.

Typical practice for cooling is to design for indoor conditions of 75°F db and a maximum of 50% to 65% rh. For heating, 68°F to 72°F db and 30% rh are common design values.

Outdoor Conditions. Outdoor design conditions for load calculations should be selected from location-specific climate data in Chapter 4 or according to local code requirements as applicable.

Cooling. The 1% values from Figure 4-4 or Table 4-7 climate data are generally appropriate. As previously emphasized, oversized cooling equipment results in poor system performance. Extremely hot events are necessarily of short duration (conditions always moderate each night); therefore, sacrificing comfort under typical conditions to meet occasional extremes is not recommended.

Load calculations also require the daily range of the dry-bulb and coincident wet-bulb temperatures and wind speed. These values can also be found in Figure 4-4 or Table 4-7, although wind speed is commonly assumed to be 7.5 mph.

Typical buildings in middle latitudes generally experience maximum cooling requirements in midsummer (July in the northern hemisphere and January in the southern hemisphere). For this reason, the RLF method is based on midsummer solar gains. However, this pattern does not always hold. Buildings at low latitudes or with significant south-facing glazing (north-facing in the southern hemisphere) should be analyzed at several times of the year using the RHB method. Local experience can provide guidance as to when maximum cooling is probable. For example, it is common for south-facing buildings in mild northern-hemisphere climates to have peak cooling loads in the fall because of low sun angles.

Heating. General practice is to use the 99% condition from Figure 4-4 or Table 4-7. Heating load calculations ignore solar and internal gains, providing a built-in safety factor. However, the designer should consider two additional factors:

- Many locations experience protracted (several-day) cold periods during which the outdoor temperature remains below the 99% value.
- Wind is a major determinant of infiltration. Residences with significant leakage (e.g., older houses) may have peak heating demand under conditions other than extreme cold, depending on site wind patterns.

Depending on the application and system type, the designer should consider using the 99.6% value or the mean minimum extreme as the heating design temperature. Alternatively, the heating load can be calculated at the 99% condition and a safety factor applied when equipment is selected. This additional capacity can also serve to meet pickup loads under non-extreme conditions.

Adjacent Buffer Spaces. Residential buildings often include unconditioned buffer spaces, such as garages, attics, crawlspaces, basements, or enclosed porches. Accurate load calculations require the adjacent air temperature. In many cases, a simple, conservative estimate is adequate, especially for heating calculations. For example, it is generally reasonable to assume that, under heating design conditions, adjacent uninsulated garages, porches, and attics are at outdoor temperature. Another reasonable assumption is that the temperature in an adjacent, unheated, *insulated* room is the mean of the indoor and outdoor temperatures.

6.2.4 Building Data

Component Areas. To perform load calculations efficiently and reliably, standard methods must be used for determining building surface areas. For fenestration, the definition of component area must be consistent with associated ratings.

Gross area. It is both efficient and conservative to derive gross surface areas from outside building dimensions, ignoring wall and floor thicknesses. Thus, floor areas should be measured to the outside of adjacent exterior walls or to the centerline of adjacent partitions. When apportioning to rooms, facade area should be divided at partition centerlines. Wall height should be taken as floor-to-floor.

Using outside dimensions avoids separate accounting of floor edge and wall comer conditions. Further, it is standard practice in residential construction to define floor area in terms of outside dimensions, so outside-dimension takeoffs yield areas that can be readily checked against building plans (e.g., the sum of room areas should equal the plan floor area). Although outside-dimension procedures are recommended as expedient for load calculations, they are not consistent with rigorous definitions used in building-related standards. However, the inconsistencies are not significant in the load calculation context.

Fenestration area. Fenestration includes exterior windows, skylights, and doors. Fenestration U-factor and SHGC ratings (discussed below) are based on the entire product area, including frames. Thus, for load calculations, fenestration area is the area of the rough opening in the wall or roof, less installation clearances (projected product area A_{pf}). Installation clearances can be neglected; it is acceptable to use the rough opening as an approximation of A_{pf}

Net area. Net surface area is the gross surface area less fenestration area (rough opening or A_{pf} contained within the surface).

Volume. Building volume is expediently calculated by multiplying floor area by floor-to-floor height. This produces a conservative estimate of enclosed air volume because wall and floor volumes are included in the total. More precise calculations are possible but are generally not justified in this context.

Construction Characteristics.

U-factors. Except for fenestration, construction U-factors should be calculated using procedures in Chapter 5 or taken from manufacturer's. data, if available. U-factors should be evaluated under heating (winter) conditions.

Fenestration. Fenestration is characterized by U-factor and solar heat gain coefficient (SHGC), which apply to the entire assembly (including frames). If available, rated values should be used. Ratings can be obtained from product literature, product label, or published listings. For unrated products (e.g., in existing construction), the U-factor and SHGC can be estimated using Table 6-2. Fenestration U-factors are evaluated under heating (winter) design conditions but are used for both heating and cooling calculations.

Relatively few types of glazing are encountered in residential applications. Single-glazed clear, double-glazed clear, and double-glazed low-emissivity (“low-e”) glass predominate. Single glazing is now rare in new construction but common in older homes. Triple-glazing, reflective glass, and heat-absorbing glass are encountered occasionally. The properties of windows equipped with storm windows should be estimated from data for a similar configuration with an additional pane. For example, data for clear, double glazing should be used for a clear single-glazed window with a storm window. Fenestration interior and exterior shading must be included in cooling load calculations.

6.2.5 Load Components

Below-Grade Surfaces. For cooling calculations, heat flow into the ground is usually ignored because it is difficult to quantify. Surfaces adjacent to the ground are modeled as if well insulated on the outside, so there is no overall heat transfer, but diurnal heat storage effects are included. Heating calculations must include loss via slabs and basement walls and floors, as discussed in the heating load section.

Infiltration. Infiltration is generally a significant component of both cooling and heating loads. Refer to Chapter 5 for a discussion of residential air leakage. The simplified residential models found in that chapter can be used to calculate infiltration rates for load calculations. Infiltration should be evaluated for the entire building, not individual rooms or zones. Natural infiltration leakage rates are modified by mechanical pressurization caused by unbalanced ventilation or duct leakage.

Table 6-2 Typical Fenestration Characteristics

(Table 2, Chapter 17, 2017 ASHRAE Handbook—Fundamentals)

					Frame									
					Operable					Fixed				
Glazing Type	**Glazing Layers**	**ID**[b]	**Property**[c,d]	**Center of Glazing**	**Aluminum**	**Aluminum with Thermal Break**	**Reinforced Vinyl/Aluminum Clad Wood**	**Wood/Vinyl**	**Insulated Fiberglass/Vinyl**	**Aluminum**	**Aluminum with Thermal Break**	**Reinforced Vinyl/Aluminum Clad Wood**	**Wood/Vinyl**	**Insulated Fiberglass/Vinyl**
Clear	1	1a	U	1.04	1.27	1.08	0.90	0.89	0.81	1.13	1.07	0.98	0.98	0.94
			SHGC	0.86	0.75	0.75	0.64	0.64	0.64	0.78	0.78	0.75	0.75	0.75
	2	5a	U	0.48	0.81	0.60	0.53	0.51	0.44	0.64	0.57	0.50	0.50	0.48
			SHGC	0.76	0.67	0.67	0.57	0.57	0.57	0.69	0.69	0.67	0.67	0.67
	3	29a	U	0.31	0.67	0.46	0.40	0.39	0.34	0.49	0.42	0.36	0.35	0.34
			SHGC	0.68	0.60	0.60	0.51	0.51	0.51	0.62	0.62	0.60	0.60	0.60
Low-e, low-solar	2	25a	U	0.30	0.67	0.47	0.41	0.39	0.33	0.48	0.41	0.36	0.35	0.33
			SHGC	0.41	0.37	0.37	0.31	0.31	0.31	0.38	0.38	0.36	0.36	0.36
	3	40c	U	0.27	0.64	0.43	0.37	0.36	0.31	0.45	0.39	0.33	0.32	0.31
			SHGC	0.27	0.25	0.25	0.21	0.21	0.21	0.25	0.25	0.24	0.24	0.24
Low-e, high-solar	2	17c	U	0.35	0.71	0.51	0.44	0.42	0.36	0.53	0.46	0.40	0.39	0.37
			SHGC	0.70	0.62	0.62	0.52	0.52	0.52	0.64	0.64	0.61	0.61	0.61
	3	32c	U	0.33	0.69	0.47	0.41	0.40	0.35	0.50	0.44	0.38	0.37	0.36
			SHGC	0.62	0.55	0.55	0.46	0.46	0.46	0.56	0.56	0.54	0.54	0.54
Heat-absorbing	1	1c	U	1.04	1.27	1.08	0.90	0.89	0.81	1.13	1.07	0.98	0.98	0.94
			SHGC	0.73	0.64	0.64	0.54	0.54	0.54	0.66	0.66	0.64	0.64	0.64
	2	5c	U	0.48	0.81	0.60	0.53	0.51	0.44	0.64	0.57	0.50	0.50	0.48
			SHGC	0.62	0.55	0.55	0.46	0.46	0.46	0.56	0.56	0.54	0.54	0.54
	3	29c	U	0.31	0.67	0.46	0.40	0.39	0.34	0.49	0.42	0.36	0.35	0.34
			SHGC	0.34	0.31	0.31	0.26	0.26	0.26	0.31	0.31	0.30	0.30	0.30
Reflective	1	1l	U	1.04	1.27	1.08	0.90	0.89	0.81	1.13	1.07	0.98	0.98	0.94
			SHGC	0.31	0.28	0.28	0.24	0.24	0.24	0.29	0.29	0.27	0.27	0.27
	2	5p	U	0.48	0.81	0.60	0.53	0.51	0.44	0.64	0.57	0.50	0.50	0.48
			SHGC	0.29	0.27	0.27	0.22	0.22	0.22	0.27	0.27	0.26	0.26	0.26
	3	29c	U	0.31	0.67	0.46	0.40	0.39	0.34	0.49	0.42	0.36	0.35	0.34
			SHGC	0.34	0.31	0.31	0.26	0.26	0.26	0.31	0.31	0.30	0.30	0.30

[b] ID = Chapter 15 in the 2013 *ASHRAE Handbook—Fundamentals* glazing type identifier. [c] U = U-factor, Btu/h·ft^2·°F [d] SHGC = solar heat gain coefficient

Leakage rate. Air leakage rates are specified either as airflow rate Q or air exchanges per hour (ACH), related as follows:

$$Q_i = \text{ACH}\ (V/60) \qquad (6\text{-}5)$$

$$\text{ACH} = \frac{60Q_i}{V} \qquad (6\text{-}6)$$

where

Q_i = infiltration airflow rate, cfm

ACH = air exchange rate, ach

V = building volume, ft^3

The infiltration airflow rate depends on two factors:

- The building effective leakage area (envelope leaks plus other air leakage paths, notably flues) and its distribution among ceilings, walls, floors, and flues.
- The driving pressure caused by buoyancy (stack effect) and wind.

These factors can be evaluated separately and combined using Equation 6-7.

$$Q_i = A_L\text{IDF} \qquad (6\text{-}7)$$

where

A_L = building effective leakage area (including flue) at reference pressure difference = 0.016 in. of water, assuming discharge coefficient CD = 1, in^2

IDF = infiltration driving force, cfm/in^2

The following sections provide procedures for determining A_L and IDF.

Leakage area. There are several ways to characterize building leakage, depending on reference pressure differences and assumed discharge coefficient. The only accurate procedure for determining A_L is by measurement using a pressurization test (commonly called a blower door test). For buildings in design, a pressurization test is not possible and leakage area must be assumed for design purposes. Leakage can be estimated using a simple approach based on an assumed average leakage per unit of building surface area:

Table 6-3 Unit Leakage Areas

(Table 3, Chapter 17, 2017 ASHRAE Handbook—Fundamentals)

Construction	Description	A_{ul} (in.^{2}/ft^2)
Tight	Construction supervised by air-sealing specialist	0.01
Good	Carefully sealed construction by knowledgeable builder	0.02
Average	Typical current production housing	0.04
Leaky	Typical pre-1970 houses	0.08
Very leaky	Old houses in original condition	0.15

$$A_L = A_{es} A_{ul} \tag{6-8}$$

where

A_{es} = building exposed surface area, ft^2

A_{ul} = unit leakage area, in^2/ft^2 (from Table 6-3)

A_{u1} is the leakage area per unit surface area; suitable design values are found in Table 6-3.

In Equation 6-8, A_{es} is the total building surface area at the envelope pressure boundary, defined as all above-grade surface area that separates the outdoors from conditioned or semiconditioned space. Table 6-4 provides guidance for evaluating A_{es}.

IDF. To determine IDF, Barnaby and Spitler (2005) derived the following relationship:

$$\text{IDF} = \frac{I_0 + H|\Delta t|[I_1 + I_2(A_{L,\text{flue}}/A_L)]}{1000} \tag{6-9}$$

where

	Cooling 7.5 mph	Heating 15 mph
I_0	343	698
I_1	0.88	0.81
I_2	0.28	0.53

H = building average stack height, ft (typically 8 to 10 ft per story)

Δt = difference between indoor and outdoor temperatures, °F

$A_{L,\text{flue}}$ = flue effective leakage area at reference pressure difference = 0.016 in. of water, assuming CD = 1, in^2 (total for flues serving furnaces, domestic water heaters, fireplaces, or other vented equipment, evaluated assuming associated equipment is not operating and with dampers in closed position)

The building stack height H is the average height difference between the ceiling and floor (or grade, if the floor is below grade). Thus, for buildings with vented crawlspaces, the crawlspace height is not included. For basement or slab-on-grade construction, H is the average height of the ceiling above grade. Generally, there is significant leakage between basements and spaces above, so above-grade basement height should be included whether or not the basement is fully conditioned. With suitable adjustments for grade level, H can also be estimated as V/A_{cf} (conditioned floor area).

Table 6-5 shows *IDF* values, assuming $A_{L,\text{flue}} = 0$.

Table 6-4 Evaluation of Exposed Surface Area

(Table 4, Chapter 17, 2017 ASHRAE Handbook—Fundamentals)

Situation	Include	Exclude
Ceiling/roof combination (e.g., cathedral ceiling without attic)	Gross surface area	
Ceiling or wall adjacent to attic	Ceiling or wall area	Roof area
Wall exposed to ambient	Gross wall area (including fenestration area)	
Wall adjacent to unconditioned buffer space (e.g., garage or porch)	Common wall area	Exterior wall area
Floor over open or vented crawlspace	Floor area	Crawlspace wall area
Floor over sealed crawlspace	Crawlspace wall area	Floor area
Floor over conditioned or semiconditioned basement	Above-grade basement wall area	Floor area
Slab floor		Slab area

Table 6-5 Typical IDF Values, cfm/in.^{2}

(Table 5, Chapter 17, 2017 ASHRAE Handbook—Fundamentals)

	Heating Design Temperature, °F					Cooling Design Temperature, °F		
H, ft	–40	–20	0	20	40	85	95	105
8	1.40	1.27	1.14	1.01	0.88	0.41	0.48	0.55
10	1.57	1.41	1.25	1.09	0.92	0.43	0.52	0.61
12	1.75	1.55	1.36	1.16	0.97	0.45	0.55	0.66
14	1.92	1.70	1.47	1.24	1.02	0.47	0.59	0.71
16	2.10	1.84	1.58	1.32	1.06	0.48	0.62	0.76
18	2.27	1.98	1.69	1.40	1.11	0.50	0.66	0.82
20	2.45	2.12	1.80	1.48	1.15	0.52	0.69	0.87
22	2.62	2.27	1.91	1.55	1.20	0.54	0.73	0.92
24	2.80	2.41	2.02	1.63	1.24	0.55	0.76	0.98

Ventilation.

Whole-building ventilation. Because of energy efficiency concerns, residential construction has become significantly tighter over the last several decades. Natural leakage rates are often insufficient to maintain acceptable indoor air quality. ASHRAE Standard 62.2-2016 specifies the required minimum whole-building ventilation rate as

$$Q_v = 0.01A_{cf} + 7.5\ (N_{br} + 1) \tag{6-10}$$

where

Q_v = required ventilation flow rate, cfm

A_{cf} = building conditioned floor area, ft^2

N_{br} = number of bedrooms (not less than 1)

Certain mild climates are exempted from this standard; local building authorities ultimately dictate actual requirements. It is expected that whole-building ventilation will become more common because of a combination of regulation and consumer demand. The load effect of Q_v must be included in both cooling and heating calculations.

Distribution Losses. Air leakage and heat losses from duct systems frequently impose substantial equipment loads in excess of building requirements. The magnitude of losses depends on the location of duct runs, their surface areas, surrounding temperatures, duct wall insulation, and duct airtightness. These values are usually difficult to accurately

determine at the time of preconstruction load calculations and must be estimated using assumed values so that selected equipment capacity is sufficient.

6.3 Cooling Load Methodology

A cooling load calculation determines total sensible cooling load from heat gain (1) through opaque surfaces (walls, floors, ceilings, and doors), (2) through transparent fenestration surfaces (windows, skylights, and glazed doors), (3) caused by infiltration and ventilation, and (4) because of occupancy. The latent portion of the cooling load is evaluated separately. Although the entire structure may be considered a single zone, equipment selection and system design should be based on room-by-room calculations. For proper design of the distribution system, the conditioned airflow required by each room must be known.

6.3.1 Peak Load Computation

To select a properly sized cooling unit, the peak or maximum load (block load) for each zone must be computed. The block load for a single-family detached house with one central system is the sum of all the room loads. If the house has a separate system for each zone, each zone block load is required. When a house is zoned with one central cooling system, the system size is based on the entire house block load, whereas zone components, such as distribution ducts, are sized using zone block loads.

In multifamily structures, each living unit has a zone load that equals the sum of the room loads. For apartments with separate systems, the block load for each unit establishes the system size. Apartment buildings having a central cooling system with fan-coils in each apartment require a block load calculation for the complete structure to size the central system; each unit load establishes the size of the fan-coil and air distribution system for each apartment.

6.3.2 Opaque Surfaces

Heat gain through walls, floors, ceilings, and doors is caused by (1) the air temperature difference across such surfaces and (2) solar gains incident on the surfaces. The heat capacity of typical construction moderates and delays building heat gain. This effect is modeled in detail in the computerized RHB method, resulting in accurate simultaneous load estimates. The RLF method uses the following to estimate cooling load:

$$q_{\text{opq}} = A \times \text{CF}_{\text{opq}} \tag{6-11}$$

$$\text{CF}_{\text{opq}} = U(\text{OF}_t \Delta t + \text{OF}_b + \text{OF}_r \text{DR}) \tag{6-12}$$

where

q_{opq} = opaque surface cooling load, Btu/h
A = net surface area, ft^2
CF = surface cooling factor, Btu/h·ft^2
U = construction U-factor, Btu/h·ft^2·°F
Δt = cooling design temperature difference, °F
OF_t, OF_b, OF_r = opaque-surface cooling factors (Table 6-6)
DR = cooling daily range, °F

Table 6-6 Opaque Surface Cooling Factor Coefficients

(Table 7, Chapter 17, 2017 ASHRAE Handbook—Fundamentals)

Surface Type	OF_t	OF_b, °F	OF_r
Ceiling or wall adjacent to vented attic	0.62	$25.7\alpha_{\text{roof}} - 8.1$	–0.19
Ceiling/roof assembly	1	$68.9\alpha_{\text{roof}} - 12.6$	–0.36
Wall (wood frame) or door with solar exposure	1	14.8	–0.36
Wall (wood frame) or door (shaded)	1	0	–0.36
Floor over ambient	1	0	–0.06
Floor over crawlspace	0.33	0	–0.28
Slab floor (see Slab Floor section)			

α_{roof} = roof solar absorptance (see Table 6-7)

Table 6-7 Roof Solar Absorptance α_{roof}

(Table 8, Chapter 17, 2017 ASHRAE Handbook—Fundamentals)

	Color			
Material	**White**	**Light**	**Medium**	**Dark**
Asphalt shingles	0.75	0.75	0.85	0.92
Tile	0.30	0.40	0.80	0.80
Metal	0.35	0.50	0.70	0.90
Elastomeric coating	0.30			

Source: Summarized from Parker et al. 2000

OF factors, found in Table 6-6, represent construction-specific physical characteristics. OF values less than 1 capture the buffering effect of attics and crawlspaces, OF_b represents incident solar gain, and OF_r captures heat storage effects by reducing the effective temperature difference. Note also that CF can be viewed as CF = $U \times$ CLTD, the formulation used in prior residential and nonresidential methods.

As shown in Table 6-6, roof solar absorptance has a significant effect on ceiling cooling load contribution. Table 6-7 shows typical values for solar absorptance of residential roofing materials.

6.3.3 Slab Floors

Slab floors produce a slight reduction in cooling load, as follows:

$$q_{\text{apq}} = A \times \text{CF}_{\text{slab}} \tag{6-13}$$

$$\text{CF}_{\text{slab}} = 0.51 - 2.5 h_{\text{srf}} \tag{6-14}$$

where

A = area of slab, ft^2
CF_{slab} = slab cooling factor, Btu/h·ft^2
h_{srf} = effective surface conductance, including resistance of slab covering material such as carpet = $1/(R_{\text{cvr}} + 0.68)$, Btu/h·ft^2·°F. Representative R_{cvr} values are found in Table 6-8.

6.3.4 Transparent Fenestration Surfaces

Cooling load associated with nondoor fenestration is calculated as follows:

$$q_{\text{fen}} = A \times \text{CF}_{\text{fen}} \tag{6-15}$$

$$\text{CF}_{\text{fen}} = U(\Delta t - 0.46\text{DR}) + \text{PXI} \times \text{SHGC} \times \text{IAC} \times \text{FF}_s \tag{6-16}$$

Table 6-8 Thermal Resistance of Floor Coverings

Description	Thermal Resistance r_c, $ft^2 \cdot h \cdot °F/Btu$
Bare concrete, no covering	0
Asphalt tile	0.05
Rubber tile	0.05
Light carpet	0.60
Light carpet with rubber pad	1.00
Light carpet with light pad	1.40
Light carpet with heavy pad	1.70
Heavy carpet	0.80
Heavy carpet with rubber pad	1.20
Heavy carpet with light pad	1.60
Heavy carpet with heavy pad	1.90
3/8 in.hardwood	0.54
5/8 in. wood floor (oak)	0.57
1/2 in. oak parquet and pad	0.68
Linoleum	0.12
Marble floor and mudset	0.18
Rubber pad	0.62
Prime urethane underlayment, 3/8 in.	1.61
48 oz. waffled sponge rubber	0.78
Bonded urethane, 1/2 in.	2.09

Notes:
1. Carpet pad thickness should not be more than 1/4 in.
2. Total thermal resistance of carpet is more a function of thickness than of fiber type.
3. Generally, thermal resistance (R-value) is approximately 2.6 times the total carpet thickness in inches.
4. Before carpet is installed, verify that the backing is resistant to long periods of continuous heat up to 120°F

Table 6-9 Peak Irradiance, $Btu/h{\cdot}ft^2$

(Table 10, Chapter 17, 2017 ASHRAE Handbook—Fundamentals)

		Latitude								
Exposure		**20°**	**25°**	**30°**	**35°**	**40°**	**45°**	**50°**	**55°**	**60°**
North	E_D	38	34	31	31	33	37	44	53	65
	E_d	35	31	27	24	21	18	15	13	11
	E_t	73	64	58	55	53	55	59	66	75
Northeast/Northwest	E_D	140	140	140	141	142	143	145	147	150
	E_d	50	47	44	42	39	37	35	33	32
	E_t	190	187	184	182	181	180	180	181	182
East/West	E_D	158	166	174	181	187	192	198	202	206
	E_d	59	56	54	52	50	49	47	46	45
	E_t	217	223	228	233	237	241	245	248	251
Southeast/Southwest	E_D	76	95	112	129	144	158	171	183	193
	E_d	63	61	59	58	56	55	54	53	52
	E_t	139	156	172	186	200	213	225	236	245
South	E_D	0	6	36	64	90	115	139	161	181
	E_d	39	62	61	60	59	58	57	56	56
	E_t	39	69	97	123	149	173	196	217	237
Horizontal	E_D	247	250	250	248	243	234	223	210	193
	E_d	54	54	54	54	54	54	54	54	54
	E_t	300	304	304	302	297	288	277	263	247

where

q_{fen} = fenestration cooling load, Btu/h
A = fenestration area (including frame), ft^2
CF_{fen} = surface cooling factor, Btu/h·ft
U = fenestration NFRC heating U-factor, $Btu/h{\cdot}ft^2{\cdot}°F$ (Table 6-2)
Δt = cooling design temperature difference, °F
PXI = peak exterior irradiance, including shading modifications, $Btu/h{\cdot}ft^2$ [see Equations 6-17 and 6-18]

Table 6-10 Peak Irradiance Equations

(Table 9, Chapter 17, 2017 ASHRAE Handbook—Fundamentals)

Horizontal surfaces

$$E_t = 258.7 + 3.233L - 0.0572L^2$$

$$E_d = \min(E_t, 53.9)$$

$$E_D = E_t - E_d$$

Vertical surfaces

$$\phi = \left|\frac{\psi}{180}\right| \text{(normalized exposure, 0 – 1)}$$

$$E_t = 134.2 + 393\phi - 1630\phi^3 + 1011\phi^4 + 11.17\phi L + 0.0452\phi L^2 - 4.086L - 0.1956L^2 + [0.2473L^2/(\phi + 1)]$$

$$E_d = \min 12.9 - 11.96\phi^2 + 0.0958\phi L - \frac{36.74\sqrt[4]{L}}{\phi + 1}$$

$$E_D = E_t - E_d$$

where

E_t, E_d, E_D = peak hourly total, diffuse, and direct irradiance, $Btu/h{\cdot}ft^2$
L = site latitude, °N
ψ = exposure (surface azimuth), ° from south (–180 to +180)

Table 6-11 Exterior Attachment Transmission

Attachment	T_x
None	1.0
Exterior insect screen	0.64 (See Chapter 15, Table 14, 2017 *ASHRAE Handbook—Fundamentals*)
Shade screen	Manufacturer shading coefficient (SC) value, typically 0.4 to 0.6

Note: See Brunger et al. (1999) regarding insect screens

Table 6-12 Shade Line Factors (SLFs)

	Latitude								
Exposure	**20°**	**25°**	**30°**	**35°**	**40°**	**45°**	**50°**	**55°**	**60°**
North	2.8	2.1	1.4	1.5	1.7	1.0	0.8	0.9	0.8
Northeast/Northwest	1.4	1.5	1.6	1.2	1.3	1.3	0.9	0.9	0.8
East/West	1.2	1.2	1.1	1.1	1.1	1.0	1.0	0.9	0.8
Southeast/Southwest	2.1	1.8	2.0	1.7	1.5	1.6	1.4	1.2	1.1
South	20.0	14.0	6.9	4.7	3.3	2.7	2.1	1.7	1.4

Note: Shadow length below overhang = SLF × D_{oh}

SHGC= fenestration rated or estimated NFRC solar heat gain coefficient (Table 6-2)
IAC = interior shading attenuation coefficient
FF_s = fenestration solar load factor, Table 6- 13

Although solar gain occurs throughout the day, the cooling load contribution of fenestration correlates well with the peak hour irradiance incident on the fenestration exterior. PXI is calculated as follows:

$$\text{PXI} = T_x E_t \text{ (unshaded fenestration)} \quad (6\text{-}17)$$

$$\text{PXI} = T_x[E_d + (1 - F_{shd})E_D] \text{ (shaded fenestration)} \quad (6\text{-}18)$$

where

PXI = peak exterior irradiance, $Btu/h{\cdot}ft^2$
E_t, E_d, E_D = peak total, diffuse, and direct irradiance (Table 6-9 or 6-10), $Btu/h{\cdot}ft^2$
T_x = transmission of exterior attachment (insect screen or shade screen) (Table 6-11)

F_{shd} = fraction of fenestration shaded by permanent overhangs, fins, or environmental obstacles (Equation 6-19)

For horizontal or vertical surfaces, peak irradiance values can be obtained from Table 6-9 for primary exposures or from Table 6-10 equations for any exposure. Skylights with slope less than 30° from horizontal should be treated as horizontal.

Common window coverings can significantly reduce fenestration solar gain. Table 6-11 shows transmission values for typical attachments.

The shaded fraction F_{shd} can be taken as 1 for any fenestration shaded by adjacent structures during peak hours. Simple overhang shading can be estimated using the following:

$$F_{\mathrm{shd}} = \min\left[1, \max\left(0, \frac{\mathrm{SLF} \times D_{\mathrm{oh}} - X_{\mathrm{oh}}}{h}\right)\right] \qquad (6\text{-}19)$$

where

SLF = shade line factor from Table 6-12

D_{oh} = depth of overhang (from plane of fenestration), ft

X_{oh} = vertical distance from top of fenestration to overhang, ft

h = height of fenestration, ft

The shade line factor (SLF) is the ratio of the vertical distance a shadow falls beneath the edge of an overhang to the depth of the overhang, so the shade line equals the SLF times the overhang depth. Table 6-12 shows SLFs for July 21 averaged over the hours of greatest solar intensity on each exposure.

Fenestration solar load factors FF_S depend on fenestration exposure and are found in Table 6-13. The values represent the fraction of transmitted solar gain that contributes to peak cooling load. It is thus understandable that morning (east) values are lower than afternoon (west) values. Higher values are included for multifamily buildings with limited exposure.

Interior shading significantly reduces solar gain and is ubiquitous in residential buildings. Field studies have shown that a large fraction of windows feature some sort of shading. Therefore, in all but special circumstances, interior shading should be assumed when calculating cooling loads. In the RLF method, the interior attenuation coefficient (IAC) model is used, as described in Chapter 15, 2013 *Fundamentals*. Residential values from that chapter are consolidated in Table 6-14.

In some cases, it is reasonable to assume that a shade is partially open. For example, drapes are often partially open to admit daylight. IAC values are computed as follows:

$$\mathrm{IAC} = 1 + F_{\mathrm{cl}}(\mathrm{IAC}_{\mathrm{cl}} - 1) \qquad (6\text{-}20)$$

where

IAC = interior attenuation coefficient of fenestration with partially closed shade

F_{cl} = shade fraction closed (0 to 1)

$\mathrm{IAC}_{\mathrm{cl}}$ = interior attenuation coefficient of fully closed configuration (Table 6-14)

Table 6-13 Fenestration Solar Load Factors FF_s

Exposure	Single Family Detached	Multifamily
North	0.44	0.27
Northeast	0.21	0.43
East	0.31	0.56
Southeast	0.37	0.54
South	0.47	0.53
Southwest	0.58	0.61
West	0.56	0.65
Northwest	0.46	0.57
Horizontal	0.58	0.73

6.3.5 Internal Gain

The contributions of occupants, lighting, and appliance gains to peak sensible and latent loads can be estimated as

$$q_{\mathrm{ig},s} = 464 + 0.7A_{\mathrm{cf}} + 75N_{\mathrm{oc}} \qquad (6\text{-}21)$$

$$q_{\mathrm{ig},l} = 68 + 0.07A_{\mathrm{cf}} + 41N_{\mathrm{oc}} \qquad (6\text{-}22)$$

where

$q_{\mathrm{ig},s}$ = sensible cooling load from internal gains, Btu/h

$q_{\mathrm{ig}\ 1}$ = latent cooling load from internal gains, Btu/h

A_{cf} = conditioned floor area of building, ft^2

N_{oc} = number of occupants (unknown, estimate as $N_{\mathrm{br}} + 1$)

Predicted gains are typical for US homes. Further allowances should be considered when unusual lighting intensities or other equipment are in continuous use during peak cooling hours. In critical situations where intermittent high occupant density or other internal gains are expected, a parallel cooling system should be considered. For room-by-room calculations, $g_{\mathrm{ig},s}$ should be evaluated for the entire conditioned area and allocated to kitchen and living spaces.

6.3.6 Total Latent Load

The latent cooling load is the result of three predominant moisture sources: outdoor air (infiltration and ventilation), occupants, and miscellaneous sources, such as cooking, laundry, and bathing.

These components, discussed in previous sections, combine to yield the total latent load:

$$q_l = q_{\mathrm{vi},l} + q_{\mathrm{ig},l} \qquad (6\text{-}23)$$

where

q_l = total latent load, Btu/h

$q_{\mathrm{vi},l}$ = ventilation/infiltration latent gain, Btu/h

$q_{\mathrm{ig},l}$ = internal latent gain, Btu/h

Because air conditioning systems are usually controlled by a thermostat, latent cooling is a side effect of equipment operation. During periods of significant latent gain but mild temperatures, there is little cooling operation, resulting in unacceptable indoor humidity. Multispeed equipment, combined temperature/humidity control, and dedicated dehumidification should be considered to address this condition.

6.3.7 Summary of RLF Cooling Load Equations

Table 6-15 contains a brief list of equations used in the cooling load calculation procedure described in this chapter.

6.4 Heating Load Methodology

During the coldest months, sustained periods of cold and cloudy weather with relatively small variation in outdoor conditions may occur. In this situation, heat loss from the space will be relatively constant and in the absence of internal gains will peak during the early morning hours. Therefore, for design purposes, the heat loss is usually estimated based on steady-state heat transfer for some reasonable design temperature. This simplified approach can be used to estimate a heating load for the “worst case” conditions that can reasonably be anticipated during a heating season. Traditionally this is considered as the load that must be met under design interior and exterior conditions, including infiltration and/or ventilation, but in the absence of solar effect (at night or on cloudy winter days) and before the periodic presence of people, lights, and appliances can begin to

Table 6-14 Interior Attenuation Coefficients (IAC_{cl})

		Drapes			Roller Shades			Blinds	
		Open-Weave	Closed-Weave		Opaque		Translucent		
Glazing Layers	Glazing Type (ID*)	Light	Dark	Light	Dark	White	Light	Medium	White
1	Clear (1a)	0.64	0.71	0.45	0.64	0.34	0.44	0.74	0.66
	Heat absorbing (1c)	0.68	0.72	0.50	0.67	0.40	0.49	0.76	0.69
2	Clear (5a)	0.72	0.81	0.57	0.76	0.48	0.55	0.82	0.74
	Low-e high-solar (17c)	0.76	0.86	0.64	0.82	0.57	0.62	0.86	0.79
	Low-e low-solar (25a)	0.79	0.88	0.68	0.85	0.60	0.66	0.88	0.82
	Heat absorbing (5c)	0.73	0.82	0.59	0.77	0.51	0.58	0.83	0.76

*Chapter 15 glazing identifier

Table 6-15 Summary of RLF Cooling Load Equations

Load Source	Equation	Tables and Notes
Exterior opaque surfaces	$q_{opq} = A \times CF$	
	$CF = U(OF_t \Delta t + OF_b + OF_r DR)$	OF factors from Table 6-6
Exterior transparent surfaces	$q_{fen} = A \times CF$	PXI from Table 6-9 plus adjustments
	$CF = U(\Delta t - 0.49DR) + PXI \times SHGC \times IAC \times FF_s$	FF_s from Table 6-13
Partitions to unconditioned space	$q = AU\Delta t$	Δt = temperature difference across partition
Ventilation/infiltration	$q_s = C_s Q \Delta t$	See Chapter 5
Occupants and appliances	$q_{ig,s} = 464 + 0.7A_{cf} + 75N_{oc}$	
Distribution	$q_d = F_{dl} \Sigma q$	F_{dl} from Table 6-16
Total sensible load	$q_s = q_d + \Sigma q$	
Latent load	$q_l = q_{vi,l} + q_{ig}$	
Ventilation/infiltration	$q_{vi,l} = C_l Q \Delta W$	
Internal gain	$q_{ig,l} = 68 + 0.07A_{cf} + 41N_{oc}$	

Table 6-16 Typical Duct Loss/Gain Factor

		1 Story						2 or More Stories					
	Supply/Return Leakage	11%/11%			5%/5%			11%/11%			5%/5%		
Duct Location	Insulation $ft^2 \cdot h \cdot °F/Btu$	R-0	R-4	R-8	R-0	R-4	R-8	R-0	R-4	R-8	R-0	R-4	R-8
Conditioned space		No loss ($F_{dl} = 0$)											
Attic	C	1.26	0.71	0.63	0.68	0.33	0.27	1.02	0.66	0.60	0.53	0.29	0.25
	H/F	0.49	0.29	0.25	0.34	0.16	0.13	0.41	0.26	0.24	0.27	0.14	0.12
	H/HP	0.56	0.37	0.34	0.34	0.19	0.16	0.49	0.35	0.33	0.28	0.17	0.15
Basement	C	0.12	0.09	0.09	0.07	0.05	0.04	0.11	0.09	0.09	0.06	0.04	0.04
	H/F	0.28	0.18	0.16	0.19	0.10	0.08	0.24	0.17	0.15	0.16	0.09	0.08
	H/HP	0.23	0.17	0.16	0.14	0.09	0.08	0.20	0.16	0.15	0.12	0.08	0.07
Crawlspace	C	0.16	0.12	0.11	0.10	0.06	0.05	0.14	0.12	0.11	0.08	0.06	0.05
	H/F	0.49	0.29	0.25	0.34	0.16	0.13	0.41	0.26	0.24	0.27	0.14	0.12
	H/HP	0.56	0.37	0.34	0.34	0.19	0.16	0.49	0.35	0.33	0.28	0.17	0.15

Values calculated for ASHRAE Standard 152 default duct system surface area using model of Francisco and Palmiter (1999). Values are provided as guidance only; losses can differ substantially for other conditions and configurations. Assumed surrounding temperatures:

Cooling (C): $t_o = 95°F$, $t_{attic} = 120°F$, $t_b = 68°F$, $t_{crawl} = 72°F$ Heating/furnace (H/F) and heating/heating pump (H/HP): $t_o = 32°F$, $t_{attic} = 32°F$, $t_b = 64°F$, $t_{crawl} = 32°F$

have an offsetting effect. The primary orientation is thus toward identification of adequately sized heating equipment to handle the normal worst-case condition.

Prior to designing a heating system, the engineer must estimate the maximum probable (design) heat loss of each room or space to be heated, based on maintaining a selected indoor air temperature during periods of design outdoor weather conditions. Heat losses may be divided into two groups: (1) transmission losses or heat transmitted through the confining walls, floor, ceiling, glass, or other surfaces and (2) infiltration losses or heat required to warm outdoor air that leaks in through cracks and crevices, around doors and windows, or through open doors and windows, or heat required to warm outdoor air used for ventilation.

The ideal solution is to select a system with a capacity at maximum output just equal to the heating load that develops when the most severe weather conditions for the locality occur. However, where night setback is used, some excess capacity may be needed unless the owner understands that under some conditions of operation, it may be impossible to elevate the set-back temperature.

The heating load is normally estimated for the winter design temperature, which usually occurs at night; therefore, no credit is taken for the heat given off by such internal sources as persons, lights, and equipment.

Table 6-17 contains a brief list of equations used in the heating load calculation procedures described in this chapter.

6.4.1 General Procedure

To calculate a design heating load, prepare the following information about building design and weather data at design conditions.

1. Select outdoor design weather conditions: temperature, wind direction, and wind speed.
2. Select the indoor air temperature to be maintained in each space and the humidity level of the return air, if a humidifier is to be installed, during design weather conditions.
3. Estimate temperatures in adjacent unheated spaces and, if there are below-grade spaces, determine the ground surface temperature at design winter conditions.
4. Select or compute heat transfer coefficients for outside walls and glass; for inside walls, non-basement floors, and ceilings, if these are next to unheated spaces; and for the roof if it is next to heated spaces.
5. Determine the net area of outside wall, glass, and roof next to heated spaces, as well as any cold walls, floors, or ceilings next to unheated spaces. These determinations can be made from building plans or from the actual buildings, using inside dimensions.
6. Compute heat transmission losses for each kind of wall, glass, floor, ceiling, and roof in the building by multiplying the overall heat transfer coefficient in each case by the area of the surface and the temperature difference between indoor air and outdoor air, adjacent space air, or ground surface, as appropriate.
7. Compute heat losses from grade-level slab floors using the heat loss rate per unit length of exposed perimeter, using the method in this chapter.
8. Compute the energy associated with infiltration of cold air around outside doors, windows, porous building materials, and other openings. These unit values depend on the kind or width of crack, wind speed, and the temperature difference between indoor and outdoor air.
9. When positive ventilation using outdoor air is provided by the conditioning unit, the energy required to warm and humidify the ventilation outdoor air to the space conditions must be provided by the unit. Unless mechanical ventilation is sufficient to maintain the building at a slightly positive pressure at all time (producing exfiltration and preventing infiltration), the heating unit must provide for both ventilation and natural infiltration losses.
10. Sum the coincidental transmission losses or heat transmitted through the confining walls, floor, ceiling, glass, and other surfaces, and the energy associated with cold air entering by infiltration and/or the ventilation air, to obtain the total heating load.
11. Include the pickup loads that may be required in intermittently heated buildings or when using night thermostat setback. Pickup loads frequently require an increase in heating capacity to bring the temperature of structure, air, and material contents to the specified temperature.

Table 6-17 Summary of Heating Load Calculation Equations

Load Source	Equation	Tables and Notes
Exterior surfaces above grade	$q = UA\,\Delta t$	$\Delta t = t_i - t_o$
Partitions to unconditioned buffer space	$q = UA\,\Delta t$	Δt = temp. difference across partition
Walls below grade	$q = U_{\text{avg,bw}}A(t_{\text{in}} - t_{\text{gr}})$	
Floors on grade	$q = F_p p \Delta t$	
Floors below grade	$q = U_{\text{avg,bf}}\,A(t_{\text{in}} - t_{\text{gr}})$	
Ventilation/infiltration	$q_{\text{vi}} = C_s Q\,\Delta t$	From Chapter 5
Total sensible load	$q_s = \Sigma q$	

With the exception of slab-on-grade heat transfer, the basic formula for the heat loss by conductive and convective heat transfer through any surface is

$$q = AU\,(t_i - t_o) \qquad (6\text{-}24)$$

where

q = heat transfer through wall, roof, ceiling, floor, or glass, Btu/h (W)

A = area of wall, glass, roof, ceiling, floor, or other exposed surface, ft^2 (m^2)

U = air-to-air (ground surface) heat transfer coefficient, $\text{Btu/h·ft}^2\text{·°F}$ [$\text{W/(m}^2\text{·K)}$]

t_i = indoor air temperature near surface involved, °F (°C)

t_o = outdoor air temperature, temperature of adjacent unheated space, or ground surface temperature, °F (°C)

Example 6-1 Calculate the design heat loss q_t for the following wall:

Wall size: 10 ft high by 50 ft long

Wall construction: 4 in. face brick; building paper, vapor seal, two layers of mopped 15 lb felt; 4 in. concrete block (normal weight); 1 in. glass fiber, organic bonded; 3/8 in. painted plasterboard.

Design conditions: indoor, 72°F dry bulb, relative humidity = 30%; outdoor, 32°F dry bulb

Solution:

	R-Value
Outdoor air	0.17
Brick	0.44
Felt	0.12
Block	1.11
Glass fiber	4.00
Plasterboard	0.32
Inside air	0.68
	$\Sigma R = 6.84$

$$U = 1/R = 1/6.84 = 0.146 \text{ Btu/h·ft}^2\text{·°F}$$

$$A = 10 \times 50 = 500 \text{ ft}^2$$

$$\Delta t = 72\text{°F} - 32\text{°F} = 40\text{°F}$$

$$q_t = UA\Delta t = 0.146 \times 500 \times 40$$
$$= 2920 \text{ Btu/h } (0.856 \text{ kW})$$

6.4.2 Selecting Heating Design Conditions

The ideal solution to a basic heating system design is a plant with a maximum output capacity equal to the heating load that develops with the most severe local weather conditions. However, this solution is usually uneconomical. Weather records show that severe weather conditions do not repeat annually. If heating systems were designed for maximum weather conditions, excess capacity would exist during most of the system's operating life.

In many cases, an occasional failure of a heating plant to maintain a preselected indoor design temperature during brief periods of severe weather is not critical. However, the successful completion of some industrial or commercial processes may depend on close regulation of indoor temperatures. The specific requirements for each building should be carefully evaluated.

Before selecting an outdoor design temperature from Chapter 4, the designer should consider the following:

- Is the type of structure heavy, medium, or light?
- Is the structure insulated?
- Is the structure exposed to high wind?
- Is the load from infiltration or ventilation high?
- Is there more glass area than normal?
- During what part of the day will the structure be used?
- What is the nature of occupancy?
- Will there be long periods of operation at reduced indoor temperature?
- What is the amplitude between local maximum and minimum daily temperatures?
- Are there local conditions that cause significant variation from temperatures reported by the Weather Bureau?
- What auxiliary heating devices will be in the building?

Before selecting an outdoor design temperature, the designer must keep in mind that, if the outdoor to indoor design temperature difference is exceeded, the indoor temperature may fall, depending on the thermal mass of the structure and its contents, whether or not the internal load was included in calculations, and the duration of the cold period.

The effect of wind on the heating requirements of any building should be considered because:

- Wind movement increases the heat transmission of walls, glass, and roof, affecting poorly insulated walls to a much greater extent than well-insulated walls.
- Wind increases the infiltration of cold air through cracks around doors and windows and even through building materials themselves.

Although 72°F to 75°F are the most commonly selected indoor temperatures for comfort heating design, local code requirements must be checked. ASHRAE Standard 55 and Chapter 9 of the 2017 *Fundamentals* provide additional details on selecting indoor design conditions, as well as Table 4-4.

6.4.3 Heat Loss from Above-Grade Exterior Surfaces

All above-grade surfaces exposed to outdoor conditions (walls, doors, ceilings, fenestration, and raised floors) are treated identically, as follows:

$$q = A \times \text{HF} \tag{6-25}$$

$$\text{HF} = U\,\Delta t$$

where HF is the heating load factor in Btu/h·ft^2.

Two ceiling configurations are common:

- For ceiling/roof combinations (e.g., flat roof or cathedral ceiling), the U-factor should be evaluated for the entire assembly.
- For well-insulated ceilings (or walls) adjacent to vented attic space, the U-factor should be that of the insulated assembly only (the roof is omitted) and the attic temperature assumed to equal the heating design outdoor temperature.

6.4.4 Heat Loss Through Below-Grade Surfaces

Heat transfer through basement walls and floors to the ground depends on (1) the difference between the air tempera-

ture within the room and that of the ground and outdoor air, (2) the material of the walls or floor, and (3) conductivity of the surrounding earth. Conductivity of the earth is usually unknown. Because of thermal inertia, ground temperature varies with depth, and there is a substantial time lag between changes in outdoor air temperatures and corresponding changes in ground temperature. As a result, ground-coupled heat transfer is less amenable to steady-state representation than is the case for above-grade building elements.

An approximate method for estimating below-grade heat loss finds the steady-state heat loss to the ground surface, as follows:

$$\mathrm{HF} = U_{\mathrm{avg}}\,(t_{\mathrm{in}} - t_{\mathrm{gr}}) \tag{6-26}$$

where

U_{avg} = average U-factor for below-grade surface, Btu/h·ft^2·°F
t_{in} = below-grade space air temperature, °F
t_{gr} = design ground surface temperature, °F

The effect of soil heat capacity means that none of the usual external design air temperatures are suitable values for t_{gr}. Ground surface temperature fluctuates about an annual mean value by amplitude A, which varies with geographic location and surface cover. The minimum ground surface temperature, suitable for heat loss estimates, is therefore

$$t_{\mathrm{gr}} = t_m - A \tag{6-27}$$

where

t_m = mean winter temperature, estimated from the winter average air temperature or from well-water temperature
A = ground surface temperature amplitude from Figure 6-1

The value of the soil thermal conductivity k varies widely with soil type and moisture content. A typical value of 0.8 Btu/h·ft·°F was used to tabulate U-factors with an R-value of approximately 1.47 h·ft^2·°F/Btu for uninsulated concrete walls. For these parameters, representative values for $U_{\mathrm{avg,bw}}$ are shown in Table 6-18. Representative values of $U_{\mathrm{avg,bf}}$ for uninsulated basement floors are shown in Table 6-19.

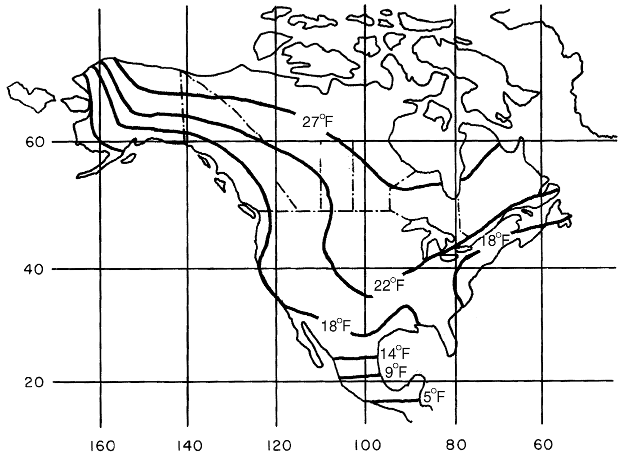


Fig. 6-1 Ground Temperature Amplitude

6.4.5 Heat Loss From On-Grade Surfaces

Concrete slab floors may be (1) unheated, relying for warmth on heat delivered above floor level by the heating system, or (2) heated, containing heated pipes or ducts that constitute a radiant slab or a portion of it for complete or partial heating of the house. Heat loss from a concrete slab floor is mostly through the perimeter rather than through the floor and into the ground. Total heat loss is more nearly proportional to the length of the perimeter than to the area of the floor. The simplified approach that treats heat loss as proportional to slab perimeter allows slab heat loss to be estimated for both unheated and heated slab floors:

$$q = P \times \mathrm{HF}$$

$$\mathrm{HF} = F_p\,\Delta t, \text{ or}$$

$$q = F_p P\,(t_i - t_o) \tag{6-28}$$

where

q = heat loss through the perimeter, Btu/h (W)
F_p = heat loss coefficient, Btu/h· °F· ft of perimeter [W/(m·K)]
P = perimeter of exposed edge of floor, ft (m)
t_i = indoor temperature, °F (°C)
t_o = outdoor design temperature, °F (°C)

Representative heat loss coefficients for slab-on-grade floors are available from Table 6-20.

Example 6-2 Determine the heat loss for a basement in St. Louis, Missouri, which is 60 ft by 40 ft by 8 ft high, of standard concrete construction and entirely below grade. Average winter temperature in St. Louis is 44°F.

Design $\Delta t = t_i - (t_a - A) = 70 - (44 - 22) = 48$ (Figure 6-1)

Wall Average U-Factors (Table 6-18): 0.157 Btu/h·ft°F

$$\mathrm{HF} = U \times \Delta t = 0.157\,(48) = 7.54$$

Wall loss = HF × A = 7.54 (200 × 8) = 12060 Btu/h

Floor Heat Loss (Table 6-20)

Table 6-18 Average U-Factor for Basement Walls with Uniform Insulation

	$U_{\mathrm{avg,bw}}$ from grade to depth, Btu/h·ft^2·°F			
Depth, ft	**Uninsulated**	**R-5**	**R-10**	**R-15**
1	0.4321	0.1351	0.080	0.057
2	0.331	0.121	0.075	0.054
3	0.273	0.110	0.070	0.052
4	0.235	0.101	0.066	0.050
5	0.208	0.094	0.063	0.048
6	0.187	0.088	0.060	0.046
7	0.170	0.083	0.057	0.044
8	0.157	0.078	0.055	0.043

Soil conductivity = 0.8 Btu/h·ft·°F; insulation is over entire depth.

Table 6-19 Average U-Factor for Basement Floors

z_f (depth of floor below grade), ft	$U_{avg,bf}$, Btu/h·ft^2·F			
	w_b (shortest width of basement), ft			
	20	24	28	32
1	0.064	0.057	0.052	0.047
2	0.054	0.048	0.044	0.040
3	0.047	0.042	0.039	0.036
4	0.042	0.038	0.035	0.033
5	0.038	0.035	0.032	0.030
6	0.035	0.032	0.030	0.028
7	0.032	0.030	0.028	0.026

Soil conductivity is 0.8 Btu/h·ft·°F; floor is uninsulated.

$$U_{\text{floor}} = 0.026 \text{ Btu/h·ft}^2\text{·°F}$$

$$\text{Area} = 60 \times 40 = 2400 \text{ ft}^2$$

$$\text{Floor heat loss} = 0.026\ (2400)\ 48 = 3000 \text{ Btu/h}$$

$$q_{\text{total}} = 12060 + 3000 = 15{,}060 \text{ Btu/h (4.41 kW)}$$

6.4.6 Heat Loss To Buffer Spaces

Heat loss to adjacent unconditioned or semiconditioned spaces can be calculated using a heating load factor based on the partition temperature difference:

$$Q = \text{HF} \times A \qquad (6\text{-}29)$$

where

$$\text{HF} = U(t_i - t_b).$$

Buffer space air temperature t_b can be estimated using procedures discussed in Section 4.4. Generally, simple approximations are sufficient except where the partition surface is poorly insulated.

6.4.7 Infiltration Heat Loss

Heat loss due to infiltration of outdoor air can be divided into sensible and latent components. The energy quantity that raises the temperature of outdoor infiltrating air up to indoor air temperature is the sensible component. The energy quantity associated with net loss of moisture from the space is the latent component. These calculations are discussed in Chapter 5.

Example 6-3 The west brick wall of a residence in Louisville, Kentucky, has a net area (excluding windows and doors) of 506 ft^2 and a U-factor of 0.067 Btu/h·ft^2·°F. Determine the heating load for the wall, Btu/h.

Solution:

$$q_s = UA\Delta t = 0.067\ (506)\ (72 - 8) = 2170 \text{ Btu/h}$$

Example 6-4 A large window, essentially all glass, 10 ft by 5 ft, is located in the west wall of a residence in Kansas City, Missouri, where the indoor and outdoor winter design conditions have been selected as 72°F and 6°F, respectively. To reduce energy requirements, the window is heat-absorbing double glass (e = 0.40), 1/2 in. air space, wood framing, and has inside draperies that are closed at peak conditions. Determine the heating load for the window, Btu/h.

Solution:

Table 6-20 Heat Loss Coefficient F_p of Slab Floor Construction

Construction	Insulation	F_p, Btu/h·ft·°F
8 in. block wall, brick facing	Uninsulated floor	0.68
	R-5.4 from edge to footer	0.50
4 in. block wall, brick facing	Uninsulated floor	0.84
	R-5.4 from edge to footer	0.49
Studded wall, stucco	Uninsulated floor	1.20
	R-5.4 from edge to footer	0.53
Poured concrete wall with duct near perimeter*	Uninsulated floor	2.12
	R-5.4 from edge to footer	0.72

*Weighted average temperature of the heating duct was assumed at 110°F during heating season (outdoor air temperature less than 65°F).

From the Chapter 5 table for fenestration U-factors,

$$U_{\text{window w/o draperies}} = 0.45 \text{ Btu/h·ft}^2\text{·°F}$$

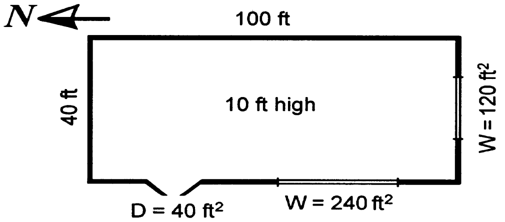
The major effect of the draperies will be to create an air space that will add approximately 1°F·ft^2·F/Btu of thermal resistance. Thus,

$$R_{\text{with drapes}} = 1/U_{w/o} = 1/0.45 + 1 = 3.22$$

$$U_{\text{with drapes}} = 1/3.22 = 0.31 \text{ Btu/h·ft}^2\text{·°F}$$

$$q_s = UA\Delta t = 0.31\ (50)\ (72 - 6) = 1023 \text{ Btu/h}$$

Example 6-5 For the light commercial/residential building described below, determine the design heating load. The outside winter design temperature is 6°F and inside winter design conditions are 72°F, 30% rh. The 8 in. concrete floor is a slab-on-grade with R = 5.4 insulation, properly installed. There are 4900 (base 65°F) degree-days.



Infiltration: 0.5 air changes per hour (ACH)

Wall: 1 in. dark stucco on 4 in. regular concrete, U = 0.350 Btu/h·ft^2·°F

Roof: 8 in. lightweight concrete, U = 0.12 Btu/h·ft^2·°F

Door: 2 in. solid wood, U = 0.42 Btu/h·ft^2·°F, A = 40 ft^2

Windows:

South: 1/8 in. regular sheet, A = 120 ft^2, U = 1.30 Btu/h·ft^2·°F

West: Insulating glass, regular sheet out/regular sheet in, 1/4 in. air space, 1/4 in. thick glass; A = 240 ft^2, U = 0.69 Btu/h·ft^2·°F

Solution:

Construction Element	U	A	Δt	q
East wall	0.35	1000	66	23,100
North wall	0.35	400	66	9,240
West wall	0.35	720	66	16,632
South wall	0.35	280	66	6,468
Door	0.42	40	66	1,135
South glass	1.30	120	66	10,296

Construction Element	U	A	Δt	q
West glass	0.69	240	66	10,930
Roof	0.12	4000	66	31,680
Floor	$F_2 = 0.49$	$P = 280$	66	9,055
			Subtotal =	118,500

Infiltration:

$$q_s = 1.10Q(t_i - t_o)$$
$$q_l = 4840Q(W_i - W_o)$$

where, at 0.5 ach,

$$Q = 0.5(40 \times 100 \times 10)/60 = 333$$

Thus,

$$q_s = 1.10 \times 333 \times 66 = 24{,}200 \text{ Btu/h}$$

$$q_l = 4840 \times 333 \times (0.005 - 0.001) = 6447 \text{ Btu/h}$$

Total design heating load =

$$118{,}500 + 24{,}200 + 6400 = 149{,}100 \text{ Btu/h}$$

Example 6-6 Redo the design heating load for the building of Example 6-5 if, instead of a slab floor, the building is constructed over a full, conditioned basement that has a height of 8 ft, all below grade. Basement walls are insulated with $R = 10$.

Solution:

All values are the same as for the previous problem except that the previously calculated floor loss must be replaced with losses for below-grade walls and floor.

$$\text{Below-grade wall loss} = U_w A[t_i - (t_a - A)]$$
$$\text{Below-grade floor loss} = U_f A[t_i - (t_a - A)]$$

The average winter air temperature t_a (Chapter 4) is found to be 43.1°F, and the ground temperature amplitude from Fig. 6-1 is 22°F.

For the wall, $U_w = 0.055$ (Table 6-18).

$$q_{\text{wall}} = 0.055(280 \times 8)[72 - (43.1 - 22)] = 6270 \text{ Btu/h}$$

For the floor, $U \cong 0.026$.

$$q_{\text{floor}} = 0.026 \times 4000 \times 50.9 = 5294 \text{ Btu/h}$$

Thus, total load =

$$149{,}100 - 9055 + 6270 + 5294 = 151{,}609 \text{ Btu/h}$$

Example 6-7 Redo the design heating load for the building of Example 6-5 if, instead of a slab floor on grade, the 8 in. concrete floor is above an open parking garage.

Solution:

All values are the same as for Example 6-5 except that the previously calculated floor loss must be replaced with one appropriately calculated for air-to-air heat transfer.

The air beneath the floor in the garage may be considered to be at outdoor air temperature and with the normal winter wind velocity. Thus, the U-factor for the floor can be determined as

$$U = \frac{1}{0.92 + 0.075 \times 8 + 0.17} = 0.59 \text{ Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

The design heat loss becomes

$$q = 0.59 \times 4000 \times 66 = 156{,}213 \text{ Btu/h}$$

Total design heating load =

$$149{,}100 - 9055 + 156{,}213 = 296{,}300 \text{ Btu/h}$$

Note: Floor needs several inches of insulation.

6.4.8 Heating Load Summary Sheet

When preparing a set of design heating loads for a multispace building, either a computer spreadsheet software program or a summary sheet such as provided in Figure 6-2 might prove useful. A summary of heating load calculations is given in Table 6-17.

6.5 Nomenclature

Symbols

A = area, ft^2; ground surface temperature amplitude, °F
A_{cf} = building conditioned floor area, ft^2
A_L = building effective leakage area (including flue) at 0.016 in. of water assuming $C_D = 1$, in.^{2}
C_l = air latent heat factor, 4840 Btu/h·cfm at sea level
C_s = air sensible heat factor, 1.1 Btu/h·cfm·°F at sea level
C_t = air total heat factor, 4.5 Btu/h·cfm·(Btu/lb) at sea level
CF = cooling load factor, Btu/h·ft^2
D_{oh} = depth of overhang (from plane of fenestration), ft
DR = daily range of outdoor dry-bulb temperature, °F
E = peak irradiance for exposure, Btu/h·ft^2
F_{dl} = distribution loss factor
F_p = heat loss coefficient per unit length of perimeter, Btu/h·ft·°F
F_{shd} = shaded fraction
FF = coefficient for CF$_{\text{fen}}$
G = internal gain coefficient
h_{srf} = effective surface conductance, including resistance of slab covering material such as carpet, $1/(R_{\text{cvr}} + 0.68)$ Btu/h·ft^2·°F
Δh = indoor/outdoor enthalpy difference, Btu/lb
H = height, ft
HF = heating (load) factor, Btu/h·ft^2
I = infiltration coefficient
IAC = interior shading attenuation coefficient
k = conductivity, Btu/h·ft·°F
LF = load factor, Btu/h·ft^2
OF = coefficient for CF$_{opq}$
p = perimeter or exposed edge of floor, ft
PXI = peak exterior irradiance, including shading modifications, Btu/h·ft^2
q = heating or cooling load, Btu/h
Q = air volumetric flow rate, cfm
R = insulation thermal resistance, h·ft^2·°F/Btu
SHGC= fenestration rated or estimated NFRC solar heat gain coefficient
SLF = shade line factor

HEAT LOSS CALCULATION SHEET

Job Name____

Location____

Date ____

Design Conditions

t_o = ___________ t_i = ___________ Δt = _________

rh_o =___________ rh_i = ___________

W_o = ___________ W_i = ___________ ΔW = _________

Room Number or Name												
Length of Exposed Slab Perim., ft												
Room Dimension, Height, Length, Width, ft												
Component	**No.**	***U* or *U′***	**Δt or ΔW**	**Area**	**Btu/h Sens.**	**Btu/h Lat.**	**Area**	**Btu/h Sens.**	**Btu/h Lat.**	**Area**	**Btu/h Sens.**	**Btu/h Lat.**
Gross Exposed Walls and Partitions												
Windows and Doors												
Net Exposed Walls and Partitions												
Ceilings and Roofs												
Floors												
Infiltration												
Heat Loss, Subtotal												
Duct Heat Loss												
Total Heat Loss												
Air Quantity, cfm												

Fig. 6-2 Heat Loss Calculation Sheet

t = temperature, °F

T_x = solar transmission of exterior attachment

Δt = design dry-bulb temperature difference (cooling or heating), °F

U = construction U-factor, Btu/h·ft^2·°F (for fenestration, NFRC rated *heating* U-factor)

w = width, ft

ΔW = indoor-outdoor humidity ratio difference, lb$_w$/lb$_{da}$

X_{oh} = vertical distance from top of fenestration to overhang, ft

z = depth below grade, ft

α_{roof} = roof solar absorptance

ε = heat/energy recovery ventilation (HRV/ERV) effectiveness

Subscripts

avg = average

b = base (as in OF$_b$) or basement

bal = balanced

bf = basement floor

bw = basement wall

br = bedrooms

ceil = ceiling

cf = conditioned floor

cvr = floor covering

D = direct

d = diffuse

da = dry air

dl = distribution loss

env = envelope

Table 6-21 Example House Characteristics

Component	Description	Factors
Roof/ceiling	Flat wood frame ceiling (insulated with R-30 fiberglass) beneath vented attic with medium asphalt shingle roof	U = 0.031 Btu/h·ft^2·°F α_{roof} = 0.85 (Table 6-7)
Exterior walls	Wood frame, exterior wood sheathing, interior gypsum board, R-13 fiberglass insulation	U = 0.090 Btu/h·ft^2·°F
Doors	Wood, solid core	U = 0.40 Btu/h·ft^2·°F
Floor	Slab on grade with heavy carpet over rubber pad; R-5 edge insulation to 3 ft below grade	R_{cvr} = 1.2 /h·ft^2·°F/Btu (Table 3, Chapter 6, 2008 *ASHRAE Handbook—HVAC Systems and Equipment*) F_p = 0.5 Btu/h·ft·°F (estimated from Table 6-20)
Windows	Clear double-pane glass in wood frames. Half fixed, half operable with insect screens (except living room picture window, which is fixed). 2 ft eave overhang on east and west with eave edge at same height as top of glazing for all windows. Allow for typical interior shading, half closed.	Fixed: U = 0.50 Btu/h·ft^2·°F; SHGC = 0.67 (Table 6-2) Operable: U = 0.51 Btu/h·ft^2·°F; SHGC = 0.57 (Table 6-2) T_x = 0.6 (Table 6-11) IAC_{cl} = 0.6 (estimated from Table 6-14)
Construction	Good	A_{ul} = 0.02 in^2/ft^2 (Table 6-3)

es = exposed surface
exh = exhaust
fen = fenestration
floor = floor
gr = ground
hr = heat recovery
i = infiltration
in = indoor
ig = internal gain
l = latent
o = outdoor
oc = occupant
oh = overhang
opq = opaque
oth = other
r = daily range (as in OF_r)
rhb = calculated with RHB method
s = sensible or solar
shd = shaded
slab = slab
srf = surface
sup = supply
t = total or temperature (as in OF_t)
ul = unit leakage
unbal = unbalanced
v = ventilation
vi = ventilation/infiltration
w = water
wall = wall

6.6 Load Calculation Example

A single-family detached house with floor plan shown in Figure 6-3 is located in Atlanta, Georgia, USA. Construction characteristics are documented in Table 6-21. Using the RLF method, find the block (whole-house) design cooling and heating loads. A furnace/air-conditioner forced-air system is planned with a well-sealed and well-insulated (R-8 wrap) attic duct system.

Table 6-22 Example House Design Conditions

Item	Heating	Cooling	Notes
Latitude	—	—	33.64°N
Elevation	—	—	1027 ft
Indoor temperature	68°F	75°F	
Indoor relative humidity	N/A	50%	No humidification
Outdoor temperature	26°F	92°F	Cooling: 1% value Rounded values Heating: 99%
Daily range	N/A	18°F	
Outdoor wet bulb	N/A	74°F	MCWB* at 1%
Wind speed	15 mph	7.5 mph	Default assumption
Design Δt	42°F	17°F	
Moisture difference		0.0052 lb/lb	Psychrometric chart

*MCWB = mean coincident wet bulb

Solution

Design Conditions. Table 6-22 summarizes design conditions. Typical indoor conditions are assumed. Outdoor conditions should be determined from Chapter 4, 2017 *ASHRAE Handbook—Fundamentals*, however slightly modified values were selected.

Component Quantities. Areas and lengths required for load calculations are derived from plan dimensions (Figure 6-3). Table 6-23 summarizes these quantities.

Opaque Surface Factors. Heating and cooling factors are derived for each component condition. Table 6-24 shows the resulting factors and their sources.

Window Factors. Deriving cooling factor values for windows requires identifying all unique glazing configurations in the house. Equation 6-16 input items indicate that the variations for this case are exposure, window height (with overhang shading), and frame type (which determines U-factor, SHGC, and the presence of insect screen). CF derivation for all configurations is summarized in Table 6-25.

For example, CF for operable 3 ft high windows facing west (the second row in Table 6-25) is derived as follows:

- U-factor and SHGC are found in Table 6-2.
- Each operable window is equipped with an insect screen. From Table 6-12, T_x = 0.6 for this arrangement.

Table 6-23 Example House Opaque Surface Factors

Component	U, Btu/h·ft^2·°F or F_p, Btu/h·ft·°F	Heating HF	 Reference	Cooling OF$_t$	 OF$_b$	 OF$_r$	 CF	 Reference
Ceiling	0.031	1.30	Equation 34	0.62	13.75	–0.19	0.65	Table 7 Equation 21
Wall	0.090	3.78		1	14.80	–0.36	2.31	
Garage wall	0.090	3.78		1	0.00	–0.36	0.98	
Door	0.400	16.8		1	14.80	–0.36	10.27	
Floor perimeter	0.500	21.0	Chapter 18, Equation 42					
Floor area				0.59	–2.5/(0.68 + 1.20) = –1.33		–0.74	Equation 23

Table 6-24 Example House Window Factors

Exposure	Height, ft	Frame	U, Btu/h·ft^2·°F Table 2	HF Eq. 34	T_x Table 12	F_{shd} Eq. 28	PXI Eq. 27	SHGC Table 2	IAC Eq. 29	FF$_s$ Table 14	CF Eq. 25
West	3	Fixed	0.50	21.0	1	0.73	108	0.67	0.80	0.56	36.9
	3	Operable	0.51	21.4	0.64	0.73	69	0.57	0.80	0.56	22.3
	6	Fixed	0.50	21.0	1	0.37	172	0.67	0.80	0.56	56.1
	6	Operable	0.51	21.4	0.64	0.37	110	0.57	0.80	0.56	32.7
	8	Fixed	0.50	21.0	1	0.28	187	0.67	0.80	0.56	60.9
South	4	Fixed	0.50	21.0	1	0.00	131	0.67	0.80	0.47	37.6
	4	Operable	0.51	21.4	0.64	0.00	84	0.57	0.80	0.47	22.7
East	3	Fixed	0.50	21.0	1	0.73	108	0.67	0.80	0.31	22.5
	3	Operable	0.51	21.4	0.64	0.73	69	0.57	0.80	0.31	14.4
	4	Fixed	0.50	21.0	1	0.55	140	0.67	0.80	0.31	27.8
	4	Operable	0.51	21.4	0.64	0.55	89	0.57	0.80	0.31	17.3

Table 6-25 Example House Envelope Loads

Component	HF	CF	Quantity, ft^2 or ft	Heating Load, Btu/h	Cooling Load, Btu/h
Ceiling	1.30	0.65	2088	2714	1363
Wall	3.78	2.31	1180	4460	2727
Garage wall	3.78	0.98	384	1452	376
Door	16.8	10.27	42	706	431
Floor perimeter	21.0		220	4620	
Floor area		–0.74	2088		–1545
W-Fixed-3	21.0	36.9	4.5	95	166
W-Operable-3	21.4	22.3	4.5	96	100
W-Fixed-6	21.0	56.1	12	252	673
W-Operable-6	21.4	32.7	12	257	393
W-Fixed-8	21.0	60.9	48	1008	2921
S-Fixed-4	21.0	37.6	8	168	301
S-Operable-4	21.4	22.7	8	171	181
E-Fixed-3	21.0	22.5	4.5	95	101
E-Operable-3	21.4	14.4	4.5	96	65
E-Fixed-4	21.0	27.8	24	504	667
E-Operable-4	21.4	17.3	24	514	416
Envelope totals				17,207	9336

- Overhang shading is evaluated with Equation 6-19. For west exposure and latitude 34°, Table 6-12 shows SLF = 1.1. Overhang depth (D_{oh}) is 2 ft and the window-overhang distance (X_{oh}) is 0 ft. With window height h of 3 ft, $F_s = 0.73$ (73% shaded).
- PXI depends on peak irradiance and shading. Approximating site latitude as 35°N, Table 6-9 shows $E_D = 177$ and $E_d = 60$ Btu/h·ft^2 for west exposure. Equation 6-18 combines these values with T_x and F_s to find PXI = $0.6[52 + (1 - 0.73)177] = 69$ Btu/h·ft^2.

Table 6-26 Example House Total Sensible Loads

Item	Heating Load, Btu/h	Cooling Load, Btu/h
Envelope	17,207	9336
Infiltration/ventilation	5914	1627
Internal gain		2226
Subtotal	23,121	13,189
Distribution loss	3006	3561
Total sensible load	26,126	16,750

- All windows are assumed to have some sort of interior shading in the half-closed position. Use (Equation 6-20) with $F_{\text{cl}} = 0.5$ and $\text{IAC}_{\text{cl}} = 0.6$ (per Table 6-21) to derive IAC = 0.8.
- FF$_s$ is taken from Table 6-13 for west exposure.
- Finally, inserting the preceding values into Equation 6-16 gives CF = $0.51(17 - 0.46 \times 17) + 69 \times 0.57 \times 0.80 \times 0.56 = 22.3$ Btu/h·ft^2.

Envelope Loads. Given the load factors and component quantities, heating and cooling loads are calculated for each envelope element, as shown in Table 6-26.

Infiltration and Ventilation. From Table 6-3, A_{ul} for this house is 0.02 in.^{2}/ft^2 of exposed surface area. Applying Equation 6-8 yields $A_L = A_{\text{es}} \times A_{\text{ul}} = 3848 \times 0.02 = 77$ in^2. Using Table 6-5, estimate heating and cooling IDF to be 1.0 and 0.48 cfm/in^2, respectively [alternatively, Equation 6-9 could be used to find IDF values]. Apply Equation 6-7 to find the infiltration leakage rates and Equation 6-6 to convert the rate to air changes per hour:

$$Q_{i,h} = 77 \times 1.0 = 77 \text{ cfm } (0.28 \text{ ach})$$

$$Q_{i,c} = 77 \times 0.48 = 36 \text{ cfm } (0.13 \text{ ach})$$

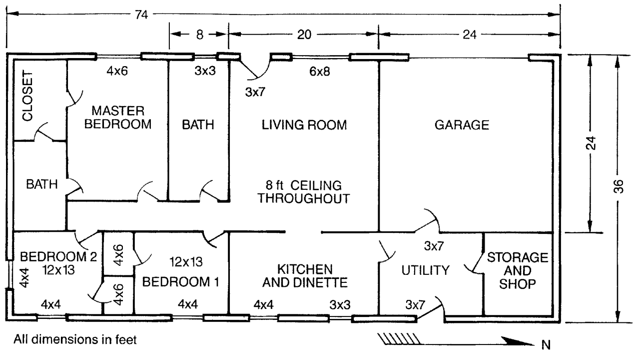


Fig. 6-3 Example House

Calculate the ventilation outdoor air requirement with Equation 6-10 using A_{cf} = 2088 ft^2 and N_{br} = 3, resulting in Q_v = 51 cfm. For design purposes, assume that this requirement is met by a mechanical system with balanced supply and exhaust flow rates ($Q_{\text{unbal}} = 0$).

Find the combined infiltration/ventilation flow rates with Equation 14, Chapter 17, 2017 *ASHRAE Handbook—Fundamentals*:

$$Q_{\text{vi},h} = 51 + \max(0, 77 + 0.5 \times 0) = 128 \text{ cfm}$$

$$Q_{\text{vi},c} = 51 + \max(0, 36 + 0.5 \times 0) = 87 \text{ cfm}$$

At Atlanta's elevation of 1027 ft, elevation adjustment of heat factors results in a small (4%) reduction in air heat transfer; thus, adjustment is unnecessary, resulting in C_s = 1.10 Btu/h·°F·cfm. Use Equation 6-1 with $Q_{\text{bal,hr}} = 0$ and $Q_{\text{bal,oth}} = 0$ to calculate the sensible infiltration/ventilation loads:

$$q_{\text{vi},s,h} = 1.1 \times 128 \times 42 = 5914 \text{ Btu/h}$$

$$q_{\text{vi},s,c} = 1.1 \times 87 \times 17 = 1627 \text{ Btu/h}$$

Note: Using an estimate of 0.4 ach for good construction would result in an infiltration rate of 142 cfm, slightly higher than the 128 calculated by this procedure.

Internal Gain. Apply Equation 6-21 to find the sensible cooling load from internal gain:

$$q_{\text{ig},s} = 464 + 0.7 \times 2088 + 75\ (3 + 1) = 2226 \text{ Btu/h}$$

Distribution Losses and Total Sensible Load. Table 6-27 summarizes the sensible load components. Distribution loss factors F_{dl} are estimated at 0.13 for heating and 0.27 for cooling.

Latent Load. Use Equation 6-2 with C_l = 4840 Btu/h·cfm, $Q_{\text{vi},c}$ = 87 cfm, $Q_{\text{bal,oth}} = 0$, and $\Delta W = 0.0052$ to calculate the infiltration/ventilation latent load = 2187 Btu/h. Use Equation 6-23 to find the latent load from internal gains = 378 Btu/h. Therefore, the total latent cooling load is 2565 Btu/h.

6.7 Problems

6.1 Determine which of the following walls of 150 ft^2 gross area will have the greatest heat loss:

(a) Wall of 25% single glass and the remainder brick veneer (U = 0.25 Btu/h·ft^2·°F)
(b) Wall of 50% double-glazed windows with the remainder of the wall brick veneer (U = 0.25 Btu/h·ft^2·°F)
(c) Wall of 10% single-pane glass and 90% of 6 in. poured concrete with h_o = 6.0 and h_i = 1.6 Btu/h·ft^2

6.2 A house has a pitched roof with an area of 159 m^2 and a U of 1.6 W/(m^2·K). The ceiling beneath the roof has an area of 133 m^2 and a U of 0.42 W/(m^2·K). The attic is unvented in winter for which the design conditions are –19°C outside and 22°C inside. Determine the heat loss through the ceiling. [Ans: 1.88 kW (6400 Btu/h)]

6.3 Determine the design winter heat loss through each of the following components of a building located in Minneapolis, Minnesota:

(a) Wall having 648 ft^2 of area and construction of 4 in. face brick; 3/4 in. plywood sheathing; 2 1/2 in. glass fiber insulation in 2 by 4 stud space (16 in. on centers); 1/2 in. plasterboard interior wall.
(b) A 2185 ft^2 ceiling topped by a 2622 ft^2 hip roof. The ceiling consists of 1/2 in. acoustical tile with R-19 insulation between the 2 by 6 (16 in. on centers) ceiling joists. The roof has asphalt shingles on 3/4 in. plywood sheathing on the roof rafters. The attic is unvented in winter.
(c) Two 4 ft by 6 ft single-pane glass windows with storm windows.

6.4 If the building of Problem 6.3 is a residence having a volume of 17,480 ft^3 that is equipped with a humidifier set for 25% rh, determine:

(a) Sensible heat load due to infiltration
(b) Latent heat load due to infiltration

6.5 For a frame building with design conditions of 72°F indoor and 12°F outdoor, determine the heat loss through each of the following components:

(a) Slab floor, 56 ft by 28 ft, on grade without perimeter insulation [Ans: 12,100 Btu/h]
(b) Single-glass double-hung window, 3 ft by 5 ft, with storm window in common metal frame [Ans: 729 Btu/h]
(c) 1 3/8 in. thick solid wood door, 3 ft by 7 ft, with wood storm door [Ans: 731 Btu/h]
(d) Sliding patio door, 6 ft by 7 ft, metal frame with double insulating glass having 1/4 in. air space [Ans: 2041 Btu/h]

6.6 Determine the heat loss for a basement in Chicago, Illinois, which is 8 m by 12 m by 2.1 m high, of standard concrete construction, and entirely below grade.

6.7 A residence located in Chicago, Illinois, has a total ceiling area of 1960 ft^2 and consists of 3/8 in. gypsum board on 2

by 6 ceiling joists. Six inches of fiberglass (mineral/glass wool) insulation fills the space between the joists. The pitched roof has asphalt shingles on 25/32 in. solid wood sheathing with no insulation between the rafters. The ratio of roof area to ceiling area is 1:3. The attic is unvented in winter. For winter design conditions, including a 72°F inside dry bulb at the 5 ft line, determine

(a) Outside design temperature, °F
(b) Appropriate temperature difference, °F
(c) Appropriate overall coefficient U, Btu/h·ft^2·°F
(d) Ceiling heat loss q, Btu/h

6.8 A residential building, 30 ft by 100 ft, located in Des Moines, Iowa, has a conditioned space that extends 9 ft below grade level. Determine the design heat loss from the uninsulated below-grade concrete walls and floor.

6.9 Determine the heating load and specify the furnace for the following residence (located in St. Louis, Missouri) with

(a) 1 in. fiberglass wall insulation and 2 in. fiberglass ceiling insulation
(b) Full wall fiberglass insulation and 4 in. fiberglass ceiling insulation

Basic Plan

Wall Construction: Face brick, 25/32 in. insulating board sheathing, 2 by 4 studs on 16 in. centers, 3/8 in. gypsum board interior

Ceiling: 2 by 6 ceiling joists, 16 in. on center, no flooring above, 3/8 in. gypsum board ceiling

Roof: Asphalt shingles on solid wood sheathing, 2 by 6 rafters, no insulation between rafters, no ceiling applied to rafters, 1:4 pitch, 1 ft overhang on eaves, no overhang on gables

Full basement: Heated, 10 in. concrete walls, all below grade, 4 in. concrete floor over 4 in. gravel

Fireplaces: One in living room on first floor

Garage: Attached but unheated

Windows:

W1: 3 ft by 5 ft single-glazed, double-hung wood sash, weather stripped with storm window
W2: 10 ft by 5 1/2 ft picture window, double glazed, 1/2 in. airspace
W3: 5 ft by 3 ft wood sash casement, double glazed, 1/2 in. airspace
W4: 3 ft by 3 ft wood sash casement, double glazed, 1/2 in. airspace

Doors:

D1: 3 ft by 6 ft 8 in., 1 3/4 in. solid with glass storm door
D2: Sliding glass door, two section, each 3 ft by 6 ft, 8 in. double-glazed, 1/2 in. airspace, aluminum frame

[Ans: (b) 51,000 Btu/h (14.7 kW)]

6.10 Determine the total conductance loss through the wall panel as shown below. The window has a wooden sill and the plate glass ($U = 1.06$) covers 85% of the window area.

[Ans: 9640 Btu/h (2.77 kW)]

6.11 Calculate, for design purposes, the heat losses from a room of a building as shown in the diagram, if the outside ambient is 0°F.

[Ans: 46,800 Btu/h (13.7 kW)]

6.12 A room has three 760 mm by 1520 mm well-fitted double-hung windows. For design conditions of −1°C and 21°C, calculate (a) heating load from air leakage and (b) heating load from transmission through the windows.

6.13 A residence has a total ceiling area of 1960 ft^2 and consists of 3/8 in. gypsum board on 2 in. by 6 in. ceiling joists. Six inches of fiberglass (mineral/glass wool) insulation fills the space between the joists. The effect of the joists themselves can be neglected. The pitched roof has asphalt shingles on 5/8 in. plywood with no insulation between the rafters. The ratio of roof area to ceiling area is 1:3. The attic contains louvers that remain open all year. The residence is located in Louisville, Kentucky. For winter design conditions, determine: (a) appropriate temperature difference Δt, (b) overall coefficient U, and (c) ceiling heat loss.

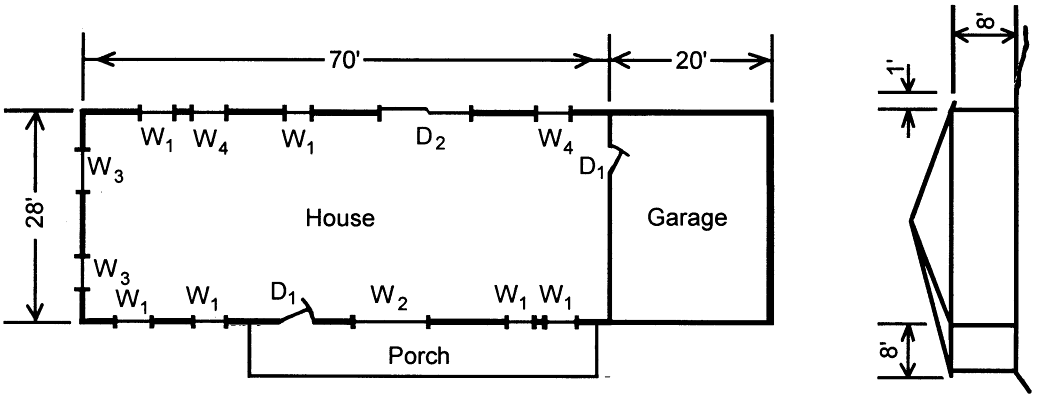


Diagram for Problem 6.9

6.14 Estimate the heat loss from the uninsulated slab floor of a frame house having dimensions of 18 m by 38 m. The house is maintained at 22°C. Outdoor design temperature is –15°C in a region with 5400 kelvin days.

[Ans: 8.6 kW]

6.15 Repeat Problem 6.14 for the case where insulation [$R = 0.9\ (\mathrm{m^2 \cdot K})/\mathrm{W}$] is applied to the slab edge and extended below grade to the frost line.

6.16 To preclude attic condensation, an attic ventilation rate of 59 L/s is provided with outdoor air at –13°C. The roof area is 244 $\mathrm{m^2}$ and $U_{\mathrm{roof}} = 2.7\ \mathrm{W/(m^2 \cdot K)}$. The ceiling area is 203 $\mathrm{m^2}$ and $U_{\mathrm{clg}} = 0.30\ \mathrm{W/(m^2 \cdot K)}$. Inside design temperature is 22°C. Determine the ceiling heat loss W with ventilation and compare to the loss if there had been no ventilation.

6.17 For a residence in Roanoke, Virginia, the hip roof consisting of asphalt shingles on 1/2 in. plywood has an area of 2950 $\mathrm{ft^2}$. The 2300 $\mathrm{ft^2}$ ceiling consists of 3/8 in. plasterboard on 2 by 6 joists on 24 in. centers. The attic has forced ventilation at the rate of 325 cfm. Determine the attic air temperature at winter design conditions.

6.18 Solve the following:

(a) A 115 ft by 10 ft high wall in Minneapolis, Minnesota, consists of face brick, a 3/4 in. air gap, 8 in. cinder aggregate concrete blocks, 1 in. organic bonded glass fiber insulation, and 4 in. clay tile interior. Determine the design heat loss through the wall in winter, Btu/h.

(b) If the wall of Part (a) is converted to 60% single-glazed glass, what is the winter design heat loss through the total wall, Btu/h?

6.19 Determine the design heating load for a residence, 30 ft by 100 ft by 10 ft, to be located in Windsor Locks, Connecticut, which has an uninsulated slab-on-grade concrete floor. The construction consists of

Walls: 4 in. face brick, 3/4 in. plywood sheathing, 4 in. cellular glass insulation, and 1/2 in. plasterboard

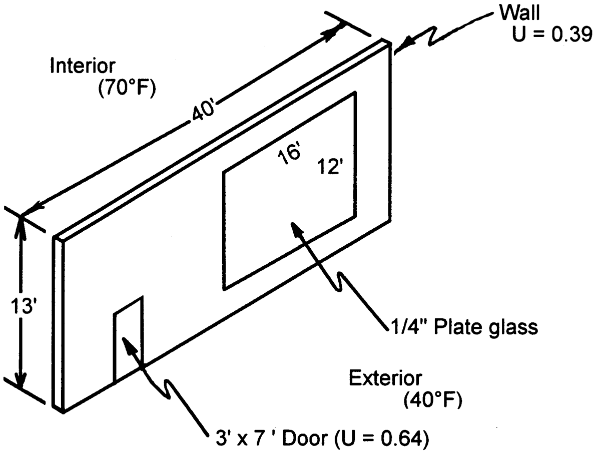
Ceiling/roof: 3 in. lightweight concrete deck, built-up roofing, 2 in. of rigid, expanded rubber insulation, and a drop ceiling of 1/2 in. acoustical tiles, some 18 in. below the roof.

Windows: 45% of each wall is double-pane, nonoperable, metal-framed glass (1/4 in. air gap).

Doors: Two 3 ft by 7 ft, 1.75 in. thick, solid wood doors are located in each wall.

6.20 As an attempt to minimize energy requirements, a new residence has been constructed in Dallas, Texas (100°F dry bulb; 20°F daily range; $W = 0.0156$). Size the air-conditioning unit (Btu/h) for this residence with the following features:

- No windows
- Inside design conditions 75°F, 60% rh ($W = 0.0112$)
- One 3 ft by 7 ft, 2 in. thick wood door with storm door on south side, $U = 0.26\ \mathrm{Btu/h{\cdot}ft^2{\cdot}°F}$, one 3 ft by 7 ft, 2 in. thick



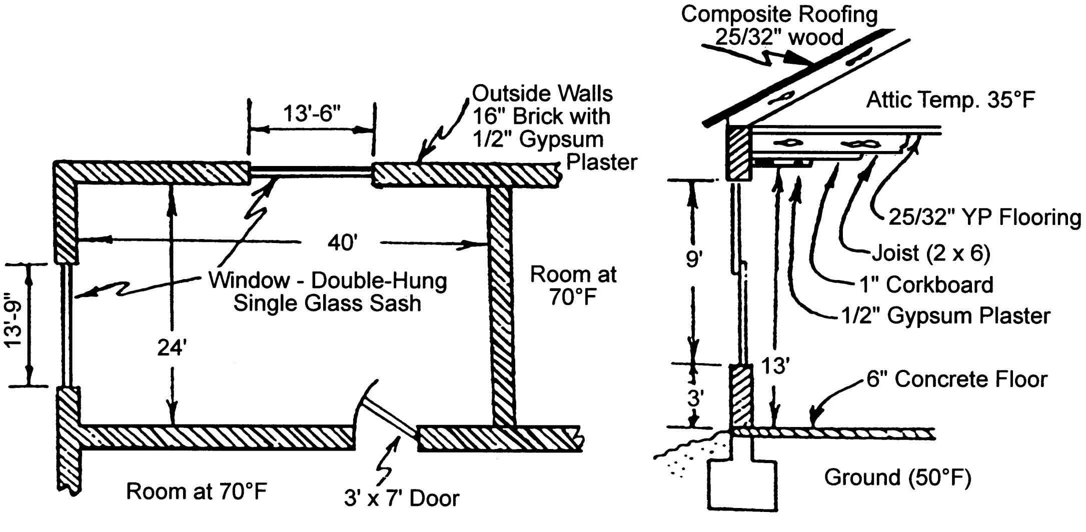


Diagram for Problem 6.11

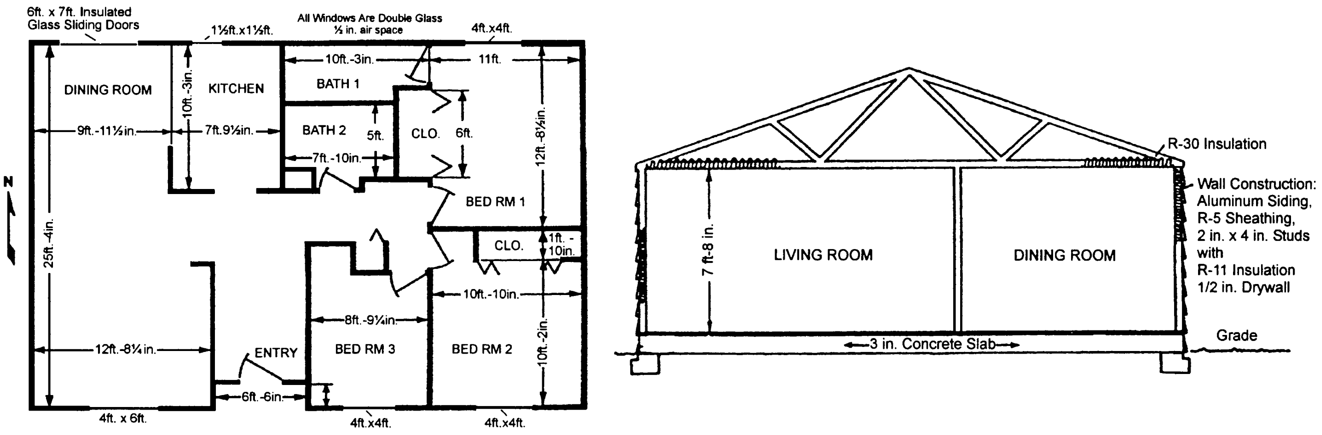


Diagram for Problem 6.21

wood door with storm door on east side, U = 0.26 Btu/h·ft^2·°F

- Overall size: 70 ft by 28 ft by 8 ft high
- Walls are frame construction with 2 by 6 studs and full insulation for $U = 0.043$
- Attic has natural ventilation, 12 in. fiberglass insulation, and overall U of roof and ceiling of 0.021 based on ceiling area, light-colored roof
- Infiltration is so small that outdoor air must be brought in at a rate of 9 cfm/person
- Estimated occupancy at design condition is 15 people
- Fluorescent tights rated at 320 W will be on all the time
- Floor is concrete slab on ground.

6.21 Determine design heating and cooling loads for the residence shown in the figure and located in Manhattan, Kansas.

6.22 Determine the cooling load and specify the central air-conditioning system for the following residence in St. Louis, Missouri, having:

(a) black asphalt shingles, full-wall fiberglass insulation, 4 in. fiberglass ceiling insulation, no drapes, no attic fan
(b) same as (a) except silver-white asphalt shingles
(c) same as (a) except lined drapes at all windows
(d) same as (a) except large attic vent fan

Basic Plan

Wall construction: Face brick, 25/32 in. insulating board sheathing, 2 by 4 studs on 16 in. centers, 3/8 in. gypsum board interior

Ceiling: 2 by 6 ceiling joists, 16 in. on-canter, no flooring above, 3/8 in. gypsum board ceiling

Roof: Asphalt shingles on solid wood sheathing, 2 by 6 rafters, no insulation between rafters, no ceiling applied to rafters, 1:4 pitch, 1 ft overhang on eaves, no overhang on gables

Full basement: Heated, 10 in. concrete watts, all below grade, 4 in. concrete floor over 4 in. gravel

Fireplace: One in living room of first floor

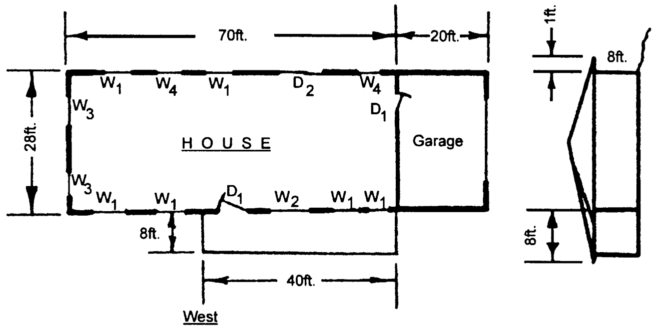
Garage: Attached but unheated

Windows:

W1: 3 ft by 5 ft single-glazed, double-hung wood sash, weather stripped with storm window
W2: 10 ft by 5.5 ft picture window, double-glazed, 1/2 in. airspace
W3: 5 ft by 3 ft wood-sash casement, double-glazed, 1/2 in. airspace
W4: 3 ft by 3 ft wood-sash casement, double-glazed, 1/2 in. airspace

Doors:

D1: 3 ft by 6 ft, 8 in., 1 3/4 in. solid with glass storm door
D2: Sliding glass door, two 3 ft by 6 ft 8 in sections, double-glazed, 1/2 in. airspace, aluminum frame



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SI Figures and Tables

Table 6-1 SI RLF Limitations

Item	Valid Range	Notes
Latitude	20° S to 60° N	Also approximately valid for 20° S to 60° S with N and S orientations reversed for southern hemisphere.
Date	July 21	Application must be summer peaking. Buildings in mild climates with significant SE/S/SW glazing may experience maximum cooling load in fall or even winter. Use RHB if local experience indicates this is a possibility.
Elevation	Less than 2000 m	RLF factors assume 50 m elevation. With elevation-corrected C_s, method is acceptably accurate except at very high elevations.
Climate	Warm/hot	Design-day average outdoor temperature assumed to be above indoor design temperature.
Construction	Lightweight residential construction (wood or metal framing, wood or stucco siding)	May be applied to masonry veneer over frame construction; results are conservative. Use RHB for structural masonry or unconventional construction.
Fenestration area	0% to 15% of floor area on any façade, 0% to 30% of floor area total	Spaces with high fenestration fraction should be analyzed with RHB.
Fenestration tilt	Vertical or horizontal	Skylights with tilt less than 30° can be treated as horizontal. Buildings with significant sloped glazing areas should be analyzed with RHB.
Occupancy	Residential	Applications with high internal gains and/or high occupant density should be analyzed with RHB or nonresidential procedures.
Temperature swing	1.7 Ks	
Distribution losses	Typical	Applications with extensive duct runs in unconditioned spaces should be analyzed with RHB.

Table 6-2 SI Typical Fenestration Characteristics

					Frame									
					Operable					Fixed				
Glazing Type	**Glazing Layers**	**ID[b]**	**Property[c,d]**	**Center of Glazing**	**Aluminum**	**Aluminum with Thermal Break**	**Reinforced Vinyl/Aluminum Clad Wood**	**Wood/Vinyl**	**Insulated Fiberglass/Vinyl**	**Aluminum**	**Aluminum with Thermal Break**	**Reinforced Vinyl/Aluminum Clad Wood**	**Wood/Vinyl**	**Insulated Fiberglass/Vinyl**
Clear	1	1a	*U*	5.91	7.24	6.12	5.14	5.05	4.61	6.42	6.07	5.55	5.55	5.35
			SHGC	0.86	0.75	0.75	0.64	0.64	0.64	0.78	0.78	0.75	0.75	0.75
	2	5a	*U*	2.73	4.62	3.42	3.00	2.87	5.83	3.61	3.22	2.86	2.84	2.72
			SHGC	0.76	0.67	0.67	0.57	0.57	0.57	0.69	0.69	0.67	0.67	0.67
	3	29a	*U*	1.76	3.80	2.60	2.25	2.19	1.91	2.76	2.39	2.05	2.01	1.93
			SHGC	0.68	0.60	0.60	0.51	0.51	0.51	0.62	0.62	0.60	0.60	0.60
Low-e, low-solar	2	25a	*U*	1.70	3.83	2.68	2.33	2.21	1.89	2.75	2.36	2.03	2.01	1.90
			SHGC	0.41	0.37	0.37	0.31	0.31	0.31	0.38	0.38	0.36	0.36	0.36
	3	40c	*U*	1.02	3.22	2.07	1.76	1.71	1.45	2.13	1.76	1.44	1.40	1.33
			SHGC	0.27	0.25	0.25	0.21	0.21	0.21	0.25	0.25	0.24	0.24	0.24
Low-e, high-solar	2	17c	U	1.99	4.05	2.89	2.52	2.39	2.07	2.99	2.60	2.26	2.24	2.13
			SHGC	0.70	0.62	0.62	0.52	0.52	0.52	0.64	0.64	0.61	0.61	0.61
	3	32c	*U*	1.42	3.54	2.36	2.02	1.97	1.70	2.47	2.10	1.77	1.73	1.66
			SHGC	0.62	0.55	0.55	0.46	0.46	0.46	0.56	0.56	0.54	0.54	0.54
Heat-absorbing	1	1c	*U*	5.91	7.24	6.12	5.14	5.05	4.61	6.42	6.07	5.55	5.55	5.35
			SHGC	0.73	0.64	0.64	0.54	0.54	0.54	0.66	0.66	0.64	0.64	0.64
	2	5c	*U*	2.73	4.62	3.42	3.00	2.87	2.53	3.61	3.22	2.86	2.84	2.72
			SHGC	0.62	0.55	0.55	0.46	0.46	0.46	0.56	0.56	0.54	0.54	0.54
	3	29c	*U*	1.76	3.80	2.60	2.25	2.19	1.91	2.76	2.39	2.05	2.01	1.93
			SHGC	0.34	0.31	0.31	0.26	0.26	0.26	0.31	0.31	0.30	0.30	0.30
Reflective	1	1l	*U*	5.91	7.24	6.12	5.14	5.05	4.61	6.42	6.07	5.55	5.55	5.35
			SHGC	0.31	0.28	0.28	0.24	0.24	0.24	0.29	0.29	0.27	0.27	0.27
	2	5p	*U*	2.73	4.62	3.42	3.00	2.87	2.53	3.61	3.22	2.86	2.84	2.72
			SHGC	0.29	0.27	0.27	0.22	0.22	0.22	0.27	0.27	0.26	0.26	0.26
	3	29c	*U*	1.76	3.80	2.60	2.25	2.19	1.91	2.76	2.39	2.05	2.01	1.93
			SHGC	0.34	0.31	0.31	0.26	0.26	0.26	0.31	0.31	0.30	0.30	0.30

[a] Data are from Chapter 15 in the 2017 *ASHRAE Handbook—Fundamentals*, Tables 4 and 14 for selected combinations.

[b] ID = Chapter 15 in the 2017 *ASHRAE Handbook—Fundamentals* glazing type identifier.

[c] U = U-factor, W/(m^2·K)

[d] SHGC = solar heat gain coefficient

Table 6-3 SI Unit Leakage Areas

Construction	Description	A_{ul} (cm^2/m^2)
Tight	Construction supervised by air-sealing specialist	0.7
Good	Carefully sealed construction by knowledgeable builder	1.4
Average	Typical current production housing	2.8
Leaky	Typical pre-1970 houses	5.6
Very leaky	Old houses in original condition	10.4

Table 6-4 SI Evaluation of Exposed Surface Area

Situation	Include	Exclude
Ceiling/roof combination (e.g., cathedral ceiling without attic)	Gross surface area	
Ceiling or wall adjacent to attic	Ceiling or wall area	Roof area
Wall exposed to ambient	Gross wall area (including fenestration area)	
Wall adjacent to unconditioned buffer space (e.g., garage or porch)	Common wall area	Exterior wall area
Floor over open or vented crawlspace	Floor area	Crawlspace wall area
Floor over sealed crawlspace	Crawlspace wall area	Floor area
Floor over conditioned or semiconditioned basement	Above-grade basement wall area	Floor area
Slab floor		Slab area

Table 6-5 SI Typical IDF Values, L/(s·cm^2)

	Heating Design Temperature, °C					Cooling Design Temperature, °C			
H, m	–40	–30	–20	–10	0	10	30	35	40
2.5	0.10	0.095	0.086	0.077	0.069	0.060	0.031	0.035	0.040
3	0.11	0.10	0.093	0.083	0.072	0.061	0.032	0.038	0.043
4	0.14	0.12	0.11	0.093	0.079	0.065	0.034	0.042	0.049
5	0.16	0.14	0.12	0.10	0.086	0.069	0.036	0.046	0.055
6	0.18	0.16	0.14	0.11	0.093	0.072	0.039	0.050	0.061
7	0.20	0.17	0.15	0.12	0.10	0.075	0.041	0.051	0.068
8	0.22	0.19	0.16	0.14	0.11	0.079	0.043	0.058	0.074

Table 6-6 SI Opaque Surface Cooling Factor Coefficients

Surface Type	OF_t	OF_b, °F	OF_r
Ceiling or wall adjacent to vented attic	0.62	$25.7\,\alpha_{roof} - 8.1$	–0.19
Ceiling/roof assembly	1	$68.9\,\alpha_{roof} - 12.6$	–0.36
Wall (wood frame) or door with solar exposure	1	14.8	–0.36
Wall (wood frame) or door (shaded)	1	0	–0.36
Floor over ambient	1	0	–0.06
Floor over crawlspace	0.33	0	–0.28
Slab floor (see Slab Floor section)			

α_{roof} = roof solar absorptance (see Table 6-7)

Table 6-7 SI Roof Solar Absorptance a_{roof}

	Color			
Material	White	Light	Medium	Dark
Asphalt shingles	0.75	0.75	0.85	0.92
Tile	0.30	0.40	0.80	0.80
Metal	0.35	0.50	0.70	0.90
Elastomeric coating	0.30			

Source: Summarized from Parker et al. 2000

Table 6-8 SI Thermal Resistance of Floor Coverings

Description	Thermal Resistance r_c, ft^2·h·°F/Btu
Bare concrete, no covering	0
Asphalt tile	0.05
Rubber tile	0.05
Light carpet	0.60
Light carpet with rubber pad	1.00
Light carpet with light pad	1.40
Light carpet with heavy pad	1.70
Heavy carpet	0.80
Heavy carpet with rubber pad	1.20
Heavy carpet with light pad	1.60
Heavy carpet with heavy pad	1.90
3/8 in. hardwood	0.54
5/8 in. wood floor (oak)	0.57
1/2 in. oak parquet and pad	0.68
Linoleum	0.12
Marble floor and mudset	0.18
Rubber pad	0.62
Prime urethane underlayment, 3/8 in.	1.61
48 oz. waffled sponge rubber	0.78
Bonded urethane, 1/2 in.	2.09

Notes:

1. Carpet pad thickness should not be more than 1/4 in.
2. Total thermal resistance of carpet is more a function of thickness than of fiber type.
3. Generally, thermal resistance (R-value) is approximately 2.6 times the total carpet thickness in inches.
4. Before carpet is installed, verify that the backing is resistant to long periods of continuous heat up to 120°F

Table 6-9 SI Peak Irradiance, W/m^2

		Latitude								
Exposure		20°	25°	30°	35°	40°	45°	50°	55°	60°
North	E_D	106	98	98	104	117	138	167	203	106
	E_d	97	85	74	65	56	48	41	34	97
	E_t	203	183	172	169	174	187	208	237	203
Northeast/Northwest	E_D	442	442	444	447	451	457	465	474	442
	E_d	149	139	131	124	117	111	106	100	149
	E_t	590	582	575	571	568	568	570	574	590
East/West	E_D	524	548	570	590	608	624	638	651	524
	E_d	178	171	164	159	154	149	145	141	178
	E_t	702	719	734	748	761	773	783	792	702
Southeast/Southwest	E_D	299	355	407	455	499	540	577	610	299
	E_d	193	187	182	178	174	170	167	164	193
	E_t	492	542	589	632	673	710	744	775	492
South	E_D	21	114	203	286	365	439	509	574	21
	E_d	197	192	188	185	182	180	177	175	197
	E_t	218	306	391	471	547	619	686	749	218
Horizontal	E_D	788	790	782	765	739	705	661	608	788
	E_d	170	170	170	170	170	170	170	170	170
	E_t	958	960	952	935	909	875	831	778	958

Table 6-10 SI Peak Irradiance Equations

Horizontal surfaces

$$E_t = 258.7 + 3.233L - 0.0572L^2$$

$$E_d = \min(E_t, 53.9)$$

$$E_D = E_t - E_d$$

Vertical surfaces

$$\phi = \left|\frac{\psi}{180}\right| \text{(normalized exposure, 0 – 1)}$$

$$E_t = 134.2 + 393\phi - 1630\phi^3 + 1011\phi^4 + 11.17\phi L + 0.0452\phi L^2 - 4.086L - 0.1956L^2 + [0.2473L^2/(\phi + 1)]$$

$$E_d = \min 12.9 - 11.96\phi^2 + 0.0958\phi L - \frac{36.74\sqrt[4]{L}}{\phi + 1}$$

$$E_D = E_t - E_d$$

where

E_t, E_d, E_D = peak hourly total, diffuse, and direct irradiance, Btu/h·ft^2
L = site latitude, °N
ψ = exposure (surface azimuth), ° from south (–180 to +180)

Table 6-11 SI Exterior Attachment Transmission

Attachment	T_x
None	1.0
Exterior insect screen	0.64 (see Chapter 15, Table 14G, 2017 *ASHRAE Handbook—Fundamental*s [SI])
Shade screen	Manufacturer SC value, typically 0.4 to 0.6

Note: See Brunger et al. (1999) regarding insect screens

Table 6-12 SI Shade Line Factors (SLFs)

	Latitude								
Exposure	**20°**	**25°**	**30°**	**35°**	**40°**	**45°**	**50°**	**55°**	**60°**
North	2.8	2.1	1.4	1.5	1.7	1.0	0.8	0.9	0.8
Northeast/Northwest	1.4	1.5	1.6	1.2	1.3	1.3	0.9	0.9	0.8
East/West	1.2	1.2	1.1	1.1	1.1	1.0	1.0	0.9	0.8
Southeast/Southwest	2.1	1.8	2.0	1.7	1.5	1.6	1.4	1.2	1.1
South	20.0	14.0	6.9	4.7	3.3	2.7	2.1	1.7	1.4

Note: Shadow length below overhang = SLF × D_{oh}

Table 6-13 SI Fenestration Solar Load Factors FF_s

Exposure	**Single Family Detached**	**Multifamily**
North	0.44	0.27
Northeast	0.21	0.43
East	0.31	0.56
Southeast	0.37	0.54
South	0.47	0.53
Southwest	0.58	0.61
West	0.56	0.65
Northwest	0.46	0.57
Horizontal	0.58	0.73

Table 6-14 SI Interior Attenuation Coefficients (IAC_{cl})

Glazing Layers	Glazing Type (ID*)	Drapes			Roller Shades			Blinds	
		Open-Weave	Closed-Weave		Opaque		Translucent		
		Light	Dark	Light	Dark	White	Light	Medium	White
1	Clear (1a)	0.64	0.71	0.45	0.64	0.34	0.44	0.74	0.66
	Heat absorbing (1c)	0.68	0.72	0.50	0.67	0.40	0.49	0.76	0.69
2	Clear (5a)	0.72	0.81	0.57	0.76	0.48	0.55	0.82	0.74
	Low-e high-solar (17c)	0.76	0.86	0.64	0.82	0.57	0.62	0.86	0.79
	Low-e low-solar (25a)	0.79	0.88	0.68	0.85	0.60	0.66	0.88	0.82
	Heat absorbing (5c)	0.73	0.82	0.59	0.77	0.51	0.58	0.83	0.76

*Chapter 15 glazing identifier

Table 6-15 SI Summary of RLF Cooling Load Equations

Load Source	Equation	Tables and Notes
Exterior opaque surfaces	$q_{opq} = A \times CF$	
	$CF = U(OF_t \Delta t + OF_b + OF_r DR)$	OF factors from Table 7, Chapter 29, 2005 *ASHRAE Handbook—Fundamentals*
Exterior transparent surfaces	$q_{fen} = A \times CF$	PXI from Table 9 plus adjustments, Chapter 29, 2005 *ASHRAE Handbook—Fundamentals*
	$CF = U(\Delta t - 0.46DR) + PXI \times SHGC \times IAC \times FF_s$	FF_s from Table 13, Chapter 29, 2005 *ASHRAE Handbook—Fundamentals*
Partitions to unconditioned space	$q = AU\Delta t$	Δt = temperature difference across partition
Ventilation/infiltration	$q_s = C_s Q \Delta t$	See Common Data and Procedures section
Occupants and appliances	$q_{ig,s} = 136 + 2.2A_{cf} + 22N_{oc}$	
Distribution	$q_d = F_{dl} \Sigma q$	F_{dl} from Table 6, Chapter 17, 2009 *ASHRAE Handbook—Fundamentals*
Total sensible load	$q_s = q_d + \Sigma q$	
Latent load	$q_l = q_{vi,l} + q_{ig,}$	
Ventilation/infiltration	$q_{vi,l} = C_l Q \Delta W$	
Internal gain	$q_{ig,l} = 20 + 0.22A_{cf} + 12N_{oc}$	

Table 6-16 SI Typical Duct Loss/Gain Factors

		1 Story						2 or More Stories					
	Supply/Return Leakage	11%/11%			5%/5%			11%/11%			5%/5%		
Duct Location	Insulation	R-0	R-	R-	R-0	R-	R-	R-0	R-	R-	R-0	R-	R-
Conditioned space		No loss ($F_{dl} = 0$)											
Attic	C	1.26	0.71	0.63	0.68	0.33	0.27	1.02	0.66	0.60	0.53	0.29	0.25
	H/F	0.49	0.29	0.25	0.34	0.16	0.13	0.41	0.26	0.24	0.27	0.14	0.12
	H/HP	0.56	0.37	0.34	0.34	0.19	0.16	0.49	0.35	0.33	0.28	0.17	0.15
Basement	C	0.12	0.09	0.09	0.07	0.05	0.04	0.11	0.09	0.09	0.06	0.04	0.04
	H/F	0.28	0.18	0.16	0.19	0.10	0.08	0.24	0.17	0.15	0.16	0.09	0.08
	H/HP	0.23	0.17	0.16	0.14	0.09	0.08	0.20	0.16	0.15	0.12	0.08	0.07
Crawlspace	C	0.16	0.12	0.11	0.10	0.06	0.05	0.14	0.12	0.11	0.08	0.06	0.05
	H/F	0.49	0.29	0.25	0.34	0.16	0.13	0.41	0.26	0.24	0.27	0.14	0.12
	H/HP	0.56	0.37	0.34	0.34	0.19	0.16	0.49	0.35	0.33	0.28	0.17	0.15

Values calculated for ASHRAE *Standard* 152 default duct system surface area using model of Francisco and Palmiter (1999). Values are provided as guidance only; losses can differ substantially for other conditions and configurations. Assumed surrounding temperatures:

Cooling (C): t_o = , t_{attic} = , t_b = , t_{crawl} =

Heating/furnace (H/F) and heating/heating pump (H/HP): t_o = , t_{attic} = , t_b = , t_{crawl} =

Table 6-17 SI Average U-Factor for Basement Walls with Uniform Insulation

Depth, m	$U_{avg,bw}$ from grade to depth, W/(m^2·K)			
	Uninsulated	R-0.88	R-1.76	R-2.64
0.3	2.468	0.769	0.458	0.326
0.6	1.898	0.689	0.427	0.310
0.9	1.571	0.628	0.401	0.296
1.2	1.353	0.579	0.379	0.283
1.5	1.195	0.539	0.360	0.272
1.8	1.075	0.505	0.343	0.262
2.1	0.980	0.476	0.328	0.252
2.4	0.902	0.450	0.315	0.244

Soil conductivity = 1.4 W/(m·K); insulation is over entire depth.

Table 6-18 SI Average U-Factor for Basement Floors

z_f (depth of floor below grade), m	$U_{avg,bf}$, W/(m^2·K)			
	w_b (shortest width of basement), m			
	6	7	8	9
0.3	0.370	0.335	0.307	0.283
0.6	0.310	0.283	0.261	0.242
0.9	0.271	0.249	0.230	0.215
1.2	0.242	0.224	0.208	0.195
1.5	0.220	0.204	0.190	0.179
1.8	0.202	0.188	0.176	0.166
2.1	0.187	0.175	0.164	0.155

Soil conductivity is 1.4 W/(m·K); floor is uninsulated.

Table 6-19 SI Heat Loss Coefficient F_p of Slab Floor Construction

Construction	Insulation	F_p, W/(m·K)
200 mm block wall, brick facing	Uninsulated floor	1.17
	R-0.95 (m^2·K)/W from edge to footer	0.86
4 in. block wall, brick facing	Uninsulated floor	1.45
	R-0.95 (m^2·K)/W from edge to footer	0.85
Studded wall, stucco	Uninsulated floor	2.07
	R-0.95 (m^2·K)/W from edge to footer	0.92
Poured concrete wall with duct near perimeter*	Uninsulated floor	3.67
	R-0.95 (m^2·K)/W from edge to footer	1.24

*Weighted average temperature of the heating duct was assumed at 43°C during heating season (outdoor air temperature less than 18°C).

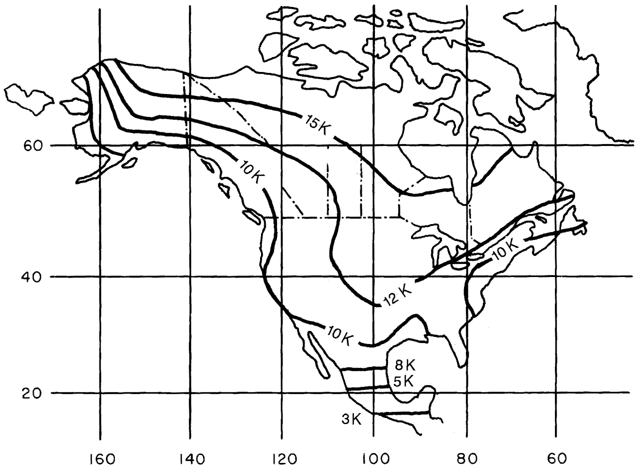


Fig. 6-1 SI Ground Temperature Amplitude

Chapter 7

NONRESIDENTIAL COOLING AND HEATING LOAD CALCULATIONS

This chapter presents the methodology for determining the air-conditioning cooling and heating loads used for sizing cooling and heating equipment for nonresidential buildings. A more detailed discussion of the cooling and heating loads for buildings is given in Chapters 15 and 18 of the 2017 *ASHRAE Handbook—Fundamentals*. Another excellent source of information is the *Load Calculation Applications Manual* (ASHRAE 2014).

7.1 Principles

Heating and cooling load calculations are the primary basis for the design and selection of most heating and air-conditioning systems and components. These calculations are necessary to determine the size of piping, ducting, diffusers, air handlers, boilers, chillers, coils, compressors, fans, and every other component of the systems that condition indoor environments. Cooling and heating load calculations will directly or indirectly affect the first cost of building construction, the comfort and productivity of building occupants, and operation and energy consumption.

Heating and cooling loads are the rates of energy input (heating) or removal (cooling) required to maintain an indoor environment at a desired combination of temperature and humidity. Heating and cooling systems are designed, sized, and controlled to accomplish that energy transfer. The amount of heating or cooling required at any particular time varies widely, depending on external (e.g., outside temperature) and internal (e.g., number of people present) factors. Peak design heating and cooling load calculations seek to determine the maximum rate of heating and of cooling energy transfer needed at any time in the year. This chapter discusses common elements of load calculations and several methods of making load estimates, but it focuses on ASHRAE's Radiant Time Series (RTS) method.

Cooling loads result from many conductive, convective, and radiative heat transfer processes through the building envelope and from internal sources and system components. Building components or contents that may affect cooling loads include the following:

External: Walls, roofs, windows, partitions, ceilings, and floors

Internal: Lights, people, appliances, and equipment

Infiltration: Air leakage and moisture migration

System:Ventilation air, duct leakage, reheat, and fan and pump power.

The variables affecting cooling load calculations are numerous, often difficult to define precisely, and always intricately interrelated. Many cooling load components vary in magnitude over a wide range during a 24-hour period. Since these cyclic changes in load components are often not in phase with each other, each must be analyzed to establish the resultant maximum cooling load for a building or zone. A zoned system (a system of conditioning equipment serving several independent areas, each with its own temperature control) needs no greater total cooling load capacity than the largest hourly summary of simultaneous zone loads throughout a design day; however, it must handle the peak cooling load for each zone at its individual peak hour. At certain times of the day during the heating or intermediate seasons, some zones may need heating while others need cooling.

Calculation Accuracy. The concept of determining the cooling load for a given building must be kept in perspective. A proper cooling load calculation gives values adequate for proper performance. Variation in the heat transmission coefficient of typical building materials and composite assemblies, the differing motivations and skills of those who physically construct the building, and the manner in which the building is actually operated are some of the variables that make a numerically precise calculation impossible. While the designer uses reasonable procedures to account for these factors, the calculation can never be more than a good estimate of the actual cooling load.

Heat Flow Rates. In air-conditioning design, four related heat flow rates, each of which varies with time, must be differentiated: (1) space heat gain, (2) space cooling load, (3) space heat extraction rate, and (4) cooling coil load.

Space Heat Gain. This instantaneous rate of heat gain is the rate heat enters into and/or is generated within a space at a given instant. Heat gain is classified by (1) the mode in which it enters the space and (2) whether it is a sensible or latent gain.

Mode of Entry. The modes of heat gain may be (1) solar radiation through transparent surfaces; (2) heat conduction through exterior walls and roofs; (3) heat conduction through interior partitions, ceilings, and floors; (4) heat generated within the space by occupants, lights, and appliances; (5) energy transfer as a result of ventilation and infiltration of outdoor air; or (6) miscellaneous heat gains.

Sensible or Latent Heat. Sensible heat gain is directly added to the conditioned space by conduction, convection,

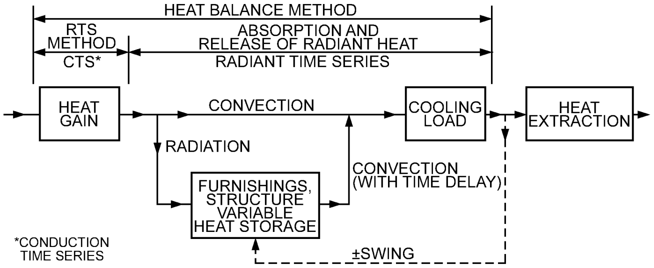


Fig. 7-1 Origin of Difference Between Magnitude of Instantaneous Heat Gain and Instantaneous Cooling Load
(Figure 1, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

and/or radiation. Latent heat gain occurs when moisture is added to the space (e.g., from vapor emitted by occupants and equipment). To maintain a constant humidity ratio, water vapor must condense on the cooling apparatus at a rate equal to its rate of addition into the space. The amount of energy required to offset the latent heat gain essentially equals the product of the rate of condensation and the latent heat of condensation. In selecting cooling apparatus, it is necessary to distinguish between sensible and latent heat gain. Every cooling apparatus has a maximum latent heat removal capacity for particular operating conditions.

Space Cooling Load. This is the rate at which heat must be removed from the space to maintain a constant space air temperature. The sum of all space instantaneous heat gains at any given time does not necessarily (or even frequently) equal the cooling load for the space at that same time.

Radiant Heat Gain. Space heat gain by radiation is not immediately converted into cooling load. Radiant energy must first be absorbed by the surfaces that enclose the space (walls, floor, and ceiling) and the objects in the space (furniture, etc.). As soon as these surfaces and objects become warmer than the space air, some of their heat is transferred in the air space by convection. The composite heat storage capacity of these surfaces and objects determines the rate at which their respective surface temperatures increase for a given radiant input and thus governs the relationship between the radiant portion of heat gain and its corresponding part of the space cooling load (Figure 7-1). The thermal storage effect is critically important in differentiating between instantaneous heat gain for a given space and its cooling load for that moment. Predicting the nature and magnitude of this elusive phenomenon in order to estimate a realistic cooling load for a particular combination of circumstances has long been a subject of major interest to design engineers.

Space Heat Extraction Rate. The rate at which heat is removed from the conditioned space equals the space cooling load only to the degree that room air temperature is held constant. In conjunction with intermittent operation of the cooling equipment, the control system characteristics usually permit a minor cyclic variation or swing in room temperature. Therefore, a proper simulation of the control system gives a more realistic value of energy removal over a fixed period than using the values of the space cooling load. This concept is primarily important for estimating energy use over time; however, it is not needed to calculate design peak cooling load for equipment selection.

Cooling Coil Load. The rate at which energy is removed at the cooling coil that serves one or more conditioned spaces equals the sum of the instantaneous space cooling loads (or space heat extraction rate if it is assumed that the space temperature does not vary) for all the spaces served by the coil, plus any external loads. Such external loads include heat gain by the distribution system between the individual spaces and the cooling equipment, and outdoor air heat and moisture introduced into the distribution system through the cooling equipment.

Cooling Load Estimation in Practice. Frequently, a cooling load must be calculated before every parameter in the conditioned space can be properly or completely defined. An example is a cooling load estimate for a new building with many floors of unleased spaces where detailed partition requirements, furnishings, lighting selection, and layout cannot be predefined. Potential tenant modifications once the building is occupied also must be considered. The load estimating process requires proper engineering judgment that includes a thorough understanding of heat balance fundamentals.

7.1.1 Heat Balance Fundamentals

The calculation of cooling load for a space involves calculating a surface-by-surface conductive, convective, and radiative heat balance for each room surface and a convective heat balance for the room air. Sometimes called "the exact solution," these principles form the foundation for all other methods described in this chapter.

To calculate space cooling load directly by heat balance procedures requires a laborious solution of energy balance equations involving the space air, surrounding walls and windows, infiltration and ventilation air, and internal energy sources. To demonstrate the calculation principle, consider a sample room enclosed by four walls, a ceiling, and a floor, with infiltration air and normal internal energy sources. The energy exchange at each inside surface at a given time can be calculated from the following equation:

$$q_{i,\theta} = \left[h_{ci}(t_{a,\theta} - t_{i,\theta}) + \sum_{j = 1 \neq i}^{m} g_{ij}(t_{j,\theta} - t_{i,\theta}) \right] A_i + \mathrm{RS}_{i,\theta} + \mathrm{RL}_{i,\theta} + \mathrm{RE}_{i,\theta} \tag{7-1}$$

$$\text{for } i = 1, 2, 3, 4, 5, 6$$

where

m = number of surfaces in room (six in this case)

$q_{i,\theta}$ = rate of heat conducted into surface i at inside surface at time θ

A_i = area of surface i

h_{ci} = convective heat transfer coefficient at interior surface i

g_{ij} = a radiation heat transfer factor between interior surface *i* and interior surface *j*
$t_{a,\theta}$ = inside air temperature at time θ
$t_{i,\theta}$ = average temperature of interior surface *i* at time θ
$\mathrm{RS}_{i,\theta}$ = rate of solar energy coming through windows and absorbed by surface *i* at time θ
$\mathrm{RL}_{i,\theta}$ = rate of heat radiated from lights and absorbed by surface *i* at time θ
$\mathrm{RE}_{i,\theta}$ = rate of heat radiated from equipment and occupants and absorbed by surface *i* at time θ

Conduction Transfer Functions. The equations governing conduction within the six surfaces cannot be solved independently of Equation 7-1, since the energy exchanges occurring within the room affect the inside surface conditions, in turn affecting the internal conduction. Consequently, the six formulations of Equation 7-1 must be solved simultaneously with the governing equations of conduction within the six surfaces in order to calculate the space cooling load. Typically, these equations are formulated as conduction transfer functions (CTFs) in the form

$$q_{\mathrm{in},\theta} = \sum_{m=1}^{M} Y_{k,m} t_{o,\theta-m+1} - \sum_{m=1}^{M} Z_{k,m} t_{o,\theta-m+1} + \sum_{m=1}^{M} F_m q_{\mathrm{in},\theta-m} \tag{7-2}$$

where

q = rate of heat conducted into a specific surface at a specific hour
in = inside surface subscript
k = order of CTF
m = time index variable
M = number of nonzero CTF values
o = outside surface subscript
t = temperature
θ = time
Y = cross CTF values
Z = interior CTF values
F_m = flux history coefficients

Space Air Energy Balance. Note that the interior surface temperature, $t_{i,\theta}$ in Equation 7-1 and $t_{\mathrm{in},\theta}$ in Equation 7-2, requires simultaneous solution. In addition, Equation 7-3, which represents an energy balance on the space air, must also be solved simultaneously.

$$Q_{L,\theta} = \left[\sum_{i=1}^{6} h_{ci}(t_{i,\theta} - t_{a,\theta})\right] A_i + \rho C V_{L,\theta}(t_{o,\theta} - t_{a,\theta}) + \rho C V_{v,\theta}(t_{v,\theta} - t_{a,\theta}) + \mathrm{RS}_{a,\theta} + \mathrm{RL}_{a,\theta} + \mathrm{RE}_{a,\theta} \tag{7-3}$$

where

ρ = air density
C = air specific heat
$V_{L,\theta}$ = volume flow rate of outdoor air infiltrating into room at time θ
$t_{o,\theta}$ = outdoor air temperature at time θ
$V_{v,\theta}$ = volume rate of flow of ventilation air at time θ
$t_{v,\theta}$ = ventilation air temperature at time θ
$\mathrm{RS}_{a,\theta}$ = rate of solar heat coming through windows and convected into room air at time θ
$\mathrm{RL}_{a,\theta}$ = rate of heat from lights convected into room air at time θ
$\mathrm{RE}_{a,\theta}$ = rate of heat from equipment and occupants and convected into room air at time θ

Note that the space air temperature is allowed to float. By fixing the space air temperature, the cooling load need not be determined simultaneously.

This rigorous approach to calculating space cooling load is impractical without the speed at which computations can be done by digital computers. Computer programs that calculate instantaneous space cooling loads in this exact manner are primarily oriented to energy use calculations over extended periods (Mitalas and Stephenson 1967; Buchberg 1958).

The transfer function concept is a simplification to the strict heat balance calculation procedure. In the transfer function concept, Mitalas and Stephenson (1967) used room thermal response factors. In their procedure, room surface temperatures and cooling load were first calculated by the rigorous method just described for several typical constructions representing offices, schools, and dwellings of heavy, medium, and light construction. In these calculations, components such as solar heat gain, conductive heat gain, or heat gain from the lighting, equipment, and occupants were simulated by pulses of unit strength. The transfer functions were then calculated as numerical constants representing the cooling load corresponding to the input excitation pulses. Once these transfer functions were determined for typical constructions they were assumed independent of input pulses, thus permitting cooling loads to be determined without the more rigorous calculation. Instead, the calculation requires simple multiplication of the transfer functions by a time-series representation of heat gain and subsequent summation of these products. The same transfer function concept can be applied to calculating heat gain components themselves, as explained later.

7.1.2 Total Equivalent Temperature Differential Method

In the total equivalent temperature differential (TETD) method, the response factor technique was used with a number of representative wall and roof assemblies from which data were derived to calculate TETD values as functions of sol-air temperature and maintained room temperature. Various components of space heat gain are calculated using associated TETD values, and the results are added to internal heat gain elements to get an instantaneous total rate of space heat gain. This gain is converted to an instantaneous space cooling load by the time-averaging (TA) technique of averaging the

radiant portions of the heat gain load components for the current hour with related values from an appropriate period of immediately preceding hours. This technique provides a rational means to deal quantitatively with the thermal storage phenomenon, but it is best solved by computer because of its complexity.

7.1.3 Transfer Function Method

Although similar in principle to TETD/TA, the transfer function method (TFM) (Mitalas 1972) applies a series of weighting factors, or conduction transfer function (CTF) coefficients, to the various exterior opaque surfaces and to differences between sol-air temperature and inside space temperature to determine heat gain with appropriate reflection of thermal inertia of such surfaces. Solar heat gain through glass and various forms of internal heat gain are calculated directly for the load hour of interest. The TFM next applies a second series of weighting factors, or coefficients of room transfer functions (RTF), to heat gain and cooling load values from all load elements having radiant components to account for the thermal storage effect in converting heat gain to cooling load. Both evaluation series consider data from several previous hours as well as the current hour. RTF coefficients relate specifically to the spatial geometry, configuration, mass, and other characteristics of the space so as to reflect weighted variations in thermal storage effect on a time basis rather than a straight-line average.

Transfer Functions. These coefficients relate an output function at a given time to the value of one or more driving functions at a given time and at a set period immediately preceding. The CTF described in this chapter is no different from the thermal response factor used for calculating wall or roof heat conduction, while the RTF is the weighting factor for obtaining cooling load components (ASHRAE 1975). While the TFM is scientifically appropriate and technically sound for a specific cooling load analysis, its computational complexity requires significant computer use for effective application in a commercial design environment.

7.1.4 Heat Balance Method (HB)

The estimation of cooling load for a space involves calculating a surface-by-surface conductive, convective, and radiative heat balance for each room surface and a convective heat balance for the room air. Sometimes called the **exact solution**, these principles form the foundation for all methods described in this chapter.

Some of the computations required by this rigorous approach to calculating space cooling load make the use of digital computers essential. The heat balance procedure is not new. Many energy calculation programs have used it in some form for many years. The first implementation that incorporated all the elements to form a complete method was NBSLD (Kusuda 1967). The heat balance procedure is also implemented in both the BLAST and TARP energy analysis programs (Walton 1983). Prior to the implementation of ASHRAE Research Project 875, the method had never been described completely or in a form applicable to cooling load calculations. The papers resulting from RP-875 describe the heat balance procedure in detail (Pedersen et al. 1997; Liesen and Pedersen 1997; McClellan and Pedersen 1997).

Description of Heat Balance Model. All calculation procedures involve some kind of model. All models require simplifying assumptions and therefore are approximate. The most fundamental assumption is that the air in the thermal zone can be modeled as **well mixed**, meaning it has a uniform temperature throughout the zone. ASHRAE Research Project 664 established that this assumption is valid over a wide range of conditions.

The next major assumption is that the surfaces of the room (walls, windows, floor, etc.) can be treated as having:

- Uniform surface temperatures
- Uniform longwave (LW) and shortwave (SW) irradiation
- Diffuse radiating surfaces
- One-dimensional heat conduction within

The resulting formulation is called the **heat balance model**. It is important to note that the foregoing assumptions, although common, are quite restrictive and set certain limits on the information that can be obtained from the model.

7.1.5 Radiant Time Series Method (RTS)

The radiant time series (RTS) method is a new simplified method for performing design cooling load calculations that is derived from the heat balance (HB) method described above. It effectively replaces all other simplified (non-heat-balance) methods, such as the transfer function method (TFM), the cooling load temperature difference/cooling load factor (CLTD/CLF) method, and the total equivalent temperature difference/time averaging (TETD/TA) method.

The casual observer might well ask why yet another load calculation method is necessary. This method was developed in response to the desire to offer a method that is rigorous, yet does not require iterative calculations, and that quantifies each component contribution to the total cooling load. In addition, it is desirable for the user to be able to inspect and compare the coefficients for different construction and zone types in a form illustrating their relative impact on the result. These characteristics of the RTS method make it easier to apply engineering judgment during the cooling load calculation process.

The RTS method is suitable for peak design load calculations, but it should not be used for annual energy simulations due to its inherent limiting assumptions. The RTS method, while simple in concept, involves too many calculations to be used practically as a manual method, although it can easily be implemented in a simple computerized spreadsheet. For a manual cooling load calculation method, refer to the CLTD/CLF method included in the 1997 *ASHRAE Handbook—Fundamentals*.

7.2 Initial Design Considerations

To calculate a space cooling load, detailed building design information and weather data at selected design conditions are required. Generally, the following steps should be followed:

Building Characteristics. Characteristics of the building, such as building materials, component size, external surface colors, and shape, can usually be obtained from building plans and specifications.

Configuration. Determine building location, orientation, and external shading from plans and specifications. Shading from adjacent buildings can be determined by a site plan or by visiting the proposed site. The probable permanence of shading should be evaluated before it is included in the calculations. Possible high ground-reflected solar radiation from adjacent water, sand, parking lots, or solar load from adjacent reflective buildings should not be overlooked.

Outdoor Design Conditions. Obtain appropriate weather data and select outdoor design conditions. Weather data can be obtained from local weather stations or (in the United States) from the National Climatic Data Center (NCDC), Asheville, North Carolina 28801. (See Chapter 4 for outdoor design conditions for a large number of weather stations.) The designer should exercise judgment to ensure that results are consistent with expectations. Prevailing wind velocity and the relationship of a project site to a selected weather station should also be considered.

Indoor Design Conditions. Select indoor design conditions such as indoor dry-bulb temperature, indoor wet-bulb temperature, and ventilation rate. Include permissible variations and control limits.

Internal Heat Gains and Operating Schedules. Obtain planned density and a proposed schedule of lighting, occupancy, internal equipment, appliances, and processes that contribute to the internal thermal load.

Areas. Use consistent methods for calculation of building areas. For fenestration, the definition of a component's area must be consistent with associated ratings.

Gross surface area. It is efficient and conservative to derive gross surface areas from outside building dimensions, ignoring wall and floor thicknesses and avoiding separate accounting for floor edge and wall corner conditions. Measure floor areas to the outside of adjacent walls or to the center line of adjacent partitions. When apportioning to rooms, façade area should be divided at partition centerlines. Wall height should be taken as floor-to-floor height. The outside dimension is recommended as expedient for load calculations, but it may not be consistent with rigorous definitions used in building-related standards. However, the resulting differences do not introduce significant errors in the estimated design cooling and heating loads.

Fenestration area. Fenestration ratings [U-factor and solar heat gain coefficient (SHGC)] are based on the entire product area, including frames. Thus, for load calculations, fenestration area is the area of the rough opening in the wall or roof.

Net surface area. Net surface area is the gross surface area minus any fenestration area.

Additional Considerations. The proper design and sizing of all-air or air-and-water central air-conditioning systems require more than calculation of the cooling load in the space to be conditioned. The type of air-conditioning system, fan energy, fan location, duct heat loss and gain, duct leakage, heat-extraction lighting systems, and type of return air system all affect system load and component sizing. Adequate system design and component sizing require that system performance be analyzed as a series of psychrometric processes.

7.3 Heat Gain Calculation Concepts

The primary weather-related variable influencing a building's cooling load is solar radiation. The effect of solar radiation is more pronounced and immediate on exposed, nonopaque surfaces. The calculation of solar heat gain and conductive heat transfer through various glazing materials and associated mounting frames is discussed in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals*.

7.3.1 Heat Gain Through Exterior Walls and Roofs

Heat gain through exterior opaque surfaces is derived in the same way as for fenestration areas. It differs primarily as a function of the mass and nature of wall or roof construction, since those elements affect the rate of conductive transfer through the composite assembly to the interior surface.

Sol-Air Temperature. This is the temperature of the outdoor air that, in the absence of all radiation changes, gives the same rate of heat entry into the surface as would the combination of incident solar radiation, radiant energy exchange with the sky and other outdoor surroundings, and convective heat exchange with outdoor air.

Heat Gain Through Exterior Surfaces. The heat balance at a sunlit surface gives the heat flux into the surface q/A in Btu/h·ft^2 (W/m^2), as

$$q/A = \alpha I_t + h_o(t_o - t_s) - \varepsilon\Delta R \tag{7-4}$$

where

α = absorptance of surface for solar radiation

I_t = total solar radiation incident on surface, Btu/h·ft^2 (W/m^2)

h_o = coefficient of heat transfer by longwave radiation and convection at outer surface, Btu/h·ft^2·°F [W/(m^2·K)]

t_o = outdoor air temperature, °F (°C)

t_s = surface temperature, °F (°C)

ε = hemispherical emittance of surface

ΔR = difference between longwave radiation incident on surface from sky and surroundings and radiation emitted by blackbody at outdoor air temperature, Btu/h·ft^2 (W/m^2)

Table 7-1 Sol-Air Temperature (t_e) for July 21, 40° N Latitude

$t_e = t_o + \alpha I_t/h_o - \varepsilon \Delta R/h_o$

		Light Colored Surface, α/h_o = 0.15											Dark Colored Surface, α/h_o = 0.30								
Time	Air Temp. t_o, °F	N	NE	E	SE	S	SW	W	NW	HOR	Time	Air Temp. t_o, °F	N	NE	E	SE	S	SW	W	NW	HOR
1	76	76	76	76	76	76	76	76	76	69	1	76	76	76	76	76	76	76	76	76	69
2	76	76	76	76	76	76	76	76	76	69	2	76	76	76	76	76	76	76	76	76	69
3	75	75	75	75	75	75	75	75	75	68	3	75	75	75	75	75	75	75	75	75	68
4	74	74	74	74	74	74	74	74	74	67	4	74	74	74	74	74	74	74	74	74	67
5	74	74	74	74	74	74	74	74	74	67	5	74	74	75	75	74	74	74	74	74	67
6	74	80	93	95	84	76	76	76	76	72	6	74	85	112	115	94	77	77	77	77	77
7	75	80	99	106	94	78	78	78	78	81	7	75	84	124	136	113	81	81	81	81	94
8	77	81	99	109	101	82	81	81	81	92	8	77	85	121	142	125	86	85	85	85	114
9	80	85	96	109	106	88	85	85	85	102	9	80	90	112	138	131	96	89	89	89	131
10	83	88	91	105	107	95	88	88	88	111	10	83	94	100	127	131	107	94	94	94	145
11	87	93	93	99	106	102	93	93	93	118	11	87	98	99	111	125	118	100	98	98	156
12	90	96	96	96	102	106	102	96	96	122	12	90	101	101	102	114	123	114	102	101	162
13	93	99	99	99	99	108	112	105	99	124	13	93	104	104	104	106	124	131	117	105	162
14	94	99	99	99	99	106	118	116	102	122	14	94	105	105	105	105	118	142	138	111	156
15	95	100	100	100	100	103	121	124	111	117	15	95	105	104	104	104	111	146	153	127	146
16	94	98	98	98	98	99	118	126	116	109	16	94	102	102	102	102	103	142	159	138	131
17	93	98	96	96	96	96	112	124	117	99	17	93	102	99	99	99	99	131	154	142	112
18	91	97	93	93	93	93	101	112	110	89	18	91	102	94	94	94	94	111	132	129	94
19	87	87	87	87	87	87	87	87	87	80	19	87	87	87	87	87	87	87	88	88	80
20	85	85	85	85	85	85	85	85	85	78	20	85	85	85	85	85	85	85	85	85	78
21	83	83	83	83	83	83	83	83	83	76	21	83	83	83	83	83	83	83	83	83	76
22	81	81	81	81	81	81	81	81	81	74	22	81	81	81	81	81	81	81	81	81	74
23	79	79	79	79	79	79	79	79	79	72	23	79	79	79	79	79	79	79	79	79	72
24	77	77	77	77	77	77	77	77	77	70	24	77	77	77	77	77	77	77	77	77	70
Avg.	**83**	**86**	**88**	**90**	**90**	**87**	**90**	**90**	**88**	**90**	**Avg.**	**83**	**89**	**94**	**99**	**97**	**93**	**97**	**99**	**94**	**104**

Note: Sol-air temperatures are calculated based on $\varepsilon \Delta R/h_o$ = 7°F for horizontal surfaces and 0°F for vertical surfaces.

Assuming the rate of heat transfer can be expressed in terms of the sol-air temperature t_e:

$$q/A = h_o(t_e - t_s) \tag{7-5}$$

From Equations (7-4) and (7-5):

$$t_e = t_o + \alpha I_t/h_o - \varepsilon \Delta R/h_o \tag{7-6}$$

For horizontal surfaces that receive longwave radiation from the sky only, an appropriate value of ΔR is about 20 Btu/h·ft^2, so if $\varepsilon = 1$ and $h_o = 3.0$ Btu/h·ft^2·°F, the longwave correction term is about –7°F.

Because vertical surfaces receive longwave radiation from the ground and surrounding buildings, as well as from the sky, accurate ΔR values are difficult to determine. When solar radiation intensity is high, surfaces of terrestrial objects usually have a higher temperature than the outdoor air; thus, their longwave radiation compensates to some extent for the sky's low emittance. Therefore, it is assumed that $\Delta R = 0$ for vertical surfaces.

The sol-air temperatures in Table 7-1 are calculated based on $\varepsilon \Delta R/h_o = -7$°F for horizontal surfaces and 0°F for vertical surfaces; total solar intensity values for the calculation were the same as those used to evaluate the solar heat gain factors (SHGF) for July 21 at 40° N latitude. These values of I_t incorporate diffuse radiation from a clear sky and ground reflection but make no allowance for reflection from adjacent walls.

Table 7-2 Percentage of Daily Range

Time, h	%	Time, h	%	Time, h	%
1	87	9	71	17	10
2	92	10	56	18	21
3	96	11	39	19	34
4	99	12	23	20	47
5	100	13	11	21	58
6	98	14	3	22	68
7	93	15	0	23	76
8	84	16	3	24	82

Surface Colors. Sol-air temperature values are given for two values of the parameter α/h_o (Table 7-1); 0.15 is appropriate for a light-colored surface, while 0.30 is the usual maximum value for this parameter (i.e., for a dark-colored surface or any surface for which the permanent lightness cannot be reliably anticipated).

Air Temperature Cycle. The air temperature cycle used to calculate sol-air temperatures is given in Column 2, Table 7-1. These values are obtained by using the daily temperature range and the percent (%) difference from Table 7-2. Sol-air temperatures can be adjusted to any other air temperature cycle by adding or subtracting the difference between the desired air temperature and the air temperature value given in Column 2.

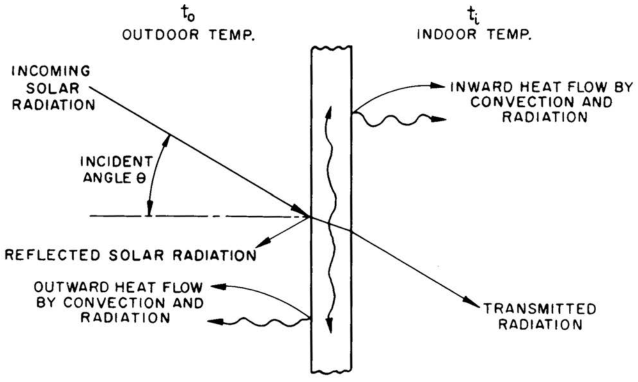


Fig. 7-2 Instantaneous Heat Balance for Sunlit Glazing Material
(Figure 18, Chapter 15, 2017 ASHRAE Handbook—Fundamentals)

Hourly Air Temperatures. The hourly air temperatures in Column 2, Table 7-1 are for a location with a design temperature of 95°F and a range of 21°F. To compute corresponding temperatures for other locations, select a suitable design temperature and note the outdoor daily range.

For each hour, take the percentage of the daily range indicated in Table 7-2 and subtract from the design temperature.

7.3.2 Heat Gain Through Fenestration

Fenestration is the term used here to designate any light-transmitting opening in a building wall or roof. The opening may be glazed with single or multiple sheet, plate or float glass, pattern glass, plastic panels, or glass block. Interior or exterior shading devices are usually employed, and some glazing systems incorporate integral sun control devices.

Calculating heat transfer through fenestration is explained in detail in Chapter 15 of the 2017 *ASHRAE Handbook—Fundamentals* while Chapter 18 presents only the portion of this operation required in the calculation of space cooling load due to heat transfer through fenestration.

Heat admission or loss through fenestration areas is affected by many factors, including

- Solar radiation intensity and incident angle
- Outdoor-indoor temperature difference
- Velocity and direction of airflow across the exterior and interior fenestration surfaces
- Low-temperature radiation exchange between the surfaces of the fenestration and the surroundings
- Exterior and/or interior shading

When solar radiation strikes an unshaded window (Figure 7-2), part of the radiant energy (8% for uncoated clear glass) is reflected back outdoors, part is absorbed within the glass (from 5 to 50%, depending upon the composition and thickness of the glass), and the remainder is transmitted directly indoors to become part of the cooling load. The solar heat gain is the sum of the transmitted radiation and the portion of the absorbed radiation that flows inward.

The total instantaneous rate of heat gain through a glazing material can be obtained from the heat balance between a unit area of fenestration and its thermal environment:

$$\boxed{\text{Total heat transmission through glass}} = \boxed{\text{Heat flow due to outdoor-indoor temp. difference}} + \boxed{\text{Inward flow of absorbed solar radiation}} + \boxed{\text{Radiation transmitted through glass}}$$

In this equation, the last two terms on the right are present only when the fenestration is irradiated and are therefore related to the incident radiation. The first term occurs whether or not the sun is shining, since it represents the heat flow through fenestration by thermal conduction.

Combining the last two terms,

$$\boxed{\text{Total heat transmission through glass}} = \boxed{\text{Conductive heat gain}} + \boxed{\text{Solar heat gain}}$$

In this way, heat gain is divided into two components: (1) the conductive heat gain (or loss), due to differences in outdoor and indoor air temperature, and (2) the solar heat gain (SHG), due to transmitted and absorbed solar energy. The total load through fenestration is the sum of the load due to conductive heat gain and the load due to solar heat gain.

Whether or not sunlight is present, heat flows through fenestration by thermal conduction, as expressed by

$$\boxed{\text{Conductive heat flow}} = \boxed{\text{Overall coefficient of heat transfer}} \times \boxed{\text{Outdoor-indoor temperature difference}}$$

or

$$q/A = U(t_o - t_i) \tag{7-7}$$

where

q/A = instantaneous rate of heat transfer through fenestration
U = overall coefficient of heat transfer for the glazing
t_o = outdoor air temperature
ti = inside air temperature

Values of the overall coefficient of heat transfer for a number of widely used fenestrations are in Chapter 15 of the 2017 *ASHRAE Handbook—Fundamentals* (also see Chapter 5). Table 7-3 includes some useful solar equations.

For fenestration heat gain, the following equations apply:

Direct beam solar heat gain q_b:

$$q_b = AE_D\text{SHGC}(\Theta)\text{IAC} \tag{7-8}$$

Diffuse solar heat gain q_d:

$$q_d = A(E_d + E_r)(\text{SHGC})_D\text{IAC} \tag{7-9}$$

Conductive heat gain q_c:

$$q_c = UA(T_{\text{out}} - T_{\text{in}}) \tag{7-10}$$

Total fenestration heat gain q:

Table 7-3 Solar Heat Gain

(Table 14, Chapter 30, 2005 ASHRAE Handbook—Fundamentals)

Solar Angles

All angles are in degrees. The solar azimuth ϕ and the surface azimuth ψ are measured in degrees from south; angles to the east of south are negative, and angles to the west of south are positive. Calculate solar altitude, azimuth, and surface incident angles as follows:

Apparent solar time AST, in decimal hours:

$$\text{AST} = \text{LST} + \text{ET}/60 + (\text{LSM} - \text{LON})/15$$

Hour angle H, degrees:

$$H = 15(\text{hours of time from local solar noon}) = 15(\text{AST} - 12)$$

Solar altitude β:

$$\sin\beta = \cos L \cos\delta \; \cos H + \sin L \sin\delta$$

Solar azimuth ϕ:

$$\cos\phi = (\sin\beta \; \sin L - \sin\delta)/(\cos\beta \; \cos L)$$

Surface-solar azimuth γ:

$$\gamma = \phi - \psi$$

Incident angle θ:

$$\cos\theta = \cos\beta \; \cos\gamma \; \sin\Sigma + \sin\beta \; \cos\Sigma$$

where

ET = equation of time, decimal minutes
L = latitude
LON = local longitude, decimal degrees of arc
LSM = local standard time meridian, decimal degrees of arc
= 60° for Atlantic Standard Time
= 75° for Eastern Standard Time
= 90° for Central Standard Time
= 105° for Mountain Standard Time
= 120° for Pacific Standard Time
= 135° for Alaska Standard Time
= 150° for Hawaii-Aleutian Standard Time
LST = local standard time, decimal hours
δ = solar declination, °
ψ = surface azimuth, °
Σ = surface tilt from horizontal, horizontal = 0°

Values of ET and δ are given in Table 2 of Chapter 35, 2011 *ASHRAE Handbook—HVAC Applications* for the 21st day of each month.

Direct, Diffuse, and Total Solar Irradiance

Direct normal irradiance E_{DN}

If $\beta > 0$ $$E_{DN} = \left[\frac{A}{\exp(B/\sin\beta)}\right]\text{CN}$$

Otherwise, $E_{DN} = 0$

Surface direct irradiance E_D

If $\cos\theta > 0$ $E_D = E_{DN}\cos\theta$

Otherwise, $E_D = 0$

Ratio Y of sky diffuse on vertical surface to sky diffuse on horizontal surface

If $\cos\theta > -0.2$ $Y = 0.55 + 0.437\cos\theta + 0.313\cos^2\theta$

Otherwise, $Y = 0.45$

Diffuse irradiance E_d

Vertical surfaces	$E_d = CYE_{DN}$
Surfaces other than vertical	$E_d = CE_{DN}(1 + \cos\Sigma)/2$
Ground-reflected irradiance	$E_r = E_{DN}(C + \sin\beta)\rho_g(1 - \cos\Sigma)/2$
Total surface irradiance	$E_t = E_D + E_d + E_r$

where

A = apparent solar constant
B = atmospheric extinction coefficient
C = sky diffuse factor
CN = clearness number multiplier for clear/dry or hazy/humid locations. See Figure 5 in Chapter 33 of the 2003 *ASHRAE Handbook—HVAC Applications* for CN values.
E_d = diffuse sky irradiance
E_r = diffuse ground-reflected irradiance
ρ_g = ground reflectivity

Values of A, B, and C are given in Table 1of Chapter 35, 2011 *ASHRAE Handbook—HVAC Applications* for the 21st day of each month. Values of ground reflectivity ρ_g are given in Table 10 of Chapter 31.

$$q = q_b + q_d + q_c \tag{7-11}$$

where

A = window area, ft²
E_D, E_d, and E_r = direct, diffuse, and ground-reflected irradiance, calculated using the equations in Table 7-3
SHGC(Θ) = direct solar heat gain coefficient as a function of incident angle q; may be interpolated between values in Table 7-4
$(\text{SHGC})_D$ = diffuse solar heat gain coefficient (also referred to as hemispherical SHGC); from Table 7-4
T_{in} = inside temperature, °F
T_{out} = outside temperature, °F
U = overall U-factor, including frame and mounting orientation from Table 5-16, Btu/h·ft²·°F
IAC = inside shading attenuation coefficient, = 1.0 if no inside shading device

If specific window manufacturer's SHGC and U-factor data are available, those should be used. For fenestration equipped with inside shading (blinds, drapes, or shades), IAC is listed in Table 7-8. The inside shading attenuation coefficients given are used to calculate both direct and diffuse solar heat gains.

Fenestration ratings (U-factor and SHGC) are based on the entire product area, including frames. Thus, for load calculations, fenestration area is the area of the entire opening in the wall or roof.

Nonuniform exterior shading, caused by roof overhangs, side fins, or building projections, requires separate hourly calculations for the externally shaded and unshaded areas of the window in question, with the inside shading SHGC still used to account for any internal shading devices. The areas, shaded and unshaded, depend on the location of the shadow line on a surface in the plane of the glass.

To account for the different types of fenestration and shading devices, used the inside shading attenuation coefficient IAC, which relates the solar heat gain through a glazing system under a specific set of conditions to the solar heat gain through the reference glazing material under the same conditions.

Most fenestration has some type of internal shading to provide privacy and aesthetic effects, as well as to give varying degrees of sun control. The IAC values and other glazing are given in Tables 7-4 though 7-11 for various fenestrations and shading device combinations.

Table 7-8 gives values of IAC (derived from measurements) for a variety of glazing and shading combinations.

Table 7-4 Visible Transmittance (T_v), Solar Heat Gain Coefficient (SHGC), Solar Transmittance (T), Front Reflectance (R^f), Back Reflectance (R^b), and Layer Absorptances (A^f_n) for Glazing and Window Systems

(From Table 10, Chapter 15, 2017 ASHRAE Handbook—Fundamentals)

					Center-of-Glazing Properties							Total Window SHGC at Normal Incidence				Total Window T_v at Normal Incidence			
	Glazing System				Incidence Angles							Aluminum		Other Frames		Aluminum		Other Frames	
ID	Glass Thick., in.		Center Glazing T_v		Normal 0.00	40.00	50.00	60.00	70.00	80.00	Hemis., Diffuse	Operable	Fixed	Operable	Fixed	Operable	Fixed	Operable	Fixed
Uncoated Single Glazing																			
1a	1/8	CLR	0.90	SHGC	0.86	0.84	0.82	0.78	0.67	0.42	0.78	0.78	0.79	0.70	0.76	0.80	0.81	0.72	0.79
				T	0.83	0.82	0.80	0.75	0.64	0.39	0.75								
				R^f	0.08	0.08	0.10	0.14	0.25	0.51	0.14								
				R^b	0.08	0.08	0.10	0.14	0.25	0.51	0.14								
				A^f_1	0.09	0.10	0.10	0.11	0.11	0.11	0.10								
1b	1/4	CLR	0.88	SHGC	0.81	0.80	0.78	0.73	0.62	0.39	0.73	0.74	0.74	0.66	0.72	0.78	0.79	0.70	0.77
				T	0.77	0.75	0.73	0.68	0.58	0.35	0.69								
				R^f	0.07	0.08	0.09	0.13	0.24	0.48	0.13								
				R^b	0.07	0.08	0.09	0.13	0.24	0.48	0.13								
				A^f_1	0.16	0.17	0.18	0.19	0.19	0.17	0.17								
1c	1/8	BRZ	0.68	SHGC	0.73	0.71	0.68	0.64	0.55	0.34	0.65	0.67	0.67	0.59	0.65	0.61	0.61	0.54	0.60
				T	0.65	0.62	0.59	0.55	0.46	0.27	0.56								
				R^f	0.06	0.07	0.08	0.12	0.22	0.45	0.12								
				R^b	0.06	0.07	0.08	0.12	0.22	0.45	0.12								
				A^f_1	0.29	0.31	0.32	0.33	0.33	0.29	0.31								
1d	1/4	BRZ	0.54	SHGC	0.62	0.59	0.57	0.53	0.45	0.29	0.54	0.57	0.57	0.50	0.55	0.48	0.49	0.43	0.48
				T	0.49	0.45	0.43	0.39	0.32	0.18	0.41								
				R^f	0.05	0.06	0.07	0.11	0.19	0.42	0.10								
				R^b	0.05	0.68	0.66	0.62	0.53	0.33	0.10								
				A^f_1	0.46	0.49	0.50	0.51	0.49	0.41	0.48								
1e	1/8	GRN	0.82	SHGC	0.70	0.68	0.66	0.62	0.53	0.33	0.63	0.64	0.64	0.57	0.62	0.73	0.74	0.66	0.72
				T	0.61	0.58	0.56	0.52	0.43	0.25	0.53								
				R^f	0.06	0.07	0.08	0.12	0.21	0.45	0.11								
				R^b	0.06	0.07	0.08	0.12	0.21	0.45	0.11								
				A^f_1	0.33	0.35	0.36	0.37	0.36	0.31	0.35								
1f	1/4	GRN	0.76	SHGC	0.60	0.58	0.56	0.52	0.45	0.29	0.54	0.55	0.55	0.49	0.53	0.68	0.68	0.61	0.67
				T	0.47	0.44	0.42	0.38	0.32	0.18	0.40								
				R^f	0.05	0.06	0.07	0.11	0.20	0.42	0.10								
				R^b	0.05	0.06	0.07	0.11	0.20	0.42	0.10								
				A^f_1	0.47	0.50	0.51	0.51	0.49	0.40	0.49								
1g	1/8	GRY	0.62	SHGC	0.70	0.68	0.66	0.61	0.53	0.33	0.63	0.64	0.64	0.57	0.62	0.55	0.56	0.50	0.55
				T	0.61	0.58	0.56	0.51	0.42	0.24	0.53								
				R^f	0.06	0.07	0.08	0.12	0.21	0.44	0.11								
				R^b	0.06	0.07	0.08	0.12	0.21	0.44	0.11								
				A^f_1	0.33	0.36	0.37	0.37	0.37	0.32	0.35								
1h	1/4	GRY	0.46	SHGC	0.59	0.57	0.55	0.51	0.44	0.28	0.52	0.54	0.54	0.48	0.52	0.41	0.41	0.37	0.40
				T	0.46	0.42	0.40	0.36	0.29	0.16	0.38								
				R^f	0.05	0.06	0.07	0.10	0.19	0.41	0.10								
				R^b	0.05	0.06	0.07	0.10	0.19	0.41	0.10								
				A^f_1	0.49	0.52	0.54	0.54	0.52	0.43	0.51								
1i	1/4	BLUGRN	0.75	SHGC	0.62	0.59	0.57	0.54	0.46	0.30	0.55	0.57	0.57	0.50	0.55	0.67	0.68	0.60	0.66
				T	0.49	0.46	0.44	0.40	0.33	0.19	0.42								
				R^f	0.06	0.06	0.07	0.11	0.20	0.43	0.11								
				R^b	0.06	0.06	0.07	0.11	0.20	0.43	0.11								
				A^f_1	0.45	0.48	0.49	0.49	0.47	0.38	0.48								
Reflective Single Glazing																			
1j	1/4	SS on CLR 8%	0.08	SHGC	0.19	0.19	0.19	0.18	0.16	0.10	0.18	0.18	0.18	0.16	0.17	0.07	0.07	0.06	0.07
				T	0.06	0.06	0.06	0.05	0.04	0.03	0.05								
				R^f	0.33	0.34	0.35	0.37	0.44	0.61	0.36								
				R^b	0.50	0.50	0.51	0.53	0.58	0.71	0.52								
				A^f_1	0.61	0.61	0.60	0.58	0.52	0.37	0.57								
1k	1/4	SS on CLR 14%	0.14	SHGC	0.25	0.25	0.24	0.23	0.20	0.13	0.23	0.24	0.24	0.21	0.22	0.12	0.13	0.11	0.12
				T	0.11	0.10	0.10	0.09	0.07	0.04	0.09								
				R^f	0.26	0.27	0.28	0.31	0.38	0.57	0.30								
				R^b	0.44	0.44	0.45	0.47	0.52	0.67	0.46								

Table 7-4 Visible Transmittance (T_v), Solar Heat Gain Coefficient (SHGC), Solar Transmittance (T), Front Reflectance (R^f), Back Reflectance (R^b), and Layer Absorptances (A_n^f) for Glazing and Window Systems (*Continued*)

(From Table 10, Chapter 15, 2017 ASHRAE Handbook—Fundamentals)

ID	Glazing System: Glass Thick., in.	Glazing System	Center Glazing T_v		Center-of-Glazing Properties, Incidence Angles: Normal 0.00	40.00	50.00	60.00	70.00	80.00	Hemis., Diffuse	Total Window SHGC at Normal Incidence: Aluminum Operable	Aluminum Fixed	Other Frames Operable	Other Frames Fixed	Total Window T_v at Normal Incidence: Aluminum Operable	Aluminum Fixed	Other Frames Operable	Other Frames Fixed
				A_1^f	0.63	0.63	0.62	0.60	0.55	0.39	0.60								
1l	1/4	SS on CLR 20%	0.20	SHGC	0.31	0.30	0.30	0.28	0.24	0.16	0.28	0.29	0.29	0.26	0.28	0.18	0.18	0.16	0.18
				T	0.15	0.15	0.14	0.13	0.11	0.06	0.13								
				R^f	0.21	0.22	0.23	0.26	0.34	0.54	0.25								
				R^b	0.38	0.38	0.39	0.41	0.48	0.64	0.41								
				A_1^f	0.64	0.64	0.63	0.61	0.56	0.40	0.60								
1m	1/4	SS on GRN 14%	0.12	SHGC	0.25	0.25	0.24	0.23	0.21	0.14	0.23	0.24	0.24	0.21	0.22	0.11	0.11	0.10	0.11
				T	0.06	0.06	0.06	0.06	0.04	0.03	0.06								
				R^f	0.14	0.14	0.16	0.19	0.27	0.49	0.18								
				R^b	0.44	0.44	0.45	0.47	0.52	0.67	0.46								
				A_1^f	0.80	0.80	0.78	0.76	0.68	0.48	0.75								
1n	1/4	TI on CLR 20%	0.20	SHGC	0.29	0.29	0.28	0.27	0.23	0.15	0.27	0.27	0.27	0.24	0.26	0.18	0.18	0.16	0.18
				T	0.14	0.13	0.13	0.12	0.09	0.06	0.12								
				R^f	0.22	0.22	0.24	0.26	0.34	0.54	0.26								
				R^b	0.40	0.40	0.42	0.44	0.50	0.65	0.43								
				A_1^f	0.65	0.65	0.64	0.62	0.57	0.40	0.62								
1o	1/4	TI on CLR 30%	0.30	SHGC	0.39	0.38	0.37	0.35	0.30	0.20	0.35	0.36	0.36	0.32	0.35	0.27	0.27	0.24	0.26
				T	0.23	0.22	0.21	0.19	0.16	0.09	0.20								
				R^f	0.15	0.15	0.17	0.20	0.28	0.50	0.19								
				R^b	0.32	0.33	0.34	0.36	0.43	0.60	0.36								
				A_1^f	0.63	0.65	0.64	0.62	0.57	0.40	0.62								
Uncoated Double Glazing																			
5a	1/8	CLR CLR	0.81	SHGC	0.76	0.74	0.71	0.64	0.50	0.26	0.66	0.69	0.70	0.62	0.67	0.72	0.73	0.65	0.71
				T	0.70	0.68	0.65	0.58	0.44	0.21	0.60								
				R^f	0.13	0.14	0.16	0.23	0.36	0.61	0.21								
				R^b	0.13	0.14	0.16	0.23	0.36	0.61	0.21								
				A_1^f	0.10	0.11	0.11	0.12	0.13	0.13	0.11								
				A_2^f	0.07	0.08	0.08	0.08	0.07	0.05	0.07								
5b	1/4	CLR CLR	0.78	SHGC	0.70	0.67	0.64	0.58	0.45	0.23	0.60	0.64	0.64	0.57	0.62	0.69	0.70	0.62	0.69
				T	0.61	0.58	0.55	0.48	0.36	0.17	0.51								
				R^f	0.11	0.12	0.15	0.20	0.33	0.57	0.18								
				R^b	0.11	0.12	0.15	0.20	0.33	0.57	0.18								
				A_1^f	0.17	0.18	0.19	0.20	0.21	0.20	0.19								
				A_2^f	0.11	0.12	0.12	0.12	0.10	0.07	0.11								
5c	1/8	BRZ CLR	0.62	SHGC	0.62	0.60	0.57	0.51	0.39	0.20	0.53	0.57	0.57	0.50	0.55	0.55	0.56	0.50	0.55
				T	0.55	0.51	0.48	0.42	0.31	0.14	0.45								
				R^f	0.09	0.10	0.12	0.16	0.27	0.49	0.15								
				R^b	0.12	0.13	0.15	0.21	0.35	0.59	0.19								
				A_1^f	0.30	0.33	0.34	0.36	0.37	0.34	0.33								
				A_2^f	0.06	0.06	0.06	0.06	0.05	0.03	0.06								
5d	1/4	BRZ CLR	0.47	SHGC	0.49	0.46	0.44	0.39	0.31	0.17	0.41	0.45	0.45	0.40	0.43	0.42	0.42	0.38	0.41
				T	0.38	0.35	0.32	0.27	0.20	0.08	0.30								
				R^f	0.07	0.08	0.09	0.13	0.22	0.44	0.12								
				R^b	0.10	0.11	0.13	0.19	0.31	0.55	0.17								
				A_1^f	0.48	0.51	0.52	0.53	0.53	0.45	0.50								
				A_2^f	0.07	0.07	0.07	0.07	0.06	0.04	0.07								
5e	1/8	GRN CLR	0.75	SHGC	0.60	0.57	0.54	0.49	0.38	0.20	0.51	0.55	0.55	0.49	0.53	0.67	0.68	0.60	0.66
				T	0.52	0.49	0.46	0.40	0.30	0.13	0.43								
				R^f	0.09	0.10	0.12	0.16	0.27	0.50	0.15								
				R^b	0.12	0.13	0.15	0.21	0.35	0.60	0.19								
				A_1^f	0.34	0.37	0.38	0.39	0.39	0.35	0.37								
				A_2^f	0.05	0.05	0.05	0.04	0.04	0.03	0.04								
5f	1/4	GRN CLR	0.68	SHGC	0.49	0.46	0.44	0.39	0.31	0.17	0.41	0.45	0.45	0.40	0.43	0.61	0.61	0.54	0.60
				T	0.39	0.36	0.33	0.29	0.21	0.09	0.31								
				R^f	0.08	0.08	0.10	0.14	0.23	0.45	0.13								
				R^b	0.10	0.11	0.13	0.19	0.31	0.55	0.17								

Table 7-4 Visible Transmittance (T_v), Solar Heat Gain Coefficient (SHGC), Solar Transmittance (T), Front Reflectance (R^f), Back Reflectance (R^b), and Layer Absorptances (A_n^f) for Glazing and Window Systems (*Continued*)

(From Table 10, Chapter 15, 2017 ASHRAE Handbook—Fundamentals)

					Center-of-Glazing Properties							Total Window SHGC at Normal Incidence				Total Window T_v at Normal Incidence			
	Glazing System				Incidence Angles							Aluminum		Other Frames		Aluminum		Other Frames	
ID	Glass Thick., in.		Center Glazing T_v		Normal 0.00	40.00	50.00	60.00	70.00	80.00	Hemis., Diffuse	Operable	Fixed	Operable	Fixed	Operable	Fixed	Operable	Fixed
				A_1^f	0.49	0.51	0.05	0.53	0.52	0.43	0.50								
				A_2^f	0.05	0.05	0.05	0.05	0.04	0.03	0.05								
5g	1/8	GRY CLR	0.56	SHGC	0.60	0.57	0.54	0.48	0.37	0.20	0.51	0.55	0.55	0.49	0.53	0.50	0.50	0.45	0.49
				T	0.51	0.48	0.45	0.39	0.29	0.12	0.42								
				R^f	0.09	0.09	0.11	0.16	0.26	0.48	0.14								
				R^b	0.12	0.13	0.15	0.21	0.34	0.59	0.19								
				A_1^f	0.34	0.37	0.39	0.40	0.41	0.37	0.37								
				A_2^f	0.05	0.06	0.06	0.05	0.05	0.03	0.05								
5h	1/4	GRY CLR	0.41	SHGC	0.47	0.44	0.42	0.37	0.29	0.16	0.39	0.43	0.43	0.38	0.42	0.36	0.37	0.33	0.36
				T	0.36	0.32	0.29	0.25	0.18	0.07	0.28								
				R^f	0.07	0.07	0.08	0.12	0.21	0.43	0.12								
				R^b	0.10	0.11	0.13	0.18	0.31	0.55	0.17								
				A_1^f	0.51	0.54	0.56	0.57	0.56	0.47	0.53								
				A_2^f	0.07	0.07	0.07	0.06	0.05	0.03	0.06								
5i	1/4	BLUGRN CLR	0.67	SHGC	0.50	0.47	0.45	0.40	0.32	0.17	0.43	0.46	0.46	0.41	0.44	0.60	0.60	0.54	0.59
				T	0.40	0.37	0.34	0.30	0.22	0.10	0.32								
				R^f	0.08	0.08	0.10	0.14	0.24	0.46	0.13								
				R^b	0.11	0.11	0.14	0.19	0.31	0.55	0.17								
				A_1^f	0.47	0.49	0.50	0.51	0.50	0.42	0.48								
				A_2^f	0.06	0.06	0.06	0.05	0.04	0.03	0.05								
5j	1/4	HI-P GRN CLR	0.59	SHGC	0.39	0.37	0.35	0.31	0.25	0.14	0.33	0.36	0.36	0.32	0.35	0.53	0.53	0.47	0.52
				T	0.28	0.26	0.24	0.20	0.15	0.06	0.22								
				R^f	0.06	0.07	0.08	0.12	0.21	0.43	0.11								
				R^b	0.10	0.11	0.13	0.19	0.31	0.55	0.17								
				A_1^f	0.62	0.65	0.65	0.65	0.62	0.50	0.63								
				A_2^f	0.03	0.03	0.03	0.03	0.02	0.01	0.03								
Reflective Double Glazing																			
5k	1/4	SS on CLR 8%, CLR	0.07	SHGC	0.13	0.12	0.12	0.11	0.10	0.06	0.11	0.13	0.13	0.11	0.12	0.06	0.06	0.06	0.06
				T	0.05	0.05	0.04	0.04	0.03	0.01	0.04								
				R^f	0.33	0.34	0.35	0.37	0.44	0.61	0.37								
				R^b	0.38	0.37	0.38	0.40	0.46	0.61	0.40								
				A_1^f	0.61	0.61	0.60	0.58	0.53	0.37	0.56								
				A_2^f	0.01	0.01	0.01	0.01	0.01	0.01	0.01								
5l	1/4	SS on CLR 14%, CLR	0.13	SHGC	0.17	0.17	0.16	0.15	0.13	0.08	0.16	0.17	0.16	0.14	0.15	0.12	0.12	0.10	0.11
				T	0.08	0.08	0.08	0.07	0.05	0.02	0.07								
				R^f	0.26	0.27	0.28	0.31	0.38	0.57	0.30								
				R^b	0.34	0.33	0.34	0.37	0.44	0.60	0.36								
				A_1^f	0.63	0.64	0.64	0.63	0.61	0.56	0.60								
				A_2^f	0.02	0.02	0.02	0.02	0.02	0.02	0.02								
5m	1/4	SS on CLR 20%, CLR	0.18	SHGC	0.22	0.21	0.21	0.19	0.16	0.09	0.20	0.21	0.21	0.18	0.20	0.16	0.16	0.14	0.16
				T	0.12	0.11	0.11	0.09	0.07	0.03	0.10								
				R^f	0.21	0.22	0.23	0.26	0.34	0.54	0.25								
				R^b	0.30	0.30	0.31	0.34	0.41	0.59	0.33								
				A_1^f	0.64	0.64	0.63	0.62	0.57	0.41	0.61								
				A_2^f	0.03	0.03	0.03	0.03	0.02	0.02	0.03								
5n	1/4	SS on GRN 14%, CLR	0.11	SHGC	0.16	0.16	0.15	0.14	0.12	0.08	0.14	0.16	0.16	0.14	0.14	0.10	0.10	0.09	0.10
				T	0.05	0.05	0.05	0.04	0.03	0.01	0.04								
				R^f	0.14	0.14	0.16	0.19	0.27	0.49	0.18								
				R^b	0.34	0.33	0.34	0.37	0.44	0.60	0.36								
				A_1^f	0.80	0.80	0.79	0.76	0.69	0.49	0.76								
				A_2^f	0.01	0.01	0.01	0.01	0.01	0.01	0.01								
5o	1/4	TI on CLR 20%, CLR	0.18	SHGC	0.21	0.20	0.19	0.18	0.15	0.09	0.18	0.20	0.20	0.18	0.19	0.16	0.16	0.14	0.16
				T	0.11	0.10	0.10	0.08	0.06	0.03	0.09								
				R^f	0.22	0.22	0.24	0.27	0.34	0.54	0.26								

Table 7-4 Visible Transmittance (T_v), Solar Heat Gain Coefficient (SHGC), Solar Transmittance (T), Front Reflectance (R^f), Back Reflectance (R^b), and Layer Absorptances (A_n^f) for Glazing and Window Systems (*Continued*)

(From Table 10, Chapter 15, 2017 ASHRAE Handbook—Fundamentals)

					Center-of-Glazing Properties							Total Window SHGC at Normal Incidence				Total Window T_v at Normal Incidence			
	Glazing System				Incidence Angles							Aluminum		Other Frames		Aluminum		Other Frames	
ID	Glass Thick., in.		Center Glazing T_v		Normal 0.00	40.00	50.00	60.00	70.00	80.00	Hemis., Diffuse	Operable	Fixed	Operable	Fixed	Operable	Fixed	Operable	Fixed
				R^b	0.32	0.31	0.32	0.35	0.42	0.59	0.35								
				A_1^f	0.65	0.66	0.65	0.63	0.58	0.41	0.62								
				A_2^f	0.02	0.02	0.02	0.02	0.02	0.01	0.02								
5p	1/4	TI on CLR 30%, CLR	0.27	SHGC	0.29	0.28	0.27	0.25	0.20	0.12	0.25	0.27	0.27	0.24	0.26	0.24	0.24	0.22	0.24
				T	0.18	0.17	0.16	0.14	0.10	0.05	0.15								
				R^f	0.15	0.15	0.17	0.20	0.29	0.51	0.19								
				R^b	0.27	0.27	0.28	0.31	0.40	0.58	0.31								
				A_1^f	0.64	0.64	0.63	0.62	0.58	0.43	0.61								
				A_2^f	0.04	0.04	0.04	0.04	0.03	0.02	0.04								

KEY:

CLR = clear, BRZ = bronze, GRN = green, GRY = gray, BLUGRN = blue-green, SS = stainless steel reflective coating, TI = titanium reflective coating

Reflective coating descriptors include percent visible transmittance as *x*%.

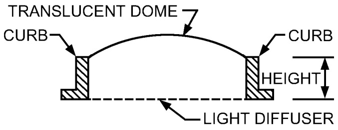
HI-P GRN = high-performance green tinted glass, LE = low-emissivity coating

T_v = visible transmittance, T = solar transmittance, SHGC = solar heat gain coefficient, and H. = hemispherical SHGC

ID #s refer to U-factors in Table 4, except for products 49 and 50.

Table 7-5a Solar Heat Gain Coefficients for Domed Horizontal Skylights

(Table 11, Chapter 15, 2017 ASHRAE Handbook—Fundamentals)



Dome	Light Diffuser (Translucent)	Curb Height, in.	Curb Width-to-Height Ratio	Solar Heat Gain Coefficient	Visible Transmittance
Clear	Yes	0	∞	0.53	0.56
$\tau = 0.86$	$\tau = 0.58$	9	5	0.50	0.58
		12	2.5	0.44	0.59
Clear		0	∞	0.86	0.91
$\tau = 0.86$	None	9	5	0.77	0.91
		12	2.5	0.70	0.91
Translucent		0	∞	0.50	0.46
$\tau = 0.52$	None	12	2.5	0.40	0.32
Translucent		0	∞	0.30	0.25
$\tau = 0.27$	None	9	5	0.26	0.21
		12	2.5	0.24	0.18

Source: Laouadi et al. (2003), Schutrum and Ozisik (1961).

Table 7-5b Performance Characteristics of Typical TDDs

Glazing of Collector/Diffuser	Single/ Single Layer	Double/ Double Layer	Double/ Single Layer	Single/ Double Layer	Single/ Triple Layer
Visible transmittance*	0.77	0.63	0.72	0.66	0.58
SHGC* (10 in. insulation above ceiling)	0.32	0.24	0.27	0.28	0.24
SHGC* (10 in. insulation under roof)	0.34	0.33	0.32	0.35	0.34
U-factor, Btu/h· ft^2·°F (10 in. insulation above ceiling)	0.62	0.38	0.62	0.38	0.27
U-factor (10 in. insulation under roof)	1.34	0.83	0.83	1.34	1.33

*VT and SHGC calculated at solar incidence angle of 50° at due south; this angle is close to annual averaged incidence angle (48.84°) of NFRC *Standard* 203-2014.

The IAC bears a certain similarity to the shading coefficient. There is, however, an important difference: we must calculate the solar heat flux through the unshaded glazing at the appropriate angle before applying the IAC. With the shading coefficient, only the angular dependence of single glazing was included (through the now-discarded SHGF). The effectiveness of any internal shading device depends on its ability to reflect incoming solar radiation back through the fenestration before it can be absorbed and converted into heat within the building. Table 7-10 lists approximate values of solar-optical properties for the typical indoor shading devices described in Tables 7-8 and 7-9.

Skylights. Skylight solar heat gain strongly depends on the configuration of the space below or adjacent to (i.e., in sloped applications) the skylight formed by the skylight curb and any associated light well.

Five aspects must be considered: (1) transmittance and absorptance of the skylight unit, (2) transmitted solar flux that reaches the aperture of the light well, (3) whether that aperture is covered by a diffuser, (4) transmitted solar flux that strikes the walls of the light well, and (5) reflectance of the walls of the light well. Data for flat skylights, which may be considered as sloped glazings, are found in Tables 5-16 and 7-5a.

The following categories of skylights all admit daylight from the roof plane but differ markedly in their design.

Domed Skylights. Solar and total heat gains for domed skylights can be determined by the same procedure used for windows. Table 7-5a gives SHGCs for plastic domed sky-

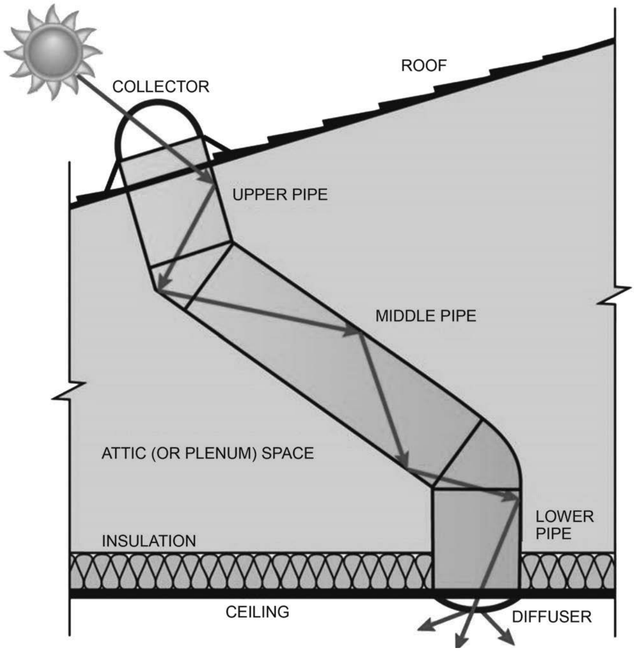


Fig. 7-3 Generalized Tubular Daylighting Device
(Figure 16, Chapter 15, 2017 ASHRAE Handbook—Fundamentals))

lights at normal incidence (Shutrum and Ozisik 1961). Manufacturers' literature has further details. Given the poorly defined incident angle conditions for domed skylights, it is best to use these values without correction for incident angle, together with the correct (angle-dependent) value of incident solar irradiance. Results should be considered approximate. In the absence of other data, these values may also be used to make estimates for skylights on slanted roofs.

Tubular Daylighting Devices (TDDs). Tubular daylighting devices collect and channel daylight (diffuse sky and sunbeam light) from the roof of a building into interior spaces (Figure 7-3). TDDs are also known as *mirror light pipes* or *tubular skylights.* They are an alternative to conventional skylights and have the advantages of energy savings (small area relative to the amount of useful light they admit), lower solar heat gains, relative ease of installation, and tube lengths (depending on the tube length-to-diameter ratio) of up to 42 ft {13 m}. TDD technologies undergo continuous development to meet the increasing needs for high standards of energy efficiency in buildings and glare-free lighting. The energy-saving potential of TDDs is well recognized (Allen 1997; Carter 2008; Laouadi 2004), and TDDs are encouraged or mandated, particularly in commercial buildings [CEC 2010; see also U.S. Green Building Council's (USGBC) Leadership in Energy and Environmental Design (LEED) program]. A variant known as a hybrid TDD (HTDD) is available from some manufacturers. In an HTDD, more than one material or geometry is present along the length of the tube (e.g., a round tube that transitions to a square ceiling diffuser).

The natural daylight that TDDs deliver is beneficial to the visual comfort, health, and well-being of building occupants (Boyce et al. 2003; Carter and Al-Marwaee 2009). The daylighting performance (light output) of a TDD depends on many location and material parameters, but a typical device can illuminate an area of up to 300 ft^2. The area of coverage depends on the height of the ceiling: the higher the ceiling, the more widely the light will be uniformly distributed.

TDD Components. TDDs typically consist of (1) a collector on the roof to gather sunbeams and diffuse sky light, (2) a hollow pipe guide in the plenum/attic space to channel light downwards, and (3) a light diffuser at ceiling level to spread light indoors (see Figure 7-3).

Table 7-5b shows the computed optical and thermal characteristics of a typical TDD made up of a 1/8 in. PMMA collector and a 1/8 in. polycarbonate diffuser with an NFRC standard-size pipe of 14 in. diameter and aspect ratio of 2.14. The tube reflectivity is 99% and 76% for the visible and solar spectrum, respectively. The NFRC Technical Document 100 boundary conditions (NFRC 2014) are used for the U-factor calculation. The table shows that, although the number of layers in the collector or diffuser has only a small effect on visible transmittance, as expected it has a significant effect on thermal performance. The table does not show any best arrangement, but provides information that can help a designer select an arrangement that best suits a particular application in residential or commercial buildings.

During certain seasons of the year and for some exposures, horizontal projections can result in considerable reductions in solar heat gain by providing shade. This is particularly applicable to south, southeast, and southwest exposures during the late spring, summer, and early fall. On east and west exposures during the entire year, and on southerly exposures during the winter, the solar altitude is generally so low that horizontal projections, in order to be effective, would have to be excessively long.

A vertical fin or projection P_V required to produce a given shadow width S_V on a window or wall for any given time of day and year is related to the vertical surface-solar azimuth (see Table 7-12) by

$$P_V = S_V / |\tan\gamma| \tag{7-12}$$

The horizontal projection P_H required to produce a given shadow height S_H on a window or wall for any time of day or year is related to the profile angle Ω(Table 7-13) by

$$P_H = S_H / \tan\Omega \tag{7-13}$$

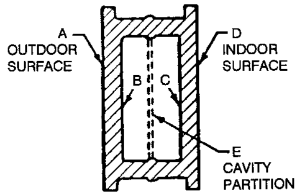
The profile angle, also called the vertical shadow line angle, is the angle between a line perpendicular to the plane of the window and the projection of the earth-sun line in a vertical plane that is also normal to the window and is given by

$$\tan\Omega = \tan\beta / \cos\gamma$$

The use of an overhang to shade glass is an excellent way of reducing the solar heat gain. Use the values for north glass when reducing solar heat gain for a shaded window. North glass does not have sun shining directly on it, so the SHGFs listed for it are based on only diffuse solar radiation (as on shaded glass). If a window is partially shaded, use the north

Table 7-6 Solar Heat Gain Coefficients and U-Factors for Standard Hollow Glass Block Wall Panels

(Table 13, Chapter 15, 2017 ASHRAE Handbook—Fundamentals)

	Type of Glass Block[a]	Description of Glass Block	Solar Heat Gain Coefficient		U-Factor,[c] Btu/(h·ft^2·°F)
			In Sun	In Shade[b]	
	Type I	Glass colorless or aqua A, D: Smooth B, C: Smooth or wide ribs, or flutes horizontal or vertical, or shallow configuration E: None	0.57	0.35	0.51
	Type IA	Same as Type I except ceramic enamel on A	0.23	0.17	0.51
	Type II	Same as Type I except glass fiber screen partition E	0.38	0.30	0.48
	Type III	Glass colorless or aqua A, D: Narrow vertical ribs or flutes. B, C: Horizontal light-diffusing prisms, or horizontal light-directing prisms E: Glass fiber screen	0.29	0.23	0.48
	Type IIIA	Same as Type III except E: Glass fiber screen with green ceramic spray coating or glass fiber screen and gray glass or glass fiber screen with light-selecting prisms	0.22	0.16	0.48
	Type IV	Same as Type I except reflective oxide coating on A	0.14	0.10	0.51

[a] All values are for 7 3/4 by 7 3/4 by 3 7/8 in. block, set in light-colored mortar. For 11 3/4 by 11 3/4 by 3 7/8 in. block, increase coefficients by 15%, and for 5 3/4 by 5 3/4 by 3 7/8 in. block reduce coefficients by 15%.

[b] For NE, E, and SE panels in shade, add 50% to values listed for panels in shade.

[c] Values shown are identical for all size block.

Table 7-7 Unshaded Fractions (F_u) and Exterior Solar Attenuation Coefficients (EAC) for Louvered Sun Screens

Profile Angle	Group 1			Group 2			Group 3			Group 4		
	Transmittance	F_u	EAC	Transmittance	F_u	EAC	Transmittance	F_u	EAC	Transmittance	F_u	EAC
10°	0.23	0.20	0.15	0.25	0.13	0.02	0.4	0.33	0.18	0.48	0.29	0.3
20°	0.06	0.02	0.15	0.14	0.03	0.02	0.32	0.24	0.18	0.39	0.2	0.3
30°	0.04	0.00	0.15	0.12	0.01	0.02	0.21	0.13	0.18	0.28	0.08	0.3
≥ 40°	0.04	0.00	0.15	0.11	0.00	0.02	0.07	0.00	0.18	0.2	0.00	0.3

Group 1: Black, width over spacing ratio 1.15/1; 23 louvers/in. Group 2: Light color; high reflectance, otherwise same as Group 1. Group 3: Black or dark color; w/s ratio 0.85/1; 17 louvers/in. Group 4: Light color or unpainted aluminum; high reflectance; otherwise same as Group 3. U-factor = 0.85 Btu/(h·ft^2·°F) for all groups when used with single glazing.

value of SHGF for the shaded portion and the regular (east, west, south, etc.) SHGF for the rest of the window.

A window with a significant depth of reveal generally has part of its glass area shaded by the mullions and the transom. The area that is shaded varies throughout the day, but it can be estimated by treating mullions and transoms as vertical and horizontal projections.

The width S_V of the shadow from a mullion that projects a distance P_V beyond the plane of the glass is

$$S_V = P_V|\tan\gamma| \tag{7-14}$$

where γ is the wall-solar azimuth. When γ is greater than 90°, the entire window is in the shade.

Similarly, the height of the shadow cast by a transom that projects P_H beyond the plane of the glass is

$$S_H = P_H\tan\beta/\cos\gamma \tag{7-15}$$

where β is the solar altitude. The solar angles are shown in Figure 7-3.

Thus, the sunlit area of the window is

$$\text{Area in Sun} = (\text{Width} - S_V)(\text{Height} - S_H) \tag{7-16}$$

7.3.3 Heat Gain Through Interior Surfaces

Whenever a conditioned space is adjacent to a space with a different temperature, heat transfer through the separating physical section must be considered. The heat transfer rate q, in Btu/h, is given by

$$q = UA(t_b - t_i) \tag{7-17}$$

where

U = coefficient of overall heat transfer between the adjacent and the conditioned space, Btu/h·ft^2·°F

A = area of separating section concerned, ft^2

t_b = average air temperature in adjacent space, °F

t_i = air temperature in conditioned space, °F

Temperature t_b may have any value over a considerable range according to conditions in the adjacent space. The temperature in a kitchen or boiler room may be as much as 15°F to 50°F above the outdoor air temperature. The actual temperatures in adjoining spaces should be measured wherever practicable; where nothing is known except that the adjacent space is of conventional construction and contains no heat sources, $t_b - t_i$ may be considered to be the difference

Table 7-8 Interior Solar Attenuation Coefficients (IAC) for Single or Double Glazings Shaded by Interior Venetian Blinds or Roller Shades

Glazing System[a]	Nominal Thickness[b] Each Pane, in.	Glazing Solar Transmittance[b]		Glazing SHGC[b]	IAC				
					Venetian Blinds		Roller Shades		
		Outer Pane	Single or Inner Pane		Medium	Light	Opaque Dark	Opaque White	Translucent Light
Single Glazing Systems									
Clear, residential	1/8[c]		0.87 to 0.80	0.86	0.75[d]	0.68[d]	0.82	0.40	0.45
Clear, commercial	1/4 to 1/2		0.80 to 0.71	0.82					
Clear, pattern	1/8 to 1/2		0.87 to 0.79						
Heat absorbing, pattern	1/8			0.59					
Tinted	3/16, 7/32		0.74, 0.71						
Above glazings, automated blinds[e]				0.86	0.64	0.59			
Above glazings, tightly closed vertical blinds				0.85	0.30	0.26			
Heat absorbing[f]	1/4		0.46	0.59	0.84	0.78	0.66	0.44	0.47
Heat absorbing, pattern	1/4								
Tinted	1/8, 1/4		0.59, 0.45						
Heat absorbing or pattern			0.44 to 0.30	0.59	0.79	0.76	0.59	0.41	0.47
Heat absorbing	3/8		0.34						
Heat absorbing or pattern			0.29 to 0.15						
			0.24	0.37	0.99	0.94	0.85	0.66	0.73
Reflective coated glass				0.26 to 0.52	0.83	0.75			
Double Glazing Systems[g]									
Clear double, residential	1/8	0.87	0.87	0.76	0.71[d]	0.66[d]	0.81	0.40	0.46
Clear double, commercial	1/4	0.80	0.80	0.70					
Heat absorbing double[f]	1/4	0.46	0.8	0.47	0.72	0.66	0.74	0.41	0.55
Reflective double				0.17 to 0.35	0.90	0.86			
Other Glazings (Approximate)					0.83	0.77	0.74	0.45	0.52
± Range of Variation[h]					0.15	0.17	0.16	0.21	0.21

[a] Systems listed in the same table block have same IAC.

[b] Values or ranges given for identification of appropriate IAC value; where paired, solar transmittances and thicknesses correspond. SHGC is for unshaded glazing at normal incidence.

[c] Typical thickness for residential glass.

[d] From measurements by Van Dyke and Konen (1982) for 45° open venetian blinds, 35° solar incidence, and 35° profile angle.

[e] Use these values only when operation is automated for exclusion of beam solar (as opposed to daylight maximization). Also applies to tightly closed horizontal blinds.

[f] Refers to gray, bronze, and green tinted heat-absorbing glass (on exterior pane in double glazing)

[g] Applies either to factory-fabricated insulating glazing units or to prime windows plus storm windows.

[h] The listed approximate IAC value may be higher or lower by this amount, due to glazing/shading interactions and variations in the shading properties (e.g., manufacturing tolerances).

Table 7-9 Between-Glass Solar Attenuation Coefficients (BAC) for Double Glazing with Between-Glass Shading

Type of Glass	Nominal Thickness, Each Pane	Solar Transmittance[a]		Description of Air Space	Type of Shading		
					Venetian Blinds		Louvered Sun Screen
		Outer Pane	Inner Pane		Light	Medium	
Clear out, Clear in	3/32, 1/8 in.	0.87	0.87	Shade in contact with glass or shade separated from glass by air space.	0.33	0.36	0.43
Clear out, Clear in	1/4 in.	0.80	0.80	Shade in contact with glass-voids filled with plastic.	—	—	0.49
Heat-absorbing[b] out, Clear in				Shade in contact with glass or shade separated from glass by air space.	0.28	0.30	0.37
	1/4 in.	0.46	0.80	Shade in contact with glass-voids filled with plastic.	—	—	0.41

[a] Refer to manufacturers' literature for exact values.

[b] Refers to gray, bronze and green tinted heat-absorbing glass.

Table 7-10 Properties of Representative Indoor Shading Devices Shown in Tables 7-8 and 7-9

Indoor Shade	Solar-Optical Properties (Normal Incidence)		
	Transmittance	Reflectance	Absorptance
Venetian blinds[a] (ratio of slat width to slat spacing 1.2, slat angle 45°)			
Light-colored slat	0.05	0.55	0.40
Medium-colored slat	0.05	0.35	0.60
Vertical blinds			
White louvers	0.00	0.77	0.23
Roller shades			
Light shades (translucent)	0.25	0.60	0.15
White shade (opaque)	0.00	0.65	0.35
Dark-colored shade (opaque)	0.00	0.20	0.80

[a] Values in this table and Tables 19 and 20 are based on horizontal venetian blinds. However, tests show that these values can be used for vertical blinds with good accuracy.

between the outdoor air and conditioned-space design dry-bulb temperatures minus 5°F (2.8°C). In some cases, the air temperature in the adjacent space corresponds to the outdoor air temperature or higher.

For floors in direct contact with the ground, or over an underground basement that is neither ventilated nor conditioned, heat transfer may be neglected for cooling load estimates.

7.3.4 Heat Sources in Conditioned Spaces

People. Representative rates at which heat and moisture are given off by human beings in different states of activity are listed in Table 7-14. Often, these sensible and latent heat gains constitute a large fraction of the total load. For short occupancy, the extra heat and moisture brought in by people may be significant.

The conversion of sensible heat gain from people to space cooling load is affected by the thermal storage characteristics of that space, and it is thus subject to application of appropriate room transfer functions (RTF). Latent heat gains are instantaneous.

Lighting. Since lighting is often the major component of space load, an accurate estimate of the space heat gain it imposes is needed. The rate of heat gain at any moment can be different from the heat equivalent of power supplied instantaneously to those lights.

Only part of the energy from lights is in the form of convective heat, which is immediately picked up by the air-conditioning apparatus. The remaining portion is in the form of radiation that affects the conditioned space once it has been absorbed and re-released by the walls, floors, furniture, etc. This absorbed energy contributes to space cooling load only after a time lag, so part of this energy is reradiating after the lights have been turned off (Figure 7-5).

Time lag effect should always be considered when calculating cooling load, since load felt by the space can be lower than the instantaneous heat gain generated, and peak load for the space may be affected.

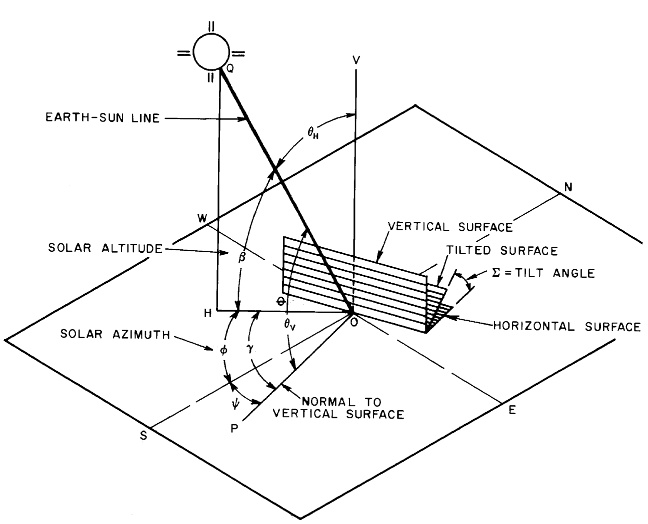


Fig. 7-4 Solar Angles

The primary source of heat from lighting comes from the light-emitting elements (lamps), although additional heat may be generated from associated components in the light fixtures housing such lamps.

Generally, the instantaneous rate of heat gain from electric lighting may be calculated from

$$q_{el} = 3.41 W F_{\text{ul}} F_{\text{sa}} \tag{7-18}$$

where

q_{el} = heat gain, Btu/h
W = total light wattage
F_{ul} = lighting use factor
F_{sa} = lighting special allowance factor

Total light wattage is obtained from the ratings of all lamps installed, both for general illumination and for display use.

The **lighting use factor** is the ratio of the wattage in use for the conditions under which the load estimate is being made to the total installed wattage. For commercial applications, such as stores, the use factor generally is 1.0.

The **special allowance factor** is for fluorescent fixtures and/or fixtures that are either ventilated or installed so that only part of their heat goes to the conditioned space. For fluorescent or high-intensity discharge fixtures, the special allowance factor accounts primarily for ballast losses. Table 7-15 shows that the special allowance factor for a two-lamp fluorescent fixture ranges from 0.94 for T8 lamps with an electronic ballast to 1.21 for energy-saver T12 lamps with a standard electromagnetic ballast. High-intensity discharge fixtures, such as metal halide, may have special allowance factors varying from 1.07 to 1.44, depending on the lamp wattage and quantity of lamps per fixture, and should be dealt with individually. There is a wide variety of lamp and ballast combinations is available, and ballast catalog data provide the overall fixture wattage.

An alternative procedure is to estimate the lighting heat gain on a per square foot basis. Such an approach may be required when final lighting plans are not available. Table 7-16 shows the maximum lighting power density (LPD) (lighting heat gain per square foot) allowed by ASHRAE Standard 90.1-2010 for a range of space types.

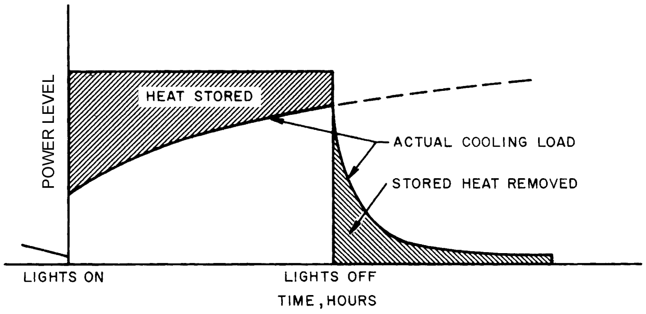


Fig. 7-5 Thermal Storage Effect in Cooling Load from Lights

(Figure 2, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

Table 7-11 Interior Solar Attenuation Coefficients for Single and Insulating Glass with Draperies

Glazing	Glass Trans-mission	Glazing SHGC (No Drapes)	IAC A	B	C	D	E	F	G	H	I	J
Single glass												
1/8 in. clear	0.86	0.87	0.87	0.82	0.74	0.69	0.64	0.59	0.53	0.48	0.42	0.37
1/4 in. clear	0.8	0.83	0.84	0.79	0.74	0.68	0.63	0.58	0.53	0.47	0.42	0.37
1/2 in. clear	0.71	0.77	0.84	0.80	0.75	0.69	0.64	0.59	0.55	0.49	0.44	0.40
1/4 in. heat absorbing	0.46	0.58	0.85	0.81	0.78	0.73	0.69	0.66	0.61	0.57	0.54	0.49
1/2 in. heat absorbing	0.24	0.44	0.86	0.84	0.80	0.78	0.76	0.72	0.68	0.66	0.64	0.60
Reflective coated	—	0.52	0.95	0.90	0.85	0.82	0.77	0.72	0.68	0.63	0.60	0.55
	—	0.44	0.92	0.88	0.84	0.82	0.78	0.76	0.72	0.68	0.66	0.62
	—	0.35	0.90	0.88	0.85	0.83	0.80	0.75	0.73	0.70	0.68	0.65
	—	0.26	0.83	0.80	0.80	0.77	0.77	0.77	0.73	0.70	0.70	0.67
Insulating glass, 1/4-in. air space												
(1/8 in. out and 1/8 in. in)	0.76	0.77	0.84	0.80	0.73	0.71	0.64	0.60	0.54	0.51	0.43	0.40
Insulating glass 1/2-in. air space												
Clear out and clear in	0.64	0.72	0.80	0.75	0.70	0.67	0.63	0.58	0.54	0.51	0.45	0.42
Heat absorbing out and clear in	0.37	0.48	0.89	0.85	0.82	0.78	0.75	0.71	0.67	0.64	0.60	0.58
Reflective coated	—	0.35	0.95	0.93	0.93	0.90	0.85	0.80	0.78	0.73	0.70	0.70
	—	0.26	0.97	0.93	0.90	0.90	0.87	0.87	0.83	0.83	0.80	0.80
	—	0.17	0.95	0.95	0.90	0.90	0.85	0.85	0.80	0.80	0.75	0.75

Interior Solar Attenuation (IAC)

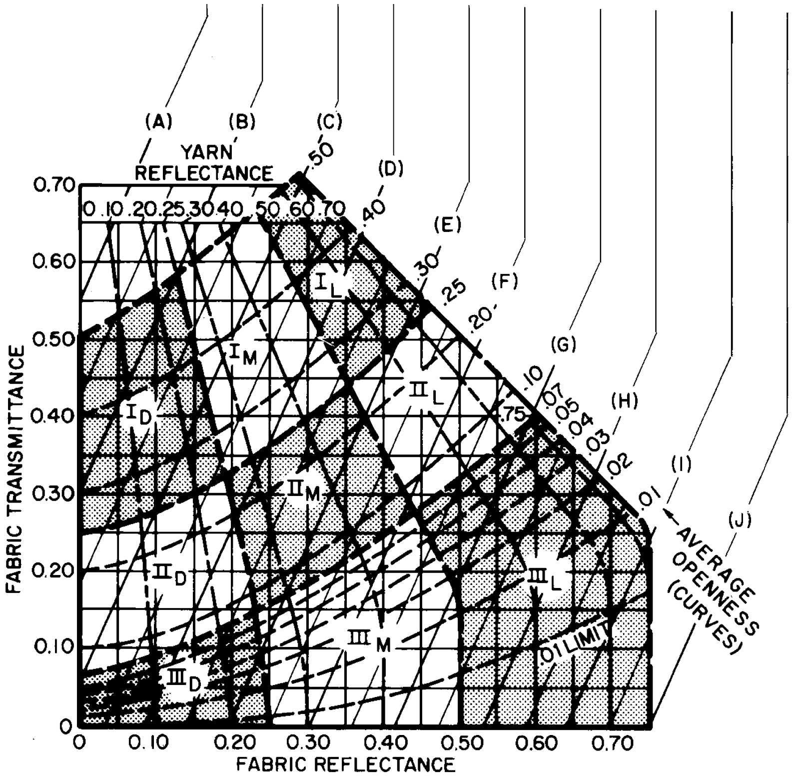
Notes:

1. Interior attenuation coefficients are for draped fabrics.
2. Other properties are for fabrics in flat orientation.
3. Use fabric reflectance and transmittance to obtain accurate IAC values.
4. Use openness and yarn reflectance or openness and fabric reflectance to obtain the various environmental characteristics, or to obtain approximate IAC values.

Classification of Fabrics

I = Open weave
II = Semiopen weave
III = Closed weave

D = Dark color
M = Medium color
L = Light color



To obtain fabric designator (III_L, I_M, etc.). Using either (1) fabric transmittance and fabric reflectance coordinates, or (2) openness and yarn reflectance coordinates, find a point on the chart and note the designator for that area.

To obtain interior attenuation (IAC). (1) Locate drapery fabric as a point using its known properties, or approximate using its fabric classification designator. For accuracy, use fabric transmittance and fabric reflectance; (2) follow diagonal IAC lines to lettered columns in the table. Find IAC value in selected column on line corresponding to glazing used. For example, IAC is 0.4 for 1/4-in. clear single glass with III_L drapery (Column H).

In addition to determining the lighting heat gain, the fraction of lighting heat gain that enters the conditioned space may need to be distinguished from the fraction that enters an unconditioned space; of the former category, the distribution between radiative and convective heat gain must be established.

Fisher and Chantrasrisalai (2006) experimentally studied 12 luminaire types and recommended five different categories of luminaires, as shown in Table 7-17. The table provides a range of design data for the conditioned space fraction, short-wave radiative fraction, and long-wave radiative fraction under typical operating conditions: airflow rate of 1 cfm/ft², supply air temperature between 59°F and 62°F, and room air temperature between 72°F and 75°F. The recommended fractions in Table 7-17 are based on lighting heat input rates range of 0.9 to 2.6 W/ft^2. For higher design power input, the lower bounds of the space and short-wave fractions should be used; for design power input below this range, the upper bounds of the space and short-wave fractions should be used. The **space fraction** in the table is the fraction of lighting heat gain that goes to the room; the fraction going to the plenum can be computed as 1 – the space fraction. The **radiative fraction** is the radiative part of the lighting heat gain that goes to the room. The convective fraction of the lighting heat gain that goes to the room is 1 – the radiative fraction. Using values in the middle of the range yields sufficiently accurate results. However, values that better suit a specific situation may be determined according to the notes for Table 7-17.

Table 7-17's data apply to both ducted and nonducted returns. However, application of the data, particularly the ceiling plenum fraction, may vary for different return configura-

Table 7-12 Surface Orientations and Azimuths, Measured from South

	N	NE	E	SE	S	SW	W	NW
Surface azimuth, ψ	180°	–135°	–90°	–45°	0°	45°	90°	135°

Table 7-13 Solar Position and Profile Angles for 40° N Latitude

	Solar	Solar Time		Profile (Shadow Line) Angles												Angles of Incidence Vertical Surfaces													Solar
DATE	Time	ALT	AZ	N	NNE	NE	ENE	E	ESE	SE	SSE	S	SSW	SW	WSW	N	NNE	NE	ENE	E	ESE	SE	SSE	S	SSW	SW	WSW	HOR	Time
DEC	0800	5	53			35	11	7	6	6	6	9	21					82	60	37	16	10	31	53	76			85	1600
	0900	14	42				37	20	15	14	15	18	30	78					71	50	29	14	24	44	65	87		76	1500
	1000	21	29				72	38	26	21	21	23	31	54					84	63	43	26	22	35	55	75		69	1400
	1100	25	15					61	37	28	25	26	31	43	75					76	56	38	26	29	44	63	83	65	1300
	1200	27	0					90	53	35	28	27	28	35	53					90	70	51	34	27	34	51	70	63	1200
JAN	0800	8	55			38	15	10	8	8	10	14	34					80	58	36	15	13	34	56	78			82	1600
+	0900	17	44				40	24	18	17	18	23	37	87					70	48	29	17	27	46	68	89		73	1500
NOV	1000	24	31				72	41	29	24	24	27	36	61					82	62	43	27	25	38	57	77		66	1400
	1100	21	16					63	41	32	29	29	35	48	78					76	57	40	29	32	47	65	84	62	1300
	1200	30	0					90	56	39	32	30	32	39	56					90	71	52	37	30	37	52	71	60	1200
FEB	0700	4	72		43	9	6	4	4	5	7	14					85	63	41	18	6	27	50	72				86	1700
+	0800	15	62			43	23	17	15	15	19	29	69					74	52	32	16	22	41	63	84			75	1600
OCT	0900	24	50			80	45	31	25	24	27	35	56					86	65	46	30	25	36	54	74			66	1500
	1000	32	35				70	47	36	32	33	38	50	75					79	61	44	33	34	46	63	82		58	1400
	1100	37	19					67	49	40	37	39	45	60	85					75	58	45	37	41	53	69	87	53	1300
	1200	39	0					90	65	49	41	39	41	49	65					90	73	57	44	39	44	57	73	51	1200
MAR	0700	11	80		43	13	13	12	12	14	21	50					78	56	34	15	17	37	58	80				79	1700
+	0800	23	70		85	45	29	24	23	25	31	50					88	67	47	30	23	33	51	71				67	1600
SEP	0900	33	57			72	48	37	33	33	38	50	75					80	61	45	34	35	46	63	81			57	1500
	1000	42	42				69	53	45	42	45	50	64	87					76	60	48	42	45	56	71	88		48	1400
	1100	48	23				90	71	57	50	48	50	57	71					90	75	62	52	48	52	62	75		42	1300
	1200	50	0					90	72	59	52	50	52	59	72					90	76	63	54	50	54	63	76	40	1200
APR	0600	7	99	40	14	9	8	8	9	12	29					81	59	37	15	12	32	54	77					83	1800
+	0700	19	89		42	26	20	19	20	26	41	88					69	48	29	19	29	48	68	89				71	1700
AUG	0800	30	79		71	46	35	31	31	35	47	72					80	61	44	32	32	44	62	81				60	1600
	0900	41	67			67	51	44	41	43	51	66	90					74	58	46	41	46	58	73	90			49	1500
	1000	51	51			85	69	58	52	51	55	63	77					86	72	61	53	51	57	67	80			39	1400
	1100	59	29				86	73	64	60	59	62	69	81					87	75	66	60	69	63	71	82		31	1300
	1200	62	0					90	78	69	63	62	63	69	78					90	80	70	64	62	64	70	80	28	1200
MAY	0500	2	115	5	3	2	2	2	3	6						65	43	20	3	25	47	70						88	1900
+	0600	13	106	40	20	15	13	13	16	25	62					75	53	32	14	20	40	61	83					77	1800
JUL	0700	24	97	75	42	30	25	24	27	36	58					84	64	44	28	25	37	55	76					66	1700
	0800	35	87		65	47	38	35	37	44	59	86					74	57	43	36	40	53	70	88				55	1600
	0900	47	76		82	64	53	48	47	51	61	77					84	69	57	48	47	54	66	80				43	1500
	1000	57	61			80	68	61	58	58	63	73	86					82	70	62	58	59	65	75	86			33	1400
	1100	66	37				84	75	69	66	67	71	77	87					84	76	70	66	67	71	78	87		24	1300
	1200	70	0					90	82	76	71	70	71	76	82					90	82	76	72	70	72	76	82	20	1200
JUN	0500	4	117	9	6	4	4	5	7	14						63	40	18	6	28	50	72						86	1900
	0600	15	108	40	22	16	15	16	19	31	75					72	51	30	15	23	43	64	86					75	1800
	0700	26	100	71	42	31	27	26	30	40	66					81	61	43	29	28	40	59	79					64	1700
	0800	37	91	89	63	47	39	37	40	48	64					89	72	55	42	37	43	56	73					53	1600
	0900	49	80		79	63	54	49	50	54	65	82					82	68	56	90	50	57	69	84				41	1500
	1000	60	66			78	68	62	60	61	67	77	89					80	70	63	60	62	69	78	89			30	1400
	1100	69	42				83	76	71	69	70	74	81	89					83	76	71	69	70	75	81	89		21	1300
	1200	73	0					90	84	78	75	73	75	78	84					90	84	78	75	73	75	78	84	17	1200
				N	NNW	NW	WNW	W	WSW	SW	SSW	S	SSE	SE	ESE	N	NNW	NW	WNW	W	WSW	SW	SSW	S	SSE	SE	ESE	HOR	

Dates vary year to year within plus or minus two days of the twenty-first day of the month.

Table 7-14 Representative Rates at Which Heat and Moisture are Given off by Humans in Different States of Activity

(Table 1, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

		Total Heat, Btu/h		Sensible Heat, Btu/h	Latent Heat, Btu/h	% Sensible Heat that is Radiant[b]	
Degree of Activity	Location	Adult Male	Adjusted, M/F[a]			Low V	High V
Seated at theater	Theater	390	350	245	105	60	27
Seated, very light work	Offices, hotels, apartments	450	400	245	155		
Moderately active office work	Offices, hotels, apartments	475	450	250	200		
Standing, light work; walking	Department store; retail store	550	450	250	200	58	38
Walking, standing	Drug store, bank	550	500	250	250		
Sedentary work	Restaurant[c]	490	550	275	275		
Light bench work	Factory	800	750	275	475		
Moderate dancing	Dance hall	900	850	305	545	49	35
Walking 3 mph; light machine work	Factory	1000	1000	375	625		
Bowling[d]	Bowling alley	1500	1450	580	870		
Heavy work	Factory	1500	1450	580	870	54	19
Heavy machine work; lifting	Factory	1600	1600	635	965		
Athletics	Gymnasium	2000	1800	710	1090		

Notes:

1. Tabulated values are based on 75°F room dry-bulb temperature. For 80°F room dry bulb, total heat remains the same, but sensible heat values should be decreased by approximately 20%, and latent heat values increased accordingly.
2. Also see Table 4, Chapter 9, for additional rates of metabolic heat generation.
3. All values are rounded to nearest 5 Btu/h.

[a] Adjusted heat gain is based on normal percentage of men, women, and children for the application listed, and assumes that gain from an adult female is 85% of that for an adult male, and gain from a child is 75% of that for an adult male.

[b] Values approximated from data in Table 6, Chapter 9, where V is air velocity with limits shown in that table.

[c] Adjusted heat gain includes 60 Btu/h for food per individual (30 Btu/h sensible and 30 Btu/h latent).

[d] Figure one person per alley actually bowling, and all others as sitting (400 Btu/h) or standing or walking slowly (550 Btu/h).

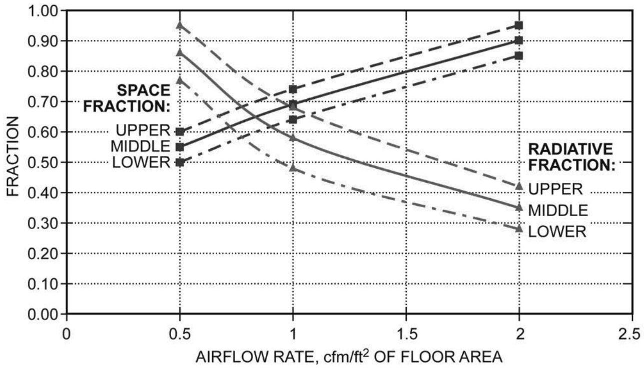


Fig. 7-6 Lighting Heat Gain Parameters for Recessed Fluorescent Luminaire Without Lens

(Figure 3, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

tions. For instance, for a room with a ducted return, although a portion of the lighting energy initially dissipated to the ceiling plenum is quantitatively equal to the plenum fraction, a large portion of this energy would likely end up as the conditioned space cooling load and a small portion would end up as the cooling load to the return air.

If the space airflow rate is different from the typical condition (i.e., about 1 cfm/ft^2), Figure 7-6 can be used to estimate the lighting heat gain parameters. Design data shown in Figure 7-6 are only applicable for the recessed fluorescent luminaire without lens.

Although design data presented in Table 7-17 and Figure 7-6 can be used for a vented luminaire with side-slot returns, they are likely not applicable for a vented luminaire with lamp compartment returns, because in the latter case, all heat convected in the vented luminaire is likely to go directly to the ceiling plenum, resulting in zero convective fraction and a much lower space fraction. Therefore, the design data should only be used for a configuration where conditioned air is returned through the ceiling grille or luminaire side slots.

For other luminaire types, it may be necessary to estimate the heat gain for each component as a fraction of the total lighting heat gain by using judgment to estimate heat-to-space and heat-to-return percentages.

Because of the directional nature of downlight luminaires, a large portion of the short-wave radiation typically falls on the floor. When converting heat gains to cooling loads in the RTS method, the solar **radiant time factors (RTFs)** may be more appropriate than nonsolar RTFs. (Solar RTFs are calculated assuming most solar radiation is intercepted by the floor; nonsolar RTFs assume uniform distribution by area over all interior surfaces.) This effect may be significant for rooms where lighting heat gain is high and for which solar RTFs are significantly different from nonsolar RTFs.

For ventilated or recessed fixtures, manufacturers' or other data must be sought to establish the fraction of the total wattage that may be expected to enter the conditioned space directly (and subject to time lag effect) versus that which must be picked up by return air or in some other appropriate manner.

Light Heat Components. Cooling load caused by lights recessed into ceiling cavities is made up of two components: one part (known as the **heat-to-space** load) comes from the light heat directly contributing to the space heat gain, and the other is the light heat released into the above-ceiling cavity, which (if used as a return air plenum) is mostly picked up by the return air that passes over or through the light fixtures. In such a ceiling return air plenum, this second part of the load (sometimes referred to as **heat-to-return**) never enters the conditioned space. It does, however, add to the overall load and significantly influences the load calculation.

Even though the total cooling load imposed on the cooling coil from these two components remains the same, the larger the fraction of heat output picked up by the return air, the more

the space cooling load is reduced. The minimum required airflow rate for the conditioned space decreases as the space cooling load decreases. Supply fan power decreases accordingly, which results in reduced energy consumption for the system and possibly reduced equipment size as well.

For ordinary design load estimation, the heat gain for each component may be calculated simply as a fraction of the total lighting load by using judgment to estimate heat-to-space and heat-to-return percentages (Mitalas and Kimura 1971).

Return Air Light Fixtures. Two generic types of return air light fixture are available—those that allow and those that do not allow return air to flow through the lamp chamber. The first type is sometimes called a heat-of-light fixture. The percentage of light heat released through the plenum side of various ventilated fixtures can be obtained from lighting fixture manufacturers. For representative data, see Nevins et al. (1971). Even unventilated fixtures lose some heat to plenum spaces; however, most of the heat ultimately enters the conditioned space from a dead-air plenum or is picked up by return air via ceiling return air openings. The percentage of heat to return air ranges from 40% to 60% for heat-to-return ventilated fixtures or 15% to 25% for unventilated fixtures.

Plenum Temperatures. As heat from lighting is picked up by the return air, the temperature differential between the ceiling space and the conditioned space causes part of that heat to flow from the ceiling back to the conditioned space. Return air from the conditioned space can be ducted to capture light heat without passing through a ceiling plenum as such, or the ceiling space can be used as a return air plenum, causing the distribution of light heat to be handled in distinctly different ways. Most plenum temperatures do not rise more than 1 to 3°F above space temperature, thus generating only a relatively small thermal gradient for heat transfer through plenum surfaces but a relatively large percentage reduction in space cooling load.

Energy Balance. Where the ceiling space is used as a return air plenum, an energy balance requires that the heat picked up from the lights into the return air do one or more of the following:

1. Become a part of the cooling load to the return air (represented by a temperature rise of the return air as it passes through the ceiling space)
2. Be partially transferred back into the conditioned space through the ceiling material below
3. Be partially lost (from the space) through the floor surfaces above the plenum

In a multistory building, the conditioned space frequently gains heat through its floor from a similar plenum below, offsetting the loss just mentioned. The radiant component of heat leaving the ceiling or floor surface of a plenum is normally so small that all such heat transfer is considered convective in calculations.

Figure 7-7 shows a schematic diagram of a typical return air plenum. Equations (7-19) through (7-23), using the sign convention as shown in Figure 7-6, represent the heat balance of a return air plenum design for a typical interior room in a multifloor building:

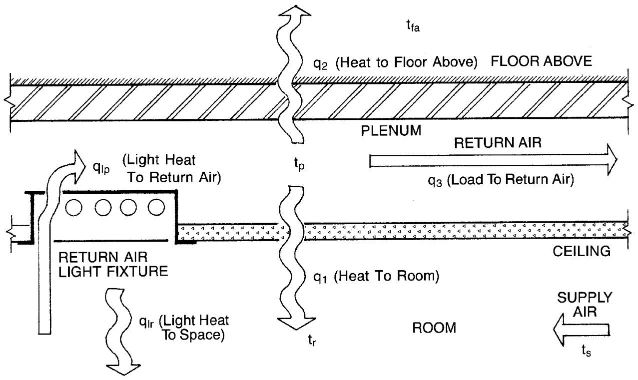


Fig. 7-7 Heat Balance of Typical Ceiling Return Plenum
(Figure 15, Chapter 18, 2013 ASHRAE Handbook—Fundamentals)

$$q_1 = U_c A_c (t_p - t_r) \tag{7-19}$$

$$q_2 = U_f A_f (t_p - t_{\mathrm{fa}}) \tag{7-20}$$

$$q_3 = 1.1 Q (t_p - t_r) \tag{7-21}$$

$$q_{lp} - q_2 - q_1 - q_3 = 0 \tag{7-22}$$

$$Q = \frac{q_r + q_1}{1.1(t_r - t_s)} \tag{7-23}$$

where

q_1 = heat gain to space from plenum through ceiling, Btu/h
q_2 = heat loss from plenum through floor above, Btu/h
q_3 = heat gain “pickup” by return air, Btu/h
Q = return airflow, cfm
q_{lp} = light heat gain to plenum via return air, Btu/h
q_{lr} = light heat gain to space, Btu/h
q_f = heat gain from plenum below, through floor, Btu/h
q_w = heat gain from exterior wall, Btu/h
q_r = space cooling load, Btu/h, including appropriate treatment of q_{lr}, q_f, and/or q_w
t_p = plenum temperature, °F
t_r = space temperature, °F
t_{fa} = space temperature of floor above, °F
t_s = supply air temperature, °F

By substituting Equations (7-19), (7-20), (7-21), and (7-23) into the heat balance equation (7-23), t_p can be found as the resultant return air temperature or plenum temperature. The results, although rigorous and best solved by computer, are important in determining the cooling load, which affects equipment size selection, future energy consumption, and other factors.

Equations (7-19) through (7-23) are simplified to illustrate the heat balance relationship. Heat gain into a return air ple-

Table 7-15 Typical Nonincandescent Light Fixtures

Description	Ballast	Watts/Lamp	Lamps/Fixture	Lamp Watts	Fixture Watts	Special Allowance Factor
Compact Fluorescent Fixtures						
Twin, (1) 5 W lamp	Mag-Std	5	1	5	9	1.80
Twin, (1) 7 W lamp	Mag-Std	7	1	7	10	1.43
Twin, (1) 9 W lamp	Mag-Std	9	1	9	11	1.22
Quad, (1) 13 W lamp	Mag-Std	13	1	13	17	1.31
Quad, (2) 18 W lamp	Mag-Std	18	2	36	45	1.25
Quad, (2) 22 W lamp	Mag-Std	22	2	44	48	1.09
Quad, (2) 26 W lamp	Mag-Std	26	2	52	66	1.27
Twin, (2) 40 W lamp	Mag-Std	40	2	80	85	1.06
Quad, (1) 13 W lamp	Electronic	13	1	13	15	1.15
Quad, (1) 26 W lamp	Electronic	26	1	26	27	1.04
Quad, (2) 18 W lamp	Electronic	18	2	36	38	1.06
Quad, (2) 26 W lamp	Electronic	26	2	52	50	0.96
Twin or multi, (2) 32 W lamp	Electronic	32	2	64	62	0.97
Fluorescent Fixtures						
(1) 18 in., T8 lamp	Mag-Std	15	1	15	19	1.27
(1) 18 in., T12 lamp	Mag-Std	15	1	15	19	1.27
(2) 18 in., T8 lamp	Mag-Std	15	2	30	36	1.20
(2) 18 in., T12 lamp	Mag-Std	15	2	30	36	1.20
(1) 24 in., T8 lamp	Mag-Std	17	1	17	24	1.41
(1) 24 in., T12 lamp	Mag-Std	20	1	20	28	1.40
(2) 24 in., T12 lamp	Mag-Std	20	2	40	56	1.40
(1) 24 in., T12 HO lamp	Mag-Std	35	1	35	62	1.77
(2) 24 in., T12 HO lamp	Mag-Std	35	2	70	90	1.29
(1) 24 in., T8 lamp	Electronic	17	1	17	16	0.94
(2) 24 in., T8 lamp	Electronic	17	2	34	31	0.91
(1) 36 in., T12 lamp	Mag-Std	30	1	30	46	1.53
(2) 36 in., T12 lamp	Mag-Std	30	2	60	81	1.35
(1) 36 in., T12 ES lamp	Mag-Std	25	1	25	42	1.68
(2) 36 in., T12 ES lamp	Mag-Std	25	2	50	73	1.46
(1) 36 in., T12 HO lamp	Mag-Std	50	1	50	70	1.40
(2) 36 in., T12 HO lamp	Mag-Std	50	2	100	114	1.14
(2) 36 in., T12 lamp	Mag-ES	30	2	60	74	1.23
(2) 36 in., T12 ES lamp	Mag-ES	25	2	50	66	1.32
(1) 36 in., T12 lamp	Electronic	30	1	30	31	1.03
(1) 36 in., T12 ES lamp	Electronic	25	1	25	26	1.04
(1) 36 in., T8 lamp	Electronic	25	1	25	24	0.96
(2) 36 in., T12 lamp	Electronic	30	2	60	58	0.97
(2) 36 in., T12 ES lamp	Electronic	25	2	50	50	1.00
(2) 36 in., T8 lamp	Electronic	25	2	50	46	0.92
(2) 36 in., T8 HO lamp	Electronic	25	2	50	50	1.00
(2) 36 in., T8 VHO lamp	Electronic	25	2	50	70	1.40
(1) 48 in., T12 lamp	Mag-Std	40	1	40	55	1.38
(2) 48 in., T12 lamp	Mag-Std	40	2	80	92	1.15
(3) 48 in., T12 lamp	Mag-Std	40	3	120	140	1.17
(4) 48 in., T12 lamp	Mag-Std	40	4	160	184	1.15
(1) 48 in., T12 ES lamp	Mag-Std	34	1	34	48	1.41
(2) 48 in., T12 ES lamp	Mag-Std	34	2	68	82	1.21
(3) 48 in., T12 ES lamp	Mag-Std	34	3	102	100	0.98
(4) 48 in., T12 ES lamp	Mag-Std	34	4	136	164	1.21
(1) 48 in., T12 ES lamp	Mag-ES	34	1	34	43	1.26
(2) 48 in., T12 ES lamp	Mag-ES	34	2	68	72	1.06
(3) 48 in., T12 ES lamp	Mag-ES	34	3	102	115	1.13
(4) 48 in., T12 ES lamp	Mag-ES	34	4	136	144	1.06
(1) 48 in., T8 lamp	Mag-ES	32	1	32	35	1.09
(2) 48 in., T8 lamp	Mag-ES	32	2	64	71	1.11
(3) 48 in., T8 lamp	Mag-ES	32	3	96	110	1.15
(4) 48 in., T8 lamp	Mag-ES	32	4	128	142	1.11
(1) 48 in., T12 ES lamp	Electronic	34	1	34	32	0.94
(2) 48 in., T12 ES lamp	Electronic	34	2	68	60	0.88
(3) 48 in., T12 ES lamp	Electronic	34	3	102	92	0.90
(4) 48 in., T12 ES lamp	Electronic	34	4	136	120	0.88
(1) 48 in., T8 lamp	Electronic	32	1	32	32	1.00
(2) 48 in., T8 lamp	Electronic	32	2	64	60	0.94
(3) 48 in., T8 lamp	Electronic	32	3	96	93	0.97
(4) 48 in., T8 lamp	Electronic	32	4	128	120	0.94
(1) 60 in., T12 lamp	Mag-Std	50	1	50	63	1.26
(2) 60 in., T12 lamp	Mag-Std	50	2	100	128	1.28
(1) 60 in., T12 HO lamp	Mag-Std	75	1	75	92	1.23
(2) 60 in., T12 HO lamp	Mag-Std	75	2	150	168	1.12
(1) 60 in., T12 ES VHO lamp	Mag-Std	135	1	135	165	1.22
(2) 60 in., T12 ES VHO lamp	Mag-Std	135	2	270	310	1.15
(1) 60 in., T12 HO lamp	Mag-ES	75	1	75	88	1.17
(2) 60 in., T12 HO lamp	Mag-ES	75	2	150	176	1.17
(1) 60 in., T12 lamp	Electronic	50	1	50	44	0.88
(2) 60 in., T12 lamp	Electronic	50	2	100	88	0.88
(1) 60 in., T12 HO lamp	Electronic	75	1	75	69	0.92
(2) 60 in., T12 HO lamp	Electronic	75	2	150	138	0.92
(1) 60 in., T8 lamp	Electronic	40	1	40	36	0.90
(2) 60 in., T8 lamp	Electronic	40	2	80	72	0.90
(3) 60 in., T8 lamp	Electronic	40	3	120	106	0.88
(4) 60 in., T8 lamp	Electronic	40	4	160	134	0.84
(1) 72 in., T12 lamp	Mag-Std	55	1	55	76	1.38
(2) 72 in., T12 lamp	Mag-Std	55	2	110	122	1.11
(3) 72 in., T12 lamp	Mag-Std	55	3	165	202	1.22
(4) 72 in., T12 lamp	Mag-Std	55	4	220	244	1.11
(1) 72 in., T12 HO lamp	Mag-Std	85	1	85	120	1.41
(2) 72 in., T12 HO lamp	Mag-Std	85	2	170	220	1.29
(1) 72 in., T12 VHO lamp	Mag-Std	160	1	160	180	1.13
(2) 72 in., T12 VHO lamp	Mag-Std	160	2	320	330	1.03
(2) 72 in., T12 lamp	Mag-ES	55	2	110	122	1.11
(4) 72 in., T12 lamp	Mag-ES	55	4	220	244	1.11
(2) 72 in., T12 HO lamp	Mag-ES	85	2	170	194	1.14
(4) 72 in., T12 HO lamp	Mag-ES	85	4	340	388	1.14
(1) 72 in., T12 lamp	Electronic	55	1	55	68	1.24
(2) 72 in., T12 lamp	Electronic	55	2	110	108	0.98
(3) 72 in., T12 lamp	Electronic	55	3	165	176	1.07
(4) 72 in., T12 lamp	Electronic	55	4	220	216	0.98
(1) 96 in., T12 ES lamp	Mag-Std	60	1	60	75	1.25
(2) 96 in., T12 ES lamp	Mag-Std	60	2	120	128	1.07
(3) 96 in., T12 ES lamp	Mag-Std	60	3	180	203	1.13
(4) 96 in., T12 ES lamp	Mag-Std	60	4	240	256	1.07
(1) 96 in., T12 ES HO lamp	Mag-Std	95	1	95	112	1.18
(2) 96 in., T12 ES HO lamp	Mag-Std	95	2	190	227	1.19
(3) 96 in., T12 ES HO lamp	Mag-Std	95	3	285	380	1.33
(4) 96 in., T12 ES HO lamp	Mag-Std	95	4	380	454	1.19
(1) 96 in., T12 ES VHO lamp	Mag-Std	185	1	185	205	1.11
(2) 96 in., T12 ES VHO lamp	Mag-Std	185	2	370	380	1.03
(3) 96 in., T12 ES VHO lamp	Mag-Std	185	3	555	585	1.05
(4) 96 in., T12 ES VHO lamp	Mag-Std	185	4	740	760	1.03
(2) 96 in., T12 ES lamp	Mag-ES	60	2	120	123	1.03
(3) 96 in., T12 ES lamp	Mag-ES	60	3	180	210	1.17
(4) 96 in., T12 ES lamp	Mag-ES	60	4	240	246	1.03
(2) 96 in., T12 ES HO lamp	Mag-ES	95	2	190	207	1.09
(4) 96 in., T12 ES HO lamp	Mag-ES	95	4	380	414	1.09

Table 7-15 Typical Nonincandescent Light Fixtures *(Continued)*

Description	Ballast	Watts/Lamp	Lamps/Fixture	Lamp Watts	Fixture Watts	Special Allowance Factor	Description	Ballast	Watts/Lamp	Lamps/Fixture	Lamp Watts	Fixture Watts	Special Allowance Factor
(1) 96 in., T12 ES lamp	Electronic	60	1	60	69	1.15	(1) 96 in., T8 HO lamp	Electronic	59	1	59	68	1.15
(2) 96 in., T12 ES lamp	Electronic	60	2	120	110	0.92	(1) 96 in., T8 VHO lamp	Electronic	59	1	59	71	1.20
(3) 96 in., T12 ES lamp	Electronic	60	3	180	179	0.99	(2) 96 in., T8 lamp	Electronic	59	2	118	109	0.92
(4) 96 in., T12 ES lamp	Electronic	60	4	240	220	0.92	(3) 96 in., T8 lamp	Electronic	59	3	177	167	0.94
(1) 96 in., T12 ES HO lamp	Electronic	95	1	95	80	0.84	(4) 96 in., T8 lamp	Electronic	59	4	236	219	0.93
(2) 96 in., T12 ES HO lamp	Electronic	95	2	190	173	0.91	(2) 96 in., T8 HO lamp	Electronic	86	2	172	160	0.93
(4) 96 in., T12 ES HO lamp	Electronic	95	4	380	346	0.91	(4) 96 in., T8 HO lamp	Electronic	86	4	344	320	0.93
(1) 96 in., T8 lamp	Electronic	59	1	59	58	0.98							
Circular Fluorescent Fixtures													
Circlite, (1) 20 W lamp	Mag-PH	20	1	20	20	1.00	(2) 8 in. circular lamp	Mag-RS	22	2	44	52	1.18
Circlite, (1) 22 W lamp	Mag-PH	22	1	22	20	0.91	(1) 12 in. circular lamp	Mag-RS	32	1	32	31	0.97
Circline, (1) 32 W lamp	Mag-PH	32	1	32	40	1.25	(2) 12 in. circular lamp	Mag-RS	32	2	64	62	0.97
(1) 6 in. circular lamp	Mag-RS	20	1	20	25	1.25	(1) 16 in. circular lamp	Mag-Std	40	1	40	35	0.88
(1) 8 in. circular lamp	Mag-RS	22	1	22	26	1.18							
High-Pressure Sodium Fixtures													
(1) 35 W lamp	HID	35	1	35	46	1.31	(1) 250 W lamp	HID	250	1	250	295	1.18
(1) 50 W lamp	HID	50	1	50	66	1.32	(1) 310 W lamp	HID	310	1	310	365	1.18
(1) 70 W lamp	HID	70	1	70	95	1.36	(1) 360 W lamp	HID	360	1	360	414	1.15
(1) 100 W lamp	HID	100	1	100	138	1.38	(1) 400 W lamp	HID	400	1	400	465	1.16
(1) 150 W lamp	HID	150	1	150	188	1.25	(1) 1000 W lamp	HID	1000	1	1000	1100	1.10
(1) 200 W lamp	HID	200	1	200	250	1.25							
Metal Halide Fixtures													
(1) 32 W lamp	HID	32	1	32	43	1.34	(1) 250 W lamp	HID	250	1	250	295	1.18
(1) 50 W lamp	HID	50	1	50	72	1.44	(1) 400 W lamp	HID	400	1	400	458	1.15
(1) 70 W lamp	HID	70	1	70	95	1.36	(2) 400 W lamp	HID	400	2	800	916	1.15
(1) 100 W lamp	HID	100	1	100	128	1.28	(1) 750 W lamp	HID	750	1	750	850	1.13
(1) 150 W lamp	HID	150	1	150	190	1.27	(1) 1000 W lamp	HID	1000	1	1000	1080	1.08
(1) 175 W lamp	HID	175	1	175	215	1.23	(1) 1500 W lamp	HID	1500	1	1500	1610	1.07
Mercury Vapor Fixtures													
(1) 40 W lamp	HID	40	1	40	50	1.25	(1) 250 W lamp	HID	250	1	250	290	1.16
(1) 50 W lamp	HID	50	1	50	74	1.48	(1) 400 W lamp	HID	400	1	400	455	1.14
(1) 75 W lamp	HID	75	1	75	93	1.24	(2) 400 W lamp	HID	400	2	800	910	1.14
(1) 100 W lamp	HID	100	1	100	125	1.25	(1) 700 W lamp	HID	700	1	700	780	1.11
(1) 175 W lamp	HID	175	1	175	205	1.17	(1) 1000 W lamp	HID	1000	1	1000	1075	1.08

Abbreviations: Mag = electromagnetic; ES = energy saver; Std = standard; HID = high-intensity discharge; HO = high output; VHO = very high output; PH = preheat; RS = rapid start

num is not limited to the heat of lights alone. Exterior walls directly exposed to the ceiling space will transfer heat directly to or from the return air. For single-story buildings or the top floor of a multistory building, the roof heat gain or loss enters or leaves the ceiling plenum rather than entering or leaving the conditioned space directly. The supply air quantity calculated by Equation 7-23 is for the conditioned space under consideration only, and it is assumed equal to the return air quantity.

The amount of airflow through a return plenum above a conditioned space may not be limited to that supplied into the space under consideration; it will, however, have no noticeable effect on plenum temperature if the surplus comes from an adjacent plenum operating under similar conditions. Where special conditions exist, heat balance Equations (7-19) through (7-23) must be modified appropriately. Finally, even though the building's thermal storage has some effect, the amount of heat entering the return air is small and may be considered as convective for calculation purposes.

Power. Instantaneous heat gain from equipment operated by electric motors within a conditioned space is calculated as follows:

$$q_{em} = 2545(P/E_M)F_{\mathrm{LM}}F_{\mathrm{MU}} \tag{7-24}$$

where

q_{em} = heat equivalent of equipment operation, Btu/h
P = motor horsepower rating
E_M = motor efficiency, as decimal fraction < 1.0
F_{LM} = motor load factor, 1.0 or decimal fraction < 1.0
F_{UM} = motor use factor, 1.0 or decimal fraction < 1.0

The motor use factor may be applied when motor use is known to be intermittent with significant nonuse during all hours of operation (i.e., overhead door operator, and so forth). For conventional applications, its value is 1.0.

The motor load factor is the fraction of the rated load delivered under the conditions of the cooling load estimate. In

Table 7-16 Lighting Heat Gain Parameters for Typical Operating Conditions

(Table 3, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

Luminaire Category	Space Fraction	Radiative Fraction	Notes
Recessed fluorescent luminaire without lens	0.64 to 0.74	0.48 to 0.68	• Use middle values in most situations • May use higher space fraction, and lower radiative fraction for luminaire with side-slot returns • May use lower values of both fractions for direct/indirect luminaire • May use higher values of both fractions for ducted returns
Recessed fluorescent luminaire with lens	0.40 to 0.50	0.61 to 0.73	• May adjust values in the same way as for recessed fluorescent luminaire without lens
Downlight compact fluorescent luminaire	0.12 to 0.24	0.95 to 1.0	• Use middle or high values if detailed features are unknown • Use low value for space fraction and high value for radiative fraction if there are large holes in luminaire's reflector
Downlight incandescent luminaire	0.70 to 0.80	0.95 to 1.0	• Use middle values if lamp type is unknown • Use low value for space fraction if standard lamp (i.e. A-lamp) is used • Use high value for space fraction if reflector lamp (i.e. BR-lamp) is used
Non-in-ceiling fluorescent luminaire	1.0	0.5 to 0.57	• Use lower value for radiative fraction for surface-mounted luminaire • Use higher value for radiative fraction for pendant luminaire
Recessed LED troffer partial aperture diffuser	0.49 to 0.64	0.37 to 0.47	• Use middle value in most cases. • May use higher space fraction for ducted return configuration and lower space fraction for high supply air temperature. • May use higher radiant value for ducted return configuration and lower value for large supply airflow rate.
Recessed LED troffer uniform diffuser	0.44 to 0.66	0.32 to 0.41	• Use middle value in most cases. • May use higher space fraction for smaller supply airflow rate and lower value for larger supply airflow rate. • May use higher radiant value for ducted return configuration and lower value for larger supply airflow rate.
Recessed high-efficacy LED troffer	0.59	0.51	
Recessed LED downlight	0.40 to 0.56	0.15 to 0.18	• Use middle value in most cases. • May use higher space fraction value for high supply air temperature and lower value for smaller air flowrate. • May use higher radiant value for dimming control and lower value for large supply air flowrate.
Recessed LED retrofit kit 2×4	0.41 to 0.53	0.31 to 0.42	• Use middle value in most cases. • May use higher space fraction value for large supply air flowrate and lower value for ducted return configuration. • May use higher radiant value for ducted return configuration and lower value for larger supply airflow rate.
Recessed LED color tuning fixture	0.53 to 0.56	0.40 to 0.42	Use middle value in most cases.
High-bay LED fixture	1.0	0.42 to 0.51	Use middle value in most cases.
Linear pendant LED fixture	1.0	0.55 to 0.60	Use middle value in most cases.

Sources: Fisher and Chantrasrisalai (2006); Zhou et al. 2016.

Equation 7-24, both the motor and the driven equipment are assumed to be within the conditioned space. If the motor is outside the space or airstream, Equation 7-25 is used:

$$q_{em} = 2545PF_{\text{LM}}F_{\text{UM}} \tag{7-25}$$

When the motor is inside the conditioned space or airstream but the driven machine is outside, Equation 7-26 is used:

$$q_{em} = 2545P[(1-E_M)/E_M]F_{\text{LM}}F_{\text{UM}} \tag{7-26}$$

Equation 7-26 also applies to a fan or pump in the conditioned space that exhausts air or pumps fluid outside that space.

Average efficiencies, and related data representative of typical electric motors, generally derived from the lower efficiencies reported by several manufacturers of open, drip-proof motors, are given in Table 7-18. Unless the manufacturers' technical literature indicates otherwise, the heat gain may be divided equally between radiant and convective components for subsequent cooling load calculations.

Table 7-19 gives minimum efficiencies and related data representative of typical electric motors from ASHRAE Standard 90.1-2013. The actual value should be obtained from the manufacturer. If the motor is underloaded or overloaded, efficiency could vary from the manufacturer's listing.

Appliances. In a cooling load estimate, heat gain from all appliances—electrical, gas, or steam—should be considered. Food preparation equipment is among the most common types of heat-producing appliances found in conditioned areas. Appliance surfaces contribute most of the heat to commercial kitchens. When installed under an effective hood, cooling load is independent of the fuel or energy used for similar equipment performing the same operations. Because the heat is primarily radiant energy from the appliance surfaces and cooking utensils, convected and latent heat are negligible.

To establish a heat gain value, actual input values and various factors, efficiencies, or other judgmental modifiers are

Table 7-17 Lighting Power Densities Using Space-by-Space Method

(Table 2, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

Common Space Types[a]	LPD, W/ft^2
Atrium	
≤40 ft high	0.03/ft total height
>40 ft high	0.40 + 0.02/ft total height
Audience Seating Area	
In auditorium	0.63
In convention center	0.82
In gymnasium	0.65
In motion picture theater	1.14
In penitentiary	0.28
In performing arts theater	2.43
In religious building	1.53
In sports arena	0.43
All other audience seating areas	0.43
Banking Activity Area	1.01
Breakroom (*See* **Lounge/Breakroom)**	
Classroom/Lecture Hall/Training Room	
In penitentiary	1.34
All other classrooms/lecture halls/training rooms	1.24
Conference/Meeting/Multipurpose Room	1.23
Confinement Cells	0.81
Copy/Print Room	0.72
Corridor[b]	
In facility for visually impaired (and not used primarily by staff)[c]	0.92
In hospital	0.99
In manufacturing facility	0.41
All other corridors	0.66
Courtroom	1.72
Computer Room	1.71
Dining Area	
In penitentiary	0.96
In facility for visually impaired (and not used primarily by staff)[c]	2.65
In bar/lounge or leisure dining	1.07
In cafeteria or fast food dining	0.65
In family dining	0.89
All other dining areas	0.65
Electrical/Mechanical Room[f]	0.42
Emergency Vehicle Garage	0.56
Food Preparation Area	1.21
Guest Room	0.91
Laboratory	
In or as classroom	1.43
All other laboratories	1.81
Laundry/Washing Area	0.60
Loading Dock, Interior	0.47
Lobby	
In facility for the visually impaired (and not used primarily by staff)[c]	1.80
For elevator	0.64
In hotel	1.06
In motion picture theater	0.59
In performing arts theater	2.00
All other lobbies	0.90
Locker Room	0.75
Lounge/Breakroom	
In health care facility	0.92
All other lounges/breakrooms	0.73
Enclosed and ≤250 ft^2	1.11
Enclosed and >250 ft^2	1.11
Open plan	0.98
Office	
Enclosed	1.11
Open plan	0.98
Parking Area, Interior	0.19
Pharmacy Area	1.68
Restroom	
In facility for the visually impaired (and not used primarily by staff)[c]	1.21
All other restrooms	0.98
Sales Area[d]	1.44
Seating Area, General	0.54
Stairway	
Space containing stairway determines LPD and control requirements for stairway.	
Stairwell	0.69
Storage Room	
<50 ft^2	1.24
All other storage rooms	0.63
Vehicular Maintenance Area	0.67

Building-Specific Space Types*	LPD, W/ft^2
Facility for Visually Impaired[c]	
Chapel (used primarily by residents)	2.21
Recreation room/common living room (and not used primarily by staff)	2.41
Automotive (*See* **Vehicular Maintenance Area**)	
Convention Center, Exhibit Space	1.45
Dormitory/Living Quarters	0.38
Fire Station, Sleeping Quarters	0.22
Gymnasium/Fitness Center	
Exercise area	0.72
Playing area	1.20
Health Care Facility	
Exam/treatment room	1.66
Imaging room	1.51
Medical supply room	0.74
Nursery	0.88
Nurses' station	0.71
Operating room	2.48
Patient room	0.62
Physical therapy room	0.91
Recovery room	1.15
Library	
Reading area	1.06
Stacks	1.71
Manufacturing Facility	
Detailed manufacturing area	1.29
Equipment room	0.74
Extra-high-bay area (>50 ft floor-to-ceiling height)	1.05
High-bay area (25 to 50 ft floor-to-ceiling height)	1.23
Low bay area (<25 ft floor-to-ceiling height)	1.19
Museum	
General exhibition area	1.05
Restoration room	1.02
Performing Arts Theater, Dressing Room	0.61
Post Office, Sorting Area	0.94
Religious Buildings	
Fellowship hall	0.64
Worship/pulpit/choir area	1.53
Retail Facilities	
Dressing/fitting room	0.71
Mall concourse	1.10
Sports Arena, Playing Area	
For Class I facility	3.68
For Class II facility	2.40
For Class III facility	1.80
For Class IV facility	1.20
Transportation Facility	
In baggage/carousel area	0.53
In airport concourse	0.36
At terminal ticket counter	0.80
Warehouse—Storage Area	
For medium to bulky, palletized items	0.58
For smaller, hand-carried items[e]	0.95

Source: ASHRAE *Standard* 90.1-2013.

[a] In cases where both a common space type and a building-specific type are listed, the building-specific space type applies.

[b] In corridors, extra lighting power density allowance is granted when corridor width is <8 ft and is not based on room/corridor ratio (RCR).

[c] A facility for the visually impaired one that can be documented as being designed to comply with light levels in ANSI/IES RP-28 and is (or will be) licensed by local/state authorities for either senior long-term care, adult daycare, senior support, and/or people with special visual needs.

[d] For accent lighting, see section 9.6.2(b) of ASHRAE *Standard* 90.1-2013.

[e] Sometimes called a picking area.

[f] An additional 0.53 W/ft^2 is allowed *only* if this additional lighting is controlled separately from the base allowance of 0.42 W/ft^2.

preferred. Where specific rating data are unavailable (nameplate missing, equipment not yet purchased, and so forth), recommended heat gains tabulated in this chapter may be used. In estimating the appliance load, probabilities of simultaneous use and operation for different appliances located in the same space must be considered.

Radiation contributes up to 32% of the hourly heat input to hooded appliances (for a conservative radiation factor F_{RA} = 0.32). Radiant heat temperature rises can be substantially reduced by shielding the fronts of cooking appliances. These reductions amount to 61% with glass panels and 78% with polished aluminum shielding. A floor-slot air curtain in front of appliances reduces the radiant temperature rise by 15%.

For each meal served, the heat transferred to the dining space is approximately 50 Btu/h, 75% of which is sensible and 25% is latent.

The maximum hourly input can be estimated as 50% of the total nameplate or catalog input q_i ratings because of the diversity of appliance use and the effect of thermostatic controls, giving a usage factor F_{UA} = 0.50. Therefore, the maximum hourly heat gain q_m for generic types of **electric and steam appliance**s installed under a hood can be estimated from the following equations:

$$q_a = q_i F_{UA} F_{RA} \tag{7-27}$$

or

$$q_a = 0.16 q_i \tag{7-28}$$

Direct fuel-fired cooking appliances require more heat input than electric or steam equipment of the same type and size. In the case of gas fuel, the American Gas Association established an overall figure of approximately 60% more. Where appliances are installed under an effective hood, only radiant heat adds to the cooling load; convected and latent heat from the cooking process and combustion products are exhausted and do not enter the kitchen. To compensate for 60% higher input ratings, Equation 7-29 must be used with **fuel-fired appliances**, since the appliance surface temperatures are the same and the extra heat input combustion products are exhausted to outdoors. This correction is made by introducing a flue loss factor (F_{FL}) of 1.60 as follows:

$$q_a = (q_i F_{UA} F_{RA}) / F_{FL} \tag{7-29}$$

or

$$q_a = 0.10 q_i \tag{7-30}$$

Factors for seven typical electrical and steam appliances are found in Table 7-20.

Unhooded Equipment. For all cooking appliances not installed under an exhaust hood or directly vent-connected and located in the conditioned area, the heat gain may be estimated as 50% (F_U = 0.50) or the rated hourly input, regardless of the type of energy or fuel used. On average, 34% of the heat may be assumed to be latent and the remaining 66% sensible. Note that cooking appliances ventilated by "ductless" hoods should be treated as unhooded appliances from the perspective of estimating heat gain. In other words, all energy consumed by the appliance and all moisture produced by the cooking process is introduced to the kitchen as a sensible or latent cooling load.

Table 7-18 Heat Gain from Typical Electric Motors

				Location of Motor and Driven Equipment with Respect to Conditioned Space or Airstream		
				A	B	C
Motor Name-plate or Rated Horse-power	Motor Type	Nominal rpm	Full Load Motor Effi-ciency, %	Motor in, Driven Equip-ment in, Btu/h	Motor out, Driven Equip-ment in, Btu/h	Motor in, Driven Equip-ment out, Btu/h
0.05	Shaded pole	1500	35	360	130	240
0.08	Shaded pole	1500	35	580	200	380
0.125	Shaded pole	1500	35	900	320	590
0.16	Shaded pole	1500	35	1160	400	760
0.25	Split phase	1750	54	1180	640	540
0.33	Split phase	1750	56	1500	840	660
0.50	Split phase	1750	60	2120	1270	850
0.75	3-Phase	1750	72	2650	1900	740
1	3-Phase	1750	75	3390	2550	850
1.5	3-Phase	1750	77	4960	3820	1140
2	3-Phase	1750	79	6440	5090	1350
3	3-Phase	1750	81	9430	7640	1790
5	3-Phase	1750	82	15,500	12,700	2790
7.5	3-Phase	1750	84	22,700	19,100	3640
10	3-Phase	1750	85	29,900	24,500	4490
15	3-Phase	1750	86	44,400	38,200	6210
20	3-Phase	1750	87	58,500	50,900	7610
25	3-Phase	1750	88	72,300	63,600	8680
30	3-Phase	1750	89	85,700	76,300	9440
40	3-Phase	1750	89	114,000	102,000	12,600
50	3-Phase	1750	89	143,000	127,000	15,700
60	3-Phase	1750	89	172,000	153,000	18,900
75	3-Phase	1750	90	212,000	191,000	21,200
100	3-Phase	1750	90	283,000	255,000	28,300
125	3-Phase	1750	90	353,000	318,000	35,300
150	3-Phase	1750	91	420,000	382,000	37,800
200	3-Phase	1750	91	569,000	509,000	50,300
250	3-Phase	1750	91	699,000	636,000	62,900

Recommended Heat Gain Values. As an alternative procedure, Table 7-21 lists recommended rates of heat gain from typical commercial cooking appliances. The data in the "with hood" columns assume installation under a properly designed exhaust hood connected to a mechanical fan exhaust system.

Hospital and Laboratory Equipment. Hospital and laboratory equipment items are major sources of heat gain in conditioned spaces. Care must be taken in evaluating the probability and duration of simultaneous usage when many components are concentrated in one area, such as a laboratory, an operating room, etc. Commonly, heat gain from equipment in a laboratory ranges from 15 to 70 Btu/h·ft^2 or, in laboratories with outdoor exposure, as much as four times the heat gain from all other sources combined.

Medical Equipment. It is more difficult to provide generalized heat gain recommendations for medical equipment than for general office equipment because medical equipment is

Table 7-19 Minimum Nominal Full-Load Efficiency for 60 Hz NEMA General-Purpose Electric Motors (Subtype I) Rated 600 V or Less (Random Wound)*

(Table 4A, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

	Open Drip-Proof Motors			Totally Enclosed Fan-Cooled Motors		
Number of Poles ⇒	**2**	**4**	**6**	**2**	**4**	**6**
Synchronous Speed (RPM) ⇒	**3600**	**1800**	**1200**	**3600**	**1800**	**1200**
Motor Horsepower						
1	77.0	85.5	82.5	77.0	85.5	82.5
1.5	84.0	86.5	86.5	84.0	86.5	87.5
2	85.5	86.5	87.5	85.5	86.5	88.5
3	85.5	89.5	88.5	86.5	89.5	89.5
5	86.5	89.5	89.5	88.5	89.5	89.5
7.5	88.5	91.0	90.2	89.5	91.7	91.0
10	89.5	91.7	91.7	90.2	91.7	91.0
15	90.2	93.0	91.7	91.0	92.4	91.7
20	91.0	93.0	92.4	91.0	93.0	91.7
25	91.7	93.6	93.0	91.7	93.6	93.0
30	91.7	94.1	93.6	91.7	93.6	93.0
40	92.4	94.1	94.1	92.4	94.1	94.1
50	93.0	94.5	94.1	93.0	94.5	94.1
60	93.6	95.0	94.5	93.6	95.0	94.5
75	93.6	95.0	94.5	93.6	95.4	94.5
100	93.6	95.4	95.0	94.1	95.4	95.0
125	94.1	95.4	95.0	95.0	95.4	95.0
150	94.1	95.8	95.4	95.0	95.8	95.8
200	95.0	95.8	95.4	95.4	96.2	95.8

Source: ASHRAE *Standard* 90.1-2013.

*Nominal efficiencies established in accordance with NEMA *Standard* MG1. Design A and Design B are National Electric Manufacturers Association (NEMA) design class designations for fixed-frequency small and medium AC squirrel-cage induction motors.

Table 7-20 Heat Gain Factors of Typical Appliances under Hoods

Appliance	**Usage Factor F_U**	**Radiation Factor F_R**	**Load Factor $F_L = F_U F_R$ Elec/Steam**
Griddle	0.16	0.45	0.07
Fryer	0.06	0.43	0.03
Convection oven	0.42	0.17	0.07
Charbroiler	0.83	0.29	0.24
Open-top range			
without oven	0.34	0.46	0.16
Hot-top range			
without oven	0.79	0.47	0.37
with oven	0.59	0.48	0.28
Steam cooker	0.13	0.30	0.04

much more varied in type and in application. Some heat gain testing has been done and can be presented, but the equipment included represents only a small sample of the type of equipment that may be encountered.

The data presented for medical equipment in Table 7-22 are relevant for portable and bench-top equipment. Medical equipment is very specific and can vary greatly from application to application. The data are presented to provide guidance in only the most general sense. For large equipment, such as MRI, engineers must obtain heat gain from the manufacturer.

Laboratory Equipment. Equipment in laboratories is similar to medical equipment in that it will vary significantly from space to space. Chapter 16 of the 2015 *ASHRAE Handbook—HVAC Applications* discusses heat gain from equipment, which may range from 5 to 25 W/ft^2 in highly automated laboratories. Table 7-23 lists some values for laboratory equipment, but, as is the case for medical equipment, it is for general guidance only.

Office Equipment

Computers, printers, copiers, etc., can generate very significant heat gains, sometimes greater than all other gains combined. ASHRAE research project RP-822 developed a method to measure the actual heat gain from equipment in buildings and the radiant/convective percentages (Hosni et al. 1998; Jones et al. 1998). This methodology was then incorporated into ASHRAE research project RP-1055 and applied to a wide range of equipment (Hosni et al. 1999) as a follow-up to independent research by Wilkins and McGaffin (1994) and Wilkins et al. (1991). Komor (1997) found similar results. Analysis of measured data showed that results for office equipment could be generalized, but results from laboratory and hospital equipment proved too diverse. The following general guidelines for office equipment are a result of these studies.

Nameplate Versus Measured Energy Use. Nameplate data rarely reflect the actual power consumption of office equipment. Actual power consumption is assumed to equal total (radiant plus convective) heat gain, but its ratio to the nameplate value varies widely. ASHRAE research project RP-1055 (Hosni et al. 1999) found that, for general office equipment with nameplate power consumption of less than 1000 W, the actual ratio of total heat gain to nameplate ranged from 25% to 50%, but when all tested equipment is considered, the range is broader. Generally, if the nameplate value is the only information known and no actual heat gain data are available for similar equipment, it is conservative to use 50% of nameplate as heat gain and more nearly correct if 25% of nameplate is used. Much better results can be obtained, however, by considering heat gain to be predictable based on the type of equipment. However, if the device has a mainly resistive internal electric load (e.g., a space heater), the nameplate rating may be a good estimate of its peak energy dissipation.

Computers. Based on tests by Hosni et al. (1999) and Wilkins and McGaffin (1994), nameplate values on computers should be ignored when performing cooling load calculations. Table 7-24 presents typical heat gain values for computers with varying degrees of safety factor.

Monitors. Based on monitors tested by Hosni et al. (1999), heat gain for cathode ray tube (CRT) monitors correlates approximately with screen size as

$$q_{\text{mon}} = 5S - 20 \tag{7-31}$$

where

q_{mon} = sensible heat gain from monitor, W

S = nominal screen size, in.

Table 7-24 shows typical values.

Flat-panel monitors have replaced CRT monitors in many workplaces. Power consumption, and thus heat gain, for flat-panel displays are significantly lower than for CRTs. Consult

Table 7-21A Recommended Rates of Radiant and Convective Heat Gain from Unhooded Electric Appliances During Idle (Ready-to-Cook) Conditions

(Table 5A, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

	Energy Rate, Btu/h		Rate of Heat Gain, Btu/h				Usage Factor	Radiation Factor
Appliance	Rated	Standby	Sensible Radiant	Sensible Convective	Latent	Total	F_U	F_R
Cabinet: hot serving (large), insulated[a]	6,800	1,200	400	800	0	1,200	0.18	0.33
hot serving (large), uninsulated	6,800	3,500	700	2,800	0	3,500	0.51	0.20
proofing (large)[a]	17,400	1,400	1,200	0	200	1,400	0.08	0.86
proofing (small 15-shelf)	14,300	3,900	0	900	3,000	3,900	0.27	0.00
Cheesemelter[b]	8,200	3,300	1,500	1,800	0	3,300	0.41	0.45
Coffee brewing urn	13,000	1,200	200	300	700	1,200	0.09	0.17
Drawer warmers, 2-drawer (moist holding)[a]	4,100	500	0	0	200	200	0.12	0.00
Egg cooker[b]	8,100	850	200	650	0	850	0.10	0.26
Espresso machine*	8,200	1,200	400	800	0	1,200	0.15	0.33
Food warmer: steam table (2-well-type)	5,100	3,500	300	600	2,600	3,500	0.69	0.09
Freezer (small)	2,700	1,100	500	600	0	1,100	0.41	0.45
Fryer, countertop, open deep fat[b]	15,700	1,500	700	800	0	1,500	0.09	0.47
Griddle, countertop[b]	27,300	6,100	2,900	3,200	0	6,100	0.22	0.48
Hot dog roller[b]	5,500	4,200	900	3,300	0	4,200	0.77	0.22
Hot plate: single element, high speed	3,800	3,400	1,100	2,300	0	3,400	0.89	0.32
Hot-food case (dry holding)[a]	31,100	2,500	900	1,600	0	2,500	0.08	0.36
Hot-food case (moist holding)[a]	31,100	3,300	900	1,800	600	3,300	0.11	0.27
Induction hob, countertop[b]	17,100	0	0	0	0	0	0.00	0.00
Microwave oven: commercial	5,800	0	0	0	0	0	0.00	0.00
Oven: countertop conveyorized bake/finishing[b]	17,100	13,500	2,500	11,000	0	13,500	0.79	0.18
Panini[b]	6,100	2,300	700	1,600	0	2,300	0.37	0.29
Popcorn popper[b]	2,900	400	100	300	0	400	0.14	0.24
Rapid-cook oven (quartz-halogen)[a]	41,000	0	0	0	0	0	0.00	0.00
Rapid-cook oven (microwave/convection)[b]	19,400	3,900	300	3,600	0	3,900	0.20	0.08
Reach-in refrigerator[a]	4,800	1,200	300	900	0	1,200	0.25	0.25
Refrigerated prep table[a]	2,000	900	600	300	0	900	0.45	0.67
Rice cooker[b]	5,300	300	50	250	0	300	0.05	0.17
Soup warmer[b]	2,700	1,300	0	200	1,100	1,300	0.49	0.00
Steamer (bun)[b]	5,100	700	100	600	0	700	0.13	0.16
Steamer, countertop[b]	28,300	1,200	0	800	400	1,200	0.04	0.00
Toaster: 4-slice pop up (large): cooking	6,100	3,000	200	1,400	1,000	2,600	0.49	0.07
contact (vertical)[b]	8,900	2,600	600	2,000	0	2,600	0.29	0.24
conveyor (large)	32,800	10,300	3,000	7,300	0	10,300	0.31	0.29
small conveyor[b]	6,000	5,800	1,200	4,600	0	5,800	0.98	0.21
Tortilla grill[b]	7,500	3,600	900	2,700	0	3,600	0.47	0.25
Waffle iron[b]	9,200	900	200	700	0	900	0.10	0.22

Sources: Swierczyna et al. (2008, 2009); with the following exceptions as noted.

[a] Swierczyhna et al. (2009) only.

[b] Additions and updates from ASHRAE research project RP-1631 (Kong and Zhang 2016; Kong et al 2016).

manufacturers' literature for average power consumption data for use in heat gain calculations.

Laser Printers. Hosni et al. (1999) found that power consumption, and therefore the heat gain, of laser printers depended largely on the level of throughput for which the printer was designed. Smaller printers tend to be used more intermittently, and larger printers may run continuously for longer periods.

Table 7-25 presents data on laser printers. These data can be applied by taking the value for continuous operation and then applying an appropriate diversity factor. This would likely be most appropriate for larger open office areas. Another approach, which may be appropriate for a single room or small area, is to take the value that most closely matches the expected operation of the printer with no diversity. New data for printers is given in Table 7-25B.

Copiers. Hosni et al. (1999) also tested five photocopy machines, including desktop and office (freestanding high-volume copiers) models. Larger machines used in production environments were not addressed. Table 7-25 summarizes the results. Desktop copiers rarely operate continuously, but office copiers frequently operate continuously for periods of an hour or more. Large, high-volume photocopiers often include provisions for exhausting air outdoors; if so equipped, the direct-to-space or system makeup air heat gain needs to be included in the load calculation. Also, when the air is dry, humidifiers are often operated near copiers to limit static electricity; if this occurs during cooling mode, their load on HVAC systems should be considered.

Miscellaneous Office Equipment. Table 7-26 presents data on miscellaneous office equipment such as vending machines and mailing equipment.

Table 7-21B Recommended Rates of Radiant and Convective Heat Gain from Unhooded Electric Appliances During Cooking Conditions

(Table 5B, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

	Energy Rate, Btu/h		Rate of Heat Gain, Btu/h					
Appliance	Rated	Cooking	Sensible Radiant	Sensible Convective	Latent	Total	Usage Factor F_U	Radiation Factor F_R
Cheesemelter	8,200	9,300	1,500	3,700	2,000	7,200	1.13	0.16
Egg cooker	8,100	4,100	200	1,300	2,200	3,700	0.50	0.05
Fryer, countertop, open deep fryer	15,700	13,000	700	1,700	5,600	8,000	0.83	0.05
Griddle, countertop	27,300	11,200	2,900	2,200	4,400	9,500	0.41	0.26
Hot dog roller	5,500	5,400	900	2,100	2,300	5,300	0.99	0.17
Hot plate, single burner	3,800	3,400	1,100	2,100	200	3,400	0.90	0.32
Induction hob, countertop	17,100	2,200	0	1,100	1,100	2,200	0.13	0.00
Oven, conveyor	17,100	14,600	2,500	8,400	700	11,600	0.86	0.17
Microwave	5,800	8,100	0	3,200	3,400	6,600	1.39	0.00
Rapid cook	19,400	7,900	300	4,200	2,600	7,100	0.41	0.04
Panini grill	6,100	4,700	700	2,400	500	3,600	0.76	0.14
Popcorn popper	2,900	2,000	100	800	700	1,600	0.68	0.05
Rice cooker	5,300	4,000	50	300	200	550	0.75	0.01
Soup warmer	2,700	2,900	0	300	2,400	2,700	1.05	0.00
Steamer (bun)	5,100	2,700	100	800	1,700	2,600	0.53	0.04
Steamer, countertop	28,300	26,400	0	1,700	23,700	25,400	0.93	0.00
Toaster, conveyor	6,000	5,800	1,200	3,300	1,300	5,800	0.98	0.21
Vertical	8,900	6,300	600	2,400	1,100	4,100	0.71	0.10
Tortilla grill	7,500	7,500	900	4,300	2,300	7,500	1.00	0.12
Waffle maker	9,200	4,000	200	1,200	1,900	3,300	0.44	0.05

Source: ASHRAE research project RP-1631 (Zhang et al. 2015).

Table 7-21C Recommended Rates of Radiant Heat Gain from Hooded Electric Appliances During Idle (Ready-to-Cook) Conditions

(Table 5C, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

	Energy Rate, Btu/h		Rate of Heat Gain, Btu/h		
Appliance	Rated	Standby	Sensible Radiant	Usage Factor F_u	Radiation Factor F_r
Broiler: underfired 3 ft	36,900	30,900	10,800	0.84	0.35
Cheesemelter*	12,300	11,900	4,600	0.97	0.39
Fryer: kettle	99,000	1,800	500	0.02	0.28
Fryer: open deep-fat, 1-vat	47,800	2,800	1,000	0.06	0.36
Fryer: pressure	46,100	2,700	500	0.06	0.19
Griddle: double sided 3 ft (clamshell down)*	72,400	6,900	1,400	0.1	0.2
Griddle: double sided 3 ft (clamshell up)*	72,400	11,500	3,600	0.16	0.31
Griddle: flat 3 ft	58,400	11,500	4,500	0.2	0.39
Griddle-small 3 ft*	30,700	6,100	2,700	0.2	0.44
Induction cooktop*	71,700	0	0	0	0
Induction wok*	11,900	0	0	0	0
Oven: combi: combi-mode*	56,000	5,500	800	0.1	0.15
Oven: combi: convection mode	56,000	5,500	1,400	0.1	0.25
Oven: convection full-size	41,300	6,700	1,500	0.16	0.22
Oven: convection half-size*	18,800	3,700	500	0.2	0.14
Pasta cooker*	75,100	8,500	0	0.11	0
Range top: top off/oven on*	16,600	4,000	1,000	0.24	0.25
Range top: 3 elements on/oven off	51,200	15,400	6,300	0.3	0.41
Range top: 6 elements on/oven off	51,200	33,200	13,900	0.65	0.42
Range top: 6 elements on/oven on	67,800	36,400	14,500	0.54	0.4
Range: hot-top	54,000	51,300	11,800	0.95	0.23
Rotisserie*	37,900	13,800	4,500	0.36	0.33
Salamander*	23,900	23,300	7,000	0.97	0.3
Steam kettle: large (60 gal) simmer lid down*	110,600	2,600	100	0.02	0.04
Steam kettle: small (40 gal) simmer lid down*	73,700	1,800	300	0.02	0.17
Steamer: compartment: atmospheric*	33,400	15,300	200	0.46	0.01
Tilting skillet/braising pan	32,900	5,300	0	0.16	0

Source: Swierczyna et al. (2008, 2009). Items with an asterisk appear only in Swierczyna (2009).

Table 7-21D Recommended Rates of Radiant Heat Gain from Hooded Gas Appliances During Idle (Ready-to-Cook) Conditions

(Table 5D, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

	Energy Rate, Btu/h		Rate of Heat Gain, Btu/h		
Appliance	Rated	Standby	Sensible Radiant	Usage Factor F_u	Radiation Factor F_r
Broiler: batch*	95,000	69,200	8,100	0.73	0.12
Broiler: chain (conveyor)	132,000	96,700	13,200	0.73	0.14
Broiler: overfired (upright)*	100,000	87,900	2,500	0.88	0.03
Broiler: underfired 3 ft	96,000	73,900	9,000	0.77	0.12
Fryer: doughnut	44,000	12,400	2,900	0.28	0.23
Fryer: open deep-fat, 1 vat	80,000	4,700	1,100	0.06	0.23
Fryer: pressure	80,000	9,000	800	0.11	0.09
Griddle: double sided 3 ft (clamshell down)*	108,200	8,000	1,800	0.07	0.23
Griddle: double sided 3 ft (clamshell up)*	108,200	14,700	4,900	0.14	0.33
Griddle: flat 3 ft	90,000	20,400	3,700	0.23	0.18
Oven: combi: combi-mode*	75,700	6,000	400	0.08	0.07
Oven: combi: convection mode	75,700	5,800	1,000	0.08	0.17
Oven: convection full-size	44,000	11,900	1,000	0.27	0.08
Oven: conveyor (pizza)	170,000	68,300	7,800	0.4	0.11
Oven: deck	105,000	20,500	3,500	0.2	0.17
Oven: rack mini-rotating*	56,300	4,500	1,100	0.08	0.24
Pasta cooker*	80,000	23,700	0	0.3	0
Range top: top off/oven on*	25,000	7,400	2,000	0.3	0.27
Range top: 3 burners on/oven off	120,000	60,100	7,100	0.5	0.12
Range top: 6 burners on/oven off	120,000	120,800	11,500	1.01	0.1
Range top: 6 burners on/oven on	145,000	122,900	13,600	0.85	0.11
Range: wok*	99,000	87,400	5,200	0.88	0.06
Rethermalizer*	90,000	23,300	11,500	0.26	0.49
Rice cooker*	35,000	500	300	0.01	0.6
Salamander*	35,000	33,300	5,300	0.95	0.16
Steam kettle: large (60 gal) simmer lid down*	145,000	5,400	0	0.04	0
Steam kettle: small (10 gal) simmer lid down*	52,000	3,300	300	0.06	0.09
Steam kettle: small (40 gal) simmer lid down	100,000	4,300	0	0.04	0
Steamer: compartment: atmospheric *	26,000	8,300	0	0.32	0
Tilting skillet/braising pan	104,000	10,400	400	0.1	0.04

Source: Swierczyna et al. (2008, 2009). Items with an asterisk appear only in Swierczyna (2009).

Table 7-21E Recommended Rates of Radiant Heat Gain from Hooded Solid Fuel Appliances During Idle (Ready-to-Cook) Conditions

(Table 5E, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

	Energy Rate, Btu/h	Rate of Heat Gain, Btu/h			
Appliance	Rated	Standby	Sensible	Usage Factor F_u	Radiation Factor F_r
Broiler: solid fuel: charcoal	40 lb	42,000	6200	N/A	0.15
Broiler: solid fuel: wood (mesquite)*	40 lb	49,600	7000	N/A	0.14

Source: Swierczyna et al. (2008, 2009). Items with an asterisk appear only in Swierczyna (2009).

Diversity. The ratio of measured peak electrical load at equipment panels to the sum of the maximum electrical load of each individual item of equipment is the usage diversity. A small, one- or two-person office containing equipment listed in Tables 7-24 to 7-26 usually contributes heat gain to the space at the sum of the appropriate listed values. Progressively larger areas with many equipment items always experience some degree of usage diversity resulting from whatever percentage of such equipment is not in operation at any given time.

Wilkins and McGaffin (1994) measured diversity in 23 areas within five different buildings totaling over 275,000 ft^2. Diversity was found to range between 37 and 78%, with the average (normalized based on area) being 46%. Figure 7-7 illustrates the relationship between nameplate, sum of peaks, and actual electrical load with diversity accounted for, based on the average of the total area tested. Data on actual diversity can be used as a guide, but diversity varies significantly with occupancy. The proper diversity factor for an office of mail-order catalog telephone operators is different from that for an office of sales representatives who travel regularly.

ASHRAE research project RP-1093 derived diversity profiles for use in energy calculations (Abushakra et al. 2004; Claridge et al. 2004). Those profiles were derived from available measured data sets for a variety of office buildings, and indicated a range of peak weekday diversity factors for lighting ranging from 70 to 85% and for receptacles (appliance load) between 42 and 89%.

Heat Gain per Unit Area. Wilkins and Hosni (2000, 2011) and Wilkins and McGaffin (1994) summarized research on a heat gain per unit area basis. Diversity testing showed that the actual heat gain per unit area, or load factor, ranged from 0.44 to 1.08 W/ft^2, with an average (normalized based on area) of 0.81 W/ft^2. Spaces tested were fully occu-

Table 7-21F Recommended Rates of Radiant and Convective Heat Gain from Warewashing Equipment During Idle (Standby) or Washing Conditions

(Table 5F, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

	Energy Rate, Btu/h		Rate of Heat Gain, Btu/h						
			Unhooded				Hooded		
Appliance	Rated	Standby/ Washing	Sensible Radiant	Sensible Convective	Latent	Total	Sensible Radiant	Usage Factor F_U	Radiation Factor F_R
Dishwasher: conveyor type, hot-water sanitizing, washing	46,800	N/A	0	12,100	47,000	59,100	0	N/A	0.00
Standby	46,800	5,700	0	1,600	4,100	5,700	0	0.12	0.00
Dishwasher: conveyor type, chemical sanitizing, washing	46,800	43,600	0	11,100	35,400	46,500	0	0.93	0.00
Standby	46,800	5,700	0	1,600	4,100	5,700	0	0.12	0.00
Dishwasher: door type, hot-water sanitizing, washing	60,100	18,500	0	7,600	25,200	32,800	0	0.31	0.00
With heat recovery and vapor reduction	51,900	27,100	0	5,800	13,100	18,900	0	0.52	0.00
Standby	18,400	1,200	0	2,280	4,170	6,450	0	0.35	0.00
Dishwasher: door type, chemical sanitizing, washing	30,000	15,600	0	3,900	13,200	17,100	0	0.52	0.00
Standby	18,400	1,200	0	900	300	1,200	0	0.07	0.00
Dishwasher: door type, chemical sanitizing, dump and fill, washing	6,100	3,000	0	2,900	4,200	7,100	0	0.49	0.00
Standby	6,100	3,000	0	0	0	0	0	0.49	0.00
Pot and pan washer: door type, hot-water sanitizing, washing	53,200	36,400	0	6,000	23,500	29,500	0	0.68	0.00
With heat recovery and vapor reduction	53,200	35,200	0	5,500	19,000	24,500	0	0.66	0.00
Dishwasher: under-counter type, hot-water sanitizing, washing	28,500	7,600	800	3,200	6,900	10,900	800	0.27	0.11
With heat recovery and vapor reduction	26,600	22,800	0	2,000	1,100	3,100	0	0.86	0.00
Standby	26,600	1,700	800	500	400	1,700	800	0.06	0.47
Dishwasher: under-counter type, chemical sanitizing, washing	28,500	6,900	0	2,200	4,900	7,100	0	0.24	0.00
Standby	26,600	1,700	800	500	400	1,700	0	0.06	0.47
Booster heater	130,000	0	500	0	0	0	500	0	N/A

Sources: PG&E (2010-2016), Swierczyna et al. (2008) and (2009).

Table 7-22 Recommended Heat Gain from Typical Medical Equipment

(Table 6, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

Equipment	Nameplate, W	Peak, W	Average, W
Anesthesia system	250	177	166
Blanket warmer	500	504	221
Blood pressure meter	180	33	29
Blood warmer	360	204	114
ECG/RESP	1440	54	50
Electrosurgery	1000	147	109
Endoscope	1688	605	596
Harmonical scalpel	230	60	59
Hysteroscopic pump	180	35	34
Laser sonics	1200	256	229
Optical microscope	330	65	63
Pulse oximeter	72	21	20
Stress treadmill	N/A	198	173
Ultrasound system	1800	1063	1050
Vacuum suction	621	337	302
X-ray system	968		82
X-ray system	1725	534	480
X-ray system	2070		18

Source: Hosni et al. (1999).

pied and highly automated, comprising 21 unique areas in five buildings, with a computer and monitor at every workstation. Table 7-27 presents a range of load factors with a subjective description of the type of space to which they would apply. The medium load density is likely to be appropriate for most standard office spaces. Medium/heavy or heavy load densities may be encountered but can be considered extremely conservative estimates even for densely populated and highly automated spaces. Table 7-28 indicates applicable diversity factors.

Table 7-23 Recommended Heat Gain from Typical Laboratory Equipment

(Table 7, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

Equipment	Nameplate, W	Peak, W	Average, W
Analytical balance	7	7	7
Centrifuge	138	89	87
Centrifuge	288	136	132
Centrifuge	5500	1176	730
Electrochemical analyzer	50	45	44
Electrochemical analyzer	100	85	84
Flame photometer	180	107	105
Fluorescent microscope	150	144	143
Fluorescent microscope	200	205	178
Function generator	58	29	29
Incubator	515	461	451
Incubator	600	479	264
Incubator	3125	1335	1222
Orbital shaker	100	16	16
Oscilloscope	72	38	38
Oscilloscope	345	99	97
Rotary evaporator	75	74	73
Rotary evaporator	94	29	28
Spectronics	36	31	31
Spectrophotometer	575	106	104
Spectrophotometer	200	122	121
Spectrophotometer	N/A	127	125
Spectro fluorometer	340	405	395
Thermocycler	1840	965	641
Thermocycler	N/A	233	198
Tissue culture	475	132	46
Tissue culture	2346	1178	1146

Source: Hosni et al. (1999).

Radiant/Convective Split. ASHRAE research project RP-1482 (Hosni and Beck 2008) is examining the radiant/convective split for common office equipment; the most important differentiating feature is whether the equipment

Table 7-24A Recommended Heat Gain for Typical Desktop Computers

(Table 8A, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

Description	Nameplate Power,[a] W	Peak Heat Gain,[b, d] W
Manufacturer 1		
3.0 GHz processor, 4 GB RAM, $n = 1$	NA	83
3.3 GHz processor, 8 GB RAM, $n = 8$	NA	50
3.5 GHz processor, 8 GB RAM, $n = 2$	NA	42
3.6 GHz processor, 16 GB RAM, $n = 2$	NA	66
3.3 GHz processor, 16 GB RAM, $n = 2$	NA	52
4.0 GHz processor, 16 GB RAM, $n = 1$	NA	83
3.3 GHz processor, 8 GB RAM, $n = 1$	NA	84
3.7 GHz processor, 32 GB RAM, $n = 1$	750	116
	NA	102
3.5 GHz processor, 16 GB RAM, $n = 3$[c]	550	144
	NA	93
Manufacturer 2		
3.6 GHz processor, 32 GB RAM, $n = 8$	NA	80
3.6 GHz processor, 16 GB RAM, $n = 1$	NA	78
3.4 GHz processor, 32 GB RAM, $n = 1$	NA	72
3.4 GHz processor, 24 GB RAM, $n = 1$	NA	86
3.50 GHz processor, 4 GB RAM, $n = 1$	NA	26
3.3 GHz processor, 8 GB RAM, $n = 1$	NA	78
3.20 GHz processor, 8 GB RAM, $n = 1$	NA	61
3.20 GHz processor, 4 GB RAM, $n = 1$	NA	44
2.93 GHz processor, 16 GB RAM, $n = 1$	NA	151
2.67 GHz processor, 8 GB RAM, $n = 1$	NA	137
Average 15-min peak power consumption (range)	82 (26-151)	

Source: Bach and Sarfraz (2017)

n = number of tested equipment of same configuration.

[a] Nameplate for desktop computer is present on its power supply, which is mounted inside desktop, hence not accessible for most computers, where NA = not available.

[b] For equipment peak heat gain value, highest 15-min interval of recorded data is listed in tables.

[c] For tested equipment with same configuration, increasing power supply size does not increase average power consumption.

[d] Approximately 90% convective heat gain and 10% radiative heat gain.

Table 7-24B Recommended Heat Gain for Typical Laptops and Laptop Docking Station

(Table 8A, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

Equipment	Description	Nameplate Power,[a] W	Peak Heat Gain,[b, c] W
Laptop computer	Manufacturer 1, 2.6 GHz processor, 8 GB RAM, $n = 1$	NA	46
	Manufacturer 2, 2.4 GHz processor, 4 GB RAM, $n = 1$	NA	59
Average 15-min peak power consumption (range)		53 (46-59)	
Laptop with docking station	Manufacturer 1, 2.7 GHz processor, 8 GB RAM, $n = 1$	NA	38
	1.6 GHz processor, 8 GB RAM, $n = 2$	NA	45
	2.0 GHz processor, 8 GB RAM, $n = 1$	NA	50
	2.6 GHz processor, 4 GB RAM, $n = 1$	NA	51
	2.4 GHz processor, 8 GB RAM, $n = 1$	NA	40
	2.6 GHz processor, 8 GB RAM, $n = 1$	NA	35
	2.7 GHz processor, 8 GB RAM, $n = 1$	NA	59
	3.0 GHz processor, 8 GB RAM, $n = 3$	NA	70
	2.9 GHz processor, 32 GB RAM, $n = 3$	NA	58
	3.0 GHz processor, 32 GB RAM, $n = 1$	NA	128
	3.7 GHz processor, 32 GB RAM, $n = 1$	NA	63
	3.1 GHz processor, 32 GB RAM, $n = 1$	NA	89
Average 15-min peak power consumption (range)		61 (26-151)	

Source: Bach and Sarfraz (2017)

n = number of tested equipment of same configuration.

[a] Voltage and amperage information for laptop computer and laptop docking station is available on power supply nameplates; however, nameplate does not provide information on power consumption, where NA = not available.

[b] For equipment peak heat gain value, the highest 15-min interval of recorded data is listed in tables.

[c] Approximately 75% convective heat gain and 25% radiative heat gain.

Table 7-24C Recommended Heat Gain for Typical Tablet PC

(Table 8A, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

Description	Nameplate Power,[a] W	Peak Heat Gain,[b] W
1.7 GHz processor, 4 GB RAM, $n = 1$	NA	42
2.2 GHz processor, 16 GB RAM, $n = 1$	NA	40
2.3 GHz processor, 8 GB RAM, $n = 1$	NA	30
2.5 GHz processor, 8 GB RAM, $n = 1$	NA	31
Average 15-min peak power consumption (range)	36 (31-42)	

Source: Bach and Sarfraz (2017)

n = number of tested equipment of same configuration.

[a] Voltage and amperage information for tablet PC is available on power supply nameplate; however, nameplate does not provide information on power consumption, where NA = not available.

[b] For equipment peak heat gain value, highest 15-min interval of recorded data is listed in tables.

had a cooling fan. Footnotes in Tables 7-24 and 7-25 summarizes those results.

7.3.5 Ventilation and Infiltration Air

Wind and pressure differences cause outdoor air to infiltrate into the cracks around doors and windows, resulting in localized sensible and latent heat gains. Also, some outdoor ventilation air is needed to eliminate any odors. This outdoor air imposes a cooling and dehumidifying load on the cooling coil because heat and/or moisture must be removed from the air. Heat gains due to infiltration and ventilation can be computed using equations in Chapter 5

These equations are valid for calculating the cooling load due to infiltration of outdoor air and also due to the positive introduction of air for ventilation, provided it is introduced directly into the space.

7.3.6 Moisture Transfer Through Permeable Building Materials

In the usual comfort air-conditioning application, moisture transfer through walls is often neglected, since the actual rate is small and the corresponding latent heat gain is insignificant. On the other hand, industrial applications frequently call for a low moisture content in a conditioned space. Here, moisture transfer cannot be neglected, as the latent heat gain accompanying this transfer may be of greater magnitude than any other latent heat gain. Under these conditions, proper calculation of the moisture transfer due to both air infiltration and diffusion through building materials is important.

7.3.7 Miscellaneous Sources of Heat

Fans that circulate air through HVAC systems add energy to the system by one or all of the following processes:

Table 7-24D Recommended Heat Gain for Typical Monitors

(Table 8A, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

Description[a]	Nameplate Power, W	Peak Heat Gain,[b, c] W
Manufacturer 1		
1397 mm LED flat screen, $n = 1$ (excluded from average because atypical size)	240	50
686 mm LED flat screen, $n = 2$	40	26
546 mm LED flat screen, $n = 2$	29	25
Manufacturer 2		
1270 mm 3D LED flat screen, $n = 1$ (excluded from average because atypical size)	94	49
Manufacturer 3		
864 mm LCD curved screen, $n = 1$ (excluded from average because atypical size and curved)	130	48
584 mm LED flat screen, $n = 3$	50	17
584 mm LED flat screen, $n = 1$	38	21
584 mm LED flat screen, $n = 1$	38	14
Manufacturer 4		
610 mm LED flat screen, $n = 1$	42	25
Manufacturer 5		
600 mm LED flat screen, $n = 1$	26	17
546 mm LED flat screen, $n = 1$	29	22
Manufacturer 6		
546 mm LED flat screen, $n = 1$	28	24
Average 15-min peak power consumption (range)	21 (14-26)	

Source: Bach and Sarfraz (2017)

n = number of tested equipment of same configuration.

[a] Screens with atypical size and shape are excluded for calculating average 15-min peak power consumption.

[b] For equipment peak heat gain value, highest 15-min interval of recorded data is listed in tables.

[c] Approximately 60% convective heat gain and 40% radiative heat gain.

Temperature rise in the airstream from fan inefficiency. Depending on the equipment, fan efficiencies generally range between 50 and 70%, with an average value of 65%. Thus, some 35% of the energy required by the fan appears as instantaneous heat gain to the air being transported.

Temperature rise in the airstream as a consequence of air static and velocity pressure. The fan energy that creates pressure to move air spreads throughout the entire air transport system in the process of conversion into sensible heat. Designers commonly assume that the temperature change equivalent of this heat occurs in one point in the system, depending on fan location.

Temperature rise from heat generated by motor and drive inefficiencies. The relatively small gains from fan motors and drives are normally disregarded unless the motor and/or drive are physically located within the conditioned airstream. Equations (7-24), (7-25), and (7-26) may be used to estimate heat gains from typical motors. Belt drive losses are often estimated as 3% of the rated motor power.

The location of each fan relative to other elements (primarily the cooling coil), the type of system (single zone, multizone, double-duct, terminal reheat, VAV, etc.), and the type of equipment control (space temperature alone, space temperature and relative humidity, and so forth) must be known before the analysis can be completed. A fan located upstream of the cooling coil (blow-through supply fan, return air fan, outdoor air fan) adds the heat equivalent to its inefficiency to the airstream temperature at that point; the cooling coil sees this as elevated entering dry-bulb temperature. A fan located downstream of the cooling coil raises the dry-bulb temperature of air leaving the cooling coil. This rise can be offset by reducing the cooling coil temperature, or alternatively, by increasing airflow across the cooling coil.

Unless return air duct systems are extensive or subjected to rigorous conditions, only the heat gained or lost by supply duct systems is significant. It is estimated as a percentage of space sensible cooling load (normally about 1%) and applied to the air dry-bulb temperature of the air leaving the coil in the form of an equivalent temperature reduction. .

Losses from air leakage out of (or into) ductwork or equipment can be greater than conventional duct heat gain or loss but is normally about the same or less. Outward duct leakage is a direct loss of cooling and/or dehumidifying capacity and must be offset by increased airflow (sometimes reduced supply air temperature) unless it enters the conditioned space directly. Inward duct leakage causes temperature and/or humidity variations, but it is often ignored under ordinary circumstances due to the low temperature and pressure differentials involved

A well-designed and properly installed duct system should not leak more than 1 to 3% of the total system airflow. HVAC equipment and volume control units connected into a duct system should be delivered from manufacturers with allowable leakage rates not exceeding 1 or 2% of maximum airflow. Where duct systems are specified to be sealed and leak-tested, both low- and high-pressure types can be constructed and required to fall in this range. Latent heat considerations are frequently ignored.

Poorly designed or installed duct systems can have leakage rates of 10 to 30%. Leakage from low-pressure lighting troffer connections lacking proper taping and sealing runs up to 35% or more of the terminal air supply. Such extremes can ruin the validity of any load calculation. As such, they may not affect overall system loads enough to cause problems; they will, however, always adversely impact required supply air quantities for most air-conditioning systems. Also, using uninsulated supply ductwork running through return air plenums results in high thermal leakage, loss of space cooling capability by the supply air, and condensation difficulties during a warm start-up.

7.4 Description of Radiant Time Series (RTS)

Design cooling loads are based on the assumption of **steady-periodic conditions (**i.e., the design day's weather, occupancy, and heat gain conditions are identical to those for preceding days such that the loads repeat on an identical 24 h cyclical basis). Thus, the heat gain for a particular component at a particular hour is the same as 24 h prior, which is the same as 48 h prior, etc. This assumption is the basis for the RTS derivation from the HB method.

Table 7-25A Recommended Heat Gain from Typical Laser Printers and Copiers

(Table 9, Chapter 18, 2013 ASHRAE Handbook Fundamentals)

Equipment	Description	Nameplate Power, W	Average Power, W	Radiant Fraction
Laser printer, typical desktop, small-office type[a]	Printing speed up to 10 pages per minute	430	137	0.30[a]
	Printing speed up to 35 pages per minute	890	74	0.30[a]
	Printing speed up to 19 pages per minute	508	88	0.30[a]
	Printing speed up to 17 pages per minute	508	98	0.30[a]
	Printing speed up to 19 pages per minute	635	110	0.30[a]
	Printing speed up to 24 page per minute	1344	130	0.30[a]
Multifunction (copy, print, scan)[b]	Small, desktop type	600	30	d
		40	15	d
	Medium, desktop type	700	135	d
Scanner[b]	Small, desktop type	19	16	d
Copy machine[c]	Large, multiuser, office type	1750	800 (idle 260 W)	d (idle 0.00[c])
		1440	550 (idle 135 W)	d (idle 0.00[c])
		1850	1060 (idle 305 W)	d (idle 0.00[c])
Fax machine	Medium	936	90	d
	Small	40	20	d
Plotter	Manufacturer A	400	250	d
	Manufacturer B	456	140	d

Source: Hosni and Beck (2008).

[a] Various laser printers commercially available and commonly used in personal offices were tested for power consumption in print mode, which varied from 75 to 140 W, depending on model, print capacity, and speed. Average power consumption of 110 W may be used. Split between convection and radiation is approximately 70/30%.

[b] Small multifunction (copy, scan, print) systems use about 15 to 30 W; medium-sized ones use about 135 W. Power consumption in idle mode is negligible. Nameplate values do not represent actual power consumption and should not be used. Small, single-sheet scanners consume less than 20 W and do not contribute significantly to building cooling load.

[c] Power consumption for large copy machines in large offices and copy centers ranges from about 550 to 1100 W in copy mode. Consumption in idle mode varies from about 130 to 300 W. Count idle-mode power consumption as mostly convective in cooling load calculations.

[d] Split between convective and radiant heat gain was not determined for these types of equipment.

Cooling load calculations must address two time-delay effects inherent in building heat transfer processes: (1) delay of conductive heat gain through opaque massive exterior surfaces (walls, roofs, or floors) and (2) delay of radiative heat gain conversion to cooling loads.

Exterior walls and roofs conduct heat due to temperature differences between outdoor and indoor air. In addition, solar energy on exterior surfaces is absorbed, then transferred by conduction to the building interior. Due to the mass and thermal capacity of the wall or roof construction materials, there is a substantial time delay in heat input at the exterior surface becoming heat gain at the interior surface.

As described earlier in this chapter, most heat sources transfer energy to a room by a combination of convection and radiation. (See Table 7-29). The convection part of heat gain immediately becomes cooling load. The radiation part must first be absorbed by the finishes and mass of the interior room surfaces and becomes cooling load only when it is later transferred by convection from those surfaces to the room air. Thus, radiant heat gains become cooling loads over a delayed period of time.

Overview of the RTS Method

Figure 7-9 gives an overview of the radiant time series method. In the calculation of solar radiation, transmitted solar heat gain through windows, sol-air temperature, and infiltration, the RTS method is exactly the same as previous simplified methods (TFM and TETD/TA). Important areas that are different include the computation of conductive heat gain, the splitting of all heat gains into radiant and convective portions, and the conversion of radiant heat gains into cooling loads.

The RTS method accounts for both conduction time delay and radiant time delay effects by multiplying hourly heat gains by 24 h time series. The time series multiplication, in effect, distributes heat gains over time. Series coefficients, which are called **radiant time factors** and **conduction time factors**, are derived using the heat balance method. Radiant time factors reflect the percentage of an earlier radiant heat gain that becomes cooling load during the current hour. Likewise, conduction time factors reflect the percentage of an earlier heat gain at the exterior of a wall or roof that becomes heat gain at the inside during the current hour. By definition, each radiant or conduction time series must total 100%.

These series can be used to easily compare the time-delay impact of one construction versus another. This ability to compare choices is of particular benefit in the design process, when all construction details may not have been decided. Comparison can illustrate the magnitude of difference between the choices, allowing the engineer to apply judgment and make more informed assumptions in estimating the load.

RTS Procedure

The general procedure for calculating cooling load for each load component (lights, people, walls, roofs, windows, appliances, etc.) with RTS is as follows:

1. Calculate 24 h profile of component heat gain for design day (for conduction, first account for conduction time delay by applying conduction time series).

Table 7-25B Recommended Heat Gain for Typical Printers

(Table 9, Chapter 18, 2017 ASHRAE Handbook Fundamentals)

Equipment	Description	Max. Printing Speed, Pages per Minute	Nameplate Power, W	Peak Heat Gain,[a] W
Multifunction printer (copy, print, scan)	Large, multiuser, office type	40	1010	540 (Idle 29 W)
		30	1300	303 (Idle 116 W)
		28	1500	433 (Idle 28 W)
	Average 15-min peak power consumption (range)		425 (303-540)	
	Multiuser, medium-office type	35	900	732 (Idle 18 W)
	Desktop, small-office type	25	470	56 (Idle 3 W)
Monochrome printer	Desktop, medium-office type	55	1000	222
		45	680	61
	Average 15-min peak power consumption (range)		142 (61-222)	
Color printer	Desktop, medium-office type	40	620	120
Laser printer	Desktop, small-office type	14	310	89
		24	495	67
		26	1090	65
	Average 15-min peak power consumption (range)		74 (65-89)	
Plotter	Manufacturer 1		1600	571
	Manufacturer 2		270	173
	Average 15-min peak power consumption (range)		372 (173-571)	
Fax machine	Medium		1090	92
	Small		600	46
	Average 15-min peak power consumption (range)		69 (46-92)	

Source: Bach and Sarfraz (2017)

[a] Approximately 70% convective heat gain and 30% radiative heat gain.

2. Split heat gains into radiant and convective parts (see Table 7-27 for radiant and convective fractions).
3. Apply appropriate radiant time series to radiant part of heat gains to account for time delay in conversion to cooling load.
4. Sum convective part of heat gain and delayed radiant part of heat gain to determine cooling load for each hour for each cooling load component.

After calculating cooling loads for each component for each hour, sum those to determine the total cooling load for each hour and select the hour with the peak load for design of the air-conditioning system. This process should be repeated for multiple design months to determine the month when the peak load occurs, especially with windows on southern exposures (northern exposure in southern latitudes), which can result in higher peak room cooling loads in winter months than in summer.

Conduction Heat Gain

In the RTS method, conduction through exterior walls and roofs is calculated using conduction time series (CTS).

Table 7-26 Recommended Heat Gain from Miscellaneous Office Equipment

(Table 10, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

Equipment	Nameplate Power,[a] W	Peak Heat Gain,[b] W
Vending machine		
Drinks, 280 to 400 items	NA	940
Snacks	NA	54
Food (e.g., for sandwiches)	NA	465
Thermal binding machine, 2 single documents up to 340 pages	350	28.5
Projector, resolution 1024 × 768	340	308
Paper shredder, up to 28 sheets	1415	265
Electric stapler, up to 45 sheets	NA	1.5
Speakers	220	15
Temperature-controlled electronics soldering station	95	16
Cell phone charger	NA	5
Battery charger		
40 V	NA	19
AA	NA	5.5
Microwave oven, 7 to 9 gal	1000 to 1550	713 to 822
Coffee maker		
Single cup	1400	385
Up to 12 cups	950	780
With grinder	1350	376
Coffee grinder, up to 12 cups	NA	73
Tea kettle, up to 6 cups	1200	1200
Dorm fridge, 3.1 ft^3	NA	57
Freezer, 18 ft^3	130	125
Fridge, 18 to 28 ft^3	NA	387 to 430
Ice maker and dispenser, 20 lb bin capacity	NA	658
Top mounted bottled water cooler	NA	114 to 350
Cash register	25	9
Touch screen computer, 15 in. standard LCD and 2.2 GHz processor	NA	58
Self-checkout machine	NA	15

Source: Bach and Sarfraz (2017)

[a] For some equipment, nameplate power consumption is not available, where NA = not available.

[b] For equipment peak heat gain value, highest 15-min interval of recorded data is listed in tables.

Wall and roof conductive heat input at the exterior is defined by the familiar conduction Equation 7-32 as

$$q_{i,\theta-n} = UA(t_{e,\theta-n} - t_{rc}) \tag{7-32}$$

where

$q_{i,\theta-n}$ = conductive heat input for the surface n hours ago
U = overall heat transfer coefficient for the surface
A = surface area, ft^2
$t_{e,\theta-n}$ = sol-air temperature, °F, n hours ago
t_{rc} = presumed constant room air temperature, °F

Conductive heat gain through walls or roofs can be calculated using conductive heat inputs for the current and past 23 h and conduction time series, as illustrated in Equation 7-33:

$$q_\theta = c_0 q_{i,\theta} + c_1 q_{i,\theta-1} + c_2 q_{i,\theta-2} + c_3 q_{i,\theta-3} + \dots + c_{23} q_{i,\theta-23} \tag{7-33}$$

Table 7-27 Recommended Load Factors for Various Types of Offices
(Table 11, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

Type of Use	Load Factor*, W/ft^2	Description
100% laptop, docking station		
light	0.34	167 ft^2/workstation, all laptop docking station use, 1 printer per 10
medium	0.46	125 ft^2/workstation, all laptop docking station use, 1 printer per 10
50% laptop, docking station		
light	0.44	167 ft^2/workstation, 50% laptop docking station/50% desktop, 1 printer per 10
medium	0.59	125 ft^2/workstation, 50% laptop docking station/50% desktop, 1 printer per 10
100% desktop		
light	0.54	167 ft^2/workstation, all desktop use, 1 printer per 10
medium	0.72	125 ft^2/workstation, all desktop use, 1 printer per 10
100% laptop, docking station		
2 screens	0.69	125 ft^2/workstation, all laptop docking station use, 2 screens, 1 printer per 10
100% desktop		
2 screens	0.84	125 ft^2/workstation, all laptop use, 2 screens, 1 printer per 10
3 screens	0.96	125 ft^2/workstation, all desktop use, 3 screens, 1 printer per 10
100% desktop		
heavy, 2 screens	1.02	85 ft^2/workstation, all desktop use, 2 screens, 1 printer per 8
heavy, 3 screens	1.16	85 ft^2/workstation, all desktop use, 3 screens, 1 printer per 8
100% laptop, docking station		
full on, 2 screens	1.14	85 ft^2/workstation, all laptop docking use, 2 screens, 1 printer per 8, no diversity
100% desktop		
full on, 2 screens	1.33	85 ft^2/workstation, all desktop use, 2 screens, 1 printer per 8, no diversity
full on, 3 screens	1.53	85 ft^2/workstation, all desktop use, 3 screens, 1 printer per 8, no diversity

Source: Bach and Sarfraz (2017)
*Medium-office type monochrome printer is used for load factor calculator with 15-min peak power consumption of 142 W.

Table 7-28 Recommended Diversity Factors for Office Equipment
(Table 12, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

Equipment	Diversity Factor, %
Desktop PC	75
Laptop docking station	70
Notebook computer	75[b]
Screen	70
Printer	45

Source: Bach and Sarfraz (2017)
[a] 2013 *ASHRAE Handbook—Fundamentals*
[b] Insufficient data from RP-1742; values based on previous data from 2013 *ASHRAE Handbook—Fundamentals* and judgment of Bach and Sarfraz (2017).

where

q_θ = hourly conductive heat gain, Btu/h, for the surface
$q_{i,\theta}$ = heat input for the current hour
$q_{i,\theta-n}$ = heat input n hours ago
c_0, c_1, etc.=conduction time factors

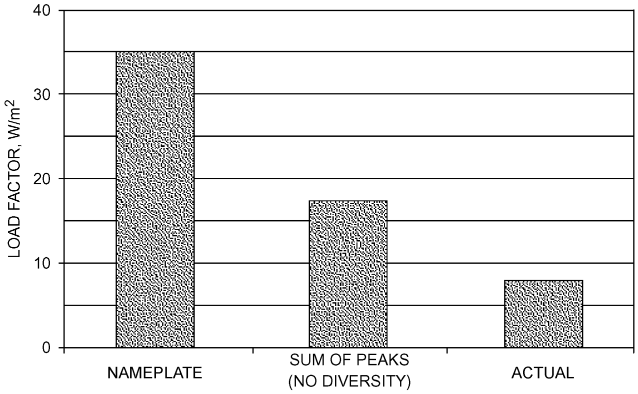


Fig. 7-8 Office Equipment Load Factor Comparison
(Wilkins and McGaffin 1994)
(Figure 4, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

Conduction time factors for representative wall and roof types are included in Tables 7-30 and 7-31. Those values were derived by first calculating conduction transfer functions for each example wall and roof construction. The assumption of steady-periodic heat input conditions for design load calculations allowed the conduction transfer functions to be reformulated into periodic response factors as demonstrated by Spitler and Fisher (1999a). The periodic response factors were further simplified by dividing the 24 periodic response factors by the respective overall wall or roof U-factor to form the conduction time series (CTS). The CTS factors can then be used in Equation 7-33 and provide a means for comparison of time delay characteristics between different wall and roof constructions. Construction material data used in the calculations for walls and roofs included in Tables 7-30 and 7-31 are listed in Table 7-32.

Heat gains calculated for walls or roofs using periodic response factors (and thus CTS) are identical to those calculated using conduction transfer functions for the steady periodic conditions assumed in design cooling load calculations.

7.5 Cooling Load Calculation Using RTS

The instantaneous cooling load is defined as the rate at which heat energy is convected to the zone air at a given point in time. The computation of cooling load is complicated by the radiant exchange between surfaces, furniture, partitions, and other mass in the zone. Most heat gain sources transfer energy by both convection and radiation. Radiative heat transfer introduces to the process a time dependency that is not easily quantified. Radiation is absorbed by the thermal masses in the zone and then later transferred by convection into the space. This process creates a time lag and dampening effect. The convection portion of heat gains, on the other hand, is assumed to immediately become cooling load in the hour in which that heat gain occurs.

Heat balance procedures calculate the radiant exchange between surfaces based on their surface temperatures and emissivities, but they typically rely on estimated "radiative-convective splits" to determine the contribution of internal loads, including people, lighting, appliances, and equipment, to the radiant exchange. The radiant time series procedure further simplifies the heat balance procedure by also relying on an estimated radiative-convective split of wall and roof conductive heat gain instead of simultaneously solving for the instantaneous convective and radiative heat transfer from each surface, as is done in the heat balance procedure.

Table 7-29 Convective and Radiant Percentages of Total Sensible Heat Gain

Heat Gain Source	Radiant Heat, %	Convective Heat, %
Transmitted solar, no inside shade	100	0
Window solar, with inside shade	63	37
Absorbed (by fenestration) solar	63	37
Fluorescent lights, suspended, unvented	67	33
Fluorescent lights, recessed, vented to return air	59	41
Fluorescent lights, recessed, vented to return air and supply air	19	81
Incandescent lights	80	20
People	70	30
Conduction, exterior walls	63	37
Conduction, exterior roofs	84	16
Infiltration and ventilation	0	100
Machinery and appliances	20 to 80	80 to 20

Sources: Pedersen et al. (1998), Hosni et al. (1999).

Thus, the cooling load for each load component (lights, people, walls, roofs, windows, appliances, etc.) for a particular hour is the sum of the convective portion of the heat gain for that hour plus the time-delayed portion of radiant heat gains for that hour and the previous 23 h. Table 7-29 contains recommendations for splitting each of the heat gain components into convective and radiant portions.

The radiant time series method converts the radiant portion of hourly heat gains to hourly cooling loads using radiant time factors, the coefficients of the radiant time series. Radiant time factors are used to calculate the cooling load for the current hour on the basis of current and past heat gains. The radiant time series for a particular zone gives the time-dependent response of the zone to a single pulse of radiant energy. The series shows the portion of the radiant pulse that is convected to the zone air for each hour. Thus, r_0 represents the fraction of the radiant pulse convected to the zone air in the current hour r_1 in the previous hour, and so on. The radiant time series thus generated is used to convert the radiant portion of hourly heat gains to hourly cooling loads.

Two different radiant time series are used: **solar**, for directly transmitted solar heat gain (radiant energy assumed to be distributed to the floor and furnishings only), and **nonsolar** for all other types of heat gains (radiant energy assumed to be uniformly distributed on all internal surfaces). Nonsolar RTS apply to radiant heat gains from people, lights, appliances, walls, roofs, and floors. Also, for diffuse solar heat gain and direct solar

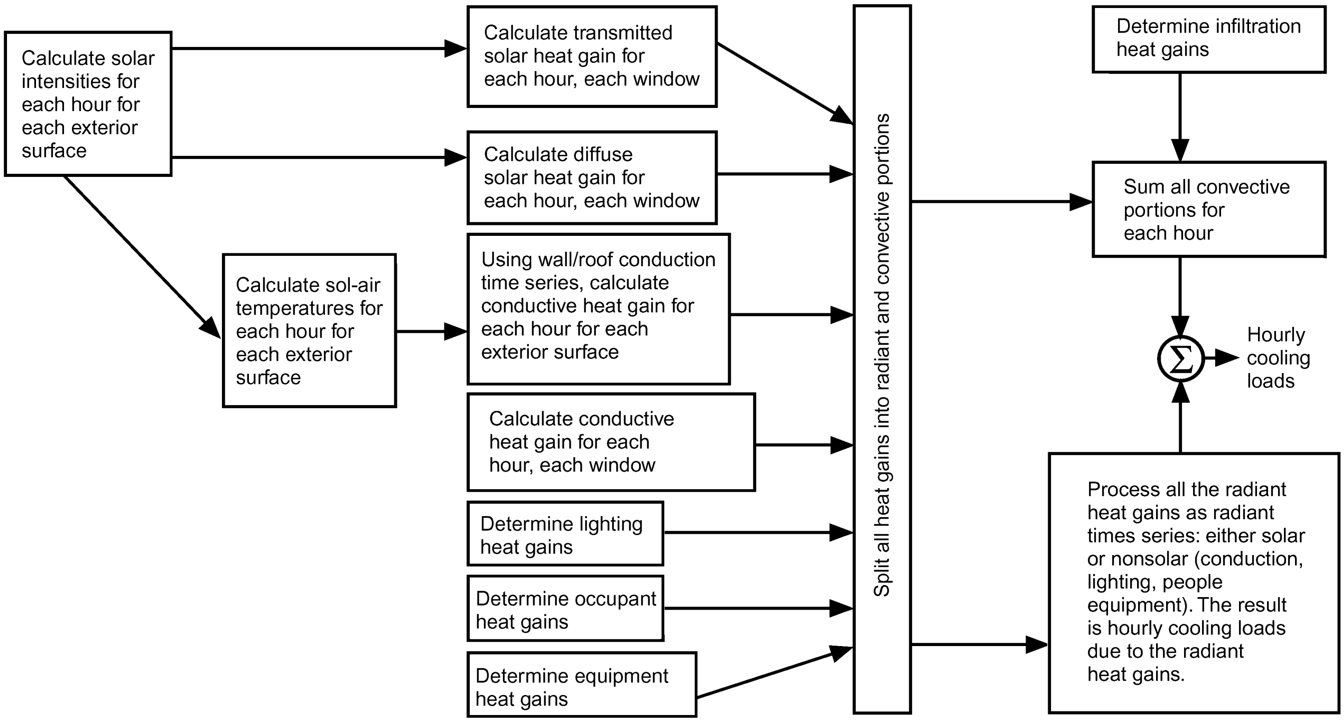


Fig. 7-9 Overview of Radiant Time Series Method
(Figure 8, Chapter 18, 2017 ASHRAE Handbook—Fundamentals)

Table 7-30 Wall Conduction Time Series (CTS)

(Table 16, Chapter 18, *2013 ASHRAE Handbook—Fundamentals*)

	CURTAIN WALLS			STUD WALLS				EIFS			BRICK WALLS									
Wall Number =	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
U-Factor, Btu/h·ft^2·°F	0.075	0.076	0.075	0.074	0.074	0.071	0.073	0.118	0.054	0.092	0.101	0.066	0.050	0.102	0.061	0.111	0.124	0.091	0.102	0.068
Total *R*	13.3	13.2	13.3	13.6	13.6	14.0	13.8	8.5	18.6	10.8	9.9	15.1	20.1	9.8	16.3	9.0	8.1	11.0	9.8	14.6
Mass, lb/ft^2	6.3	4.3	16.4	5.2	17.3	5.2	13.7	7.5	7.8	26.8	42.9	44.0	44.2	59.6	62.3	76.2	80.2	96.2	182.8	136.3
Thermal Capacity, Btu/ft^2·°F	1.5	1.0	3.3	1.2	3.6	1.6	3.0	1.8	1.9	5.9	8.7	8.7	8.7	11.7	12.4	15.7	15.3	19.0	38.4	28.4
Hour	**Conduction Time Factors, %**																			
0	18	25	8	19	6	7	5	11	2	1	0	0	0	1	2	2	1	3	4	3
1	58	57	45	59	42	44	41	50	25	2	5	4	1	1	2	2	1	3	4	3
2	20	15	32	18	33	32	34	26	31	6	14	13	7	2	2	2	3	3	4	3
3	4	3	11	3	13	12	13	9	20	9	17	17	12	5	3	4	6	3	4	4
4	0	0	3	1	4	4	4	3	11	9	15	15	13	8	5	5	7	3	4	4
5	0	0	1	0	1	1	2	1	5	9	12	12	13	9	6	6	8	4	4	4
6	0	0	0	0	1	0	1	0	3	8	9	9	11	9	7	6	8	4	4	5
7	0	0	0	0	0	0	0	0	2	7	7	7	9	9	7	7	8	5	4	5
8	0	0	0	0	0	0	0	0	1	6	5	5	7	8	7	7	8	5	4	5
9	0	0	0	0	0	0	0	0	0	6	4	4	6	7	7	6	7	5	4	5
10	0	0	0	0	0	0	0	0	0	5	3	3	5	7	6	6	6	5	4	5
11	0	0	0	0	0	0	0	0	0	5	2	2	4	6	6	6	6	5	5	5
12	0	0	0	0	0	0	0	0	0	4	2	2	3	5	5	5	5	5	5	5
13	0	0	0	0	0	0	0	0	0	4	1	2	2	4	5	5	4	5	5	5
14	0	0	0	0	0	0	0	0	0	3	1	2	2	4	5	5	4	5	5	5
15	0	0	0	0	0	0	0	0	0	3	1	1	1	3	4	4	3	5	4	4
16	0	0	0	0	0	0	0	0	0	3	1	1	1	3	4	4	3	5	4	4
17	0	0	0	0	0	0	0	0	0	2	1	1	1	2	3	4	3	4	4	4
18	0	0	0	0	0	0	0	0	0	2	0	0	1	2	3	3	2	4	4	4
19	0	0	0	0	0	0	0	0	0	2	0	0	1	2	3	3	2	4	4	4
20	0	0	0	0	0	0	0	0	0	2	0	0	0	1	3	3	2	4	4	4
21	0	0	0	0	0	0	0	0	0	1	0	0	0	1	2	2	1	4	4	4
22	0	0	0	0	0	0	0	0	0	1	0	0	0	1	2	2	1	4	4	3
23	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	3	4	3
Total Percentage	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
Layer ID from outside to inside (see Table 7-30)	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01
	F09	F08	F10	F08	F10	F11	F07	F06	F06	F06	M01	M01	M01	M01	M01	M01	M01	M01	M01	M01
	F04	F04	F04	G03	G03	G02	G03	I01	I01	I01	F04	F04	F04	F04	F04	F04	F04	F04	F04	F04
	I02	I02	I02	I04	I04	I04	I04	G03	G03	G03	I01	G03	I01	I01	M03	I01	I01	I01	I01	M15
	F04	F04	F04	G01	G01	G04	G01	F04	I04	M03	G03	I04	G03	M03	I04	M05	M01	M13	M16	I04
	G01	G01	G01	F02	F02	F02	F02	G01	G01	F04	F04	G01	I04	F02	G01	G01	F02	F04	F04	G01
	F02	F02	F02	—	—	—	—	F02	F02	G01	G01	F02	G01	—	F02	F02	—	G01	G01	F02
	—	—	—	—	—	—	—	—	—	F02	F02	—	F02	—	—	—	—	F02	F02	—

Wall Number Descriptions

1. Spandrel glass, R-10 insulation board, gyp board
2. Metal wall panel, R-10 insulation board, gyp board
3. 1 in. stone, R-10 insulation board, gyp board
4. Metal wall panel, sheathing, R-11 batt insulation, gyp board
5. 1 in. stone, sheathing, R-11 batt insulation, gyp board
6. Wood siding, sheathing, R-11 batt insulation, 1/2 in. wood
7. 1 in. stucco, sheathing, R-11 batt insulation, gyp board
8. EIFS finish, R-5 insulation board, sheathing, gyp board
9. EIFS finish, R-5 insulation board, sheathing, R-11 batt insulation, gyp board
10. EIFS finish, R-5 insulation board, sheathing, 8 in. LW CMU, gyp board
11. Brick, R-5 insulation board, sheathing, gyp board
12. Brick, sheathing, R-11 batt insulation, gyp board
13. Brick, R-5 insulation board, sheathing, R-11 batt insulation, gyp board
14. Brick, R-5 insulation board, 8 in. LW CMU
15. Brick, 8 in. LW CMU, R-11 batt insulation, gyp board
16. Brick, R-5 insulation board, 8 in. HW CMU, gyp board
17. Brick, R-5 insulation board, brick
18. Brick, R-5 insulation board, 8 in. LW concrete, gyp board
19. Brick, R-5 insulation board, 12 in. HW concrete, gyp board
20. Brick, 8 in. HW concrete, R-11 batt insulation, gyp board

heat gain from fenestration with inside shading (blinds, drapes, etc.), the nonsolar RTS should be used. Radiation from those sources is assumed to be more uniformly distributed onto all room surfaces.

Representative solar and nonsolar RTS data for light, medium, and heavyweight constructions are provided in Tables 7-33 and 7-34. Those were calculated using the zone characteristics listed in Table 7-35.

Comparison of RTS with Previous Methods. The user may question what benefits may be expected now that the TFM, TETD/TA, and CLTD/CLF procedures presented in earlier editions have been superseded (*not* invalidated or discredited). The primary benefit will be improved accuracy, with reduced dependency upon purely subjective input (such as determining a proper time-averaging period for TETD/TA or ascertaining appropriate safety factors to add to the

Table 7-30 Wall Conduction Time Series (CTS) (*Continued*)

(Table 16, Chapter 18, *2013 ASHRAE Handbook—Fundamentals*)

	CONCRETE BLOCK WALL						PRECAST AND CAST-IN-PLACE CONCRETE WALLS								
Wall Number =	**21**	**22**	**23**	**24**	**25**	**26**	**27**	**28**	**29**	**30**	**31**	**32**	**33**	**34**	**35**
U-Factor, Btu/h·ft^2·°F	0.067	0.059	0.073	0.186	0.147	0.121	0.118	0.074	0.076	0.115	0.068	0.082	0.076	0.047	0.550
Total *R*	14.8	16.9	13.7	5.4	6.8	8.2	8.4	13.6	13.1	8.7	14.7	12.2	13.1	21.4	1.8
Mass, lb/ft^2	22.3	22.3	46.0	19.3	21.9	34.6	29.5	29.6	53.8	59.8	56.3	100.0	96.3	143.2	140.0
Thermal Capacity, Btu/ft^2·°F	4.8	4.8	10.0	4.1	4.7	7.4	6.1	6.1	10.8	12.1	11.4	21.6	20.8	30.9	30.1
Hour	**Conduction Time Factors, %**														
0	0	1	0	1	0	1	1	0	1	2	1	3	1	2	1
1	4	1	2	11	3	1	10	8	1	2	2	3	2	2	2
2	13	5	8	21	12	2	20	18	3	3	3	4	5	3	4
3	16	9	12	20	16	5	18	18	6	5	6	5	8	3	7
4	14	11	12	15	15	7	14	14	8	6	7	6	9	5	8
5	11	10	11	10	12	9	10	11	9	6	8	6	9	5	8
6	9	9	9	7	10	9	7	8	9	6	8	6	8	6	8
7	7	8	8	5	8	8	5	6	9	6	7	5	7	6	8
8	6	7	7	3	6	8	4	4	8	6	7	5	6	6	7
9	4	6	6	2	4	7	3	3	7	6	6	5	6	6	6
10	3	5	5	2	3	6	2	2	7	5	6	5	5	6	6
11	3	4	4	1	3	6	2	2	6	5	5	5	5	5	5
12	2	4	3	1	2	5	1	2	5	5	5	4	4	5	4
13	2	3	2	1	2	4	1	1	4	5	4	4	4	5	4
14	2	3	2	0	1	4	1	1	4	4	4	4	3	4	4
15	1	3	2	0	1	3	1	1	3	4	3	4	3	4	3
16	1	2	1	0	1	3	0	1	2	4	3	4	3	4	3
17	1	2	1	0	1	2	0	0	2	3	3	4	2	4	3
18	1	2	1	0	0	2	0	0	1	3	2	4	2	4	2
19	0	1	1	0	0	2	0	0	1	3	2	3	2	3	2
20	0	1	1	0	0	2	0	0	1	3	2	3	2	3	2
21	0	1	1	0	0	2	0	0	1	3	2	3	2	3	1
22	0	1	1	0	0	1	0	0	1	3	2	3	1	3	1
23	0	1	0	0	0	1	0	0	1	2	2	2	1	3	1
Total Percentage	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
Layer ID from outside to inside (see Table 7-30)	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01
	M03	M08	F07	M08	M08	M09	M11	M11	M11	F06	M13	F06	M15	M16	M16
	I04	I04	M05	F02	F04	F04	I01	I04	I02	I01	I04	I02	I04	I05	F02
	G01	G01	I04	—	G01	G01	F04	G01	M11	M13	G01	M15	G01	G01	—
	F02	F02	G01	—	F02	F02	G01	F02	F02	G01	F02	G01	F02	F02	—
	—	—	F02	—	—	—	F02	—	—	F02	—	F02	—	—	—

Wall Number Descriptions

21. 8 in. LW CMU, R-11 batt insulation, gyp board
22. 8 in. LW CMU with fill insulation, R-11 batt insulation, gyp board
23. 1 in. stucco, 8 in. HW CMU, R-11 batt insulation, gyp board
24. 8 in. LW CMU with fill insulation
25. 8 in. LW CMU with fill insulation, gyp board
26. 12 in. LW CMU with fill insulation, gyp board
27. 4 in. LW concrete, R-5 board insulation, gyp board
28. 4 in. LW concrete, R-11 batt insulation, gyp board
29. 4 in. LW concrete, R-10 board insulation, 4 in. LW concrete
30. EIFS finish, R-5 insulation board, 8 in. LW concrete, gyp board
31. 8 in. LW concrete, R-11 batt insulation, gyp board
32. EIFS finish, R-10 insulation board, 8 in. HW concrete, gyp board
33. 8 in. HW concrete, R-11 batt insulation, gyp board
34. 12 in. HW concrete, R-19 batt insulation, gyp board
35. 12 in. HW concrete

"rounded off" TFM results). As a generic example, the space sensible cooling load for the traditional little ASHRAE store building (used for example purposes since the 1940s) was calculated by means of the heat balance procedure and independently calculated by application of the radiant time series procedure, with each set of results plotted as one of the load profile curves of Figure 7-9. Also plotted on this chart are the corresponding curves produced by the TFM and TETD/TA methodologies in the 1997 edition of this chapter. Users may draw their own conclusions from this chart.

As part of the presentation of this method, RTS Method Load Calculation spreadsheets are available at **www.ashrae.org/PHVAC8**.

7.6 Heating Load Calculations

Techniques for estimating design heating load for commercial, institutional, and industrial applications are essentially the same as those for estimating design cooling loads, with the following exceptions:

- Credit for solar or internal heat gains is not included.

Table 7-31 Roof Conduction Time Series (CTS), Layers, U-Factors, Mass and Thermal Capacity

(Table 17, Chapter 18, *2009 ASHRAE Handbook—Fundamentals*)

	SLOPED FRAME ROOFS						WOOD DECK		METAL DECK ROOFS					CONCRETE ROOFS					
Roof Number	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**
U-factor, Btu/h·ft^2·°F	0.044	0.040	0.045	0.041	0.042	0.041	0.69	0.058	0.080	0.065	0.057	0.036	0.052	0.054	0.052	0.051	0.056	0.055	0.042
Total *R*	22.8	25.0	22.2	24.1	23.7	24.6	14.5	17.2	12.6	15.4	17.6	27.6	19.1	18.6	19.2	19.7	18.0	18.2	23.7
Mass, lb/ft^2	5.5	4.3	2.9	7.1	11.4	7.1	10.0	11.5	4.9	6.3	5.1	5.6	11.8	30.6	43.9	57.2	73.9	97.2	74.2
Thermal Capacity, Btu/ft·°F	1.3	0.8	0.6	2.3	3.6	2.3	3.7	3.9	1.4	1.6	1.4	1.6	2.8	6.6	9.3	12.0	16.3	21.4	16.2
Hour	**Conduction Time Factors, %**																		
0	6	10	27	1	1	1	0	1	18	4	8	1	0	1	2	2	2	3	1
1	45	57	62	17	17	12	7	3	61	41	53	23	10	2	2	2	2	3	2
2	33	27	10	31	34	25	18	8	18	35	30	38	22	8	3	3	5	3	6
3	11	5	1	24	25	22	18	10	3	14	7	22	20	11	6	4	6	5	8
4	3	1	0	14	13	15	15	10	0	4	2	10	14	11	7	5	7	6	8
5	1	0	0	7	6	10	11	9	0	1	0	4	10	10	8	6	7	6	8
6	1	0	0	4	3	6	8	8	0	1	0	2	7	9	8	6	6	6	7
7	0	0	0	2	1	4	6	7	0	0	0	0	5	7	7	6	6	6	7
8	0	0	0	0	0	2	5	6	0	0	0	0	4	6	7	6	6	6	6
9	0	0	0	0	0	1	3	5	0	0	0	0	3	5	6	6	5	5	5
10	0	0	0	0	0	1	3	5	0	0	0	0	2	5	5	6	5	5	5
11	0	0	0	0	0	1	2	4	0	0	0	0	1	4	5	5	5	5	5
12	0	0	0	0	0	0	1	4	0	0	0	0	1	3	5	5	4	5	4
13	0	0	0	0	0	0	1	3	0	0	0	0	1	3	4	5	4	4	4
14	0	0	0	0	0	0	1	3	0	0	0	0	0	3	4	4	4	4	3
15	0	0	0	0	0	0	1	3	0	0	0	0	0	2	3	4	4	4	3
16	0	0	0	0	0	0	0	2	0	0	0	0	0	2	3	4	3	4	3
17	0	0	0	0	0	0	0	2	0	0	0	0	0	2	3	4	3	4	3
18	0	0	0	0	0	0	0	2	0	0	0	0	0	1	3	3	3	3	2
19	0	0	0	0	0	0	0	2	0	0	0	0	0	1	2	3	3	3	2
20	0	0	0	0	0	0	0	1	0	0	0	0	0	1	2	3	3	3	2
21	0	0	0	0	0	0	0	1	0	0	0	0	0	1	2	3	3	3	2
22	0	0	0	0	0	0	0	1	0	0	0	0	0	1	2	3	2	2	2
23	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	2	2	2	2
	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
Layer ID from outside to inside (see Table 7-30)	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01	F01
	F08	F08	F08	F12	F14	F15	F13	F13	F13	F13	F13	F13	M17	F13	F13	F13	F13	F13	F13
	G03	G03	G03	G05	G05	G05	G03	G03	G03	G03	G03	G03	F13	G03	G03	G03	G03	G03	M14
	F05	F05	F05	F05	F05	F05	I02	I02	I02	I02	I03	I02	G03	I03	I03	I03	I03	I03	F05
	I05	I05	I05	I05	I05	I05	G06	G06	F08	F08	F08	I03	I03	M11	M12	M13	M14	M15	I05
	G01	F05	F03	F05	F05	F05	F03	F05	F03	F05	F03	F08	F08	F03	F03	F03	F03	F03	F16
	F03	F16	—	G01	G01	G01	—	F16	—	F16	—	—	F03	—	—	—	—	—	F03
	—	F03	—	F03	F03	F03	—	F03	—	F03	—	—	—	—	—	—	—	—	—

Roof Number Descriptions

1. Metal roof, R-19 batt insulation, gyp board
2. Metal roof, R-19 batt insulation, suspended acoustical ceiling
3. Metal roof, R-19 batt insulation
4. Asphalt shingles, wood sheathing, R-19 batt insulation, gyp board
5. Slate or tile, wood sheathing, R-19 batt insulation, gyp board
6. Wood shingles, wood sheathing, R-19 batt insulation, gyp board
7. Membrane, sheathing, R-10 insulation board, wood deck
8. Membrane, sheathing, R-10 insulation board, wood deck, suspended acoustical ceiling
9. Membrane, sheathing, R-10 insulation board, metal deck
10. Membrane, sheathing, R-10 insulation board, metal deck, suspended acoustical ceiling
11. Membrane, sheathing, R-15 insulation board, metal deck
12. Membrane, sheathing, R-10 plus R-15 insulation boards, metal deck
13. 2-in. concrete roof ballast, membrane, sheathing, R-15 insulation board, metal deck
14. Membrane, sheathing, R-15 insulation board, 4-in. LW concrete
15. Membrane, sheathing, R-15 insulation board, 6-in. LW concrete
16. Membrane, sheathing, R-15 insulation board, 8-in. LW concrete
17. Membrane, sheathing, R-15 insulation board, 6-in. HW concrete
18. Membrane, sheathing, R-15 insulation board, 8-in. HW concrete
19. Membrane, 6-in HW concrete, R-19 batt insulation, suspended acoustical ceiling

- Thermal storage effect of building structure is ignored.

This simplified approach is justified because it evaluates worst-case conditions that can reasonably occur during a heating season. Thus, the worst-case load is based on the following:

- Design interior and exterior conditions
- Infiltration and/or ventilation
- No solar effect
- No heat gains from lights, people, and appliances

Typical new commercial and retail spaces have nighttime unoccupied periods at a setback temperature where no ventilation is required, building lights and equipment are off, and heat loss is primarily through conduction. Before being occupied, buildings are warmed to the occupied temperature. During occupied time, building lights, equipment, and people heat gains can offset conductive heat loss, although some perimeter heat may be required, leaving ventilation load as the primary heating load.

Table 7-32 Thermal Properties and Code Numbers of Layers Used in Wall and Roof Descriptions for Tables 7-28 and 7-29

(Table 18, Chapter 18, *2013 ASHRAE Handbook—Fundamentals*)

Layer ID	Description	Thickness, in.	Conductivity, Btu·in/h·ft^2·°F	Density, lb/ft^3	Specific Heat, Btu/lb·°F	Resistance, ft^2·°F·h/Btu	*R*	Mass, lb/ft^2	Thermal Capacity, Btu/ft^2·°F	Notes
F01	Outside surface resistance	—	—	—	—	0.25	0.25	—	—	1
F02	Inside vertical surface resistance	—	—	—	—	0.68	0.68	—	—	2
F03	Inside horizontal surface resistance	—	—	—	—	0.92	0.92	—	—	3
F04	Wall air space resistance	—	—	—	—	0.87	0.87	—	—	4
F05	Ceiling air space resistance	—	—	—	—	1.00	1.00	—	—	5
F06	EIFS finish	0.375	5.00	116.0	0.20	—	0.08	3.63	0.73	6
F07	1 in. stucco	1.000	5.00	116.0	0.20	—	0.20	9.67	1.93	6
F08	Metal surface	0.030	314.00	489.0	0.12	—	0.00	1.22	0.15	7
F09	Opaque spandrel glass	0.250	6.90	158.0	0.21	—	0.04	3.29	0.69	8
F10	1 in. stone	1.000	22.00	160.0	0.19	—	0.05	13.33	2.53	9
F11	Wood siding	0.500	0.62	37.0	0.28	—	0.81	1.54	0.43	10
F12	Asphalt shingles	0.125	0.28	70.0	0.30	—	0.44	0.73	0.22	
F13	Built-up roofing	0.375	1.13	70.0	0.35	—	0.33	2.19	0.77	
F14	Slate or tile	0.500	11.00	120.0	0.30	—	0.05	5.00	1.50	
F15	Wood shingles	0.250	0.27	37.0	0.31	—	0.94	0.77	0.24	
F16	Acoustic tile	0.750	0.42	23.0	0.14	—	1.79	1.44	0.20	11
F17	Carpet	0.500	0.41	18.0	0.33	—	1.23	0.75	0.25	12
F18	Terrazzo	1.000	12.50	160.0	0.19	—	0.08	13.33	2.53	13
G01	5/8 in. gyp board	0.625	1.11	50.0	0.26	—	0.56	2.60	0.68	
G02	5/8 in. plywood	0.625	0.80	34.0	0.29	—	0.78	1.77	0.51	
G03	1/2 in. fiberboard sheathing	0.500	0.47	25.0	0.31	—	1.06	1.04	0.32	14
G04	1/2 in. wood	0.500	1.06	38.0	0.39	—	0.47	1.58	0.62	15
G05	1 in. wood	1.000	1.06	38.0	0.39	—	0.94	3.17	1.24	15
G06	2 in. wood	2.000	1.06	38.0	0.39	—	1.89	6.33	2.47	15
G07	4 in. wood	4.000	1.06	38.0	0.39	—	3.77	12.67	4.94	15
I01	R-5, 1 in. insulation board	1.000	0.20	2.7	0.29	—	5.00	0.23	0.07	16
I02	R-10, 2 in. insulation board	2.000	0.20	2.7	0.29	—	10.00	0.45	0.13	16
I03	R-15, 3 in. insulation board	3.000	0.20	2.7	0.29	—	15.00	0.68	0.20	16
I04	R-11, 3-1/2 in. batt insulation	3.520	0.32	1.2	0.23	—	11.00	0.35	0.08	17
I05	R-19, 6-1/4 in. batt insulation	6.080	0.32	1.2	0.23	—	19.00	0.61	0.14	17
I06	R-30, 9-1/2 in. batt insulation	9.600	0.32	1.2	0.23	—	30.00	0.96	0.22	17
M01	4 in. brick	4.000	6.20	120.0	0.19	—	0.65	40.00	7.60	18
M02	6 in. LW concrete3 block	6.000	3.39	32.0	0.21	—	1.77	16.00	3.36	19
M03	8 in. LW concrete block	8.000	3.44	29.0	0.21	—	2.33	19.33	4.06	20
M04	12 in. LW concrete block	12.000	4.92	32.0	0.21	—	2.44	32.00	6.72	21
M05	8 in. concrete block	8.000	7.72	50.0	0.22	—	1.04	33.33	7.33	22
M06	12 in. concrete block	12.000	9.72	50.0	0.22	—	1.23	50.00	11.00	23
M07	6 in. LW concrete block (filled)	6.000	1.98	32.0	0.21	—	3.03	16.00	3.36	24
M08	8 in. LW concrete block (filled)	8.000	1.80	29.0	0.21	—	4.44	19.33	4.06	25
M09	12 in. LW concrete block (filled)	12.000	2.04	32.0	0.21	—	5.88	32.00	6.72	26
M10	8 in. concrete block (filled)	8.000	5.00	50.0	0.22	—	1.60	33.33	7.33	27
M11	4 in. lightweight concrete	4.000	3.70	80.0	0.20	—	1.08	26.67	5.33	
M12	6 in. lightweight concrete	6.000	3.70	80.0	0.20	—	1.62	40.00	8.00	
M13	8 in. lightweight concrete	8.000	3.70	80.0	0.20	—	2.16	53.33	10.67	
M14	6 in. heavyweight concrete	6.000	13.50	140.0	0.22	—	0.44	70.00	15.05	
M15	8 in. heavyweight concrete	8.000	13.50	140.0	0.22	—	0.59	93.33	20.07	
M16	12 in. heavyweight concrete	12.000	13.50	140.0	0.22	—	0.89	140.0	30.10	
M17	2 in. LW concrete roof ballast	2.000	1.30	40	0.20	—	1.54	6.7	1.33	28

Notes: The following notes give sources for the data in this table.

1. Chapter 26, Table 1 for 7.5 mph wind
2. Chapter 26, Table 1 for still air, horizontal heat flow
3. Chapter 26, Table 1 for still air, downward heat flow
4. Chapter 26, Table 3 for 1.5 in. space, 90°F, horizontal heat flow, 0.82 emittance
5. Chapter 26, Table 3 for 3.5 in. space, 90°F, downward heat flow, 0.82 emittance
6. EIFS finish layers approximated by Chapter 26, Table 4 for 3/8 in. cement plaster, sand aggregate
7. Chapter 33, Table 3 for steel (mild)
8. Chapter 26, Table 4 for architectural glass
9. Chapter 26, Table 4 for marble and granite
10. Chapter 26, Table 4, density assumed same as Southern pine
11. Chapter 26, Table 4 for mineral fiberboard, wet molded, acoustical tile
12. Chapter 26, Table 4 for carpet and rubber pad, density assumed same as fiberboard
13. Chapter 26, Table 4, density assumed same as stone
14. Chapter 26, Table 4 for nail-base sheathing
15. Chapter 26, Table 4 for Southern pine
16. Chapter 26, Table 4 for expanded polystyrene
17. Chapter 26, Table 4 for glass fiber batt, specific heat per glass fiber board
18. Chapter 26, Table 4 for clay fired brick
19. Chapter 26, Table 4, 16 lb block, 8 in. × 16 in. face
20. Chapter 26, Table 4, 19 lb block, 8 in. × 16 in. face
21. Chapter 26, Table 4, 32 lb block, 8 in. × 16 in. face
22. Chapter 26, Table 4, 33 lb normal weight block, 8 in. × 16 in. face
23. Chapter 26, Table 4, 50 lb normal weight block, 8 in. × 16 in. face
24. Chapter 26, Table 4, 16 lb block, vermiculite fill
25. Chapter 26, Table 4, 19 lb block, 8 in. × 16 in. face, vermiculite fill
26. Chapter 26, Table 4, 32 lb block, 8 in. × 16 in. face, vermiculite fill
27. Chapter 26, Table 4, 33 lb normal weight block, 8 in. × 16 in. face, vermiculite fill
28. Chapter 26, Table 4 for 40 lb/ft^3 LW concrete

Table 7-33 Representative Nonsolar RTS Values for Light to Heavy Construction

																			Interior Zones					
	Light						Medium						Heavy						Light		Medium		Heavy	
% Glass	With Carpet			No Carpet			With Carpet			No Carpet			With Carpet			No Carpet			With Carpet	No Carpet	With Carpet	No Carpet	With Carpet	No Carpet
	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%						
Hour	Radiant Time Factor, %																							
0	47	50	53	41	43	46	46	49	52	31	33	35	34	38	42	22	25	28	46	40	46	31	33	21
1	19	18	17	20	19	19	18	17	16	17	16	15	9	9	9	10	9	9	19	20	18	17	9	9
2	11	10	9	12	11	11	10	9	8	11	10	10	6	6	5	6	6	6	11	12	10	11	6	6
3	6	6	5	8	7	7	6	5	5	8	7	7	4	4	4	5	5	5	6	8	6	8	5	5
4	4	4	3	5	5	5	4	3	3	6	5	5	4	4	4	5	5	4	4	5	3	6	4	5
5	3	3	2	4	3	3	2	2	2	4	4	4	4	3	3	4	4	4	3	4	2	4	4	4
6	2	2	2	3	3	2	2	2	2	4	3	3	3	3	3	4	4	4	2	3	2	4	3	4
7	2	1	1	2	2	2	1	1	1	3	3	3	3	3	3	4	4	4	2	2	1	3	3	4
8	1	1	1	1	1	1	1	1	1	3	2	2	3	3	3	4	3	3	1	1	1	3	3	4
9	1	1	1	1	1	1	1	1	1	2	2	2	3	3	2	3	3	3	1	1	1	2	3	3
10	1	1	1	1	1	1	1	1	1	2	2	2	3	2	2	3	3	3	1	1	1	2	3	3
11	1	1	1	1	1	1	1	1	1	2	2	2	2	2	2	3	3	3	1	1	1	2	2	3
12	1	1	1	1	1	1	1	1	1	1	1	1	2	2	2	3	3	3	1	1	1	1	2	3
13	1	1	1	0	1	0	1	1	1	1	1	1	2	2	2	3	3	2	1	1	1	1	2	3
14	0	0	1	0	1	0	1	1	1	1	1	1	2	2	2	3	2	2	1	0	1	1	2	3
15	0	0	1	0	0	0	1	1	1	1	1	1	2	2	2	2	2	2	0	0	1	1	2	3
16	0	0	0	0	0	0	1	1	1	1	1	1	2	2	2	2	2	2	0	0	1	1	2	3
17	0	0	0	0	0	0	1	1	1	1	1	1	2	2	2	2	2	2	0	0	1	1	2	2
18	0	0	0	0	0	0	1	1	1	1	1	1	2	2	1	2	2	2	0	0	1	1	2	2
19	0	0	0	0	0	0	0	1	0	0	1	1	2	2	1	2	2	2	0	0	1	0	2	2
20	0	0	0	0	0	0	0	0	0	0	1	1	2	1	1	2	2	2	0	0	0	0	2	2
21	0	0	0	0	0	0	0	0	0	0	1	1	2	1	1	2	2	2	0	0	0	0	2	2
22	0	0	0	0	0	0	0	0	0	0	1	0	1	1	1	2	2	2	0	0	0	0	1	2
23	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	2	2	1	0	0	0	0	1	2
	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100

Table 7-34 Representative Solar RTS Values for Light to Heavy Construction

	Light						Medium						Heavy					
% Glass	With Carpet			No Carpet			With Carpet			No Carpet			With Carpet			No Carpet		
	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%	10%	50%	90%
Hour	Radiant Time Factor, %																	
0	53	55	56	44	45	46	52	54	55	28	29	29	47	49	51	26	27	28
1	17	17	17	19	20	20	16	16	15	15	15	15	11	12	12	12	13	13
2	9	9	9	11	11	11	8	8	8	10	10	10	6	6	6	7	7	7
3	5	5	5	7	7	7	5	4	4	7	7	7	4	4	3	5	5	5
4	3	3	3	5	5	5	3	3	3	6	6	6	3	3	3	4	4	4
5	2	2	2	3	3	3	2	2	2	5	5	5	2	2	2	4	4	4
6	2	2	2	3	2	2	2	1	1	4	4	4	2	2	2	3	3	3
7	1	1	1	2	2	2	1	1	1	4	3	3	2	2	2	3	3	3
8	1	1	1	1	1	1	1	1	1	3	3	3	2	2	2	3	3	3
9	1	1	1	1	1	1	1	1	1	3	3	3	2	2	2	3	3	3
10	1	1	1	1	1	1	1	1	1	2	2	2	2	2	2	3	3	3
11	1	1	1	1	1	1	1	1	1	2	2	2	2	2	1	3	3	2
12	1	1	1	1	1	0	1	1	1	2	2	2	2	1	1	2	2	2
13	1	1	0	1	0	0	1	1	1	2	2	2	2	1	1	2	2	2
14	1	0	0	0	0	0	1	1	1	1	1	1	2	1	1	2	2	2
15	1	0	0	0	0	0	1	1	1	1	1	1	1	1	1	2	2	2
16	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	2	2	2
17	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	2	2	2
18	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	2	2	2
19	0	0	0	0	0	0	0	0	0	1	1	1	1	1	1	2	2	2
20	0	0	0	0	0	0	0	0	0	1	1	1	1	1	1	2	2	2
21	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	2	2	2
22	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	2	1	1
23	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	2	1	1
	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100

Table 7-35 RTS Representative Zone Construction for Tables 7-31 and 7-32

Construction Class	Exterior Wall	Roof/Ceiling	Partitions	Floor	Furnishings
Light	steel siding, 2 in. insulation, air space, 3/4 in. gyp	4 in. LW concrete, ceiling air space, acoustic tile	3/4 in. gyp, air space, 3/4 in. gyp	acoustic tile, ceiling air space, 4 in. LW concrete	1 in. wood at 50% of floor area
Medium	4 in. face brick, 2 in. insulation, air space, 3/4 in. gyp	4 in. HW concrete, ceiling air space, acoustic tile	3/4 in. gyp, air space, 3/4 in. gyp	acoustic tile, ceiling air space, 4 in. HW concrete	1 in. wood at 50% of floor area
Heavy	4 in. face brick, 8 in. HW concrete air space, 2 in. insulation, 3/4 in. gyp	8 in. HW concrete, ceiling air space, acoustic tile	3/4 in. gyp, 8 in. HW concrete block, 3/4 in. gyp	acoustic tile, ceiling air space, 8 in. HW concrete	1 in. wood at 50% of floor area

A combined warm-up/safety allowance of 20% to 25% is fairly common but varies depending on the particular climate, building use, and type of construction. Engineering judgment must be applied for the particular project.

7.7 Design Loads Calculation Example

To illustrate the cooling and heating design load calculation procedures presented in this chapter, as taken from Chapter 18, 2013 *ASHRAE Handbook—Fundamentals*, an example problem was developed by the ASHRAE Technical Committee responsible for Chapter 18.

This example problem has been developed based on the ASHRAE headquarters building located in Atlanta, Georgia. This example is a two-story office building of approximately 35,000 ft^2, including a variety of common office functions and occupancies. In addition to demonstrating calculation procedures, a hypothetical design/construction process is discussed to illustrate (1) application of load calculations and (2) the need to develop reasonable assumptions when specific data is not yet available, as often occurs in everyday design processes.

Table 7-36 provides a summary of RTS load calculation procedures.

7.7.1 Single-Room Example

Calculate the peak heating and cooling loads for the office room shown in Figure 7-11, for Atlanta, Georgia. The room is on the second floor of a two-story building and has two vertical exterior exposures, with a flat roof above.

Room Characteristics.

Area: 130 ft^2.

Floor: Carpeted 5 in. concrete slab on metal deck above a conditioned space.

Roof: Flat metal deck topped with rigid closed-cell polyisocyanurate foam core insulation (R = 30), and light-colored membrane roofing. Space above 9 ft suspended acoustical tile ceiling is used as a return air plenum. Assume 30% of the cooling load from the roof is directly absorbed in the return airstream without becoming room load. Use roof U = 0.032 Btu/h·ft^2·°F.

Spandrel wall: Spandrel bronze-tinted glass, opaque, backed with air space, rigid mineral fiber insulation (R = 5.0), mineral fiber batt insulation (R = 13), and 5/8 in. gypsum wall board. Use spandrel wall U = 0.077 Btu/h·ft^2·°F.

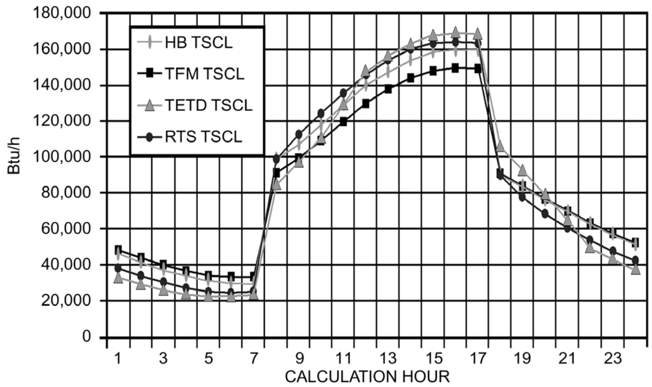


Fig. 7-10 Load Profile Comparison

(Figure 13, Chapter; 29, 2001 ASHRAE Handbook—Fundamentals)

Brick wall: Light-brown-colored face brick (4 in.), lightweight concrete block (6 in.), rigid continuous insulation (R = 5), mineral fiber batt insulation (R = 13), and gypsum wall board (5/8 in.). Use brick wall U = 0.08 Btu/h·ft^2·°F.

Windows: Double glazed, 1/4 in. bronze-tinted outdoor pane, 1/2 in. air space and 1/4 in. clear indoor pane with light-colored interior miniblinds. Window normal solar heat gain coefficient (SHGC) = 0.49. Windows are nonoperable and mounted in aluminum frames with thermal breaks having overall combined U = 0.56 Btu/h·ft^2·°F (based on Type 5d from Tables 4 and 10 of Chapter 15 of 2013 *ASHRAE Handbook—Fundamentals*). Indoor attenuation coefficients (IACs) for indoor miniblinds are based on light venetian blinds (assumed louver reflectance = 0.8 and louvers positioned at 45° angle) with heat-absorbing double glazing (Type 5d from Table 13B of Chapter 15 of 2013 *ASHRAE Handbook—Fundamentals*), IAC(0) = 0.74, IAC(60) = 0.65, IAD(diff) = 0.79, and radiant fraction = 0.54. Each window is 6.25 ft1.91 m wide by 6.4 ft tall for an area per window = 40 ft^2.

South exposure:

Orientation = 30° east of true south
Window area = 40 ft^2
Spandrel wall area = 60 ft^2
Brick wall area = 60 ft^2

West exposure:

Orientation = 60° west of south
Window area = 40 ft^2
Spandrel wall area = 60 ft^2
Brick wall area = 40 ft^2

Table 7-36 Summary of RTS Load Calculation Procedures

(Table and equation references are for Chapter 18 of the 2017 ASHRAE Handbook—Fundamentals unless otherwise noted.)

Equation — **Equation No. in Chapter**

External Heat Gain

Sol-Air Temperature

$$t_e = t_o + \frac{\alpha E_t}{h_o} - \frac{\varepsilon \Delta R}{h_o} \tag{29}$$

where

t_e = sol-air temperature, °F
t_o = outdoor air temperature, °F
a = absorptance of surface for solar radiation
E_t = total solar radiation incident on surface, Btu/h·ft^2
h_o = coefficient of heat transfer by long-wave radiation and convection at outer surface, Btu/h·ft^2·°F
ε = hemispherical emittance of surface
ΔR = difference between long-wave radiation incident on surface from sky and surroundings and radiation emitted by blackbody at outdoor air temperature, Btu/h·ft^2; 20 for horizontal surfaces; 0 for vertical surfaces

Wall and Roof Transmission

$$q_\theta = \mathrm{c}_0 q_{i,\theta} + c_1 q_{i,\theta\text{-}1} + c_2 q_{i,\theta\text{-}2} + \cdots + c_{23} q_{i,\theta\text{-}23} \tag{31}$$

$$q_{i,\theta\text{-}n} = UA(t_{e,\theta\text{-}n} - t_{rc}) \tag{30}$$

where

q_θ = hourly conductive heat gain for surface, Btu/h
$q_{i,\theta}$ = heat input for current hour
$q_{i,\theta\text{-}n}$ = conductive heat input for surface n hours ago, Btu/h
c_0, c_1, etc. = conduction time factors
U = overall heat transfer coefficient for surface, Btu/h·ft^2·°F
A = surface area, ft^2

Fenestration Transmission

$$q_c = UA(T_{out} - T_{in}) \tag{14}$$

where

q = fenestration transmission heat gain, Btu/h
U = overall U-factor, including frame and mounting orientation from Table 4 of Chapter 15, Btu/h·ft^2·°F
A = window area, ft^2
T_{in} = indoor temperature, °F
T_{out} = outdoor temperature, °F

Fenestration Solar

T_{out} = outdoor temperature, °F

$$q_b = AE_{t,b}\,\mathrm{SHGC}(\theta)\mathrm{IAC}(\theta,\Omega) \tag{12}$$

$$q_d = A(E_{t,d} + E_{t,r})\langle \mathrm{SHGC}\rangle_D\,\mathrm{IAC}_D \tag{13}$$

where

q_b = beam solar heat gain, Btu/h
q_d = diffuse solar heat gain, Btu/h
A = window area, ft^2
$E_{t,b}$, $E_{t,d}$, and $E_{t,r}$ = beam, sky diffuse, and ground-reflected diffuse irradiance, calculated using equations in Chapter 14
SHGC(θ) = beam solar heat gain coefficient as a function of incident angle θ; may be interpolated between values in Table 10 of Chapter 15
IAC(θ.Ω) = indoor solar attenuation coefficient for beam solar heat gain coefficient; = 1.0 if no indoor shading device. IAC(θ.Ω) is a function of shade type and, depending on type, may also be a function of beam solar angle of incidence θ and shade geometry
IAC$_D$ = indoor solar attenuation coefficient for diffuse solar heat gain coefficient; = 1.0 if not indoor shading device. IAC$_D$ is a function of shade type and, depending on type, may also be a function of shade geometry

Partitions, Ceilings, Floors Transmission

$$q = UA(t_b - t_i) \tag{32}$$

where

q = heat transfer rate, Btu/h
U = coefficient of overall heat transfer between adjacent and conditioned space, Btu/h·ft^2·°F
A = area of separating section concerned, ft^2
t_b = average air temperature in adjacent space, °F
t_i = air temperature in conditioned space, °F

Internal Heat Gain

Occupants

$$q_s = q_{s,per} N$$

$$q_l = q_{l,per} N$$

where

q_s = occupant sensible heat gain, Btu/h
q_l = occupant latent heat gain, Btu/h
$q_{l,per}$ = latent heat gain per person, Btu/h·person; see Table 1
N = number of occupants

Lighting

$$q_{el} = 3.41 W F_{ul} F_{\mathrm{sa}} \tag{1}$$

where

q_{el} = heat gain, Btu/h
W = total light wattage, W
F_{ul} = lighting use factor
F_{sa} = lighting special allowance factor
3.41 = conversion factor

Electric Motors

$$q_{em} = 2545(P/E_M)F_{UM}F_{LM} \tag{2}$$

where

q_{em} = heat equivalent of equipment operation, Btu/h
P = motor power rating, hp
E_M = motor efficiency, decimal fraction <1.0
F_{UM} = motor use factor, 1.0 or decimal fraction <1.0
F_{LM} = motor load factor, 1.0 or decimal fraction <1.0
2545 = conversion factor, Btu/h·hp

Hooded Cooking Appliances

$$q_s = q_{input} F_U F_R$$

where

q_s = sensible heat gain, Btu/h
q_{input} = nameplate or rated energy input, Btu/h
F_U = usage factor; see Tables 5B, 5C, 5D
F_R = radiation factor; see Tables 5B, 5C, 5D

For other appliances and equipment, find q_s for
Unhooded cooking appliances: Table 5A
Other kitchen equipment: Table 5E
Hospital and laboratory equipment: Tables 6 and 7
Computers, printers, scanners, etc.: Tables 8 and 9
Miscellaneous office equipment: Table 10

Find q_l for
Unhooded cooking appliances: Table 5A
Other kitchen equipment: Table 5E

Ventilation and Infiltration Air Heat Gain

$$q_s = 1.10 Q_s \Delta t \tag{9}$$

$$q_l = 60 \times 0.075 \times 1076 Q_s \Delta W = 4840 Q_s \Delta W \tag{10}$$

where

q_s = sensible heat gain due to infiltration, Btu/h

Table 7-36 Summary of RTS Load Calculation Procedures *(Continued)*

(Table and equation references are for Chapter 18 of the 2013 ASHRAE Handbook—Fundamentals unless otherwise noted.)

Equation	Equation No. in Chapter	Equation	Equation No. in Chapter

q_l = latent heat gain due to infiltration, Btu/h
Q_s = infiltration airflow at standard air conditions, cfm
t_o = outdoor air temperature, °F
t_i = indoor air temperature, °F
W_o = outdoor air humidity ratio, lb/lb
W_i = indoor air humidity ratio, lb/lb
1.10 = air sensible heat factor at standard air conditions, Btu/h·cfm
4840 = air latent heat factor at standard air conditions, Btu/h·cfm

Instantaneous Room Cooling Load

$$Q_s = \Sigma Q_{i,r} + \Sigma Q_{i,c}$$

$$Q_l = \Sigma q_{i,l}$$

where

Q_s = room sensible cooling load, Btu/h
$Q_{i,r}$ = radiant portion of sensible cooling load for current hour, resulting from heat gain element *i*, Btu/h
$Q_{i,c}$ = convective portion of sensible cooling load, resulting from heat gain element *i*, Btu/h
Q_l = room latent cooling load, Btu/h
$q_{i,l}$ = latent heat gain for heat gain element *i*, Btu/h

Radiant Portion of Sensible Cooling Load

$$Q_{i,r} = Q_{r,\theta}$$

$$Q_{r,\theta} = r_0 q_{r,\theta} + r_1 q_{r,\theta-1} + r_2 q_{r,\theta-2} + r_3 q_{r,\theta-3} + \cdots + r_{23} q_{r,\theta-23} \qquad (33)$$

where

$Q_{r,\theta}$ = radiant cooling load Q_r for current hour θ, Btu/h
$q_{r,\theta}$ = radiant heat gain for current hour, Btu/h
$q_{r,\theta-n}$ = radiant heat gain n hours ago, Btu/h
r_0, r_1, etc. = radiant time factors; see Table 19 for radiant time factors for nonsolar heat gains: wall, roof, partition, ceiling, floor, fenestration transmission heat gains, and occupant, lighting, motor, appliance heat gain. Also used for fenestration diffuse solar heat gain; see Table 20 for radiant time factors for fenestration beam solar heat gain.

$$q_{r,\theta} = q_{i,s} F_r$$

where

$q_{i,s}$ = sensible heat gain from heat gain element *i*, Btu/h
F_r = fraction of heat gain that is radiant.

Data Sources:

Wall transmission: see Table 14
Roof transmission: see Table 14
Floor transmission: see Table 14
Fenestration transmission: see Table 14
Fenestration solar heat gain: see Table 14, Chapter 18 and Tables 14A to 14G, Chapter 15
Lighting: see Table 3
Occupants: see Tables 1 and 14
Hooded cooking appliances: see Tables 5B, 5C, and 5D
Unhooded cooking appliances: see Table 5A
Other appliances and equipment: see Tables 5E, 8, 9, 10, and 14
Infiltration: see Table 14
Lighting: see Table 3

Convective Portion of Sensible Cooling Load

$$Q_{i,c} = q_{i,c}$$

where $q_{i,c}$ is convective portion of heat gain from heat gain element *i*, Btu/h.

$$q_{i,c} = q_{i,s}(1 - F_r)$$

where

$q_{i,s}$ = sensible heat gain from heat gain element *i*, Btu/h
F_r = fraction of heat gain that is radiant; see row for radiant portion for sources of radiant fraction data for individual heat gain elements

Occupancy: 1 person from 8:00 AM to 5:00 PM.

Lighting: One 4-lamp pendant fluorescent 8 ft type. The fixture has four 32 W T-8 lamps plus electronic ballasts (special allowance factor 0.85 per manufacturer's data), for a total of 110 W for the room. Operation is from 7:00 AM to 7:00 PM. Assume 0% of the cooling load from lighting is directly absorbed in the return airstream without becoming room load, per Table 3 in Chapter 18 of 2013 *ASHRAE Handbook—Fundamentals*.

Equipment: One computer and a personal printer are used, for which an allowance of 1 W/ft^2 is to be accommodated by the cooling system, for a total of 130 W for the room. Operation is from 8:00 AM to 5:00 PM.

Infiltration: For purposes of this example, assume the building is maintained under positive pressure during peak cooling conditions and therefore has no infiltration. Assume that infiltration during peak heating conditions is equivalent to one air change per hour.

Weather data: Per Chapter 14 of 2013 *ASHRAE Handbook—Fundamentals*, for Atlanta, Georgia, latitude = 33.64, longitude = 84.43, elevation = 1027 ft above sea level, 99.6% heating design dry-bulb temperature = 21.5°F. For cooling load calculations, use 5% dry-bulb/coincident wet-bulb monthly design day profile calculated per Chapter 14. See Table 7-37 or temperature profiles used in these examples.

Indoor design conditions: 72°F for heating; 75°F with 50% rh for cooling.

Cooling Loads Using RTS Method. Traditionally, simplified cooling load calculation methods have estimated the total cooling load at a particular design condition by independently calculating and then summing the load from each component (walls, windows, people, lights, etc). Although the actual heat transfer processes for each component do affect each other, this simplification is appropriate for design load calculations and useful to the designer in understanding the relative contribution of each component to the total cooling load.

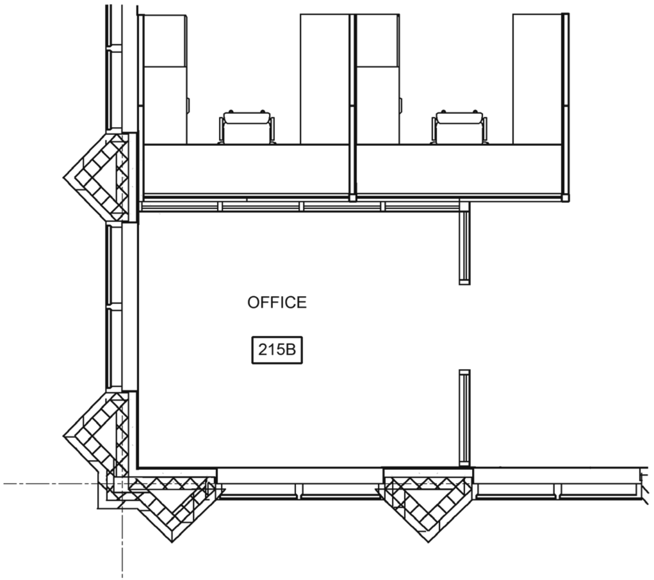


Fig. 7-11 Single-Room Example Office
(Figure 16, Chapter 18, 2013 ASHRAE Handbook—Fundamentals)

Cooling loads are calculated with the RTS method on a component basis similar to previous methods. The following example parts illustrate cooling load calculations for individual components of this single room for a particular hour and month. Equations used are summarized in Table 7-36.

Part 1. Internal cooling load using radiant time series. Calculate the cooling load from lighting at 3:00 PM for the previously described office.

Solution:

First calculate the 24 h heat gain profile for lighting, then split those heat gains into radiant and convective portions, apply the appropriate RTS to the radiant portion, and sum the convective and radiant cooling load components to determine total cooling load at the designated time. Using Equation 1, the lighting heat gain profile, based on the occupancy schedule indicated is

$q_1 = (110\text{ W})3.41(0\%) = 0$
$q_2 = (110\text{ W})3.41(0\%) = 0$
$q_3 = (110\text{ W})3.41(0\%) = 0$
$q_4 = (110\text{ W})3.41(0\%) = 0$
$q_5 = (110\text{ W})3.41(0\%) = 0$
$q_6 = (110\text{ W})3.41(0\%) = 0$
$q_7 = (110\text{ W})3.41(100\%) = 375$
$q_8 = (110\text{ W})3.41(100\%) = 375$
$q_9 = (110\text{ W})3.41(100\%) = 375$
$q_{10} = (110\text{ W})3.41(100\%) = 375$
$q_{11} = (110\text{ W})3.41(100\%) = 375$
$q_{12} = (110\text{ W})3.41(100\%) = 375$
$q_{13} = (110\text{ W})3.41(100\%) = 375$
$q_{14} = (110\text{ W})3.41(100\%) = 375$
$q_{15} = (110\text{ W})3.41(100\%) = 375$
$q_{16} = (110\text{ W})3.41(100\%) = 375$
$q_{17} = (110\text{ W})3.41(100\%) = 375$
$q_{18} = (110\text{ W})3.41(100\%) = 375$
$q_{19} = (110\text{ W})3.41(0\%) = 0$
$q_{20} = (110\text{ W})3.41(0\%) = 0$
$q_{21} = (110\text{ W})3.41(0\%) = 0$
$q_{22} = (110\text{ W})3.41(0\%) = 0$
$q_{23} = (110\text{ W})3.41(0\%) = 0$
$q_{24} = (110\text{ W})3.41(0\%) = 0$

The convective portion is simply the lighting heat gain for the hour being calculated times the convective fraction for non-in-ceiling fluorescent luminaire (pendant), from Table 3 of Chapter 18, 2013 *ASHRAE Handbook—Fundamentals*:

$$Q_{c,15} = (375)(43\%) = 161.3\text{ Btu/h}$$

The radiant portion of the cooling load is calculated using lighting heat gains for the current hour and past 23 h, the radiant fraction from Table 3 of Chapter 18, 2013 *ASHRAE Handbook—Fundamentals* (57%), and radiant time series from Table 19, in accordance with Equation 34. From Table 19 of Chapter 18, 2013 *ASHRAE Handbook—Fundamentals*, select the RTS for medium-weight construction, assuming 50% glass and carpeted floors as representative of the described construction. Thus, the radiant cooling load for lighting is

$$\begin{aligned}
Q_{r,15} &= r_0(0.48)q_{15} + r_1(0.48)q_{14} + r_2(0.48)q_{13} + r_3(0.48)q_{12} \\
&\quad + \cdots + r_{23}(0.48)q_{16} \\
&= (0.49)(0.57)(375) + (0.17)(0.57)(375) \\
&\quad + (0.09)(0.57)(375) + (0.05)(0.57)(375) + (0.03)(0.57)(375) \\
&\quad + (0.02)(0.57)(375) + (0.02)(0.57)(375) + (0.01)(0.57)(375) \\
&\quad + (0.01)(0.57)(375) + (0.01)(0.57)(0) + (0.01)(0.57)(0) \\
&\quad + (0.01)(0.57)(0) + (0.01)(0.57)(0) + (0.01)(0.57)(0) \\
&\quad + (0.01)(0.5748)(0) + (0.01)(0.57)(0) + (0.01)(0.57)(0) \\
&\quad + (0.01)(0.57)(0) \\
&\quad + (0.01)(0.57)(0) + (0.01)(0.57)(0) + (0.00)(0.57)(0) \\
&\quad + (0.00)(0.57)(375) + (0.00)(0.57)(375) \\
&\quad + (0.00)(0.57)(375) = 190.3\text{ Btu/h}
\end{aligned}$$

The total lighting cooling load at the designated hour is thus

$$Q_{\text{light}} = Q_{c,15} + Q_{r,15} = 161.3 + 190.3 = 351.6\text{ Btu/h}$$

See Table 7-38 for the office's lighting usage, heat gain, and cooling load profiles.

Part 2. Wall cooling load using sol-air temperature, conduction time series and radiant time series. Calculate the cooling load contribution from the spandrel wall section facing 60° west of south at 3:00 PM local standard time in July for the previously described office.

Solution:

Determine the wall cooling load by calculating (1) sol-air temperatures at the exterior surface, (2) heat input based on sol-air temperature, (3) delayed heat gain through the mass of the wall to the interior surface using conduction time series, and (4) delayed space cooling load from heat gain using radiant time series.

First, calculate the sol-air temperature at 3:00 PM local standard time (LST) (4:00 PM daylight saving time) on July 21 for a vertical, dark-colored wall surface, facing 60° west of

Table 7-37 Monthly/Hourly Design Temperatures (5% Conditions) for Atlanta, GA, °F

Hour	January		February		March		April		May		June		July		August		September		October		November		December	
	db	wb	db	wb	db	wb	db	wb	db	wb	db	wb	db	wb	db	wb	db	wb	db	wb	db	wb	db	wb
1	44.8	44.0	47.4	45.7	53.1	48.7	59.4	54.3	66.4	61.9	72.2	66.7	73.9	68.9	73.8	69.3	69.5	64.9	60.1	56.8	52.5	50.8	46.4	46.3
2	44.0	43.4	46.6	45.1	52.1	48.2	58.5	53.9	65.6	61.6	71.4	66.4	73.0	68.7	73.1	69.1	68.7	64.6	59.3	56.4	51.7	50.3	45.6	45.6
3	43.4	43.0	46.0	44.8	51.5	47.9	57.8	53.6	65.0	61.4	70.8	66.2	72.4	68.5	72.5	68.9	68.2	64.4	58.7	56.2	51.0	50.0	45.0	45.0
4	42.8	42.6	45.3	44.4	50.8	47.5	57.2	53.3	64.4	61.2	70.2	66.0	71.8	68.3	71.9	68.7	67.6	64.2	58.1	55.9	50.4	49.6	44.4	44.4
5	42.4	42.3	44.9	44.1	50.3	47.3	56.7	53.1	64.0	61.0	69.8	65.9	71.4	68.2	71.5	68.6	67.2	64.1	57.7	55.7	50.0	49.4	44.0	44.0
6	42.8	42.6	45.3	44.4	50.8	47.5	57.2	53.3	64.4	61.2	70.2	66.0	71.8	68.3	71.9	68.7	67.6	64.2	58.1	55.9	50.4	49.6	44.4	44.4
7	44.2	43.6	46.8	45.3	52.4	48.3	58.8	54.0	65.8	61.7	71.6	66.5	73.2	68.8	73.2	69.1	68.9	64.7	59.5	56.5	51.9	50.4	45.8	45.8
8	47.7	46.0	50.4	47.5	56.3	50.3	62.6	55.6	69.3	63.0	75.1	67.6	76.7	69.8	76.5	70.2	72.2	65.9	63.0	58.1	55.4	52.4	49.1	48.2
9	51.6	48.7	54.4	50.0	60.7	52.4	67.0	57.5	73.1	64.5	78.9	68.8	80.6	71.0	80.2	71.3	75.8	67.2	66.8	59.8	59.4	54.6	52.9	50.9
10	55.0	51.1	57.9	52.2	64.6	54.4	70.8	59.1	76.6	65.8	82.3	69.9	84.1	72.0	83.5	72.3	79.0	68.3	70.3	61.3	62.9	56.6	56.3	53.2
11	58.1	53.2	61.1	54.1	68.0	56.1	74.3	60.6	79.6	66.9	85.4	70.9	87.2	73.0	86.4	73.2	81.9	69.3	73.3	62.7	66.0	58.3	59.2	55.3
12	60.1	54.7	63.2	55.4	70.3	57.2	76.5	61.5	81.7	67.7	87.4	71.6	89.2	73.6	88.4	73.8	83.8	70.0	75.4	63.6	68.1	59.5	61.2	56.7
13	61.8	55.8	64.9	56.5	72.2	58.1	78.4	62.3	83.3	68.3	89.0	72.1	90.9	74.1	89.9	74.3	85.3	70.6	77.0	64.3	69.8	60.4	62.8	57.8
14	62.8	56.5	65.9	57.1	73.3	58.7	79.5	62.8	84.3	68.7	90.0	72.4	91.9	74.4	90.9	74.6	86.3	70.9	78.0	64.8	70.8	61.0	63.8	58.5
15	62.8	56.5	65.9	57.1	73.3	58.7	79.5	62.8	84.3	68.7	90.0	72.4	91.9	74.4	90.9	74.6	86.3	70.9	78.0	64.8	70.8	61.0	63.8	58.5
16	61.6	55.6	64.6	56.3	71.9	58.0	78.1	62.2	83.1	68.2	88.8	72.0	90.7	74.0	89.7	74.2	85.2	70.5	76.8	64.3	69.6	60.3	62.6	57.7
17	59.9	54.5	63.0	55.3	70.1	57.1	76.3	61.4	81.5	67.6	87.2	71.5	89.0	73.5	88.2	73.8	83.6	69.9	75.2	63.5	67.9	59.4	61.0	56.6
18	57.9	53.1	60.9	54.0	67.8	56.0	74.0	60.5	79.4	66.9	85.2	70.8	87.0	72.9	86.2	73.2	81.7	69.3	73.1	62.6	65.8	58.2	59.0	55.2
19	54.8	51.0	57.7	52.0	64.3	54.3	70.6	59.0	76.4	65.7	82.1	69.9	83.9	72.0	83.3	72.3	78.9	68.2	70.1	61.3	62.7	56.5	56.1	53.1
20	52.6	49.4	55.4	50.6	61.8	53.0	68.1	58.0	74.2	64.9	79.9	69.2	81.7	71.3	81.2	71.6	76.8	67.5	67.9	60.3	60.4	55.2	53.9	51.6
21	50.8	48.1	53.5	49.4	59.7	52.0	66.0	57.1	72.3	64.2	78.1	68.6	79.8	70.7	79.5	71.1	75.0	66.9	66.0	59.4	58.5	54.2	52.1	50.3
22	48.9	46.8	51.6	48.3	57.7	50.9	64.0	56.2	70.5	63.5	76.3	68.0	78.0	70.2	77.7	70.5	73.3	66.3	64.2	58.6	56.7	53.1	50.3	49.0
23	47.5	45.9	50.2	47.4	56.1	50.2	62.4	55.5	69.1	62.9	74.9	67.5	76.5	69.8	76.4	70.1	72.0	65.8	62.8	58.0	55.2	52.3	49.0	48.1
24	46.1	44.9	48.7	46.4	54.4	49.4	60.8	54.8	67.7	62.4	73.4	67.1	75.1	69.3	75.0	69.7	70.6	65.3	61.4	57.3	53.7	51.5	47.6	47.1

Table 7-38 Cooling Load Component: Lighting, Btu/h

Hour	Usage Profile, %	Heat Gain, Btu/h			Nonsolar RTS Zone Type 8, %	Radiant Cooling Load	Total Sensible Cooling Load	% Lighting to Return 26%	Room Sensible Cooling Load
		Total	Convective 43%	Radiant 57%					
1	0	—	—	—	49	26	26	—	26
2	0	—	—	—	17	26	26	—	26
3	0	—	—	—	9	24	24	—	24
4	0	—	—	—	5	21	21	—	21
5	0	—	—	—	3	19	19	—	19
6	0	—	—	—	2	17	17	—	17
7	100	375	161	214	2	120	281	—	281
8	100	375	161	214	1	154	315	—	315
9	100	375	161	214	1	171	332	—	332
10	100	375	161	214	1	180	341	—	341
11	100	375	161	214	1	184	345	—	345
12	100	375	161	214	1	186	347	—	347
13	100	375	161	214	1	188	349	—	349
14	100	375	161	214	1	188	349	—	349
15	100	375	161	214	1	190	352	—	352
16	100	375	161	214	1	192	354	—	354
17	100	375	161	214	1	195	356	—	356
18	100	375	161	214	1	197	358	—	358
19	0	—	—	—	1	94	94	—	94
20	0	—	—	—	1	60	60	—	60
21	0	—	—	—	0	43	43	—	43
22	0	—	—	—	0	34	34	—	34
23	0	—	—	—	0	30	30	—	30
24	0	—	—	—	0	28	28	—	28
Total		4,501	1,936	2,566	1	2,566	4,501	—	4,501

south, located in Atlanta, Georgia (latitude = 33.64, longitude = 84.43), solar taub = 0.440 and taud = 2.202 from monthly Atlanta weather data for July (Table 1 in Chapter 14, 2013 *ASHRAE Handbook—Fundamentals*). From Table 7-3, the calculated outdoor design temperature for that month and time is 92°F. The ground reflectivity is assumed $\rho_g = 0.2$.

Sol-air temperature is calculated using Equation 30. For the dark-colored wall, $\alpha/h_o = 0.30$, and for vertical surfaces, $\varepsilon\,\Delta R/h_o = 0$. The solar irradiance E_t on the wall must be determined using the equations in Chapter 14, 2013 *ASHRAE Handbook—Fundamentals*:

***Solar Angles*:**

ψ = southwest orientation = +60°

Σ = surface tilt from horizontal (where horizontal = 0°) = 90° for vertical wall surface

3:00 PM LST = hour 15

Calculate solar altitude, solar azimuth, surface solar azimuth, and incident angle as follows:

From Table 2 in Chapter 14 of 2013 *ASHRAE Handbook—Fundamentals*, solar position data and constants for July 21 are

$ET = -6.4$ min

$\delta = 20.4°$

$E_o = 419.8\ \text{Btu/h}\cdot\text{ft}^2$

Local standard meridian (LSM) for Eastern Time Zone = 75°.

Apparent solar time AST

$$\begin{aligned} AST &= LST + ET/60 + (LSM - LON)/15 \\ &= 15 + (-6.4/60) + [(75 - 84.43)/15] \\ &= 14.2647 \end{aligned}$$

Hour angle H, degrees

$$\begin{aligned} H &= 15(AST - 12) \\ &= 15(14.2647 - 12) \\ &= 33.97° \end{aligned}$$

Solar altitude β

$$\begin{aligned} \sin\beta &= \cos L \cos\delta \cos H + \sin L \sin\delta \\ &= \cos(33.64)\cos(20.4)\cos(33.97) + \sin(33.64)\sin(20.4) \\ &= 0.841 \\ \beta &= \sin^{-1}(0.841) = 57.2° \end{aligned}$$

Solar azimuth ϕ

$$\begin{aligned} \cos\phi &= (\sin\beta \sin L - \sin\delta)/(\cos\beta\cos L) \\ &= [(\sin(57.2)\sin(33.64) - \sin(20.4)]/[\cos(57.2)\cos(33.64)] \\ &= 0.258 \\ \phi &= \cos^{-1}(0.253) = 75.05° \end{aligned}$$

Surface-solar azimuth γ

$$\begin{aligned} \gamma &= \phi - \psi \\ &= 75.05 - 60 \\ &= 15.05° \end{aligned}$$

Incident angle θ

$$\begin{aligned} \cos\theta &= \cos\beta\cos g \sin\Sigma + \sin\beta\cos\Sigma \\ &= \cos(57.2)\cos(15.05)\sin(90) + \sin(57.2)\cos(90) \\ &= 0.523 \\ \theta &= \cos^{-1}(0.523) = 58.45° \end{aligned}$$

Beam normal irradiance E_b

$$\begin{aligned} E_b &= E_o \exp(-\tau_b m^{ab}) \\ m &= \text{relative air mass} \\ &= 1/[\sin\beta + 0.50572(6.07995 + \beta)^{-1.6364}], \\ &\quad \beta \text{ expressed in degrees} \\ &= 1.18905 \\ ab &= \text{beam air mass exponent} \\ &= 1.454 - 0.406\tau_b - 0.268\tau_d + 0.021\tau_b\tau_d \\ &= 0.7055705 \\ E_b &= 419.8\exp[-0.556(1.8905^{0.7055705})] \\ &= 255.3\ \text{Btu/h}\cdot\text{ft}^2 \end{aligned}$$

Surface beam irradiance $E_{t,b}$

$$\begin{aligned} E_{t,b} &= E_b\cos\theta \\ &= (255.3)\cos(58.5) \\ &= 133.6\ \text{Btu/h}\cdot\text{ft}^2 \end{aligned}$$

Ratio Y of sky diffuse radiation on vertical surface to sky diffuse radiation on horizontal surface

$$\begin{aligned} Y &= 0.55 + 0.437\cos\theta + 0.313\cos^2\theta \\ &= 0.55 + 0.437\cos(58.45) + 0.313\cos^2(58.45) \\ &= 0.8644 \end{aligned}$$

Diffuse irradiance E_d ***– Horizontal surfaces***

$$\begin{aligned} E_d &= E_o\exp(-\tau_d m^{ad}) \\ ad &= \text{diffuse air mass exponent} \\ &= 0.507 + 0.205\tau_b - 0.080\,\tau_d - 0.190\,\tau_b\tau_d \\ &= 0.2369528 \\ E_d &= E_o\exp(-\tau_d m^{ad}) \\ &= 419.8\exp[-2.202(1.8905^{0.2369528})] \\ &= 42.3\ \text{Btu/h}\cdot\text{ft}^2 \end{aligned}$$

Diffuse irradiance E_d ***– Vertical surfaces***

$$\begin{aligned} E_{t,d} &= E_d Y \\ &= (42.3)(0.864) \\ &= 36.6\ \text{Btu/h}\cdot\text{ft}^2 \end{aligned}$$

Ground reflected irradiance $E_{t,r}$

$$\begin{aligned} E_{t,r} &= (E_b\sin\beta + E_d)\rho_g(1 - \cos\Sigma)/2 \\ &= [255.\,3\sin(57.2) + 42.3](0.2)[1 - \cos(90)]/2 \\ &= 25.7\ \text{Btu/h}\cdot\text{ft}^2 \end{aligned}$$

Total surface irradiance E_t

$$\begin{aligned} E_t &= E_D + E_d + E_r \\ &= 133.6 + 36.6 + 25.7 \\ &= 195.9\ \text{Btu/h}\cdot\text{ft}^2 \end{aligned}$$

Sol-air temperature [from Equation 30]:

$$\begin{aligned} T_e &= t_o + \alpha E_t/h_o - \varepsilon\Delta R/h_o \\ &= 91.9 + (0.30)(195.9) - 0 \\ &= 150.7°\text{F} \end{aligned}$$

This procedure is used to calculate the sol-air temperatures for each hour on each surface. Because of the tedious solar angle and intensity calculations, using a simple computer spreadsheet or other computer software can reduce the effort involved. A spreadsheet was used to calculate a 24 h sol-air temperature profile for the data of this example. See Table 7-39A for the solar angle and intensity calculations and Table 7-

39B for the sol-air temperatures for this wall surface and orientation.

Conductive heat gain is calculated using Equations 31 and 32. First, calculate the 24 h heat input profile using Equation 31 and the sol-air temperatures for a southwest-facing wall with dark exterior color:

$q_{i,1} = (0.077)(60)(73.9 - 75) = -5$ Btu/h
$q_{i,2} = (0.077)(60)(73 - 75) = -9$
$q_{i,3} = (0.077)(60)(72.4 - 75) = -12$
$q_{i,4} = (0.077)(60)(71.8 - 75) = -15$
$q_{i,5} = (0.077)(60)(71.4 - 75) = -17$
$q_{i,6} = (0.077)(60)(72.8 - 75) = -10$
$q_{i,7} = (0.077)(60)(77.4 - 75) = 11$
$q_{i,8} = (0.077)(60)(84.1 - 75) = 42$
$q_{i,9} = (0.077)(60)(90.8 - 75) = 73$
$q_{i,10} = (0.077)(60)(96.7 - 75) = 100$
$q_{i,11} = (0.077)(60)(101.5 - 75) = 122$
$q_{i,12} = (0.077)(60)(105.5 - 75) = 141$
$q_{i,13} = (0.077)(60)(122.4 - 75) = 219$
$q_{i,14} = (0.077)(60)(139.6 - 75) = 298$
$q_{i,15} = (0.077)(60)(150.7 - 75) = 350$
$q_{i,16} = (0.077)(60)(153.7 - 75) = 363$
$q_{i,17} = (0.077)(60)(147.7 - 75) = 336$
$q_{i,18} = (0.077)(60)(131.7 - 75) = 262$
$q_{i,19} = (0.077)(60)(103.1 - 75) = 130$
$q_{i,20} = (0.077)(60)(81.7 - 75) = 31$
$q_{i,21} = (0.077)(60)(79.8 - 75) = 22$
$q_{i,22} = (0.077)(60)(78.0 - 75) = 14$
$q_{i,23} = (0.077)(60)(76.5 - 75) = 7$
$q_{i,24} = (0.077)(60)(75.1 - 75) = 0$

Next, calculate wall heat gain using conduction time series. The preceding heat input profile is used with conduction time series to calculate the wall heat gain. From Table 16 of Chapter 18, 2013 *ASHRAE Handbook—Fundamentals*, the most similar wall construction is wall number 1. This is a spandrel glass wall that has similar mass and thermal capacity. Using Equation 32, the conduction time factors for wall 1 can be used in conjunction with the 24 h heat input profile to determine the wall heat gain at 3:00 PM LST:

$$\begin{aligned} q_{15} &= c_0 q_{i,15} + c_1 q_{i,14} + c_2 q_{i,13} + c_3 q_{i,12} + \cdots + c_{23} q_{i,14} \\ &= (0.18)(350) + (0.58)(298) + (0.20)(219) + (0.04)(141) \\ &\quad + (0.00)(122) + (0.00)(100) + (0.00)(73) + (0.00)(42) \\ &\quad + (0.00)(11) + (0.00)(-10) + (0.00)(-17) + (0.00)(-15) \\ &\quad + (0.00)(-12) + (0.00)(-9) + (0.00)(-5) + (0.00)(0) \\ &\quad + (0.00)(7) + (0.00)(14) + (0.00)(22) + (0.00)(31) \\ &\quad + (0.00)(130) + (0.00)(262) + (0.00)(336) + \\ &\qquad (0.00)(363) = 285 \text{ Btu/h} \end{aligned}$$

Because of the tedious calculations involved, a spreadsheet is used to calculate the remainder of a 24 h heat gain profile indicated in Table 7-39B for the data of this example.

Finally, calculate wall cooling load using radiant time series. Total cooling load for the wall is calculated by summing the convective and radiant portions. The convective portion is simply the wall heat gain for the hour being calculated times the convective fraction for walls from Table 14 of Chapter 18, 2013 *ASHRAE Handbook—Fundamentals* (54%):

$$Q_c = (285)(0.54) = 154 \text{ Btu/h}$$

The radiant portion of the cooling load is calculated using conductive heat gains for the current and past 23 h, the radiant fraction for walls from Table 14 of Chapter 18, 2013 *ASHRAE Handbook—Fundamentals* (46%), and radiant time series from Table 19, in accordance with Equation 34. From Table 19, select the RTS for medium-weight construction, assuming 50% glass and carpeted floors as representative for the described construction. Use the wall heat gains from Table 39B for 24 h design conditions in July. Thus, the radiant cooling load for the wall at 3:00 PM is

$$\begin{aligned} Q_{r,15} &= r_0(0.46)q_{i,15} + r_1(0.46)\, q_{i,14} + r_2(0.46)\, q_{i,13} + r_3(0.46) \\ &\quad q_{i,12} \\ &\quad + \cdots + r_{23}(0.46)\, q_{i,16} \\ &= (0.49)(0.46)(285) + (0.17)(0.46)(214) + \\ &\quad (0.09)(0.46)(150) \\ &\quad + (0.05)(0.46)(119) + (0.03)(0.46)(96) + \\ &\quad (0.02)(0.46)(69) \\ &\quad + (0.02)(0.46)(39) + (0.01)(0.46)(11) + (0.01)(0.46)(- \\ &\quad 8) \\ &\quad + (0.01)(0.46)(-15) + (0.01)(0.46)(-14) + \\ &\quad (0.01)(0.46)(-12) \\ &\quad + (0.01)(0.46)(-9) + (0.01)(0.46)(-4) + \\ &\quad (0.01)(0.46)(1) \\ &\quad + (0.01)(0.46)(8) + (0.01)(0.46)(15) + \\ &\quad (0.01)(0.46)(27) + (0.01)(0.46)(58) + \\ &\quad (0.01)(0.46)(147) + (0.00)(0.46)(257) \\ &\quad + (0.00)(0.46)(329) + (0.00)\,(0.46)(353) + \\ &\quad (0.00)(0.46)(337) = 93 \text{ Btu/h} \end{aligned}$$

The total wall cooling load at the designated hour is thus

$$Q_{\text{wall}} = Q_c + Q_{r15} = 154 + 93 = 247 \text{ Btu/h}$$

Again, a simple computer spreadsheet or other software is necessary to reduce the effort involved. A spreadsheet was used with the heat gain profile to split the heat gain into convective and radiant portions, apply RTS to the radiant portion, and total the convective and radiant loads to determine a 24 h cooling load profile for this example, with results in Table 39B.

Part 3. Window cooling load using radiant time series. Calculate the cooling load contribution, with and without indoor shading (venetian blinds) for the window area facing 60° west of south at 3:00 PM in July for the conference room example.

Solution:

First, calculate the 24 h heat gain profile for the window, then split those heat gains into radiant and convective portions, apply the appropriate RTS to the radiant portion, then sum the convective and radiant cooling load components to determine total window cooling load for the time. The window heat gain components are calculated using Equations (13) to (15). From Part 2, at hour 15 LST (3:00 PM):

$E_{t,b} = 133.6 \text{ Btu/h·ft}^2$

Table 7-39A Wall Component of Solar Irradiance

						Direct Beam Solar			Diffuse Solar Heat Gain					
Local Standard Hour	Apparent Solar Time	Hour Angle *H*	Solar Altitude β	Solar Azimuth φ	Air Solar Mass *m*	E_b, Direct Normal Btu/h·ft^2	Surface Incident Angle θ	Surface Direct Btu/h·ft^2	E_d, Diffuse Horizontal, Btu/h·ft^2	Ground Diffuse Btu/h·ft^2	*Y* Ratio	Sky Diffuse Btu/h·ft^2	Subtotal Diffuse Btu/h·ft^2	Total Surface Irradiance Btu/h·ft^2
1	0.26	–176	–36	–175	0.0	0.0	117.4	0.0	0.0	0.0	0.4500	0.0	0.0	0.0
2	1.26	–161	–33	–159	0.0	0.0	130.9	0.0	0.0	0.0	0.4500	0.0	0.0	0.0
3	2.26	–146	–27	–144	0.0	0.0	144.5	0.0	0.0	0.0	0.4500	0.0	0.0	0.0
4	3.26	–131	–19	–132	0.0	0.0	158.1	0.0	0.0	0.0	0.4500	0.0	0.0	0.0
5	4.26	–116	–9	–122	0.0	0.0	171.3	0.0	0.0	0.0	0.4500	0.0	0.0	0.0
6	5.26	–101	3	–113	16.91455	16.5	172.5	0.0	5.7	0.6	0.4500	2.6	3.2	3.2
7	6.26	–86	14	–105	3.98235	130.7	159.5	0.0	19.8	5.2	0.4500	8.9	14.1	14.1
8	7.26	–71	27	–98	2.22845	193.5	145.9	0.0	29.3	11.6	0.4500	13.2	24.8	24.8
9	8.26	–56	39	–90	1.58641	228.3	132.3	0.0	36.0	18.0	0.4500	16.2	34.2	34.2
10	9.26	–41	51	–81	1.27776	248.8	118.8	0.0	40.7	23.5	0.4500	18.3	41.8	41.8
11	10.26	–26	63	–67	1.11740	260.9	105.6	0.0	43.8	27.7	0.4553	19.9	47.6	47.6
12	11.26	–11	74	–39	1.04214	266.9	92.6	0.0	45.4	30.1	0.5306	24.1	54.2	54.2
13	12.26	4	76	16	1.02872	268.0	80.2	45.5	45.7	30.6	0.6332	29.0	59.6	105.1
14	13.26	19	69	57	1.07337	264.3	68.7	96.2	44.7	29.1	0.7505	33.6	62.7	158.8
15	14.26	34	57	75	1.18905	255.3	58.45	133.6	42.3	25.7	0.8644	36.6	62.3	195.9
16	15.26	49	45	86	1.41566	239.2	50.4	152.4	38.4	20.7	0.9555	36.7	57.4	209.9
17	16.26	64	32	94	1.86186	212.2	45.8	148.1	32.7	14.6	1.0073	33.0	47.6	195.7
18	17.26	79	20	102	2.89735	165.3	45.5	115.8	24.7	8.1	1.0100	24.9	33.1	148.9
19	18.26	94	8	109	6.84406	76.0	49.7	49.1	13.0	2.4	0.9631	12.5	14.9	64.0
20	19.26	109	–3	117	0.0	0.0	57.5	0.0	0.0	0.0	0.8755	0.0	0.0	0.0
21	20.26	124	–14	127	0.0	0.0	67.5	0.0	0.0	0.0	0.7630	0.0	0.0	0.0
22	21.26	139	–23	138	0.0	0.0	79.0	0.0	0.0	0.0	0.6452	0.0	0.0	0.0
23	22.26	154	–30	151	0.0	0.0	91.3	0.0	0.0	0.0	0.5403	0.0	0.0	0.0
24	23.26	169	–35	167	0.0	0.0	104.2	0.0	0.0	0.0	0.4618	0.0	0.0	0.0

Table 7-39B Wall Component of Sol-Air Temperatures, Heat Input, Heat Gain, Cooling Load

							Heat Gain, Btu/h					
								Convective	Radiant			
Local Standard Hour	Total Surface Irradiance Btu/h·ft^2	Outdoor Temp., °F	Sol-Air Temp., °F	Indoor Temp., °F	Heat Input, Btu/h	CTS Type 1, %	Total	54%	46%	Nonsolar RTS Zone Type 8, %	Radiant Cooling Load, Btu/h	Total Cooling Load, Btu/h
1	0.0	73.9	73.9	75	–5	18	1	1	1	49	16	16
2	0.0	73.0	73.0	75	–9	58	–4	–2	–2	17	12	10
3	0.0	72.4	72.4	75	–12	20	–9	–5	–4	9	10	5
4	0.0	71.8	71.8	75	–15	4	–12	–6	–5	5	8	2
5	0.0	71.4	71.4	75	–17	0	–14	–8	–7	3	7	–1
6	3.2	71.8	72.8	75	–10	0	–15	–8	–7	2	6	–2
7	14.1	73.2	77.4	75	11	0	–8	–4	–4	2	7	2
8	24.8	76.7	84.1	75	42	0	11	6	5	1	11	17
9	34.2	80.6	90.8	75	73	0	39	21	18	1	18	39
10	41.8	84.1	96.7	75	100	0	69	37	32	1	27	64
11	47.6	87.2	101.5	75	122	0	96	52	44	1	36	88
12	54.2	89.2	105.5	75	141	0	119	64	55	1	43	108
13	105.1	90.9	122.4	75	219	0	150	81	69	1	53	134
14	158.8	91.9	139.6	75	298	0	214	115	98	1	71	186
15	195.9	91.9	150.7	75	350	0	285	154	131	1	93	247
16	209.9	90.7	153.7	75	363	0	337	182	155	1	114	296
17	195.7	89.0	147.7	75	336	0	353	191	162	1	127	318
18	148.9	87.0	131.7	75	262	0	329	177	151	1	128	306
19	64.0	83.9	103.1	75	130	0	257	139	118	1	115	253
20	0.0	81.7	81.7	75	31	0	147	79	67	1	86	165
21	0.0	79.8	79.8	75	22	0	58	32	27	0	56	88
22	0.0	78.0	78.0	75	14	0	27	14	12	0	38	52
23	0.0	76.5	76.5	75	7	0	15	8	7	0	27	35
24	0.0	75.1	75.1	75	0	0	8	4	4	0	20	25

$E_{t,d}$ = 36.6 Btu/h·ft^2
E_r = 25.7 Btu/h·ft^2
θ = 58.45°

From Chapter 15, Table 10, 2013 *ASHRAE Handbook—Fundamentals* for glass type 5d,

$$\text{SHGC}(\theta) = \text{SHGC}(58.45) = 0.3978 \text{ (interpolated)}$$

$$\langle \text{SHGC} \rangle_D = 0.41$$

From Chapter 15, Table 13B, 2013 *ASHRAE Handbook—Fundamentals*, for light-colored blinds (assumed louver reflectance = 0.8 and louvers positioned at 45° angle) on double-glazed, heat-absorbing windows (Type 5d from Table 13B of Chapter 15), IAC(0) = 0.74, IAC(60) = 0.65, IAC(diff) = 0.79, and radiant fraction = 0.54. Without blinds, IAC = 1.0. Therefore, window heat gain components for hour 15, without blinds, are

$$q_{b15} = AE_{t,b}\,\text{SHGC}(\theta)(\text{IAC}) = (40)(133.6)(0.3978)(1.00) = 2126 \text{ Btu/h}$$

$$q_{d15} = A(E_{t,d} + E_r)\langle \text{SHGC} \rangle_D(\text{IAC}) = (40)(36.6 + 25.7)(0.41)(1.00) = 1021 \text{ Btu/h}$$

$$q_{c15} = UA(t_{\text{out}} - t_{\text{in}}) = (0.56)(40)(91.9 - 75) = 379 \text{ Btu/h}$$

This procedure is repeated to determine these values for a 24 h heat gain profile, shown in Table 7-30.

Total cooling load for the window is calculated by summing the convective and radiant portions. For windows with indoor shading (blinds, drapes, etc.), the direct beam, diffuse, and conductive heat gains may be summed and treated together in calculating cooling loads. However, in this example, the window does not have indoor shading, and the direct beam solar heat gain should be treated separately from the diffuse and conductive heat gains. The direct beam heat gain, without indoor shading, is treated as 100% radiant, and solar RTS factors from Table 20 are used to convert the beam heat gains to cooling loads. The diffuse and conductive heat gains can be totaled and split into radiant and convective portions according to Table 14, and nonsolar RTS factors from Table 19 are used to convert the radiant portion to cooling load.

The solar beam cooling load is calculated using heat gains for the current hour and past 23 h and radiant time series from Table 20, in accordance with Equation 39. From Table 20, select the solar RTS for medium-weight construction, assuming 50% glass and carpeted floors for this example. Using Table 7-40 values for direct solar heat gain, the radiant cooling load for the window direct beam solar component is

$$\begin{aligned} Q_{b,15} &= r_0 q_{b,15} + r_1 q_{b,14} + r_2 q_{b,13} + r_3 q_{b,12} + \cdots + r_{23} q_{b,16} \\ &= (0.54)(2126) + (0.16)(1234) + (0.08)(302) + \\ &\quad (0.04)(0) + (0.03)(0) + (0.02)(0) + (0.01)(0) + \\ &\quad (0.01)(0) + (0.01)(0) + (0.01)(0) + (0.01)(0) + \\ &\quad (0.01)(0) + (0.01)(0) + (0.01)(0) + (0.01)(0) + \\ &\quad (0.01)(0) + (0.01)(0) + (0.01)(0) + (0.01)(0) \\ &\quad + (0.00)(0) + (0.00)(865) + (0.00)(2080) + \\ &\quad (0.00)(2656) + (0.00)(2670) = 1370 \text{ Btu/h} \end{aligned}$$

Table 7-40 Window Component of Heat Gain (No Blinds or Overhang)

	Beam Solar Heat Gain						Diffuse Solar Heat Gain							Conduction		
Local Std. Hour	Beam Normal, Btu/h·ft^2	Surface Incident Angle	Surface Beam, Btu/h·ft^2	Beam SHGC	Adjusted Beam IAC	Beam Solar Heat Gain, Btu/h	Diffuse Horiz. E_d, Btu/h·ft^2	Ground Diffuse, Btu/h·ft^2	Y Ratio	Sky Diffuse, Btu/h·ft^2	Subtotal Diffuse, Btu/h·ft^2	Hemis. SHGC	Diff. Solar Heat Gain, Btu/h	Outside Temp., °F	Conduction Heat Gain, Btu/h	Total Window Heat Gain, Btu/h
1	0.0	117.4	0.0	0.000	1.000	0	0.0	0.0	0.4500	0.0	0.0	0.410	0	73.9	–25	–25
2	0.0	130.9	0.0	0.000	1.000	0	0.0	0.0	0.4500	0.0	0.0	0.410	0	73.0	–45	–45
3	0.0	144.5	0.0	0.000	1.000	0	0.0	0.0	0.4500	0.0	0.0	0.410	0	72.4	–58	–58
4	0.0	158.1	0.0	0.000	1.000	0	0.0	0.0	0.4500	0.0	0.0	0.410	0	71.8	–72	–72
5	0.0	171.3	0.0	0.000	1.000	0	0.0	0.0	0.4500	0.0	0.0	0.410	0	71.4	–81	–81
6	16.5	172.5	0.0	0.000	0.000	0	5.7	0.6	0.4500	2.6	3.2	0.410	52	71.8	–72	–19
7	130.7	159.5	0.0	0.000	0.000	0	19.8	5.2	0.4500	8.9	14.1	0.410	231	73.2	–40	191
8	193.5	145.9	0.0	0.000	0.000	0	29.3	11.6	0.4500	13.2	24.8	0.410	406	76.7	38	444
9	228.3	132.3	0.0	0.000	0.000	0	36.0	18.0	0.4500	16.2	34.2	0.410	560	80.6	125	686
10	248.8	118.8	0.0	0.000	0.000	0	40.7	23.5	0.4500	18.3	41.8	0.410	686	84.1	204	890
11	260.9	105.6	0.0	0.000	0.000	0	43.8	27.7	0.4553	19.9	47.6	0.410	781	87.2	273	1055
12	266.9	92.6	0.0	0.000	0.000	0	45.4	30.1	0.5306	24.1	54.2	0.410	890	89.2	318	1208
13	268.0	80.2	45.5	0.166	1.000	302	45.7	30.6	0.6332	29.0	59.6	0.410	977	90.9	356	1635
14	264.3	68.7	96.2	0.321	1.000	1234	44.7	29.1	0.7505	33.6	62.7	0.410	1028	91.9	379	2640
15	255.3	58.4	133.6	0.398	1.000	2126	42.3	25.7	0.8644	36.6	62.3	0.410	1021	91.9	379	3526
16	239.2	50.4	152.4	0.438	1.000	2670	38.4	20.7	0.9555	36.7	57.4	0.410	942	90.7	352	3964
17	212.2	45.8	148.1	0.448	1.000	2656	32.7	14.6	1.0073	33.0	47.6	0.410	781	89.0	314	3751
18	165.3	45.5	115.8	0.449	1.000	2080	24.7	8.1	1.0100	24.9	33.1	0.410	542	87.0	269	2892
19	76.0	49.7	49.1	0.441	1.000	865	13.0	2.4	0.9631	12.5	14.9	0.410	244	83.9	199	1309
20	0.0	57.5	0.0	0.403	0.000	0	0.0	0.0	0.8755	0.0	0.0	0.410	0	81.7	150	150
21	0.0	67.5	0.0	0.330	0.000	0	0.0	0.0	0.7630	0.0	0.0	0.410	0	79.8	108	108
22	0.0	79.0	0.0	0.185	0.000	0	0.0	0.0	0.6452	0.0	0.0	0.410	0	78.0	67	67
23	0.0	91.3	0.0	0.000	1.000	0	0.0	0.0	0.5403	0.0	0.0	0.410	0	76.5	34	34
24	0.0	104.2	0.0	0.000	1.000	0	0.0	0.0	0.4618	0.0	0.0	0.410	0	75.1	2	2

This process is repeated for other hours; results are listed in Table 7-41.

For diffuse and conductive heat gains, the radiant fraction according to Table 14 is 46%. The radiant portion is processed using nonsolar RTS coefficients from Table 19. The results are listed in Tables 30 and 31. For 3:00 PM, the diffuse and conductive cooling load is 1297 Btu/h.

The total window cooling load at the designated hour is thus

$$Q_{\text{window}} = Q_b + Q_{\text{diff + cond}} = 1370 + 1297 = 2667 \text{ Btu/h}$$

Again, a computer spreadsheet or other software is commonly used to reduce the effort involved in calculations. The spreadsheet illustrated in Table 7-40 is expanded in Table 7-41 to include splitting the heat gain into convective and radiant portions, applying RTS to the radiant portion, and totaling the convective and radiant loads to determine a 24 h cooling load profile for a window without indoor shading.

If the window has an indoor shading device, it is accounted for with the indoor attenuation coefficients (IAC), the radiant fraction, and the RTS type used. If a window has no indoor shading, 100% of the direct beam energy is assumed to be radiant and solar RTS factors are used. However, if an indoor shading device is present, the direct beam is assumed to be interrupted by the shading device, and a portion immediately becomes cooling load by convection. Also, the energy is assumed to be radiated to all surfaces of the room, therefore nonsolar RTS values are used to convert the radiant load into cooling load.

IAC values depend on several factors: (1) type of shading device, (2) position of shading device relative to window, (3) reflectivity of shading device, (4) angular adjustment of shading device, as well as (5) solar position relative to the shading device. These factors are discussed in detail in Chapter 15 of 2013 *ASHRAE Handbook—Fundamentals*. For this example with venetian blinds, the IAC for beam radiation is treated separately from the diffuse solar gain. The direct beam IAC must be adjusted based on the profile angle of the sun. At 3:00 PM in July, the profile angle of the sun relative to the window surface is 58°. Calculated using Equation 39 from Chapter 15, the beam IAC = 0.653. The diffuse IAC is 0.79. Thus, the window heat gains, with light-colored blinds, at 3:00 PM are

$$q_{b15} = AE_D \,\text{SHGC}(\theta)(\text{IAC}) = (40)(133.6)(0.3978)(0.653) = 1388 \text{ Btu/h}$$

$$q_{d15} = A(E_d + E_r)\langle \text{SHGC} \rangle_D(\text{IAC})_D = (40)(36.6 + 25.7)(0.41)(0.79) = 807 \text{ Btu/h}$$

$$q_{c15} = UA(t_{\text{out}} - t_{\text{in}}) = (0.56)(40)(91.9 - 75) = 379 \text{ Btu/h}$$

Because the same radiant fraction and nonsolar RTS are applied to all parts of the window heat gain when indoor shading is present, those loads can be totaled and the cooling load calculated after splitting the radiant portion for processing with non-

Table 7-41 Window Component of Cooling Load (No Blinds or Overhang)

	Unshaded Direct Beam Solar (if AC = 1)						Shaded Direct Beam (AC < 1.0) + Diffuse + Conduction									
Local Standard Hour	Beam Heat Gain, Btu/h	Convective 0%, Btu/h	Radiant 100%, Btu/h	Solar RTS, Zone Type 8, %	Radiant Btu/h	Cooling Load, Btu/h	Beam Heat Gain, Btu/h	Diffuse Heat Gain, Btu/h	Conduction Heat Gain, Btu/h	Total Heat Gain, Btu/h	Convective 54%, Btu/h	Radiant 46%, Btu/h	Nonsolar RTS, Zone Type 8	Radiant Btu/h	Cooling Load, Btu/h	Window Cooling Load, Btu/h
1	0	0	0	54	119	119	0	0	–25	–25	–13	–11	49%	59	45	165
2	0	0	0	16	119	119	0	0	–45	–45	–24	–21	17%	49	24	144
3	0	0	0	8	119	119	0	0	–58	–58	–31	–27	9%	41	9	129
4	0	0	0	4	119	119	0	0	–72	–72	–39	–33	5%	32	–6	113
5	0	0	0	3	119	119	0	0	–81	–81	–44	–37	3%	25	–19	100
6	0	0	0	2	119	119	0	52	–72	–19	–10	–9	2%	32	22	141
7	0	0	0	1	119	119	0	231	–40	191	103	88	2%	78	181	301
8	0	0	0	1	116	116	0	406	38	444	240	204	1%	148	388	504
9	0	0	0	1	104	104	0	560	125	686	370	315	1%	225	596	700
10	0	0	0	1	83	83	0	686	204	890	481	409	1%	300	780	863
11	0	0	0	1	56	56	0	781	273	1055	569	485	1%	365	935	991
12	0	0	0	1	29	29	0	890	318	1208	652	556	1%	426	1078	1108
13	302	0	302	1	172	172	0	977	356	1333	720	613	1%	480	1200	1372
14	1234	0	1234	1	715	715	0	1028	379	1406	759	647	1%	521	1281	1995
15	2126	0	2126	1	1370	1370	0	1021	379	1400	756	644	1%	541	1297	2666
16	2670	0	2670	1	1893	1893	0	942	352	1294	699	595	1%	530	1229	3122
17	2656	0	2656	1	2090	2090	0	781	314	1094	591	503	1%	487	1078	3168
18	2080	0	2080	1	1890	1890	0	542	269	811	438	373	1%	411	849	2739
19	865	0	865	1	1211	1211	0	244	199	444	240	204	1%	302	542	1753
20	0	0	0	0	549	549	0	0	150	150	81	69	1%	196	277	826
21	0	0	0	0	322	322	0	0	108	108	58	49	0%	145	203	525
22	0	0	0	0	213	213	0	0	67	67	36	31	0%	112	148	361
23	0	0	0	0	157	157	0	0	34	34	18	15	0%	89	107	265
24	0	0	0	0	128	128	0	0	2	2	1	1	0%	72	73	201

Table 7-42 Window Component of Cooling Load (With Blinds, No Overhang)

	Unshaded Direct Beam Solar (if AC = 1)						Shaded Direct Beam (AC < 1.0) + Diffuse + Conduction									
Local Standard Hour	Beam Heat Gain, Btu/h	Convective 0%, Btu/h	Radiant 100%, Btu/h	Solar RTS, Zone Type 8, %	Radiant Btu/h	Cooling Load, Btu/h	Beam Heat Gain, Btu/h	Diffuse Heat Gain, Btu/h	Conduction Heat Gain, Btu/h	Total Heat Gain, Btu/h	Convective 54%, Btu/h	Radiant 46%, Btu/h	Nonsolar RTS, Zone Type 8	Radiant, Btu/h	Cooling Load, Btu/h	Window Cooling Load, Btu/h
1	0	0	0	1	0	0	0	0	–25	–25	–11	–13	49%	105	94	94
2	0	0	0	0	0	0	0	0	–45	–45	–21	–24	17%	90	70	70
3	0	0	0	0	0	0	0	0	–58	–58	–27	–31	9%	81	54	54
4	0	0	0	0	0	0	0	0	–72	–72	–33	–39	5%	72	39	39
5	0	0	0	0	0	0	0	0	–81	–81	–37	–44	3%	63	26	26
6	0	0	0	0	0	0	0	41	–72	–30	–14	–16	2%	70	56	56
7	0	0	0	0	0	0	0	183	–40	143	66	77	2%	114	180	180
8	0	0	0	0	0	0	0	321	38	359	165	194	1%	183	348	348
9	0	0	0	0	0	0	0	443	125	568	261	307	1%	260	522	522
10	0	0	0	0	0	0	0	542	204	746	343	403	1%	331	674	674
11	0	0	0	0	0	0	0	617	273	891	410	481	1%	391	801	801
12	0	0	0	0	0	0	0	703	318	1021	470	551	1%	443	913	913
13	0	0	0	0	0	0	196	772	356	1325	609	715	1%	540	1149	1149
14	0	0	0	0	0	0	802	812	379	1992	916	1076	1%	751	1668	1668
15	0	0	0	0	0	0	1388	807	379	2574	1184	1390	1%	987	2171	2171
16	0	0	0	0	0	0	1784	744	352	2880	1325	1555	1%	1170	2495	2495
17	0	0	0	0	0	0	1816	617	314	2747	1263	1483	1%	1221	2484	2484
18	0	0	0	0	0	0	1458	428	269	2156	992	1164	1%	1103	2094	2094
19	0	0	0	0	0	0	624	193	199	1017	468	549	1%	774	1242	1242
20	0	0	0	0	0	0	0	0	150	150	69	81	1%	434	503	503
21	0	0	0	0	0	0	0	0	108	108	49	58	0%	290	339	339
22	0	0	0	0	0	0	0	0	67	67	31	36	0%	209	240	240
23	0	0	0	0	0	0	0	0	34	34	15	18	0%	160	176	176
24	0	0	0	0	0	0	0	0	2	2	1	1	0%	128	129	129

solar RTS. This is illustrated by the spreadsheet results in Table 7-42. The total window cooling load with venetian blinds at 3:00 PM = 2171 Btu/h.

Part 4. Window cooling load using radiant time series for window with overhang shading. Calculate the cooling load contribution for the previous example with the addition of a 10 ft overhang shading the window.

Solution:

In Chapter 15 of 2013 *ASHRAE Handbook—Fundamentals*, methods are described and examples provided for calculating the area of a window shaded by attached vertical or horizontal projections. For 3:00 PM LST IN July, the solar position calculated in previous examples is

$$\text{Solar altitude } \beta = 57.2°$$

$$\text{Solar azimuth } \phi = 75.1°$$

$$\text{Surface-solar azimuth } \gamma = 15.1°$$

From Chapter 15 of 2013 *ASHRAE Handbook—Fundamentals*, Equation 33, profile angle $\Omega$is calculated by

$$\tan\Omega = \tan\beta/\cos\gamma = \tan(57.2)/\cos(15.1) = 1.6087$$

$$\Omega = 58.1°$$

From Chapter 15 of 2013 *ASHRAE Handbook—Fundamentals*, Equation 40, shadow height S_H is

$$S_H = P_H \tan\Omega = 10(1.6087) = 16.1 \text{ ft}$$

Because the window is 6.4 ft tall, at 3:00 PM the window is completely shaded by the 10 ft deep overhang. Thus, the shaded window heat gain includes only diffuse solar and conduction gains. This is converted to cooling load by separating the radiant portion, applying RTS, and adding the resulting radiant cooling load to the convective portion to determine total cooling load. Those results are in Table 7-43. The total window cooling load = 1098 Btu/h.

Part 5. Room cooling load total. Calculate the sensible cooling loads for the previously described office at 3:00 PM in July.

Solution:

The steps in the previous example parts are repeated for each of the internal and external loads components, including the southeast facing window, spandrel and brick walls, the southwest facing brick wall, the roof, people, and equipment loads. The results are tabulated in Table 7-44. The total room sensible cooling load for the office is 3674 Btu/h at 3:00 PM in July. When this calculation process is repeated for a 24 h design day for each month, it is found that the peak room sensible cooling load actually occurs in July at hour 14 (2:00 PM solar time) at 3675 Btu/h as indicated in Table 7-45.

Although simple in concept, these steps involved in calculating cooling loads are tedious and repetitive, even using the "simplified" RTS method; practically, they should be performed using a computer spreadsheet or other program. The calculations should be repeated for multiple design conditions (i.e., times of day, other months) to determine the maximum cooling load for mechanical equipment sizing. Example

Table 7-43 Window Component of Cooling Load (With Blinds and Overhang)

	Overhang and Fins Shading					Shaded Direct Beam (AC < 1.0) + Diffuse + Conduction									
Local Standard Hour	Surface Solar Azimuth	Profile Angle	Shadow Width, ft	Shadow Height, ft	Direct Sunlit Area, ft^2	Beam Heat Gain, Btu/h	Diffuse Heat Gain, Btu/h	Conduction Heat Gain, Btu/h	Total Heat Gain, Btu/h	Convective 54%, Btu/h	Radiant 46%, Btu/h	Nonsolar RTS, Zone Type 8	Radiant, Btu/h	Cooling Load, Btu/h	Window Cooling Load, Btu/h
1	–235	52	0.0	0.0	0.0	0	0	–25	–25	–13	–11	49%	55	42	42
2	–219	40	0.0	0.0	0.0	0	0	–45	–45	–24	–21	17%	43	19	19
3	–204	29	0.0	0.0	0.0	0	0	–58	–58	–31	–27	9%	36	4	4
4	–192	19	0.0	0.0	0.0	0	0	–72	–72	–39	–33	5%	28	–11	–11
5	–182	9	0.0	0.0	0.0	0	0	–81	–81	–44	–37	3%	20	–23	–23
6	–173	–3	0.0	0.0	0.0	0	41	–72	–30	–16	–14	2%	26	10	10
7	–165	–15	0.0	0.0	0.0	0	183	–40	143	77	66	2%	64	141	141
8	–158	–28	0.0	0.0	0.0	0	321	38	359	194	165	1%	122	316	316
9	–150	–43	0.0	0.0	0.0	0	443	125	568	307	261	1%	189	496	496
10	–141	–58	0.0	0.0	0.0	0	542	204	746	403	343	1%	253	656	656
11	–127	–73	0.0	0.0	0.0	0	617	273	891	481	410	1%	310	791	791
12	–99	–87	0.0	0.0	0.0	0	703	318	1021	551	470	1%	363	914	914
13	–44	80	0.0	6.4	0.0	0	772	356	1128	609	519	1%	409	1018	1018
14	–3	69	0.0	6.4	0.0	0	812	379	1190	643	548	1%	443	1085	1085
15	15	58	0.0	6.4	0.0	0	807	379	1186	640	545	1%	457	1098	1098
16	26	48	0.0	6.4	0.0	0	744	352	1096	592	504	1%	449	1040	1040
17	34	38	0.0	6.4	0.0	0	617	314	930	502	428	1%	412	915	915
18	42	26	0.0	4.9	18.9	344	428	269	1041	562	479	1%	427	990	990
19	49	12	0.0	2.2	53.0	414	193	199	806	435	371	1%	380	816	816
20	57	–6	0.0	0.0	0.0	0	0	150	150	81	69	1%	219	300	300
21	67	–32	0.0	0.0	0.0	0	0	108	108	58	49	0%	154	212	212
22	78	–64	0.0	0.0	0.0	0	0	67	67	36	31	0%	113	150	150
23	91	87	0.0	0.0	0.0	0	0	34	34	18	15	0%	87	106	106
24	107	67	0.0	0.0	0.0	0	0	2	2	1	1	0%	70	71	71

spreadsheets for computing each cooling load component using conduction and radiant time series have been compiled and are available from ASHRAE. To illustrate the full building example discussed previously, those individual component spreadsheets have been compiled to allow calculation of cooling and heating loads on a room by room basis as well as for a "block" calculation for analysis of overall areas or buildings where detailed room-by-room data are not available.

7.7.2 Single-Room Example Peak Heating Load

Although the physics of heat transfer that creates a heating load is identical to that for cooling loads, a number of traditionally used simplifying assumptions facilitate a much simpler calculation procedure. As described in the Heating Load Calculations section, design heating load calculations typically assume a single outdoor temperature, with no heat gain from solar or internal sources, under steady-state conditions. Thus, space heating load is determined by computing the heat transfer rate through building envelope elements ($UA\Delta T$) plus heat required because of outdoor air infiltration.

Part 6. Room heating load. Calculate the room heating load for the previous described office, including infiltration airflow at one air change per hour.

Solution:

Because solar heat gain is not considered in calculating design heating loads, orientation of similar envelope elements may be ignored and total areas of each wall or window type combined. Thus, the total spandrel wall area = 60 + 60 = 120 ft^2, total brick wall area = 60 + 40 = 100 ft^2, and total window area = 40 + 40 = 80 ft^2. For this example, use the U-factors that were used for cooling load conditions. In some climates, higher prevalent winds in winter should be considered in calculating U-factors (see Chapter 25 of 2013 *ASHRAE Handbook—Fundamentals* for information on calculating U-factors and surface heat transfer coefficients appropriate for local wind conditions). The 99.6% heating design dry-bulb temperature for Atlanta is 21.5°F and the indoor design temperature is 72°F. The room volume with a 9 ft ceiling = 9 × 130 = 1170 ft^3. At one air change per hour, the infiltration airflow = 1 × 1170/60 = 19.5 cfm. Thus, the heating load is

Windows:	0.56 × 80 × (72 – 21.5) =	2262 Btu/h
Spandrel Wall:	0.077 × 120 × (72 – 21.5)=	467
Brick Wall:	0.08 × 100 × (72 – 21.5)=	404
Roof:	0.032 × 130 × (72 – 21.5)=	210
Infiltration:	19.5 × 1.1 × (72 – 21.5)=	1083
Total Room Heating Load:		4426 Btu/h

Additional examples of nonresidential cooling and heating load calculations are given in Chapter 18 of the 2017 *ASHRAE Handbook—Fundamentals.*

7.8 Problems

7.1 The exterior windows are of double insulating glass with 0.25 in. (6-mm) airspace and have metal sashes. Determine the design U-factor for cooling for the window.

7.2 A store in Lafayette, Indiana, is on the northeast corner of an intersection with one street running due north. The bottom of the show windows are 2 ft, 6 in. above the sidewalk; the show windows are 7 ft high. An aluminum awning with a 3 in. rise per horizontal foot is to be hung with the bottom strut at the window header. Both south and west awnings are to have the same dimensions.

(a) What minimum distance should the strut extend from the building to keep the shade line on the windows at 3 PM sun time?

(b) Which face of the building governs the awning dimensions?

(c) Where will the shade line be at 3 PM on the other face of the building?

(d) What is the elevation of the top of the awnings above the sidewalk?

7.3 Calculate the solar radiation entering through clear glass as shown at right. [Ans 692 Btu/h]

7.4 Solve the following:

(a) Determine the solar angle of incidence for a vertical wall facing 15° west of south when the sun has an azimuth of 79.2° west of south and an altitude of 75.7°. [Ans: 83.8°]

(b) Find the solar incident angle (for direct solar radiation) for a vertical surface facing southeast at 8:30 AM CST on October 22 at 32° N latitude and 95° W longitude.
[Ans: 28.4°]

7.5 What environmental factors affect the solar intensity reaching the earth's surface?

7.6 Determine the heat being dissipated by 50 pendant mounted fluorescent luminaires with four 40 W lamps in each luminaire.

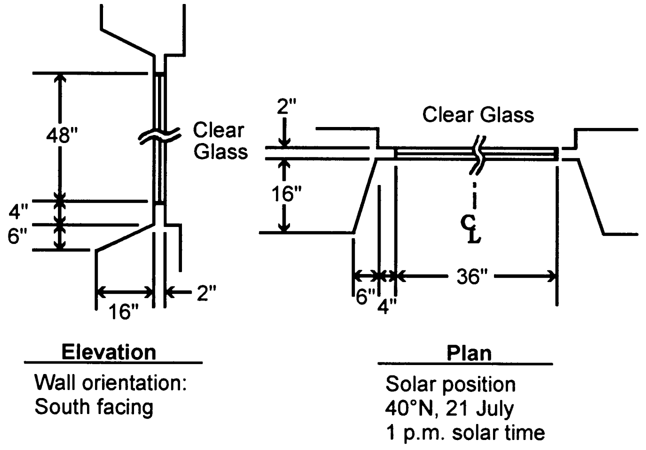
7.7 How much sensible, latent, and total heat is contributed by 50 customers shopping in a drugstore?

7.8 A 1 hp motor driving a pump is located in a space to be air conditioned. Determine heat dissipated to the space from the motor and pump. [Ans: 3390 Btu/h]

7.9 Calculate the heat gain to a room from a 21-lb deep fat fryer (8 kW input rating) if (a) hooded and (b) nonhooded.

7.10 Calculate the maximum heat gain through the floor for a room directly over a boiler room. The air temperature at the underside of the floor is 100°F, and the room air temperature desired close to the floor is 70°F. The floor is 4 in. concrete with vinyl tile finish.

[Ans: 21 Btu/h·ft^2]



7.11 An air-conditioning unit serves an office having the following areas:

Description	Size	Occupancy
General office	25 ft by 50 ft	75 ft^2 per person
Director's room	25 ft by 25 ft	16 people
Conference room	10 ft by 25 ft	Plush furnishings
5 private offices	10 ft by 10 ft	Smoking permitted

What quantity of outdoor air must be brought into the air-conditioning unit for ventilation?

7.12 Suppose the fan of the air-conditioning unit in Problem 7.11 supplies 3200 cfm to the ductwork.

(a) How many air changes per hour are being used? (Assume a ceiling height of 9 ft.)

(b) What is the percentage of outdoor air?

(c) Suppose the minimum recommended quantities of total air and outdoor air are used, what will be the percentage of outdoor air?

7.13 A small parts assembly area in a factory has a working force of 25 men and occupies a space 27.4 m by 9.1 m with a 3 m ceiling. Smoking is not allowed. Determine

(a) Sensible heat load from the occupants

(b) Latent heat load from the occupants

(c) Moisture added from the occupants

(d) The minimum volume of outdoor air for ventilation

(e) Suitable summer design inside dry-bulb temperature

7.14 The office portion of a multistory commercial building, located at 40° N latitude, is shown in the sketch on the next page. Neglecting any outdoor air load, for August 21 determine the cooling load at 4 PM if it is located at 40° N.

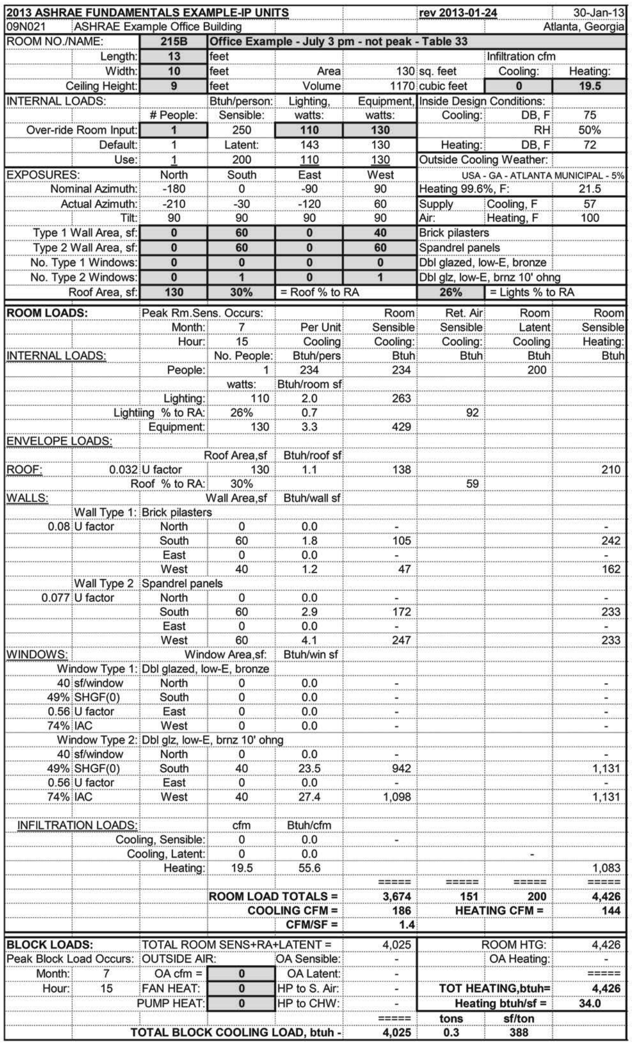
Inside design: 75°F, $W = 0.0102$

Outside design: 95°F, 22°F daily range, $W = 0.0168$

For Office Portion:

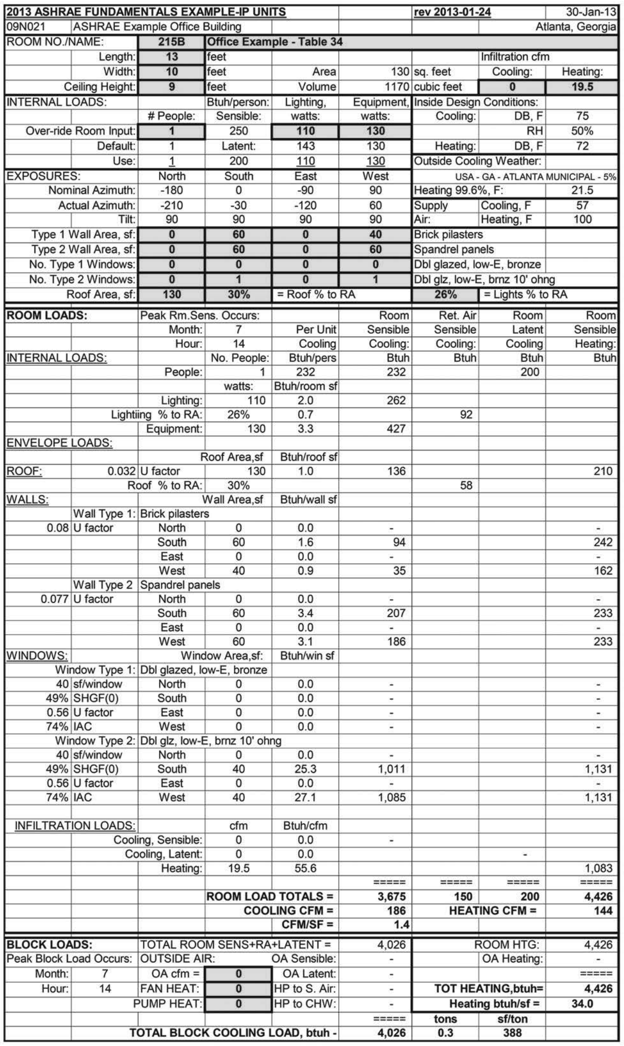
West wall: Net area = 230 ft^2, $U = 0.333$

Table 7-44 Single-Room Example Cooling Load (July 3:00 PM) for ASHRAE Example Office Building, Atlanta, GA



2013 ASHRAE FUNDAMENTALS EXAMPLE-IP UNITS						rev 2013-01-24		30-Jan-13
09N021	ASHRAE Example Office Building							Atlanta, Georgia
ROOM NO./NAME:		215B	Office Example - July 3 pm - not peak - Table 33					
	Length:	13	feet				Infiltration cfm	
	Width:	10	feet	Area	130	sq. feet	Cooling:	Heating:
	Ceiling Height:	9	feet	Volume	1170	cubic feet	0	19.5
INTERNAL LOADS:			Btuh/person:	Lighting,	Equipment,	Inside Design Conditions:		
		# People:	Sensible:	watts:	watts:	Cooling:	DB, F	75
	Over-ride Room Input:	1	250	110	130		RH	50%
	Default:	1	Latent:	143	130	Heating:	DB, F	72
	Use:	1	200	110	130	Outside Cooling Weather:		
EXPOSURES:		North	South	East	West	USA - GA - ATLANTA MUNICIPAL - 5%		
	Nominal Azimuth:	-180	0	-90	90	Heating 99.6%, F:		21.5
	Actual Azimuth:	-210	-30	-120	60	Supply	Cooling, F	57
	Tilt:	90	90	90	90	Air:	Heating, F	100
	Type 1 Wall Area, sf:	0	60	0	40	Brick pilasters		
	Type 2 Wall Area, sf:	0	60	0	60	Spandrel panels		
	No. Type 1 Windows:	0	0	0	0	Dbl glazed, low-E, bronze		
	No. Type 2 Windows:	0	1	0	1	Dbl glz, low-E, brnz 10' ohng		
	Roof Area, sf:	130	30%	= Roof % to RA		26%	= Lights % to RA	
ROOM LOADS:		Peak Rm.Sens. Occurs:			Room	Ret. Air	Room	Room
		Month:	7	Per Unit	Sensible	Sensible	Latent	Sensible
		Hour:	15	Cooling	Cooling:	Cooling:	Cooling	Heating:
INTERNAL LOADS:			No. People:	Btuh/pers	Btuh	Btuh	Btuh	Btuh
		People:	1	234	234		200	
			watts:	Btuh/room sf				
		Lighting:	110	2.0	263			
		Lightiing % to RA:	26%	0.7		92		
		Equipment:	130	3.3	429			
ENVELOPE LOADS:								
			Roof Area,sf	Btuh/roof sf				
ROOF:	0.032	U factor	130	1.1	138			210
		Roof % to RA:	30%			59		
WALLS:			Wall Area,sf	Btuh/wall sf				
	Wall Type 1:	Brick pilasters						
0.08	U factor	North	0	0.0	-			-
		South	60	1.8	105			242
		East	0	0.0	-			-
		West	40	1.2	47			162
	Wall Type 2	Spandrel panels						
0.077	U factor	North	0	0.0	-			-
		South	60	2.9	172			233
		East	0	0.0	-			-
		West	60	4.1	247			233
WINDOWS:			Window Area,sf:	Btuh/win sf				
	Window Type 1:	Dbl glazed, low-E, bronze						
40	sf/window	North	0	0.0	-			-
49%	SHGF(0)	South	0	0.0	-			-
0.56	U factor	East	0	0.0	-			-
74%	IAC	West	0	0.0	-			-
	Window Type 2:	Dbl glz, low-E, brnz 10' ohng						
40	sf/window	North	0	0.0	-			-
49%	SHGF(0)	South	40	23.5	942			1,131
0.56	U factor	East	0	0.0	-			-
74%	IAC	West	40	27.4	1,098			1,131
INFILTRATION LOADS:			cfm	Btuh/cfm				
		Cooling, Sensible:	0	0.0	-			
		Cooling, Latent:	0	0.0			-	
		Heating:	19.5	55.6				1,083
					=====	=====	=====	=====
				ROOM LOAD TOTALS =	**3,674**	**151**	**200**	**4,426**
				COOLING CFM =	**186**		**HEATING CFM =**	**144**
				CFM/SF =	**1.4**			
BLOCK LOADS:		TOTAL ROOM SENS+RA+LATENT =			4,025		ROOM HTG:	4,426
Peak Block Load Occurs:		OUTSIDE AIR:		OA Sensible:	-		OA Heating:	-
Month:	7	OA cfm =	0	OA Latent:	-			=====
Hour:	15	FAN HEAT:	0	HP to S. Air:	-		**TOT HEATING,btuh=**	**4,426**
		PUMP HEAT:	0	HP to CHW:	-		**Heating btuh/sf =**	**34.0**
					=====	**tons**	**sf/ton**	
		TOTAL BLOCK COOLING LOAD, btuh -			**4,025**	**0.3**	**388**	

Table 7-45 Single-Room Example Peak Cooling Load (Sept. 5:00 PM) for ASHRAE Example Office Building, Atlanta, GA



2013 ASHRAE FUNDAMENTALS EXAMPLE-IP UNITS						rev 2013-01-24		30-Jan-13
09N021	ASHRAE Example Office Building							Atlanta, Georgia
ROOM NO./NAME:		215B	Office Example - Table 34					
	Length:	13	feet				Infiltration cfm	
	Width:	10	feet	Area	130	sq. feet	Cooling:	Heating:
	Ceiling Height:	9	feet	Volume	1170	cubic feet	0	19.5
INTERNAL LOADS:			Btuh/person:	Lighting,	Equipment,	Inside Design Conditions:		
		# People:	Sensible:	watts:	watts:	Cooling:	DB, F	75
	Over-ride Room Input:	1	250	110	130		RH	50%
	Default:	1	Latent:	143	130	Heating:	DB, F	72
	Use:	1	200	110	130	Outside Cooling Weather:		
EXPOSURES:		North	South	East	West	USA - GA - ATLANTA MUNICIPAL - 5%		
	Nominal Azimuth:	-180	0	-90	90	Heating 99.6%, F:		21.5
	Actual Azimuth:	-210	-30	-120	60	Supply	Cooling, F	57
	Tilt:	90	90	90	90	Air:	Heating, F	100
	Type 1 Wall Area, sf:	0	60	0	40	Brick pilasters		
	Type 2 Wall Area, sf:	0	60	0	60	Spandrel panels		
	No. Type 1 Windows:	0	0	0	0	Dbl glazed, low-E, bronze		
	No. Type 2 Windows:	0	1	0	1	Dbl glz, low-E, brnz 10' ohng		
	Roof Area, sf:	130	30%	= Roof % to RA		26%	= Lights % to RA	
ROOM LOADS:		Peak Rm.Sens. Occurs:			Room	Ret. Air	Room	Room
		Month:	7	Per Unit	Sensible	Sensible	Latent	Sensible
		Hour:	14	Cooling	Cooling:	Cooling:	Cooling	Heating:
INTERNAL LOADS:			No. People:	Btuh/pers	Btuh	Btuh	Btuh	Btuh
		People:	1	232	232		200	
			watts:	Btuh/room sf				
		Lighting:	110	2.0	262			
		Lightiing % to RA:	26%	0.7		92		
		Equipment:	130	3.3	427			
ENVELOPE LOADS:								
			Roof Area,sf	Btuh/roof sf				
ROOF:	0.032	U factor	130	1.0	136			210
		Roof % to RA:	30%			58		
WALLS:			Wall Area,sf	Btuh/wall sf				
	Wall Type 1:	Brick pilasters						
0.08	U factor	North	0	0.0	-			-
		South	60	1.6	94			242
		East	0	0.0	-			-
		West	40	0.9	35			162
	Wall Type 2	Spandrel panels						
0.077	U factor	North	0	0.0	-			-
		South	60	3.4	207			233
		East	0	0.0	-			-
		West	60	3.1	186			233
WINDOWS:			Window Area,sf:	Btuh/win sf				
	Window Type 1:	Dbl glazed, low-E, bronze						
40	sf/window	North	0	0.0	-			-
49%	SHGF(0)	South	0	0.0	-			-
0.56	U factor	East	0	0.0	-			-
74%	IAC	West	0	0.0	-			-
	Window Type 2:	Dbl glz, low-E, brnz 10' ohng						
40	sf/window	North	0	0.0	-			-
49%	SHGF(0)	South	40	25.3	1,011			1,131
0.56	U factor	East	0	0.0	-			-
74%	IAC	West	40	27.1	1,085			1,131
INFILTRATION LOADS:			cfm	Btuh/cfm				
		Cooling, Sensible:	0	0.0	-			
		Cooling, Latent:	0	0.0			-	
		Heating:	19.5	55.6				1,083
					=====	=====	=====	=====
				ROOM LOAD TOTALS =	**3,675**	**150**	**200**	**4,426**
				COOLING CFM =	**186**		**HEATING CFM =**	**144**
				CFM/SF =	**1.4**			
BLOCK LOADS:		TOTAL ROOM SENS+RA+LATENT =			4,026		ROOM HTG:	4,426
Peak Block Load Occurs:		OUTSIDE AIR:		OA Sensible:	-		OA Heating:	-
Month:	7	OA cfm =	0	OA Latent:	-			=====
Hour:	14	FAN HEAT:	0	HP to S. Air:	-		**TOT HEATING,btuh=**	**4,426**
		PUMP HEAT:	0	HP to CHW:	-		**Heating btuh/sf =**	**34.0**
					=====	**tons**	**sf/ton**	
		TOTAL BLOCK COOLING LOAD, btuh -			**4,026**	**0.3**	**388**	

4 in. red face brick, 4 in. low weight concrete block, and 0.75 in. plaster

Window: Area = 90 ft^2, $U = 0.59$

double pane, regular sheet, insulating glass

People: 4, moderately active office work

Lights: 800 W fluorescent, on continuously

7.15 Calculate the cooling load for the office shown in the diagram.

Location: Indianapolis, Indiana

Occupancy: 180 people

Factory temperature: Assume 5°F below outside ambient

Equipment: 35 business machines, each 110 V at 4.0 A

East and south walls: 8 in. concrete block, $U = 0.3$ Btu/h·ft^2·°F

North and west walls: Concrete block, $U = 0.5$ Btu/h·ft^2·°F

Floor transmission: Neglect

Ceiling: $U = 0.5$ Btu/h·ft^2·°F

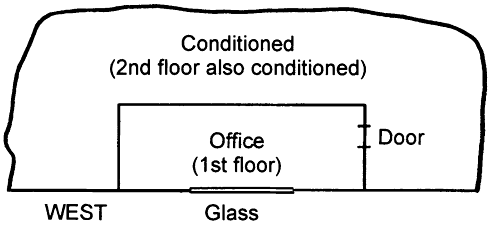


Diagram for Problem 7.15

Lights: 6 W/ft^2

Windows: Insulating glass, 0.25 in. gray plate outside, regular plate inside

Direct-expansion packaged air-conditioning unit will be used.

Locate equipment in factory west of office.

Calculate the heat gain for this office.

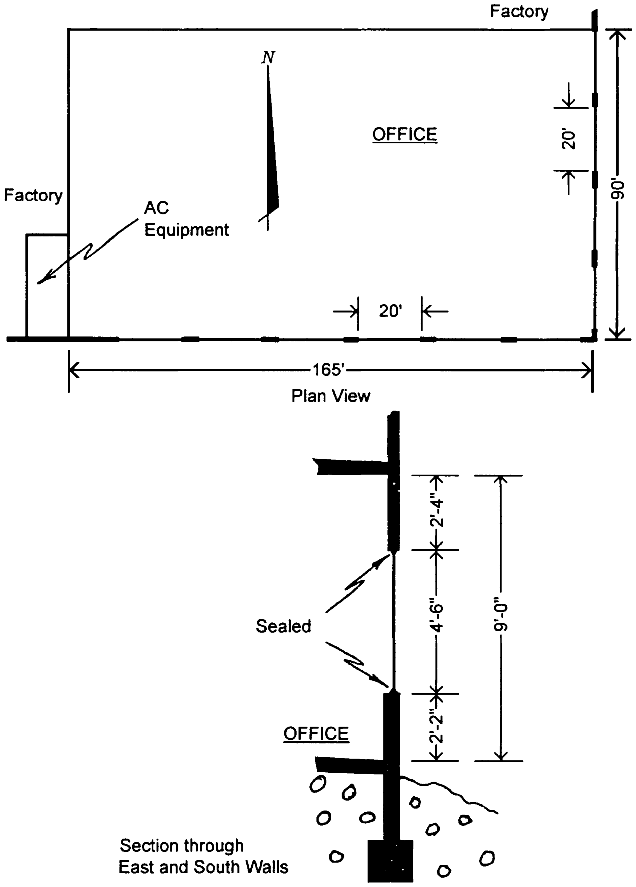


Diagram for Problem 7.16

7.16 An architect provides the following data for a new cafeteria in Tampa, Florida:

Size: 80 ft wide by 100 ft deep; 30 ft deep area in rear is for kitchen and storage. Wall separating this area from main dining area is made of concrete block.

Walls: 4 in. concrete block with an outside facing of 1/2 in. cement mortar; 3/8 in. air space filled with fiberglass insulation, backed by 1 mil aluminum foil; 3/8 in. gypsum wallboard; two coats light green paint; front wall has a canopy projection of 4 ft (located 12 ft above ground level).

Windows: On front wall: 80% of front wall is composed of 1/4 in. gray plate glass, backed by fiberglass draperies of a medium-colored yarn of a close weave; windows are well sealed. On side walls: No windows. On back walls: No windows.

Roof: Flat, stone aggregation on tar base outside; 3/8 in. roofing board; 1/2 in. roof insulation; metal decking; 2 ft air space between steel beams (for lighting fixtures and ductwork); suspended acoustical ceiling.

Floor: 8 in. concrete slab with 1/4 in. floor tile: 1 in. edge insulation.

Doors: Two sets of double swinging doors on opposite sides of front wall; single bank; 7 ft wide (total of two doors); 7 ft high; peak traffic expected is 300 people per hour. There are two service entrance doors on rear, normally closed, 3 ft by 7 ft, with slight cracks all around.

Occupancy: Maximum expected occupancy is 300 between the hours of 5:00 PM and 7:00 PM.

Internal equipment: Usual cooking, washing, and food service storage trays for a cafeteria; a refrigerated food locker is in the rear.

Lighting: Indirect neon lights on ceiling.

Location: Front faces southwest; located in Tampa, Florida.

Calculate the sensible cooling load (using your best judgment as to the number of people present at off-hours) for 10 AM, 2 PM, and 6 PM. Consider the cafeteria as a single zone and plan to air condition the kitchen also.

7.17 Size the cooling system by determining the design cooling loads for the following pharmacy building to be built in Tulsa, Oklahoma.

Overall size: 60 ft by 130 ft by 12 ft high [long sides on east and west]

Walls: 8-inch concrete block with normal weight sand and gravel aggregazte with 1-in. stucco on the outside and 3/4-in. cement plaster with sand aggregate on the inside

Doors: Double 2-1/4-in. solid core doors (6 ft by 7 ft) on north and south

Window: One 6 ft by 90 ft double glass with thermal break frames, 1/8-in. thick glass with 3/8-in. air gap, with translucent roller shades, on west side

Roof/Ceiling: 4-in. lightweight concrete with 3/8-in. built-up roofing on the exterior and 3/4-in. cement plaster with sand aggregate on the inside

Carefully state all assumptions.

7.18 Solve the following:

(a) A 115 ft by 10 ft high wall in Minneapolis, Minnesota, consists of face brick, a 3/4-in. air gap, 8-in. cinder aggregate concrete blocks, 1-in. organic bonded glass fiber insulation, and 4-in. clay tile interior. Determine the design heat loss through the wall in winter in Btu/h.

(b) If the wall of Part (a) is converted to 60% single-glazed glass, what is the winter design heat loss through the total wall in Btu/h?

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SI Tables and Figures

Table 7-1 SI Sol-Air Temperatures (t_e) for July 21, 40° N Latitude

$t_e = t_o + \alpha I_t/h_o - \varepsilon \Delta R/h_o$

	Air Temp.	Light Colored Surface, $\alpha/h_o = 0.026$										Air Temp.	Dark Colored Surface, $\alpha/h_o = 0.052$								
Time	**t_o, °C**	**N**	**NE**	**E**	**SE**	**S**	**SW**	**W**	**NW**	**HOR**	**Time**	**t_o, °C**	**N**	**NE**	**E**	**SE**	**S**	**SW**	**W**	**NW**	**HOR**
1	25.4	25.4	25.4	25.4	25.4	25.4	25.4	25.4	25.4	21.5	1	25.4	25.4	25.4	25.4	25.4	25.4	25.4	25.4	25.4	21.5
2	24.9	24.9	24.9	24.9	24.9	24.9	24.9	24.9	24.9	21.0	2	24.9	24.9	24.9	24.9	24.9	24.9	24.9	24.9	24.9	21.0
3	24.4	24.4	24.4	24.4	24.4	24.4	24.4	24.4	24.4	20.5	3	24.4	24.4	24.4	24.4	24.4	24.4	24.4	24.4	24.4	20.5
4	24.1	24.1	24.1	24.1	24.1	24.1	24.1	24.1	24.1	20.2	4	24.1	24.1	24.1	24.1	24.1	24.1	24.1	24.1	24.1	20.2
5	24.0	24.1	24.2	24.2	24.1	24.0	24.0	24.0	24.0	20.1	5	24.0	24.2	24.4	24.3	24.1	24.0	24.0	24.0	24.0	20.2
6	24.2	27.2	34.5	35.5	29.8	25.1	25.1	25.1	25.1	22.9	6	24.2	30.2	44.7	46.7	35.4	26.0	26.0	26.0	26.0	25.5
7	24.8	27.3	38.1	41.5	35.2	26.5	26.4	26.4	26.4	28.1	7	24.8	29.7	51.5	58.2	45.6	28.2	28.0	28.0	28.0	35.4
8	25.8	28.1	38.0	43.5	38.9	28.2	28.0	28.0	28.0	33.8	8	25.8	30.5	50.1	61.2	52.1	30.7	30.1	30.1	30.1	45.8
9	27.2	29.9	35.9	43.1	41.2	31.5	29.8	29.8	29.8	39.2	9	27.2	32.5	44.5	58.9	55.1	35.8	32.3	32.3	32.3	55.1
10	28.8	31.7	33.4	40.8	41.8	35.4	31.8	31.7	31.7	43.9	10	28.8	34.5	38.0	52.8	54.9	42.0	34.7	34.5	34.5	62.8
11	30.7	33.7	34.0	37.4	41.1	39.0	34.2	33.7	33.7	47.7	11	30.7	36.8	37.2	44.0	51.5	47.4	37.7	36.8	36.8	68.5
12	32.5	35.6	35.6	35.9	39.1	41.4	39.1	35.9	35.6	50.1	12	32.5	38.7	38.7	39.3	45.7	50.4	45.7	39.3	38.7	71.6
13	33.8	36.8	36.8	36.8	37.3	42.1	44.2	40.5	37.1	50.8	13	33.8	39.9	39.9	39.9	40.8	50.5	54.6	47.1	40.3	71.6
14	34.7	37.6	37.6	37.6	37.7	41.3	47.7	46.7	39.3	49.8	14	34.7	40.4	40.4	40.4	40.6	47.9	60.8	58.7	43.9	68.7
15	35.0	37.7	37.6	37.6	37.6	39.3	49.0	50.9	43.7	47.0	15	35.0	40.3	40.1	40.1	40.1	43.6	62.9	66.7	52.3	62.9
16	34.7	37.0	36.9	36.9	36.9	37.1	47.8	52.4	46.9	42.7	16	34.7	39.4	39.0	39.0	39.0	39.6	61.0	70.1	59.0	54.7
17	33.9	36.4	35.5	35.5	35.5	35.6	44.3	50.6	47.2	37.2	17	33.9	38.8	37.1	37.1	37.1	37.3	54.7	67.3	60.6	44.5
18	32.7	35.7	33.6	33.6	33.6	33.6	38.3	44.0	43.0	31.4	18	32.7	38.7	34.5	34.5	34.5	34.5	43.9	55.2	53.2	34.0
19	31.3	31.4	31.3	31.3	31.3	31.3	31.4	31.5	31.5	27.4	19	31.3	31.5	31.3	31.3	31.3	31.3	31.4	31.6	31.7	27.5
20	29.8	29.8	29.8	29.8	29.8	29.8	29.8	29.8	29.8	25.9	20	29.8	29.8	29.8	29.8	29.8	29.8	29.8	29.8	29.8	25.9
21	28.6	28.6	28.6	28.6	28.6	28.6	28.6	28.6	28.6	24.7	21	28.6	28.6	28.6	28.6	28.6	28.6	28.6	28.6	28.6	24.7
22	27.5	27.5	27.5	27.5	27.5	27.5	27.5	27.5	27.5	23.6	22	27.5	27.5	27.5	27.5	27.5	27.5	27.5	27.5	27.5	23.6
23	26.6	26.6	26.6	26.6	26.6	26.6	26.6	26.6	26.6	22.7	23	26.6	26.6	26.6	26.6	26.6	26.6	26.6	26.6	26.6	22.7
24	26.0	26.0	26.0	26.0	26.0	26.0	26.0	26.0	26.0	22.1	24	26.0	26.0	26.0	26.0	26.0	26.0	26.0	26.0	26.0	22.1
Avg.	**29.0**	**30.0**	**32.0**	**33.0**	**32.0**	**31.0**	**32.0**	**33.0**	**32.0**	**32.0**	**Avg.**	**29.0**	**32.0**	**35.0**	**37.0**	**37.0**	**34.0**	**37.0**	**37.0**	**35.0**	**40.0**

Note: Sol-air temperatures are calculated based on $\varepsilon \Delta R/h_o = -3.9$°C for horizontal surfaces and 0°C for vertical surfaces.

Table 7-4 SI Visible Transmittance (T_v), Solar Heat Gain Coefficient (SHGC), Solar Transmittance (T), Front Reflectance (R^f), Back Reflectance (R^b), and Layer Absorptance A_n^f for Glazing and Window Systems

(Table 10, Chapter 15, 2017 ASHRAE Handbook—Fundamentals [SI])

					Center-of-Glazing Properties							Total Window SHGC at Normal Incidence				Total Window T_v at Normal Incidence			
	Glazing System				Incidence Angles							Aluminum		Other Frames		Aluminum		Other Frames	
ID	Glass Thick., mm		Center Glazing T_v		Normal 0.00	40.00	50.00	60.00	70.00	80.00	Hemis., Diffuse	Operable	Fixed	Operable	Fixed	Operable	Fixed	Operable	Fixed
Uncoated Single Glazing																			
1a	3	CLR	0.90	SHGC	0.86	0.84	0.82	0.78	0.67	0.42	0.78	0.78	0.79	0.70	0.76	0.80	0.81	0.72	0.79
				T	0.83	0.82	0.80	0.75	0.64	0.39	0.75								
				R^f	0.08	0.08	0.10	0.14	0.25	0.51	0.14								
				R^b	0.08	0.08	0.10	0.14	0.25	0.51	0.14								
				A_1^f	0.09	0.10	0.10	0.11	0.11	0.11	0.10								
1b	6	CLR	0.88	SHGC	0.81	0.80	0.78	0.73	0.62	0.39	0.73	0.74	0.74	0.66	0.72	0.78	0.79	0.70	0.77
				T	0.77	0.75	0.73	0.68	0.58	0.35	0.69								
				R^f	0.07	0.08	0.09	0.13	0.24	0.48	0.13								
				R^b	0.07	0.08	0.09	0.13	0.24	0.48	0.13								
				A_1^f	0.16	0.17	0.18	0.19	0.19	0.17	0.17								
1c	3	BRZ	0.68	SHGC	0.73	0.71	0.68	0.64	0.55	0.34	0.65	0.67	0.67	0.59	0.65	0.61	0.61	0.54	0.60
				T	0.65	0.62	0.59	0.55	0.46	0.27	0.56								
				R^f	0.06	0.07	0.08	0.12	0.22	0.45	0.12								
				R^b	0.06	0.07	0.08	0.12	0.22	0.45	0.12								
				A_1^f	0.29	0.31	0.32	0.33	0.33	0.29	0.31								
1d	6	BRZ	0.54	SHGC	0.62	0.59	0.57	0.53	0.45	0.29	0.54	0.57	0.57	0.50	0.55	0.48	0.49	0.43	0.48
				T	0.49	0.45	0.43	0.39	0.32	0.18	0.41								
				R^f	0.05	0.06	0.07	0.11	0.19	0.42	0.10								
				R^b	0.05	0.68	0.66	0.62	0.53	0.33	0.10								
				A_1^f	0.46	0.49	0.50	0.51	0.49	0.41	0.48								
1e	3	GRN	0.82	SHGC	0.70	0.68	0.66	0.62	0.53	0.33	0.63	0.64	0.64	0.57	0.62	0.73	0.74	0.66	0.72
				T	0.61	0.58	0.56	0.52	0.43	0.25	0.53								
				R^f	0.06	0.07	0.08	0.12	0.21	0.45	0.11								
				R^b	0.06	0.07	0.08	0.12	0.21	0.45	0.11								
				A_1^f	0.33	0.35	0.36	0.37	0.36	0.31	0.35								
1f	6	GRN	0.76	SHGC	0.60	0.58	0.56	0.52	0.45	0.29	0.54	0.55	0.55	0.49	0.53	0.68	0.68	0.61	0.67
				T	0.47	0.44	0.42	0.38	0.32	0.18	0.40								
				R^f	0.05	0.06	0.07	0.11	0.20	0.42	0.10								
				R^b	0.05	0.06	0.07	0.11	0.20	0.42	0.10								
				A_1^f	0.47	0.50	0.51	0.51	0.49	0.40	0.49								
1g	3	GRY	0.62	SHGC	0.70	0.68	0.66	0.61	0.53	0.33	0.63	0.64	0.64	0.57	0.62	0.55	0.56	0.50	0.55
				T	0.61	0.58	0.56	0.51	0.42	0.24	0.53								
				R^f	0.06	0.07	0.08	0.12	0.21	0.44	0.11								
				R^b	0.06	0.07	0.08	0.12	0.21	0.44	0.11								
				A_1^f	0.33	0.36	0.37	0.37	0.37	0.32	0.35								
1h	6	GRY	0.46	SHGC	0.59	0.57	0.55	0.51	0.44	0.28	0.52	0.54	0.54	0.48	0.52	0.41	0.41	0.37	0.40
				T	0.46	0.42	0.40	0.36	0.29	0.16	0.38								
				R^f	0.05	0.06	0.07	0.10	0.19	0.41	0.10								
				R^b	0.05	0.06	0.07	0.10	0.19	0.41	0.10								
				A_1^f	0.49	0.52	0.54	0.54	0.52	0.43	0.51								
1i	6	BLUGRN	0.75	SHGC	0.62	0.59	0.57	0.54	0.46	0.30	0.55	0.57	0.57	0.50	0.55	0.67	0.68	0.60	0.66
				T	0.49	0.46	0.44	0.40	0.33	0.19	0.42								
				R^f	0.06	0.06	0.07	0.11	0.20	0.43	0.11								
				R^b	0.06	0.06	0.07	0.11	0.20	0.43	0.11								
				A_1^f	0.45	0.48	0.49	0.49	0.47	0.38	0.48								
Reflective Single Glazing																			
1j	6	SS on CLR 8%	0.08	SHGC	0.19	0.19	0.19	0.18	0.16	0.10	0.18	0.18	0.18	0.16	0.17	0.07	0.07	0.06	0.07
				T	0.06	0.06	0.06	0.05	0.04	0.03	0.05								
				R^f	0.33	0.34	0.35	0.37	0.44	0.61	0.36								
				R^b	0.50	0.50	0.51	0.53	0.58	0.71	0.52								
				A_1^f	0.61	0.61	0.60	0.58	0.52	0.37	0.57								
1k	6	SS on CLR 14%	0.14	SHGC	0.25	0.25	0.24	0.23	0.20	0.13	0.23	0.24	0.24	0.21	0.22	0.12	0.13	0.11	0.12
				T	0.11	0.10	0.10	0.09	0.07	0.04	0.09								

Table 7-4 SI Visible Transmittance (T_v), Solar Heat Gain Coefficient (SHGC), Solar Transmittance (T), Front Reflectance (R^f), Back Reflectance (R^b), and Layer Absorptance A_n^f for Glazing and Window Systems (*Continued*)

(Table 10, Chapter 15, 2017 ASHRAE Handbook—Fundamentals [SI])

					Center-of-Glazing Properties							Total Window SHGC at Normal Incidence				Total Window T_v at Normal Incidence			
		Glazing System			Incidence Angles							Aluminum		Other Frames		Aluminum		Other Frames	
ID	Glass Thick., mm		Center Glazing T_v		Normal 0.00	40.00	50.00	60.00	70.00	80.00	Hemis., Diffuse	Operable	Fixed	Operable	Fixed	Operable	Fixed	Operable	Fixed
				R^f	0.26	0.27	0.28	0.31	0.38	0.57	0.30								
				R^b	0.44	0.44	0.45	0.47	0.52	0.67	0.46								
				A_1^f	0.63	0.63	0.62	0.60	0.55	0.39	0.60								
1l	6	SS on CLR 20%	0.20	SHGC	0.31	0.30	0.30	0.28	0.24	0.16	0.28	0.29	0.29	0.26	0.28	0.18	0.18	0.16	0.18
				T	0.15	0.15	0.14	0.13	0.11	0.06	0.13								
				R^f	0.21	0.22	0.23	0.26	0.34	0.54	0.25								
				R^b	0.38	0.38	0.39	0.41	0.48	0.64	0.41								
				A_1^f	0.64	0.64	0.63	0.61	0.56	0.40	0.60								
1m	6	SS on GRN 14%	0.12	SHGC	0.25	0.25	0.24	0.23	0.21	0.14	0.23	0.24	0.24	0.21	0.22	0.11	0.11	0.10	0.11
				T	0.06	0.06	0.06	0.06	0.04	0.03	0.06								
				R^f	0.14	0.14	0.16	0.19	0.27	0.49	0.18								
				R^b	0.44	0.44	0.45	0.47	0.52	0.67	0.46								
				A_I^f	0.80	0.80	0.78	0.76	0.68	0.48	0.75								
1n	6	TI on CLR 20%	0.20	SHGC	0.29	0.29	0.28	0.27	0.23	0.15	0.27	0.27	0.27	0.24	0.26	0.18	0.18	0.16	0.18
				T	0.14	0.13	0.13	0.12	0.09	0.06	0.12								
				R^f	0.22	0.22	0.24	0.26	0.34	0.54	0.26								
				R^b	0.40	0.40	0.42	0.44	0.50	0.65	0.43								
				A_1^f	0.65	0.65	0.64	0.62	0.57	0.40	0.62								
1o	6	TI on CLR 30%	0.30	SHGC	0.39	0.38	0.37	0.35	0.30	0.20	0.35	0.36	0.36	0.32	0.35	0.27	0.27	0.24	0.26
				T	0.23	0.22	0.21	0.19	0.16	0.09	0.20								
				R^f	0.15	0.15	0.17	0.20	0.28	0.50	0.19								
				R^b	0.32	0.33	0.34	0.36	0.43	0.60	0.36								
				A_1^f	0.63	0.65	0.64	0.62	0.57	0.40	0.62								
Uncoated Double Glazing																			
5a	3	CLR CLR	0.81	SHGC	0.76	0.74	0.71	0.64	0.50	0.26	0.66	0.69	0.70	0.62	0.67	0.72	0.73	0.65	0.71
				T	0.70	0.68	0.65	0.58	0.44	0.21	0.60								
				R^f	0.13	0.14	0.16	0.23	0.36	0.61	0.21								
				R^b	0.13	0.14	0.16	0.23	0.36	0.61	0.21								
				A_1^f	0.10	0.11	0.11	0.12	0.13	0.13	0.11								
				A_2^f	0.07	0.08	0.08	0.08	0.07	0.05	0.07								
5b	6	CLR CLR	0.78	SHGC	0.70	0.67	0.64	0.58	0.45	0.23	0.60	0.64	0.64	0.57	0.62	0.69	0.70	0.62	0.69
				T	0.61	0.58	0.55	0.48	0.36	0.17	0.51								
				R^f	0.11	0.12	0.15	0.20	0.33	0.57	0.18								
				R^b	0.11	0.12	0.15	0.20	0.33	0.57	0.18								
				A_1^f	0.17	0.18	0.19	0.20	0.21	0.20	0.19								
				A_2^f	0.11	0.12	0.12	0.12	0.10	0.07	0.11								
5c	3	BRZ CLR	0.62	SHGC	0.62	0.60	0.57	0.51	0.39	0.20	0.53	0.57	0.57	0.50	0.55	0.55	0.56	0.50	0.55
				T	0.55	0.51	0.48	0.42	0.31	0.14	0.45								
				R^f	0.09	0.10	0.12	0.16	0.27	0.49	0.15								
				R^b	0.12	0.13	0.15	0.21	0.35	0.59	0.19								
				A_1^f	0.30	0.33	0.34	0.36	0.37	0.34	0.33								
				A_2^f	0.06	0.06	0.06	0.06	0.05	0.03	0.06								
5d	6	BRZ CLR	0.47	SHGC	0.49	0.46	0.44	0.39	0.31	0.17	0.41	0.45	0.45	0.40	0.43	0.42	0.42	0.38	0.41
				T	0.38	0.35	0.32	0.27	0.20	0.08	0.30								
				R^f	0.07	0.08	0.09	0.13	0.22	0.44	0.12								
				R^b	0.10	0.11	0.13	0.19	0.31	0.55	0.17								
				A_1^f	0.48	0.51	0.52	0.53	0.53	0.45	0.50								
				A_2^f	0.07	0.07	0.07	0.07	0.06	0.04	0.07								
5e	3	GRN CLR	0.75	SHGC	0.60	0.57	0.54	0.49	0.38	0.20	0.51	0.55	0.55	0.49	0.53	0.67	0.68	0.60	0.66
				T	0.52	0.49	0.46	0.40	0.30	0.13	0.43								
				R^f	0.09	0.10	0.12	0.16	0.27	0.50	0.15								
				R^b	0.12	0.13	0.15	0.21	0.35	0.60	0.19								
				A_1^f	0.34	0.37	0.38	0.39	0.39	0.35	0.37								
				A_2^f	0.05	0.05	0.05	0.04	0.04	0.03	0.04								
5f	6	GRN CLR	0.68	SHGC	0.49	0.46	0.44	0.39	0.31	0.17	0.41	0.45	0.45	0.40	0.43	0.61	0.61	0.54	0.60
				T	0.39	0.36	0.33	0.29	0.21	0.09	0.31								

Table 7-4 SI Visible Transmittance (T_v), Solar Heat Gain Coefficient (SHGC), Solar Transmittance (T), Front Reflectance (R^f), Back Reflectance (R^b), and Layer Absorptance A_n^f for Glazing and Window Systems (*Continued*)

(Table 10, Chapter 15, 2017 ASHRAE Handbook—Fundamentals [SI])

					Center-of-Glazing Properties							Total Window SHGC at Normal Incidence				Total Window T_v at Normal Incidence			
	Glazing System				Incidence Angles							Aluminum		Other Frames		Aluminum		Other Frames	
ID	Glass Thick., mm		Center Glazing T_v		Normal 0.00	40.00	50.00	60.00	70.00	80.00	Hemis., Diffuse	Operable	Fixed	Operable	Fixed	Operable	Fixed	Operable	Fixed
				R^f	0.08	0.08	0.10	0.14	0.23	0.45	0.13								
				R^b	0.10	0.11	0.13	0.19	0.31	0.55	0.17								
				A_1^f	0.49	0.51	0.05	0.53	0.52	0.43	0.50								
				A_2^f	0.05	0.05	0.05	0.05	0.04	0.03	0.05								
5g	3	GRY CLR	0.56	SHGC	0.60	0.57	0.54	0.48	0.37	0.20	0.51	0.55	0.55	0.49	0.53	0.50	0.50	0.45	0.49
				T	0.51	0.48	0.45	0.39	0.29	0.12	0.42								
				R^f	0.09	0.09	0.11	0.16	0.26	0.48	0.14								
				R^b	0.12	0.13	0.15	0.21	0.34	0.59	0.19								
				A_1^f	0.34	0.37	0.39	0.40	0.41	0.37	0.37								
				A_2^f	0.05	0.06	0.06	0.05	0.05	0.03	0.05								
5h	6	GRY CLR	0.41	SHGC	0.47	0.44	0.42	0.37	0.29	0.16	0.39	0.43	0.43	0.38	0.42	0.36	0.37	0.33	0.36
				T	0.36	0.32	0.29	0.25	0.18	0.07	0.28								
				R^f	0.07	0.07	0.08	0.12	0.21	0.43	0.12								
				R^b	0.10	0.11	0.13	0.18	0.31	0.55	0.17								
				A_1^f	0.51	0.54	0.56	0.57	0.56	0.47	0.53								
				A_2^f	0.07	0.07	0.07	0.06	0.05	0.03	0.06								
5i	6	BLUGRN CLR	0.67	SHGC	0.50	0.47	0.45	0.40	0.32	0.17	0.43	0.46	0.46	0.41	0.44	0.60	0.60	0.54	0.59
				T	0.40	0.37	0.34	0.30	0.22	0.10	0.32								
				R^f	0.08	0.08	0.10	0.14	0.24	0.46	0.13								
				R^b	0.11	0.11	0.14	0.19	0.31	0.55	0.17								
				A_1^f	0.47	0.49	0.50	0.51	0.50	0.42	0.48								
				A_2^f	0.06	0.06	0.06	0.05	0.04	0.03	0.05								
5j	6	HI-P GRN CLR	0.59	SHGC	0.39	0.37	0.35	0.31	0.25	0.14	0.33	0.36	0.36	0.32	0.35	0.53	0.53	0.47	0.52
				T	0.28	0.26	0.24	0.20	0.15	0.06	0.22								
				R^f	0.06	0.07	0.08	0.12	0.21	0.43	0.11								
				R^b	0.10	0.11	0.13	0.19	0.31	0.55	0.17								
				A_1^f	0.62	0.65	0.65	0.65	0.62	0.50	0.63								
				A_2^f	0.03	0.03	0.03	0.03	0.02	0.01	0.03								
Reflective Double Glazing																			
5k	6	SS on CLR 8%, CLR	0.07	SHGC	0.13	0.12	0.12	0.11	0.10	0.06	0.11	0.13	0.13	0.11	0.12	0.06	0.06	0.06	0.06
				T	0.05	0.05	0.04	0.04	0.03	0.01	0.04								
				R^f	0.33	0.34	0.35	0.37	0.44	0.61	0.37								
				R^b	0.38	0.37	0.38	0.40	0.46	0.61	0.40								
				A_1^f	0.61	0.61	0.60	0.58	0.53	0.37	0.56								
				A_2^f	0.01	0.01	0.01	0.01	0.01	0.01	0.01								
5l	6	SS on CLR 14%, CLR	0.13	SHGC	0.17	0.17	0.16	0.15	0.13	0.08	0.16	0.17	0.16	0.14	0.15	0.12	0.12	0.10	0.11
				T	0.08	0.08	0.08	0.07	0.05	0.02	0.07								
				R^f	0.26	0.27	0.28	0.31	0.38	0.57	0.30								
				R^b	0.34	0.33	0.34	0.37	0.44	0.60	0.36								
				A_1^f	0.63	0.64	0.64	0.63	0.61	0.56	0.60								
				A_2^f	0.02	0.02	0.02	0.02	0.02	0.02	0.02								
5m	6	SS on CLR 20%, CLR	0.18	SHGC	0.22	0.21	0.21	0.19	0.16	0.09	0.20	0.21	0.21	0.18	0.20	0.16	0.16	0.14	0.16
				T	0.12	0.11	0.11	0.09	0.07	0.03	0.10								
				R^f	0.21	0.22	0.23	0.26	0.34	0.54	0.25								
				R^b	0.30	0.30	0.31	0.34	0.41	0.59	0.33								
				A_1^f	0.64	0.64	0.63	0.62	0.57	0.41	0.61								
				A_2^f	0.03	0.03	0.03	0.03	0.02	0.02	0.03								
5n	6	SS on GRN 14%, CLR	0.11	SHGC	0.16	0.16	0.15	0.14	0.12	0.08	0.14	0.16	0.16	0.14	0.14	0.10	0.10	0.09	0.10
				T	0.05	0.05	0.05	0.04	0.03	0.01	0.04								
				R^f	0.14	0.14	0.16	0.19	0.27	0.49	0.18								
				R^b	0.34	0.33	0.34	0.37	0.44	0.60	0.36								
				A_1^f	0.80	0.80	0.79	0.76	0.69	0.49	0.76								
				A_2^f	0.01	0.01	0.01	0.01	0.01	0.01	0.01								
5o	6	TI on CLR 20%, CLR	0.18	SHGC	0.21	0.20	0.19	0.18	0.15	0.09	0.18	0.20	0.20	0.18	0.19	0.16	0.16	0.14	0.16
				T	0.11	0.10	0.10	0.08	0.06	0.03	0.09								
				R^f	0.22	0.22	0.24	0.27	0.34	0.54	0.26								
				R^b	0.32	0.31	0.32	0.35	0.42	0.59	0.35								
				A_1^f	0.65	0.66	0.65	0.63	0.58	0.41	0.62								
				A_2^f	0.02	0.02	0.02	0.02	0.02	0.01	0.02								

Table 7-4 SI Visible Transmittance (T_v), Solar Heat Gain Coefficient (SHGC), Solar Transmittance (T), Front Reflectance (R^f), Back Reflectance (R^b), and Layer Absorptance A^f_n for Glazing and Window Systems (*Continued*)

(Table 10, Chapter 15, 2017 ASHRAE Handbook—Fundamentals [SI])

	Glazing System				Center-of-Glazing Properties							Total Window SHGC at Normal Incidence				Total Window T_v at Normal Incidence			
					Incidence Angles							Aluminum		Other Frames		Aluminum		Other Frames	
ID	Glass Thick., mm		Center Glazing T_v		Normal 0.00	40.00	50.00	60.00	70.00	80.00	Hemis., Diffuse	Operable	Fixed	Operable	Fixed	Operable	Fixed	Operable	Fixed
5p	6	TI on CLR 30%, CLR	0.27	SHGC	0.29	0.28	0.27	0.25	0.20	0.12	0.25	0.27	0.27	0.24	0.26	0.24	0.24	0.22	0.24
				T	0.18	0.17	0.16	0.14	0.10	0.05	0.15								
				R^f	0.15	0.15	0.17	0.20	0.29	0.51	0.19								
				R^b	0.27	0.27	0.28	0.31	0.40	0.58	0.31								
				A^f_1	0.64	0.64	0.63	0.62	0.58	0.43	0.61								
				A^f_2	0.04	0.04	0.04	0.04	0.03	0.02	0.04								
Low-e Double Glazing, e = 0.2 on surface 2																			
17a	3	LE CLR	0.76	SHGC	0.65	0.64	0.61	0.56	0.43	0.23	0.57	0.59	0.60	0.53	0.58	0.68	0.68	0.61	0.67
				T	0.59	0.56	0.54	0.48	0.36	0.18	0.50								
				R^f	0.15	0.16	0.18	0.24	0.37	0.61	0.22								
				R^b	0.17	0.18	0.20	0.26	0.38	0.61	0.24								
				A^f_1	0.20	0.21	0.21	0.21	0.20	0.16	0.20								
				A^f_2	0.07	0.07	0.08	0.08	0.07	0.05	0.07								
17b	6	LE CLR	0.73	SHGC	0.60	0.59	0.57	0.51	0.40	0.21	0.53	0.55	0.55	0.49	0.53	0.65	0.66	0.58	0.64
				T	0.51	0.48	0.46	0.41	0.30	0.14	0.43								
				R^f	0.14	0.15	0.17	0.22	0.35	0.59	0.21								
				R^b	0.15	0.16	0.18	0.23	0.35	0.57	0.22								
				A^f_1	0.26	0.26	0.26	0.26	0.25	0.19	0.25								
				A^f_2	0.10	0.11	0.11	0.11	0.10	0.07	0.10								
Low-e Double Glazing, e = 0.2 on surface 3																			
17c	3	CLR LE	0.76	SHGC	0.70	0.68	0.65	0.59	0.46	0.24	0.61	0.64	0.64	0.57	0.62	0.68	0.68	0.1	0.67
				T	0.59	0.56	0.54	0.48	0.36	0.18	0.50								
				R^f	0.17	0.18	0.20	0.26	0.38	0.61	0.24								
				R^b	0.15	0.16	0.18	0.24	0.37	0.61	0.22								
				A^f_1	0.11	0.12	0.13	0.13	0.14	0.15	0.12								
				A^f_2	0.14	0.14	0.14	0.13	0.11	0.07	0.13								
17d	6	CLR LE	0.73	SHGC	0.65	0.63	0.60	0.54	0.42	0.21	0.56	0.59	0.60	0.53	0.58	0.65	0.66	0.58	0.64
				T	0.51	0.48	0.46	0.41	0.30	0.14	0.43								
				R^f	0.15	0.16	0.18	0.23	0.35	0.57	0.22								
				R^b	0.14	0.15	0.17	0.22	0.35	0.59	0.21								
				A^f_1	0.17	0.19	0.20	0.21	0.22	0.22	0.19								
				A^f_2	0.17	0.17	0.17	0.15	0.13	0.07	0.16								
17e	3	BRZ LE	0.58	SHGC	0.57	0.54	0.51	0.46	0.35	0.18	0.48	0.52	0.52	0.46	0.51	0.52	0.52	0.46	0.51
				T	0.46	0.43	0.41	0.36	0.26	0.12	0.38								
				R^f	0.12	0.12	0.14	0.18	0.28	0.50	0.17								
				R^b	0.14	0.15	0.17	0.23	0.35	0.60	0.21								
				A^f_1	0.31	0.34	0.35	0.37	0.38	0.35	0.34								
				A^f_2	0.11	0.11	0.10	0.10	0.08	0.04	0.10								
17f	6	BRZ LE	0.45	SHGC	0.45	0.42	0.40	0.35	0.27	0.14	0.38	0.42	0.42	0.37	0.40	0.40	0.41	0.36	0.40
				T	0.33	0.30	0.28	0.24	0.17	0.07	0.26								
				R^f	0.09	0.09	0.10	0.14	0.23	0.44	0.13								
				R^b	0.13	0.14	0.16	0.21	0.34	0.58	0.20								
				A^f_1	0.48	0.51	0.52	0.54	0.53	0.45	0.50								
				A^f_2	0.11	0.11	0.10	0.09	0.07	0.04	0.09								
17g	3	GRN LE	0.70	SHGC	0.55	0.52	0.50	0.44	0.34	0.17	0.46	0.50	0.51	0.45	0.49	0.62	0.63	0.56	0.62
				T	0.44	0.41	0.38	0.33	0.24	0.11	0.36								
				R^f	0.11	0.11	0.13	0.17	0.27	0.48	0.16								
				R^b	0.14	0.15	0.17	0.23	0.35	0.60	0.21								
				A^f_1	0.35	0.38	0.39	0.41	0.42	0.37	0.38								
				A^f_2	0.11	0.10	0.10	0.09	0.07	0.04	0.09								
17h	6	GRN LE	0.61	SHGC	0.41	0.39	0.36	0.32	0.25	0.13	0.34	0.38	0.38	0.34	0.36	0.54	0.55	0.49	0.54
				T	0.29	0.26	0.24	0.21	0.15	0.06	0.23								
				R^f	0.08	0.08	0.09	0.13	0.22	0.43	0.13								
				R^b	0.13	0.14	0.16	0.21	0.34	0.58	0.20								
				A^f_1	0.53	0.57	0.58	0.59	0.58	0.48	0.56								

Table 7-4 SI Visible Transmittance (T_v), Solar Heat Gain Coefficient (SHGC), Solar Transmittance (T), Front Reflectance (R^f), Back Reflectance (R^b), and Layer Absorptance A_n^f for Glazing and Window Systems (*Continued*)

(Table 10, Chapter 15, 2017 ASHRAE Handbook—Fundamentals [SI])

					Center-of-Glazing Properties							Total Window SHGC at Normal Incidence				Total Window T_v at Normal Incidence			
	Glazing System				Incidence Angles							Aluminum		Other Frames		Aluminum		Other Frames	
ID	Glass Thick., mm		Center Glazing T_v		Normal 0.00	40.00	50.00	60.00	70.00	80.00	Hemis., Diffuse	Operable	Fixed	Operable	Fixed	Operable	Fixed	Operable	Fixed
				A_2^f	0.10	0.09	0.09	0.08	0.06	0.03	0.08								
17i	3	GRY LE	0.53	SHGC	0.54	0.51	0.49	0.44	0.33	0.17	0.46	0.50	0.50	0.44	0.48	0.47	0.48	0.42	0.47
				T	0.43	0.40	0.38	0.33	0.24	0.11	0.35								
				R^f	0.11	0.11	0.13	0.17	0.27	0.48	0.16								
				R^b	0.14	0.15	0.17	0.22	0.35	0.60	0.21								
				A_1^f	0.36	0.39	0.40	0.42	0.42	0.38	0.39								
				A_2^f	0.10	0.10	0.10	0.09	0.07	0.04	0.09								
17j	6	GRY LE	0.37	SHGC	0.39	0.37	0.35	0.31	0.24	0.13	0.33	0.36	0.36	0.32	0.35	0.33	0.33	0.30	0.33
				T	0.27	0.25	0.23	0.20	0.14	0.06	0.21								
				R^f	0.09	0.09	0.11	0.14	0.23	0.44	0.14								
				R^b	0.13	0.14	0.16	0.22	0.34	0.58	0.20								
				A_1^f	0.55	0.58	0.59	0.59	0.58	0.48	0.56								
				A_2^f	0.09	0.09	0.08	0.07	0.06	0.03	0.08								
17k	6	BLUGRN LE	0.62	SHGC	0.45	0.42	0.40	0.35	0.27	0.14	0.37	0.42	0.42	0.37	0.40	0.55	0.56	0.50	0.55
				T	0.32	0.29	0.27	0.23	0.17	0.07	0.26								
				R^f	0.09	0.09	0.10	0.14	0.23	0.44	0.13								
				R^b	0.13	0.14	0.16	0.21	0.34	0.58	0.20								
				A_1^f	0.48	0.51	0.53	0.54	0.54	0.45	0.51								
				A_2^f	0.11	0.10	0.10	0.09	0.07	0.03	0.09								
17l	6	HI-P GRN LE	0.55	0.241	0.34	0.31	0.30	0.26	0.20	0.11	0.28	0.32	0.32	0.28	0.30	0.49	0.50	0.44	0.48
				T	0.22	0.19	0.18	0.15	0.10	0.04	0.17								
				R^f	0.07	0.07	0.08	0.11	0.20	0.41	0.11								
				R^b	0.13	0.14	0.16	0.21	0.33	0.58	0.20								
				A_1^f	0.64	0.67	0.68	0.68	0.66	0.53	0.65								
				A_2^f	0.08	0.07	0.06	0.06	0.04	0.02	0.06								
Low-e Double Glazing, e = 0.1 on surface 2																			
21a	3	LE CLR	0.76	SHGC	0.65	0.64	0.62	0.56	0.43	0.23	0.57	0.59	0.60	0.53	0.58	0.68	0.68	0.61	0.67
				T	0.59	0.56	0.54	0.48	0.36	0.18	0.50								
				R^f	0.15	0.16	0.18	0.24	0.37	0.61	0.22								
				R^b	0.17	0.18	0.20	0.26	0.38	0.61	0.24								
				A_1^f	0.20	0.21	0.21	0.21	0.20	0.16	0.20								
				A_2^f	0.07	0.07	0.08	0.08	0.07	0.05	0.07								
21b	6	LE CLR	0.72	SHGC	0.60	0.59	0.57	0.51	0.40	0.21	0.53	0.55	0.55	0.49	0.53	0.64	0.65	0.58	0.63
				T	0.51	0.48	0.46	0.41	0.30	0.14	0.43								
				R^f	0.14	0.15	0.17	0.22	0.35	0.59	0.21								
				R^b	0.15	0.16	0.18	0.23	0.35	0.57	0.22								
				A_1^f	0.26	0.26	0.26	0.26	0.25	0.19	0.25								
				A_2^f	0.10	0.11	0.11	0.11	0.10	0.07	0.10								
Low-e Double Glazing, e = 0.1 on surface 3																			
21c	3	CLR LE	0.75	SHGC	0.60	0.58	0.56	0.51	0.40	0.22	0.52	0.55	0.55	0.49	0.53	0.67	0.68	0.60	0.66
				T	0.48	0.45	0.43	0.37	0.27	0.13	0.40								
				R^f	0.26	0.27	0.28	0.32	0.42	0.62	0.31								
				R^b	0.24	0.24	0.26	0.29	0.38	0.58	0.28								
				A_1^f	0.12	0.13	0.14	0.14	0.15	0.15	0.13								
				A_2^f	0.14	0.15	0.15	0.16	0.16	0.10	0.15								
21d	6	CLR LE	0.72	SHGC	0.56	0.55	0.52	0.48	0.38	0.20	0.49	0.51	0.52	0.46	0.50	0.64	0.65	0.58	0.63
				T	0.42	0.40	0.37	0.32	0.24	0.11	0.35								
				R^f	0.24	0.24	0.25	0.29	0.38	0.58	0.28								
				R^b	0.20	0.20	0.22	0.26	0.34	0.55	0.25								
				A_1^f	0.19	0.20	0.21	0.22	0.23	0.22	0.21								
				A_2^f	0.16	0.17	0.17	0.17	0.16	0.10	0.16								
21e	3	BRZ LE	0.57	SHGC	0.48	0.46	0.44	0.40	0.31	0.17	0.42	0.44	0.44	0.39	0.43	0.51	0.51	0.46	0.50
				T	0.37	0.34	0.32	0.27	0.20	0.08	0.30								
				R^f	0.18	0.17	0.19	0.22	0.30	0.50	0.21								
				R^b	0.23	0.23	0.25	0.29	0.37	0.57	0.28								

Table 7-4 SI Visible Transmittance (T_v), Solar Heat Gain Coefficient (SHGC), Solar Transmittance (T), Front Reflectance (R^f), Back Reflectance (R^b), and Layer Absorptance A_n^f for Glazing and Window Systems (*Continued*)

(Table 10, Chapter 15, 2017 ASHRAE Handbook—Fundamentals [SI])

	Glazing System		Center Glazing		Center-of-Glazing Properties							Total Window SHGC at Normal Incidence				Total Window T_v at Normal Incidence			
					Incidence Angles							Aluminum		Other Frames		Aluminum		Other Frames	
ID	Glass Thick., mm		Center Glazing T_v		Normal 0.00	40.00	50.00	60.00	70.00	80.00	Hemis., Diffuse	Operable	Fixed	Operable	Fixed	Operable	Fixed	Operable	Fixed
				A_1^f	0.34	0.37	0.38	0.39	0.39	0.35	0.37								
				A_2^f	0.11	0.12	0.12	0.12	0.11	0.07	0.11								
21f	6	BRZ LE	0.45	SHGC	0.39	0.37	0.35	0.31	0.24	0.13	0.33	0.36	0.36	0.32	0.35	0.40	0.41	0.36	0.40
				T	0.27	0.24	0.22	0.19	0.13	0.05	0.21								
				R^f	0.12	0.12	0.13	0.16	0.24	0.44	0.16								
				R^b	0.19	0.20	0.22	0.25	0.34	0.55	0.24								
				A_1^f	0.51	0.54	0.55	0.56	0.55	0.46	0.53								
				A_2^f	0.10	0.10	0.10	0.10	0.09	0.05	0.10								
21g	3	GRN LE	0.68	SHGC	0.46	0.44	0.42	0.38	0.30	0.16	0.40	0.42	0.43	0.38	0.41	0.61	0.61	0.54	0.60
				T	0.36	0.32	0.30	0.26	0.18	0.08	0.28								
				R^f	0.17	0.16	0.17	0.20	0.29	0.48	0.20								
				R^b	0.23	0.23	0.25	0.29	0.37	0.57	0.27								
				A_1^f	0.38	0.41	0.42	0.43	0.43	0.38	0.40								
				A_2^f	0.10	0.11	0.11	0.11	0.10	0.06	0.10								
21h	6	GRN LE	0.61	SHGC	0.36	0.33	0.31	0.28	0.22	0.12	0.30	0.34	0.34	0.30	0.32	0.54	0.55	0.49	0.54
				T	0.24	0.21	0.19	0.16	0.11	0.05	0.18								
				R^f	0.11	0.10	0.11	0.14	0.22	0.43	0.14								
				R^b	0.19	0.20	0.22	0.25	0.34	0.55	0.24								
				A_1^f	0.56	0.59	0.61	0.61	0.59	0.48	0.58								
				A_2^f	0.09	0.09	0.09	0.08	0.08	0.04	0.08								
21i	3	GRY LE	0.52	SHGC	0.46	0.44	0.42	0.38	0.30	0.16	0.39	0.42	0.43	0.38	0.41	0.46	0.47	0.42	0.46
				T	0.35	0.32	0.30	0.25	0.18	0.08	0.28								
				R^f	0.16	0.16	0.17	0.20	0.28	0.48	0.20								
				R^b	0.23	0.23	0.25	0.29	0.37	0.57	0.27								
				A_1^f	0.39	0.42	0.43	0.44	0.44	0.38	0.41								
				A_2^f	0.10	0.11	0.11	0.11	0.10	0.06	0.10								
21j	6	GRY LE	0.37	SHGC	0.34	0.32	0.30	0.27	0.21	0.12	0.28	0.32	0.32	0.28	0.30	0.33	0.33	0.30	0.33
				T	0.23	0.20	0.18	0.15	0.11	0.04	0.17								
				R^f	0.11	0.11	0.12	0.15	0.23	0.44	0.15								
				R^b	0.20	0.20	0.22	0.25	0.34	0.55	0.24								
				A_1^f	0.58	0.60	0.61	0.61	0.59	0.48	0.59								
				A_2^f	0.08	0.08	0.08	0.08	0.07	0.04	0.08								
21k	6	BLUGRN LE	0.62	SHGC	0.39	0.37	0.34	0.31	0.24	0.13	0.33	0.36	0.36	0.32	0.35	0.55	0.56	0.50	0.55
				T	0.28	0.25	0.23	0.20	0.14	0.06	0.22								
				R^f	0.12	0.12	0.13	0.16	0.24	0.44	0.16								
				R^b	0.23	0.23	0.25	0.28	0.37	0.57	0.27								
				A_1^f	0.51	0.54	0.56	0.56	0.55	0.46	0.53								
				A_2^f	0.08	0.09	0.08	0.08	0.08	0.05	0.08								
21l	6	HI-P GRN W/LE CLR	0.57	SHGC	0.31	0.30	0.29	0.26	0.21	0.12	0.27	0.29	0.29	0.26	0.28	0.51	0.51	0.46	0.50
				T	0.22	0.21	0.19	0.17	0.12	0.06	0.18								
				R^f	0.07	0.07	0.09	0.13	0.22	0.46	0.12								
				R^b	0.23	0.23	0.24	0.28	0.37	0.57	0.27								
				A_1^f	0.67	0.68	0.67	0.66	0.62	0.46	0.65								
				A_2^f	0.04	0.05	0.05	0.05	0.04	0.03	0.04								
Low-e Double Glazing, e = 0.05 on surface 2																			
25a	3	LE CLR	0.72	SHGC	0.41	0.40	0.38	0.34	0.27	0.14	0.36	0.38	0.38	0.34	0.36	0.64	0.65	0.58	0.63
				T	0.37	0.35	0.33	0.29	0.22	0.11	0.31								
				R^f	0.35	0.36	0.37	0.40	0.47	0.64	0.39								
				R^b	0.39	0.39	0.40	0.43	0.50	0.66	0.42								
				A_1^f	0.24	0.26	0.26	0.27	0.28	0.23	0.26								
				A_2^f	0.04	0.04	0.04	0.04	0.03	0.03	0.04								
25b	6	LE CLR	0.70	SHGC	0.37	0.36	0.34	0.31	0.24	0.13	0.32	0.34	0.34	0.30	0.33	0.62	0.63	0.56	0.62
				T	0.30	0.28	0.27	0.23	0.17	0.08	0.25								
				R^f	0.30	0.30	0.32	0.35	0.42	0.60	0.34								

Table 7-4 SI Visible Transmittance (T_v), Solar Heat Gain Coefficient (SHGC), Solar Transmittance (T), Front Reflectance (R^f), Back Reflectance (R^b), and Layer Absorptance A_n^f for Glazing and Window Systems (*Continued*)

(Table 10, Chapter 15, 2017 ASHRAE Handbook—Fundamentals [SI])

					Center-of-Glazing Properties							Total Window SHGC at Normal Incidence				Total Window T_v at Normal Incidence			
	Glazing System				Incidence Angles							Aluminum		Other Frames		Aluminum		Other Frames	
ID	Glass Thick., mm		Center Glazing T_v		Normal 0.00	40.00	50.00	60.00	70.00	80.00	Hemis., Diffuse	Operable	Fixed	Operable	Fixed	Operable	Fixed	Operable	Fixed
				R^b	0.35	0.35	0.35	0.38	0.44	0.60	0.37								
				A_1^f	0.34	0.35	0.35	0.36	0.35	0.28	0.34								
				A_2^f	0.06	0.07	0.07	0.06	0.06	0.04	0.06								
25c	6	BRZ W/LE CLR	0.42	SHGC	0.26	0.25	0.24	0.22	0.18	0.10	0.23	0.25	0.25	0.22	0.23	0.37	0.38	0.34	0.37
				T	0.18	0.17	0.16	0.14	0.10	0.05	0.15								
				R^f	0.15	0.16	0.17	0.21	0.29	0.51	0.20								
				R^b	0.34	0.34	0.35	0.37	0.44	0.60	0.37								
				A_1^f	0.63	0.63	0.63	0.61	0.57	0.42	0.60								
				A_2^f	0.04	0.04	0.04	0.04	0.03	0.03	0.04								
25d	6	GRN W/LE CLR	0.60	SHGC	0.31	0.30	0.28	0.26	0.21	0.12	0.27	0.29	0.29	0.26	0.28	0.53	0.54	0.48	0.53
				T	0.22	0.21	0.20	0.17	0.13	0.06	0.18								
				R^f	0.10	0.10	0.12	0.16	0.25	0.48	0.15								
				R^b	0.35	0.34	0.35	0.37	0.44	0.60	0.37								
				A_1^f	0.64	0.64	0.64	0.63	0.59	0.43	0.62								
				A_2^f	0.05	0.05	0.05	0.05	0.04	0.03	0.05								
25e	6	GRY W/LE CLR	0.35	SHGC	0.24	0.23	0.22	0.20	0.16	0.09	0.21	0.23	0.23	0.20	0.21	0.31	0.32	0.28	0.31
				T	0.16	0.15	0.14	0.12	0.09	0.04	0.13								
				R^f	0.12	0.13	0.15	0.18	0.26	0.49	0.17								
				R^b	0.34	0.34	0.35	0.37	0.44	0.60	0.37								
				A_1^f	0.69	0.69	0.68	0.67	0.62	0.45	0.66								
				A_2^f	0.03	0.03	0.03	0.03	0.03	0.02	0.03								
25f	6	BLUE W/LE CLR	0.45	SHGC	0.27	0.26	0.25	0.23	0.18	0.11	0.24	0.26	0.25	0.22	0.24	0.40	0.41	0.36	0.40
				T	0.19	0.18	0.17	0.15	0.11	0.05	0.16								
				R^f	0.12	0.12	0.14	0.17	0.26	0.49	0.16								
				R^b	0.34	0.34	0.35	0.37	0.44	0.60	0.37								
				A_1^f	0.66	0.66	0.65	0.64	0.60	0.44	0.63								
				A_2^f	0.04	0.04	0.04	0.04	0.04	0.03	0.04								
25g	6	HI-P GRN W/LE CLR	0.53	SHGC	0.27	0.26	0.25	0.23	0.18	0.11	0.23	0.26	0.25	0.22	0.24	0.47	0.48	0.42	0.47
				T	0.18	0.17	0.16	0.14	0.10	0.05	0.15								
				R^f	0.07	0.07	0.09	0.13	0.22	0.46	0.12								
				R^b	0.35	0.34	0.35	0.38	0.44	0.60	0.37								
				A_1^f	0.71	0.72	0.71	0.69	0.64	0.47	0.68								
				A_2^f	0.04	0.04	0.04	0.04	0.03	0.02	0.04								
Triple Glazing																			
29a	3	CLR CLR CLR	0.74	SHGC	0.68	0.65	0.62	0.54	0.39	0.18	0.57	0.62	0.62	0.55	0.60	0.66	0.67	0.59	0.65
				T	0.60	0.57	0.53	0.45	0.31	0.12	0.49								
				R^f	0.17	0.18	0.21	0.28	0.42	0.65	0.25								
				R^b	0.17	0.18	0.21	0.28	0.42	0.65	0.25								
				A_1^f	0.10	0.11	0.12	0.13	0.14	0.14	0.12								
				A_2^f	0.08	0.08	0.09	0.09	0.08	0.07	0.08								
				A_3^f	0.06	0.06	0.06	0.06	0.05	0.03	0.06								
29b	6	CLR CLR CLR	0.70	SHGC	0.61	0.58	0.55	0.48	0.35	0.16	0.51	0.56	0.56	0.50	0.54	0.62	0.63	0.56	0.62
				T	0.49	0.45	0.42	0.35	0.24	0.09	0.39								
				R^f	0.14	0.15	0.18	0.24	0.37	0.59	0.22								
				R^b	0.14	0.15	0.18	0.24	0.37	0.59	0.22								
				A_1^f	0.17	0.19	0.20	0.21	0.22	0.21	0.19								
				A_2^f	0.12	0.13	0.13	0.13	0.12	0.08	0.12								
				A_3^f	0.08	0.08	0.08	0.08	0.06	0.03	0.08								
29c	6	HI-P GRN CLR CLR	0.53	SHGC	0.32	0.29	0.27	0.24	0.18	0.10	0.26	0.30	0.30	0.26	0.29	0.47	0.48	0.42	0.47
				T	0.20	0.17	0.15	0.12	0.07	0.02	0.15								
				R^f	0.06	0.07	0.08	0.11	0.20	0.41	0.11								
				R^b	0.13	0.14	0.16	0.22	0.35	0.57	0.20								
				A_1^f	0.64	0.67	0.68	0.68	0.66	0.53	0.65								

Table 7-4 SI Visible Transmittance (T_v), Solar Heat Gain Coefficient (SHGC), Solar Transmittance (T), Front Reflectance (R^f), Back Reflectance (R^b), and Layer Absorptance A^f_n for Glazing and Window Systems (*Continued*)

(Table 10, Chapter 15, 2017 ASHRAE Handbook—Fundamentals [SI])

					Center-of-Glazing Properties							Total Window SHGC at Normal Incidence				Total Window T_v at Normal Incidence			
	Glazing System				Incidence Angles							Aluminum		Other Frames		Aluminum		Other Frames	
ID	Glass Thick., mm		Center Glazing T_v		Normal 0.00	40.00	50.00	60.00	70.00	80.00	Hemis., Diffuse	Operable	Fixed	Operable	Fixed	Operable	Fixed	Operable	Fixed
				A^f_2	0.06	0.06	0.05	0.05	0.05	0.03	0.05								
				A^f_3	0.04	0.04	0.04	0.03	0.02	0.01	0.04								
Triple Glazing, e = 0.2 on surface 2																			
32a	3	LE CLR CLR	0.68	SHGC	0.60	0.58	0.55	0.48	0.35	0.17	0.51	0.55	0.55	0.49	0.53	0.61	0.61	0.54	0.60
				T	0.50	0.47	0.44	0.38	0.26	0.10	0.41								
				R^f	0.17	0.19	0.21	0.27	0.41	0.64	0.25								
				R^b	0.19	0.20	0.22	0.29	0.42	0.63	0.26								
				A^f_1	0.20	0.20	0.20	0.21	0.21	0.17	0.20								
				A^f_2	0.08	0.08	0.08	0.09	0.08	0.07	0.08								
				A^f_3	0.06	0.06	0.06	0.06	0.05	0.03	0.06								
32b	6	LE CLR CLR	0.64	SHGC	0.53	0.50	0.47	0.41	0.29	0.14	0.44	0.49	0.49	0.43	0.47	0.57	0.58	0.51	0.56
				T	0.39	0.36	0.33	0.27	0.17	0.06	0.30								
				R^f	0.14	0.15	0.17	0.21	0.31	0.53	0.20								
				R^b	0.16	0.16	0.19	0.24	0.36	0.57	0.22								
				A^f_1	0.28	0.31	0.31	0.34	0.37	0.31	0.31								
				A^f_2	0.11	0.11	0.11	0.11	0.10	0.08	0.11								
				A^f_3	0.08	0.08	0.08	0.07	0.05	0.03	0.07								
Triple Glazing, e = 0.2 on surface 5																			
32c	3	CLR CLR LE	0.68	SHGC	0.62	0.60	0.57	0.49	0.36	0.16	0.52	0.57	0.57	0.50	0.55	0.61	0.61	0.54	0.60
				T	0.50	0.47	0.44	0.38	0.26	0.10	0.41								
				R^f	0.19	0.20	0.22	0.29	0.42	0.63	0.26								
				R^b	0.18	0.19	0.21	0.27	0.41	0.64	0.25								
				A^f_1	0.11	0.12	0.13	0.14	0.15	0.15	0.13								
				A^f_2	0.09	0.10	0.10	0.10	0.10	0.08	0.10								
				A^f_3	0.11	0.11	0.11	0.10	0.08	0.04	0.10								
32d	6	CLR CLR LE	0.64	SHGC	0.56	0.53	0.50	0.44	0.32	0.15	0.47	0.51	0.52	0.46	0.50	0.57	0.58	0.1	0.56
				T	0.39	0.36	0.33	0.27	0.17	0.06	0.30								
				R^f	0.16	0.16	0.19	0.24	0.36	0.57	0.22								
				R^b	0.14	0.15	0.17	0.21	0.31	0.53	0.20								
				A^f_1	0.17	0.19	0.20	0.21	0.22	0.22	0.19								
				A^f_2	0.13	0.14	0.14	0.14	0.13	0.10	0.13								
				A^f_3	0.15	0.16	0.15	0.14	0.12	0.05	0.14								
Triple Glazing, e = 0.1 on surface 2 and 5																			
40a	3	LE CLR LE	0.62	SHGC	0.41	0.39	0.37	0.32	0.24	0.12	0.34	0.38	0.38	0.34	0.36	0.55	0.56	0.50	0.55
				T	0.29	0.26	0.24	0.20	0.13	0.05	0.23								
				R^f	0.30	0.30	0.31	0.34	0.41	0.59	0.33								
				R^b	0.30	0.30	0.31	0.34	0.41	0.59	0.33								
				A^f_1	0.25	0.27	0.28	0.30	0.32	0.27	0.28								
				A^f_2	0.07	0.08	0.08	0.08	0.07	0.06	0.07								
				A^f_3	0.08	0.09	0.09	0.09	0.07	0.04	0.08								
40b	6	LE CLR LE	0.59	SHGC	0.36	0.34	0.32	0.28	0.21	0.10	0.30	0.34	0.34	0.30	0.32	0.53	0.53	0.47	0.52
				T	0.24	0.21	0.19	0.16	0.10	0.03	0.18								
				R^f	0.34	0.34	0.35	0.38	0.44	0.61	0.37								
				R^b	0.23	0.23	0.25	0.28	0.36	0.56	0.27								
				A^f_1	0.24	0.25	0.26	0.28	0.30	0.25	0.26								
				A^f_2	0.10	0.11	0.11	0.11	0.10	0.07	0.10								
				A^f_3	0.09	0.09	0.09	0.08	0.07	0.03	0.08								
Triple Glazing, e = 0.05 on surface 2 and 4																			
49	3	LE LE CLR	0.58	SHGC	0.27	0.25	0.24	0.21	0.16	0.08	0.23	0.26	0.25	0.22	0.25	0.52	0.52	0.46	0.51
				T	0.18	0.17	0.16	0.13	0.08	0.03	0.14								
				R^f	0.41	0.41	0.42	0.44	0.50	0.65	0.44								
				R^b	0.46	0.45	0.46	0.48	0.53	0.68	0.47								
				A^f_1	0.27	0.28	0.28	0.29	0.30	0.24	0.28								
				A^f_2	0.12	0.12	0.12	0.12	0.11	0.07	0.12								

Table 7-4 SI Visible Transmittance (T_v), Solar Heat Gain Coefficient (SHGC), Solar Transmittance (T), Front Reflectance (R^f), Back Reflectance (R^b), and Layer Absorptance A_n^f for Glazing and Window Systems (*Continued*)

(Table 10, Chapter 15, 2017 ASHRAE Handbook—Fundamentals [SI])

	Glazing System				Center-of-Glazing Properties							Total Window SHGC at Normal Incidence				Total Window T_v at Normal Incidence			
					Incidence Angles							Aluminum		Other Frames		Aluminum		Other Frames	
ID	Glass Thick., mm		Center Glazing T_v		Normal 0.00	40.00	50.00	60.00	70.00	80.00	Hemis., Diffuse	Operable	Fixed	Operable	Fixed	Operable	Fixed	Operable	Fixed
				A_3^f	0.02	0.02	0.02	0.02	0.01	0.01	0.02								
50	6	LE LE CLR	0.55	SHGC	0.26	0.25	0.23	0.21	0.16	0.08	0.22	0.25	0.25	0.21	0.24	0.49	0.0	0.44	0.48
				T	0.15	0.14	0.12	0.10	0.07	0.02	0.12								
				R^f	0.33	0.33	0.34	0.37	0.43	0.60	0.36								
				R^b	0.39	0.38	0.38	0.40	0.46	0.61	0.40								
				A_1^f	0.34	0.36	0.36	0.37	0.36	0.28	0.35								
				A_2^f	0.15	0.15	0.15	0.14	0.12	0.08	0.14								
				A_3^f	0.03	0.03	0.03	0.03	0.02	0.01	0.03								

KEY:

CLR = clear, BRZ = bronze, GRN = green, GRY = gray, BLUGRN = blue-green, SS = stainless steel reflective coating, TI = titanium reflective coating

Reflective coating descriptors include percent visible transmittance as x%.

HI-P GRN = high-performance green tinted glass, LE = low-emissivity coating

T_v = visible transmittance, T = solar transmittance, SHGC = solar heat gain coefficient, and H. = hemispherical SHGC

ID #s refer to U-factors in Table 4, except for products 49 and 50.

Table 7-8 SI Interior Solar Attenuation Coefficients (IAC) for Single or Double Glazings Shaded by Interior Venetian Blinds or Roller Shades

		Glazing Solar Transmittance[b]			IAC				
					Venetian Blinds		Roller Shades		
Glazing System[a]	Nominal Thickness Each Pane, mm	Outer Pane	Single or Inner Pane	Glazing SHGC	Medium	Light	Opaque Dark	Opaque White	Translucent Light
Single Glazing Systems									
Clear, residential	3		0.87 to 0.80	0.86	0.75	0.68	0.82	0.40	0.40
Clear, commercial	6 to 13		0.80 to 0.71	0.82					
Clear, pattern	3 to 13		0.87 to 0.79						
Heat absorbing, pattern	3			0.59					
Tinted	5, 5.5		0.74, 0.71						
Above glazings, automated blinds[e]				0.86	0.64	0.59			
Above glazings, tightly closed vertical blinds				0.85	0.30	0.26			
Heat absorbing[f]	6		0.46	0.59	0.84	0.78	0.66	0.44	0.47
Heat absorbing, pattern	6								
Tinted	3, 6		0.59, 0.45						
Heat absorbing or pattern			0.44 to 0.30	0.59	0.79	0.76	0.59	0.41	0.47
Heat absorbing	10		0.34						
Heat absorbing or pattern			0.29 to 0.15						
			0.24	0.37	0.99	0.94	0.85	0.66	0.73
Reflective coated glass			0.26 to 0.52	0.83	0.75				
Double Glazing Systems[g]									
Clear double, residential	3	0.87	0.87	0.76	0.71	0.66	0.81	0.40	0.46
Clear double, commercial	6	0.80	0.80	0.70					
Heat absorbing double[f]	6	0.46	0.8	0.47	0.72	0.66	0.74	0.41	0.55
Reflective double				0.17 to 0.35	0.90	0.86			
Other Glazings (Approximate)					0.83	0.77	0.74	0.45	0.52
± Range of Variation[h]					0.15	0.17	0.16	0.21	0.21

[a] Systems listed in the same table block have same IAC.

[b] Values or ranges given for identification of appropriate IAC value; where paired, solar transmittances and thicknesses correspond. SHGC is for unshaded glazing at normal incidence.

[c] Typical thickness for residential glass.

[d] From measurements by Van Dyke and Konen (1982) for 45° open venetian blinds, 35° solar incidence, and 35° profile angle.

[e] Use these values only when operation is automated for exclusion of beam solar (as opposed to daylight maximization). Also applies to tightly closed horizontal blinds.

[f] Refers to gray, bronze, and green tinted heat-absorbing glass (on exterior pane in double glazing)

[g] Applies either to factory-fabricated insulating glazing units or to prime windows plus storm windows.

[h] The listed approximate IAC value may be higher or lower by this amount, due to glazing/shading interactions and variations in the shading properties (e.g., manufacturing tolerances).

Table 7-9 SI Between-Glass Solar Attenuation Coefficients (BAC) for Double Glazing with Between-Glass Shading

					Type of Shading		
	Nominal Thickness,	Solar Transmittance[a]			Venetian Blinds		Louvered
Type of Glass	Each Pane	Outer Pane	Inner Pane	Description of Air Space	Light	Medium	Sun Screen
Clear out, Clear in	2.43, 3 mm	0.87	0.87	Shade in contact with glass or shade separated from glass by air space.	0.33	0.36	0.43
Clear out, Clear in	6 mm	0.80	0.80	Shade in contact with glass-voids filled with plastic.	—	—	0.49
Heat-absorbing[b] out, Clear in				Shade in contact with glass or shade separated from glass by air space.	0.28	0.30	0.37
	6 mm	0.46	0.80	Shade in contact with glass-voids filled with plastic.	—	—	0.41

[a] Refer to manufacturers' literature for exact values.

[b] Refers to gray, bronze and green tinted heat-absorbing glass.

Table 7-14 SI Rates of Heat Gain from Occupants[a,b,c]

(Table 1, Chapter 18, 2017 ASHRAE Handbook—Fundamentals [SI])

		Total Heat, W				% Sensible Heat that is Radiant[b]
Degree of Activity	Location	Adult Male	Adjusted, M/F[a]	Sensible Heat, W	Latent Heat, W	Low *V*
Seated at theater	Theater	115	105	70	35	60
Seated, very light work	Offices, hotels, apartments	130	115	70	45	
Moderately active office work	Offices, hotels, apartments	140	130	75	55	
Standing, light work; walking	Department store; retail store	160	130	75	55	58
Walking, standing	Drug store, bank	160	145	75	70	
Sedentary work	Restaurant[c]	145	160	80	80	
Light bench work	Factory	235	220	80	140	
Moderate dancing	Dance hall	265	250	90	160	49
Walking 4.8 km/h; light machine work	Factory	295	295	110	185	
Bowling[d]	Bowling alley	440	425	170	255	
Heavy work	Factory	440	425	170	255	54
Heavy machine work; lifting	Factory	470	470	185	285	
Athletics	Gymnasium	585	525	210	315	

Notes:

1. Tabulated values are based on 24°C room dry-bulb temperature. For 27°C room dry bulb, total heat remains the same, but sensible heat values should be decreased by approximately 20%, and latent heat values increased accordingly.
2. Also see Table 4, Chapter 9, for additional rates of metabolic heat generation.
3. All values are rounded to nearest 5 W.

[a] Adjusted heat gain is based on normal percentage of men, women, and children for the application listed, and assumes that gain from an adult female is 85% of that for an adult male, and gain from a child is 75% of that for an adult male.

[b] Values approximated from data in Table 6, Chapter 9, where *V* is air velocity with limits shown in that table.

[c] Adjusted heat gain includes 18 W for food per individual (9 W sensible and 9 W latent).

[d] Figure one person per alley actually bowling, and all others as sitting (117 W) or standing or walking slowly (231 W).

Table 7-16 SI Heat Gain from Typical Electric Motors

					Location of Motor and Driven Equipment with Respect to Conditioned Space or Airstream		
					A	B	C
Motor Nameplate or Rated Horsepower	(kW)	Motor Type	Nominal rpm	Full Load Motor Efficiency, %	Motor in, Driven Equipment in, Watt	Motor out, Driven Equipment in, Watt	Motor in, Driven Equipment out, Watt
0.05	(0.04)	Shaded pole	1500	35	105	35	70
0.08	(0.06)	Shaded pole	1500	35	170	59	110
0.125	(0.09)	Shaded pole	1500	35	264	94	173
0.16	(0.12)	Shaded pole	1500	35	340	117	223
0.25	(0.19)	Split phase	1750	54	346	188	158
0.33	(0.25)	Split phase	1750	56	439	246	194
0.50	(0.37)	Split phase	1750	60	621	372	249
0.75	(0.56)	3-Phase	1750	72	776	557	217
1	(0.75)	3-Phase	1750	75	993	747	249
1.5	(1.1)	3-Phase	1750	77	1453	1119	334
2	(1.5)	3-Phase	1750	79	1887	1491	396
3	(2.2)	3-Phase	1750	81	2763	2238	525
5	(3.7)	3-Phase	1750	82	4541	3721	817
7.5	(5.6)	3-Phase	1750	84	6651	5596	1066
10	(7.5)	3-Phase	1750	85	8760	7178	1315
15	(11.2)	3-Phase	1750	86	13 009	11 192	1820
20	(14.9)	3-Phase	1750	87	17 140	14 913	2230
25	(18.6)	3-Phase	1750	88	21 184	18 635	2545
30	(22.4)	3-Phase	1750	89	25 110	22 370	2765
40	(30)	3-Phase	1750	89	33 401	29 885	3690
50	(37)	3-Phase	1750	89	41 900	37 210	4600
60	(45)	3-Phase	1750	89	50 395	44 829	5538
75	(56)	3-Phase	1750	90	62 115	55 962	6210
100	(75)	3-Phase	1750	90	82 918	74 719	8290
125	(93)	3-Phase	1750	90	103 430	93 172	10 342
150	(110)	3-Phase	1750	91	123 060	111 925	11 075
200	(150)	3-Phase	1750	91	163 785	149 135	14 738
250	(190)	3-Phase	1750	91	204 805	186 346	18 430

Table 7-21A SI Recommended Rates of Radiant and Convective Heat Gain from Unhooded Electric Appliances During Idle (Ready-to-Cook) Conditions

(Table 5A, Chapter 18, 2017 ASHRAE Handbook—Fundamentals [SI])

	Energy Rate, W		Rate of Heat Gain, W					
Appliance	**Rated**	**Standby**	**Sensible Radiant**	**Sensible Convective**	**Latent**	**Total**	**Usage Factor F_U**	**Radiation Factor F_R**
Cabinet: hot serving (large), insulated[a]	1993	352	117	234	0	352	0.18	0.33
hot serving (large), uninsulated	1993	1026	205	821	0	1026	0.51	0.20
proofing (large)[a]	5099	410	352	0	59	410	0.08	0.86
proofing (small-15 shelf)	4191	1143	0	264	879	1143	0.27	0.00
Cheesemelter[b]	2400	976	443	533	0	976	0.41	0.45
Coffee brewing urn	3810	352	59	88	205	352	0.08	0.17
Drawer warmers, 2-drawer (moist holding)[a]	1202	147	0	0	59	59	0.12	0.00
Egg cooker[b]	2380	249	65	184	0	249	0.10	0.26
Espresso machine[a]	2403	352	117	234	0	352	0.15	0.33
Food warmer: steam table (2-well-type)	1495	1026	88	176	762	1026	0.69	0.08
Freezer (small)	791	322	147	176	0	322	0.41	0.45
Fryer, countertop, open deep fat[b]	4600	431	202	229	0	431	0.09	0.47
Griddle, countertop[b]	8000	1771	848	923	0	1771	0.22	0.48
Hot dog roller[b]	1600	1240	267	973	0	1240	0.77	0.22
Hot plate: single element, high speed[b]	1100	982	314	668	0	982	0.89	0.32
Hot-food case (dry holding)[a]	9115	733	264	469	0	733	0.08	0.36
Hot-food case (moist holding)[a]	9115	967	264	528	176	967	0.11	0.27
Induction hob, countertop[b]	5000	0	0	0	0	0	0.00	0.00
Microwave oven: commercial[b]	1700	0	0	0	0	0	0	0.00
Oven: countertop conveyorized bake/finishing[b]	5000	3932	718	3214	0	3932	0.79	0.18
Panini[b]	1800	673	195	478	0	673	0.37	0.29
Popcorn popper[b]	850	115	28	87	0	115	0.14	0.24
Rapid-cook oven (quartz-halogen)[a]	12 016	0	0	0	0	0	0	0.00
Rapid-cook oven (microwave/convection)[b]	5700	1141	96	1045	0	1141	0.20	0.08
Reach-in refrigerator[a]	1407	352	88	264	0	352	0.25	0.25
Refrigerated prep table[a]	586	264	176	88	0	264	0.45	0.67
Rice cooker[b]	1550	82	14	68	0	82	0.05	0.17
Soup warmer[b]	800	390	0	53	337	390	0.49	0.00
Steamer (bun)[b]	1500	200	32	168	0	200	0.13	0.16
Steamer, countertop[b]	8300	344	0	248	96	344	0.04	0.00
Toaster: 4-slice pop up (large): cooking	1788	879	59	410	293	762	0.49	0.07
contact (vertical)[b]	2600	759	180	579	0	759	0.29	0.24
conveyor (large)	9613	3019	879	2139	0	3019	0.31	0.29
small conveyor[b]	1745	1702	358	1344	0	1702	0.98	0.21
Tortilla grill[b]	2200	1034	254	780	0	1034	0.47	0.25
Waffle iron[b]	2700	267	60	207	0	267	0.10	0.22

Sources: Swierczyna et al. (2008, 2009), with the following exceptions as noted.

[a] Swierczyna et al. (2009) only.

[b] Additions and updates from ASHRAE research project RP-1631 (Kong and Zhang 2016; Zhang et al. 2016).

Table 7-21C SI Recommended Rates of Radiant Heat Gain from Hooded Electric Appliances During Idle (Ready-to-Cook) Conditions

(Table 5C, Chapter 18, 2017 ASHRAE Handbook—Fundamentals [SI])

Appliance	Energy Rate, W		Rate of Heat Gain, W	Usage Factor F_U	Radiation Factor F_R
	Rated	Standby	Sensible Radiant		
Broiler: underfired 900 mm	10 814	9056	3165	0.84	0.35
Cheesemelter*	3605	3488	1348	0.97	0.39
Fryer: kettle	29 014	528	147	0.02	0.28
Fryer: open deep-fat, 1-vat	14 008	821	293	0.06	0.36
Fryer: pressure	13 511	791	147	0.06	0.19
Griddle: double sided 900 mm (clamshell down)*	21 218	2022	410	0.10	0.20
Griddle: double sided 900 mm (clamshell up)*	21 218	3370	1055	0.16	0.31
Griddle: flat 900 mm	17 115	3370	1319	0.20	0.39
Griddle-small 900 mm*	8997	1788	791	0.20	0.44
Induction cooktop*	21 013	0	0	0.00	0.00
Induction wok*	3488	0	0	0.00	0.00
Oven: combi: combi-mode*	16 411	1612	234	0.10	0.15
Oven: combi: convection mode	16 412	1612	410	0.10	0.25
Oven: convection full-size	12 103	1964	440	0.16	0.22
Oven: convection half-size*	5510	1084	147	0.20	0.14
Pasta cooker*	22 010	2491	0	0.11	0.00
Range top: top off/oven on*	4865	1172	293	0.24	0.25
Range top: 3 elements on/oven off	15 005	4513	1846	0.30	0.41
Range top: 6 elements on/oven off	15 005	9730	4074	0.65	0.42
Range top: 6 elements on/oven on	19 870	10 668	4250	0.54	0.40
Range: hot-top	15 826	15 035	3458	0.95	0.23
Rotisserie*	11 107	4044	1319	0.36	0.33
Salamander*	7004	6829	2051	0.97	0.30
Steam kettle: large (225 L), simmer lid down*	32 414	762	29	0.02	0.04
Steam kettle: small (150 L), simmer lid down*	21 599	528	88	0.02	0.17
Steamer: compartment: atmospheric*	9789	4484	59	0.46	0.01
Tilting skillet/braising pan	9642	1553	0	0.16	0.00

* Items with an asterisk appear only in Swierczyna et al. (2009); all others appear in both Swierczyna et al. (2008) and (2009).

Table 7-21D SI Recommended Rates of Radiant Heat Gain from Hooded Gas Appliances During Idle (Ready-to-Cook) Conditions

(Table 5D, Chapter 18, 2017 ASHRAE Handbook—Fundamentals [SI])

Appliance	Energy Rate, W		Rate of Heat Gain, W	Usage Factor F_U	Radiation Factor F_R
	Rated	Standby	Sensible Radiant		
Broiler: batch*	27 842	20 280	2374	0.73	0.12
Broiler: chain (conveyor)	38 685	28 340	3869	0.73	0.14
Broiler: overfired (upright)*	29 307	25 761	733	0.88	0.03
Broiler: underfired 900 mm	28 135	21 658	2638	0.77	0.12
Fryer: doughnut	12 895	3634	850	0.28	0.23
Fryer: open deep-fat, 1 vat	23 446	1377	322	0.06	0.23
Fryer: pressure	23 446	2638	234	0.11	0.09
Griddle: double sided 900 mm (clamshell down)*	31 710	2345	528	0.07	0.23
Griddle: double sided 900 mm (clamshell up)*	31 710	4308	1436	0.14	0.33
Griddle: flat 900 mm	26 376	5979	1084	0.23	0.18
Oven: combi: combi-mode*	22 185	1758	117	0.08	0.07
Oven: combi: convection mode	22 185	1700	293	0.08	0.17
Oven: convection full-size	12 895	3488	293	0.27	0.08
Oven: conveyor (pizza)	49 822	20 017	2286	0.40	0.11
Oven: deck	30 772	6008	1026	0.20	0.17
Oven: rack mini-rotating*	16 500	1319	322	0.08	0.24
Pasta cooker*	23 446	6946	0	0.30	0.00
Range top: top off/oven on*	7327	2169	586	0.30	0.27
Range top: 3 burners on/oven off	35 169	17 614	2081	0.50	0.12
Range top: 6 burners on/oven off	35 169	35 403	3370	1.01	0.10
Range top: 6 burners on/oven on	42 495	36 018	3986	0.85	0.11
Range: wok*	29 014	25 614	1524	0.88	0.06
Rethermalizer*	26 376	6829	3370	0.26	0.49
Rice cooker*	10 257	147	88	0.01	0.60
Salamander*	10 257	9759	1553	0.95	0.16
Steam kettle: large (225 L) simmer lid down*	42 495	1583	0	0.04	0.00
Steam kettle: small (38 L) simmer lid down*	15 240	967	88	0.06	0.09
Steam kettle: small (150 L) simmer lid down	29 307	1260	0	0.04	0.00
Steamer: compartment: atmospheric*	7620	2432	0	0.32	0.00
Tilting skillet/braising pan	30 479	3048	117	0.10	0.04

*Items with an asterisk appear only in Swierczyna et al. (2009); all others appear in both Swierczyna et al. (2008) and (2009).

Table 7-21E SI Recommended Rates of Radiant Heat Gain from Hooded Solid Fuel Appliances During Idle (Ready-to-Cook) Conditions

(Table 5E, Chapter 18, 2017 ASHRAE Handbook—Fundamentals [SI])

	Energy Rate,	Rate of Heat Gain,			
Appliance	**Rated**	**Standby**	**Sensible**	**Usage Factor F_U**	**Radiation Factor F_R**
Broiler: solid fuel: charcoal				N/A	0.15
Broiler: solid fuel: wood (mesquite)*				N/A	0.14

*Items with an asterisk appear only in Swierczyna et al. (2009); all others appear in both Swierczyna et al. (2008) and (2009).

Table 7-21F SI Recommended Rates of Radiant and Convective Heat Gain from Warewashing Equipment During Idle (Standby) or Washing Conditions

(Table 5F, Chapter 18, 2017 ASHRAE Handbook—Fundamentals [SI])

	Energy Rate, W		Rate of Heat Gain, W						
			Unhooded				Hooded		
Appliance	**Rated**	**Standby/ Washing**	**Sensible Radiant**	**Sensible Convective**	**Latent**	**Total**	**Sensible Radiant**	**Usage Factor F_U**	**Radiation Factor F_R**
Dishwasher: conveyor type, hot-water sanitizing, washing	13,712	N/A	0	3,545	13,771	17,316	0	N/A	0.00
Standby	13,712	1,670	0	469	1,201	1,670	0	0.12	0.00
Dishwasher: conveyor type, chemical sanitizing, washing	13,712	12,775	0	3,252	10,372	13,624	0	0.93	0.00
Standby	13,712	1,670	0	469	1,201	1,670	0	0.12	0.00
Dishwasher: door type, hot-water sanitizing, washing	17,609	5,420	0	2,227	7,384	9,610	0	0.31	0.00
With heat recovery and vapor reduction	15,207	7,940	0	1,699	3,838	5,538	0	0.52	0.00
Standby	5,391	352	0	668	1,222	1,890	0	0.35	0.00
Dishwasher: door type, chemical sanitizing, washing	8,790	4,571	0	1,143	3,868	5,010	0	0.52	0.00
Standby	5,391	352	0	264	88	352	0	0.07	0.00
Dishwasher: door type, chemical sanitizing, dump and fill, washing	1,787	879	0	850	1,231	2,080	0	0.49	0.00
Standby	1,787	879	0	0	0	0	0	0.49	0.00
Pot and pan washer: door type, hot-water sanitizing, washing	15,587	10,665	0	1,758	6,885	8,643	0	0.68	0.00
With heat recovery and vapor reduction	15,587	10,314	0	1,611	5,567	7,178	0	0.66	0.00
Dishwasher: under-counter type, hot-water sanitizing, washing	8,350	2,227	234	938	2,022	3,194	800	0.27	0.11
With heat recovery and vapor reduction	7,794	6,680	0	586	322	908	0	0.86	0.00
Standby	7,794	498	234	146	117	498	800	0.06	0.47
Dishwasher: under-counter type, chemical sanitizing, washing	8,350	2,022	0	645	1,436	2,080	0	0.24	0.00
Standby	7,794	498	0	146	117	264	0	0.06	0.00
Booster heater	38,090	0	146	0	0	0	500	0	N/A

Sources: PG&E (2010-2016), Swierczyna et al. (2008) and (2009).

Chapter 8

ENERGY ESTIMATING METHODS

This chapter discusses general techniques, as well as several simplified methods, for estimating the energy consumption of heating and cooling systems, including the variable-base degree-day and the bin concepts. Details of the more complete and sophisticated procedures may be found in Chapter 19, Energy Estimating and Modeling Methods, of the 2017 *ASHRAE Handbook—Fundamentals*. Complementary material on *energy* may be found in Chapter 34, Energy Resources, of the 2017 *ASHRAE Handbook—Fundamentals.*

8.1 General Considerations

8.1.1 Energy Resources and Sustainability

Because energy used in buildings and facilities comprises a significant amount of the total energy used for all purposes and, thus, affects energy resources, ASHRAE recognizes the "effect of its technology on the environment and natural resources to protect the welfare of posterity" (ASHRAE 2003). Many governmental agencies regulate energy conservation, often through the procedures to obtain building permits. Required efficiency values for building energy use strongly influence selection of HVAC&R systems and equipment. The HVAC&R industry deals with energy forms as they occur on or arrive at a building site. Generally, these forms are fossil fuels (natural gas, oil, and coal) and electricity. Solar and wind energy are also available at most sites, as in low-level geothermal energy for heat pumps. The term *energy source* refers to on-site energy in the form in which it arrives at or occurs in a site (e.g., electricity, gas, oil, coal). *Energy resource* refers to the raw energy that (1) is extracted from the earth, (2) is used to generate the energy sources delivered to a building site (e.g., coal used to generate electricity), or (3) occurs naturally and is available at a site (solar, wind, or geothermal energy).

The energy requirements and fuel consumption of building HVAC systems have a direct impact on the cost of operating a building and an indirect impact on the environment. This chapter takes an introductory look at methods for estimating energy use as a guide in design, for standards/code compliance, and for economic optimization. These energy estimating methods can provide quantitative energy and cost comparisons among design alternatives. A primary objective of building energy analysis is economic—to determine which of the available options has the lowest total (life-cycle) cost. Several building energy codes and standards allow the use of an energy analysis to demonstrate compliance with the energy performance goals in the code. In fact, this use of energy analysis programs may be more prevalent than actual comparative energy studies. The large number of uncontrolled and/or unknown factors related to actual building use and HVAC system control generally preclude the use of such methods for determining absolute energy consumption and should not be used to predict future energy bills.

8.1.2 Energy Estimating Techniques

Although the procedures used for estimating energy requirements vary widely in their degree of complexity, they share three elements. These elements are the calculation of (1) space load, (2) secondary equipment load, and (3) primary equipment energy requirements. *Secondary* refers to equipment that distributes the heating, cooling, or ventilating medium to the conditioned spaces; *primary* refers to the central plant equipment that converts fuel or electric energy for heating or cooling.

The **first step** in calculating energy requirements is to determine the **space load**, which is the amount of energy that must be added to or extracted from a space to maintain thermal comfort. The simplest procedures assume that the energy required to maintain comfort is a function of a single parameter—the outdoor dry-bulb temperature. More accurate methods consider solar effects, internal gains, heat storage in the walls and interiors, and the effects of wind on both the building envelope heat transfer and infiltration. The most sophisticated procedures are based on hourly profiles for climatic conditions and operational characteristics for a number of typical days of the year or a full 8760 hours of operation.

The **second step** translates the space load into a load on the **secondary system** equipment. This step must include calculation of all forms of energy required by the secondary system, which may include electrical energy to operate fans and/or pumps, as well as energy in the form of heated or chilled water.

The **third step** calculates the fuel and energy required by the **primary equipment** to meet these loads. It considers efficiencies and part-load characteristics of the equipment. Often, the different forms of energy, such as electricity, natural gas, or oil, must be tracked. In some cases where calculations are done to ensure compliance with codes or standards, these energies must be converted to the energy source or resource consumed, as opposed to that delivered to the building boundary.

Energy calculations often lead to an economic analysis that aims to establish the cost-effectiveness of conservation

measures. Thus, a thorough energy analysis provides intermediate data, such as time of energy usage and maximum demand, so that utility charges can be accurately estimated. Although not part of the energy calculations, estimated capital equipment costs should be included in the complete analysis.

The sophistication of the calculation procedure used can often be inferred from the number of separate ambient conditions and/or time increments used in the calculations. A simple procedure may use only one measure, such as annual degree-days, and is appropriate only for simple systems and applications. Such methods are called *single-method measures*. Improved accuracy may be obtained by using more information, such as the number of hours anticipated under particular conditions of operation. These methods, of which the *bin method* is the most well known, are called *simplified multiple-measure methods*. The most elaborate methods perform energy balance calculations at each hour over some period of analysis, typically one year. These *detailed simulation methods* require hourly weather data, as well as hourly estimates of internal loads such as lighting and occupants.

Because systems that consume energy in buildings are nonlinear, dynamic, and very complex, few methods other than **computer modeling** are available for accurately calculating energy consumption. The most accurate methods for calculating building energy consumption are the most costly because of their intense computational requirements and high degree of user expertise. However, the cost of the computer facilities and the software itself are typically a small fraction of the total cost of running a building energy analysis. The major costs are for learning to use the program and for the time involved in using it.

The US Department of Energy maintains an up-to-date listing of building energy software with links to other sites that describe energy modeling and load estimating tools at http://www.energytoolsdirectory.gov. A number of these software programs are available for downloading without cost.

8.2 Component Modeling and Loads

8.2.1 Loads

After peak loads have been evaluated, equipment with capacity sufficient to offset these loads must be selected. Air supplied to the space must be at the proper conditions to satisfy both the sensible and latent loads. However, peak load occurs but a few times each year, whereas partial load operation exists most of the time. With operation predominately at part load, partial load analysis for energy use and fuel cost is often as important as the sizing procedure.

Calculating instantaneous space load is a key step in any building energy simulation. The **heat balance method** and the **weighing factor method** are two methods used for these calculations. The weighing factor method and the heat balance method use conduction transfer functions (or their equivalents) to calculate transmission heat gain or loss. The principal difference is in the methods used to calculate the subsequent internal heat transfers to the room. Experience with both methods has indicated largely the same results, provided the weighing factors are determined for the specific building under analysis.

8.2.2 Secondary System Components

Secondary HVAC systems generally include all elements of the overall building energy system between a central heating and cooling plant and the building zones. The precise definition depends heavily on the building design. A secondary system typically includes air-handling equipment, air distribution systems with the associated ductwork, dampers, fans, and heating, cooling, and humidity conditioning equipment. Secondary systems also include the liquid distribution systems between the central plant and the zone and air-handling equipment, including piping, valves, and pumps.

To the extent that the secondary system consumes energy and transfers energy between the building and central plant, energy analysis can be performed by characterizing the energy consumption of the individual components and the energy transferred among system components. In fact, few of the secondary components consume energy directly, except for fans, pumps, furnaces, direct-expansion air-conditioning package units with gas-fired heaters, and inline heaters. Secondary components are divided into two categories: distribution components and heat and mass transfer components.

The distribution system of an HVAC system affects energy consumption in two ways. First fans and pumps consume electrical energy directly, based on the flow and pressures under which the device operates. Ducts and dampers, or pipes and valves, and the system control strategies affect the flow and pressures at the fan. Second, thermal energy is often transferred to (or from) the fluid due to heat transfer through pipes and ducts and due to the electrical input to fans and pumps. The analysis of system components should, therefore, account for both direct electrical energy consumption and thermal energy transfer.

Strictly speaking, performance calculations of the fan and air distribution systems in a building require a detailed pressure balance on the entire network.

While a detailed analysis of a distribution system requires flow and pressure balancing among the components, nearly all commercially available methods of energy analysis approximate the effect of the interactions with part-load performance curves. This procedure eliminates the need to calculate pressure drop through the distribution system at off-design conditions. The exact shape of the part-load curve depends on the effect of flow control on the pressure and fan efficiency and may be calculated using a detailed analysis.

8.2.3 Primary System Components

Primary HVAC systems consume energy and deliver heating and cooling to a building, usually through secondary systems. Primary equipment generally includes chillers, boilers, cooling towers, cogeneration equipment, and plant-level thermal storage equipment. In particular, primary equipment

generally represents the major energy-consuming equipment of a building, so accurate characterization of building energy use relies on accurate modeling of primary equipment energy consumption.

The energy consumption characteristics of primary equipment generally depend on equipment design, load conditions, environmental conditions, and equipment control strategies. For example, chiller performance depends on the basic equipment design features (e.g., heat exchange surfaces, compressor design), the temperatures and flow through the condenser and evaporator, and the methods for controlling the chiller at different loads and operating conditions (e.g., inlet guide vane control on centrifugal chillers to maintain leaving chilled water temperature setpoint). In general, these variables that dictate energy consumption vary constantly and require calculations on an hourly basis.

While many secondary components are readily described by fundamental engineering principles (e.g., heat exchangers, valves), the complex nature of most primary equipment has discouraged the use of first-principle models for energy calculations. Instead, the energy consumption characteristics of primary equipment have traditionally been modeled using simple equations developed by running regression analyses on manufacturers' published design data. Because published data are generally only available for full-load design conditions, additional correction functions are used to correct the full-load data to part-load conditions. The functional form of the regression equations and correction functions takes many forms, including exponentials, Fourier series, and, most of the time, second- or third-order polynomials. The selection of an appropriate functional form depends on the behavior of the equipment. In some cases, energy consumption is calculated using direct interpolation from tables of data. However, this method often requires excessive data input and computer memory.

8.3 Overall Modeling Strategies

In developing a simulation model for building energy prediction, two basic issues must be considered—modeling of components or subsystems and the overall modeling strategy. Modeling of components results in sets of equations describing the individual components. Overall modeling strategy refers to the *sequence* and *procedures* used to solve these equations. The accuracy of results and the computer resources required to achieve these results depend on the modeling strategy.

In most building energy programs, the load models are executed for every space for every hour of the simulation period. (Practically all models use one hour as the time-step, which excludes any information on phenomena occurring in a shorter time span.) The load model is followed by running models for every secondary system, one at a time, for every hour of the simulation. Finally, the plant simulation model is executed again for the entire period. Each sequential execution processes the *fixed* output of the preceding step.

This procedure is illustrated in Fig. 8-1. The solid lines represent data passed from one model to the next. The dashed lines represent information, usually provided by the user, about one model to the preceding model.

Because of this loads-systems-plants sequence, certain phenomena cannot be modeled precisely. For example, if the heat balance method for computing loads is used and some component in the system simulation model cannot meet the load, the program can only report the current load. In actuality, the space temperature should readjust until the load matches the equipment capacity, but this cannot be modeled because the loads have been precalculated and fixed. If the weighting factor method is used for loads, this problem is partially overcome because loads are continually readjusted during the system simulation. However, the weighting factor technique is based on linear mathematics, and wide departures of room temperatures from those used during execution of the load program can introduce errors.

A similar problem arises in plant simulation. For example, in an actual building, as the load on the central plant varies the supply, chilled water temperature also varies. This variation, in turn, affects the capacity of the secondary system equipment. In an actual building, when the central plant becomes overloaded, space temperatures should rise to reduce the load. However, in most energy-estimating programs, this condition cannot occur; thus, only the overload condition can be reported. These are some of the penalties associated with decoupling of the load, system, and plant models.

An alternative strategy, in which all calculations are performed at each time step, is conceivable. Here the load, system, and plant equations are solved simultaneously at each time interval. With this strategy, unmet loads and imbalances cannot occur; conditions at the plant are immediately reflected to the secondary system and then to the load model, forcing them to readjust to the instantaneous conditions throughout the building. The results of this modeling strategy are superior to those currently available, although the magnitude and importance of the improvement are uncertain.

The principal disadvantage of the alternative approach, and the reason that it has not been widely used, is that it demands more computing resources.

An economic model, as included in Fig. 8-1, calculates energy costs (and sometimes capital costs) based on the estimated required input energy. Thus, the simulation model calculates energy usage and cost for any given input weather

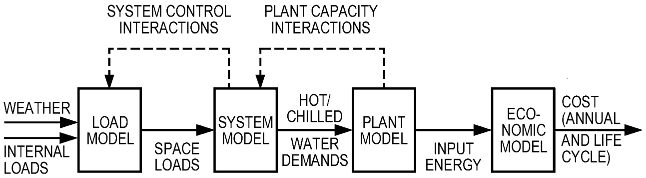

Fig. 8-1 Overall Modeling Strategy

(Figure 1, Chapter 19, 2017 ASHRAE Handbook—Fundamentals)

and internal loads. By applying this model (i.e., determining output for given inputs) at each hour (or other suitable interval), the hour-by-hour energy consumption and cost can be determined. Maintaining running sums of these quantities yields monthly or annual energy usage and costs.

These models only compare design alternatives; a large number of uncontrolled and unknown factors usually rules out such models for accurate prediction of utility bills.

Traditionally, most energy analysis programs include a set of preprogrammed models that represent various systems, such as variable air volume, terminal reheat, multizone, etc. In this scheme, the equations for each system are arranged so they can be solved sequentially. If this is not possible, then the smallest number of equations that must be solved simultaneously is solved using an appropriate technique. Furthermore, individual equations may vary from hour to hour in the simulation, depending on controls and operating conditions. For example, a dry coil uses different equations than a wet coil.

The primary disadvantage of this scheme is that it is relatively inflexible—to modify a system, the program source code may have to be modified and recompiled. Alternative strategies have viewed the system as a series of components (e.g. fan, coil, pump, duct, pipe, damper, thermostat) that may be organized in a component library. Users of the program specify the connections between the components. The program then resolves the specification of components and connections into a set of simultaneous equations.

8.4 Integration of System Models

Energy calculations for secondary systems involve construction of the complete system from the set of HVAC components. For example, a VAV system is a single-path system that controls zone temperature by modulating the airflow while maintaining a constant supply air temperature. VAV terminal units, located at each zone, adjust the quantity of air reaching each zone depending on its load requirements. Reheat coils may be included to provide required heating for perimeter zones.

This VAV system simulation consists of a central air-handling unit and a VAV terminal unit with reheat coil located at each zone, as shown in Fig. 8-2. The central air-handling unit includes a tan cooling coil, preheat coil, and an outdoor air economizer. The supply air leaving the air-handling unit is controlled to a fixed setpoint. The VAV terminal unit at each zone varies the airflow to meet the cooling load. As the zone cooling load decreases, the VAV terminal unit decreases the zone airflow until the unit reaches its minimum position. If the cooling load continues to decrease, the reheat coil will be activated to meet the zone load. As the supply air volume leaving the unit decreases, the fan power consumption will also be reduced. A variable-speed drive is used to control the supply fan.

The simulation is based on system characteristics and zone design requirements. For each zone, the inputs include the sensible and latent loads, the zone setpoint temperature, and the minimum zone supply air mass flow. System characteristics include the supply air temperature setpoint, the entering water temperature of the reheat, preheat, and cooling coils, the minimum mass flow of outdoor air, and the economizer temperature/enthalpy setpoint for minimum airflow.

The algorithm for performing the calculations for this VAV system is shown in Fig. 8-3. The algorithm directs sequential calculations of system performance. Calculations proceed from the zones forward along the return air path to the cooling coil inlet and back through the supply air path to the cooling oil discharge.

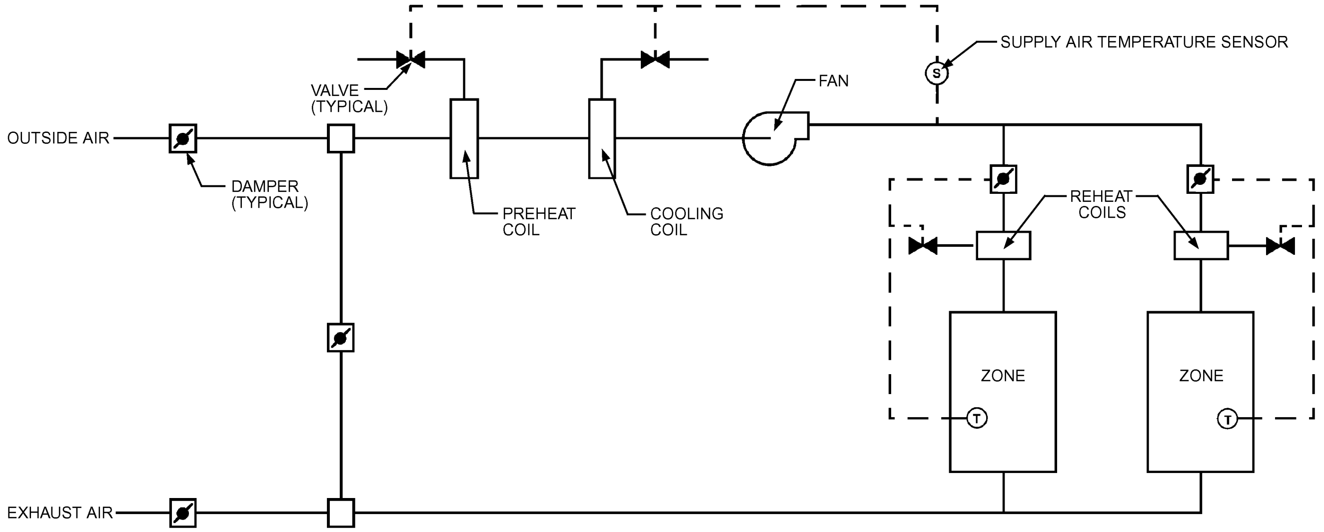

Fig. 8-2 Schematic of Variable Air Volume System with Reheat
(Figure 11, Chapter 19, 2013 ASHRAE Handbook—Fundamentals)

This basic algorithm for simulation of a VAV system might be used in conjunction with a heat-balance type of load calculation. For a weighting factor approach, it would have to be modified to allow zone temperatures to vary and a consequent readjustment of zone loads. It should also be enhanced to allow for possible limits on reheat temperature and/or cooling coil limits, zone humidity limits, outdoor air control (economizers), and/or heat-recovery devices, zone exhaust, return air fan, heat gain in the return air path because of lights, the presence of baseboard-type heaters, and more realistic control profiles. Most current building energy programs incorporate these and other features as user options, as well as algorithms for other types of systems.

The method chosen to analyze a building's energy use is determined by the purpose for the investigation.

Forward modeling begins with a description of the building system or component of interest and defines the building being modeled according to its physical description. For example, the building geometry, geographical location, type of HVAC system, wall insulation, etc., may be defined. The primary benefits of this method are that it is based on sound engineering principles and it has widespread acceptance and use in major public domain simulation codes (e.g., BLAST, DOE-2, and EnergyPlus). Figure 8-4 is a flow chart that illustrates the ordering of the analysis that is typically performed by a building energy simulation program.

Inverse modeling is based on the empirical behavior of the building as it relates to one or more driving forces. This approach is referred to as system identification, parameter identification, or inverse modeling. In this modeling approach, a structure or physical configuration of the building or system is assumed first and then important parameters are identified by a statistical analysis.

8.5 Degree-Day Methods

Degree-day methods are the simplest methods for energy analysis and are appropriate if the building use and the efficiency of the HVAC equipment are constant. Where efficiency or conditions of use vary with outdoor temperature, the consumption can be calculated for different values of outdoor temperature and multiplied by the corresponding number of hours; this approach is used in various bin methods. When the indoor temperature is allowed to fluctuate or when interior gains vary, models other than simple steady-state models must be used.

Single-measure methods for estimating cooling energy are less established than those for heating, primarily because the indoor-outdoor temperature difference during cooling is typically much smaller than under heating conditions. As a

```
BEGIN LOOP  Calculate zone related design requirements
    •  Calculate required supply airflow to meet zone load
    •  Sum actual zone mass air flow rate
    •  Sum zone latent loads
    IF zone equals last zone THEN Exit Loop
END LOOP
•  Calculate system return air temperature from zone temps
•  Assume an initial cooling coil leaving air humidity ratio

BEGIN LOOPIterate on cooling coil leaving air humidity ratio
    •  Calculate return air humidity ratio from latent loads
    •  Calculate supply fan power consumption and
       entering fan air temperature
    •  Calculate mixed air temperature and humidity
       ratio using an economizer cycle
    IF mixed air temperature is less than design
       supply air temperature THEN
         •  Calculate preheat coil load
    ELSE
         •  Calculate cooling coil load and leaving air
            humidity ratio
    ENDIF
    IF cooling coil leaving air humidity ratio converged
    THEN Exit Loop
END LOOP

BEGIN LOOP Calculate the zone reheat coil loads
    IF zone supply air temperature is greater than system
    design supply air temperature THEN
         •  Calculate reheat coil load
            (Subroutine: COILINV/HCDET)
    ENDIF
    •  Sum reheat coil loads for all zones
    IF zone equals last zone THEN Exit Loop
END LOOP
```

Fig. 8-3 Algorithm for Calculating Performance of VAV with Reheat System

(Figure 12, Chapter 19, 2013 ASHRAE Handbook—Fundamentals)

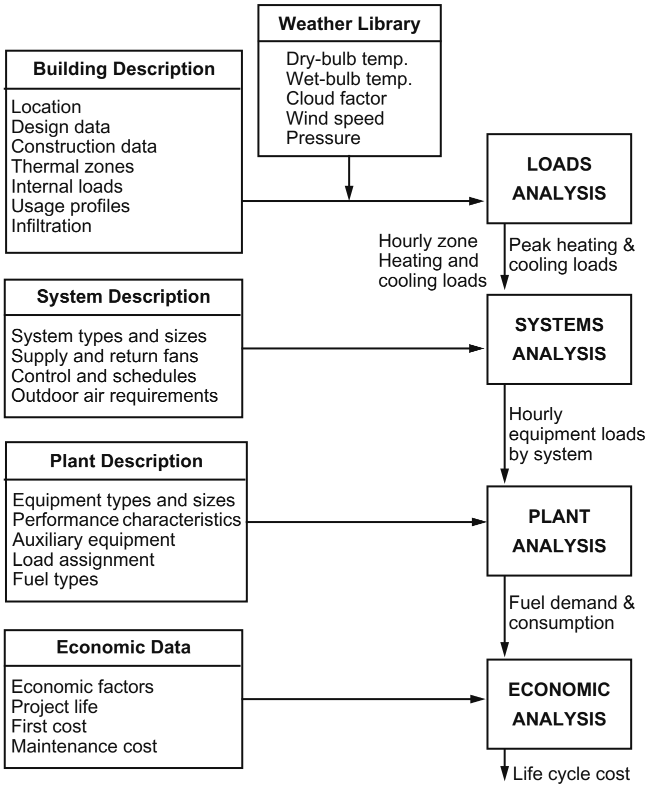

Fig. 8-4 Flow Chart for Building Energy Simulation Program

(Ayres and Stamper 1995)

result, cooling loads depend more on such factors as solar gain and internal loads (besides temperature) than do heating loads. Since these loads largely depend on specific building features, attempts to correlate cooling energy requirements against a single climate parameter have not been very successful. Nonetheless, a companion to the heating degree-day procedure has emerged, the *cooling degree-day method*, and is widely used.

Even in an age when computers can easily calculate the energy consumption of a building, the concepts of degree-days and balance-point temperature remain valuable tools. The severity of a climate can be characterized concisely in terms of degree-days. Also, the degree-day method and its generalizations can provide a simple estimate of annual loads, which can be accurate if the indoor temperature and internal gains are relatively constant and if the heating or cooling systems are to operate for a complete season. Thus, basic steady-state methods continue to be important.

Virtually all the energy-consumption estimating techniques used prior to the last few years are based on an assumed or measured set of average conditions taken from a relatively small sampling of buildings. These techniques use steady-state models based on monthly average weather and operating conditions and, for the degree-day method in particular, assume that energy use depends only on the difference between indoor and outdoor dry-bulb temperature. Thus, use of the degree-day method is restricted to small structures with envelope-dominated heating and cooling loads. Other methods should be used for larger commercial or industrial buildings, where internal, cooling-only zones are prevalent or where cooling loads are not linearly dependent on the outdoor-to-indoor temperature difference.

Any estimating method produces a more reliable result over a long period of operation than over a short period. Nearly all of the methods in use give reasonable results over a full annual heating season, but estimates for shorter periods, such as a month, may produce inaccurate results. As the period of the estimate shortens, there is more chance that some factor not directly taken into account in the estimating method will deviate from its long-term average value and thus lead to an error in the predicted energy requirement.

8.5.1 Balance-Point Temperature and Degree-Days

The *balance-point temperature* is the average outdoor temperature at which the building requires neither heating nor cooling from the HVAC system. The degree-day procedure recognizes that the heating equipment needs to meet only the heating not covered by internal sources such as lights, equipment, occupants, and solar gain. In other words, these sources provide heat down to the balance-point temperature, and below that, energy requirements of the space are proportional to the difference between the balance-point and the outside temperature.

The balance-point temperature for heating t_{bal} of a building is defined as that value of the outdoor temperature t_o at which, for the specified value of the interior temperature t_i, the total heat gain q_{gain} is equal to all space heat gains from sun, occupants, lights, and so forth.

The steady-state heating requirements of a building q_h can be expressed as

$$q_h = K(t_i - t_o) - q_{\text{gain}} \tag{8-1}$$

where

K = building loss coefficient (design heat loss divided by the design temperature difference)
t_i = indoor temperature
t_o = outdoor temperature
q_{gain} = all space heat gains not attributed to the space-conditioning system

The balance temperature is determined by setting $q_h = 0$ and solving for $t_o = t_{\text{bal}}$, so

$$t_{\text{bal}} = t_i - q_{\text{gain}}/K \tag{8-2}$$

Heating is needed only when t_o drops below t_{bal}.

With $t_{o,\text{av}}$ being the daily average value of the outdoor temperature for a particular day, the heating degree-days $\text{DD}_h(t_{\text{bal}})$ for that day are obtained as $(t_{\text{bal}} - t_{o,\text{av}})$ and for any particular period as

$$\text{DD}_h(t_{\text{bal}}) = (1\ \text{day})\Sigma(t_{\text{bal}} - t_{o,\,\text{av}}) \tag{8-3}$$

where the summation may extend over a month, the heating season, or the entire year. In connection with degree-days, the balance temperature is also known as the base of the degree-days.

Cooling degree-days can be calculated using an expression analogous to heating degree-days:

$$\text{DD}_c(t_{\text{bal}}) = (1\ \text{day})\Sigma(t_o - t_{\text{bal}}) \tag{8-4}$$

While the definition of the balance-point temperature is the same as that for heating, in a given building its numerical value for cooling is generally different from that for heating because q_{gain}, K, and/or t_i can be different.

8.5.2 Seasonal Efficiency

The seasonal efficiency of heating equipment η depends on such factors as steady-state efficiency, sizing, cycling effects, and energy conservation devices installed. Sometimes it is much lower than and other times it is comparable to steady-state efficiency. Expressions have been developed from information supplied by the National Institute of Standards and Technology that estimate the seasonal efficiency for a variety of furnaces if information on rated input and output is available. For the case where heat loss from the ducting is neglected, the series of equations is

$$\eta = \eta_{\text{si}}\text{CF}_{\text{pl}} \tag{8-5}$$

where η_{ss} is the steady-state efficiency (rated output/input).

The term CF_{pl} is a trait of the part-load efficiency of the heating equipment, which may be calculated as follows:

Gas-forced air furnaces

With pilot:

$$CF_{pl} = 0.6328 + 0.5738RLC - 0.3323(RLC)^2$$

With intermittent ignition:

$$CF_{pl} = 0.7791 + 0.1983RLC - 0.0711(RLC)^2$$

Oil furnaces without stack damper

$$CF_{pl} = 0.7092 + 0.6515RLC - 0.4711(RLC)^2$$

Resistance electric furnaces

$$CF_{pl} = 1.0$$

RLC is defined as follows:

$$RLC = \frac{K(t_{bal} - t_{od})}{CHT}$$

where

K = building loss coefficient

t_{od} = outside design temperature

CHT = rated output of equipment

8.5.3 Heating Degree-Day Method

The degree-day procedure for estimating heating energy requirements is based on the assumption that, on a long-term average, solar and internal gains offset heat loss when the mean daily outdoor temperature is equal to the balance-point temperature and that fuel consumption is proportional to the difference between the mean daily temperature and the balance-point temperature. In other words, if the balance-point temperature is 60°F, then on a day when the mean temperature is 50°F (10°F below 60°F), 10 times as much fuel is consumed as on days when the mean temperature is 59°F (1°F below 60°F). This basic concept can be represented in an equation stating that energy consumption is directly proportional to the number of degree-days in the estimation period with the heat loss per degree difference being constant.

Estimating the theoretical seasonal energy requirement of a conventional heating system (gas furnace, oil furnace, electric furnace, and so forth) using the degree-day method is simple because the efficiency of the system is assumed to be constant regardless of outdoor temperature. The theoretical heating requirement is calculated as

$$\frac{\text{Design heat loss (Btu/h)} \times 24 \times \text{degree-days}}{\text{Design temperature difference}}$$

The general form of the degree-day equation is

$$E = \frac{q_L(DD)24}{\eta(HV)\Delta t} C_D \tag{8-6}$$

where

E = fuel or energy consumption for the estimate period, units of fuel

q_L = design heat loss, including infiltration and ventilation, Btu/h

DD = number of degree-days for the estimate period

Δt = design temperature difference, °F

η = efficiency of heating system [Eq. (8-5)], also designated on an annual basis as the annual fuel utilization efficiency (AFUE)

HV = heating value of fuel, Btu/unit of fuel

C_D = 1 if DD are based on t_{bal}, or

C_D = 0.77 if 65°F is arbitrarily assumed as the balance temperature

Typical heating values HV are given in Table 8-1; additional values for η are provided in Chapter 19.

Heating degree-days or degree-hours for a balance-point temperature of 65°F have been widely tabulated based on the observation that this has represented average conditions in typical buildings in the **past**. Today, use of the 65°F base will considerably overestimate the energy consumption due to improved building construction as well as increased internal loads in recent years. The errors inherent in the established base of 65°F (18.3°C) may be adjusted by using an empirical correction factor C_D. The best single value for C_D is taken from the DOE HSPF methodology, which uses the correction factor 0.77 as the multiplier for the design heat loss per degree to provide a more appropriate building loss coefficient K to account for internal heat gains in energy estimation. However, the best approach is to avoid the arbitrary 65°F base, using the variable-base degree-day approach instead.

Both heating degree days and cooling degree days can be found to several base temperatures in Figure 4-4 for many locations in the US. The CD-ROM of the 2017 *ASHRAE Handbook—Fundamentals* contains this weather data for many other locations.

Example 8-1 A residence located in St. Joseph, Missouri, has a design heating load of 84,000 Btu/h. Determine

(a) Gallons of No. 2 fuel oil used for heating season with warm air system

(b) Electrical energy used for heating if electric baseboard units are used

Assume the indoor design temperature is 72°F and outdoor design temperature is 10.2°F.

Table 8-1 Heating Values

Fuel	Heating Value of Fuel
Natural gas	1050 Btu/ft^3 (39 MJ/m^3)
Propane	90,000 Btu/gal (25 GJ/m^3)
No. 2 fuel oil	140,000 Btu/gal (39 GJ/m^3)

Solution:

From Equation (8-6) [with $\eta = 1$ and HV = 1],

$$E = \frac{84{,}000 \times 5435 \times 24}{[72 - (10.2)]}0.77$$
$$= 136{,}500{,}000 \text{ Btu/h}$$

$$\text{(a) No. 2 fuel oil} = \frac{136.5 \times 10^6}{140{,}000 \times 0.75} \text{ (assumed)}$$
$$= 1300 \text{ gallons}$$

$$\text{(b) Heating electrical energy} = \frac{136.5 \times 10^6}{3413} = 40{,}000 \text{ kWh}$$

8.5.4 Variable-Base Heating Degree-Day

The variable-base degree-day method (VBDD) is a generalization of the degree-day method. It retains the original degree-day concept but counts degree-days based on the actual balance-point temperature rather than the now outdated 65°F.

The degree-day method, like any steady-state method, is unreliable for estimating the consumption during mild weather. Despite such problems, the degree-day method (using an appropriate base temperature) can give remarkably accurate results for the annual heating energy of single-zone buildings dominated by gains through the walls and roof and/or ventilation.

Table 8-2 provides the degree-days in °F to several bases for various locations in the United States. Annual degree-day tabulations for seven base temperatures in °C at 1195 stations in Canada are in Canadian Climate Normals (Atmospheric Environmental Service 1981).

Heating and cooling degree-day summary data for over 4000 US stations are available online at http://www.5.ncdc.noaa.gov/climatenormals/CLIM81_Sup_02.pdf (NCDC 2002a, 2002b). This publication presents annual heating degree-day normals to the following bases (°F): 65, 60, 57, 55, 50, 45, and 40; and annual cooling degree-day normals to the following bases (°F): 70, 65, 60, 57, 55, 50, and 45.

8.5.5 Cooling Degree-Day Method

Cooling degree-days are also available to several base temperatures as provided in Table 8-2.

Cooling energy is predicted in a similar fashion as described for heating earlier:

$$E_C = \frac{q_g(\text{CDD})24}{1000(\text{SEER})\Delta t_d} \tag{8-7}$$

where

E_C = energy consumed for cooling, kWh
q_g = design cooling load, Btu/h (kW)
CDD = cooling degree-days
Δt_d = cooling design temperature difference, °F (°C)
SEER = seasonal energy efficiency ratio, (Btu/h)/W (W/W)

The variable-base cooling degree-day procedure for estimating cooling energy requirements is similar to the procedure for heating. If no cooling is done by ventilation, the balance temperature t_{bal} is calculated in the same way as the balance temperature for heating, except that summer values for cooling set temperature t_i, total heat gains q_g, and loss coefficient K must be used. The only significant difference from heating in the calculation procedure is that q_g customarily includes a latent term q_l for cooling.

The degree-day method assumes that t_{bal} is constant, which is not well satisfied in practice. Solar gains are zero at night, and internal gains tend to be highest during the evening. Also, during the intermediate season, heat gains can be eliminated, and the onset of mechanical cooling can be postponed by opening windows or increasing the ventilation. Therefore, cooling degree-hours can be used to better represent the period when equipment is operating than cooling degree-days because degree-days assume uninterrupted equipment operation as long as there is a cooling load.

Example 8-2 For a residence located in El Paso, Texas, the design cooling load is 10.7 kW (36,500 Btu/h). Determine:

(a) Annual energy requirements for cooling
(b) Cost of this energy if a unit having an SEER of 11.5 is selected and the electric rate is 6.9¢/kWh

Solution:

Table 8-2: Cooling degree-days = 2098
Summer design = 98.3°F
(a) Using cooling degree method,

$$\text{Cooling} = [36{,}500/(98.3 - 78)] \times 2098 \times 24$$
$$= 90{,}534{,}000 \text{ Btu}$$
$$E_C = (90{,}534{,}000)/(1000 \times 11.5) = 7872 \text{ kWh}$$

$$\text{(b) Cost} = 7872 \text{ kWh} \times \$0.069/\text{kWh} = \$543$$

Example 8-3 A small dental office, 30 ft by 100 ft, located in downtown Chicago, Illinois, has design heating and cooling loads of 240,000 Btu/h and 85,000 Btu/h, respectively. Design conditions of 72°F inside and −3°F outside were used for winter while 78°F inside and 91°F outside were used for summer. The average interior heat gain during the winter has been estimated at 8 kW. Estimate the annual heating and cooling energy requirements and the corresponding energy costs using the national average rates given in the table below, if:

(a) Electric baseboard heating units and a "high-efficiency" air conditioner (SEER = 14) are used
(b) A condensing gas furnace and the same air conditioner are used

Unit Fuel Prices

Natural gas	\$0.62/therm
No. 2 fuel oil	\$1.06/gallon
Electricity	\$0.072/kWh

Solution:

The heat loss per °F, $K = 240{,}000/[72 - (-3)] = 3200$

The balance temperature with an internal gain of 8 kW is determined from Eq. 8-2 as

$$t_{\text{bal}} = 72 - (8 \times 3413)/3200$$
$$= 72 - 8.5 = 63.5°\text{F}$$

Table 8-2 Degree-Days to Several Bases (in °F) for Various US Locations

State	Heating					Cooling				
	Base					Base				
	65	60	55	50	45	65	60	55	50	45
ALABAMA										
Birmingham	2844	1995	1333	838	488	1928	2916	4073	5403	6877
Huntsville	3302	2414	1670	1103	686	1808	2747	3828	5090	6492
Mobile	1684	1062	619	330	148	2577	3780	5162	6698	8342
Montgomery	2269	1508	945	547	282	2238	3302	4568	5991	7551
ALASKA										
Anchorage AP	10911	9122	7492	6081	4896	0	36	224	647	1279
Annette AP	7053	5315	3773	2513	1543	14	98	386	940	1803
Barrow AP	20265	18440	16615	14803	13009	0	0	0	13	44
Barter Island AP	19994	18169	16344	14528	12738	0	0	0	9	44
Bethel AP	13203	11404	9695	8140	6835	0	26	142	411	938
Bettles AP	15925	14180	12548	11060	9718	17	97	289	626	1110
Big Delta AP	13698	11985	10410	8977	7735	34	145	395	787	1370
Cold Bay AP	9865	8040	6230	4532	3095	0	0	16	138	533
Fairbanks AP	14345	12661	11115	9714	8451	52	196	467	898	1468
Gulkana	13938	12162	10507	8985	7648	9	63	228	537	1027
Homer	10364	8539	6745	5133	3840	0	0	24	240	777
Juneau AP	9007	7222	5557	4107	2925	0	39	197	573	1219
King Salmon AP	11582	9773	8047	6563	5304	0	12	112	456	1023
Kodiak	8860	7049	5327	3819	2593	0	7	117	436	1032
Kotzebue AP	16039	14237	12491	10852	9337	0	23	102	268	598
McGrath AP	14487	12736	11107	9634	8348	14	88	284	642	1184
Nome AP	14325	12503	10721	9047	7528	0	0	46	197	503
St. Paul Island AP	11119	9294	7469	5667	4021	0	0	0	24	199
Shemya AP	9735	7910	6085	4298	2693	0	0	0	30	254
Summit FAA AP	14368	12556	10790	9146	7640	0	10	71	253	578
Talkeetna	11708	9934	8306	6848	5609	6	57	254	620	1207
Unalakleet	14027	12238	10515	8943	7565	0	31	138	391	842
Yakutat AP	9533	7711	5942	4420	3181	0	0	56	362	947
ARIZONA										
Flagstaff	7322	5776	4421	3267	2299	140	416	894	1562	2418
Phoenix	1552	899	431	165	45	3508	4680	6039	7596	9297
Prescott FAA AP	4456	3303	2321	1507	883	882	1560	2400	3414	4612
Tucson	1752	1050	541	229	65	2814	3937	5253	6765	8431
Winslow	4733	3623	2683	1882	1249	1203	1921	2802	3828	5018
Yuma	1005	507	211	59	8	4195	5518	7045	8719	10498
ARKANSAS										
Fort Smith	3336	2442	1687	1075	613	2022	2949	4015	5239	6595
Little Rock	3354	2442	1687	1075	624	1925	2843	3908	5128	6496
CALIFORNIA										
Bakersfield	2185	1367	760	371	147	2179	3185	4400	5835	7437
Bishop	4313	3179	2230	1437	848	1037	1728	2603	3641	4875
Blue Canyon	5704	4271	3037	2015	1206	302	698	1283	2079	3106
Daggett FAA AP	2203	1420	824	410	166	2729	3765	5004	6415	7996
Eureka	4679	2925	1494	607	194	0	55	460	1414	2816
Fresno	2650	1724	995	493	205	1671	2563	3667	4986	6525
Long Beach	1606	772	292	70	8	985	1982	3325	4928	6696
Los Angeles Int'l.	1819	833	295	66	7	615	1464	2755	4348	6115
Los Angeles Civic Center	1245	522	158	26	0	1185	2289	3747	5442	7244
Mount Shasta	5890	4458	3215	2177	1338	286	680	1263	2045	3035
Oakland	2909	1570	714	263	61	128	622	1598	2963	4587
Red Bluff	2688	1762	1018	505	208	1904	2803	3895	5196	6727
Sacramento	2843	1837	1043	493	186	1159	1971	3011	4286	5812
Sacramento City	2587	1627	893	406	148	1291	2158	3249	4584	6151
Sandberg	4427	3177	2107	1250	622	800	1374	2123	3100	4293
San Diego	1507	648	213	42	9	722	1694	3084	4746	6532
San Francisco	3042	1668	769	289	67	108	550	1496	2832	4438
San Francisco Fed. Bldg.	3080	1576	608	169	25	39	368	1230	2619	4298
Santa Maria	3053	1624	690	229	42	84	484	1377	2738	4380
Stockton	2806	1835	1072	537	219	1259	2100	3167	4455	5958
COLORADO										
Alamosa	8609	7029	5654	4473	3457	88	329	780	1428	2227
Colorado Springs	6473	5131	3954	2949	2089	461	945	1592	2417	3383
Denver–Stapleton	6016	4723	3601	2653	1852	625	1159	1857	2739	3759
Denver–City	5505	4246	3175	2271	1533	742	1312	2071	2993	4074
Eagle AP	8426	6864	5505	4319	3317	117	385	845	1487	2313
Grand Junction	5605	4441	3425	2551	1814	1140	1810	2619	3565	4653
Pueblo	5394	4221	3220	2351	1628	981	1632	2456	3412	4514
CONNECTICUT										
Bridgeport	5461	4264	3216	2321	1583	735	1362	2140	3064	4152
Hartford	6350	5085	3971	3005	2173	584	1143	1855	2715	3706
DISTRICT OF COLUMBIA										
Washington, D.C.–Dulles AP	5005	3881	2898	2054	1380	940	1636	2474	3458	4616
Washington, D.C.–Nat'l. AP	4211	3182	2293	1563	984	1415	2210	3152	4237	5489
DELAWARE										
Wilmington Ncastle	4940	3824	2839	2003	1330	992	1697	2537	3525	4675
FLORIDA										
Apalachicola	1361	792	426	189	67	2663	3930	5377	6967	8669
Daytona Beach	897	480	215	71	25	2919	4321	5881	7563	9341

Table 8-2 Degree-Days to Several Bases (in °F) for Various US Locations (*Continued*)

	Heating					Cooling				
	Base					Base				
State	65	60	55	50	45	65	60	55	50	45
Fort Myers	457	189	56	9	0	3711	5265	6958	8741	10553
Jacksonville	1327	788	429	194	70	2596	3890	5349	6938	8641
Key West	59	7	0	0	0	4888	6660	8474	10299	12124
Lakeland	678	330	128	40	7	3298	4774	6398	8135	9927
Miami	206	54	8	0	0	4038	5715	7494	9306	11131
Orlando	733	370	151	48	9	3226	4686	6291	8016	9806
Pensacola	1578	991	575	302	135	2695	3932	5341	6893	8551
Tallahassee	1563	996	550	279	116	2563	3792	5200	6755	8415
Tampa	718	364	151	50	14	3366	4836	6447	8172	9963
West Palm Beach	299	88	27	0	0	3786	5408	7159	8960	10785
GEORGIA										
Athens	2975	2084	1370	830	462	1722	2661	3767	5052	6508
Atlanta	3095	2189	1461	911	524	1589	2511	3604	4880	6316
Augusta	2547	1729	1106	652	348	1995	2999	4204	5573	7094
Columbus	2378	1600	1010	593	313	2143	3188	4429	5831	7375
Macon	2240	1492	934	536	271	2294	3373	4643	6068	7626
Rome	3342	2422	1653	1068	637	1615	2514	3576	4816	6210
Savannah	1952	1258	751	413	185	2317	3444	4766	6248	7851
HAWAII										
Hilo	0	0	0	0	0	3066	4887	6712	8537	10362
Honolulu	0	0	0	0	0	4221	6046	7871	9695	11521
Kahului	0	0	0	0	0	3732	5555	7380	9205	11030
Lihue AP	0	0	0	0	0	3719	5535	7360	9185	11010
IDAHO										
Boise	5833	4533	3399	2434	1626	714	1233	1929	2793	3811
Idaho Falls	8619	7129	5800	4609	3590	237	573	1064	1703	2511
Lewiston	5464	4158	3050	2112	1366	657	1186	1886	2780	3856
Pocatello	7063	5687	4454	3401	2504	437	883	1477	2252	3177
ILLINOIS										
Cairo	3833	2895	2090	1447	925	1806	2687	3710	4893	6197
Chicago Midway AP	6127	4952	3912	2998	2219	925	1575	2361	3272	4317
Chicago O'Hare AP	6497	5245	4163	3220	2404	664	1243	1986	2863	3872
Decatur	5344	4247	3293	2461	1778	1197	1925	2797	3791	4932
Moline	6395	5202	4170	3263	2462	893	1530	2324	3236	4262
Peoria	6098	4930	3910	3013	2239	968	1631	2431	3360	4412
Rockford	6845	5600	4507	3555	2713	714	1298	2032	2899	3883
Springfield	5558	4437	3468	2615	1913	1116	1821	2670	3654	4772
INDIANA										
Evansville	4624	3578	2685	1929	1327	1364	2139	3064	4140	5367
Fort Wayne	6209	4992	3930	2996	2193	748	1358	2117	3008	4030
Indianapolis	5577	4430	3431	2568	1856	974	1653	2478	3441	4554
South Bend	6462	5213	4118	3156	2333	695	1271	2002	2865	3867
IOWA										
Burlington Radio KBUR	6149	4988	3970	3077	2308	994	1657	2466	3396	4447
Des Moines	6710	5521	4470	3546	2728	928	1561	2335	3235	4243
Dubuque	7277	5992	4871	3893	3028	606	1146	1850	2697	3657
Mason City AP	7901	6586	5430	4425	3529	580	1088	1763	2581	3508
Sioux City	6953	5745	4674	3738	2898	932	1545	2298	3182	4177
Spencer	7770	6474	5329	4334	3448	641	1171	1857	2685	3620
Waterloo	7415	6153	5040	4070	3199	675	1236	1950	2806	3760
KANSAS										
Concordia	5623	4498	3509	2646	1912	1302	1998	2832	3795	4886
Dodge City	5046	3963	3011	2184	1512	1411	2153	3022	4025	5176
Goodland	6119	4891	3804	2857	2041	925	1515	2253	3131	4139
Russell FAA AP	5312	4220	3259	2423	1735	1485	2219	3081	4076	5210
Topeka	5243	4152	3203	2378	1700	1361	2093	2974	3974	5112
Wichita	4687	3654	2750	1977	1346	1673	2464	3386	4438	5638
KENTUCKY										
Covington	5070	3965	3001	2189	1527	1080	1798	2654	3672	4834
Lexington	4729	3652	2743	1963	1350	1197	1941	2858	3904	5120
Louisville	4640	3584	2676	1906	1303	1268	2032	2942	4005	5227
LOUISIANA										
Alexandria	2200	1443	880	490	234	2193	3260	4525	5958	7531
Baton Rouge	1670	1036	582	295	120	2585	3775	5150	6685	8340
Lake Charles	1498	908	500	240	91	2739	3978	5391	6956	8638
New Orleans, Aud. Park	1343	805	439	202	82	2876	4165	5622	7215	8916
New Orleans, N.O.	1465	893	492	239	96	2706	3960	5383	6956	8638
Shreveport	2167	1438	883	490	233	2538	3634	4906	6335	7903
MAINE										
Bangor	7950	6496	5222	4103	3122	268	640	1194	1896	2740
Caribou	9632	8044	6634	5409	4319	128	365	784	1379	2118
Old Town FAA AP	8648	7133	5800	4628	3589	209	519	1016	1660	2454
Portland	7498	6035	4764	3658	2705	252	616	1169	1890	2758
MARYLAND										
Baltimore	4729	3631	2682	1873	1236	1108	1840	2708	3728	4918
MASSACHUSETTS										
Blue Hill	6335	5020	3885	2895	2071	457	968	1659	2493	3498
Boston	5621	4383	3313	2405	1659	661	1250	2000	2920	4000

Table 8-2 Degree-Days to Several Bases (in °F) for Various US Locations (*Continued*)

	Heating					Cooling				
	Base					Base				
State	**65**	**60**	**55**	**50**	**45**	**65**	**60**	**55**	**50**	**45**
Nantucket AP	5929	4520	3323	2311	1513	284	708	1332	2143	3170
Worcester	6848	5498	4326	3296	2421	387	863	1514	2303	3259
MICHIGAN										
Alpena	8518	6982	5635	4473	3464	208	497	981	1642	2459
Detroit	6419	5167	4072	3113	2280	654	1227	1961	2823	3815
Flint	7041	5705	4540	3529	2640	438	923	1586	2399	3335
Grand Rapids	6801	5514	4383	3396	2524	575	1108	1807	2646	3598
Houghton Lake	8347	6861	5579	4455	3486	250	590	1132	1832	2689
Lansing	6904	5608	4464	3470	2595	535	1059	1747	2578	3528
Marquette	8351	6835	5517	4379	3378	216	531	1031	1725	2549
Muskegon	6890	5550	4390	3373	2482	469	953	1620	2428	3360
Sault Ste Marie	9193	7614	6215	5017	3971	139	386	816	1443	2217
Traverse City AP	7698	6272	5035	3953	3013	376	773	1362	2101	2989
MINNESOTA										
Duluth	9756	8185	6793	5581	4540	176	425	864	1482	2259
Intnl Falls	10547	8995	7623	6413	5348	176	454	908	1523	2283
Minneapolis–St. Paul	8159	6842	5677	4668	3765	585	1097	1758	2575	3491
Rochester	8227	6868	5682	4643	3733	474	943	1579	2370	3280
St. Cloud	8868	7481	6255	5187	4241	426	862	1468	2220	3098
MISSISSIPPI										
Jackson	2300	1548	988	590	319	2316	3394	4664	6086	7639
Meridian	2388	1621	1042	623	339	2231	3289	4538	5940	7483
MISSOURI										
Columbia Region	5078	3997	3064	2259	1605	1269	2009	2901	3919	5089
Kansas City	5161	4089	3157	2351	1694	1421	2169	3061	4085	5249
St Louis	4750	3701	2798	2031	1419	1475	2252	3174	4232	5445
St Joseph	5435	4341	3378	2544	1847	1334	2064	2925	3911	5046
Springfield	4570	3517	2611	1844	1235	1382	2149	3068	4126	5342
MONTANA										
Billings	7265	5898	4697	3641	2766	498	951	1581	2354	3298
Butte	9719	8059	6557	5225	4078	58	222	545	1038	1718
Cut Bank AP	9033	7474	6096	4907	3886	140	406	856	1489	2299
Dillon AP	8354	6821	5457	4255	3237	199	492	953	1570	2382
Glasgow	8969	7572	6329	5238	4302	438	867	1449	2185	3074
Great Falls	7652	6248	5022	3965	3074	339	760	1365	2132	3066
Havre	8687	7282	6073	5005	4104	395	818	1432	2191	3113
Helena	8190	6710	5389	4247	3258	256	606	1105	1786	2629
Kalispell	8554	6959	5542	4304	3233	117	348	755	1342	2096
Lewistown FAA AP	8586	7038	5676	4487	3467	192	468	933	1567	2379
Miles City AP	7889	6562	5392	4369	3479	752	1252	1905	2706	3641
Missoula	7931	6410	5066	3884	2876	188	497	970	1616	2428
NEBRASKA										
Grand Island	6420	5224	4166	3239	2434	1036	1662	2428	3326	4345
Lincoln AP	6218	5062	4040	3139	2362	1148	1809	2611	3536	4585
Lincoln	6012	4875	3870	2993	2234	1187	1865	2685	3634	4701
Norfolk	6981	5745	4663	3710	2863	925	1520	2263	3131	4118
North Platte	6743	5470	4345	3354	2509	802	1359	2060	2898	3874
Omaha–Eppley	6049	4907	3911	3037	2290	1173	1862	2691	3637	4715
Omaha–North	6601	5400	4349	3427	2624	949	1573	2346	3249	4270
Scottsbluff	6774	5473	4304	3289	2415	666	1188	1845	2653	3605
Valentine	7300	6006	4859	3847	2956	736	1267	1945	2758	3692
NEVADA										
Elko	7483	6027	4714	3586	2625	342	706	1228	1910	2785
Ely	7814	6327	5004	3826	2829	207	550	1052	1694	2526
Las Vegas	2601	1770	1120	625	306	2946	3938	5114	6443	7950
Lovelock FAA	5990	4695	3550	2579	1747	684	1217	1894	2743	3470
Reno	6022	4612	3387	2360	1534	329	739	1344	2150	3140
Tonopah	5900	4610	3492	2532	1723	631	1167	1869	2739	3753
Winnemucca	6629	5241	3994	2931	2015	407	845	1423	2185	3096
NEW HAMPSHIRE										
Concord	7360	5967	4757	3682	2762	349	781	1394	2150	3051
Mount Washington	13878	12053	10253	8534	6960	0	01	25	132	379
NEW JERSEY										
Atlantic City	4946	3783	2784	1941	1267	864	1533	2349	3339	4485
Atlantic City Marina	4693	3534	2530	1713	1076	835	1503	2317	3333	4517
Newark	5034	3911	2920	2074	1391	1024	1721	2543	3533	4677
Trenton	4947	3818	2832	1996	1323	968	1661	2493	3482	4634
NEW MEXICO										
Albuquerque	4292	3234	2330	1557	963	1316	2080	2996	4053	5288
Clayton	5207	3999	2966	2089	1374	767	1380	2176	3120	4231
Roswel1	3697	2729	1898	1226	706	1560	2417	3412	4566	5872
Truth or Consequences	3392	2447	1636	1007	542	1558	2429	3447	4647	6008
Tucumcari FAA	4047	3015	2135	1415	858	1357	2148	3096	4200	5467
Zuni FAA	5515	4507	3381	2437	1648	473	983	1685	2567	3605
NEW YORK										
Albany	6888	5596	4451	3457	2595	574	1111	1787	2619	3583
Binghamton	7285	5908	4714	3677	2767	369	820	1452	2231	3151
Buffalo	6927	5591	4429	3403	2508	437	928	1590	2388	3319
Massena FAA	8237	6827	5596	4510	3552	343	759	1352	2088	2958

Table 8-2 Degree-Days to Several Bases (in °F) for Various US Locations (*Continued*)

	Heating					Cooling				
	Base					Base				
State	65	60	55	50	45	65	60	55	50	45
New York Central Park	4848	3739	2771	1958	1299	1068	1784	2636	3653	4814
New York JFK Int'l. AP	5184	4023	2994	2130	1422	861	1520	2321	3278	4395
New York LaGuardia	4909	3787	2806	1980	1311	1048	1752	2587	3589	4740
Oswego East	6792	5444	4274	3243	2376	435	915	1570	2360	3319
Rochester	6719	5417	4285	3291	2434	531	1062	1750	2580	3549
Syracuse	6678	5379	4250	3267	2429	551	1081	1778	2621	3607
NORTH CAROLINA										
Asheville	4237	3129	2224	1488	937	872	1587	2508	3595	4868
Cape Hatteras	2731	1846	1166	702	380	1550	2485	3635	4991	6500
Charlotte	3218	2300	1552	984	585	1596	2503	3579	4842	6263
Greensboro	3825	2811	1984	1324	825	1341	2158	3149	4318	5640
Raleigh–Durham	3514	2542	1744	1123	670	1394	2242	3273	4482	5850
Wilmington	2433	1632	1028	610	321	1964	2995	4225	5622	7162
NORTH DAKOTA										
Bismarck	9044	7656	6425	5326	4374	487	928	1518	2248	3116
Fargo	9271	7891	6663	5573	4615	473	919	1515	2251	3122
Minot FAA	9407	7964	6685	5564	4573	370	758	1299	2002	2837
Williston	9161	7753	6504	5387	4450	422	841	1415	2128	3011
OHIO										
Akron–Canton	6224	4971	3883	2936	2129	634	1205	1943	2820	3839
Cincinnati Abbe Obs	4844	3763	2830	2040	1412	1188	1931	2819	3864	5060
Cincinnati AP	5070	3965	3001	2189	1527	1080	1798	2654	3672	4834
Cleveland	6154	4901	3819	2876	2079	613	1183	1926	2807	3836
Columbus	5702	4513	3480	2597	1846	809	1449	2244	3183	4257
Dayton	5641	4483	3468	2600	1866	936	1603	2414	3370	4460
Mansfield	5818	4618	3573	2679	1917	818	1445	2225	3152	4219
Toledo Express	6381	5136	4049	3091	2274	685	1268	2001	2870	3877
Youngstown	6426	5145	4032	3054	2232	518	1065	1774	2621	3623
OKLAHOMA										
Oklahoma City	3695	2760	1962	1326	809	1876	2768	3788	4980	6289
Tulsa	3680	2750	1950	1306	778	1949	2850	3865	5052	6347
OREGON										
Astoria	5295	3620	2233	1215	570	13	159	5961	1415	2598
Burns	7212	5740	4436	3299	2343	289	649	1161	1851	2724
Eugene	4739	3313	2141	1226	607	239	638	1286	2201	3417
Meacham	7863	6249	4817	3556	2495	103	317	712	1275	2034
Medford	4930	3614	2496	1577	882	562	1077	1779	2685	3813
North Bend AP	4688	2985	1642	756	292	0	131	597	1553	2913
Pendleton	5240	3968	2868	1970	1264	656	1211	1935	2858	3982
Portland	4792	3385	2234	1333	708	300	711	1378	2309	3520
Redmond AP	6643	5106	3767	2621	1680	170	459	943	1620	2512
Salem	4852	3424	2246	1317	667	232	620	1272	2169	3355
Sexton Summit	6430	4859	3477	2311	1374	137	381	837	1499	2386
PACIFIC										
Guam	0	0	0	0	0	5011	6836	8661	10486	12311
Johnston AP	0	0	0	0	0	5086	6911	8736	10561	12386
Koror	0	0	0	0	0	6008	7833	9658	11483	13308
Kwajalein AP	0	0	0	0	0	6164	7989	9814	11639	13464
Majuro AP	0	0	0	0	0	5904	7729	9554	11379	13204
Pago Pago AP	0	0	0	0	0	5325	7150	8975	10800	12625
Ponape	0	0	0	0	0	5652	7477	9302	11127	12952
Truk, Moen I, AP	0	0	0	0	0	5888	7713	9536	11363	13188
Wake	0	0	0	0	0	5455	7280	9105	10930	12755
Yap AP	0	0	0	0	0	5916	7741	9566	11391	13216
PENNSYLVANIA										
Allentown	5827	4618	3550	2633	1843	772	1392	2150	3053	4088
Bradford AP	7804	6294	5006	3894	2931	170	482	1022	1735	2596
Erie	6851	5485	4304	3283	2411	373	832	1482	2282	3235
Harrisburg	5224	4087	3097	2238	1541	1025	1711	2545	3511	4644
Philadelphia	4865	3753	2788	1965	1312	1104	1817	2671	3679	4849
Pittsburgh City	5278	4135	3138	2294	1603	948	1630	2456	3440	4573
Pittsburgh AP	5930	4694	3637	2720	1938	647	1240	2004	2914	3961
Wilkes-Barre/Scranton	6277	5018	3923	2972	2149	608	1181	1909	2778	3783
Williamsport	5981	4757	3695	2764	1971	698	1299	2059	2952	3986
PUERTO RICO										
San Juan	0	0	0	0	0	4982	6807	8632	10457	12282
RHODE ISLAND										
Block Island	5771	4432	3289	2306	1517	359	844	1523	2368	3409
Providence	5972	4682	3565	2599	1803	532	1067	1774	2625	3662
SOUTH CAROLINA										
Charleston AP	2146	1406	864	496	240	2078	3163	4454	5903	7478
Charleston City	1904	1230	741	412	188	2354	3502	4839	6334	7937
Columbia	2598	1783	1154	686	374	2087	3094	4292	5647	7159
Florence	2566	1748	1127	676	374	1952	2960	4171	5538	7060
Greenville–Spartenburg	3163	2246	1493	921	519	1573	2477	3552	4809	6229
SOUTH DAKOTA										
Aberdeen	8617	7267	6078	5014	4087	566	1046	1678	2440	3337
Huron	8055	6751	5600	4582	3678	711	1239	1912	2714	3641
Pierre AP	7677	6401	5271	4273	3409	858	1406	2102	2928	3889

Table 8-2 Degree-Days to Several Bases (in °F) for Various US Locations (*Continued*)

	Heating					Cooling				
	Base					Base				
State	65	60	55	50	45	65	60	55	50	45
Rapid City	7324	5982	4799	3762	2868	661	1148	1796	2575	3511
Sioux Falls	7838	6543	5401	4394	3498	719	1253	1933	2746	3681
TENNESSEE										
Bristol	4306	3255	2373	1646	1093	1107	1880	2823	3922	5197
Chattanooga	3505	2574	1785	1180	737	1636	2526	3566	4791	6169
Knoxville	3478	2557	1775	1187	744	1569	2475	3518	4753	6135
Memphis	3227	2352	1624	1058	640	2029	2984	4077	5339	6744
Nashville	3696	2758	1964	1338	852	1694	2576	3613	4812	6151
Oak Ridge	3944	2955	2119	1445	933	1367	2202	3187	4338	5656
TEXAS										
Abilene	2610	1801	1162	664	342	2466	3481	4670	5995	7498
Amarillo	4183	3156	2278	1548	976	1433	2230	3177	4274	5527
Austin	1737	1097	620	316	127	2903	4095	5443	6962	8600
Brownsville	650	336	146	54	19	3874	5385	7020	3753	10543
Corpus Christi	930	514	243	98	28	3474	4880	6438	8111	9872
Dallas	2290	1544	949	526	250	2755	3835	5073	6467	8016
Del Rio	1523	923	494	230	801	3363	4596	5986	7548	9222
El Paso	2678	1833	1149	653	326	2098	3077	4229	5548	7048
Fort Worth	2382	1616	1007	562	274	2587	3642	4862	6239	7775
Galveston	1224	704	369	157	54	3004	4312	5800	7413	9139
Houston	1434	864	471	215	81	2889	4147	5576	7150	8835
Laredo No. 2	876	481	230	87	32	4137	5568	7143	8824	10593
Lubbock	3545	2603	1807	1163	666	1647	2535	3559	4745	6068
Lufkin AP	1940	1253	731	385	163	2592	3730	5033	6512	8114
Midland	2621	1808	1159	656	333	2245	3258	4434	5757	7258
Port Arthur	1518	924	504	238	86	2798	4028	5431	6990	8669
San Angelo	2240	1498	918	493	227	2702	3789	5031	6432	7993
San Antonio	1570	956	518	242	92	2994	4206	5594	7146	8818
Victoria	1227	702	364	150	51	3140	4440	5925	7537	9262
Waco	2058	1357	807	437	195	2863	3988	5271	6717	8303
Wichita Falls	2904	2061	1384	832	451	2611	3594	4741	6015	7458
UTAH										
Blanding	6163	4869	3732	2757	1912	600	1129	1827	2670	3646
Bryce Canyon AP	9133	7459	5949	4616	3480	41	193	505	1005	1686
Cedar City AP	6137	4833	3690	2717	1897	615	1130	1813	2671	3678
Milford	6412	5109	3957	2969	2121	688	1212	1885	2721	3704
Salt Lake City	5983	4733	3633	2676	1864	927	1502	2221	3094	4108
Wendover	5760	4558	3511	2621	1870	1137	1760	2538	3475	4547
VERMONT										
Burlington	7876	6488	5270	4190	3246	396	833	1440	2180	3066
VIRGINIA										
Lynchburg	4233	3172	2269	1536	966	1100	1861	2783	3873	5128
Norfolk	3488	2516	1710	1100	663	1441	2284	3315	4530	5918
Richmond	3939	2916	2061	1388	866	1353	2157	3127	4276	5580
Roanoke	4307	3234	2326	1580	1011	1030	1778	2690	3771	5029
Wallops Island	4240	3170	2268	1531	978	1107	1865	2788	3881	5149
WASHINGTON										
Olympia	5530	3970	2653	1617	854	101	365	880	1657	2731
Omak	6858	5476	4253	3230	2355	522	965	1573	2367	3324
Quillayute	5951	4232	2750	1603	813	8	116	458	1137	2172
Seattle–Tacoma	5185	3657	2386	1416	731	129	423	984	1832	2981
Seattle (Urban)	4727	3269	2091	1194	602	183	549	1197	2127	3358
Spokane	6835	5420	4173	3088	2188	388	797	1377	2120	3040
Stampede Pass	9400	7643	6006	4532	3256	16	83	274	623	1176
Walla Walla	4835	3616	2600	1760	1126	862	1471	2279	3260	4457
Yakima	6009	4655	3483	2502	1688	479	945	1604	2452	3455
WEST INDIES										
Swan Island	0	0	0	0	0	5809	7634	9459	11284	13109
WEST VIRGINIA										
Beckley	5615	4356	3279	2390	1652	490	1061	1809	2745	3833
Charleston	4590	3500	2590	1809	1216	1055	1790	2699	3750	4981
Elkins	5975	4659	3533	2616	1834	389	905	1601	2508	3555
Huntington	4624	3533	2624	1843	1249	1098	1829	2746	3790	5020
Parkersburg	4817	3720	2786	1987	1363	1045	1770	2657	3686	4888
WISCONSIN										
Eau Claire AP	8388	7033	5832	4786	3860	459	928	1554	2331	3231
Green Bay	8098	6689	5473	4405	3478	386	805	1411	2168	3066
La Crosse AP	7417	6158	5050	4088	3219	695	1264	1978	2841	3798
Madison	7730	6373	5188	4156	3250	460	923	1572	2361	3279
Milwaukee	7444	6080	4898	3860	2946	450	911	1554	2342	3252
WYOMING										
Casper	7555	6167	4914	3813	2857	458	895	1468	2193	3061
Cheyenne	7255	5825	4562	3452	2512	327	734	1288	2003	2886
Lander	7869	6471	5207	4080	3140	383	814	1376	2078	2965
Rock Springs AP	8410	6922	5592	4412	3393	227	563	1059	1703	2515
Sheridan	7708	6298	5024	3935	3000	446	860	1411	2147	3037

From Table 8-2, the degree-days for Chicago, Midway, to this base can be obtained by interpolation to be 5775. For summer, the CDD to base 63.5 are found from Table 8-2 to be 1120.

(a) From Eq. 8-6, the heating energy requirements are

$$E_H = \frac{240{,}000 \times 5775 \times 24}{0.95 \times 3413 \times 75} = 129{,}900\ \text{kWh}$$

(assuming 5% loss from baseboard units through wall)

$$\text{Heating cost} = 129{,}900 \times \$0.072 = \$9350$$

From Eq. 8-7, the cooling energy is determined:

$$E_C = \frac{85{,}000 \times 1120 \times 24}{1000 \times 14 \times (91 - 78)} = 12{,}550\ \text{kWh}$$

$$\text{Cooling cost} = 12{,}550 \times \$0.072 = \$904$$

(b) From Eq. 8-6, the heating energy requirements are

$$E_H = \frac{240{,}000 \times 5775 \times 24}{0.93 \times 100{,}000 \times 75} = 4770\ \text{therm}$$

(with 1 therm = 100,000 Btu and assuming 93% AFUE from Chapter 19)

$$\text{Heating cost} = 4770 \times \$0.062 = \$2960$$

8.6 Bin Method (Heating and Cooling)

For many applications, the degree-day method should not be used, even with the variable-base method, because the heat loss coefficient K, the efficiency η of the HVAC system, or the balance-point temperature may not be sufficiently constant. The efficiency of a heat pump, for example, varies strongly with outdoor temperature, or the efficiency of the HVAC equipment may be affected indirectly by t_o when the efficiency varies with the load, which is a common situation for boilers and chillers. Furthermore, in most commercial buildings, the occupancy has a pronounced pattern, which affects heat gain, indoor temperature, and ventilation rate.

In such cases, a steady-state calculation can yield good results for the annual energy consumption if different temperature intervals and time periods are evaluated separately. This approach is known as the *bin method*, because the energy consumption E_{bin} is calculated for several values of the outdoor temperature t_o and multiplied by the number of hours N_{bin} in the temperature interval (bin) centered around that temperature:

$$E_{\text{bin}} = N_{\text{bin}} K_{\text{tot}} (t_{\text{bal}} - t_o) / \eta \qquad (8\text{-}8)$$

This equation is evaluated for each bin, and the total energy consumption is the sum of E_{bin} over all bins:

$$E = \Sigma E_{\text{bin}} \qquad (8\text{-}9)$$

In the United States, the necessary data are widely available. The bins are usually in 5°F increments and are often collected in three daily 8-hour shifts. Mean coincident wet-bulb temperature data (for each dry-bulb bin) are used to calculate latent cooling loads from infiltration and ventilation. The bin method considers both occupied and unoccupied building conditions and gives credit for internal loads by adjusting the balance point. For many applications, the number of calculations can be minimized. A residential heat pump (heating mode) could be evaluated using only the bins below the balance point without the three-shift breakdown.

The data included in Table 8-3 are annual totals for various cities in the United States. ASHRAE (1995) and USAF (1978) include monthly data and data further separated into time intervals throughout the day.

Modified Bin Method. Various refinements, such as the seasonal variation of solar gains, can be included in a bin calculation. If a separate calculation is done for each month, the heat gain could be based on the average solar heat gain of the month. The diurnal variation of solar gains can be accounted for by calculating the average solar gain for each of the hourly time periods of the bin method. If such a detailed calculation of solar gains is not necessary, a linear correlation of monthly average solar heat gains with monthly average outdoor temperature could be assumed. Using bin data for the corresponding periods, the calculation can also account for the operating schedules of commercial buildings.

The modified bin method has the advantage of allowing the use of diversified (part-load) rather than peak-load values to establish the load as a function of outdoor dry-bulb temperature. This method also allows both secondary and primary (plant) HVAC equipment effects to be included in the energy calculation. The modified bin method permits the user to predict more accurately effects such as reheat and heat recovery that can only be guessed at with the degree-day or conventional bin methods.

In the modified bin method, average solar gain profiles, average equipment and lighting use profiles, and cooling load temperature difference (CLTD) values are used to characterize time-dependent diversified loads. The CLTDs approximate the transient effects of building mass. Time dependencies resulting from scheduling are averaged over a selected period, or multiple calculation periods are established. The duration of a calculation period determines the number of bin hours included. Normally, two calculation periods, representing occupied and unoccupied hours, are sufficient. The method can be further refined by making calculations on a monthly, not annual, basis.

Degree-Day Data from Bin Data. To calculate degree-days from hourly bin data, the base or balance temperature must first be determined. When t_{bal} is known, the following summation can be used for any time scale, either monthly or annually, or for several periods of a day on either a monthly or annual basis.

$$\text{DD}_h(t_{\text{bal}}) = [\Sigma(t_{\text{bal}} - t_{\text{bin}}) N_{\text{bin}}] / 24$$

where

t_{bin} = temperature at center of bin
N_{bin} = number of hours in bin at t_{bin}

Cooling degree-days are calculated analogously from

Table 8-3 Hourly Weather Occurrences

	Outdoor Temperature, °F																		
Location	**72**	**67**	**62**	**57**	**52**	**47**	**42**	**37**	**32**	**27**	**22**	**17**	**12**	**7**	**2**	**–3**	**–8**	**–13**	**–18**
Albany, NY	588	733	740	708	652	625	647	769	793	574	404	278	184	110	63	32	10	5	4
Albuquerque, NM	767	831	719	651	687	734	741	689	552	346	154	66	21	4	1	1			
Atlanta, GA	1185	926	823	784	735	676	598	468	271	112	44	19	8	2					
Bakersfield, CA	831	898	966	977	908	746	541	247	77	7									
Birmingham, AL	1138	908	805	742	668	614	528	433	292	143	69	17	6	3					
Bismarck, ND	454	566	614	606	563	520	518	604	653	550	474	371	338	292	278	208	131	77	80
Boise, ID	492	575	643	702	786	798	878	829	522	307	148	53	26	14	6	2			
Boston, MA	676	819	804	781	766	757	828	848	674	429	256	151	74	35	4	9	1		
Buffalo, NY	646	772	760	700	666	624	647	756	849	602	426	267	170	81	5	24	2		
Burlington VT	573	670	703	694	655	603	637	716	752	561	491	336	272	216	135	81	39	17	8
Casper, WY	423	532	592	642	606	670	782	831	806	683	495	325	200	116	73	45	30	15	5
Charleston, SC	1267	1090	889	787	651	576	434	321	192	79	27	5							
Charleston, WV	912	949	767	689	661	667	607	633	630	356	252	135	73	22	7	1			
Charlotte, NC	1115	908	839	752	730	684	634	515	360	166	64	23	5	2					
Chattanooga, TN	1021	895	775	722	713	679	642	553	414	228	113	45	4	4	2				
Chicago, IL	762	769	653	592	569	543	591	800	822	551	335	196	117	85	59	25	12	3	
Cincinnati, OH	879	843	726	639	611	599	627	698	711	460	249	131	68	44	18	8	2		
Cleveland, OH	763	831	723	641	638	607	620	754	806	578	355	201	111	47	22	11	2		
Columbus, OH	774	820	720	648	622	603	658	772	730	502	280	169	94	40	20	10	4	1	
Corpus Christi, TX	1175	1041	748	551	444	302	180	83	27	9	3								
Dallas, TX	831	795	693	656	629	576	504	371	231	91	34	17	4	1					
Denver, CO	549	684	783	731	678	704	692	717	721	553	359	216	119	78	36	22	6	1	1
Des Moines, IA	707	751	681	600	585	512	510	627	747	557	405	281	211	152	104	59	23	8	1
Detroit, MI	721	783	695	633	592	566	595	808	884	618	377	248	131	61	17	4	1		
El Paso, TX	933	839	749	760	687	611	494	369	233	34	104	10	2						
Ft. Wayne, IN	728	777	699	608	569	552	601	725	905	596	381	205	124	69	40	19	6	1	
Fresno, CA	709	803	921	1006	1036	952	673	426	168	34									
Grand Rapids, MI	634	739	712	647	571	565	554	742	938	690	469	293	172	78	31	10	1	1	
Great Falls, MT	407	520	636	754	822	830	832	813	698	533	355	218	167	136	118	101	68	51	62
Harrisburg, PA	807	824	737	692	635	659	722	888	749	427	222	125	52	18	4	1			
Hartford, CT	617	755	751	752	649	575	683	807	825	552	370	233	153	77	33	11	3	2	
Houston, TX	1172	980	772	681	570	452	291	141	64	18	4	2							
Indianapolis, IN	821	815	722	585	586	579	605	712	791	551	293	152	97	60	35	13	3	2	
Jackson, MS	1168	922	790	677	618	605	484	367	224	103	41	6	2	2	1				
Jacksonville, FL	1334	975	879	692	530	355	288	154	83	24	2								
Kansas City, MO	761	723	601	572	553	562	628	591	625	407	265	175	99	51	21	4			
Knoxville, TN	1056	889	746	675	672	689	648	590	456	217	101	41	21	7	2				
Las Vegas, NV	651	644	699	786	769	716	591	396	194	44	7	1							
Little Rock, AR	940	803	725	672	638	669	605	509	363	172	50	23	5	1					
Los Angeles, CA	881	1654	2193	1904	1054	428	107	10											
Louisville, KY	869	758	693	654	619	634	649	703	631	332	169	97	45	25	8	3	1		
Lubbock, TX	833	829	688	700	642	618	620	546	490	346	180	86	33	7	5	1			
Memphis, TN	977	798	715	690	618	633	614	532	374	196	74	25	10	4					
Miami, FL	1705	810	452	277	147	71	26	4											
Milwaukee WI	597	753	749	634	585	591	611	774	913	659	421	285	176	116	83	47	18	4	3
Minneapolis, MN	621	690	695	602	588	482	500	560	632	609	514	383	311	246	186	119	62	31	16
Mobile, AL	1411	1038	882	698	609	506	377	214	109	49	7	3							
Nashville, TN	933	838	738	697	637	619	627	565	463	263	132	67	28	9	3	1	1		
New Orleans, LA	1189	987	850	692	621	449	282	128	47	9	2								
New York, NY	926	877	754	745	722	796	838	858	603	330	188	82	26	10	1				
Oklahoma City, OK	881	769	717	173	643	645	611	641	570	468	287	77	36	12	3	1			
Omaha, NB	726	721	606	558	539	543	543	655	663	511	390	287	189	135	93	40	15	1	
Philadelphia, PA	863	809	735	710	663	701	758	818	654	335	189	100	32	9					
Phoenix, AZ	762	776	767	769	659	540	391	182	57	8									
Pittsburg, PA	722	910	799	678	637	587	631	688	569	774	360	233	159	60	30	7	1		
Portland, ME	407	627	780	808	760	748	722	839	820	599	408	293	190	109	60	29	15	5	1
Portland, OR	373	581	1001	1316	1274	1271	1238	772	343	123	40	10	4	1					
Raleigh, NC	1087	937	848	762	707	672	638	527	410	236	103	38	11	1					
Reno, NV	418	477	572	690	845	909	890	829	733	530	387	277	101	37	15	4	1		
Richmond, VA	953	850	784	745	690	673	699	632	478	285	138	67	19	2	1				
Sacramento, CA	630	773	1071	1329	1298	1049	701	355	93	8									
Salt Lake City, UT	569	615	614	635	682	685	755	831	798	564	328	158	80	41	16	2			
San Antonio, TX	1086	943	789	669	569	445	387	190	94	31	11	4	1	1					
San Francisco, CA	285	665	1264	2341	2341	1153	449	99	10										
Seattle, WA	258	448	750	1272	1462	1445	1408	914	427	104	39	20	3						
Shreveport, LA	1063	886	772	679	619	609	516	361	200	72	23	6	2						
Sioux Falls, SD	566	684	669	605	522	498	501	625	712	585	520	448	293	208	152	102	59	43	18
St. Louis,MO	823	728	646	575	585	578	620	671	650	411	219	134	77	40	15	7	1		
Syracuse, NY	627	735	723	717	656	641	651	720	830	547	392	282	190	102	55	23	5	2	2
Tampa, FL	1387	1187	877	570	345	216	137	48	10	1									
Waco, TX	909	830	701	622	651	558	501	354	216	84	24	3	1						
Washington, D.C.	950	766	740	673	690	684	790	744	542	254	138	54	17	2					
Wichita, KS	758	709	641	603	589	592	611	584	607	426	273	161	85	45	14	3	1		

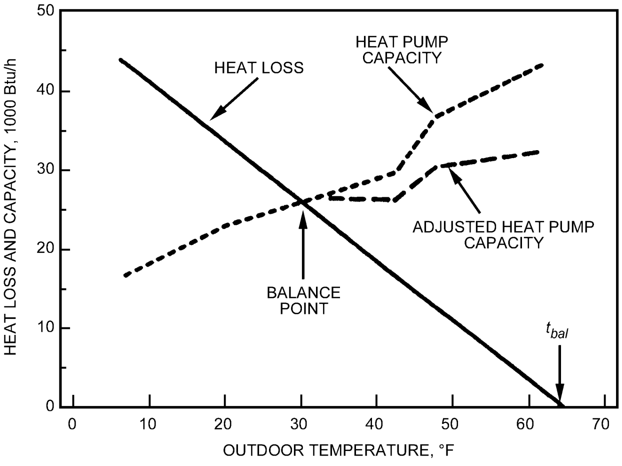

Fig. 8-5 Heat Pump Capacity and Building Load
(Figure 4, Chapter 19, *2017 ASHRAE Handbook—Fundamentals*)

$$DD_c(t_{\text{bal}}) = [\Sigma(t_{\text{bin}} - t_{\text{bal}})N_{\text{bin}}]/24$$

This method generally produces degree-day values slightly higher than published values from NOAA or the National Climatic Data Center. This small but systematic deviation can be suppressed by ignoring degree-days during the swing seasons when totals are less than a minimum, e.g., 50 to 100 Fahrenheit degree-days per month.

Weather data for use with the bin method are available from ASHRAE and in *Engineering Weather Data* (US Air Force Manual 88-29 1978). When time-of-occurrence bin data are not required, the hourly weather occurrence information in Table 8-3 may be used.

The basic data form for making a bin analysis is provided in Table 8-4.

Example 8-4 Estimate the energy requirements for a residence located in Washington, DC, with a design heat loss of 40,000 Btu/h at 53°F design temperature difference. The inside design temperature is 70°F. Average internal heat gains are estimated to be 4280 Btu/h. Assume a 3 ton heat pump with the characteristics given in Columns E and H of Table 8-5 and in Figure 8-5.

Solution: The design heat loss is based on no internal heat generation. The heat pump system energy input is the net heat requirement of the space (i.e., envelope loss minus internal heat generation). The net heat loss per degree and the heating/cooling balance temperature may be computed:

$$HL/\Delta t = 40{,}000/53 = 755 \text{ Btu/°F–h}$$

$$t_{\text{bal}} = 70 - (4280/755) = 64.3\text{°F}$$

Table 8-5 is then computed, resulting in an estimate of total electrical energy consumption for heat pump and supplemental heating of 9578 kWh.

8.7 Problems

8.1 The total design heating load on a residence in New York City is 32.8 kW (112,000 Btu/h) for an indoor temperature of 72°F. The furnace is *off* from June through September. Estimate:

(a) The annual energy requirement for heating
(b) The annual heating cost if electric heat is used with the single rate of 16¢/kWh, \$/yr
(c) The maximum savings effected if the thermostat is set back to 65°F between 10 P.M. and 6 A.M., \$/yr

8.2 Determine the cost per 1000 Btu of supplying heat in your territory for (a) oil, (b) gas, (c) direct electric heating, and (d) an air-source heat pump. In calculating gas and oil costs, assume heating plant efficiencies of 80% and 75%, respectively. In calculating heat pump costs, assume a condensing refrigerant temperature of 110°F, an evaporating refrigerant temperature 10°F below the local average winter outdoor air temperature, and an actual COP 70% of that for the reversed Carnot cycle. Assume a compressor mechanical efficiency of 80%, with the compressor located in the airstream being heated.

8.3 A home is located in Cleveland, Ohio, and has a design heat loss of 112,000 Btu/h at an inside design temperature of 72°F and an outside design temperature of 0°F. The home has an oil-fired furnace. Find the savings in gallons of fuel oil if the owner lowers the temperature in the home to 68°F between 10 P.M. and 6 A.M. every day during January.
[Ans: 13.2 gal (50 L)]

8.4 The total design heating load on a residence in Kansas City, Missouri, is 32.8 kW (112,000 Btu/h). The furnace is *off* during June through September. Estimate:

(a) Annual energy requirement for heating
(b) Annual heating cost if No. 2 fuel oil is used in a furnace with an efficiency of 80% (assume fuel oil costs 68¢/L)
(c) Maximum savings effected if the thermostat is set back from 22.2 to 18.3°C (72 to 65°F) between 10 PM and 6 AM in \$/yr

8.5 A residence located in Tulsa, Oklahoma, has a design heating load of 20 kW and a design cooling load of 9.4 kW. Determine the following:

(a) Heating energy requirements, kWh
(b) Litres of No. 2 fuel oil per season if used as heating fuel
(c) Litres of natural gas per season if used as heating fuel
(d) kWh of electric energy if used as heating fuel with baseboard units
(e) kWh of electric energy if used for air-conditioning system having $COP_{\text{seasonal}} = 3.4$
(f) Total airflow rate in L/s if a warm air system is used
(g) Total steam flow in kg/s if a steam system is used

8.6 Estimate the annual energy costs for heating and cooling a residence located in Cleveland, Ohio, having design loads of 65,000 Btu/h (heating) and 30,000 Btu/h (cooling) based on a 75°F indoor temperature. In winter the thermostat is set back

Table 8-4 Bin Data Form

Climate			House	Heat Pump							Supplemental		
A Temp. Bin, °F	B Temp. Diff. $t_{bal} - t_{bin}$	C Weather Data Bin, hours	D Heat Loss Rate, 1000 Btu/h	E Heat Pump Integrated Heating Capacity, 1000 Btu/h	F Cycling Capacity Adjustment Factor[a]	G Adjusted Heat Pump Capacity, 1000 Btu/h[b]	H Rated Electric Input, kW	I Operating Time Fraction[c]	J Heat Pump Supplied Heating, 10^6 Btu[d]	K Seasonal Heat Pump Elec. Consumption, kWh[e]	L Space Load, 10^6 Btu[f]	M Supplemental Heating Required, kWh[g]	N Total Electric Energy Consumption
62													
57													
52													
47													
42													
37													
32													
27													
22													
17													
12													
7													
2													
–3													
									TOTALS:				

[a] Cycling Capacity Adjustment Factor = 1 – $Cd(1 - x)$, where Cd = degradation coefficient (default = 0.25 unless part load factor is known) and x = building heat loss per unit capacity at temperature bin. Cycling capacity = 1 at the balance point and below.

[b] Col G = Col E × Col F

[c] Operating Time Factor equals smaller of 1 or Col D/Col G

[d] Col J = (Col I × Col G × Col C)/1000

[e] Col K = Col I × Col H × Col C

[f] Col L = Col C × Col D/1000

[g] Col M = (Col L – Col J) × 10^6/3413

[h] Col N = Col K + Col M

Table 8-5 Calculation of Annual Heating Energy Consumption for Example 8-4

(Table 2, Chapter 19, 2017 ASHRAE Handbook—Fundamentals)

Climate			House	Heat Pump							Supplemental		
A	B	C	D	E	F	G	H	I	J	K	L	M	N
Temp. Bin, °F	Temp. Diff., $t_{bal} - t_{bin}$	Weather Data Bin, h	Heat Loss Rate, 1000 Btu/h	Heat Pump Integrated Heating Capacity, 1000 Btu/h	Cycling Capacity Adjustment Factor[a]	Adjusted Heat Pump Capacity, 1000 Btu/h[b]	Rated Electric Input, kW	Operating Time Fraction[c]	Heat Pump Supplied Heating, 10^6 Btu[d]	Seasonal Heat Pump Electric Consumption, KWh[e]	Space Load, 10^6 Btu[f]	Supplemental Heating Required, kWh[g]	Total Electric Energy Consumption[h]
62	2.3	740	1.8	44.3	0.760	33.7	3.77	0.05	1.30	146	1.30	—	146
57	7.3	673	5.5	41.8	0.783	32.7	3.67	0.17	3.72	417	3.72	—	417
52	12.3	690	9.3	39.3	0.809	31.8	3.56	0.29	6.42	719	6.42	—	719
47	17.3	684	13.1	36.8	0.839	30.9	3.46	0.42	8.95	1002	8.95	—	1002
42	22.3	790	16.9	29.9	0.891	26.6	3.23	0.63	13.31	1614	13.31	—	1614
37	27.3	744	20.6	28.3	0.932	26.4	3.15	0.78	15.35	1833	15.35	—	1833
32	32.3	542	24.4	26.6	0.979	26.0	3.07	0.94	13.22	1559	13.22	—	1559
27	37.3	254	28.2	25.0	1.000	25.0	3.00	1.00	6.35	762	7.16	236	998
22	42.3	138	31.9	23.4	1.000	23.4	2.92	1.00	3.23	403	4.41	345	748
17	47.3	54	35.7	21.8	1.000	21.8	2.84	1.00	1.18	153	1.93	220	373
12	52.3	17	39.5	19.3	1.000	19.3	2.74	1.00	0.33	47	0.67	101	147
7	57.3	2	43.3	16.8	1.000	16.8	2.63	1.00	0.03	5	0.09	16	21
2	62.3	0	47.0	14.3	1.000	—	—	—	—	—	—	—	—
								Totals:	73.39	8660	76.52	917	9578

[a] Cycling Capacity Adjustment Factor = $1 - C_d(1 - x)$, where C_d = degradation coefficient (default = 0.25 unless part load factor is known) and x = building heat loss per unit capacity at temperature bin. Cycling capacity = 1 at the balance point and below. The cycling capacity adjustment factor should be 1.0 at all temperature bins if the manufacturer includes cycling effects in the heat pump capacity (Column E) and associated electrical input (Column H).

[b] Column G = Column E × Column F

[c] Operating Time Factor equals smaller of 1 or Column D/Column G

[d] Column J = (Column I × Column G × Column C)/1000

[e] Column K = Column I × Column H × Column C

[f] Column L = Column C × Column D/1000

[g] Column M = (Column L – Column J) × $10^6/3413$

[h] Column N = Column K + Column M

to 60°F for 10 hours each night. The furnace is on from October 1 through May 31. Electric baseboard heat is used. The air conditioner has an SEER of 7.3 (Btu/h)/W. Electricity costs 0.0725 $/kWh year-round.

[Ans: $2414]

8.7 A residence in St. Joseph, Missouri, has a design heating load of 68,000 Btu/h when design indoor and outdoor temperatures are 75°F and 3°F, respectively. The furnace is *off* from June through September. Determine the fuel and energy requirements for heating in:

(a) Btu/yr
(b) Gallons of No. 2 fuel oil/yr
(c) Cubic feet of natural gas/yr
(d) kWh/yr
(e) Total airflow rate in cfm if a warm air system is used
(f) Total steam flow in lb/h if a steam system is used
(g) Total water flow rate in gpm if a hydronic system is used
(h) Total electric power in kW if electric heating is used

8.8 For a residence located in New Orleans, Louisiana, the design cooling load is 12 kW (41,000 Btu/h). Determine:

(a) Annual energy requirements for cooling, kWh
(b) Cost of this energy if the electric rate is 6.5¢/kWh

8.9 An office building located in Springfield, Missouri, has a heat loss of 2,160,000 Btu/h for design condition of 75°F inside and 10°F outside. The heating system is operational between October 1 and April 30. Determine:

(a) Annual energy usage for heating
(b) Estimated fuel cost if No. 2 fuel oil is used having a heating value of 140,000 Btu/gal and costing $2.50/gal

[Ans: 2.8 × 10^9 Btu; $71,800]

8.10 A small football promotion office is being designed for Jacksonville, Florida. The design heating and cooling loads are 61,200 and 55,400 Btu/h, respectively, based on 99.6% and 1% outdoor design dry-bulb temperatures. Balance point has been estimated as 65°F.

(a) Select an appropriate heat pump from the XYZ Corporation models listed on the next page and estimate the energy costs for summer and winter if electricity is 8¢/kWh. These heat pumps are being provided at a very low cost because they ahve relatively low SEERs (10–11). The current (2017) required SEER is 14.0. AN example of the performance of a heat pump with a SEER of 14.0 is given below:

Cooling Capacity, Btu/h	49,000
EER Rating, cooling	12.00
SEER Rating, cooling	14.00
Heating Capacity at 47°F, Btu/h	47,000
Region IV HSPF Rating, heating	8.20
Heating Capacity at 17°F, Btu/h	30,200

(b) Compare the heating energy cost for the heat pump to that for a condensing gas furnace with natural gas costing $1.20 per therm.

8.11 A 1980 ft^2 residence located in Cincinnati, Ohio, has design heating and cooling loads of 74,000 Btu/h and 35,000 Btu/h, respectively. Determine:

(a) Heating energy requirements, Btu

**HEAT PUMP MODELS—XYZ CORPORATION for Problem 8.10—
Performance Data at ARI Standard Conditions**

	Cooling Capacity							Heating Capacity, 70°F Indoor Air						
	Design Conditions: ARI Rating Temperatures 80°F DB, 67°F WB Indoor, Return Air; 95°F DB Outdoor							Outdoor Air 47°F DB/43°F WB DOE High Temperature			Outdoor Air 17°F DB/15°F WB DOE Low Temperature			DOE Region IV
				Single Phase										
Model Numbers	ARI Std. Cap. Btu/h	Net Sens. Cooling Cap.	Net Lat. Btu/h	SEER	Total W	ARI Noise Rating	Approx. CFM	Btu/h	Power Input W	COP	Btu/h	Power Input W	COP	HSPF Btu/W· h
A018	18,200	13,700	4,500	11.20	1820	7.0	650	18,400	1671	3.25	9,700	1255	2.25	8.15
A024	24,200	18,000	6,200	10.80	2513	7.0	850	24,800	2162	3.35	14,400	1718	2.45	8.85
A030	30,000	22,000	8,000	10.60	3151	7.2	1050	31,800	2885	3.25	18,600	2263	2.40	8.65
A036	35,800	25,500	10,300	10.50	3837	7.4	1250	39,000	3590	3.15	22,000	2763	2.35	8.25
A042	42,500	31,900	10,600	11.20	4250	7.6	1450	43,500	3803	3.35	24,600	2929	2.45	8.85
A048	49,500	36,800	12,700	10.50	5269	7.6	1650	51,000	4596	3.25	30,000	3578	2.45	8.80
A060	60,000	43,700	16,300	10.50	6250	7.8	2050	66,000	6050	3.20	37,000	4612	2.35	8.50

A030	Heating Indoor Air Conditions, 70°F DB			
Outdoor Temperature	Btu/h	W	COP	EER
–18	8900	1800	1.45	4.94
–13	9700	1830	1.55	5.30
–8	10600	1880	1.65	5.64
–3	11800	1940	1.78	6.08
2	13200	2010	1.92	6.57
7	14800	2090	2.07	7.08
12	16600	2170	2.24	7.65
17	18600	2260	2.41	8.23
22	20500	2360	2.55	8.69
27	22700	2470	2.69	9.19
32	24900	2570	2.84	9.69
37	27300	2690	2.97	10.15
42	29600	2800	3.10	10.57
47	31800	2890	3.22	11.00
52	34500	3020	3.35	11.42
57	37000	3140	3.45	11.78
62	39400	3250	3.55	12.12
67	41800	3350	3.66	12.48
72	44100	3450	3.75	12.78
77	46400	3550	3.83	13.07
82	48600	3640	3.91	13.35

A036	Heating Indoor Air Conditions, 70°F DB			
Outdoor Temperature	Btu/h	W	COP	EER
–18	9300	2150	1.27	4.33
–13	10300	2210	1.37	4.66
–8	11700	2270	1.51	5.15
–3	13300	2350	1.66	5.66
2	15100	2440	1.81	6.19
7	17200	2550	1.98	6.75
12	19500	2650	2.15	7.33
17	22000	2760	2.34	7.97
22	24500	2910	2.47	8.42
27	27300	3050	2.62	8.95
32	30100	3190	2.76	9.44
37	33100	3330	2.91	9.94
42	36100	3480	3.04	10.37
47	39000	3610	3.17	10.80
52	42200	3770	3.28	11.19
57	45200	3910	3.39	11.56
62	48300	4050	3.49	11.93
67	51200	4190	3.58	12.22
72	54100	4310	3.68	12.55
77	56900	4440	3.75	12.82
82	59600	4550	3.84	13.10

A048	Heating Indoor Air Conditions, 70°F DB			
Outdoor Temperature	Btu/h	W	COP	EER
–18	10900	2780	1.15	3.92
–13	13200	2860	1.35	4.62
–8	15700	2950	1.56	5.32
–3	18400	3060	1.76	6.01
2	21200	3180	1.95	6.67
7	24200	3310	2.14	7.31
12	27300	3450	2.32	7.91
17	30000	3580	2.46	8.38
22	33800	3780	2.63	8.99
27	37300	3930	2.78	9.49
32	40800	4100	2.92	9.95
37	44400	4280	3.04	10.37
42	48000	4460	3.15	10.76
47	51000	4600	3.25	11.09
52	55400	4810	3.37	11.52
57	59200	4990	3.48	11.86
62	63000	5160	3.58	12.21
67	66700	5330	3.67	12.51
72	70500	5490	3.76	12.84
77	74200	5640	3.85	13.16
82	78000	5790	3.95	13.47

A060	Heating Indoor Air Conditions, 70°F DB			
Outdoor Temperature	Btu/h	W	COP	EER
–18	16800	3360	1.46	5.00
–13	18300	3470	1.55	5.27
–8	20400	3600	1.66	5.67
–3	23000	3760	1.79	6.12
2	26000	3940	1.93	6.60
7	29400	4130	2.09	7.12
12	33200	4350	2.24	7.63
17	37000	4610	2.35	8.03
22	41700	4810	2.54	8.67
27	46300	5050	2.69	9.17
32	51000	5300	2.82	9.62
37	56000	5560	2.95	10.07
42	61000	5810	3.08	10.50
47	66000	6050	3.20	10.91
52	71200	6320	3.30	11.27
57	76300	6560	3.41	11.63
62	81300	6790	3.51	11.97
67	86200	7020	3.60	12.28
72	91000	7230	3.69	12.59
77	95600	7420	3.78	12.88
82	99900	7600	3.85	13.14

(b) Gallons of No. 2 fuel oil if 75% efficient oil-fired warm air system is used
(c) Therms of natural gas if 88% efficient gas-fired warm air system is used
(d) kWh of electricity if 98% efficient baseboard units are used
(e) Required airflow, cfm, for warm air systems
(f) kWh of electricity if heat pump system (WA-36 specifications follow) including supplementary electric resistance heat is used

Performance Data for Model WA-36 Heat Pump for Problem 8-11

Air Temperature, °F	Heat Pump Output, 1000 Btu	Heat Pump Input, kW
62	44	4.5
57	43	4.4
52	41	4.3
47	39	4.1
42	36	4.0
37	33	3.9
32	30	3.7
27	27	3.6
22	24	3.5
17	22	3.3
12	19	3.2
7	17	3.1
2	15	2.9
−3	13	2.8

(g) kWh of electricity for cooling for air conditioner with SEER of 8.5 using degree-day estimation
(h) Required airflow, cfm, for air conditioning

8.12 A small commercial building located in Oklahoma City, Oklahoma, has design loads of 245,000 Btu/h, *heating*, and 162,000 Btu/h, *cooling*. Balance point for the building has been estimated at 65°F. Determine:

(a) Annual energy requirements for heating, Btu
(b) Fuel cost using LPG at $2.50/gallon, $
(c) Fuel cost using electric baseboard units with electricity at 6.7¢/kWh, $
(d) Savings if setback to 55°F is effected between 10 P.M. and 6 A.M., Monday through Saturday, and all day Sunday, %
(e) Cooling season energy cost using cooling degree-days if conditioner has SEER of 11.5 and electricity is 7¢/kWh

8.13 A small commercial building in Indianapolis, Indiana, has design heating and cooling loads of 98,000 Btu/h and 48,000 Btu/h, respectively. Internal heat gains throughout the winter are relatively steady at 4.5 kW. Electricity costs 7.1¢/kWh. Estimate:

(a) Annual heating cost if baseboard electric resistance units are used.
(b) Annual cooling cost with a conventional vapor compression air-cooled unit, using your choice of method.

Select a heat pump system for the building from the XYZ Corporation models. Determine the

(a) Annual heating cost
(b) Annual cooling cost

8.14 A small (2200 ft^2) food mart store located in Charlotte, NC, has design heating and cooling loads of 94,500 Btu/h and 57,400 Btu/h, respectively, based on inside design temperatures of 72°F (winter) and 78°F (summer). The store is open 24 hours a day and has a relatively constant internal load due to lights, food cases, people, etc., of 3.3 W/ft^2. Select a suitable heat pump for the XYZ Corporation and estimate its operating energy costs for both summer and winter if the price of electricity is 7.4¢/kWh.

8.8 Bibliography

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Chapter 9

DUCT AND PIPE SIZING

This chapter discusses the design of systems for conveying air and water. Chapter 21 of the 2017 *ASHRAE Handbook—Fundamentals* has further details on the design of duct systems. Chapter 20 from the same source gives details on space air diffusion. Chapter 22 of the 2017 *ASHRAE Handbook—Fundamentals* has additional details on pipe sizing, while Chapters 21, 44, and 46 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* have further information on fans, pumps, pipes, tubes, and fittings.

9.1 Duct Systems

An air-conditioning system must not only condition air, it must also distribute conditioned air throughout the space. Usually the conditioning fluid is distributed from a central equipment location to the individual spaces requiring environment control. For example, a fan and duct system distributes air, and a pump and piping system distributes water. Conditioned air is distributed into the room by air diffusers or grilles.

An objective of duct system design is to provide a system that, within prescribed limits of velocities, noise intensity, and space available for ducts, efficiently transmits the required flow rate of air to each space while maintaining a proper balance between investment and operating cost. When the heating, cooling, or ventilation load is established, the total flow rate of air required can be determined by methods shown in Chapters 17 and 18 of the 2013 *ASHRAE Handbook—Fundamentals*. The size of the duct system governs frictional losses and thereby the size of fan and power required to operate the duct system.

9.1.1 Pressure Changes

Resistance to airflow imposed by the supply duct system must be overcome by mechanical energy, which is ordinarily supplied by a fan. Resistance also is imposed by the return-air system, which must also be overcome by the fan. In air-conditioning and ventilating work, the pressure differences are ordinarily so small that the equations for incompressible flow can be applied. Additional simplification is obtained by considering the air to be at the standard density of 0.075 lb/ft^3 (1.2 kg/m^3).

At any cross section in a duct, the total pressure p_t is the sum of the static pressure p_s and the velocity pressure p_v.

$$p_t = p_s + p_v \tag{9-1}$$

Pressures are normally expressed in inches of water (Pa). The velocity pressure is then given by

$$p_v = \left(\frac{V}{cf_1}\right)^2 \tag{9-2}$$

where V is defined by the equation

$$V = cf_2(Q/A) \tag{9-3}$$

where

p_v = velocity pressure, in. of water (Pa)
V = mean velocity of fluid, ft/min (m/s)
Q = airflow rate, ft^3/min (L/s)
A = cross-sectional area of duct, in^2 (mm^2)
cf_1 = conversion factor, 4005 (1.29)
cf_2 = conversion factor, 144 (1000)

The following table relates the air velocity to the velocity pressure for a range of velocities that might be encountered in HVAC duct systems. These values were calculated from Equation (9-2) assuming a standard density of air as 0.075 lb$_m$/ft^3.

V fpm	**P_v** in. water	**V** fpm	**P_v** in. water	**V** fpm	**P_v** in. water
500	0.016	1800	0.202	4400	1.21
600	0.022	2000	0.249	4600	1.32
700	0.031	2200	0.302	4800	1.44
800	0.040	2400	0.359	5000	1.56
900	0.050	2600	0.421	5200	1.69
1000	0.062	2800	0.489	5400	1.82
1100	0.075	3000	0.561	5600	1.96
1200	0.090	3200	0.638	5800	2.10
1300	0.105	3400	0.721	6000	2.24
1400	0.122	3600	0.808	6200	2.40
1500	0.140	3800	0.900	6400	2.55
1600	0.160	4000	0.998	6600	2.72
1700	0.180	4200	1.10	6800	2.88

If the air is not at this standard density, Equation (9-4) may be used in place of Equation (9-2):

$$p_v = \rho\left(\frac{V}{cf_3}\right)^2 \tag{9-4}$$

where

ρ = air density, lb/ft^3 (kg/m^3)
V = fluid mean velocity, ft/min (m/s)
p_v = velocity pressure, in. of water (Pa)
cf_3 = conversion factor, 1097 (1.414)

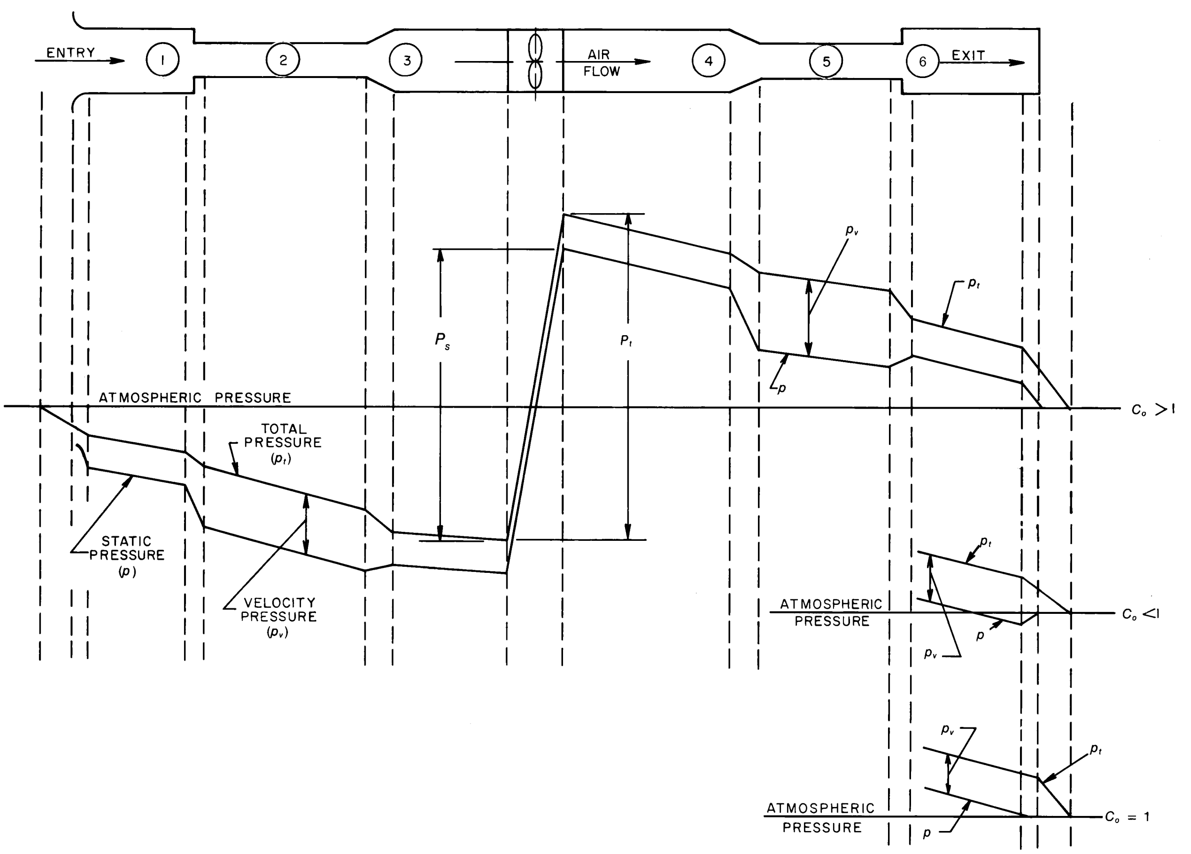


Fig. 9-1 Pressure Changes During Flow in Ducts
(Figure 7, Chapter 21, 2017 ASHRAE Handbook—Fundamentals)

The total pressure p_t is a measure of the total available energy at a cross section. In any duct system, the total pressure always decreases in the direction of the airflow. Static and velocity pressure are mutually convertible and either increase or decrease in the flow direction.

Total and static pressure changes in a simplified fan/duct system are shown in Figure 9-1. This illustrative system consists of a fan with both supply and return air ductwork. Also shown are the total and static pressure gradients referenced to atmospheric pressure.

For all constant-area sections, such as ducts and elbows, the total and static pressure losses are equal. In the case of ducts, the losses are entirely frictional, while the losses in constant-area fitting are frictional and dynamic.

At diverging sections 3 and 6, the velocity pressure decreases, the absolute value of the total pressure decreases, and the absolute value of the static pressure may increase. The increase in static pressure as shown at these sections is known as **static regain**.

At converging sections 1 and 4, the velocity pressure increases in the direction of airflow, and the absolute value of both the total and static pressure decreases. The static or total pressure loss from upstream to downstream is the difference in total/static pressure between the two sections.

At the exit, total pressure loss depends on the shape of the fitting and the flow characteristics. Exit loss coefficients can be greater than, less than, or equal to, one. For this variation of coefficients, the total and static pressure grade lines are shown in Figure 9-1. Note that when the loss coefficient is less than one, the static pressure upstream of the exit is less than atmospheric pressure (negative). The static pressure upstream of the discharge fitting can be calculated by subtracting the upstream velocity pressure from the upstream total pressure.

The entry loss also depends on the shape of the fitting. The total pressure immediately downstream of the entrance equals the difference between the upstream pressure, which is zero (atmospheric pressure), and the loss through the fitting. The static pressure at the entrance is zero, and immediately downstream, the difference between static pressure is negative, algebraically equal to the total pressure (negative) and the velocity pressure (always positive), or $p_s = p_t - p_v$.

The total system resistance to airflow is noted by p_t in Figure 9-1. The fan inlet and outlet system effect factors due to the interaction of the fan and system are not shown; only system resistances are shown. To obtain the fan static pressure p_s requirement for selecting a fan, knowing the systems' total pressure, use

$$p_s = p_t - p_{v,o} \tag{9-5}$$

where the subscript *o* refers to the discharge area of the fan.

Static pressure is used as the basis for system design; total pressure determines the actual mechanical energy that must be supplied to the system. Total pressure always decreases in the direction of flow. Note, however, in Figure 9-1, the static pressure decreases and then increases in the direction of flow. Moreover, it can even become negative (below atmospheric). Therefore, in dealing with static pressure, distinction must always be made between static pressure **loss** and static pressure **change** as a result of conversion of velocity pressure.

9.1.2 Circular Equivalents of Ducts

Rectangular Ducts. An air-handling system is usually sized first for round ducts. Then, if rectangular ducts are desired, duct sizes are selected to provide flow rates equivalent to those of the round ducts originally selected.

Rectangular ducts of aspect ratios not exceeding 8:1 usually have the same friction pressure loss for equal lengths and mean velocities of flow as do circular ducts of the same hydraulic diameter. When duct sizes are expressed in terms of hydraulic diameter (4 times area divided by perimeter), and when equations for friction loss in round and rectangular ducts are equated for equal flow rate and equal length, Equation (9-6a), giving the circular equivalent of a rectangular duct, is obtained.

$$d_c = \frac{1.30ab^{0.625}}{(a+b)^{0.250}} \tag{9-6a}$$

where

a = length of one side of rectangular duct, in. (mm)
b = length of adjacent side of rectangular duct, in. (mm)
d_c = circular equivalent of a rectangular duct for equal friction and capacity, in. (mm)

The circular equivalents of rectangular ducts for equal friction and flow rate for aspect ratios not greater than 8:1, based on Equation (9-6a), are given in Table 9-1a. Note that the mean velocity in a rectangular duct is less than its circular equivalent. Frictional losses are then obtained from Figure 9-2.

Multiplying or dividing the length of each side of a duct by a constant is the same as multiplying or dividing the equivalent round size by the same constant. Thus, if the circular equivalent of an 80 in. by 24 in. duct is required, it is twice that of a 40 in. by 12 in. duct, or $2 \times 23.0 = 46.0$ in.

Flat Oval Ducts. To convert round ducts to flat oval sizes, use Table 9-1b, which is based on Equation (9-6b) (Heyt and Diaz 1975), the circular equivalent of a flat oval duct for equal airflow, resistance, and length.

$$D_e = \frac{1.55AR^{0.625}}{P^{0.250}} \tag{9-6b}$$

where AR is the cross-sectional area of flat oval duct defined as

$$AR = (\pi a^2/4) + a(A-a)$$

and the perimeter P is calculated by

$$P = \pi a + 2(A-a)$$

where

P = perimeter of flat oval duct, in.
A = major axis of flat oval duct, in.
a = minor axis of flat oval duct, in.

Friction losses are then obtained from Figure 9-2.

9.1.3 Friction Losses

Pressure drop in a straight duct is caused by surface friction. This friction loss can be calculated by using the Air Friction Charts (Figure 9-2). The charts were built from the basic flow equation for pressure loss in circular ducts:

$$\Delta p_{fr} = f_D\,(cf_4\,L/D)\,p_v \tag{9-7}$$

where

Δp_{fr} = friction loss in terms of total pressure, in. of water (Pa)
f_D = dimensionless friction factor, which for air-conditioning work depends on Reynolds number and relative roughness of the conduit. Approximate values of f were taken from Moody (1944) where ε = 0.0003 ft (0.09 mm). It numerically equals the reciprocal of the number of duct diameters required to cause a pressure loss equal to one velocity pressure. (Figure 13, Chapter 3, 2013 *ASHRAE Handbook—Fundamentals*)
L = length of duct, ft (m)
D = inside diameter of duct, in. (mm)
p_v = velocity pressure of mean velocity, in. of water (Pa)
cf_4 = conversion factor, 12 (1).

The air friction chart is based on air with a density of 0.075 lb/ft^3 (1.2 kg/m^3) flowing through average, clean, round, galvanized metal ducts with beaded slip couplings on 48 in. (1220 mm) centers.

Variations in air temperature of the order of ±20°F from 70°F (±11°C from 20°C) have little effect on duct friction. Therefore, Figure 9-2 may be used for all air systems with temperatures from 50 to 90°F (10 to 32°C). However, the values found in the charts must be corrected for systems carrying air at much higher temperatures. To determine the friction loss in such systems, the actual flow rate or velocity existing at the nonstandard conditions must be used. For duct materials other than those indicated, and for significant variations in temperature and barometric pressure/elevation, correction factors should be applied to the Air Friction Chart values. Details concerning these correction factors can be found in Chapter 21 of the 2009 *ASHRAE Handbook—Fundamentals*.

9.1.4 Dynamic Losses

Wherever eddying flow is present, a greater loss in total pressure takes place than would occur in steady flow through a

Table 9-1A Equivalent Rectangular Duct Dimension

(Table 3, Chapter 21 of the 2017 ASHRAE Handbook—Fundamentals)

Circular Duct Diameter, in.	Length One Side of Rectangular Duct (*a*), in.																			
	4	**5**	**6**	**7**	**8**	**9**	**10**	**12**	**14**	**16**	**18**	**20**	**22**	**24**	**26**	**28**	**30**	**32**	**34**	**36**
	Length Adjacent Side of Rectangular Duct (*b*), in.																			
5	5																			
5.5	6	5																		
6	8	6																		
6.5	9	7	6																	
7	11	8	7																	
7.5	13	10	8	7																
8	15	11	9	8																
8.5	17	13	10	9																
9	20	15	12	10	8															
9.5	22	17	13	11	9															
10	25	19	15	12	10	9														
10.5	29	21	16	14	12	10														
11	32	23	18	15	13	11	10													
11.5		26	20	17	14	12	11													
12		29	22	18	15	13	12													
12.5		32	24	20	17	15	13													
13		35	27	22	18	16	14	12												
13.5		38	29	24	20	17	15	13												
14			32	26	22	19	17	14												
14.5			35	28	24	20	18	15												
15			38	30	25	22	19	16	14											
16			45	36	30	25	22	18	15											
17				41	34	29	25	20	17	16										
18				47	39	33	29	23	19	17										
19				54	44	38	33	26	22	19	18									
20					50	43	37	29	24	21	19									
21					57	48	41	33	27	23	20									
22					64	54	46	36	30	26	23	20								
23						60	51	40	33	28	25	22								
24						66	57	44	36	31	27	24	22							
25							63	49	40	34	29	26	24							
26							69	54	44	37	32	28	26	24						
27							76	59	48	40	35	31	28	25						
28								64	52	43	38	33	30	27	26					
29								70	56	47	41	36	32	29	27					
30								76	61	51	44	39	35	31	29	28				
31								82	66	55	47	41	37	34	31	29				
32								89	71	59	51	44	40	36	33	31				
33								96	76	64	54	48	42	38	35	33	30			
34									82	68	58	51	45	41	37	35	32			
35									88	73	62	54	48	44	40	37	34	32		
36									95	78	67	58	51	46	42	39	36	34		
37									101	83	71	62	55	49	45	41	38	36	34	
38									108	89	76	66	58	52	47	44	40	38	36	
39										95	80	70	62	55	50	46	43	40	37	36
40										101	85	74	65	58	53	49	45	42	39	37
41										107	91	78	69	62	56	51	47	44	41	39
42										114	96	83	73	65	59	54	50	46	44	41
43										120	102	88	77	69	62	57	53	49	46	43
44											107	93	81	73	66	60	55	51	48	45
45											113	98	86	76	69	63	58	54	50	47
46											120	103	90	80	72	66	61	56	53	49
47											126	108	95	84	76	69	64	59	55	52
48											133	114	100	89	80	73	67	62	58	54
49											140	120	105	93	84	76	70	65	60	56
50											147	126	110	98	88	80	73	68	63	59
51												132	115	102	92	83	76	71	66	61
52												139	121	107	96	87	80	74	69	64
53												145	127	112	100	91	83	77	71	67
54												152	133	117	105	95	87	80	74	70
55													139	123	110	99	91	84	78	72
56													145	128	114	104	95	87	81	75
57													151	134	119	108	98	91	84	78
58													158	139	124	112	102	94	87	81
59													165	145	130	117	107	98	91	85
60													172	151	135	122	111	102	94	88

Table 9-1B Equivalent Rectangular Duct Dimension

(Table 4, Chapter 21 of the 2017 ASHRAE Handbook—Fundamentals)

Circular Duct Diameter, in.	Minor Axis *a*, in.																
	3	4	5	6	7	8	9	10	11	12	14	16	18	20	22	24	30
	Major Axis *A*, in.																
5	8																
5.5	9	7															
6	11,12																
6.5	14	9,10	8														
7	17	12		8													
7.5	19	13	10	9													
8	22	15	11														
8.5		17,18	13,14	11	10												
9		20,21		12		10											
9.5			16	14	12	11											
10			18,19	15	13												
10.5			21	17	15	13											
11				19	16	14	12	12									
11.5				20	18		14	13									
12				22,23		16,17	15	14									
12.5				25,26	20,21				14								
13				28		19	17	16		14							
13.5				30,31		21	18										
14				33		22	20	18	16,17	15							
14.5				34,36		24,25	22	19									
15				37		27	23	21	19	17,18							
15.5				41					20								
16				44,47		30,32		23,24	22	20							
17						33,35,36		26,27	24,25	21,23	20						
18						38,39		29,30		25,26	22						
19						43,46		32,34,35		28,29	23	22					
20						49,52		37,38,40		31,32	25,27	24					
21						55,58		41		34	28,30	25	23,24				
22						61		45,48		36,37,39	31,33	27,29	26				
23								51,54		40,43	34,36	30	27	26			
24								57,60		47	39	32,33	29	28			
25								63		50	42	35	31,32	29			
26								67,70,73		53,56	45	38	34,35	31			
27								76,79		59,62	49	41	37	33			
28										65	52,55	44	40	36			
29										69,72	58	47	43	39	35		
30										75,78	61,64	51,54	46	42	38		
31										81	67	57	49	45	41	37	
32											71,74	60	53	48		40	
33											77,80	66	56	51	44		
34												69	59,62		47	43	
35												73,76	65	55,58	50	46	
36												79	68	61	53	49	
37													71	64	57	52	43
38													75,78	67	60	55	
39													81	70,73	63	59	46
40														77	66,69	62	49
41														80	72	65	52
42															75	68	55
43															79	71	
44															82	74	58
45																77	61
46																81	65
47																	68
48																	71
49																	74
50																	77
51																	80
52																	81

*Table based on Equation (26).

similar length of straight duct having a uniform cross section. The amount of this loss in excess of straight duct friction is termed **dynamic loss**.

Dynamic losses result from flow disturbances caused by fittings that change the airflow path's direction and/or area. These fittings include entries, exits, transitions, and junctions. Idelchik et al. (1986) discuss parameters affecting fluid resistance of fittings and presents loss coefficients in three forms: tables, curves, and equations.

The following dimensionless coefficient is used for fluid resistance since this coefficient has the same value in dynamically similar streams (i.e., streams with geometrically similar stretches), equal values of Reynolds number, and equal values of other criteria necessary for dynamic similarity. The fluid resistance coefficient represents the ratio of total pressure loss to velocity pressure at the referenced cross section.

$$C = \frac{\Delta p_j}{\rho(V/1097)^2} = \frac{\Delta p_j}{p_v} \qquad \text{(9-8 I-P)}$$

$$C = \frac{\Delta p_j}{\rho(V^2/2)} = \frac{\Delta p_j}{p_v} \qquad \text{(9-8 SI)}$$

where

C = local loss coefficient, dimensionless

Δp_j = fitting total pressure loss, in. of water (Pa)

ρ = density, lb_m/ft^3 (kg/m^3)

V = velocity, fpm (m/s)

p_v = velocity pressure, in. of water (Pa)

Dynamic losses occur along a duct length and cannot be separated from frictional losses. For ease of calculation, dynamic losses are assumed to be concentrated at a section (local) and to exclude friction. Frictional losses must be considered only for relatively long fittings. Generally, fitting friction losses are accounted for by measuring duct lengths from the centerline of one fitting to that of the next fitting. For fittings closely coupled (less than six hydraulic diameters apart), the flow pattern entering subsequent fittings differs from the flow pattern used to determine loss coefficients. Adequate data for these situations are unavailable.

For all fittings, except junctions, calculate the total pressure loss Δp_j at a section by

$$\Delta p_j = C_o p_{v,o} \qquad \text{(9-9)}$$

where the subscript *o* is the cross section at which the velocity pressure is referenced. The dynamic loss is based on the actual velocity in the duct, not the velocity in an equivalent noncircular duct. Where necessary (unequal area fittings), convert a loss coefficient from section *o* to section *i* by Equation (9-10), where *V* is the velocity at the respective sections.

$$C_i = \frac{C_o}{(V_i/V_o)^2} \qquad \text{(9-10)}$$

For converging and diverging flow junctions, total pressure losses through the straight (main) section are calculated as

$$\Delta p_j = C_{c,s} p_{v,c} \qquad \text{(9-10a)}$$

For total pressure losses through the branch section,

$$\Delta p_j = C_{c,b} p_{v,c} \qquad \text{(9-10b)}$$

where $P_{v,c}$ is the velocity pressure at the common section *c*, and $C_{c,s}$ and $C_{c,b}$ are losses for the straight (main) and branch flow paths, respectively, each referenced to the velocity pressure at section *c*. To convert junction local loss coefficients referenced to straight and branch velocity pressures, use Equation (9-10c).

$$C_i = \frac{C_{c,i}}{(V_i/V_c)^2} \qquad \text{(9-10c)}$$

where

C_i = local loss coefficient referenced to section being calculated (see subscripts), dimensionless

$C_{c,i}$ = straight ($C_{c,s}$) or branch ($C_{c,b}$) local loss coefficient referenced to dynamic pressure at the common section, dimensionless

V_i = velocity at section to which C_i is being referenced

V_c = velocity at common section

Subscripts:

b = branch

s = straight (main) section

c = common section

The junction of two parallel streams moving at different velocities is characterized by turbulent mixing of the streams, accompanied by pressure losses. In the course of this mixing, an exchange of momentum takes place between the particles moving at different velocities, finally resulting in the equalization of the velocity distributions in the common stream. The jet with higher velocity loses a part of its kinetic energy by transmitting it to the slower moving jet. The loss in total pressure before and after mixing is always large and positive for the higher velocity jet and increases with an increase in the amount of energy transmitted to the lower velocity jet. Consequently, the local loss coefficient, defined by Equation (9-8), will always be positive. The energy stored in the lower velocity jet increases as a result of mixing. The loss in total pressure and the local loss coefficient can, therefore, also have negative values for the lower velocity jet (Idelchik et al. 1986).

9.1.5 Ductwork Sectional Losses

Total pressure loss in a duct section is calculated by combining Equations (9-7) and (9-8) in terms of Δp, where ΣC is the sum of local loss coefficients within the duct section. Each fitting loss coefficient must be referenced to that section's velocity pressure.

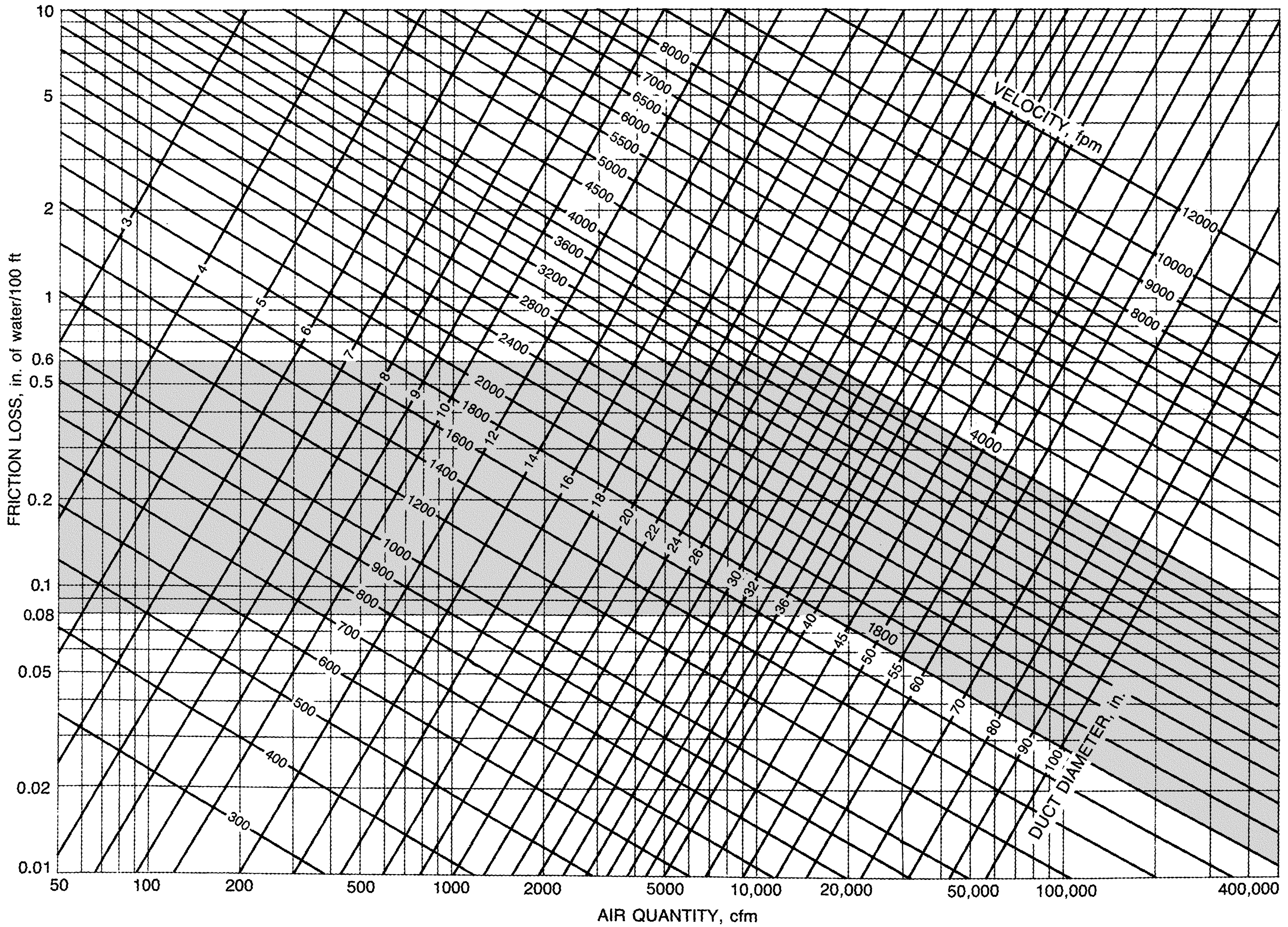


Fig. 9-2 Friction Chart for Round Duct (ρ = 0.075 lb_m/ft^3 and ε = 0.0003 ft)

Shaded Area is Normal Design Region.

(Figure 10, Chapter 21, 2017 ASHRAE Handbook—Fundamentals)

$$\Delta p = \left(\frac{12fL}{D} + \Sigma C\right)\rho(V/1097)^2 \qquad \text{(9-11 I-P)}$$

$$\Delta p = \left(\frac{1000fL}{D} + \Sigma C\right)\rho(V^2/2) \qquad \text{(9-11 SI)}$$

9.1.6 System Analysis

Since $p_t = p_s + p_v$

$$\Delta p_t = p_{t,1} - p_{t,2} \qquad \text{(9-12)}$$

where Δp_t is the change in total pressure in the direction of airflow between stations 1 and 2. For all main and branch ducts in a system, including both supply and return air ductwork, Equation (9-12) may be used for any fitting/duct or section of ductwork. The path with the greatest resistance to flow, usually the longest and most complicated, is known as the critical path. The minimum total pressure to be developed by the fan is the summation of duct/fitting losses throughout the **critical path** of the system, plus the fan system effect factors.

Thus,

$$\Delta p_t = \sum_{i=1}^{n} \Delta p_i + \mathrm{SEF}_s + \mathrm{SEF}_d \qquad \text{(9-13)}$$

where

Δp_t = fan's total pressure, in. of water (Pa)
Δp_i = component total pressure loss, in. of water (Pa)
n = number of ducts/fittings in critical path of system
SEF_s = system effect factor due to fan inlet conditions, in. of water (Pa)
SEF_d = system effect due to fan outlet conditions, in. of water (Pa)

For some simple systems, SEF_s and SEF_d can be negligible. However, for typical systems, these quantities must be estimated. Also, duct heat transfer must be considered in duct system design. Information concerning these items is given in Chapter 21 of the 2017 *ASHRAE Handbook—Fundamentals*.

9.1.7 Duct Design

This discussion refers to central station ducts for commercial and industrial heating, ventilating, and air-conditioning systems. The equal friction and static regain methods given yield the static pressure required to overcome the resistance of the ductwork, including the supply outlets and return intakes. The fan selected for the duct system must produce not only this pressure but also the additional pressure required by the central equipment such as washers or spray chambers, heating or cooling coils, and filters. Pressure losses of these components should be obtained from the manufacturers' catalogs.

Rules that should be followed in the design of ducts are

1. Convey air as directly as possible at the permissible velocities to obtain the desired results with minimum noise and greatest economy of power, material, and space.
2. Avoid sudden changes in air direction or velocity. When sudden changes are necessary at bends, use turning vanes to minimize the pressure loss.
3. Where the greatest air-carrying capacity per unit area of sheet metal is desired, make rectangular ducts as nearly square as possible. Avoid aspect ratios (ratio of width to depth) greater than 8 to 1. Where possible, maintain a ratio of 4 to 1 or less.
4. Ducts should be constructed of smooth material, such as steel or aluminum sheet metal. For ducts made from other materials, allow for the change in roughness.
5. A reasonable estimate of the flow resistances offered by the system can be obtained through the following design procedures. However, in actual installations, resistances may vary considerably from the calculated values because of variation in the smoothness of materials, types of joints used, and the ability to fabricate the system in accordance with the design. Select fans and motors to provide at least a slight factor of safety, and install dampers in each branch outlet for balancing the system.
6. Avoid obstructing ducts with piping, conduits, or structural members. Unavoidable duct obstructions must be streamlined with an easement or a tear-drop, the length of which should be at least three times the thickness of the tear-drop.

9.1.8 Design Velocities

It is impossible to give specific rules for selecting duct velocities and duct shapes (rectangular, round, or oval) without considering cost and system constraints. An ideal design has minimum owning and operating costs when all constraints on the design are considered. The velocity and friction loss rate ranges indicated in Figure 9-2 are offered as preliminary design values. In the constant velocity design method, ignore the limits on friction loss rate. Do not use Figure 9-2 indiscriminately, as noise generation throughout a system increases as the velocity increases.

A summary of recommended velocities for HVAC components encountered in built-up systems is presented in Table 9-2. Final component selection should be based on the various chapters in 2016 *ASHRAE Handbook—HVAC Systems and Equipment* or from manufacturers.

9.1.9 Design Methods

The transmission of air at high velocities has gained wide acceptance in comfort air-conditioning and ventilation systems. This acceptance is due partly to improved fans and special sound attenuation and control equipment and partly to improved design and installation methods based on a better understanding of the design and installation of high-velocity air-conditioning systems.

The design of high-velocity duct systems involves a compromise between reduction of duct size and the consequent need for higher fan power. While the duct size and air velocities are governed in large part by the space available, the maximum velocities (given later in this section) should not be exceeded without carefully examining all factors.

Table 9-2 Typical Design Velocities for HVAC Components

(Table 6, Chapter 21 of the 2013 ASHRAE Handbook—Fundamentals)

Component	Face Velocity, fpm
Terminal Units	Inlet Velocity, Maximum: 2000 Velocity Pressure, Minimum: 0.02 in. water
Louvers[a]	
Intake	
7000 cfm and greater	400
Less than 7000 cfm	See Figure 17
Exhaust	
5000 cfm and greater	500
Less than 5000 cfm	See Figure 17
Filters[b]	
Panel filters	
Viscous impingement	200 to 800
Dry extended-surface	
Flat (low efficiency)	Duct velocity
Pleated media (intermediate efficiency)	Up to 750
HEPA	250 to 500
Renewable media filters	
Moving-curtain viscous impingement	500
Heating Coils[c]	
Steam and hot water	500 to 1000 200 min., 1500 max.
Electric	
Open wire	Refer to mfg. data
Finned tubular	Refer to mfg. data
Dehumidifying Coils[d]	400 to 500
Air Washers[e]	
Spray type	Refer to mfg. data
Cell type	Refer to mfg. data
High-velocity spray type	1200 to 1800

[a] Based on assumptions presented in text.

[b] Abstracted from Chapter 29, 2012 *ASHRAE Handbook—HVAC Systems and Equipment*.

[c] Abstracted from Chapter 27, 2012 *ASHRAE Handbook—HVAC Systems and Equipment*.

[d] Abstracted from Chapter 23, 2012 *ASHRAE Handbook—HVAC Systems and Equipment*.

[e] Abstracted from Chapter 41, 2012 *ASHRAE Handbook—HVAC Systems and Equipment*.

The following principles and laws apply to all duct systems regardless of the design method used or the numerical values obtained.

1. The measure of the amount of energy required to move air from one location to another is the change (decrease) in the total pressure within the system.
2. The total pressure p_t at any location within a system is a measure of the total mechanical energy at that location. It is the sum of the static pressure and the velocity pressure.
3. In any duct system, the total pressure always decreases in the direction of airflow.
4. In any system having two or more branches, the losses in total pressure between the fan and the end of each branch are the same.
5. Static pressure and velocity pressure are mutually convertible and can either increase or decrease in the direction of flow. For example, in a straight run of duct, the static pressure decreases, the velocity pressure remains constant, and the total pressure (their sum) decreases. In a gradually diverging section (area increase), the velocity pressure decreases, the static pressure increases, and the total pressure remains the same (neglecting the small friction loss).

The most common methods of air duct design are (1) equal friction, (2) velocity reduction, and (3) static regain and variations such as total pressure. No single duct design method automatically produces the most economical duct system for all conditions; the system design with the minimum owning and operating cost depends on both the application and ingenuity of the designer.

Equal Friction Method. The principle of this method is to size a system's ductwork for a constant pressure loss per unit length of duct. At higher airflow rates, it may be necessary to limit the velocity so as not to generate objectionable noises. For an initial design, the friction loss per unit length of duct for the corresponding recommended velocities is given in Figure 9-2.

Once the system is sized, the total pressure losses for the main and branch sections from junction-to-junction/fan/terminal may be calculated and the total pressure grade line plotted. To optimize a system, additional designs are necessary to establish the annual system and power cost curves, and to find the minimum point on the total owning and operating cost curve. For system costs, only the incremental differences due to the redesign need to be considered.

After the system has been designed and the total pressure grade line plotted, sections of ductwork may be redesigned to achieve an approximate balance at the junctions without relying entirely on balancing dampers.

Velocity Reduction Method. This method consists of selecting the velocity at the fan discharge and designing for progressively lower velocities in the main duct at each junction or branch duct. For the selected velocities and known airflow rates, the various duct diameters may be read directly from Figure 9-2 and the equivalent rectangular sizes obtained from Equation (9-6). The return air ductwork is sized similarly, starting with the highest velocity at the fan suction and decreasing progressively in the direction of the return intakes. With the ducts sized and the fittings known, the total pressure losses can be calculated, the pressure gradients plotted, and the maximum pressure loss or critical path of the system established.

A refinement of this method involves sizing the branch ducts to dissipate the pressure available at the entrance to each. The pressure loss of the ductwork between the fan and first branch take-off is subtracted from the total fan pressure to obtain the available pressure at the first junction. Through trial, a branch velocity is found that results in the branch pressure loss being equal to or less than the pressure available. The procedure is repeated for each branch.

If the fan is specified so that the total pressure available for the system is known, the method consists of finding, through trial, the velocities in the main and branch ducts that will result in a pressure loss equal to or less than the pressure available. The branch ducts are sized as discussed previously.

Static Regain Method. In the static regain method, the ducts are sized so that the increase in static pressure (static regain) at each take-off offsets the pressure loss of the succeeding section of ductwork. This method is especially suited for high-velocity installations having long runs with many take-offs or terminal units. Approximately the same static pressure exists at the entrance to each branch, which simplifies outlet or terminal unit selection and system balancing. With the ductwork sized by this method, the system's total pressure losses can be calculated. A disadvantage of this method is that excessively large ducts (low velocities) result at the ends of long duct runs.

The total pressure design method is an adaptation of the static regain method. This method is advantageous since the intermediate system pressures and control of duct sizes and velocities are known.

9.1.10 Duct Design Procedures

The general procedure for duct design is as follows:

1. Study the plans of the building and arrange the supply/return outlets to provide proper distribution of air within each space. Adjust calculated actual air quantities for duct heat gains or losses and duct leakage. Adjust the supply, return, and/or exhaust air quantities to meet space pressurization requirements.
2. Select outlet sizes from manufacturers' data. (Refer to Chapter 20, Space Air Diffusion, of the 2017 *ASHRAE Handbook—Fundamentals*.)
3. Sketch the duct system, connect the supply outlets and return intakes with the central station apparatus, and avoid all structural obstructions and equipment. Be aware of system space allocations. Use round duct whenever feasible.
4. Divide the system into sections and number each section. A duct system should be divided at all points where flow, size, or shape changes. Assign fittings to the section toward the supply and return (or exhaust) terminals.
5. Size ducts by the selected design method. Calculate system total pressure loss; then select the fan (refer to Chapter 21 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*).
6. Lay out the system in detail. If duct routing and fittings vary significantly from the original design, recalculate the pressure losses. Reselect the fan if necessary.
7. Resize duct sections to approximately balance pressures at each junction.
8. Analyze the design for objectionable noise levels and specify sound attenuators as necessary.

9.1.11 Fitting Loss Coefficients

A duct fitting database, which includes more than 220 round, rectangular, and flat oval fittings, is available from ASHRAE (2016) in electronic form with the capability to be linked to duct design programs. The fittings are numbered (coded) as shown in Table 9-3.

For convenience, a selection of dynamic loss coefficients from various sources is given in Table 9-4 for use with the problems in this book. **Use of the 2016 *ASHRAE Duct Fitting Database* is recommended for actual projects.**

9.1.12 Automated Duct Design

Duct design calculations have been automated by computers. Automated duct design offers features such as (1) standardization of duct design, (2) stored loss coefficients for fittings, (3) stored duct construction and thermal insulation standards, (4) balancing analysis, (5) noise analysis, (6) duct heat gain or loss analysis, (7) material takeoffs, and (8) documentation. For available duct design programs and hardware requirements, refer to Chapter 21 of the 2017 *ASHRAE Handbook—Fundamentals.*

Table 9-3 Duct Fitting Codes

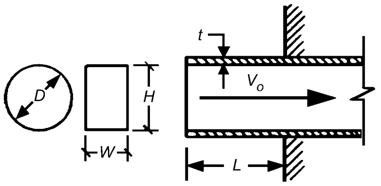
Fitting Function	Geometry	Category	Sequential Number
S: **S**upply	D: round (**D**iameter)	1. Entries	1,2,3...*n*
		2. Exits	
E: **E**xhaust/Return	R: **R**ectangular	3. Elbows	
		4. Transitions	
C: **C**ommon (supply and return)	F: **F**lat oval	5. Junctions	
		6. Obstructions	
		7. Fan and system interactions	
		8. Duct-mounted equipment	
		9. Dampers	
		10. Hoods	
		11. Straight duct	

Table 9-4 Fitting Loss Coefficients

ENTRIES

1-1 Duct Mounted in Wall (Hood, Nonenclosing, Flanged, and Unflanged) (Idelchik et al. 1986, Diagram 3-1)

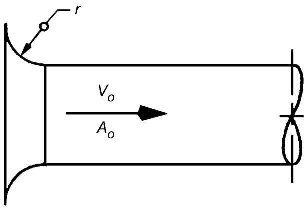
General. If entry has a screen, use Fitting 6-7 to calculate screen resistance.



Rectangular: $D = 2HW/(H + W)$

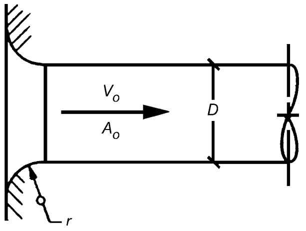
	C_o						
	L/D						
t/D	0	0.002	0.01	0.05	0.2	0.5	≥1.0
≈0	0.50	0.57	0.68	0.80	0.92	1.0	1.0
0.02	0.50	0.51	0.52	0.55	0.66	0.72	0.72
≥0.05	0.50	0.50	0.50	0.50	0.50	0.50	0.50

1-2 Smooth Converging Bellmouth Without End Wall (Idelchik et al. 1986, Diagram 34)



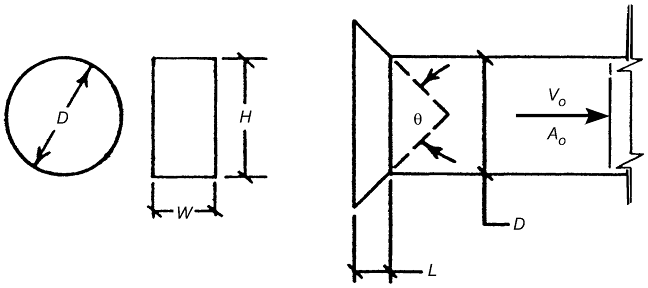
r/D	0	0.01	0.02	0.03	0.04	0.05
C_o	1.0	0.87	0.74	0.61	0.51	0.40
r/D	0.06	0.08	0.10	0.12	0.16	≥0.20
C_o	0.32	0.20	0.15	0.10	0.06	0.03

1-3 Smooth Converging Bellmouth with End Wall (Idelchik et al. 1986, Diagram 3-4)



r/D	0	0.01	0.02	0.03	0.04	0.05
C_o	0.50	0.44	0.37	0.31	0.26	0.22
r/D	0.06	0.08	0.10	0.12	0.16	≥0.20
C_o	0.20	0.15	0.12	0.09	0.06	0.03

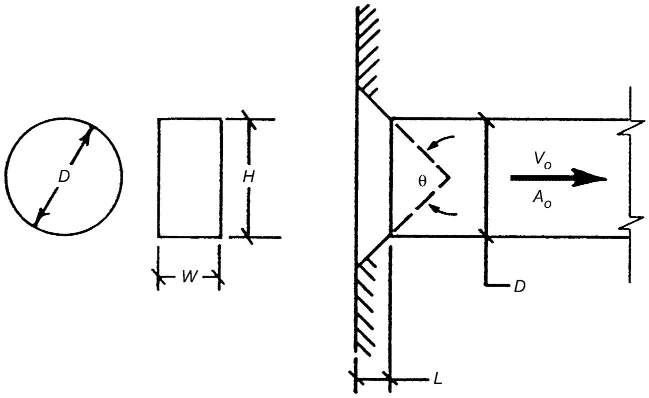
1-4 Conical Converging Bellmouth Without End Wall, Round and Rectangular



Rectangular: $D = 2HW/(H + W)$

	C_o									
	θ, degrees									
L/D	0	10	20	30	45	60	90	120	150	180
0.025	1.0	0.96	0.93	0.90	0.85	0.80	0.72	0.64	0.57	0.50
0.05	1.0	0.93	0.86	0.80	0.73	0.67	0.60	0.56	0.52	0.50
0.10	1.0	0.80	0.67	0.55	0.46	0.41	0.41	0.43	0.46	0.50
0.25	1.0	0.68	0.45	0.30	0.21	0.17	0.21	0.28	0.38	0.50
0.60	1.0	0.46	0.27	0.18	0.14	0.13	0.19	0.27	0.37	0.50
1.0	1.0	0.32	0.20	0.14	0.11	0.10	0.16	0.24	0.35	0.50

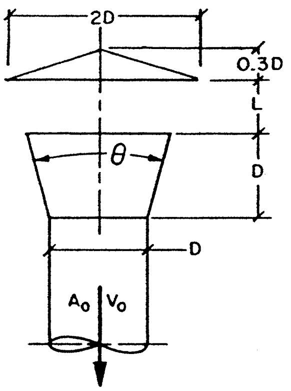
1-5 Conical Converging Bellmouth with End Wall, Round and Rectangular (Idelchik et al. 1986, Diagram 3-7)



Rectangular: $D = 2HW/(H + W)$

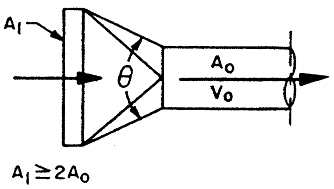
	C_o									
	θ, degrees									
L/D	0	10	20	30	45	60	90	120	150	180
0.025	0.50	0.47	0.45	0.43	0.41	0.40	0.42	0.44	0.46	0.50
0.05	0.50	0.45	0.41	0.36	0.32	0.30	0.34	0.39	0.44	0.50
0.075	0.50	0.42	0.35	0.30	0.25	0.23	0.28	0.35	0.43	0.50
0.10	0.50	0.39	0.32	0.25	0.21	0.18	0.25	0.33	0.41	0.50
0.15	0.50	0.37	0.27	0.20	0.16	0.15	0.23	0.31	0.40	0.50
0.60	0.50	0.27	0.18	0.13	0.11	0.12	0.20	0.30	0.40	0.50

1-6 Intake Hood (Idelchik et al. 1986, Diagram 3-18)



θ, degrees	C_o								
	L/D								
	0.1	0.2	0.3	0.4	0.5	0.6	0.8	1.0	4.0
0	2.63	1.83	1.53	1.39	1.31	1.19	1.08	1.06	1.0
15	1.32	0.77	0.60	0.48	0.41	0.30	0.28	0.25	0.25

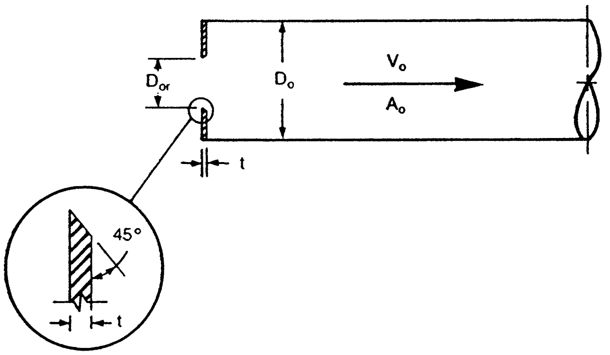
1-7 Hood, Tapered, Flanged or Unflanged (Brandt and Steffy 1946)



θ is major angle for rectangular hoods

Hood Shape: Round										
θ, degrees	0	20	40	60	80	100	120	140	160	180
C_o	1.0	0.11	0.06	0.09	0.14	0.18	0.27	0.32	0.43	0.50
Hood Shape: Square or Rectangular										
θ, degrees	0	20	40	60	80	100	120	140	160	180
C_o	1.0	0.19	0.13	0.16	0.21	0.27	0.33	0.43	0.53	0.62

1-8 Orifice, Sharp-Edged, Inlet Duct (Idelchik et al. 1986; Diagrams 3-12, 3-14, and 4-19)



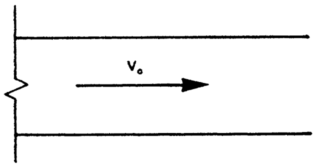
$t/D_{or} \leq 0.015$
$\mathrm{Re} = D_oV_o/\nu$
A_{or}: orifice area

Re × 10^{-3}	C_o							
	A_{or}/A_o							
	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
4	45	18	7.9	3.9	2.3	1.3	0.83	0.51
10	49	20	9.2	4.4	2.7	1.5	0.96	0.59
20	50	21	9.3	4.9	2.9	1.6	1.1	0.65
100	55	23	11.0	5.6	3.3	1.9	1.2	0.75

EXITS

General. If exit has a screen, use Fitting 6-7 to calculate screen resistance.

2-1 Exit, Abrupt, Round and Rectangular (Idelchik et al. 1986, Diagram 11-1)



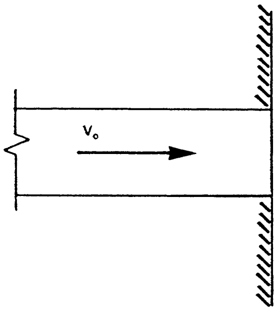
Uniform Velocity Distribution

$C_o = 1.0$

Exponential, Sinusoidal, Asymmetrical, and Parabolic Velocity Distribution

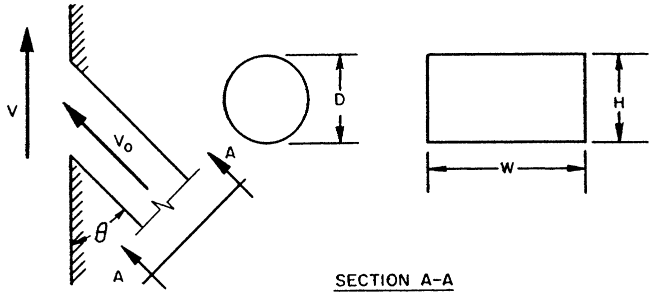
C_o varies from 1.0 to 3.67. For details, consult Idelchik (1986), Diagram 11-1.

2-2 Exit, Abrupt, Round and Rectangular, with End (Idelchik et al. 1986, Diagrams 5-2 and 5-4)



$C_o = 0.88$

2-3 Exit, Duct Flush with Wall, Flow along Wall (Idelchik et al. 1986, Diagram 11-2)



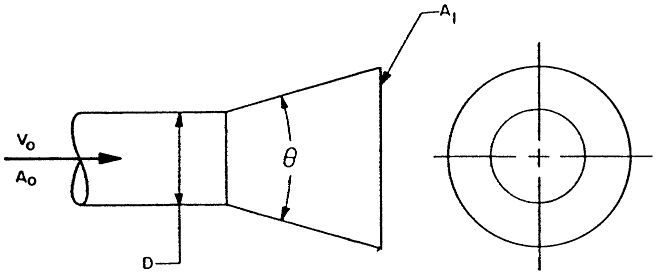
Round

	C_o				
θ,	V/V_o				
degrees	0	0.5	1.0	1.5	2.0
≤45	1.0	1.0	1.1	1.3	1.6
60	1.0	0.90	1.1	1.4	1.6
90	1.0	0.80	0.95	1.4	1.7
120	1.0	0.80	0.95	1.3	1.7
150	1.0	0.82	0.83	1.0	1.3

Rectangular

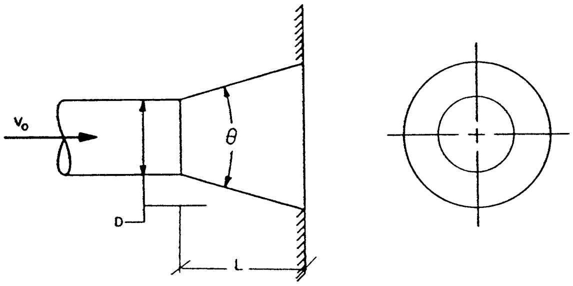
		C_o				
Aspect Ratio		V/V_o				
(H/W)	θ, degrees	0	0.5	1.0	1.5	2.0
≤0.2	≤90	1.0	0.95	1.2	1.5	1.8
	120	1.0	1.1	1.1	1.4	1.9
	150	1.0	0.95	0.95	1.4	1.8
0.5-2.0	≤45	1.0	1.0	1.1	1.3	1.6
	60	1.0	0.90	1.1	1.4	1.6
	90	1.0	0.80	0.95	1.4	1.7
	120	1.0	0.80	0.95	1.3	1.7
	150	1.0	0.82	0.83	1.0	1.3
≥5	45	1.0	0.92	0.93	1.1	1.3
	60	1.0	0.87	0.87	1.0	1.3
	90	1.0	0.82	0.80	0.97	1.2
	120	1.0	0.80	0.76	0.90	0.98

2-4 Exit, Round, Diverging (Idelchik et al. 1986, Diagram 11-3)



	C_o						
	θ, degrees						
A_1/A_o	8	10	14	20	30	45	≥60
2	0.36	0.33	0.37	0.51	0.90	1.0	1.0
4	0.24	0.21	0.28	0.40	0.70	0.99	1.0
6	0.20	0.19	0.26	0.37	0.67	0.99	1.0
10	0.18	0.16	0.24	0.36	0.68	0.99	1.0
16	0.16	0.16	0.20	0.36	0.66	0.99	1.0

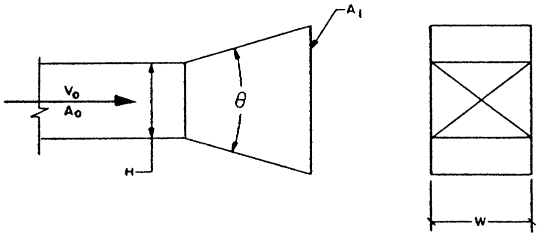
2-5 Exit, Round, with End Wall Transition (Idelchik et al. 1986, Diagram 5-8)



θ = optimum angle

L/D	0.5	1.0	2.0	3.0	4.0	5.0	6.0	8.0	10	12	14
θ, degrees	34	24	16	13	11	10	9	8	7	6	6
C_o	0.41	0.32	0.24	0.20	0.17	0.15	0.14	0.12	0.11	0.11	0.10

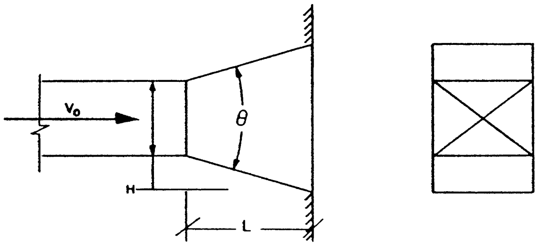
2-6 Exit, Rectangular, Two Sides Parallel, Diverging, Symmetrical (Idelchik et al. 1986, Diagram 11-6)



$0.5 \le H/W \le 2.0$

	C_o						
	θ, degrees						
A_1/A_o	8	10	14	20	30	45	≥60
2	0.50	0.51	0.56	0.63	0.80	0.96	1.0
4	0.34	0.38	0.48	0.63	0.76	0.91	1.0
6	0.32	0.34	0.41	0.56	0.70	0.84	0.96

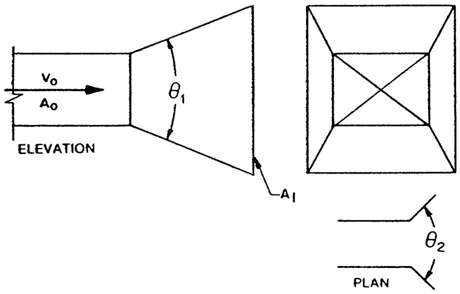
2-7 Exit, Rectangular, with Wall, Two Sides Parallel, Symmetrical, Diverging (Idelchik et al. 1986, Diagram 5-10)



θ = optimum angle

L/H	0.5	1.0	2.0	3.0	4.0	5.0	6.0	8.0	10	12	14
θ, degrees	50	35	25	21	18	16	15	13	12	11	10
C_o	0.53	0.44	0.35	0.31	0.28	0.25	0.24	0.22	0.20	0.19	0.19

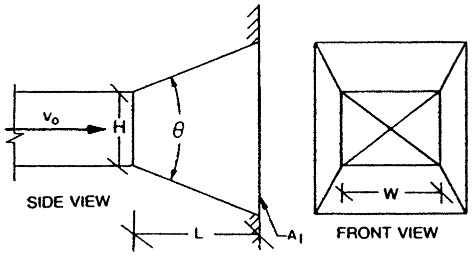
2-8 Exit, Rectangular, Pyramidal, Diverging (Idelchik et al. 1986, Diagram 11-5)



θ is larger of θ_1 and θ_2

	C_o						
	θ, degrees						
A_1/A_o	8	10	14	20	30	45	≥60
2	0.65	0.68	0.74	0.82	0.92	1.1	1.1
4	0.53	0.60	0.69	0.78	0.90	1.0	1.1
6	0.50	0.57	0.66	0.77	0.91	1.0	1.1
10	0 45	0.53	0.64	0.74	0.85	0.97	1.1

2-9 Exit, Rectangular, with Wall, Pyramidal, Diverging (Idelchik et al. 1986, Diagram 5-9)

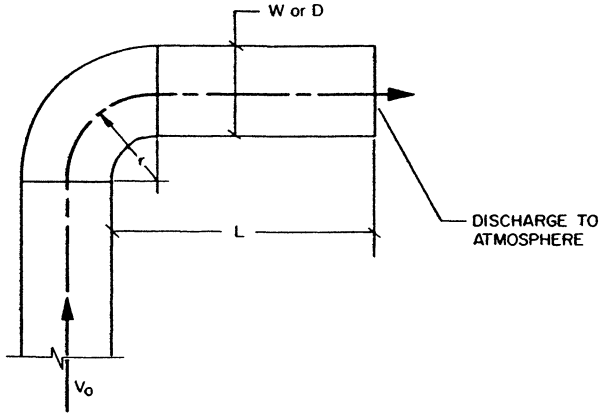


$D = 2HW/(H + W)$

θ = optimum angle

L/D	0.5	1.0	2.0	3.0	4.0	5.0	6.0	8.0	10	12	14
θ, degrees	26	19	13	11	9	8	7	6	6	5	5
C_o	0.49	0.40	0.30	0.26	0.23	0.21	0.19	0.17	0.16	0.15	0.14

2-10 Exit, Discharge to Atmosphere from a 90° Elbow, Rectangular and Round (Note: Elbow Loss Included) (Idelchik et al. 1986, Diagram 11-14)



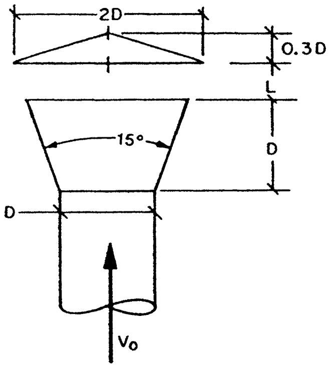
Rectangular

	C_o									
	L/W									
r/W	0	0.5	1.0	1.5	2.0	3.0	4.0	6.0	8.0	12.0
0	3.0	3.1	3.2	3.0	2.7	2.4	2.2	2.1	2.1	2.0
0.75	2.2	2.2	2.1	1.8	1.7	1.6	1.6	1.5	1.5	1.5
1.0	1.8	1.5	1.4	1.4	1.3	1.3	1.2	1.2	1.2	1.2
1.5	1.5	1.2	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1
2.5	1.2	1.1	1.1	1.0	1.0	1.0	1.0	1.0	1.0	1.0

Round (r/D = 1.0)

L/D	0.9	1.3
C_o	1.5	1.4

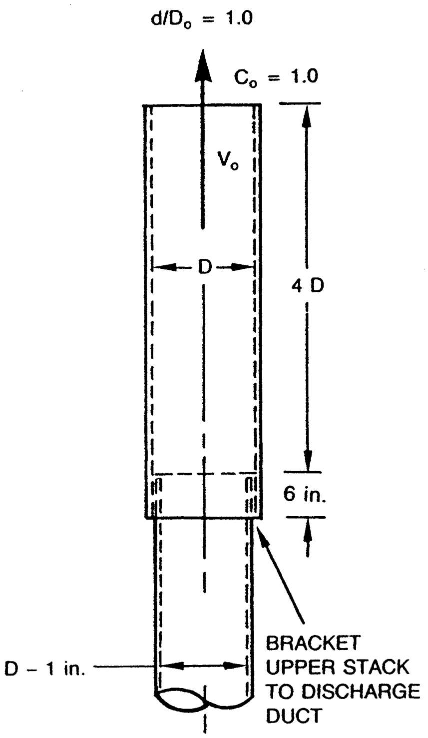
2-11 Exhaust Hood (Idelchik et al. 1986, Diagram 11-16)



Poor Design—Should Not Be Used (see Chapter 14, Figure 13)

L/D	0.1	0.2	0.25	0.3	0.35	0.4	0.5	0.6	0.8	1.0
C_o	2.6	1.2	1.0	0.80	0.70	0.65	0.60	0.60	0.60	0.60

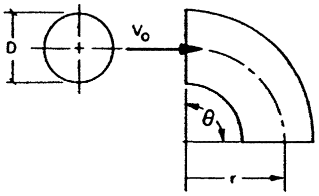
2-12 Stackhead (Idelchik et al. 1986, Diagram 11-23)



d/D	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
C_o	130	41	17	8.1	4.4	2.6	1.6	1.0

ELBOWS

3-1 Elbow, Smooth Radius (Die Stamped), Round (Locklin 1950, Equation A-10)



$C_o = K_q C'_o$

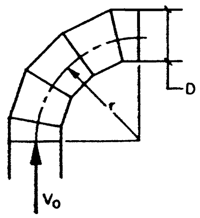
Coefficients for 90° Elbows

r/D	0.5	0.75	1.0	1.5	2.0	2.5
C'_o	0.71	0.33	0.22	0.15	0.13	0.12

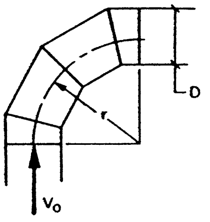
Angle Correction Factors K_q (Idelchik et al. 1986, Diagram 6-1):

θ, degrees	0	20	30	45	60	75	90	110	130	150	180
K_θ	0	0.31	0.45	0.60	0.78	0.90	1.00	1.13	1.20	1.28	1.40

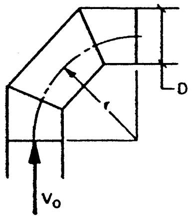
3-2 Elbows; 3-, 4-, and 5-Pieces, Round (Locklin 1950, Figure 10)



5-PIECE



4-PIECE



3-PIECE

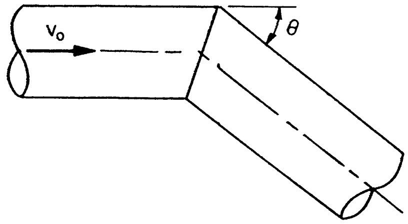
Coefficients for 90° Elbows (C'_o)

No. of Pieces	r/D			
	0.75	**1.0**	**1.5**	**2.0**
5	0.46	0.33	0.24	0.19
4	0.50	0.37	0.27	0.24
3	0.54	0.42	0.34	0.33

Angle Correction Factors K_q (Idelchik et al. 1986, Diagram 6-1)

θ, degrees	0	20	30	45	60	75	90	110	130	150	180
K_θ	0	0.31	0.45	0.60	0.78	0.90	1.00	1.13	1.20	1.28	1.40

3-3 Elbow, Mitered, Round (Idelchik et al. 1986, Diagram 6-5)



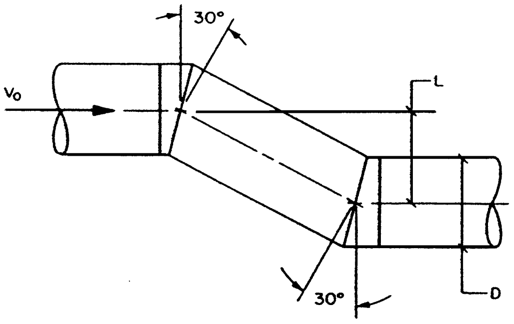
$C_o = K_{Re} C'_o$

θ, degrees	20	30	45	60	75	90
C'_o	0.08	0.16	0.34	0.55	0.81	1.2

Reynolds Number Correction Factors: ***M/hc***

Re × 10^{-4}	**1**	**2**	**3**	**4**	**6**	**8**	**10**	**≥14**
K_{Re}	1.40	1.26	1.19	1.14	1.09	1.06	1.04	1.0

3-4 Elbows, 30,° Z-Shaped, Round



$C_o = K_{Re} C'_o$

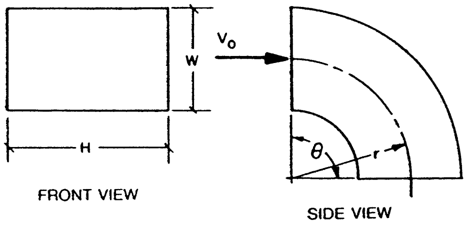
L/D	0	0.5	1.0	1.5	2.0	2.5	3.0
C'_o	0	0.15	0.15	0.16	0.16	0.16	0.16

Reynolds Number Correction Factors

Re × 10^{-4}	**1**	**2**	**3**	**4**	**6**	**8**	**10**	**≥14**
K_{Re}	1.40	1.26	1.19	1.14	1.09	1.06	1.04	1.0

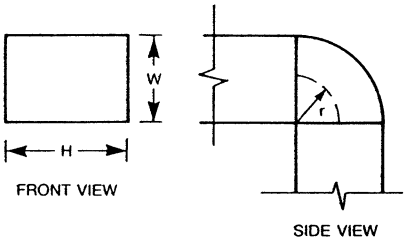
3-5 Elbow, Without Vanes, Rectangular (Idelchik et al. 1986, Diagram 6-1)

Smooth Radius



$C_o = K_\theta K_{Re} C'_o$

90°, Sharp Throat Radius Heel ($r/W = 0.5$)



$C_o = K_{Re} C'_o$

Coefficients for 90° Elbows (C'_o)

	H/W										
r/W	**0.25**	**0.5**	**0.75**	**1.0**	**1.5**	**2.0**	**3.0**	**4.0**	**5.0**	**6.0**	**8.0**
0.5	1.3	1.3	1.2	1.2	1.1	1.0	1.0	1.1	1.1	1.2	1.2
0.75	0.57	0.52	0.48	0.44	0.40	0.39	0.39	0.40	0.42	0.43	0.44
1.0	0.27	0.25	0.23	0.21	0.19	0.18	0.18	0.19	0.20	0.21	0.21
1.5	0.22	0.20	0.19	0.17	0.15	0.14	0.14	0.15	0.16	0.17	0.17
2.0	0.20	0.18	0.16	0.15	0.14	0.13	0.13	0.14	0.14	0.15	0.15

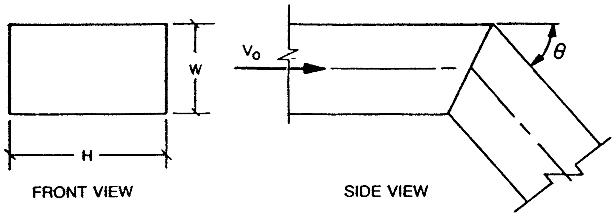
Angle Correction Factor

θ, degrees	0	20	30	45	60	75	90	110	130	150	180
K_θ	0	0.31	0.45	0.60	0.78	0.90	1.00	1.13	1.20	1.28	1.40

Reynolds Number Correction Factor (K_{Re})

	Re × 10^{-4}								
r/W	**1**	**2**	**3**	**4**	**6**	**8**	**10**	**14**	**≥20**
0.5	1.40	1.26	1.19	1.14	1.09	1.06	1.04	1.0	1.0
≥0.75	2.0	1.77	1.64	1.56	1.46	1.38	1.30	1.15	1.0

3-6 Elbow, Mitered, Rectangular (Idelchik et al. 1986, Diagram 6-5)



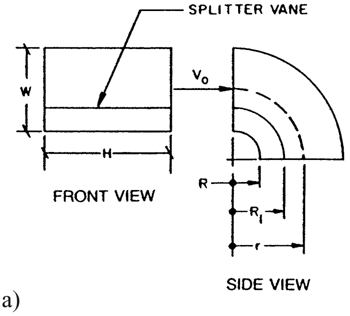
$C_o = K_{\mathrm{Re}} C'_o$

	C'_o										
θ,	*H/W*										
degrees	**0.25**	**0.5**	**0.75**	**1.0**	**1.5**	**2.0**	**3.0**	**4.0**	**5.0**	**6.0**	**8.0**
20	0.08	0.08	0.08	0.07	0.07	0.07	0.06	0.06	0.05	0.05	0.05
30	0.18	0.17	0.17	0.16	0.15	0.15	0.13	0.13	0.12	0.12	0.11
45	0.38	0.37	0.36	0.34	0.33	0.31	0.28	0.27	0.26	0.25	0.24
60	0.60	0.59	0.57	0.55	0.52	0.49	0.46	0.43	0.41	0.39	0.38
75	0.89	0.87	0.84	0.81	0.77	0.73	0.67	0.63	0.61	0.58	0.57
90	1.3	1.3	1.2	1.2	1.1	1.1	0.98	0.92	0.89	0.85	0.83

Reynolds Number Corrections Factors

Re × 10^{-4}	1	2	3	4	6	8	10	≥14
K_{Re}	1.40	1.26	1.19	1.14	1.09	1.06	1.04	1.0

3-7 Elbow, Smooth Radius with Splitter Vanes, Rectangular (Locklin 1950, Equation 10; Madison and Parker 1936)



a. One Splitter Vane

$C_o = K_\theta C'_o$
$R_1 = R/CR$

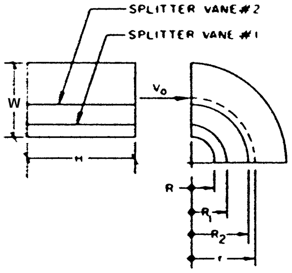
where

R = throat radius
R_1 = splitter vane radius
CR = CURVE RATIO (values from Table 3-7.a)
K_θ = angle factor (see Fitting 3-1 for values)

Table 3-7.a Coefficients for Elbows with One Splitter Vane:

			C'_o										
			H/W										
R/W	***r/W***	***CR***	**0.25**	**0.5**	**1.0**	**1.5**	**2.0**	**3.0**	**4.0**	**5.0**	**6.0**	**7.0**	**8.0**
0.05	0.55	0.218	0.52	0.40	0.43	0.49	0.55	0.66	0.75	0.84	0.93	1.0	1.1
0.10	0.60	0.302	0.36	0.27	0.25	0.24	0.30	0.35	0.39	0.42	0.46	0.49	0.52
0.15	0.65	0.361	0.28	0.21	0.18	0.19	0.20	0.22	0.25	0.26	028	0.30	0.32
0.20	0.70	0.408	0.22	0.16	0.14	0.14	0.15	0.16	0.17	0.18	0.19	0.20	0.21
0.25	0.75	0.447	0.18	013	0.11	0.11	0.11	0.12	0.13	0.14	0.14	0.15	0.15
0.30	0.80	0.480	0.15	0.11	0.09	0.09	0.09	0.09	0.10	0.10	0.11	0.11	0.12
0.35	0.85	0.509	0.13	0.09	0.08	0.07	0.07	0.08	0.08	0.08	0.08	0.09	0.09
0.40	0.90	0.535	0.11	0.08	0.07	0.06	0.06	0.06	0.06	0.07	0.07	0.07	0.07
0.45	0.95	0.557	0.10	0.07	0.06	0.05	0.05	0.05	0.05	0.05	0.06	0.06	0.06
0.50	1.00	0.577	0.09	0.06	0.05	0.05	0.04	0.04	0.04	0.05	0.05	0.05	0.05

b. Two Splitter Vanes



$C_o = K_\theta C'_o$
$R_1 = R/CR$
$R_2 = R_1/CR = R/CR^2$

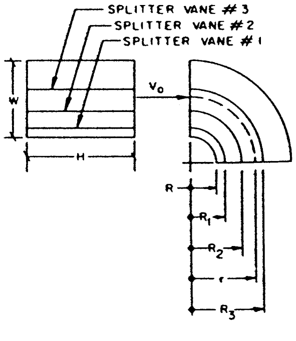
where

R = throat radius
R_1 = splitter vane # 1 radius
R_2 = splitter vane # 2 radius
CR = CURVE RATIO (value from Table 3-7.b)
K_θ = angle factor (see Note 3 for values)

Table 3-7.b Coefficients for Elbow with Two Splitter Vanes:

			C'_o										
			H/W										
R/W	***r/W***	***CR***	**0.25**	**0.5**	**1.0**	**1.5**	**2.0**	**3.0**	**4.0**	**5.0**	**6.0**	**7.0**	**8.0**
0.05	0.55	0.362	0.26	0.20	0.22	0.25	0.28	0.33	0.37	0.41	0.45	0.48	0.51
0.10	0.60	0.450	0.17	0.13	0.11	0.12	0.13	0.15	0.16	0.17	0.19	0.20	0.21
0.15	0.65	0.507	0.12	0.09	0.08	0.08	0.08	0.09	0.10	0.10	0.11	0.11	0.11
0.20	0.70	0.550	0.09	0.07	0.06	0.05	0.06	0.06	0.06	0.06	0.07	0.07	0.07
0.25	0.75	0.585	0.08	0.05	0.04	0.04	0.04	0.04	0.05	0.05	0.05	0.05	0.05
0.30	0.80	0.613	0.06	004	0.03	0.03	0.03	0.03	0.03	0.03	0.04	0.04	0.04

c. Three Splitter Vanes



$C_o = K_\theta C'_o$
$R_1 = R/CR$
$R_2 = R_1/\mathrm{CR} = R/\mathrm{CR}^2$
$R_3 = R_2/\mathrm{CR} = R/\mathrm{CR}^3$

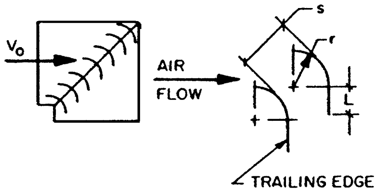
where

R = throat radius
R_1 = splitter vane # 1 radius
R_2 = splitter vane # 2 radius
R_3 = splitter vane # 3 radius
CR = curve ratio (value from Table 3-7.c)
K_θ = angle factor (see Note 3 for values)

Table 3-7.c Coefficients for Elbow with Three Splitter Vanes (C'_o):

			H/W										
R/W	***r/W***	***CR***	**0.25**	**0.5**	**1.0**	**1.5**	**2.0**	**3.0**	**4.0**	**5.0**	**6.0**	**7.0**	**8.0**
0.05	0.55	0.467	0.11	0.10	0.12	0.13	0.14	0.16	0.18	0.19	0.21	0.22	0.23
0.10	0.60	0.549	0.07	0.05	0.06	0.06	0.06	0.07	0.07	0.08	0.08	0.08	0.09

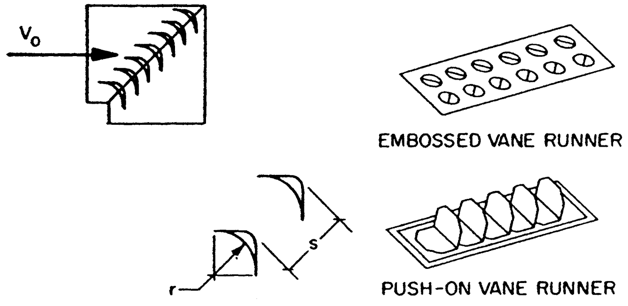
3-8 Elbow, Mitered, with Single-Thickness Vanes, Rectangular (Rozell 1974)



Design	Dimensions, in.			
No.	r	s	L	C_o
1[a]	2.0	1.5	0.75	0.12
2	4.5	2.25	0	0.15
3	4.5	3.25	1.60	0.18

[a] When extension of trailing edge is not provided for this vane, losses are approximately unchanged for single elbows, but increase considerably for elbows in series.

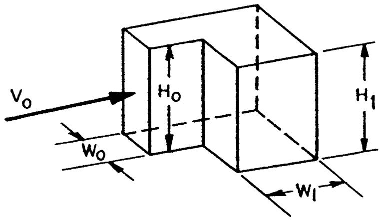
3-9 Elbow, Mitered, with Double-Thickness Vanes, Rectangular



(Rozell 1974)

			C_o				
Design	Dimensions, in.		Velocity V_o, fpm				
No.	r	s	1000	2000	3000	4000	Remarks
1	2.0	1.5	0.27	0.22	0.19	0.17	Embossed Vane Runner
2	2.0	1.5	0.33	0.29	0.26	0.23	Push-On Vane Runner
3	2.0	2.13	0.38	0.31	0.27	0.24	Embossed Vane Runner
4	4.5	3.25	0.26	0.21	0.18	0.16	Embossed Vane Runner

3-10 Elbow, Variable Inlet/Outlet Areas, Rectangular (Idelchik et al. 1986, Diagram 6-4)

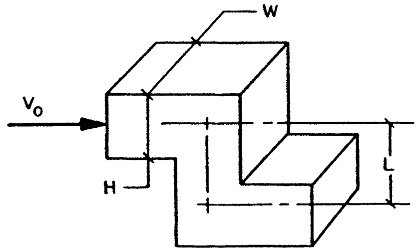


	C'_o					
	W_1/W_o					
H_o/W_o	0.6	0.8	1.2	1.4	1.6	2.0
0.25	1.8	1.4	1.1	1.1	1.1	1.1
1.0	1.7	1.4	1.0	095	0.90	0.84
4.0	1.5	1.4	0.81	0.76	0.72	0.66
∞	1.5	1.0	0.69	0.63	0.60	0.55

Reynolds Number Correction Factor

$\mathrm{Re} \times 10^{-4}$	1	2	3	4	6	8	10	≥14
K_{Re}	1.40	1.26	1.19	1.14	1.09	1.06	1.04	1.0

3-11 Elbows, 90°, Z-Shaped, Rectangular (Idelchik et al. 1986, Diagram 6-11)



$C = K\,K_{\mathrm{Re}}C'_o$

Coefficients for $W/H = 1.0$

L/H	0	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
C'_o	0	0.62	0.90	1.6	2.6	3.6	4.0	4.2	4.2	4.2
L/H	2.4	2.8	3.2	4.0	5.0	6.0	7.0	9.0	10.0	∞
C'_o	3.7	3.3	3.2	3.1	2.9	2.9	2.8	2.7	2.5	2.3

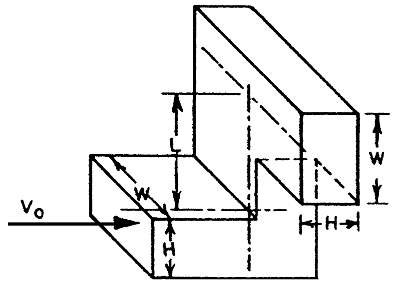
For W/H Values Other Than 1.0, Apply the Following Factor

W/H	0.25	0.50	0.75	1.0	1.5	2.0	3.0	4.0	6.0	8.0
K	1.10	1.07	1.04	1.0	0.95	0.90	9.83	0.78	0.72	0.70

Reynolds Number Correction Factor

$\mathrm{Re} \times 10^{-4}$	1	2	3	4	6	8	10	≥14
K_{Re}	1.40	1.26	1.19	1.14	1.09	1.06	1.04	1.0

3-12 Combined 90° Elbows Lying in Different Planes, Rectangular



(Idelchik et al. 1986, Diagram 6-11)

$C_o = K\,K_{\mathrm{Re}}C'_o$

Coefficients for Square Ducts

L/W	0	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
C'_o	1.2	2.4	2.9	3.3	3.4	3.4	3.4	3.3	3.2	3.1
L/W	2.4	2.8	3.2	4.0	5.0	6.0	7.0	9.0	10.0	∞
C'_o	3.2	3.2	3.2	3.0	2.9	2.8	2.7	2.5	2.4	2.3

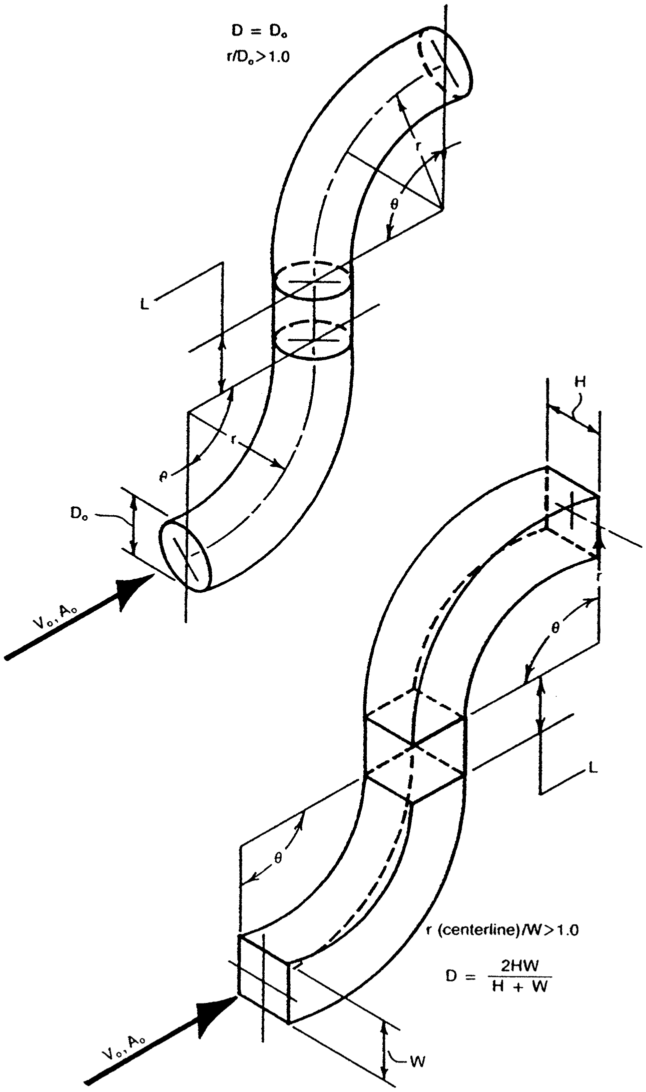
Apply the Following Factor for Other Than H/W = 1.0

H/W	0.25	0.50	0.75	1.0	1.5	2.0	3.0	4.0	6.0	8.0
K	1.10	1.07	1.04	1.0	0.95	0.90	0.83	0.78	0.72	0.70

Reynolds Number Correction Factor

$\mathrm{Re} \times 10^{-4}$	1	2	3	4	6	8	10	≥14
K_{Re}	1.40	1.26	1.19	1.14	1.09	1.06	1.04	1.0

3-13 Offset, S-Shaped (Gooseneck), Rectangular and Round (Idelchik et al. 1986, Diagram 6-16)



$C_o = KC'_o$

where

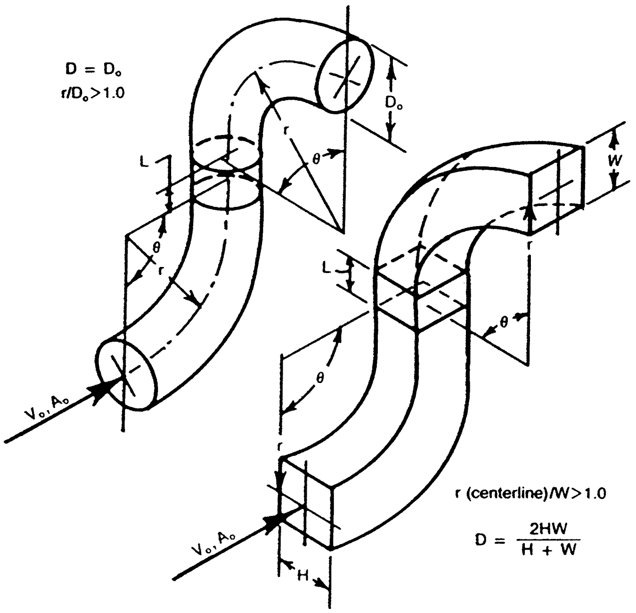
C_o = offset loss coefficient

C'_o = single elbow loss coefficient (see Fittings 3-1, 3-2, and 3-5)

	K							
θ,	***L*/*D***							
degrees	**0**	**1**	**2**	**3**	**4**	**6**	**8**	**10**
15	0.20	0.42	0.60	0.78	0.94	1.16	1.20	1.15
30	0.40	0.65	0.88	1.16	1.20	1.18	1.12	1.06
45	0.60	1.06	1.20	1.23	1.20	1.08	1.03	1.08
60	1.05	1.38	1.37	1.28	1.15	1.06	1.16	1.30
75	1.50	1.58	1.46	1.30	1.27	1.30	1.37	1.47
90	1.70	1.67	1.40	1.37	1.38	1.47	1.55	1.63

	K						
θ,	***L*/*D***						
degrees	**12**	**14**	**16**	**18**	**20**	**25**	**≥40**
15	1.08	1.05	1.02	1.00	1.10	1.25	2.0
30	1.06	1.15	1.28	1.40	1.50	1.70	2.0
45	1.17	1.30	1.42	1.55	1.65	1.80	2.0
60	1.42	1.54	1.66	1.76	1.85	1.95	2.0
75	1.57	1.68	1.75	1.80	1.88	1.97	2.0
90	1.70	1.76	1.82	1.88	1.92	1.98	2.0

3-14 Offset, S-Shaped in Two Planes 90° Apart, Rectangular and Round (Idelchik et al. 1986, Diagram 6-16)



$C_o = KC'_o$

where

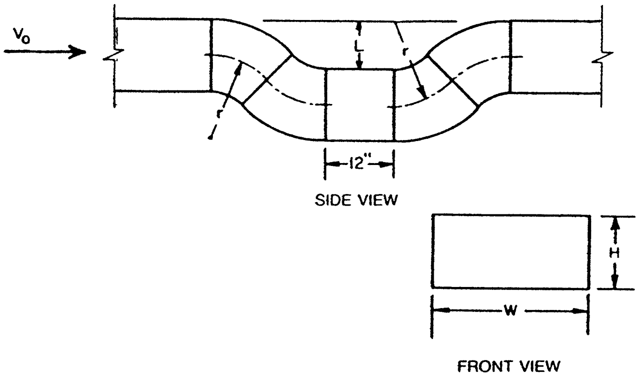
C_o = offset loss coefficient

C'_o = single elbow loss coefficient (see Fittings 3-1, 3-2, and 3-5)

	K						
θ,	***L*/*D***						
degrees	**0**	**1**	**2**	**3**	**4**	**6**	**8**
60	2.0	1.90	1.50	1.35	1.30	1.20	1.25
90	2.0	1.80	1.60	1.55	1.55	1.65	1.80

	K					
θ,	***L*/*D***					
degrees	**10**	**12**	**14**	**20**	**25**	**40**
60	1.50	1.63	1.73	1.85	1.95	2.0
90	1.90	1.93	1.98	2.0	2.0	2.0

3-15 Elbows (4), 45°, Smooth Radius, Rectangular, Arranged to Go Around an Obstruction (SMACNA 1981, Table 6-14K)

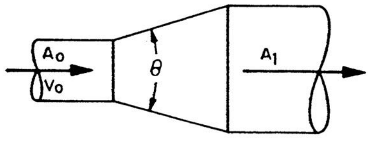


$W/H = 4$, $r/H = 1.5$, $L = 1.5H$

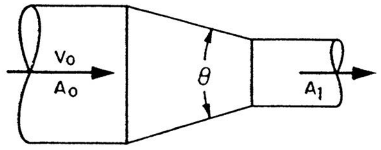
V_o, fpm	800	1200	1600	2000	2400
C_o	0.18	0.22	0.24	0.25	0.26

TRANSITIONS

4-1 Transition, Round (Idelchik et al. 1986, Diagrams 5-2 and 5-22)



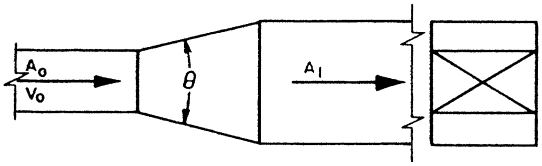
$A_o/A_1 < 1$



$A_o/A_1 > 1$

	C_o									
	θ, degrees									
A_o/A_1	10	15	20	30	45	60	90	120	150	180
0.06	0.21	0.29	0.38	0.60	0.84	0.88	0.88	0.88	0.88	0.88
0.1	0.21	0.28	0.38	0.59	0.76	0.80	0.83	0.84	0.83	0.83
0.25	0.16	0.22	0.30	0.46	0.61	0.68	0.64	0.63	0.62	0.62
0.5	0.11	0.13	0.19	0.32	0.33	0.33	0.32	0.31	0.30	0.30
1	0	0	0	0	0	0	0	0	0	0
2	0.20	0.20	0.20	0.20	0.22	0.24	0.48	0.72	0.96	1.0
4	0.80	0.64	0.64	0.64	0.88	1.1	2.7	4.3	5.6	6.6
6	1.8	1.4	1.4	1.4	2.0	2.5	6.5	10	13	15
10	5.0	5.0	5.0	5.0	6.5	8.0	19	29	37	43

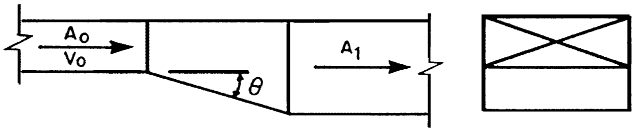
4-2 Transition, Rectangular, Two Sides Parallel, Symmetrical (Idelchik et al. 1986, Diagram 5-5)[a]



	C_o									
	θ, degrees									
A_o/A_1	10	15	30	30	45	60	90	120	150	180
0.06	0.26	0.27	0.40	0.56	0.71	0.86	1.00	0.99	0.98	0.98
0.1	0.24	0.26	0.36	0.53	0.69	0.82	0.93	0.93	0.92	0.91
0.25	0.17	0.19	0.22	0.42	0.60	0.68	0.70	0.69	0.67	0.66
0.5	0.14	0.13	0.15	0.24	0.35	0.37	0.38	0.37	0.36	0.35
1	0	0	0	0	0	0	0	0	0	0
2	0.23	0.20	0.20	0.20	0.24	0.28	0.54	0.78	1.0	1.1
4	0.81	0.64	0.64	0.64	0.88	1.1	2.8	4.4	5.7	6.6
6	1.8	1.4	1.4	1.4	2.0	2.5	6.6	10	13	15
10	5.0	5.0	5.0	5.0	6.5	8.0	19	29	37	43

[a] $A_o/A_1 > 1$ is tentative (adapted from Fitting 4-1 data).

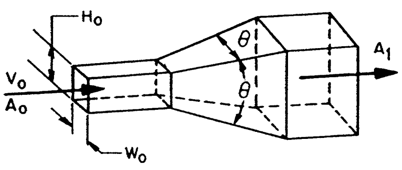
4-3 Transition, Rectangular, Three Sides Straight[a]



	C_o						
	θ, degrees						
A_o/A_1	10	15	20	30	45	60	90
0.06	0.26	0.27	0.40	0.56	0.71	0.86	1.00
0.1	0.24	0.26	0.36	0.53	0.69	0.82	0.93
0.25	0.17	0.19	0.22	0.42	0.60	0.68	0.70
0.5	0.14	0.13	0.15	0.24	0.35	0.37	0.38
1	0	0	0	0	0	0	0
2	0.23	0.20	0.20	0.20	0.24	0.28	0.54
4	0.81	0.64	0.64	0.64	0.88	1.1	2.8
6	1.8	1.4	1.4	1.4	2.0	2.5	6.6
10	5.0	5.0	5.0	5.0	6.5	8.0	19

[a] Tentative (assumed same as Fitting 4-2 data).

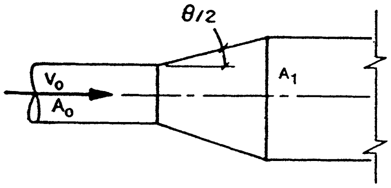
4-4 Transition, Rectangular, Pyramidal (Idelchik et al. 1986, Diagram 5-4)[a]

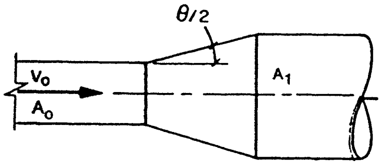


	C_o									
	θ, degrees									
A_o/A_1	10	15	20	30	45	60	90	120	150	180
0.06	0.26	0.30	0.44	0.54	0.53	0.65	0.77	0.88	0.95	0.98
0.1	0.24	0.30	0.43	0.50	0.53	0.64	0.75	0.84	0.89	0.91
0.25	0.20	0.25	0.34	0.36	0.45	0.52	0.58	0.62	0.64	0.64
0.5	0.14	0.15	0.20	0.21	0.25	0.30	0.33	0.33	0.33	0.32
1	0	0	0	0	0	0	0	0	0	0
2	0.23	0.22	0.21	0.20	0.22	0.2	0.49	0.74	0.99	1.1
4	0.84	0.68	0.68	0.64	0.88	1.1	2.7	4.3	5.6	6.6
6	1.8	1.5	1.5	1.4	2.0	2.5	6.5	10	13	15
10	5.0	5.0	5.1	5.0	6.5	8.0	19	29	37	43

[a] $A_o/A_1 > 1$ is tentative (adapted from Fitting 4-1 data).

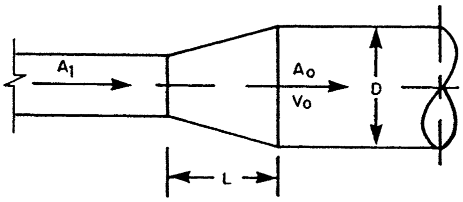
4-5 Transition, Round/Rectangular (Idelchik et al. 1986, Diagram 5-27)

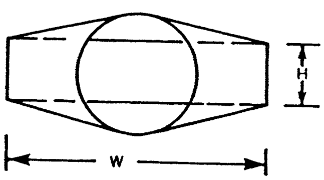




	C_o								
	θ, degrees								
A_1/A_o	8	10	14	20	30	45	60	90	180
2	0.14	0.15	0.20	0.25	0.30	0.33	0.33	0.33	0.30
4	0.20	0.25	0.34	0.45	0.52	0.58	0.62	0.64	0.64
6	0.21	0.30	0.42	0.53	0.63	0.72	0.78	0.79	0.79
≥10	0.24	0.30	0.43	0.53	0.64	0.75	0.84	0.89	0.88

4-6 Transition, Rectangular to Round (Idelchik et al. 1986, Diagram 5-26)





$\text{Re} = DV_o/\nu$
$B = W/H(A_o/A_1)^2$

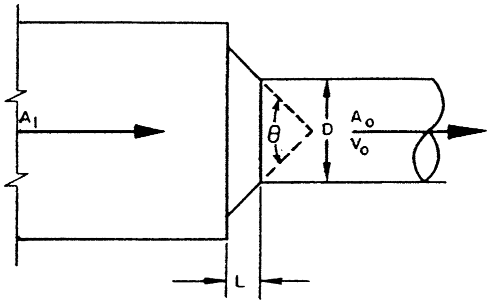
C_o for $A_o > A_1$ and $H < D$

		B							
$\text{Re} \times 10^{-4}$	L/D	0.1	0.5	1	2	5	10	20	50
1	1	0.46	0.48	0.50	0.55	0.70	0.94	1.4	2.9
	1.5	0.46	0.47	0.49	0.53	0.64	0.82	1.2	2.3
	2	0.46	0.47	0.48	0.50	0.58	0.71	0.98	1.8
	3	0.45	0.46	0.46	0.48	0.51	0.57	0.69	1.0
	4	0.45	0.46	0.46	0.46	0.48	0.51	0.56	0.73
	5	0.45	0.46	0.46	0.47	0.49	0.53	0.61	0.84
2	1	0.41	0.43	0.46	0.51	0.66	0.90	1.4	2.9
	1.5	0.41	0.43	0.45	0.48	0.59	0.78	1.1	2.2
	2	0.41	0.42	0.44	0.46	0.54	0.67	0.93	1.7
	3	0.41	0.42	0.42	0.43	0.47	0.53	0.64	0.99
	4	0.41	0.41	0.41	0.42	0.44	0.46	0.52	0.69
	5	0.41	0.41	0.42	0.43	0.45	0.49	0.57	0.80
5	1	0.31	0.33	0.35	0.40	0.55	0.79	1.3	2.8
	1.5	0.31	0.32	0.34	0.38	0.49	0.67	1.0	2.1
	2	0.31	0.32	0.33	0.36	0.43	0.57	0.83	1.6
	3	0.30	0.31	0.31	0.33	0.36	0.42	0.54	0.88
	4	0.30	0.31	0.31	0.31	0.33	0.36	0.41	0.58
	5	0.30	0.31	0.31	0.32	0.34	0.38	0.46	0.70
10	1	0.19	0.21	0.23	0.28	0.43	0.68	1.2	2.6
	1.5	0.19	0.20	0.22	0.26	0.37	0.55	0.92	2.0
	2	0.19	0.20	0.21	0.24	0.31	0.45	0.71	1.5
	3	0.19	0.19	0.20	0.21	0.24	0.30	0.42	0.77
	4	0.18	0.19	0.19	0.19	0.21	0.24	0.29	0.46
	5	0.18	0.19	0.19	0.20	0.22	0.26	0.34	0.58
20	1	0.07	0.09	0.12	0.17	0.31	0.56	1.1	2.5
	1.5	0.07	0.09	0.10	0.14	0.25	0.43	0.80	0.9
	2	0.07	0.08	0.09	0.12	0.20	0.33	0.59	1.4
	3	0.07	0.07	0.08	0.09	0.13	0.18	0.30	0.65
	4	0.07	0.07	0.07	0.08	0.10	0.12	0.18	0.34
	5	0.07	0.07	0.08	0.08	0.11	0.15	0.22	0.46
50	1	0.01	0.03	0.05	0.10	0.25	0.30	0.99	2.5
	1.5	0.01	0.02	0.04	0.08	0.19	0.37	0.74	1.8
	2	0.01	0.02	0.03	0.06	0.13	0.27	0.53	1.3
	3	0	0.01	0.02	0.03	0.06	0.12	0.24	0.58
	4	0	0.01	0.01	0.01	0.03	0.06	0.11	0.28
	5	0	0.01	0.01	0.02	0.04	0.08	0.16	0.40

C_o for $A_o < A_1$ and $H < D$

		B					
$\text{Re} \times 10^{-4}$	L/D	0.1	1	5	10	20	50
1	1	0.27	0.27	0.28	0.29	0.31	0.37
	3	0.27	0.27	0.28	0.29	0.30	0.33
	4	0.27	0.27	0.28	0.28	0.29	0.32
	5	0.27	0.27	0.27	0.27	0.27	0.27
2	1	0.25	0.25	0.26	0.27	0.29	0.35
	3	0.25	0.25	0.25	0.26	0.28	0.33
	4	0.25	0.25	0.25	0.26	0.27	0.30
	5	0.25	0.25	0.25	0.25	0.25	0.25
5	1	0.18	0.18	0.19	0.20	0.22	0.28
	3	0.18	0.18	0.19	0.20	0.22	0.27
	4	0.18	0.18	0.19	0.19	0.20	0.23
	5	0.18	0.18	0.18	0.18	0.18	0.18
10	1	0.11	0.11	0.12	0.13	0.15	0.21
	3	0.11	0.11	0.12	0.13	0.14	0.19
	4	0.11	0.11	0.12	0.12	0.13	0.16
	5	0.11	0.11	0.11	0.11	0.11	0.11
20	1	0.04	0.04	0.05	0.06	0.08	0.14
	3	0.04	0.04	0.05	0.06	0.07	0.12
	4	0.04	0.04	0.05	0.05	0.06	0.09
	5	0.04	0.04	0.04	0.04	0.04	0.04
50	1	0	0	0.01	0.02	0.04	0.10
	3	0	0	0.01	0.02	0.04	0.09
	4	0	0	0.01	0.01	0.02	0.05
	5	0	0	0	0	0	0

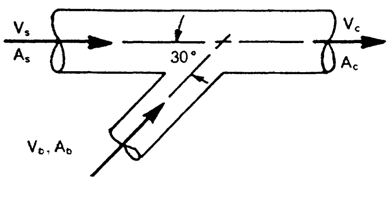
4-7 Transition, Rectangular to Round, Stepped, Conical (Idelchik et al. 1986, Diagram 4-9)



		C_o									
		θ, degrees									
A_o/A_1	L/H	0	10	20	30	45	60	90	120	150	180
0.1	0.025	0.46	0.43	0.42	0.40	0.38	0.37	0.38	0.40	0.43	0.46
	0.05	0.46	0.42	0.38	0.33	0.30	0.28	0.31	0.36	0.41	0.46
	0.075	0.46	0.39	0.32	0.28	0.23	0.21	0.26	0.32	0.39	0.46
	0.1	0.46	0.36	0.30	0.23	0.19	0.17	0.23	0.30	0.38	0.46
	0.15	0.46	0.34	0.25	0.18	0.15	0.14	0.21	0.29	0.37	0.46
	0.3	0.46	0.31	0.22	0.16	0.13	0.13	0.20	0.28	0.37	0.46
	0.6	0.46	0.25	0.17	0.12	0.10	0.11	0.19	0.27	0.36	0.46
0.25	0.025	0.40	0.38	0.36	0.35	0.33	0.32	0.33	0.35	0.37	0.40
	0.05	0.40	0.36	0.33	0.29	0.26	0.24	0.27	0.31	0.35	0.40
	0.075	0.40	0.34	0.28	0.24	0.20	0.19	0.23	0.28	0.34	0.40
	0.1	0.40	0.31	0.26	0.20	0.17	0.14	0.20	0.26	0.33	0.40
	0.15	0.40	0.30	0.22	0.16	0.13	0.12	0.18	0.25	0.32	0.40
	0.3	0.40	0.27	0.19	0.14	0.11	0.11	0.18	0.25	0.32	0.40
	0.6	0.40	0.22	0.14	0.10	0.09	0.10	0.16	0.24	0.32	0.40
0.5	0.025	0.30	0.28	0.27	0.25	0.24	0.24	0.25	0.26	0.27	0.30
	0.05	0.30	0.27	0.24	0.21	0.19	0.18	0.20	0.23	0.26	0.30
	0.075	0.30	0.25	0.21	0.18	0.15	0.14	0.17	0.21	0.25	0.30
	0.1	0.30	0.23	0.19	0.15	0.12	0.11	0.15	0.19	0.24	0.30
	0.15	0.30	0.22	0.16	0.12	0.09	0.09	0.13	0.18	0.24	0.30
	0.3	0.30	0.20	0.14	0.10	0.08	0.08	0.13	0.18	0.24	0.30
	0.6	0.30	0.16	0.11	0.08	0.07	0.07	0.12	0.17	0.23	0.30
0.8	0.025	0.15	0.14	0.13	0.13	0.12	0.12	0.12	0.13	0.14	0.15
	0.05	0.15	0.13	0.12	0.11	0.10	0.09	0.10	0.12	0.13	0.15
	0.075	0.15	0.13	0.10	0.09	0.08	0.07	0.08	0.10	0.13	0.15
	0.1	0.15	0.12	0.10	0.07	0.06	0.05	0.07	0.10	0.12	0.15
	0.15	0.15	0.11	0.08	0.06	0.05	0.04	0.07	0.09	0.12	0.15
	0.3	0.15	0.10	0.07	0.05	0.04	0.04	0.07	0.09	0.12	0.15
	0.6	0.15	0.08	0.05	0.04	0.03	0.04	0.06	0.09	0.12	0.15

JUNCTIONS (Tees, Wyes, Crosses)

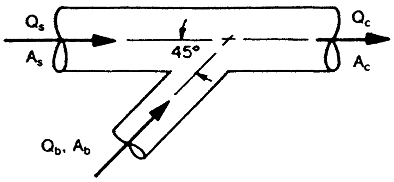
5-1 Wye, 30°, Converging (Idelchik et al. 1986, Diagram 7-1)



	Branch, $C_{c,b}$						
	A_b/A_c						
Q_b/Q_c	0.1	0.2	0.3	0.4	0.6	0.8	1.0
0	–1.0	–1.0	–1.0	–0.9	–0.9	–0.9	–0.9
0.1	0.21	–0.46	–0.57	–0.51	0.53	–0.54	–0.54
0.2	3.1	0.37	–0.06	–0.16	0.23	–0.24	–0.28
0.3	7.6	1.5	0.50	0.15	0.04	–0.06	–0.08
0.4	14	3.0	1.2	0.42	0.19	0.13	0.12
0.5	21	4.6	1.8	0.53	0.24	0.19	0.15
0.6	30	6.4	2.6	0.77	0.35	0.25	0.17
0.7	41	8.5	3.4	0.99	0.42	0.28	0.22
0.8	54	12	4.2	1.2	0.47	0.29	0.25
0.9	58	14	5.3	1.4	0.49	0.29	0.22
1.0	84	17	6.3	1.6	0.49	0.21	0.15

	Main, $C_{c,s}$						
	A_b/A_c						
Q_b/Q_c	0.1	0.2	0.3	0.4	0.6	0.8	1.0
0	0	0	0	0	0	0	0
0.1	0.02	0.11	0.13	0.15	0.16	0.17	0.17
0.2	–0.33	0.01	0.13	0.19	0.24	0.27	0.29
0.3	–1.1	–0.25	–0.01	0.10	0.22	0.30	0.35
0.4	–2.2	–0.75	–0.30	–0.05	0.17	0.26	0.36
0.5	–3.6	–1.4	–0.70	–0.35	0	0.21	0.32
0.6	–5.4	–2.4	–1.3	–0.70	–0.20	0.06	0.25
0.7	–7.6	–3.4	–2.0	–1.2	–0.50	–0.15	0.10
0.8	–10	–4.6	–2.7	–1.8	–0.90	–0.43	–0.15
0.9	–13	–6.2	–3.7	–2.6	–1.4	–0.80	–0.45
1.0	–16	–7.7	–4.8	–3.4	–1.9	–1.2	–0.75

5-2 Wye, 45° Converging, Round (Idelchik et al. 1986, Diagram 7-2)

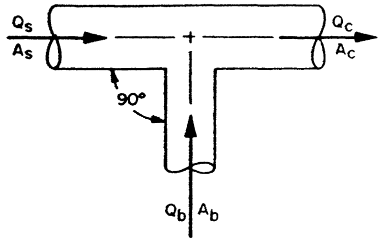


$A_s = A_c$

	$C_{c,b}$						
	A_b/A_c						
Q_b/Q_c	0.1	0.2	0 3	0.4	0.6	0.8	1.0
0	– 1.0	–1.0	–1.0	–0.90	–0.90	–0.80	–0.90
0.1	0.24	–0.45	–0.56	–0.50	–0.52	–0.53	–0.53
0.2	3.2	0.54	–0.02	–0.14	–0.21	–0.23	–0.23
0.3	8.0	1.6	0.60	0.23	0.06	0	–0.02
0.4	14	3.2	1.3	0.52	0.25	0.18	0.15
0.5	22	5.0	2.1	0.65	0.33	0.25	0.22
0.6	32	7.0	3.0	0.91	0.81	0.61	0.51
0.7	43	9.2	3.9	1.2	0.56	0.39	0.33
0.8	56	12	4.9	1.5	0.66	0.39	0.36
0.9	71	15	6.2	1.8	0.72	0.44	0.35
1.0	87	19	7.4	2.0	0.78	0.44	0.32

	$C_{c,s}$						
	A_b/A_c						
Q_b/Q_c	0.1	0.2	0.3	0.4	0.6	0.8	1.0
0	0	0	0	0	0	0	0
0.1	0.05	0.12	0.14	0.16	0.17	0.17	0.17
0.2	–0.20	0.17	0.22	0.27	0.27	0.29	0.31
0.3	–0.76	–0.13	0.08	0.20	0.28	0.32	0.40
0.4	–1.7	–0.50	–0.12	0.08	0.26	0.36	0.41
0.5	–2.8	–1.0	–0.49	–0.13	0.16	0.30	0.40
0.6	–4.3	–1.7	–0.87	–0.45	–0.04	0.20	0.33
0.7	–6.1	–2.6	–1.4	–0.85	–0.25	0.08	0.25
0.8	–8.1	3.6	–2.1	–1.3	–0.55	–0.17	0.06
0.9	–10	4.8	–2.8	–1.9	–0.88	–0.40	–0.18
1.0	–13	–6.1	–3.7	–2.6	–1.4	–0.77	–0.42

5-3 Tee, Converging, Round (Idelchik et al. 1986, Diagram 7-4)



$A_s = A_c$

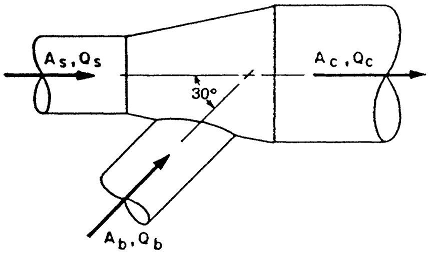
Branch, $C_{c,b}$

	A_b/A_c						
Q_s/Q_c	**0.1**	**0.2**	**03**	**0.4**	**0.6**	**0.8**	**1.0**
0	–1.0	–1.0	–1.0	–0.90	–0.90	–0.90	–0.90
0.1	0.40	–0.37	–0.51	–0.46	–0.50	–0.51	–0.52
0.2	3.8	0.72	0.17	–0.02	–0.14	–0.18	–0.24
0.3	9.2	2.3	1.0	0.44	0.21	0.11	–0.08
0.4	16	4.3	2.1	0.94	0.54	0.40	0.32
0.5	26	6.8	3.2	1.1	0.66	0.49	0.42
0.6	37	9.7	4.7	1.6	0.92	0.69	0.57
0.7	43	13	6.3	2.1	1.2	0.88	0.72
0.8	65	17	7.9	2.7	1.5	1.1	0.86
0.9	82	21	9.7	3.4	1.8	1.2	0.99
1.0	101	26	12	4.0	2.1	1.4	1.1

Main

Q_s/Q_c	**0**	**0.1**	**0.2**	**0.3**	**0.4**	**0.5**	**0.6**	**0.7**	**0.8**	**0.9**	**1.0**
$C_{c,s}$	0	0.16	0.27	0.38	0.46	0.53	0.57	0.59	0.60	0.59	0.55

5-4 Wye, 30° Converging, Round, Conical Main (Sepsy and Pies 1973)



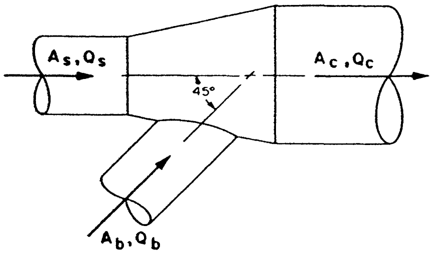
Branch, $C_{c,b}$

		Q_b/Q_s									
A_s/A_c	A_b/A_c	**0.2**	**0.4**	**0.6**	**0.8**	**1.0**	**1.2**	**1.4**	**1.6**	**1.8**	**2.0**
0.3	0.2	–2.4	–0.11	1.8	3.4	4.8	6.0	7.1	8.0	8.9	9.7
	0.3	–2.8	–1.3	0.14	0.72	1.4	2.0	2.4	2.8	3.2	3.5
0.4	0.2	–1.4	0.61	2.3	3.8	5.2	6.3	7.3	8.3	9.1	9.8
	0.3	–1.8	–0.54	0.42	1.2	1.8	2.3	2.7	3.1	3.4	3.7
	0.4	–1.9	–0.89	–0.17	0.36	0.76	1.1	1.3	1.5	1.7	1.9
0.5	0.2	–0.82	0.97	2.6	4.0	5.3	6.4	7.4	8.3	9.1	9.9
	0.3	–1.2	–0.15	0.71	1.4	2.0	2.5	2.9	3.3	3.6	3.9
	0.4	– 1.4	–0.54	0.06	0.50	0.85	1.1	1.3	1.5	1.7	1.8
	0.5	–1.4	–0.66	–0.15	0.21	0.48	0.68	0.84	0.97	1.1	1.2
0.6	0.2	–0.52	1.2	2.7	4.1	5.3	6.4	7.4	8.3	9.1	9.9
	0 3	–0.93	0.06	0.85	1.5	2.1	2.6	3.0	3.4	3.7	4.0
	0.4	–1.1	–0.37	0.16	0.55	0.86	1.1	1.3	1.4	1.6	1.8
	0.5	–1.1	–0.49	–0.06	0.25	0.48	0.66	0.79	0.90	1.0	1.1
	0.6	– 1.2	–0.55	–0.15	0.12	0.31	0.45	0.56	0.65	0.71	0.77
0.8	0.2	–0.27	1.3	2.7	4.0	5.2	6.3	7.3	8.2	9.0	9.7
	0.3	–0.67	0.18	0.90	1.5	2.0	2.5	2.9	3.3	3.6	4.0
	0.4	–0.85	–0.27	0.16	0.49	0.75	0.97	1.2	1.3	1.4	1.6
	0.5	–0.90	–0.40	–0.07	0.18	0.36	0.50	0.61	0.70	0.78	0.84
	0.6	–0.92	–0.46	–0.16	0.04	0.18	0.29	0.37	0.44	0.49	0.53
	0.7	–0.93	–0.49	–0.21	–0.03	0.10	0.19	0.25	0.30	0.34	0.37
	0.8	–0.93	–0.50	–0.24	–0.07	0.05	0.13	0.19	0.23	0.27	0.29
1.0	0.2	–0.26	1.2	2.6	3.9	5.1	6.1	7.1	8.0	8.8	9.5
	0.3	–0.65	0.12	0.79	1.4	1.9	2.4	2.8	3.1	3.5	3.8
	0.4	–0.83	–0.34	0.04	0.33	0.58	0.78	0.95	1.1	1.2	1.3
	0.5	–0.89	–0.48	–0.20	0	0.15	0.27	0.37	0.45	0.51	0.57
	0.6	–0.91	–0.54	–0.31	–0.14	–0.03	0.06	0.12	0.18	0.22	0.25
	0.8	–0.91	–0.59	–0.38	–0.25	–0.16	–0.10	–0.06	–0.03	–0.01	0.01
	1.0	–0.93	–0.60	–0.40	–0.28	–0.20	–0.14	–0.11	–0.08	–0.07	–0.06

Main, $C_{c,s}$

		Q_b/Q_c									
A_s/A_c	A_b/A_c	**0.2**	**0.4**	**0.6**	**0.8**	**1.0**	**1.2**	**1.4**	**1.6**	**1.8**	**2.0**
0.3	0.2	4.5	2.8	1.5	0.56	–0.17	–0.74	–1.2	–1.6	–1.9	–2.1
	0.3	4.6	3.1	2.0	1.2	0.57	0.08	–0.30	–0.62	–0.89	–1.1
0.4	0.2	1.6	0.85	0.16	–0.43	–0.92	–1.3	–1.7	–1.9	–2.2	–2.4
	0.3	1.7	1.1	0.58	0.13	–0.24	–0.56	–0.82	–1.1	–1.3	–1.4
	0.4	1.8	1.3	0.80	0.42	0.11	–0.15	–0.37	–0.55	–0.72	–0.86
0.5	0.2	0.67	0.18	–0.33	–0.79	–1.2	–1.5	–1.8	–2.1	–2.3	–2.5
	0.3	0.75	0.42	0.07	–0.25	–0.54	–0.80	–1.0	–1.2	–1.4	–1.5
	0.4	0.80.	0.55	0.28	0.03	–0.20	–0.40	–0.57	–0.73	–0.86	–0.98
	0.5	0.82	0.62	0.41	0.20	0.02	–0.15	–0.29	–0.42	–0.53	–0.63
0.6	0.2	0.26	–0.11	–0.54	–0.95	–1.3	–1.6	–1.9	–2.1	–2.4	–0.25
	0.3	0.34	0.13	–0.14	–0.42	–0.67	–0.90	–0.11	–0.13	–0.14	–0.16
	0.4	0.39	0.25	0.06	–0.14	–0.33	–0.51	–0.66	–0.80	–0.93	–1.0
	0.5	0.41	0.32	0.18	0.03	–1.2	–0.26	–0.38	–0.50	–0.60	–0.69
	0.6	0.43	0.37	0.26	0.14	0.02	–0.09	–0.19	–0.29	–0.37	–0.45
0.8	0.2	–0.01	–0.30	–0.67	–1.1	–1.4	–1.7	–2.0	–2.2	–2.4	–2.6
	0.3	0.07	–0.07	–0.29	–0.58	–0.76	–0.97	–1.2	–1.3	–1.5	–1.6
	0.4	0.11	0.05	–0.09	–0.26	–0.42	–0.58	–0.72	–0.85	–0.97	–1.1
	0.5	0.14	0.12	0.03	–0.09	–0.21	–0.34	–0.45	–0.55	–0.64	–0.73
	0.6	0.15	0.17	0.11	0.02	–0.07	–0.17	–0.26	–0.34	–0.42	–0.49
	0.7	0.17	0.21	0.17	0.11	0.03	–0.05	–0.12	–0.19	–0.26	–0.32
	0.5	0.17	0.23	0.22	0.17	0.11	0.05	–0.02	–0.07	–0.13	–0.18
1.0	0.2	–0.05	–0.33	–0.70	–1.1	–1.4	–1.7	–2.0	–2.2	–2.4	–2.6
	0.3	0.03	–0.10	–0.31	–0.55	–0.78	–0.98	–1.2	–1.3	–1.5	–1.6
	0.4	0.07	0.02	–0.12	–0.28	–0.44	–0.59	–0.73	–0.86	–0.98	–1.1
	0.5	0.09	0.09	0.01	–0.11	–0.23	–0.35	–0.46	–0.56	–0.65	–0.74
	0.6	0.11	0.14	0.09	0	–0.09	–0.18	–0.27	–0.35	–0.43	–0.50
	0.8	0.13	0.20	0.19	0.15	0.09	0.03	–0.03	–0.08	–0.14	–0.19
	1.0	0.14	0.24	0.25	0.24	0.20	0.16	0.12	0.08	0.04	0

5-5 Wye, 45°, Converging, Round, Conical Main (Sepsy and Pies 1973)



Branch, $C_{c,b}$

		Q_b/Q_s									
A_s/A_c	A_b/A_c	**0.2**	**0.4**	**0.6**	**0.8**	**1.0**	**1.2**	**1.4**	**1.6**	**1.8**	**2.0**
0.3	0.2	–2.4	–0.01	2.0	3.8	5.3	6.6	7.8	8.9	9.8	11
	0.3	–2.8	–1.2	0.12	1.1	1.9	2.6	3.2	3.7	4.2	4.6
0.4	0.2	–1.2	0.93	2.8	4.5	5.9	7.2	8.4	9.5	10	11
	0.3	–1.6	–0.27	0.81	1.7	2.4	3.0	3.6	4.1	4.5	4.9
	0.4	–1.8	–0.72	0.07	0.66	1.1	1.5	1.8	2.1	2.3	2.5
0.5	0.2	–0.46	1.5	3.3	4.9	6.4	7.7	8.8	9.9	11	12
	0.3	–0.94	0.25	1.2	2.0	2.7	3.3	3.8	4.2	4.7	5.0
	0.4	–1.1	–0.24	0.42	0.92	1.3	1.6	1.9	2.1	2.3	2.5
	0.5	–1.2	–0.38	0.18	0.58	0.88	1.1	1.3	1.5	1.6	1.7

Branch, $C_{c,b}$

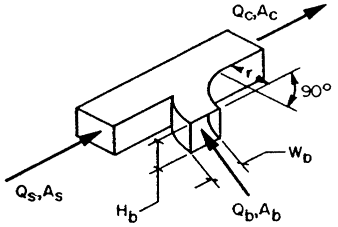
		Q_b/Q_s									
A_s/A_c	A_b/A_c	**0.2**	**0.4**	**0.6**	**0.8**	**1.0**	**12**	**1.4**	**1.6**	**1.8**	**2.0**
0.6	0.2	–0.55	1.3	3.1	4.7	6.1	7.4	8.6	9.6	11	12
	0.3	–1.1	0	0.88	1.6	2.3	2.8	3.3	3.7	4.1	4.5
	0.4	–1.2	–0.48	0.10	0.54	0.89	1.2	1.4	1.6	1.8	2.0
	0.5	–1.3	–0.62	–0.14	0.21	0.47	0.68	0.85	0.99	1.1	1.2
	0.6	–1.3	–0.69	–0.26	0.04	0.26	0.42	0.57	0.66	0.75	0.82
0.8	0.2	0.06	1.8	3.5	5.1	6.5	7.8	8.9	10	11	12
	0.3	–0.52	0.35	1.1	1.7	2.3	2.8	3.2	3.6	3.9	4.2
	0.4	–0.67	–0.05	0.43	0.80	1.1	1.4	1.6	1.8	1.9	2.1
	0.6	–0.75	–0.27	0.05	0.28	0.45	0.58	0.68	0.76	0.83	0.88
	0.7	–0.77	–0.31	–0.02	0.18	0.32	0.43	0.50	0.56	0.61	0.65
	0.8	–0.78	–0.34	–0.07	0.12	0.24	0.33	0.39	0.44	0.47	0.50
1.0	0.2	0.40	2.1	3.7	5.2	6.6	7.8	9.0	11	11	12
	0.3	–0.21	0.54	1.2	1.8	2.3	2.7	3.1	3.7	3.7	4.0
	0.4	–0.33	0.21	0.62	0.96	1.2	1.5	1.7	2.0	2.0	2.1
	0.5	–0.38	0.05	0.37	0.60	0.79	0.93	1.1	1.2	1.2	1.3
	0.6	–0.41	–0.02	0.23	0.42	0.55	0.66	0.73	0.80	0.85	0.89
	0.8	–0.44	–0.10	0.11	0.24	0.33	0.39	0.43	0.46	0.47	0.48
	1.0	–0.46	–0.14	0.05	0.16	0.23	0.27	0.29	0.30	0.30	0.29

Main, $C_{c,s}$

		Q_b/Q_s									
A_s/A_c	A_b/A_c	**0.2**	**0.4**	**0.6**	**0.8**	**1.0**	**12**	**1.4**	**1.6**	**1.8**	**2.0**
0.3	0.2	5.3	–0.01	2.0	1.1	0.34	–0.20	–0.61	–0.93	–1.2	–1.4
	0.3	5.4	3.7	2.5	1.6	1.0	0.53	0.16	–0.14	–0.38	–0.58
0.4	0.2	1.9	1.1	0.46	–0.07	–0.49	–0.83	–1.1	–1.3	–1.5	–1.7
	0.3	2.0	1.4	0.81	0.42	0.08	–0.20	–0.43	–0.62	–0.78	–0.92
	0.4	2.0	1.5	1.0	0.68	0.39	0.16	–0.04	–0.21	–0.35	–0.47
0.5	0.2	0.77	0.34	–0.09	–0.48	–0.81	–1.1	–1.3	–1.5	–1.7	–1.8
	0.3	0.85	0.56	0.25	–0.03	–0.27	–0.48	–0.67	–0.82	–0.96	–1.1
	0.4	0.88	0.66	0.43	0.21	0.02	–0.15	–0.30	–0.42	–0.54	–0.64
	0.5	0.91	0.73	0.54	0.36	0.21	0.06	–0.06	–0.17	–0.26	–0.35
0.6	0.2	0.30	0	–0.34	–0.67	–0.96	–1.2	–1.4	–1.6	–1.8	–1.9
	0.3	0.37	0.21	–0.02	–0.24	–0.44	–0.63	–0.79	–0.93	–1.1	–1.2
	0.4	0.40	0.31	0.16	–0.1	–0.16	–0.30	–0.43	–0.54	–0.64	–0.73
	0.5	0.43	0.37	0.26	0.14	0.02	–0.09	–0.20	–0.29	–0.37	–0.45
	0.6	0.44	0.41	0.33	0.24	0.14	0.05	–0.03	–0.11	–0.18	–0.25
0.8	0.2	–0.06	–0.27	–0.57	–0.86	–1.1	–1.4	–1.6	–1.7	–1.9	–2.0
	0.3	0	–0.08	–0.25	–0.43	–0.62	–0.78	–0.93	–1.1	–1.2	–1.3
	0.4	0.04	0.02	–0.08	–0.21	–0.34	–0.46	–0.57	–0.67	–0.77	–0.85
	0.5	0.06	0.08	0.02	–0.06	–0.16	–0.25	–0.34	–0.42	–0.50	–0.57
	0.6	0.07	0.12	0.09	0.03	–0.04	–0.11	–0.18	–0.25	–0.31	–0.37
	0.7	0.08	0.15	0.14	0.10	0.05	–0.01	–0.07	–0.12	–0.17	–0.22
	0.8	0.09	0.17	0.18	0.16	0.11	0.07	0.02	–0.02	–0.07	–0.11
1.0	0.2	–0.19	–0.39	–0.67	–0.96	–1.2	–1.5	–1.6	–1.8	–2.0	–2.1
	0.3	–0.12	–0.19	–0.35	–0.54	–0.71	–0.87	–1.0	–1.2	–1.3	–1.4
	0.4	–0.09	–0.10	–0.19	–0.31	–0.43	–0.55	–0.66	–0.77	–0.86	–0.94
	0.5	–0.07	–0.04	–0.09	–0.17	–0.26	–0.35	–0.44	–0.52	–0.59	–0.66
	0.6	–0.06	0	–0.02	–0.07	–0.14	–0.21	–0.28	–0.34	–0.40	–0.46
	0.8	–0.04	0.06	0.07	0.05	0.02	–0.03	–0.07	–0.12	–0.16	–0.20
	1.0	–0.3	0.09	0.13	0.13	0.11	0.08	0.06	0.03	–0.01	–0.03

5-6 Tee, Converging, Rectangular (ldelchik 1986, Diagram 7-11)

$r/w_b = 1$



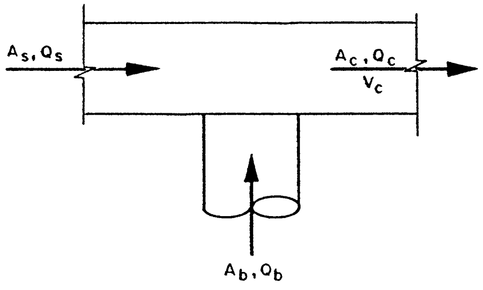
Branch, $C_{c,b}$

		Q_c/Q_c								
A_b/A_s	A_b/A_c	**0.1**	**0.2**	**~0.3**	**0.4**	**0.5**	**0.6**	**0.7**	**0.8**	**0.9**
0.33	0.25	–1.2	–0.40	0.40	1.6	3.0	4.8	6.8	8.9	11
0.5	0.5	–0.50	–0.20	0	0.25	0.45	0.70	1.0	1.5	2.0
0.67	0.5	–1.0	–0.60	–0.20	0.10	0.30	0.60	1.0	1.5	2.0
1.0	0.5	–2.2	– 1.5	–0.95	–0.50	0	0.40	0.80	1.3	1.9
1.0	1.0	–0.60	–0.30	–0.10	–0.04	0.13	0.21	0.29	0.36	0.42
1.33	1.0	–1.2	–0.80	–0.40	–0.20	0	0.16	0.24	0.32	0.38
2.0	1.0	–2.1	–1.4	–0.90	–0.50	–0.20	0	0.20	0.25	0.30

Main, $C_{c,s}$

		Q_b/Q_s								
A_b/A_s	A_b/A_c	**0.1**	**0.2**	**0.3**	**0.4**	**0.5**	**0.6**	**0.7**	**0.8**	**0.9**
0.33	0.25	0.30	0.30	0.20	–0.10	–0.45	–0.92	–1.5	– 2.0	–2.6
0.5	0.5	0.17	0.16	0.10	0	–0.08	–0.18	–0.27	–0.37	–0.46
0.67	0.5	0.27	0.35	0.32	0.25	0.12	–0.03	–0.23	–0.42	–0.58
1.0	0.5	1.2	1.1	0.90	0.65	0.35	0	–0.40	–0.80	–1.3
1.0	1.0	0.18	0.24	0.27	0.26	0.23	0.18	0.10	0	–0.12
1.33	1.0	0.75	0.36	0.38	0.35	0.27	0.18	0.05	–0.08	–0.22
2.0	1.0	0.80	0.87	0.80	0.68	0.55	0.40	0.25	0.08	–0.10

5-7 Tee, Converging, Round Tap to Rectangular Main (SMACNA 1981, Table 6-9C)



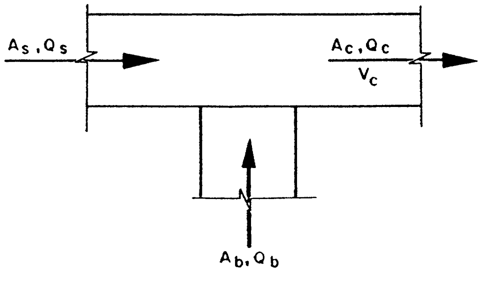
A_b/A_s	A_s/A_c	A_b/A_c
0.5	1.0	0.5

Branch, $C_{c,b}$

	Q_b/Q_c									
V_c (fpm)	**0.1**	**0.2**	**0.3**	**0.4**	**0.5**	**0.6**	**0.7**	**0.8**	**0.9**	**1.0**
<1200	–0.63	–0.55	0.13	0.23	0.78	1.30	1.93	3.10	4.88	5.60
>1200	–0.49	–0.21	0.23	0.60	1.27	2.06	2.75	3.70	4.93	5.95

For main coefficient ($C_{c,s}$), see Fitting 5-3.

5-8 Tee, Converging, Rectangular Main and Tap (SMACNA 1981, Table 6-9D)



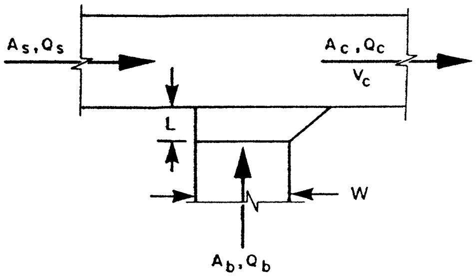
A_b/A_s	A_s/A_c	A_b/A_c
0.5	1.0	0.5

Branch, $C_{c,b}$

	Q_b/Q_c									
V_c (fpm)	**0.1**	**0.2**	**0.3**	**0.4**	**0.5**	**0.6**	**0.7**	**0.8**	**0.9**	**1.0**
<1200	–0.75	–0.53	–0.03	0.33	1.03	1.10	2.15	2.93	4.18	4.78
>1200.	–0.69	–0.21	0.23	0.67	1.17	1.66	2.67	3.36	3.93	5.13

For main coefficient ($C_{c,s}$), see Fitting 5-3.

5-9 Converging, Rectangular Main and Tap (45° Entry) (SMACNA 1981, Table-9F)



$L = 0.25\ W$, 3 in. min.

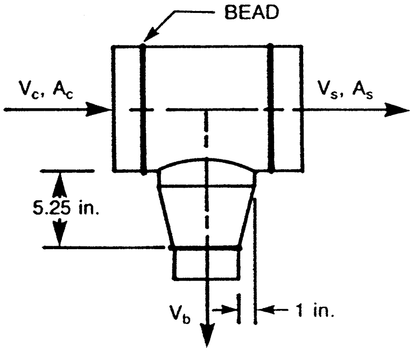
A_b/A_s	A_s/A_c	A_b/A_c
0.5	1.0	0.5

Branch, $C_{c,b}$

V_c	Q_b/Q_c									
(fpm)	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
<1200	–0.83	–0.68	–0.30	0.28	0.55	1.03	1.50	1.93	2.50	3.03
>1200	–0.72	–0.52	–0.23	0.34	0.76	1.14	1.83	2.01	2.90	3.63

For main coefficient ($C_{c,s}$), see Fitting 5-3.

5-10 Tee, Diverging, Round, Conical Branch (Jones and Sepsy 1969, Figure 12)



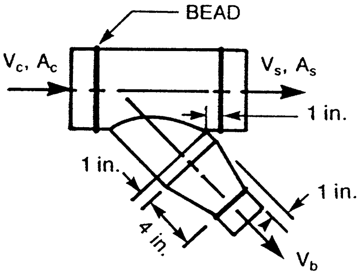
$A_c = A_s$

Branch

V_b/V_s	0	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
$C_{c,b}$	1.0	0.85	0.74	0.62	0.52	0.42	0.36	0.32	0.32	0.37	0.52

For main loss coefficient ($C_{c,s}$), see Fitting 5-23.

5-11 Wye, 45°, Diverging, Round, Conical Branch (Jones and Sepsy 1969, Figure 14)



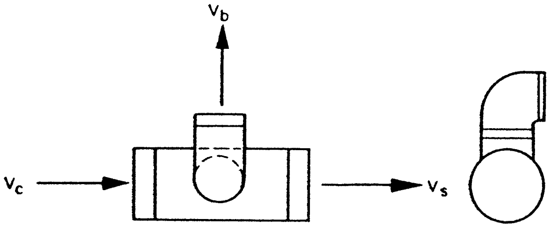
$A_c = A_s$

Branch

V_b/V_c	0	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
$C_{c,b}$	1.0	0.84	0.61	0.41	0.27	0.17	0.12	0.12	0.14	0.18	0.27

For main loss coefficient ($C_{c,s}$), see Fitting 5-23.

5-12 Tee, Diverging, Round, with 90° Elbow, Branch 90° to Main (Jones and Sepsy 1969, Figure 17)



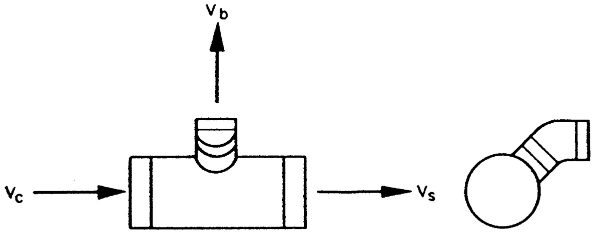
$A_r = A_s$

Branch

V_b/V_c	0	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
$C_{c,b}$	1.0	1.03	1.08	1.18	1.33	1.56	1.86	2.2	2.6	3.0	3.4

For main loss coefficient ($C_{c,s}$), see Fitting 5-23.

5-13 Tee, Diverging, Round, with 45° Elbow, Branch 90° to Main (Jones and Sepsy 1969, Figure 18)



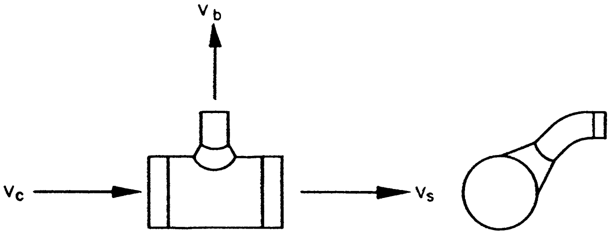
$A_c = A_s$

Branch

V_b/V_c	0	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
$C_{c,b}$	1.0	1.32	1.51	1.60	1.65	1.74	1.87	2.0	2.2	2.5	2.7

For main loss coefficient ($C_{c,s}$), see Fitting 5-23.

5-14 Tee, Diverging, Round (Conical Branch), with 45° Elbow, Branch 90° to Main (Jones and Sepsy 1969, Figure 19)



For tee geometry, see Fitting 5-10.

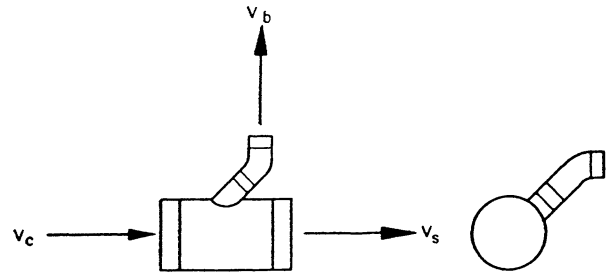
$A_c = A_s$

Branch

V_b/V_s	0	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
$C_{c,b}$	1.0	0.94	0.88	0.84	0.80	0.82	0.84	0.87	0.90	0.95	1.02

For main loss coefficient ($C_{c,s}$), see Fitting 5-23.

5-15 Wye, 45°, Round, with 60° Elbow, Branch 90° to Main (Jones and Sepsy 1969, Figure 3)

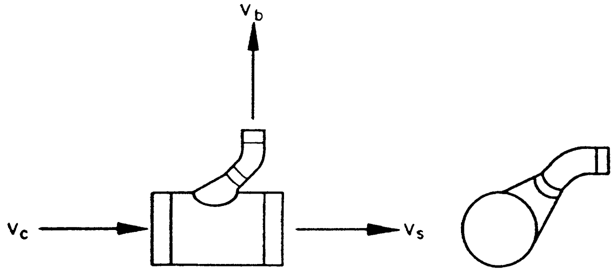


$A_c = A_s$

Branch											
V_b/V_c	0	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
$C_{c,b}$	1.0	0.88	0.77	0.68	0.65	0.69	0.73	0.88	1.14	1.54	2.2

For main loss coefficient ($C_{c,s}$), see Fitting 5-23.

5-16 Wye, 45°, Diverging, Round (Conical Branch), with 60° Elbow, Branch 90° to Main (Jones and Sepsy 1969, Figure 20)



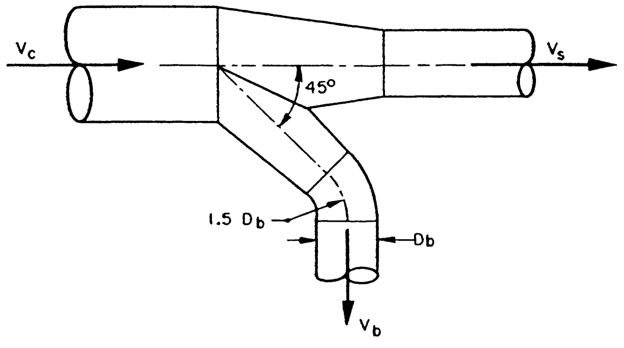
For wye geometry, see Fitting 5-11.

$A_c = A_s$

Branch											
V_b/V_c	0	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
$C_{c,b}$	1.0	0.82	0.63	0.52	0.45	0.42	0.41	0.40	0.41	0.45	0.56

For main loss coefficient ($C_{c,s}$), see Fitting 5-23.

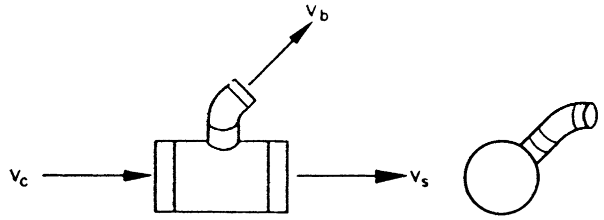
5-17 Wye, 48°, Diverging, Conical Main and Branch, with 45° Elbow,



Branch									
V_b/V_c	0.2	0.4	0.6	0.7	0.8	0.9	1.0	1.1	1.2
$C_{c,b}$	0.76	0.60	0.52	0.50	0.51	0.52	0.56	0.61	0.68
V_b/V_c	1.4	1.6	1.8	2.0	2.2	2.4	2.6	2.8	3.0
$C_{c,b}$	0.86	1.1	1.4	1.8	2.2	2.6	3.1	3.7	4.2

Main										
V_s/V_c	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
$C_{c,s}$	0.14	0.06	0.05	0.09	0.18	0.30	0.46	0.64	0.84	1.0

5-18 Tee, Diverging, Round, with 60° Elbow, Branch 45° to Main (Jones and Sepsy 1969, Figure 22)

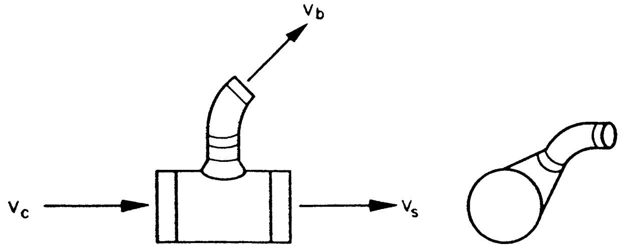


$A_c = A_s$

Branch											
V_b/V_c	0	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
$C_{c,b}$	1.0	1.06	1.15	1.29	1.45	1.65	1.89	2.2	2.5	2.9	3.3

For main loss coefficient ($C_{c,s}$), see Fitting 5-23.

5-19 Tee, Diverging, Round (Conical Branch), with 60° Elbow, Branch 45° to Main (Jones and Sepsy 1969, Figure 23)



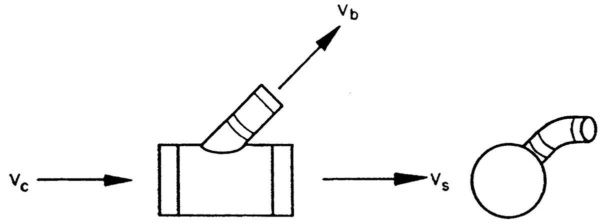
For tee geometry, see Fitting 5-10.

$A_c = A_s$

Branch											
V_b/V_c	0	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
$C_{c.b}$	1.0	0.95	0.90	0.86	0.81	0.79	0.79	0.81	0.86	0.96	1.10

For main loss coefficient ($C_{c,s}$), see Fitting 5-23.

5-20 Wye, 45°, Diverging, Round, with 30° Elbow, Branch 45° to Main (Jones and Sepsy 1969, Figure 2)

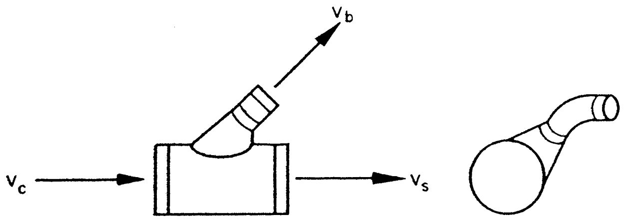


$A_c = A_s$

	Branch										
V_b/V_c	0	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
$C_{c,b}$	1.0	0.84	0.72	0.62	0.54	0.50	0.56	0.71	0.92	1.22	1.66

For main loss coefficient ($C_{c,s}$), see Fitting 5-23.

5-21 Wye, 45°, Diverging, Round (Conical Branch), with 30° Elbow,



Branch 45° to Main (Jones and Sepsy 1969, Figure 24)

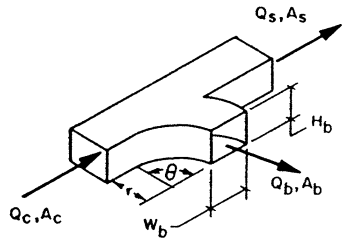
For wye geometry, see Fitting 5-11.

$A_c = A_s$

	Branch										
V_b/V_c	0	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
$C_{c,b}$	1.0	0.93	0.71	0.55	0.44	0.42	0.42	0.44	0.47	0.54	0.62

For main loss coefficient ($C_{c,s}$), see Fitting 5-23.

5-22 Tee, Diverging, Rectangular (Idelchik et al. 1986, Diagram 7-21)



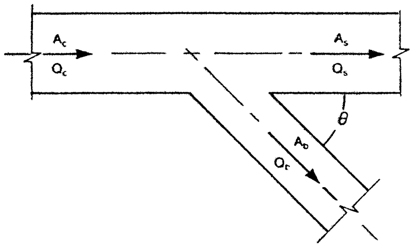
$\theta = 90°$

$r/W_b = 1.0$

		Branch, $C_{c,b}$								
		Q_b/Q_c								
A_b/A_s	A_b/A_c	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.25	0.25	0.55	0.50	0.60	0.85	1.2	1.8	3.1	4.4	6.0
0.33	0.25	0.35	0.35	0.50	0.80	1.3	2.0	2.8	3.8	5.0
0.5	0.5	0.62	0.48	0.40	0.40	0.48	0.60	0.78	1.1	1.5
0.67	0.5	0.52	0.40	0.32	0.30	0.34	0.44	0.62	0.92	1.4
1.0	0.5	0.44	0.38	0.38	0.41	0.52	0.68	0.92	1.2	1.6
1.0	1.0	0.67	0.55	0.46	0.37	0.32	0.29	0.29	0.30	0.37
1.33	1.0	0.70	0.60	0.51	0.42	0.34	0.28	0.26	0.26	0.29
2.0	1.0	0.60	0.52	0.43	0.33	0.24	0.17	0.15	0.17	0.21

		Main, $C_{c,s}$								
		Q_b/Q_c								
A_b/A_s	A_b/A_c	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.25	0.25	–0.01	–0.03	–0.01	0.05	0.13	0.21	0.29	0.38	0.46
0.33	0.25	0.08	0	–0.02	–0.01	0.02	0.08	0.16	0.24	0.34
0.5	0.5	–0.03	–0.06	–0.05	0	0.06	0.12	0.19	0.27	0.35
0.67	0.5	0.04	–0.02	–0.04	–0.03	–0.01	0.04	0.12	0.23	0.37
1.0	0.5	0.72	0.48	0.28	0.13	0.05	0.04	0.09	0.18	0.30
1.0	1.0	–0.02	–0.04	–0.04	–0.01	0.06	0.13	0.22	0.30	0.38
1.33	1.0	0.10	0	0.01	–0.03	–0.01	0.03	0.10	0.20	0.30
2.0	1.0	0.62	0.38	0.23	0.23	0.08	0.05	0.06	0.10	0.20

5-23 Wye, Diverging, Rectangular and Round (Idelchik et al. 1986, Diagrams 7-15 and 7-17)



$A_c = A_s$; $H_b = H_c$, where H is height of rectangular duct

$\theta = 30°$

	Branch, $C_{c,b}$								
	Q_b/Q_c								
A_b/A_c	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.8	0.75	0.55	0.40	0.28	0.21	0.16	0.15	0.16	0.19
0.7	0.72	0.51	0.36	0.25	0.18	0.15	0.16	0.20	0.26
0.6	0.69	0.46	0.31	0.21	0.17	0.16	0.20	0.28	0.39
0.5	0.65	0.41	0.26	0.19	0.18	0.22	0.32	0.47	0.67
0.4	0.59	0.33	0.21	0.20	0.27	0.40	0.62	0.92	1.3
0.3	0.55	0.28	0.24	0.38	0.76	1.3	2.0	3.0	4.1
0.2	0.40	0.26	0.58	1.3	2.5	4.1	6.1	8.6	11.0
0.1	0.28	1.5	4.3	8.3	15.0	—	—	—	—

$\theta = 45°$

	Branch, $C_{c,b}$								
	Q_b/Q_c								
A_b/A_c	0.1	0.2	0.3	0.4	0.5	0 6	0.7	0.8	0.9
0.8	0.78	0.62	0.49	0.40	0.34	0.31	0.32	0.35	0.40
0.7	0.71	0.59	0.47	0.38	0.34	0.32	0.35	0.41	0.50
0.6	0.74	0.56	0.44	0.37	0.35	0.36	0.43	0.54	0.68
0.5	0.71	0.52	0.41	0.38	0.40	0.45	0.59	0.78	1.0
0.4	0.66	0.47	0.40	0.43	0.54	0.69	0.95	1.3	1.7
0.3	0.66	0.48	0.52	0.73	1.2	1.8	2.7	3.7	4.9
0.2	0.56	0.56	1.0	1.8	3.2	4.9	7.1	9.6	13.0
0.1	0.60	2.1	5.1	9.3	16.0	—	—	—	—

$\theta = 60°$

	Branch, $C_{c,b}$								
	Q_b/Q_c								
A_b/A_c	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.8	0.83	0.71	0.62	0.56	0.52	0.50	0.53	0.60	0.68
0.7	0.82	0.69	0.61	0.56	0.54	0.54	0.60	0.70	0.82
0.6	0.81	0.68	0.60	0.58	0.58	0.61	0.72	0.87	1.1
0.5	0.79	0.66	0.61	0.62	0.68	0.76	0.94	1.2	1.5
0.4	0.76	0.65	0.65	0.74	0.89	1.1	1.4	1.8	2.3
0.3	0.80	0.75	0.89	1.2	1.8	2.6	3.5	4.6	6.0
0.2	0.77	0.96	1.6	2.5	4.0	6.0	8.3	11.0	—
0.1	1.0	2.9	6.2	10.0	—	—	—	—	—

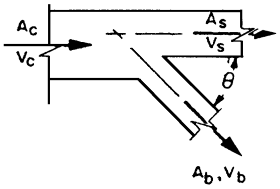
$\theta = 90°$

	Branch, $C_{c,b}$								
	Q_b/Q_c								
A_b/A_c	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0 8	0.95	0.92	0.92	0.93	0.94	0.95	1.1	1.2	1.4
0.7	0.95	0.94	0.95	0.98	1.0	1.1	1.2	1.4	1.6
0.6	0.96	0.97	1.0	1.1	1.1	1.2	1.4	1.7	2.0
0.5	0.97	1.0	1.1	1.2	1.4	1.5	1.8	2.1	2.5
0.4	0.99	1.1	1.3	1.5	1.7	2.0	2.4	3.0	3.6
0.3	1.1	1.4	1.8	2.3	3.2	4.3	5.5	6.9	8.5
0.2	1.3	1.9	2.9	4.1	6.2	8.5	11.0	—	—
0.1	2.1	4.8	8.9	14.0	—	—	—	—	—

Main

V_s/V_c	0	0.1	0.2	0.3	0.4	0.5	0.6	0.8	1.0
$C_{c,s}$	0.40	0.32	0.26	0.20	0.14	0.10	0.06	0.02	0

5-24 Diverging Wye, Rectangular (Idelchik et al. 1986, Diagrams 7-16 and 7-17)



θ = 15° to 90° and $A_c = A_s + A_b$

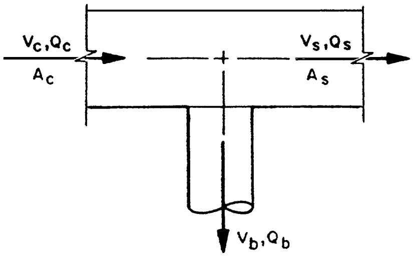
Branch, $C_{c,b}$

θ, deg.	V_b/V_c 0.1	0.2	0.3	0.4	0.5	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
15	0.81	0.65	0.51	0.38	0.28	0.20	0.11	0.06	0.14	0.30	0.51	0.76	1.0
30	0.84	0.69	0.56	0.44	0.34	0.26	0.19	0.15	0.15	0.30	0.51	0.76	1.0
45	0.87	0.74	0.63	0.54	0.45	0.38	0.29	0.24	0.23	0.30	0.51	0.76	1.0
60	0.90	0.82	0.79	0.66	0.59	0.53	0.43	0.36	0.33	0.39	0.51	0.76	1.0
90	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0

Main, $C_{c,s}$

θ, degrees	15-60	90				
		A_s/A_c				
V_s/V_c	0-1.0	0-0.4	0.5	0.6	0.7	≥0.8
0	1.0	1.0	1.0	1.0	1.0	1.0
0.1	0.81	0.81	0.81	0.81	0.81	0.81
0.2	0.64	0.64	0.64	0.64	0.64	0.64
0.3	0.50	0.50	0.52	0.52	0.50	0.50
0.4	0.36	0.36	0.40	0.38	0.37	0.36
0.5	0.25	0.25	0.30	0.28	0.27	0.25
0.6	0.16	0.16	0.23	0.20	0.18	0.16
0.8	0.04	0.04	0.17	0.10	0.07	0.04
1.0	0	0	0.20	0.10	0.05	0
1.2	0.07	0.07	0.36	0.21	0.14	0.07
1.4	0.39	0.39	0.79	0.59	0.39	—
1.6	0.90	0.90	1.4	1.2	—	—
1.8	1.8	1.8	2.4	—	—	—
2.0	3.2	3.2	4.0	—	—	—

5-25 Tee, Diverging, Rectangular Main to Round Tap (SMACNA 1981, Table 6-10T)



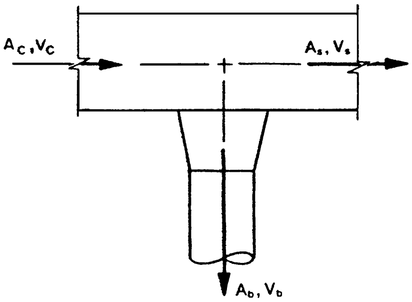
$A_c = A_s$

Branch, $C_{c,b}$

V_b/V_c	Q_b/Q_s 0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.2	1.00								
0.4	1.01	1.07							
0.6	1.14	1.10	1.08						
0.8	1.18	1.31	1.12	1.13					
1.0	1.30	1.38	1.20	1.23	1.26				
1.2	1.46	1.58	1.45	1.31	1.39	1.48			
1.4	1.70	1.82	1.65	1.51	1.56	1.64	1.71		
1.6	1.93	2.06	2.00	1.85	1.70	1.76	1.80	1.88	
1.8	2.06	2.17	2.10	2.13	2.06	1.98	1.99	2.00	2.07

For main coefficient ($C_{c,s}$), see Fitting 5-23.

5-26 Tee, Diverging, Rectangular Main to Round Tap (Conical) (Inoue et al. 1980, Korst et al. 1950)



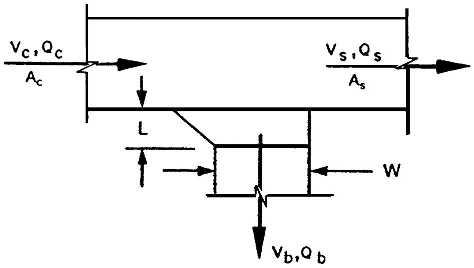
$A_c = A_s$

Branch

V_b/V_c	0.40	0.50	0.75	1.0	1.3	1.5
$C_{c,b}$	0.80	0.83	0.90	1.0	1.1	1.4

For main coefficient ($C_{c,s}$), see Fitting 5-23.

5-27 Tee, Diverging, Rectangular Main, and Tap (45° Entry) (SMACNA 1981, Table 6-10N)



Recommended[a]

$L = 0.25W$, 3 in. min.

$A_c = A_s$

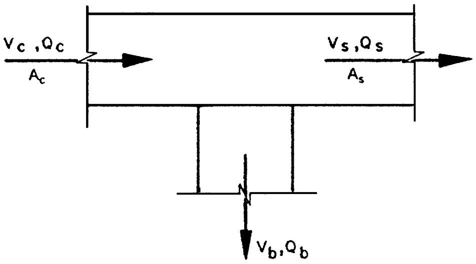
Branch, $C_{c,b}$

V_b/V_c	Q_b/Q_s 0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.2	0.91								
0.4	0.81	0.79							
0.6	0.77	0.72	0.70						
0.8	0.78	0.73	0.69	0.66					
1.0	0.78	0.98	0.85	0.79	0.74				
1.2	0.90	1.11	1.16	1.23	1.03	0.86			
1.4	1.19	1.22	1.26	1.29	1.54	0.25	0.92		
1.6	1.35	1.42	1.55	1.59	1.63	1.50	1.31	1.09	
1.8	1.44	1.50	1.75	1.74	1.72	2.24	1.63	1.40	1.17

For main coefficient ($C_{c,s}$), see Fitting 5-23.

[a] For performance study, see SMACNA (1987).

5-28 Tee, Diverging, Rectangular Main, and Tap[a] (SMACNA 1981, Table 10Q)



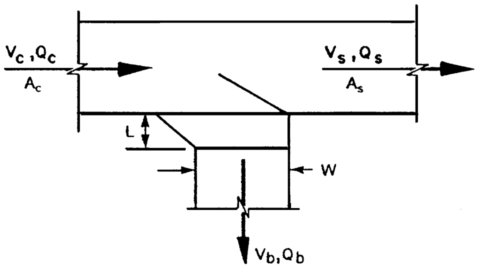
$A_c = A_s$

	Branch, $C_{c,b}$								
	Q_b/Q_c								
V_b/V_c	**0.1**	**0.2**	**0.3**	**0.4**	**0.5**	**0.6**	**0.7**	**0.8**	**0.9**
0.2	1.03								
0.4	1.04	1.01							
0 6	1.11	1.03	1.05						
0 8	1.16	1.21	1.17	1.12					
1.0	1.38	1.40	1.30	1.36	1.27				
1 2	1.52	1.61	1.68	1.91	1.47	1.66			
1 4	1.79	2.01	1.90	2.31	2.28	2.20	1.95		
1.6	2.07	2.28	2.13	2.71	2.99	2.81	2.09	2.20	
1.8	2.32	2.54	2.64	3.09	3.72	2.48	2.21	2.57	2.32

For main coefficient ($C_{c,s}$) see Fitting 5-23.

[a] For performance study see SMACNA (1987).

5-29 Tee, Diverging, Rectangular Main and Tap (45° Entry), with Damper (SMACNA 1981, Table 6-10P)



Poor; should not be used.[a]

$L = 0.25W$, 3 in. min

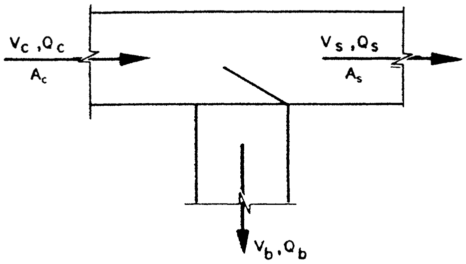
$A_c = A_s$

	Branch, $C_{c,b}$								
	Q_b/Q_c								
V_b/V_c	**0.1**	**0.2**	**0.3**	**0.4**	**0.5**	**0.6**	**0.7**	**0.8**	**0.9**
0.2	0.61								
0.4	0.46	0.61							
0.6	0.43	0.50	0.54						
0.8	0.39	0.43	0.62	0.53					
1.0	0.34	0.57	0.77	0.73	0.68				
1.2	0.37	0.64	0.85	0.98	1.07	0.83			
1.4	0.57	0.71	1.04	1.16	1.54	1.36	1.18		
1.6	0.89	1.08	1.28	1.30	1.69	2.09	1.81	1.47	
1.8	1.33	1.34	2.04	1.78	1.90	2.40	2.77	2.23	1.92

For main coefficient ($C_{c,s}$), see Fitting 5-31.

[a] For performance study, see SMACNA (1987).

5-30 Tee, Diverging, Rectangular Main and Tap, with Damper (SMACNA 1981, Table 6-10R)



Poor; should not be used.[a]

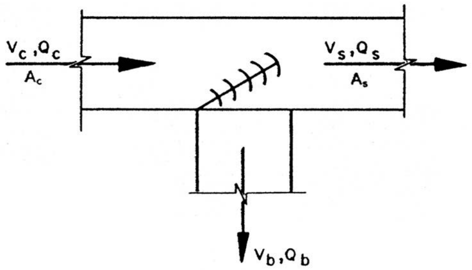
$A_c = A_s$

	Branch, $C_{c,b}$								
	Q_b/Q_c								
V_b/V_c	**0.1**	**0.2**	**0.3**	**0.4**	**0.5**	**0.6**	**0.7**	**0.8**	**0.9**
0.2	0.58								
0.4	0.67	0.64							
0.6	0.78	0.76	0.75						
0.8	0.88	0.98	0.81	1.01					
1.0	1.12	1.05	1.08	1.18	1.29				
1.2	1.49	1.48	1.40	1.51	1.70	1.91			
1.4	2.10	2.21	2.25	2.29	2.32	2.48	2.53		
1.6	2.72	3.30	2.84	3.09	3.30	3.19	3.29	3.16	
1.8	3.42	4.58	3.65	3.92	4.20	4.15	4.14	4.10	4.05

For main coefficient ($C_{c,s}$) see Fitting 5-31.

[a] For performance study see SMACNA (1987).

5-31 Tee, Diverging, Rectangular, with Extractor (SMACNA 1981, Table 6-10S)



Poor; should not be used.[a]

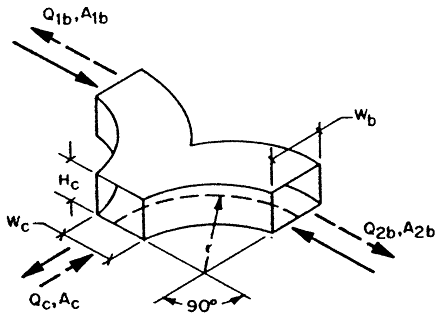
$A_c = A_s$

	Branch, $C_{c,b}$								
	Q_b/Q_c								
V_b/V_c	**0.1**	**0.2**	**0.3**	**0.4**	**0.5**	**0.6**	**0.7**	**0.8**	**0.9**
0.2	0.60								
0.4	0.62	0.69							
0.6	0.74	0.80	0.82						
0.8	0.99	1.10	0.95	0.90					
1.0	1.48	1.12	1.41	1.24	1.21				
1.2	1.91	1.33	1.43	1.52	1.55	1.64			
1.4	2.47	1.67	1.70	2.04	1.86	1.98	2.47		
1.6	3.17	2.40	2.33	2.53	2.31	2.51	3.13	3.25	
1.8	3.85	3.37	2.89	3.23	3.09	3.03	3.30	3.74	4.11

	Main								
V_b/V_c	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8
$C_{c,s}$	0.03	0.04	0.07	0.12	0.13	0.14	0.27	0.30	0.25

[a] For performance study, see SMACNA (1987).

5-32 Symmetrical Wye, Dovetail, Rectangular (Idelchik et al. 1986, Diagram 7-24)



$r/W_c = 1.5$

$Q_{1b}/Q_c = Q_{2b}/Q_c = 0.5$

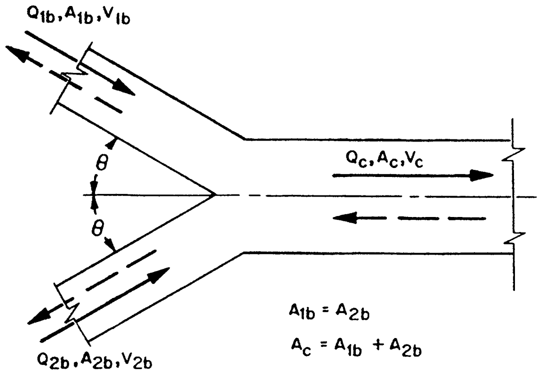
Converging

A_{1b}/A_c or A_{2b}/A_c	0.50	1.0
$C_{c,1b}$ or $C_{c,2b}$	0.23	0.07

Diverging

A_{1b}/A_c or A_{2b}/A_c	0.50	1.0
$C_{c,1b}$ or $C_{c,2b}$	0.30	0.25

5-33 Wye, Rectangular and Round (Idelchik et al. 1986, Diagram 7-30)



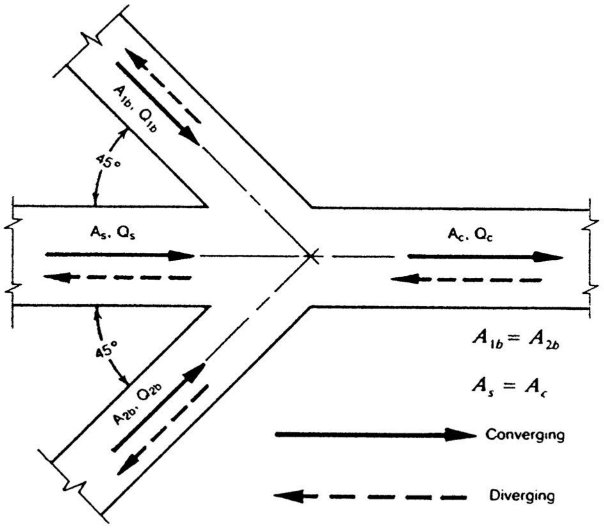
Converging $C_{c,1b}$ or $C_{c,2b}$

θ,	Q_{1b}/Q_c or Q_{2b}/Q_c										
deg.	0	0.1	0.2	03	0.4	0.5	0.6	0.7	0.8	0.9	1.0
15	–2.6	–1.9	–1.3	–0.77	–0.30	0.10	0.41	0.67	0.85	0.97	1.0
30	–2.1	–1.5	–1.0	–0.53	–0.10	0.28	0.69	0.91	1.1	1.4	1.6
45	–1.3	–0.93	–0.55	–0.16	0.20	0.56	0.92	1.3	1.6	2.0	2.3

Diverging $C_{c,1b}$ or $C_{c,2b}$

θ,	V_{1b}/V_c or V_{2b}/V_c												
deg.	0.1	0.2	03	0.4	0.5	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
15	0.81	0.65	0.51	0.38	0.28	0.20	0.11	0.06	0.14	0.30	0.51	0.76	1.0
30	0.84	0.69	0.56	0.44	0.34	0.26	0.19	0.15	0.15	0.30	0.51	0.76	1.0
45	0.87	0.74	0.63	0.54	0.45	0.38	0.29	0.24	0.23	0.30	0.51	0.76	1.0
60	0.90	0.82	0.79	0.66	0.59	0.53	0.43	0.36	0.33	0.39	0.51	0.76	1.0
90	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0

5-34 Wye (Double), 45° Rectangular and Round (Idelchik et al. 1986, Diagram 7-27)



Converging Flow

Branch, $C_{c,b}$

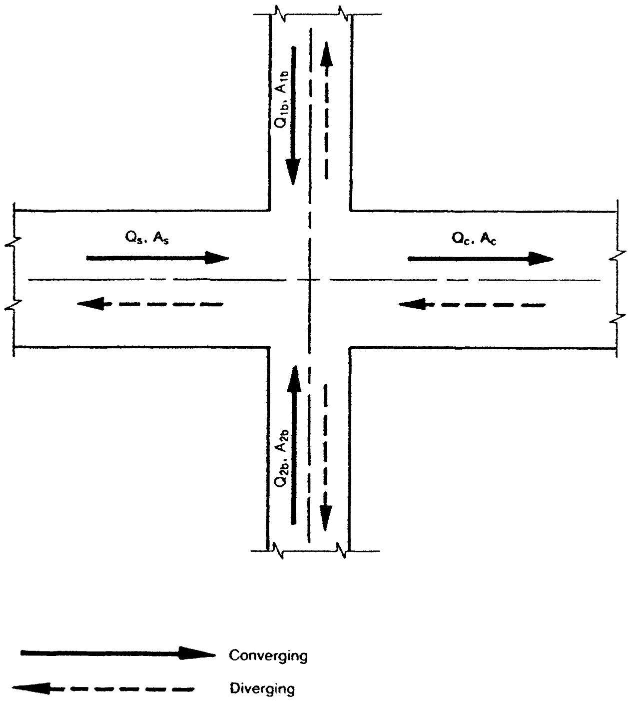
Q_{2b}/Q_{1b}	Q_{1b}/Q_c 0	0.1	0.2	0.3	0.4	0.5	0.6
				$A_{1b}/A_c = 0.2$			
0.5	–1.0	–0.36	0.59	1.8	3.2	4.9	6.8
1.0	–1.0	–0.24	0.63	1.7	2.6	3.7	—
2.0	–1.0	–0.19	0.21	0.04	—	—	—
				$A_{1b}/A_c = 0.4$			
0.5	–1.0	–0.48	–0.02	0.58	0.92	1.3	16
1.0	–1.0	–0.36	0.17	0.55	0.72	0.78	—
2.0	–1.0	–0.18	0.16	–0.06	—	—	—
				$A_{1b}/A_c = 0.6$			
0.5	–1.0	–0.50	–0.07	0.31	0.60	0.82	0.92
1.0	–1.0	–0.37	0.12	0.55	0.60	0.52	—
2.0	–1.0	–0.18	0.26	0.16	—	—	—
				$A_{1b}/A_c = 1.0$			
0.5	–1.0	–0.51	–0.09	0.25	0.50	0.65	0.64
1.0	–1.0	–0.37	0.13	0.46	0.61	0.54	—
2.0	–1.0	–0.15	0.38	0.42	—	—	—

Main, $C_{c,s}$

Q_{2b}/Q_{1b}	Q_s/Q_c 0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
						$A_{1b}/A_c = 0.2$					
0.5 & 2.0	–2.9	–1.9	–1.3	–0.80	–0.56	–0.23	–0.01	0.16	0.22	0.15	0
1.0	–2.5	–1.9	–1.3	–0.80	–0.42	–0.12	0.08	0.20	0.22	0.15	0
						$A_{1b}/A_c = 0.4$					
0.5 & 2.0	–0.98	–0.61	–0.30	–0.05	0.14	0.26	0.33	0.34	0.28	0.17	0
1.0	–0.77	–0.44	–0.16	0.05	0.21	0.31	0.36	0.35	0.29	0.17	0
						$A_{1b}/A_{cc} = 0.6$					
0.5 & 2.0	–0.32	0.08	0.11	0.27	0.37	0.43	0.44	0.40	0.31	0.18	0
1.0	–0.18	–0.04	0.21	0.34	0.42	0.46	0.46	0.41	0.31	0.18	0
						$A_{1b}/A_c = 1.0$					
0.5 & 2.0	0.11	0.36	0.46	0.53	0.57	0.56	0.52	0.44	0.33	0.18	0
1.0	0.29	0.42	0.51	0.57	0.58	0.58	0.54	0.45	0.33	0.18	0

Diverging Flow: Use Fitting 5-23.

5-35 Cross, 90°, Rectangular and Round (Idelchik et al. 1986, Diagram 7-29)



$A_{1b} = A_{2b}$
$A_s = A_c$

Converging Flow

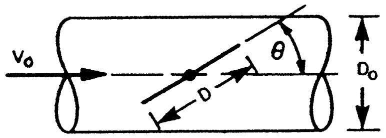
	Branch, $C_{c,b}$						
	Q_{1b}/Q_c or Q_{2b}/Q_c						
Q_{2b}/Q_{1b}	0	0.1	0.2	0.3	0.4	0.5	0.6
			$A_{1b}/A_c = 0.2$				
0.5	–0.85	–0.10	1.1	2.7	4.8	7.3	10
1.0	–0.85	–0.05	1.4	3.1	5.1	7.4	—
2.0	–0.85	–0.31	1.8	3.4	—	—	—
			$A_{1b}/A_c = 0.4$				
0.5	–0 85	–0.29	0.34	1.0	1.8	2.6	3.4
1.0	–0.85	–0.14	0.60	1.3	2.1	2.7	—
2.0	–0.85	0.12	1.0	1.7	—	—	—
			$A_{1b}/A_c = 0.6$				
0.5	–0.85	–0.32	0.20	0.72	1.2	1.7	2.1
1.0	–0.85	–0.18	0.46	1.0	1.5	1.9	—
2.0	–0.85	0.09	0.88	1.4	—	—	—
			$A_{1b}/A_c = 0.8$				
0.5	–0.85	–0.33	0.13	0.61	1.0	1.4	1.7
1.0	–0.85	–0.18	0.41	0.91	1.3	1.5	—
2.0	–0.85	0.08	0.83	1.3	—	—	—
			$A_{1b}/A_c = 1.0$				
0.5	–0.85	–0.34	0.13	0.56	0.93	1.3	1.5
1.0	–0.85	–0.19	0.39	0.86	1.2	1.4	—
2.0	–0.85	0.07	0.81	1.2	—	—	—

	Main					
Q_s/Q_c	0	0.1	0.2	0.3	0.4	0.5
$C_{c,s}$	1.2	1.2	1.2	1.1	1.1	0.96
Q_s/Q_c	0.6	0.7	0.8	0.9	1.0	
$C_{c,s}$	0.85	0.72	0.56	0.39	0.20	

Diverging Flow: Use Fitting 5-23.

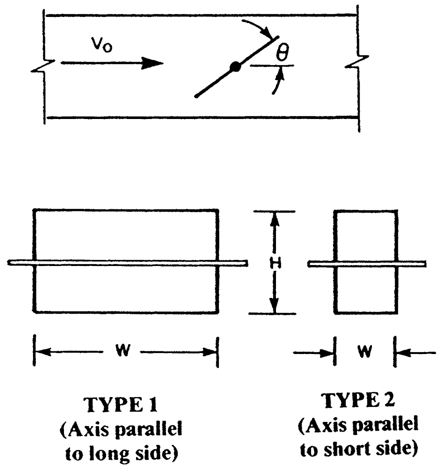
OBSTRUCTIONS

6-1 Damper, Butterfly, Round (Idelchik et al. 1986, Diagram 9-16; Zolotov 1967)



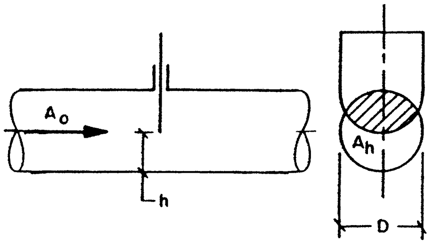
	C_o										
	θ, degrees										
D/D_o	0	10	20	30	40	50	60	70	75	80	85
0.5	019	0.27	0.37	0.49	0.61	0.74	0.86	0.96	0.99	1.0	1.0
0.6	0.19	0.32	0.48	0.69	0.94	1.2	1.5	1.7	1.8	1.9	1.9
0.7	0.19	0.37	0.64	1.0	1.5	2.1	2.8	3.5	3.7	3.9	4.1
0.8	0.19	0.45	0.87	1.6	2.6	4.1	6.1	8.4	9.4	10	10
0.9	0.19	0.54	1.2	2.5	5.0	9.6	17	30	38	45	50
1.0	0.19	0.67	1.8	4.4	11	32	113	—	—	—	—

6-2 Damper, Butterfly, Rectangular (Idelchik et al. 1986, Diagram 9-17; Zolotov 1967)



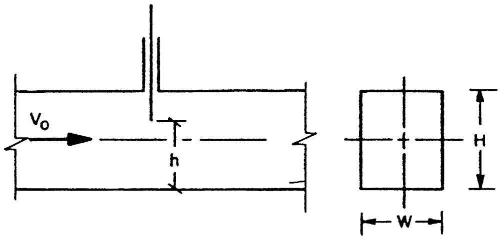
		C_o								
		θ, degrees								
Type	H/W	0	10	20	30	40	50	60	65	70
1	<0.25	0.04	0.30	1.1	3.0	8.0	23	60	100	190
1	0.25-1.0	0.08	0.33	1.2	3.3	9.0	26	70	128	210
2	>1.0	0.13	0.35	1.3	3.6	10	29	80	155	230

6-3 Damper, Gate, Round (Idelchik et al. 1986, Diagram 9-5)



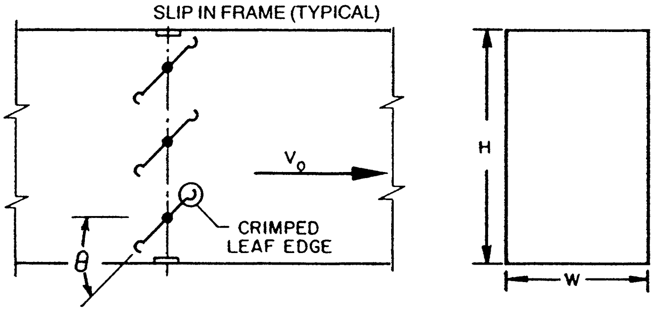
h/D	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
A_h/A_o	0.25	0.38	0.50	0.61	0.71	0.81	0.90	0.96
C_o	35	10	4.6	2.1	0.98	0.44	0.17	0.06

6-4 Damper, Gale, Rectangular (Idelchik et al. 1986, Diagram 9-5)



	h/H						
H/W	**0.3**	**0.4**	**0.5**	**0.6**	**0.7**	**0.8**	**0.9**
0.5	14	6.9	3.3	1.7	0.83	0.32	0.09
1.0	19	8.8	4.5	2.4	1.2	0.55	0.17
1.5	20	9.1	4.7	2.7	1.2	0.47	0.11
2.0	18	8.8	4.5	2.3	1.1	0.51	0.13

6-5 Damper, Rectangular, Parallel Blades (Brown and Fellows 1957)



$$L/R = \frac{NW}{2(H+W)}$$

where

N = number of damper blades

W = duct dimension parallel to blade axis, in.

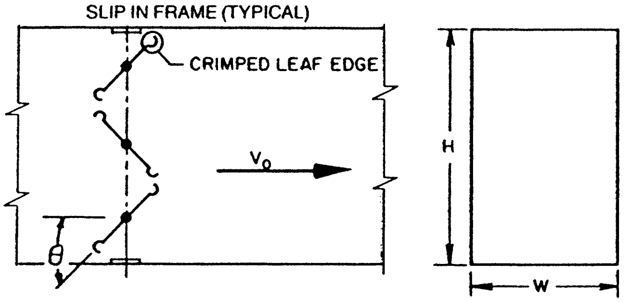
H = duct height, in.

L = sum of damper blade lengths, in.

R = perimeter of duct, in.

	C_o							
	θ, degrees							
L/R	**0**	**10**	**20**	**30**	**40**	**50**	**60**	**70**
0.3	0.52	0.79	1.4	2.3	5.0	9	14	32
0.4	0.52	0.85	1.5	2.4	5.0	9	16	38
0.5	0.52	0.92	1.5	2.4	5.0	9	18	45
0.6	0.52	0.92	1.5	2.4	5.4	9	21	45
0.8	0.52	0.92	1.5	2.5	5.4	9	22	55
1.0	0.52	1.0	1.6	2.6	5.4	10	24	65
1.5	0.52	1.0	1.6	2.7	5.4	10	28	102

6-6 Damper, Rectangular, Opposed Blades (Brown and Fellows 1957)

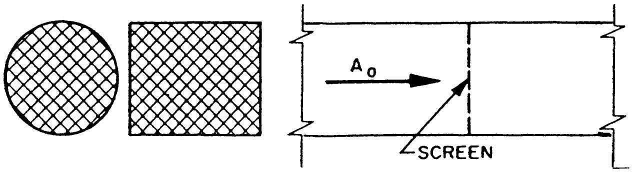


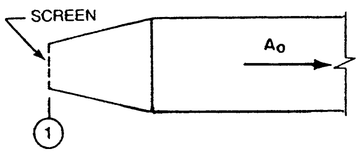
$$L/R = \frac{NW}{2(H+W)}$$

See Fitting 6-5 for definition of terms.

	C_o							
	θ, degrees							
L/R	**0**	**10**	**20**	**30**	**40**	**50**	**60**	**70**
0.3	0.52	0.85	2.1	4.1	9	21	73	284
0.4	0.52	0.92	2.2	5.0	11	28	100	332
0.5	0.52	1.0	2.3	5.4	13	33	122	377
0.6	0.52	1.0	2.3	6.0	14	38	148	411
0.8	0.52	1.1	2.4	6.6	18	54	188	495
1.0	0.52	1.2	2.7	7.3	21	65	245	547
1.5	0.52	1.4	3.2	9.0	28	107	361	677

6-7 Obstruction, Screen, Round and Rectangular (Idelchik et al. 1986, Diagram 8-6)

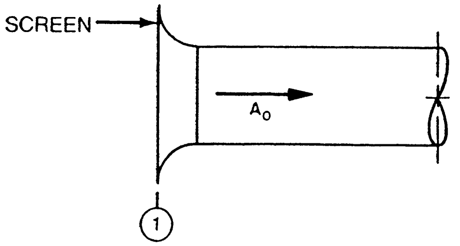




n = free area ratio of screen

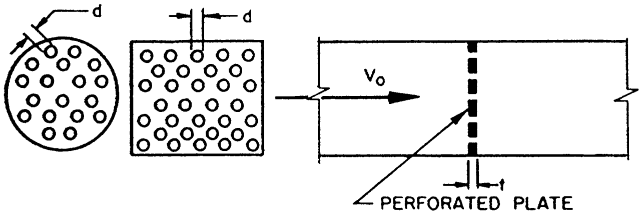
A_o = area of duct

A_1 = cross-sectional area of duct or fitting where screen is located



	C_o							
	n							
A_1/A_o	**0.3**	**0.4**	**0.5**	**0.6**	**0.7**	**0.8**	**0.9**	**1.0**
0.2	155	75	42	24	15	8.0	3.5	0
0.3	69	33	19	11	6.4	3.6	1.6	0
0.4	39	19	10	6.1	3.6	2.0	0.88	0
0.6	17	8.3	4.7	2.7	1.6	0.89	0.39	0
0.8	9.7	4.7	2.7	1.5	0.91	0.50	0.22	0
1.0	6.2	3.0	1.7	0.97	0.58	0.32	0.14	0
1.2	4.3	2.1	1.2	0.67	0.40	0.22	0.10	0
1.4	3.2	1.5	0.87	0.49	0.30	0.16	0.07	0
1.6	2.4	1.2	0.66	0.38	0.23	0.12	0.05	0
2.0	1.6	0.75	0.43	0.24	0.15	0.08	0.04	0
2.5	0.99	0.48	0.27	0.16	0.09	0.05	0.02	0
3.0	0.69	0.33	0.19	0.11	0.06	0.04	0.02	0
4.0	0.39	0.19	0.11	0.06	0.04	0.02	0.01	0
6.0	0.17	0.08	0.05	0.03	0.02	0.01	0	0

6-8 Obstruction, Perforated Plate, Thick, Round, and Rectangular (Idelchik et al. 1986, Diagram 8-6)



$t/d \geq 0.015$

$A_{or} = \pi d^2/4$

$n = \Sigma A_{or}/A_o$

where

A_o = area of duct

A_{or} = orifice area

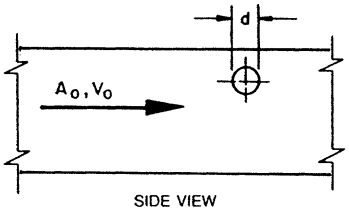
d = diameter of perforated hole

n = free area ratio of plate dimensionless

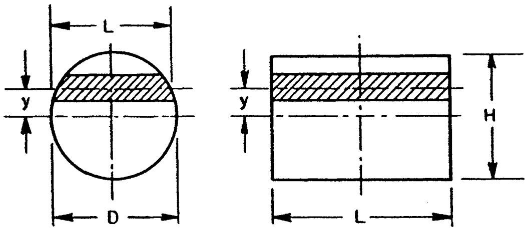
t = plate thickness

	C_o								
	n								
t/d	**0.20**	**0.25**	**0.30**	**0.40**	**0.50**	**0.60**	**0.70**	**0.80**	**0.90**
0.015	52	30	18	8.3	4.0	2.0	0.97	0.42	0.13
0.2	48	28	17	7.7	3.8	1.9	0.91	0.40	0.13
0.4	46	27	17	7.4	3.6	1.8	0.88	0.39	0.13
0.6	42	24	15	6.6	3.2	1.6	0.80	0.36	0.13

6-9 Obstruction, Smooth Cylinder in Round and Rectangular Ducts (Idelchik et al. 1986, Diagram 10-1)



SIDE VIEW



FRONT VIEW

$S_m/A_o < 0.3$

$S_m = dL$

$Re' = dV_o/v$

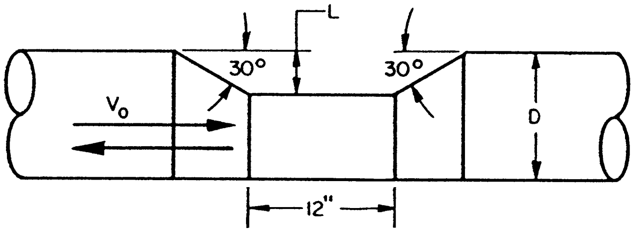
$C_o = KC'_o$

	C'_o			
	S_m/A_o			
Re′	**0.05**	**0.10**	**0.15**	**0.20**
0.1	3.9	8.4	14	19
0.5	1.5	3.2	5.2	7.1
1	0.66	1.4	2.3	3.2
5	0.30	0.64	1.1	1.4
10	0.17	0.38	0.62	0.84
50	0.11	0.24	0.38	0.52
100	0.10	0.21	0.35	0.47
500 to 200,000	0.07	0.15	0.24	0.33
3×10^5	0.07	0.16	0.26	0.35
4×10^5	0.05	0.11	0.19	0.25
5×10^5	0.04	0.09	0.14	0.19
6×10^5 to 10^6	0.02	0.05	0.07	0.10

For obstruction offset from the centerline, use the following factors:

y/D or y/H	0	0.05	0.10	0.15	0.20	0.25	0.30	0.35	0.40
K	1.0	0.97	0.93	0.89	0.84	0.79	0.74	0.67	0.58

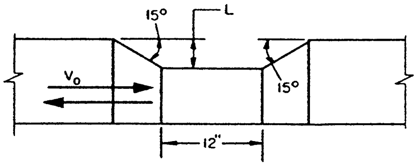
6-10 Round Duct, Depressed to Avoid an Obstruction (SMACNA 1981,Table 6-14I)



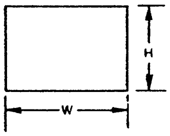
$L/D = 0.33$

$C_o = 0.24$

6-11 Rectangular Duct, Depressed to Avoid an Obstruction (SMACNA 1981, Table 6-14J)



SIDE VIEW

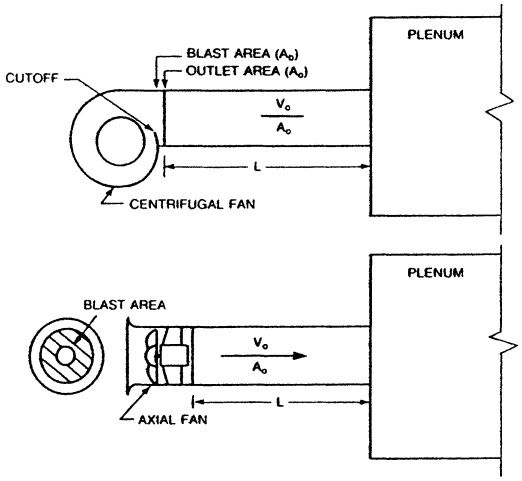


FRONT VIEW

	C_o			
	L/H			
W/H	**0.125**	**0.15**	**0.25**	**030**
1.0	0.26	0.30	0.33	0.35
4.0	0.10	0.14	0.22	0.30

FAN-SYSTEM CONNECTIONS

7-1 Fans Discharging into a Plenum (AMCA 1973, Figure 19)



Calculate effective duct length.

$$(V_o > 2500\text{ fpm: })L_e = V\sqrt{A_o}/10{,}600$$

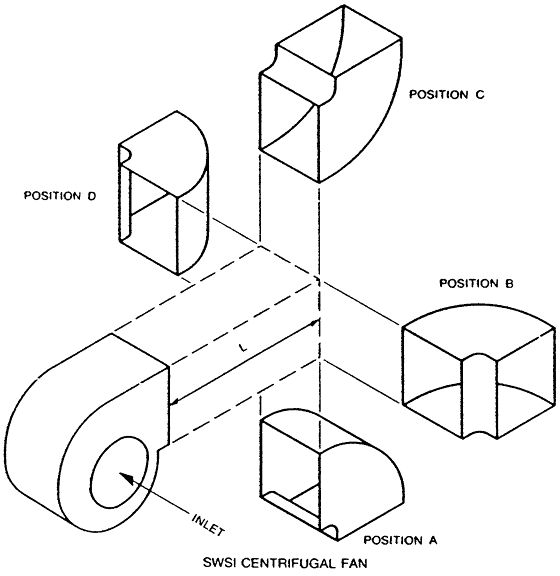
$$V_o \le 2500\text{ fpm: }\ L_e = \sqrt{A_o}/4.3$$

where

V_o = duct velocity, fpm
L_e = effective duct length, ft
A_o = duct area, in^2

	C_o				
	L/L_e				
A_b/A_o	0	0.12	0.25	0.5	≥1.0
0.4	2.0	1.0	0.40	0.18	0
0.5	2.0	1.0	0.40	0.18	0
0.6	1.0	0.66	0.33	0.14	0
0.7	0.8	0.40	0.14	0	0
0.8	0.47	0.22	0.10	0	0
0.9	0.22	0.14	0	0	0
1.0	0	0	0	0	0

7-2 Single Width Single Inlet (SWSI) Fan with an Outlet Duct Elbow (AMCA 1973, Figure 22)

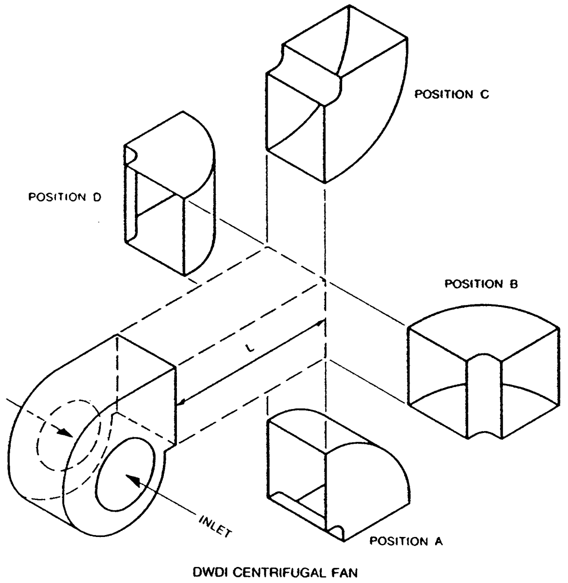


A_b = centrifugal fan blast area (see Fitting 7-1)
A_o = duct/outlet area (see Fitting 7-1)

To calculate effective duct length L_e, see Fitting 7-1.

	C_o					
	Outlet Elbow	L/L_e				
A_b/A_o	Position	0	0.12	0.25	0.5	≥1.0
0.4	A	3.2	2.7	1.8	0.84	0
	B	4.0	3.3	2.2	1.0	0
	C	5.8	4.8	3.2	1.5	0
	D	5.8	4.8	3.2	1.5	0
0.5	A	2.3	1.9	1.3	0.60	0
	B	2.8	2.4	1.6	0.72	0
	C	4.0	3.3	2.2	1.0	0
	D	4.0	3.3	2.2	1.0	0
0.6	A	1.6	1.3	0.88	0.40	0
	B	2.0	1.7	1.1	0.52	0
	C	2.9	2.4	1.6	0.76	0
	D	2.9	2.4	1.6	0.76	0
0.7	A	1.1	0.88	0.60	0.28	0
	B	1.3	1.1	0.72	0.36	0
	C	2.0	1.6	1.1	0.52	0
	D	2.0	1.6	1.1	0.52	0
0.8	A	0.76	0.64	0.44	0.20	0
	B	0.96	0.80	0.52	0.24	0
	C	1.4	1.2	0.76	0.36	0
	D	1.4	1.2	0.76	0.36	0
0.9	A	0.60	0.48	0.32	0.16	0
	B	0.76	0.64	0.44	0.20	0
	C	1.1	0.92	0.64	0.28	0
	D	1.1	0.92	0.64	0.28	0
1.0	A	0.56	0.48	0.32	0.16	0
	B	0.68	0.56	0.36	0.16	0
	C	1.0	0.84	0.56	0.26	0
	D	1.0	0.84	0.56	0.16	0

7-3 Double Width Double Inlet (DWDI) Fan with an Outlet Duct Elbow (AMCA 1973, Figure 22)

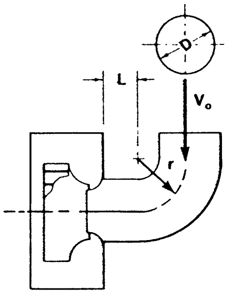


A_b = centrifugal fan blast area (see Fitting 7-1)
A_o = duct/outlet area (see Fitting 7-1)

To calculate effective duct length L_e, see Fitting 7-1.

		C_o				
	Outlet Elbow	L/L_e				
A_b/A_o	**Position**	**0**	**0.12**	**0.25**	**0.5**	**≥1.0**
0.4	A	3.2	2.7	1.8	0.84	0
	B	5.0	4.2	2.8	1.3	0
	C	5.8	4.8	3.2	1.5	0
	D	4.9	4.1	2.7	1.3	0
0.5	A	2.3	1.9	1.3	0.60	0
	B	3.6	3.0	2.0	0.90	0
	C	4 0	3-3	2.2	1.0	0
	D	3.4	2.8	19	0.88	0
0.6	A	1.6	1.3	0.88	0.40	0
	B	2.5	2.1	1.4	0.65	0
	C	2.9	2.4	1.6	0.76	0
	D	2.5	2.1	1.4	0.65	0
0.7	A	1.1	0.88	0.60	0.28	0
	B	1.7	1.4	0.90	0.45	0
	C	2 0	1.6	1.1	0.52	0
	D	1.7	1.4	0.92	0.44	0
0.8	A	0.76	0.64	0.44	0.20	0
	B	1.2	1.0	0.65	0.30	0
	C	1.4	1.2	0.76	0.36	0
	D	1.2	0.99	0.65	0.31	0
0.9	A	0.60	0.48	0.32	0.16	0
	B	0.95	0.80	0.55	0.25	0
	C	1.1	0.92	0.64	0.28	0
	D	0.95	0.78	0.54	0.24	0
1.0	A	0.56	0.48	0.32	0.16	0
	B	0.85	0.70	0.45	0.20	0
	C	1.0	0.84	0.56	0.28	0
	D	0.85	0.71	0.48	0.24	0

7-4 Nonuniform Elbow into a Fan Inlet Induced by a 90° Round Smooth Radius Elbow Without Vanes (AMCA 1973, Fig. 27)

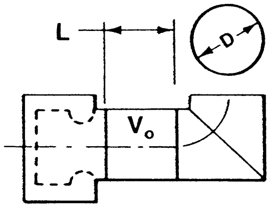


	C_o		
	L/D		
r/D	**0**	**2.0**	**≥5.0**
0.75	1.4	0.80	0.40
1.0	1.2	0.66	0.33
1.5	1.1	0.60	0.33
2.0	1.0	0.53	0.33
3.0	0.66	0.40	0.22

7-5 Nonuniform Elbow into a Fan Inlet Induced by 90° Mitered and Multipiece Elbows Without Vanes (AMCA 1973, Fig. 29)

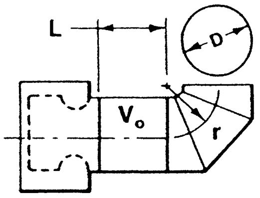
Mitered

L/D	0	2.0	≥5.0
C_o	3.2	2.0	1.0



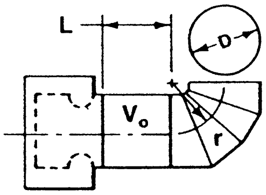
Three-Piece

	C_o		
	L/D		
r/D	**0**	**2.0**	**≥5.0**
0.50	2.5	1.6	0.80
0.75	1.6	1.0	0.47
1.0	1.2	0.66	0.33
1.5	1.1	0.60	0.33
2.0	1.0	0.53	0.33
3.0	0.80	0.47	0.26



Four-Piece or more

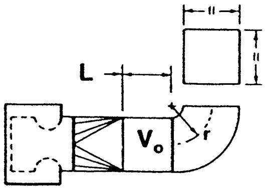
	C_o		
	L/D		
r/D	**0**	**2.0**	**≥5.0**
0.50	1.8	1.0	0.53
0.75	1.4	0.80	0.40
1.0	1.2	0.66	0.33
1.5	1.1	0.60	0.33
2.0	1.0	0.53	0.33
3.0	0.66	0.40	0.22



7-6 Nonuniform Elbow into a Fan Inlet Induced by a 90° Square Smooth Radius Elbow (AMCA 1973, Figure 35A)

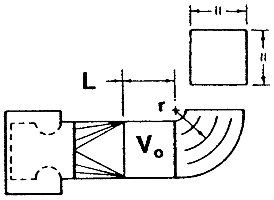
Square Elbow with an Inlet Transition,[a] No Vanes

	C_o		
	L/D		
r/D	**0**	**2.5**	**≥6.0**
0.50	2.5	1.6	0.80
0.75	2.0	1.2	0.66
1.0	1.2	0.66	0.33
1.5	1.0	0.57	0.30
2.0	0.8	0.47	0.26



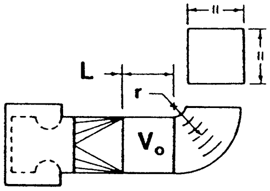
Square Elbow with an Inlet Transition,[a] Full Radius Vanes Equally Spaced

	C_o		
	L/D		
r/D	**0**	**2.5**	**≥6.0**
0.50	0.80	0.47	0.26
1.0	0.53	0.33	0.18
1.5	0.40	0.28	0.16
2.0	1.26	0.22	0.14



Square Elbow with an Inlet Transition,[a] Short Vanes per Fitting 3.8

	C_o		
	L/D		
r/D	**0**	**2.5**	**≥6.0**
0.50	0.80	0.47	0.26
1.0	0.53	0.33	0.18
1.5	0.40	0.28	0.16
2.0	0.26	0.22	0.14



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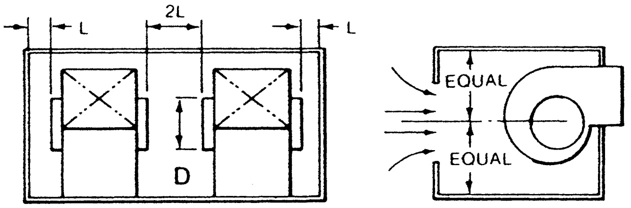
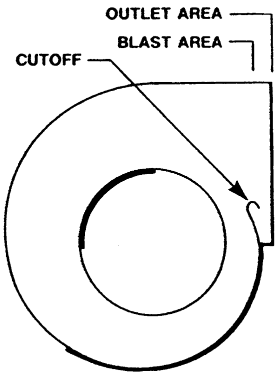
[a] The inside area of the square duct ($H \times H$) is equal to the inside area circumscribed by the fan inlet collar. The maximum angle of any converging element of the transition is 15° and, for a diverging element, 7.5°.

7-7 Fans Located in Plenums and Cabinet Enclosures (AMCA 1973, Figure 35A)



L	C_o
0.75 D	0.22
0.5 D	0.40
0.4 D	0.53
0.3 D	0.80
0.2 D	1.2

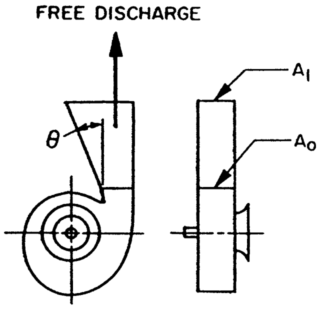
7-8 Fan Without an Outlet Diffuser (AMCA 1973, Figure 19)



Poor; should not be used.

A_b/A_o	0.4	0.5	0.6	0.7	0.8	0.9	1.0
C_o	2.0	2.0	1.0	0.80	0.47	0.22	0

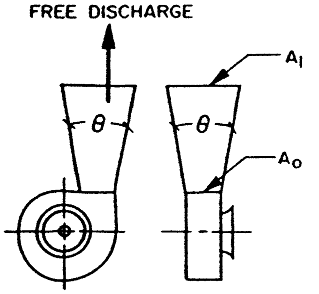
7-9 Plane Asymmetric Diffuser at Fan Outlet Without Ductwork (Idelchik et al. 1986, Diagram 11-11)



	C_o					
θ,	A_1/A_o					
degrees	1.5	2.0	2.5	3.0	3.5	4.0
10	0.51	0.34	0.25	0.21	0.18	0.17
15	0.54	0.36	0.27	0.24	0.22	0.20
20	0.55	0.38	0.31	0.27	0.25	0.24
25	0.59	0.43	0.37	0.35	0.33	0.33
30	0.63	0.50	0.46	0.44	0.43	0.42
35	0.65	0.56	0.53	0.52	0.51	0.50

If diffuser has a screen, use Fitting 6-7 to calculate screen resistance.

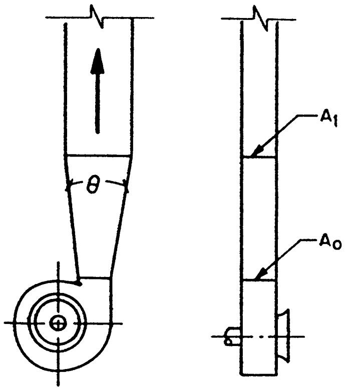
7-10 Pyramidal Diffuser at Fan Outlet Without Ductwork (Idelchik et al. 1986, Diagram 11-11)



	C_o					
θ,	A_1/A_o					
degrees	1.5	2.0	2.5	3.0	3.5	4.0
10	0.54	0.42	0.37	0.34	0.32	0.31
15	0.67	0.58	0.53	0.51	0.50	0.51
20	0.75	0.67	0.65	0.64	0.64	0.65
25	0.80	0.74	0.72	0.70	0.70	0.72
30	0.85	0.78	0.76	0.75	0.75	0.76

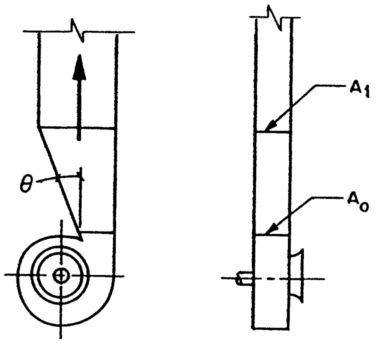
If diffuser has a screen, use Fitting 6-7 to calculate screen resistance.

7-11 Plane Symmetric Diffuser at Fan Outlet with Ductwork (Idelchik et al. 1986, Diagram 5-12)



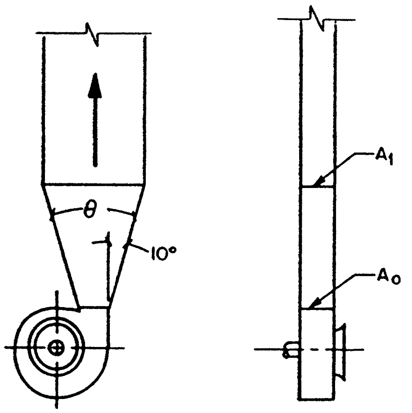
	C_o					
θ,	A_1/A_o					
degrees	1.5	2.0	2.5	3.0	3.5	4.0
10	0.05	0.07	0.09	0.10	0.11	0.11
15	0.06	0.09	0.11	0.13	0.13	0.14
20	0.07	0.10	0.13	0.15	0.16	0.16
25	0.08	0.13	0.16	0.19	0.21	0.23
30	0.16	0.29	0.39	0.32	0.34	0.35
35	0.24	0.34	0.39	0.44	0.48	0.50

7-12 Plane Asymmetric Diffuser at Fan Outlet with Ductwork (Idelchik et al. 1986, Diagram 5-13)



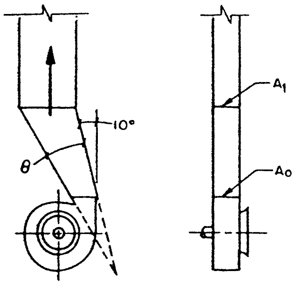
	C_o					
θ,	A_1/A_o					
degrees	1.5	2.8	2.5	3.0	3.5	4.0
10	0.08	0.09	0.10	0.10	0.11	0.11
15	0.10	0.11	0.12	0.13	0.14	0.15
20	0.12	0.14	0.15	0.16	0.17	0.18
25	0.15	0.18	0.21	0.23	0.25	0.26
30	0.18	0.25	0.30	0.33	0.35	0.35
35	0.21	0.31	0.38	0.41	0.43	0.44

7-13 Plane Asymmetric Diffuser at Fan Outlet with Ductwork (Idelchik et al. 1986, Diagram 5-14)



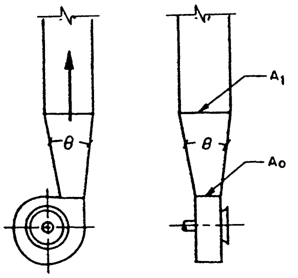
	C_o					
θ,	A_1/A_o					
degrees	1.5	2.8	2.5	3.0	3.5	4.0
10	0.05	0.08	0.11	0.13	0.13	0.14
15	0.06	0.10	0.12	0.14	0.15	0.15
20	0.07	0.11	0.14	0.15	0.16	0.16
25	0.09	0.14	0.18	0.20	0.21	0.22
30	0.13	0.18	0.23	0.26	0.28	0.29
35	0.15	0.23	0.28	0.33	0.35	0.36

7-14 Plane Asymmetric Diffuser at Fan Outlet with Ductwork (Idelchik et al. 1986, Diagram 5-15)



	C_o					
θ,	A_1/A_o					
degrees	1.5	2.8	2.5	3.0	3.5	4.0
10	0.11	0.13	0.14	0.14	0.14	0.14
15	0.13	0.15	0.16	0.17	0.18	0.18
20	0.19	0.22	0.24	0.26	0.28	0.30
25	0.29	0.32	0.35	0.37	0.39	0.40
30	0.36	0.42	0.46	0.49	0.51	0.51
35	0.44	0.54	0.61	0.64	0.66	0.66

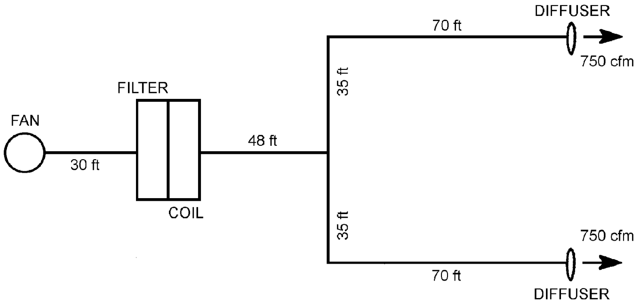
7-15 Pyramidal Diffuser at Fan Outlet with Ductwork (Idelchik et al. 1986, Diagram 5-16)



	C_o					
θ,	A_1/A_o					
degrees	1.5	2.8	2.5	3.0	3.5	4.0
10	0.10	0.18	0.21	0.23	0.24	0.25
15	0.23	0.33	0.38	0.40	0.42	0.44
20	0.31	0.43	0.48	0.53	0.56	0.58
25	0.36	0.49	0.55	0.58	0.62	0.64
30	0.42	0.53	0.59	0.64	0.67	0.69

Example 9.1. The supply duct system for the 100% outside makeup air system of a clean room as shown in the following sketch is to use rectangular, galvanized ducting throughout. An air velocity of 500 fpm is to be used. The following data are known about the components: (a) elbows are mitered, design 2 type (3-8), (b) the flow split is accomplished with a symmetrical wye (5-32), (c) the loss coefficient for the combination of transition section and outlet diffuser C_c is 1.86, (d) the HEPA filter is to be 99.9% efficient, and (e) the coil is a 4-row Series 56 with 12 fins per inch. Data for the filter and coil can be obtained from the given data.

Size each duct and determine the total pressure drop for the supply duct.



Pressure Losses, Inches of Water

Filter Pressure Losses	Velocity, fpm 250 (Clean/Dirty)*	 500 (Clean/Dirty)*
Panel 2 in. pleated 30% efficiency	0.08 / 0.90	0.28 / 0.90
Panel 4 in. pleated 30% efficiency	0.07 / 0.90	0.27 / 0.90
Bag 22 in. deep 60-65% efficiency	0.12 / 1.00	0.30 / 1.00
Bag 22 in. deep 80-85% efficiency	0.28 / 1.00	.045 / 1.00
Bag 22 in. deep 90-95% efficiency	0.50 / 1.50	0.70 / 1.50
Cartridge 12 in. deep 60-65% efficiency	0.15 / 1.50	0.29 / 1.50
Cartridge 12 in. deep 80-85% efficiency	0.27 / 1.50	0.50 / 1.50
Cartridge 12 in. deep 90-95% efficiency	0.34 / 1.50	0.68 / 1.50
HEPA 12 in. deep, 99.97% DOP**	0.60 / 2.00	1.20 / 2.00

* Unit air flow performance should be selected halfway between initial (clean) and final (dirty) filter pressure loss, or as specified.
** HEPA's shown are high flow capacity; rated for 500 fpm face velocity.

Pressure Loss Per Row, Inches of Water

Fin Type	Fins per inch	Face Velocity, fpm 200	 300	 400	 500	 600	 700
Series 56 (5/8 in. tubes)	8	0.050	0.092	0.140	0.196	0.257	0.324
	10	0.063	0.112	0.170	0.234	0.304	0.379
	12	0.075	0.133	0.199	0.271	0.360	0.435

Solution:

$$P_v = \left(\frac{500}{4005}\right)^2 = 0.0156 \text{ in. wg.}$$

Main Duct: 78 ft long, 1500 cfm, 500 fpm

$$A = \frac{1500}{500} = 3 \text{ ft}^2 = \frac{\pi D_e^2}{4}$$

$D_e = 1.95$ ft = 23.5 in.
equivalent rectangular = 30 × 16 to 42 × 12

Select 32 × 16 (lower aspect ratio and easier to match with coil face)

From Fig. 9.2, $\Delta p/100$ ft = 0.013 in. w.g.

$$\Delta p_M = 0.017 \times \frac{78}{100} = 0.013 \text{ in. w.g.}$$

Branches: 105 ft, 750 cfm, 500 fpm

$$750/500 = 1.5 \text{ ft}^2$$

$D_e = 1.38$ ft = 16.6 in.
equivalent rectangular = 16 × 16 to 34 × 8
From Fig. 9.2 $\Delta p/100$ ft = 0.022

$$\Delta p_B = 0.022 \times \frac{105}{100} = 0.023 \text{ in. wg.}$$

Elbows: (type 3-8-2) $C_o = 0.15$ $\Delta p_e = C_o p_v$

$$\Delta p_e = 0.15 \text{ (0.0156 in. w.g.)} = 0.0023 \text{ in. w.g.}$$

Wye: (type 5-32) $C_o = 0.30$

$$\Delta p_Y = 0.30 \text{ (0.0156)} = 0.0047 \text{ in. w.g.}$$

Diffusers: $C_o = 1.86$

$$\Delta p_D = 1.86 \text{ (0.0156)} = 0.029 \text{ in. w.g.}$$

$$\begin{aligned}\Delta p_{\text{TOTAL}} &= \Delta p_M + \Delta p_F + \Delta p_C + \Delta p_Y \\ &\quad + \Delta p_B + \Delta p_e + \Delta p_D \\ &= 0.010 + 1.60(\text{avg}) + (4 \times 0.271) + 0.0047 \\ &\quad + 0.023 + 0.0023 + 0.029 \\ &= 2.75 \text{ in. w.g.}\end{aligned}$$

Example 9.2. Find the total pressure loss in the straight-through section of a 90° cylindrical tee. The velocity in the main upstream section is 2000 fpm and in the main downstream is 1500 fpm. The velocity in the tee branch is 1060 fpm. Also, calculate the total pressure loss between the straight through section and the branch. The value of A_b/A_c is 0.6.

Solution:

(a) For the straight-through section

$$V_c = 2000 \text{ fpm } V_s = 1500 \text{ fpm } V_b = 1060 \text{ fpm}$$

From Equation (9-2),

$$p_{vc} = (2000/4005)^2 = 0.25 \text{ in. of water}$$

For Fitting 5-23, Table 9-4, with θ = 90°, and V_s/V_c = 0.75:

$$c_{c,s} = 0.03$$

By Equation (9-10a),

$$\Delta p_t = C_{c,s} P_{vc}$$
$$\Delta p_t = 0.03 \text{ (0.25)} = 0.0075 \text{ in. of water (negligible)}$$

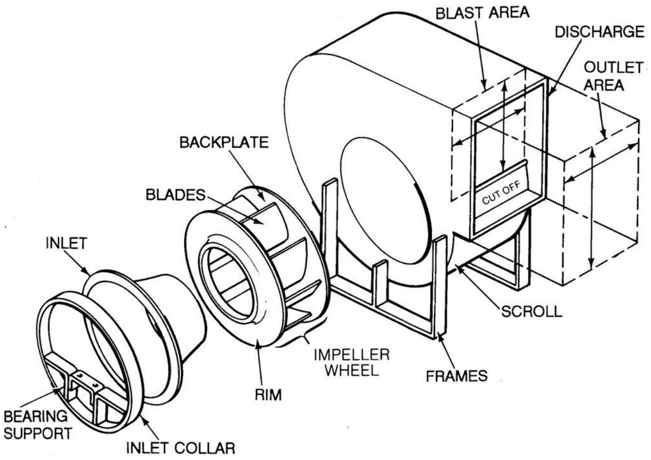


Fig. 9-3 Centrifugal Fan Components
(Figure 1, Chapter 21, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

(b) For the branch section for θ = 90° and $V_b/V_c = 0.53$ and $A_b/A_c = 0.06$, for Fitting 5-23, Table 9-4

$$c_{c,b} = 1.01$$

By Equation (9-10b),

$$\Delta p_1 = C_{c,b}$$

$$\Delta p_t = 1.01\ (0.25) = 0.25 \text{ in. w.g.}$$

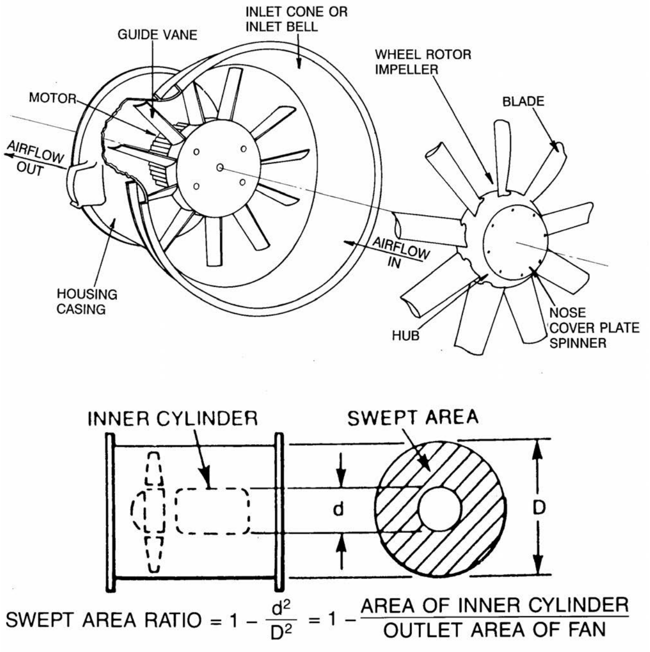
9.2 Fans

A fan is an air pump, a machine that creates a pressure difference and causes airflow. The impeller imparts to the air static and kinetic energy, which varies in proportion, depending on the type of fan.

Fans are classified as centrifugal fans or axial flow fans, according to the direction of airflow through the impeller. The general configuration of a centrifugal fan is shown in Figure 9-3. A similar description of an axial flow fan is shown in Figure 9-4. In addition to these two types, there are several subdivisions of each of the general types. A comparison of typical characteristics of some fan types is shown in Table 9-6.

9.2.1 Principles of Operation

Centrifugal fan impellers produce pressure from two related sources: (1) the centrifugal force created by rotating the air column enclosed between the blades and (2) the kinetic energy imparted to the air by its velocity leaving the impeller. This velocity is a combination of rotative velocity of the impeller and air speed relative to the impeller. When the blades are inclined forward, these two velocities are cumulative, when backward, oppositional. The forward-curved fans depend less on centrifugal force for pressure and more on velocity pressure conversion in the scroll. Conversely, fans with backward-curved blades build up more pressure by centrifugal force and less by velocity conversion in the scroll. However, since the buildup of pressure by centrifugal force is a more efficient form of energy transfer than the conversion of velocity, backward-curved blade fans generally are more efficient than forward-curved blade fans.



Note: The swept area ratio in axial fans is equivalent to the blast area ratio in centrifugal fans.

Fig. 9-4 Axial Fan Components
(Figure 2, Chapter 21, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Axial flow fans produce all of their static pressure from the change in velocity of the air passing through the impeller. These fans are divided into three subtypes. Propeller fans, customarily used at free delivery, are of relatively simple construction. The impellers usually have a small hub-to-tip ratio and are mounted in an orifice plate or inlet ring. Tubeaxial fans are mounted in a cylindrical tube. They have reduced running clearance and operate at higher tip speeds, which give the tubeaxial fan a higher static pressure capability than the propeller fan. Vaneaxial fans are essentially tubeaxial fans with guide vanes that give improved pressure characteristics and efficiency.

9.2.2 Definitions

Volume flow rate, handled by the fan, is the number of cubic feet of air per minute expressed at fan inlet conditions.

Fan total pressure rise is the fan total pressure at outlet minus the fan total pressure at inlet, in. of water.

Fan velocity pressure is the pressure corresponding to the average velocity determined from the volume flow rate and fan outlet area, in. of water.

Fan static pressure rise is the fan total pressure rise diminished by the fan velocity pressure. The fan inlet velocity

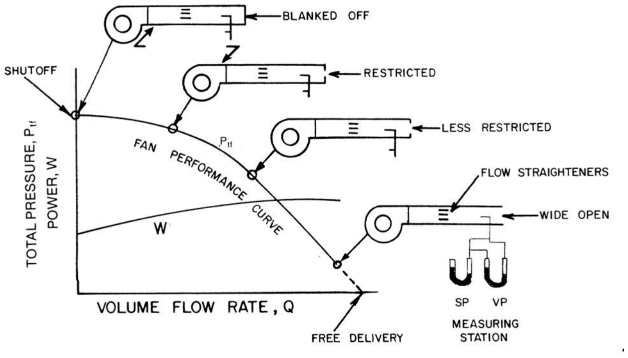


Fig. 9-5 Method of Obtaining Fan Performance Curve
(Figure 3, Chapter 21, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Table 9-5 Fan Laws
(Table 2, Chapter 21, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Law No.	Dependent Variables				Independent Variables
1a	Q_1	=	Q_2	×	$(D_1/D_2)^3\ (N_1/N_2)$
1b	p_1	=	p_2	×	$(D_1/D_2)^2\ (N_1/N_2)^2\ \rho_1/\rho_2$
1c	W_1	=	W_2	×	$(D_1/D_2)^5\ (N_1/N_2)^3\ \rho_1/\rho_2$
2a	Q_1	=	Q_2	×	$(D_1/D_2)^2\ (p_1/p_2)^{1/2}\ (\rho_2/\rho_1)^{1/2}$
2b	N_1	=	N_2	×	$(D_2/D_1)\ (p_1/p_2)^{1/2}\ (\rho_2/\rho_1)^{1/2}$
2c	W_1	=	W_2	×	$(D_1/D_2)^2\ (p_1/p_2)^{3/2}\ (\rho_2/\rho_1)^{1/2}$
3a	N_1	=	N_2	×	$(D_2/D_1)^3\ (Q_1/Q_2)$
3b	p_1	=	p_2	×	$(D_2/D_1)^4\ (Q_1/Q_2)^2\ \rho_1/\rho_2$
3c	W_1	=	W_2	×	$(D_2/D_1)^4\ (Q_1/Q_2)^3\ \rho_1/\rho_2$

Notes:
1. Subscript 1 denotes the variable for the fan under consideration. Subscript 2 denotes the variable for the tested fan.
2. For all fans laws $(\eta_t)_1 = (\eta_t)_2$ and (Point of rating)$_1$ = (Point of rating)$_2$.
3. p equals either p_{tf} or p_{sf}.

head is assumed equal to zero for fan rating purposes, in. of water.

Power output of a fan is expressed as horsepower and is based on the fan volume flow rate and the fan total pressure.

Power input of a fan is expressed as horsepower and is measured as power delivered to the fan shaft.

Mechanical efficiency (or fan total efficiency) of a fan is the ratio of power output to power input.

Static efficiency of a fan is the mechanical efficiency multiplied by the ratio of fan static pressure to fan total pressure.

Point of rating may be any point on the fan performance. curve. For each case, the particular point on the curve must be specifically defined.

9.2.3 Fan Testing

The pilot tube duct traverse can be used to explain the procedure by which a constant speed fan performance curve can be obtained (Figure 9-5). Fans are tested in accordance with *ASHRAE Standard* 51 and various AMCA standards.

The fan is tested from **shutoff** conditions to nearly **free delivery** conditions. At shutoff, the duct is completely blanked off; at free delivery, the outlet resistance is reduced to zero. Between these two conditions, various flow restrictions are placed on the end of the duct to simulate various conditions on the fan. Sufficient points are obtained to define the curve between the shutoff point and free delivery conditions.

Fans designed for duct systems are tested with a length of duct between the fan and the measuring station. This length of duct smoothes the flow of the fan and provides stable, uniform flow conditions at the plane of measurement. The measured pressures are corrected back to fan outlet conditions. Fans designed for use without ducts are tested without ductwork.

Not all sizes are tested for rating. Test information may be used to calculate the performance of larger fans that are geometrically similar, but it should not be extrapolated to smaller fans. For the performance of one fan to be determined from the known performance of another, the two fans must be dynamically similar. Strict dynamic similarity requires that the important nondimensional parameters vary in only insignificant ways. These nondimensional parameters include those that affect the aerodynamic characteristics, such as Mach number, Reynolds number, surface roughness, and gap size. (For more specific information, the manufacturer's application manual or engineering data should be consulted.)

9.2.4 Fan Laws

Fan laws relate the performance variables for any geometrically similar series of fans (Table 9-5). The variables involved are fan size D, rotational speed N, gas density ρ, volume flow rate Q, pressure p_t or p_s, power H (either air or shaft), and mechanical efficiency η_1.

Fan laws mathematically express the fact that when two fans are both members of a geometrically similar series, their performance curves are homologous. At the same point of rating (i.e., at the same relative point on the fan performance curve), efficiencies are equal. Point of rating is sometimes expressed as a stated percent of free delivery airflow. Another method of describing point of rating is the static pressure-velocity pressure ratio p_s/p_v.

Unless otherwise identified, fan performance data are based on a standard air density of 0.075 lb/ft^3 (1.2 kg/m^3). With constant size and speed, the power and pressure varies directly as the ratio of gas density to standard air density.

The application of the fan laws for a change in fan speed N to a specific size fan is illustrated in Figure 9-6. The computed P curve is derived from the base curve. For example, point E ($N_1 = 650$) is computed from point D ($N_2 = 600$) as follows:

At D, $Q_2 = 6000$ and $p_{tf2} = 1.13$
Using Fan Law 1a at Point E

$$Q_1 = 6000\ (650/600) = 6500$$

Using Fan Law 1b

$$p_{t1} = 1.13\ (650/600)^2 = 1.33$$

Table 9-6 Types of Fans

(Table 1, Chapter 21, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

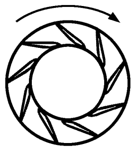
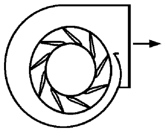
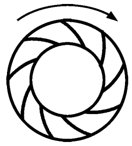
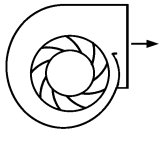
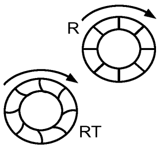
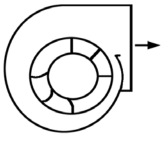
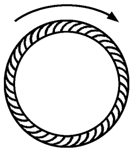
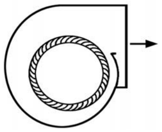
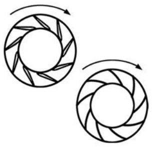
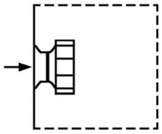
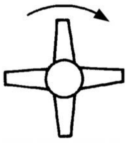
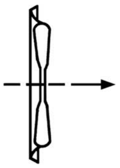
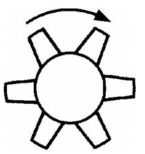
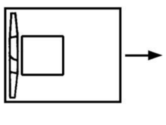
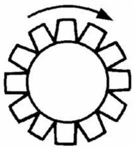
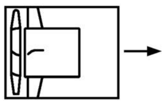
	Type	Impeller Design	Housing Design
Centrifugal Fans	Airfoil	 Blades of airfoil contour curved away from direction of rotation. Deep blades allow efficient expansion within blade passages. Air leaves impeller at velocity less than tip speed. For given duty, has highest speed of centrifugal fan designs.	 Scroll design for efficient conversion of velocity pressure to static pressure. Maximum efficiency requires close clearance and alignment between wheel and inlet.
	Backward-Inclined Backward-Curved	 Single-thickness blades curved or inclined away from direction of rotation. Efficient for same reasons as airfoil fan.	 Uses same housing configuration as airfoil design.
	Radial (R) Radial Tip (Rt)	 Higher pressure characteristics than airfoil, backward-curved, and backward-inclined fans. Curve may have a break to left of peak pressure and fan should not be operated in this area. Power rises continually to free delivery.	 Scroll similar to and often identical to other centrifugal fan designs. Fit between wheel and inlet not as critical as for airfoil and backward-inclined fans.
	Forward-Curved	 Flatter pressure curve and lower efficiency than the airfoil, backward-curved, and backward-inclined. Do not rate fan in the pressure curve dip to the left of peak static pressure. Power rises continually toward free delivery.	 Scroll similar to and often identical to other centrifugal fan designs. Fit between wheel and inlet not as critical as for airfoil and backward-inclined fans.
	Plenum/Plug	 Plenum and plug fans typically use airfoil, backward inclined, or backward curved impellers in a single inlet configuration. Relative benefits of each impeller are the same as those described for scroll housed fans.	 Plenum and plug fans are unique in that they operate with no housing. The equivalent of a housing, or plenum chamber (dashed line), depends on the application. The components of the drive system for the plug fan are located outside the airstream.
Axial Fans	Propeller	 Low efficiency. Limited to low-pressure applications. Usually low-cost impellers have two or more blades of single thickness attached to relatively small hub. Primary energy transfer by velocity pressure.	 Simple circular ring, orifice plate, or venturi. Optimum design is close to blade tips and forms smooth airfoil into wheel.
	Tubeaxial	 Somewhat more efficient and capable of developing more useful static pressure than propeller fan. Usually has 4 to 8 blades with airfoil or single-thickness cross section. Hub is usually less than half the fan tip diameter.	 Cylindrical tube with close clearance to blade tips.
	Vaneaxial	 Good blade design gives medium- to high-pressure capability at good efficiency. Most efficient have airfoil blades. Blades may have fixed, adjustable, or controllable pitch. Hub is usually greater than half fan tip diameter.	 Cylindrical tube with close clearance to blade tips. Guide vanes upstream or downstream from impeller increase pressure capability and efficiency.

Table 9-6 Types of Fans *(Continued)*

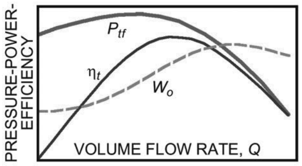
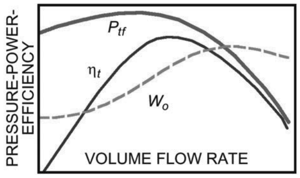
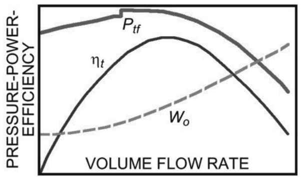
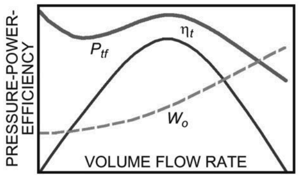
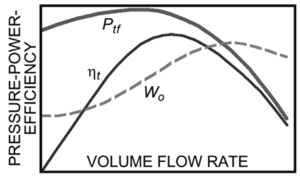
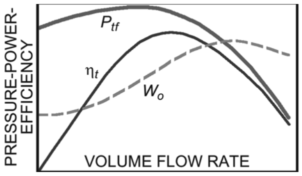
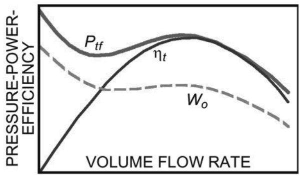
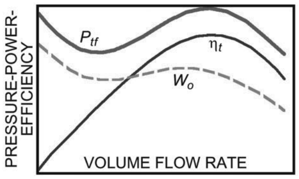
Performance Curves*	Performance Characteristics	Applications
	Highest efficiency of all centrifugal fan designs and peak efficiencies occur at 50 to 60% of wide-open volume. Fan has a non-overloading characteristic, which means power reaches maximum near peak efficiency and becomes lower, or self-limiting, toward free delivery.	General heating, ventilating, and air-conditioning applications. Usually only applied to large systems, which may be low-, medium-, or high-pressure applications. Applied to large, clean-air industrial operations for significant energy savings.
	Similar to airfoil fan, except peak efficiency slightly lower. Curved blades are slightly more efficient than straight blades.	Same heating, ventilating, and air-conditioning applications as airfoil fan. Used in some industrial applications where environment may corrode or erode airfoil blade.
	Higher pressure characteristics than airfoil and backward-curved fans. Pressure may drop suddenly at left of peak pressure, but this usually causes no problems. Power rises continually to free delivery, which is an overloading characteristic. Curved blades are slightly more efficient than straight blades.	Primarily for materials handling in industrial plants. Also for some high-pressure industrial requirements. Rugged wheel is simple to repair in the field. Wheel sometimes coated with special material. Not common for HVAC applications.
	Pressure curve less steep than that of backward-curved fans. Curve dips to left of peak pressure. Highest efficiency occurs at 40 to 50% of wide-open volume. Operate fan to right of peak pressure. Power rises continually to free delivery which is an overloading characteristic.	Primarily for low-pressure HVAC applications, such as residential furnaces, central station units, and packaged air conditioners.
	Plenum and plug fans are similar to comparable housed airfoil/backward-curved fans but are generally less efficient because of inefficient conversion of kinetic energy in discharge air stream. They are more susceptible to performance degradation caused by poor installation.	Plenum and plug fans are used in a variety of HVAC applications such as air handlers, especially where direct-drive arrangements are desirable. Other advantages of these fans are discharge configuration flexibility and potential for smaller-footprint units.
	High flow rate, but very low pressure capabilities. Maximum efficiency reached near free delivery. Discharge pattern circular and airstream swirls.	For low-pressure, high-volume air-moving applications, such as air circulation in a space or ventilation through a wall without ductwork. Used for makeup air applications.
	High flow rate, medium pressure capabilities. Pressure curve dips to left of peak pressure. Avoid operating fan in this region. Discharge pattern circular and airstream rotates or swirls.	Low- and medium-pressure ducted HVAC applications where air distribution downstream is not critical. Used in some industrial applications, such as drying ovens, paint spray booths, and fume exhausts.
	High-pressure characteristics with medium-volume flow capabilities. Pressure curve dips to left of peak pressure. Avoid operating fan in this region. Guide vanes correct circular motion imparted by impeller and improve pressure characteristics and efficiency of fan.	General HVAC systems in low-, medium-, and high-pressure applications where straight-through flow and compact installation are required. Has good downstream air distribution. Used in industrial applications in place of tubeaxial fans. More compact than centrifugal fans for same duty.

Table 9-6 Types of Fans *(Continued)*

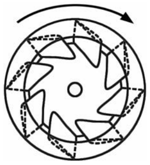
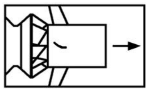
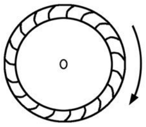
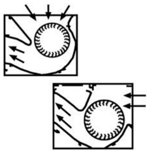
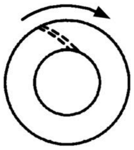
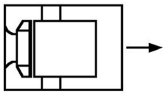
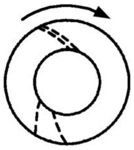
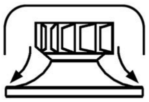
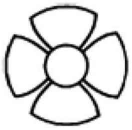
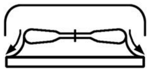
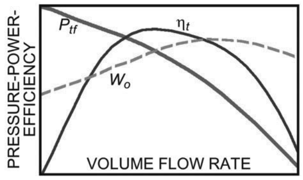
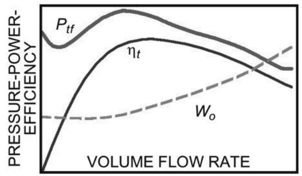
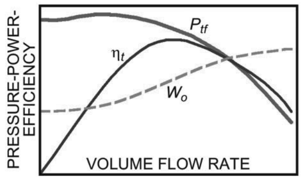
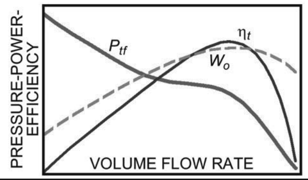
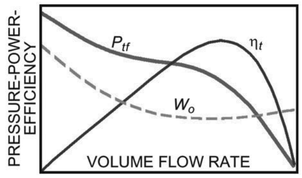
Type				Impeller Design	Housing Design
Mixed-Flow	Mixed-Flow			Combination of axial and centrifugal characteristics. Ideally suited in applications in which the air has to flow in or out axially. Higher pressure characteristic than axial fans.	 The majority of mixed-flow fans are in a tubular housing and include outlet turning vanes. Can operate without housing or in a pipe and duct.
Cross-flow	Cross-flow (Tangential)			Impeller with forward-curved blades. During rotation the flow of air passes through part of the rotor blades into the rotor. This creates an area of turbulence which, working with the guide system, deflects the airflow through another section of the rotor into the discharge duct of the fan casing. Lowest efficiency of any type of fan.	 Special designed housing for 90°or straight through airflow.
Other Designs	Tubular Centri-Fugal			Performance similar to backward-curved fan except capacity and pressure are lower. Lower efficiency than backward-curved fan. Performance curve may have a dip to the left of peak pressure.	 Cylindrical tube similar to vaneaxial fan, except clearance to wheel is not as close. Air discharges radially from wheel and turns 90° to flow through guide vanes.
	Power Roof Ventilators	Centrifugal		Low-pressure exhaust systems such as general factory, kitchen, warehouse, and some commercial installations. Provides positive exhaust ventilation, which is an advantage over gravity-type exhaust units. Centrifugal units are slightly quieter than axial units.	 Normal housing not used, because air discharges from impeller in full circle. Usually does not include configuration to recover velocity pressure component.
		Axial		Low-pressure exhaust systems such as general factory, kitchen, warehouse, and some commercial installations. Provides positive exhaust ventilation, which is an advantage over gravity-type exhaust units. Hood protects fan from weather and acts as safety guard.	 Essentially a propeller fan mounted in a supporting structure. Air discharges from annular space at bottom of weather hood.

Table 9-6 Types of Fans *(Continued)*

Performance Curves*	Performance Characteristics	Applications
	Characteristic pressure curve between axial fans and centrifugal fans. Higher pressure than axial fans and higher volume flow than centrifugal fans.	Similar HVAC applications to centrifugal fans or in applications where an axial fan cannot generate sufficient pressure rise.
	Similar to forward-curved fans. Power rises continually to free delivery, which is an overloading characteristic. Unlike all other fans, performance curves include motor characteristics. Lowest efficiency of any fan type.	Low-pressure HVAC systems such as fan heaters, fireplace inserts, electronic cooling, and air curtains.
	Performance similar to backward-curved fan, except capacity and pressure are lower. Lower efficiency than backward-curved fan because air turns 90°. Performance curve of some designs is similar to axial flow fan and dips to left of peak pressure.	Primarily for low-pressure, return air systems in HVAC applications. Has straight-through flow.
	Usually operated without ductwork; therefore, operates at very low pressure and high volume.	Centrifugal units are somewhat quieter than axial flow units. Low-pressure exhaust systems, such as general factory, kitchen, warehouse, and some commercial installations. Low first cost and low operating cost give an advantage over gravity-flow exhaust systems.
	Usually operated without ductwork; therefore, operates at very low pressure and high volume.	Low-pressure exhaust systems, such as general factory, kitchen, warehouse, and some commercial installations. Low first cost and low operating cost give an advantage over gravity-flow exhaust systems.

*These performance curves reflect general characteristics of various fans as commonly applied. They are not intended to provide complete selection criteria, because other parameters, such as diameter and speed, are not defined.

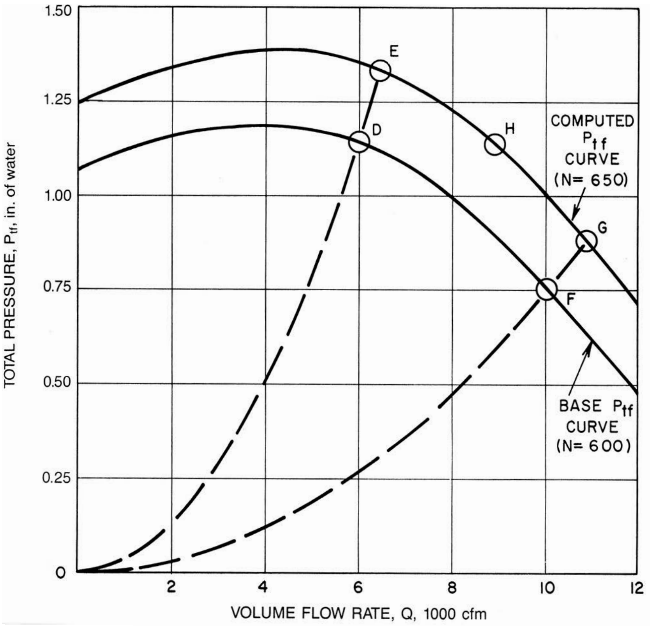


Fig. 9-6 Example Application of Fan Laws
(Figure 4, Chapter 21, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Therefore, the completed P_{tf1}, $N = 650$ curve may be generated by computing additional points from data on the base curve, such as point G from point F. If equivalent points of rating are joined, as illustrated by the dotted lines in Figure 9-6, they will lie on parabolas.

Each point on the base P_{tf} curve determines only one point on the computed curve. For example, there is no way to calculate point H from either point D or point F. It is totally unrelated to either one of these points. Point H is, however, related to some point between these two points on the base P_{tf} curve, and only that point can be used to locate point H. Furthermore, there is no way that point D can be used to calculate the position of point F on the base P_{tf} curve. The entire base curve must be defined by test.

9.2.5 Duct System Characteristics

A simplified duct system with three 90° elbows is shown in Figure 9-7. These elbows represent the resistance offered by the ductwork, heat exchangers, cabinets, dampers, grilles, and other system components. A given rate of airflow through a given system requires a definite total pressure in the system. The resulting total pressure varies as the volume flow rate squared.

The following Equation (9-14) is true for turbulent airflow systems. Heating, ventilating, and air-conditioning systems generally follow this law closely, and no serious error is introduced by its use.

$$\frac{\Delta p_2}{\Delta p_1} = \left(\frac{Q_1}{Q_2}\right)^2 \tag{9-14}$$

The discussion in this chapter is limited to turbulent flow, which is the flow regime in which most fans operate. In some systems, particularly constant or variable volume air conditioning, the air-handling devices and associated controls may produce system resistance curves that deviate widely from Equation (9-14), even though each element of the system may be described by this equation.

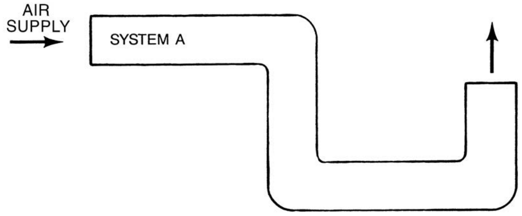


Fig. 9-7 Simplified Duct System with Resistance to Flow Represented by Three 90° Elbows
(Figure 9, Chapter 21, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

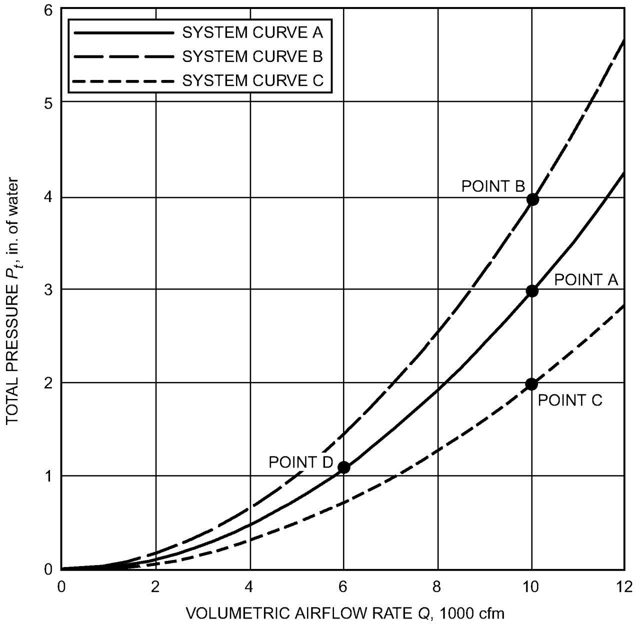


Fig. 9-8 Example System Total Pressure Loss (Δp) Curves
(Figure 10, Chapter 21, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Note that Equation (9-14) permits plotting a turbulent flow system's pressure loss (Δp) curve from one known operating condition (see Figure 9-5). The fixed system must operate at some point on this system curve as the volume flow rate changes. As an example, at point A, curve A, Figure 9-8, when the flow rate through a duct system such as that shown in Figure 9-7 is 10,000 cfm, the total pressure drop is 3 in. of water. If these values are substituted in Equation (9-14) for Δp_1 and Q_1, other points of the system's Δp curve can be determined.

For 6000 cfm

$$\Delta p_2 = 3 \text{ in. of water } (6000/10{,}000)^2 = 1.08 \text{ in. of water}$$

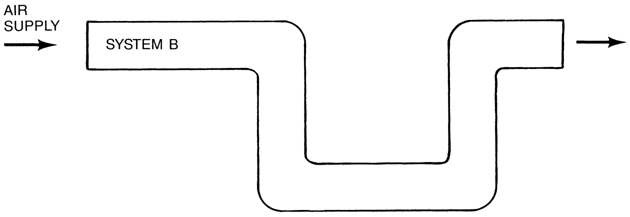


Fig. 9-9 Resistance Added to System of Figure 9-7
(Figure 11, Chapter 21, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

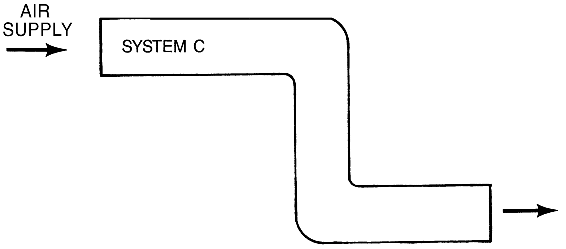


Fig. 9-10 Resistance Removed from that of Figure 9-7
(Figure 12, Chapter 21, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

If a change is made in the system so that the total pressure at design flow rate is increased, the system will no longer operate on the previous Δp curve; a new curve will be defined.

For example, in Figure 9-9, an additional elbow has been added to the schematic duct system shown in Figure 9-7, which increases the total pressure of the system. If the total pressure at 10,000 cfm is increased by 1.0 in. of water, the system total pressure drop at this point will now be 4.0 in. of water, as shown by point B in Figure 9-8. Following the procedure outlined above, a series of Δp points may be computed and a new curve plotted (Δp curve B in Figure 9-8).

If the system in Figure 9-7 is now changed by removing one of the schematic elbows, the resulting system total pressure drops below the total pressure resistance (Figure 9-10). The new Δp curve is shown in Curve C of Figure 9-8. For curve C, a total pressure reduction of 1.0 in. has been assumed when 10,000 cfm flows through the system; thus, the point of operation is at 2.0 in. of water as shown by point C. These three AP curves all follow the relationship expressed in Equation (9-14). Note also that these curves result from changes within the system itself and do not change the fan performance. During the design phase, such system total pressure changes may be due to studies of alternative duct routing, studies of differences in duct sizes, allowance for future duct extensions, or the effect of the design safety factor being applied to the system.

In an actual operating system, these three Δp curves could represent three system characteristic lines that result from three different positions of a throttling control damper. Curve C is the most open position, and curve B is the most closed position of the three positions illustrated. A control damper actually forms a continuous series of these Δp curves as it moves from a wide open position to a completely closed position and covers a much wider range of operation than illustrated here. Such curves could also represent the clogging of turbulent flow filters in a system.

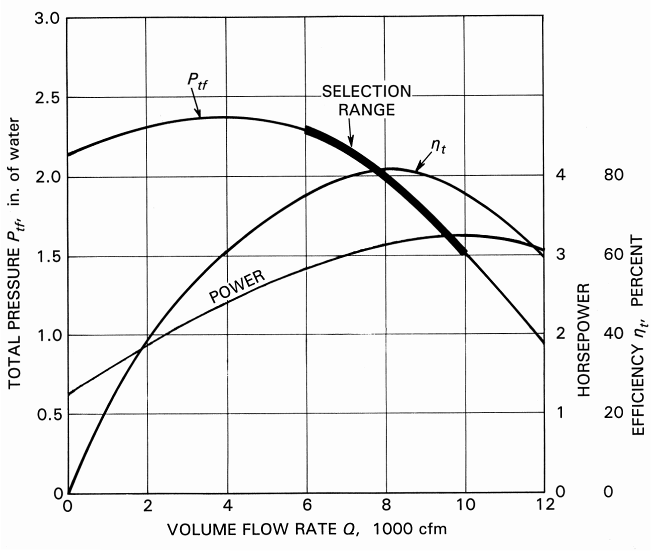


Fig. 9-11 Typical Manufacturer's Fan Performance Curve (Curve shows performance of a fixed fan size running at a fixed speed.)
(Figure 13, Chapter 20, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

9.2.6 Fan Selection

After the system pressure loss curve of the air distribution system has been defined, a fan can be selected to meet the system's requirements. Fan manufacturers present performance data in either the tabular (multi-rating table) or the graphic (curve) as in Figure 9-11, which shows the performance of a fan of a fixed size running at fixed speed. Figure 9-11b is an example of a manufacturer's fan table.

Multi-rating tables usually provide only performance data within the recommended (safe) operating range. The optimum selection area or peak efficiency point is identified in various ways by the different manufacturers. Performance data in the usual fan tables are based on arbitrary increments of flow rate and pressure. In these tables, adjacent data, either horizontally or vertically, represent different points of operation (i.e., different points of rating) on the fan performance curve. These points of rating depend solely on the fan's characteristics; they cannot be obtained one from the other by use of the fan laws. However, these points are usually close together, so intermediate points may be interpolated arithmetically with accuracy adequate for fan selection.

The selection of a fan for a particular air distribution system requires the fan pressure characteristics to fit the system pressure characteristics. Thus, the total system must be evaluated and the flow requirements and resistances existing at

the fan inlet and outlet must be known. The direct effect that certain types of installations have on fan performance must also be considered. Performance pressure changes, known as system effect factors, are a direct effect that must be added to the system resistance before fan selection.

Fan speed and power requirements are then calculated using one of the methods available from fan manufacturers. These may consist of the multi-rating tables as mentioned or of single-speed or multi-speed performance curves or graphs.

The point of operation selected must be at a desirable point on the fan curve, so that maximum efficiency and resistance to stall and pulsation can be attained (Figure 9-12). On systems where more than one point of operation is encountered during operation, the range of performance must be evaluated to determine how the selected fan will react within this complete range. This is particularly true in variable volume systems, where the fan not only experiences a change in performance, but the entire system does not follow the relationship defined in Equation (9-14). In these cases, actual losses in the system at performance extremes must be evaluated.

9.3 Air-Diffusing Equipment

Supply air outlets and diffusing equipment introduce air into a conditioned space to obtain a desired indoor atmospheric environment. Return and exhaust air are removed from the space through return and exhaust inlets. Various types of diffusing equipment are available as standard manufactured products. Refer to Chapter 20 of the 2017 *ASHRAE Handbook—Fundamentals* and Chapter 20 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for additional details concerning space air distribution.

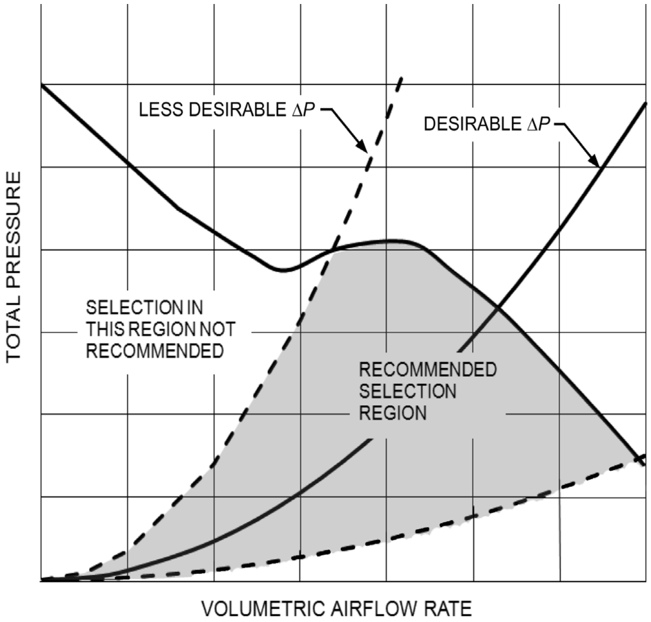


Fig. 9-12 Desirable Combination of p_{tf} *and* Δp *Curves*
(Figure 14, Chapter 21, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Air diffusion in warm air heating, ventilating, and air-conditioning systems should create the proper combination of temperature, humidity, and air motion in the occupied zone of the conditioned room [floor to 6 ft (1.8 m) above floor level]. Standard limits have been established as **acceptable, effective draft temperature**. This term comprises air temperature, air motion, relative humidity, and their physiological effect on the human body. Any variation from accepted standards of one of these elements results in discomfort to the occupants. Such discomfort may arise due to excessive room air temperature variations (horizontally, vertically, or both), excessive air motion (draft), failure to deliver or distribute the air according to the load requirements at the different locations, or overly rapid fluctuation of room temperature.

To define the difference θ in effective draft temperature between any point in the occupied zone and the control condition, investigators have used the following equation:

$$\theta = (t_x - t_c) - a(V_x - b) \quad (9\text{-}15)$$

where

t_x = local airstream dry-bulb temperature, °F
V_x = local airstream velocity, ft/min
t_c = average room dry-bulb temperature, °F
b = 30
a = 0.07

Equation (9-15) accounts for the feeling of coolness produced by air motion. It also shows that a 1.8°F temperature change is equal to a 25 fpm velocity change. In summer, the local airstream temperature t_x is below the control temperature. Hence, both temperature and velocity terms are negative when the velocity V_x is greater than 30 fpm, and both of them add to the feeling of coolness. If, in winter, t_x is above the control temperature, any air velocity above 30 fpm subtracts from the feeling of warmth produced by t_x. Therefore, it is usually possible to have zero difference in effective temperature between location “x” and the control point in winter, but not in summer.

Conditioned air is normally supplied to air outlets at velocities much higher than would be acceptable in the occupied zone. The conditioned air temperature may be above, below, or equal to the temperature of the air in the occupied zone. Proper air distribution, therefore, calls for (1) entrainment of room air by the primary airstream outside the zone of occupancy so that air motion and temperature differences are reduced to acceptable limits before the air enters the occupied zone and (2) counteraction of the natural convection and radiation effects within the room.

9.3.1 Supply Air Outlets

The correct types of outlets, properly sized and properly located, control the air pattern within the space to obtain proper air motion and temperature equalization in the occupied zone. Four types of supply outlets are commonly available: (1) grille outlets, (2) slot diffuser outlets, (3) ceiling diffuser outlets, and (4) perforated ceiling panels. These outlets have different con-

SIZE 273

Wheel diameter: 27″

Fan outlet area: 4.19 ft^2

$$\text{Maximum BHP} = 2.87\left(\frac{\text{RPM}}{1000}\right)^3$$

		½″ SP		1″ SP		1½″ SP		2″ SP		2½″ SP		3″ SP		4″ SP		5″ SP		6″ SP		7″ SP		8″ SP	
CFM	**OV**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**
3351	800	487	0.33	643	0.68	783	1.11	904	1.57	1010	2.07	1108	2.61	1280	3.74	1430	4.96	1568	6.29	1692	7.68	1809	9.15
3770	900	511	0.38	652	0.75	784	1.19	904	1.68	1010	2.21	1107	2.76	1279	3.95	1432	5.25	1566	6.61	1694	8.07	1808	9.56
4198	1000	539	0.44	665	0.82	788	1.27	904	1.78	1011	2.34	1106	2.92	1279	4.17	1429	5.51	1567	6.93	1691	8.41	1809	10.0
4608	1100	569	0.51	683	0.91	797	1.37	908	1.91	1011	2.47	1107	3.08	1278	4.38	1430	5.77	1568	7.27	1691	8.79	1808	10.4
5027	1200	601	0.58	705	1.01	810	1.48	915	2.02	1015	2.62	1108	3.24	1278	4.61	1429	6.04	1567	7.58	1691	9.17	1810	10.8
5446	1300	634	0.67	730	1.11	828	1.61	925	2.16	1020	2.77	1110	3.41	1279	4.82	1429	6.31	1556	7.91	1691	9.56	1808	11.2
5865	1400	668	0.77	758	1.23	847	1.74	939	2.31	1027	2.92	1115	3.59	1279	5.02	1429	6.58	1566	8.22	1692	9.94	1809	11.7
6284	1500	704	0.88	787	1.36	871	1.88	955	2.46	1040	3.11	1122	3.77	1282	5.26	1431	6.86	1565	8.53	1692	10.3	1809	12.1
6703	1600	739	1.01	818	1.51	896	2.05	975	2.64	1054	3.28	1133	3.98	1288	5.51	1431	7.13	1566	8.87	1691	10.7	1808	12.5
7541	1800	813	1.28	884	1.84	953	2.42	1022	3.05	1092	3.72	1163	4.44	1303	6.01	1440	7.71	1568	9.51	1692	11.4	1806	13.4
8379	2000	887	1.62	952	2.22	1015	2.86	1077	3.52	1140	4.23	1203	4.97	1329	6.58	1455	8.34	1578	10.2	1696	12.2	1809	14.3
9217	2200	962	2.01	1024	2.69	1081	3.36	1138	4.07	1194	4.81	1252	5.61	1367	7.27	1482	9.08	1595	11.0	1708	13.0	1816	15.2
10055	2400	1039	2.49	1096	3.21	1149	3.94	1202	4.71	1253	5.47	1306	6.31	1411	8.05	1515	9.88	1622	11.8	1726	13.9	1829	16.2
10893	2600	1115	3.01	1170	3.81	1220	4.59	1268	5.41	1317	6.23	1364	7.08	1460	8.87	1558	10.8	1654	12.8	1753	15.0	1850	17.3
11731	2800	1192	3.62	1244	4.49	1293	5.35	1337	6.18	1382	7.07	1427	7.96	1517	9.87	1606	11.8	1696	13.9	1787	16.1	1878	18.4
12569	3000	1270	4.32	1319	5.25	1364	6.14	1408	7.07	1451	8.01	1492	8.95	1576	10.9	1658	12.9	1741	15.0	1826	17.3	1912	19.8
13407	3200	1349	5.13	1395	6.11	1438	7.06	1480	8.05	1520	9.03	1559	10.0	1637	12.0	1714	14.1	1793	16.4	1870	18.7	1952	21.2
14245	3400	1472	6.02	1472	7.07	1513	8.09	1553	9.12	1591	10.1	1628	11.2	1703	13.3	1774	15.5	1850	17.9	1923	20.3	1997	22.7

SIZE 303

Wheel diameter: 30″

Fan outlet area: 5.17 ft^2

$$\text{Maximum BHP} = 4.83\left(\frac{\text{RPM}}{1000}\right)^3$$

		½″ SP		1″ SP		1½″ SP		2″ SP		2½″ SP		3″ SP		4″ SP		5″ SP		6″ SP		7″ SP		8″ SP	
CFM	**OV**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**
4135	800	438	0.41	579	0.84	704	1.36	813	1.94	910	2.56	997	3.21	1151	4.61	1286	6.12	1410	7.76	1523	9.47	1628	11.2
4652	900	460	0.47	586	0.92	705	1.46	813	2.07	910	2.72	996	3.41	1150	4.87	1288	6.47	1409	8.14	1524	9.96	1629	11.8
5169	1000	484	0.54	598	1.01	710	1.58	814	2.21	909	2.88	996	3.61	1150	5.14	1288	6.81	1410	8.55	1524	10.4	1628	12.3
5686	1100	512	0.62	614	1.11	717	1.69	817	2.34	910	3.05	996	3.81	1150	5.41	1286	7.12	1411	8.96	1524	10.8	1630	12.9
6203	1200	541	0.72	634	1.23	730	1.83	823	2.51	913	3.23	996	4.01	1150	5.67	1286	7.45	1409	9.35	1524	11.3	1628	13.4
6720	1300	571	0.83	657	1.37	744	1.98	832	2.66	917	3.41	998	4.21	1151	5.94	1286	7.79	1409	9.74	1522	11.7	1629	13.9
7237	1400	601	0.95	682	1.51	762	2.14	844	2.85	926	3.62	1003	4.43	1152	6.22	1286	8.12	1409	10.1	1522	12.2	1628	14.4
7754	1500	633	1.08	709	1.68	783	2.32	860	3.05	935	3.83	1011	4.67	1154	6.48	1287	8.46	1409	10.5	1522	12.7	1628	15.0
8271	1600	655	1.23	736	1.85	806	2.53	877	3.26	949	4.06	1020	4.92	1158	6.78	1288	8.79	1409	10.9	1522	13.1	1626	15.5
9305	1800	731	1.58	795	2.27	857	2.98	920	3.77	983	4.59	1046	5.48	1172	7.41	1295	9.51	1413	11.7	1522	14.1	1628	16.6
10339	2000	798	1.99	857	2.74	913	3.53	970	4.37	1025	5.22	1083	6.15	1197	8.14	1311	10.3	1420	12.6	1528	15.1	1628	17.6
11373	2200	866	2.48	922	3.32	972	4.14	1023	5.02	1074	5.93	1126	6.91	1230	8.96	1333	11.2	1437	13.6	1537	16.1	1636	18.8
12407	2400	935	3.07	986	3.96	1034	4.86	1081	5.79	1129	6.79	1175	7.78	1270	9.92	1364	12.2	1459	14.6	1553	17.2	1646	20.1
13441	2600	1003	3.72	1053	4.71	1097	5.67	1141	6.65	1184	7.68	1227	8.72	1313	10.9	1402	13.3	1488	15.8	1577	18.5	1665	21.3
14475	2800	1072	4.46	1119	5.53	1163	6.59	1204	7.66	1244	8.71	1283	9.82	1365	12.1	1445	14.6	1526	17.1	1608	19.9	1689	22.8
15509	3000	1142	5.33	1187	6.47	1227	7.57	1267	8.72	1305	9.88	1343	11.0	1417	13.4	1491	15.9	1568	18.7	1642	21.4	1720	24.4
16543	3200	1213	6.32	1255	7.53	1294	8.71	1331	9.92	1368	11.1	1402	12.3	1473	14.8	1544	17.5	1613	20.2	1685	23.2	1756	26.1
17577	3400	1284	7.42	1324	8.72	1361	9.98	1397	11.2	1431	12.5	1465	13.8	1532	16.4	1598	19.2	1664	22.0	1730	25.0	1797	28.1

SIZE 363

Wheel diameter: 37½″

Fan outlet area: 7.66 ft^2

$$\text{Maximum BHP} = 12.8\left(\frac{\text{RPM}}{1000}\right)^3$$

		½″ SP		1″ SP		1½″ SP		2″ SP		2½″ SP		3″ SP		4″ SP		5″ SP		6″ SP		7″ SP		8″ SP	
CFM	**OV**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**	**RPM**	**BHP**
6127	800	360	0.58	475	1.21	578	1.97	669	2.81	750	3.69	823	4.64	953	6.68	1065	8.84	1166	11.1	1260	13.6	1348	16.1
6893	900	377	0.67	481	1.33	580	2.11	668	2.99	784	3.93	821	4.93	952	7.08	1063	9.33	1166	11.7	1261	14.3	1348	17.0
7659	1000	397	0.78	491	1.46	582	2.26	669	3.18	747	4.16	820	5.21	950	7.46	1064	9.86	1166	12.3	1250	15.0	1347	17.8
8425	1100	419	0.91	504	1.61	589	2.44	670	3.37	748	4.41	819	5.49	948	7.82	1061	10.3	1165	12.9	1260	15.7	1347	18.6
9191	1200	442	1.03	520	1.77	598	2.63	675	3.59	749	4.54	819	5.77	946	8.21	1060	10.7	1164	13.5	1258	16.4	1346	19.4
9957	1300	466	1.19	539	1.97	611	2.85	682	3.82	752	4.89	820	6.06	945	8.55	1058	11.2	1162	14.1	1257	17.1	1345	20.2
10723	1400	491	1.36	559	2.18	626	3.09	693	4.09	758	5.18	823	6.36	946	8.96	1058	11.7	1160	14.6	1255	17.7	1344	20.9
11489	1500	517	1.55	580	2.41	643	3.35	705	4.37	767	5.49	829	6.69	947	9.34	1057	12.1	1158	15.2	1253	18.4	1340	21.6
12255	1600	543	1.76	603	2.67	662	3.64	719	4.69	778	5.82	836	7.05	950	9.73	1058	12.6	1157	15.7	1252	19.0	1339	22.4
13787	1800	596	2.23	651	3.26	703	4.31	754	5.41	806	6.61	859	7.88	962	10.6	1062	13.6	1158	16.9	1251	20.3	1336	23.9
15319	2000	650	2.81	701	3.93	748	5.08	795	6.27	842	7.52	888	8.84	981	11.6	1075	14.8	1166	18.1	1253	21.7	1338	25.4
16851	2200	704	3.47	752	4.71	796	5.96	838	7.21	881	8.55	923	9.92	1009	12.8	1094	16.0	1178	19.4	1261	23.1	1342	27.0
18383	2400	759	4.24	804	5.61	845	6.93	885	8.31	924	9.71	963	11.1	1041	14.2	1119	17.5	1197	21.0	1274	24.7	1351	28.7
19915	2600	815	5.13	857	6.61	869	8.08	934	9.54	971	11.0	1007	12.6	1078	15.7	1150	19.1	1221	22.7	1294	26.6	1364	30.5
21447	2800	871	6.15	911	7.75	948	9.33	984	10.9	1017	12.4	1051	14.1	1119	17.4	1184	20.9	1252	24.7	1319	28.6	1386	32.7
22979	3000	926	7.27	966	9.05	1000	10.7	1035	12.4	1066	14.0	1098	15.7	1162	19.3	1223	22.9	1286	26.8	1348	30.8	1410	35.0
24511	3200	984	8.61	1020	10.4	1053	12.2	1086	14.0	1117	15.8	1147	17.6	1206	21.3	1266	25.2	1324	29.2	1381	33.2	1440	37.5
26043	3400	1040	10.0	1075	12.0	1108	14.0	1138	15.8	1168	17.7	1197	19.7	1253	23.6	1309	27.6	1363	31.6	1418	35.9		

BHP shown does not include belt drive losses.

Performance shown is for General Purpose Fans with outlet ducts and with or without inlet ducts.

Fig. 9-11b Manufacturer’s Fan Performance Table

struction features and physical configurations and differ in their performance characteristics (air pattern).

9.3.2 Outlet Selection and Location Procedure

The following procedure is generally used in selecting outlet locations and types of outlets:

1. Determine the amount of air to be supplied to each room. Refer to Chapter 3 to determine air quantities for heating and cooling. Determine the amount of outdoor air to be introduced according to ventilation requirements of appropriate codes.
2. Select the type and quantity of outlets for each room, considering such factors as air quantity required, distance available for throw or radius of diffusion, structural characteristics, and architectural concepts. Table 9-7 is based on experience and typical ratings of various types of outlets. It may be used as a guide to the types of outlets applicable to various room air loadings. Special conditions such as ceiling heights greater than 8 to 12 ft (2.4 to 3.6 m), exposed duct mounting, and so forth, as well as product modifications and unusual conditions of room occupancy can all modify the information in this table. Manufacturers' rating data should be consulted to determine the suitability of the specific outlets.
3. Locate outlets to distribute the air as uniformly as possible throughout the room. Outlets may be sized and located to distribute air in various portions of the room in proportion to the heat gain or loss in those areas.
4. Select proper outlet size from manufacturers' ratings according to the air quantities, discharge velocities, distribution patterns, and sound levels. Note manufacturers' recommendations with regard to use, method of installation, minimum velocities, maximum temperature differentials, and any air distribution characteristics that may limit the performance of the outlet. Give special attention to obstructions to the normal air-distribution pattern.

9.3.3 Noise Control

Sound at an outlet is composed of the sound generation of the outlet (a function of the discharge velocity) and the transmission of systemic noise (a function of the size of the outlet). Higher-frequency sounds are caused by excessive outlet velocity, but they may also be the result of sounds generated in the duct by the moving airstream. Lower-pitched sounds are generally caused by mechanical equipment noise transmitted through the duct system and outlet. The cause of high-frequency sounds can be pinpointed as outlet or system sounds by removing the outlet during operation. A reduction in sound level indicates that the outlet is causing noise. If the sound level remains essentially unchanged, then the system is at fault.

Table 9-7 Guide to Use of Various Outlets

Type of Outlet	Air Loading of Floor Space, cfm/ft^2 [L/(s·m^2)]	Approximate Maximum Air Changes per hour for 10 ft (3 m) Ceiling
Grille	0.6 to 1.2 (3 to 6)	7
Slot	0.8 to 2.0 (4 to 10)	12
Perforated panel	0.9 to 3.0 (5 to 15)	18
Ceiling diffuser	0.9 to 5.0 (5 to 25)	30
Perforated ceiling	1.0 to 10 (5 to 50)	60

Example 9.3. A 12 in. by 18 in. high sidewall grille with an 11.25 in. by 17.25 in. core area (80% free area) has been selected. Calculate the throw to 50 fpm, 100 fpm, and 150 fpm if the airflow is 600 cfm.

Note: Chapter 20 of the 2017 *ASHRAE Handbook—Fundamentals* is required.

Solution:

From Table 1, Chapter 20, 2017 *ASHRAE Handbook—Fundamentals*, the centerline velocity constant $K = 5.0$.

Use Equation 5, Chapter 20 of the 2017 *ASHRAE Handbook—Fundamentals*, to calculate the maximum throw for an outlet.

$$X = \frac{KQ}{V_x\sqrt{A_o}}$$

$$X = \frac{5 \times 600}{V_x\sqrt{11.25 \times 17.25/144}} = \frac{2920}{V_x}$$

Solving for 50 fpm throw, $X = 2920/50 = 58$ ft.

But according to Figure 3, Chapter 20, 2017 *ASHRAE Handbook—Fundamentals*, 50 fpm is in Zone 4, which is typically 20% less than calculated, or

$$X = 58 \times 0.80 = 46 \text{ ft}$$

Solving for 100 fpm throw, $X = 2920/100 = 29$ ft
Solving for 150 fpm throw, $X = 2920/150 = 19$ ft

9.4 Pipe, Tube, and Fittings

9.4.1 Selection and Application

Listed in Tables 9-8 and 9-9 are the common sizes and dimensions for pipes and tubing used in the HVAC&R industry. Regulatory codes and voluntary standards of such organizations as the American Society of Mechanical Engineers (ASME) and American Society for Testing and Materials (ASTM) should be considered when selecting and applying these components.

9.4.2 Materials

Table 9-10 is a guide to materials used in heating and air conditioning. While steel, iron, and copper materials are most commonly used, iron and steel alloys, copper alloys, nickel and nickel alloys, and nonmetallic pipe are finding increasing applications.

Codes, dimensional standards, and material specifications cover service requirements, which consider such effects as

Table 9-8 Steel Pipe Data

(Table 2, Chapter 46, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Nominal Size, in.	Pipe OD, in.	Schedule Number or Weight[a]	Wall Thickness t, in.	Inside Diameter d, in.	Surface Area		Cross Section		Weight		Working Pressure[c] ASTM A53 B to 400°F		
					Outside, ft^2/ft	Inside, ft^2/ft	Metal Area, in^2	Flow Area, in^2	Pipe, lb/ft	Water, lb/ft	Mfr. Process	Joint Type[b]	psig
1/4	0.540	40 ST	0.088	0.364	0.141	0.095	0.125	0.104	0.424	0.045	CW	T	188
		80 XS	0.119	0.302	0.141	0.079	0.157	0.072	0.535	0.031	CW	T	871
3/8	0.675	40 ST	0.091	0.493	0.177	0.129	0.167	0.191	0.567	0.083	CW	T	203
		80 XS	0.126	0.423	0.177	0.111	0.217	0.141	0.738	0.061	CW	T	820
1/2	0.840	40 ST	0.109	0.622	0.220	0.163	0.250	0.304	0.850	0.131	CW	T	214
		80 XS	0.147	0.546	0.220	0.143	0.320	0.234	1.087	0.101	CW	T	753
3/4	1.050	40 ST	0.113	0.824	0.275	0.216	0.333	0.533	1.13	0.231	CW	T	217
		80 XS	0.154	0.742	0.275	0.194	0.433	0.432	1.47	0.187	CW	T	681
1	1.315	40 ST	0.133	1.049	0.344	0.275	0.494	0.864	1.68	0.374	CW	T	226
		80 XS	0.179	0.957	0.344	0.251	0.639	0.719	2.17	0.311	CW	T	642
1-1/4	1.660	40 ST	0.140	1.380	0.435	0.361	0.669	1.50	2.27	0.647	CW	T	229
		80 XS	0.191	1.278	0.435	0.335	0.881	1.28	2.99	0.555	CW	T	594
1-1/2	1.900	40 ST	0.145	1.610	0.497	0.421	0.799	2.04	2.72	0.881	CW	T	231
		80 XS	0.200	1.500	0.497	0.393	1.068	1.77	3.63	0.765	CW	T	576
2	2.375	40 ST	0.154	2.067	0.622	0.541	1.07	3.36	3.65	1.45	CW	T	230
		80 XS	0.218	1.939	0.622	0.508	1.48	2.95	5.02	1.28	CW	T	551
2-1/2	2.875	40 ST	0.203	2.469	0.753	0.646	1.70	4.79	5.79	2.07	CW	W	533
		80 XS	0.276	2.323	0.753	0.608	2.25	4.24	7.66	1.83	CW	W	835
3	3.500	40 ST	0.216	3.068	0.916	0.803	2.23	7.39	7.57	3.20	CW	W	482
		80 XS	0.300	2.900	0.916	0.759	3.02	6.60	10.25	2.86	CW	W	767
4	4.500	40 ST	0.237	4.026	1.178	1.054	3.17	12.73	10.78	5.51	CW	W	430
		80 XS	0.337	3.826	1.178	1.002	4.41	11.50	14.97	4.98	CW	W	695
6	6.625	40 ST	0.280	6.065	1.734	1.588	5.58	28.89	18.96	12.50	ERW	W	696
		80 XS	0.432	5.761	1.734	1.508	8.40	26.07	28.55	11.28	ERW	W	1209
8	8.625	30	0.277	8.071	2.258	2.113	7.26	51.16	24.68	22.14	ERW	W	526
		40 ST	0.322	7.981	2.258	2.089	8.40	50.03	28.53	21.65	ERW	W	643
		80 XS	0.500	7.625	2.258	1.996	12.76	45.66	43.35	19.76	ERW	W	1106
10	10.75	30	0.307	10.136	2.814	2.654	10.07	80.69	34.21	34.92	ERW	W	485
		40 ST	0.365	10.020	2.814	2.623	11.91	78.85	40.45	34.12	ERW	W	606
		XS	0.500	9.750	2.814	2.552	16.10	74.66	54.69	32.31	ERW	W	887
		80	0.593	9.564	2.814	2.504	18.92	71.84	64.28	31.09	ERW	W	1081
12	12.75	30	0.330	12.090	3.338	3.165	12.88	114.8	43.74	49.68	ERW	W	449
		ST	0.375	12.000	3.338	3.141	14.58	113.1	49.52	48.94	ERW	W	528
		40	0.406	11.938	3.338	3.125	15.74	111.9	53.48	48.44	ERW	W	583
		XS	0.500	11.750	3.338	3.076	19.24	108.4	65.37	46.92	ERW	W	748
		80	0.687	11.376	3.338	2.978	26.03	101.6	88.44	43.98	ERW	W	1076
14	14.00	30 ST	0.375	13.250	3.665	3.469	16.05	137.9	54.53	59.67	ERW	W	481
		40	0.437	13.126	3.665	3.436	18.62	135.3	63.25	58.56	ERW	W	580
		XS	0.500	13.000	3.665	3.403	21.21	132.7	72.04	57.44	ERW	W	681
		80	0.750	12.500	3.665	3.272	31.22	122.7	106.05	53.11	ERW	W	1081
16	16.00	30 ST	0.375	15.250	4.189	3.992	18.41	182.6	62.53	79.04	ERW	W	421
		40 XS	0.500	15.000	4.189	3.927	24.35	176.7	82.71	76.47	ERW	W	596
18	18.00	ST	0.375	17.250	4.712	4.516	20.76	233.7	70.54	101.13	ERW	W	374
		30	0.437	17.126	4.712	4.483	24.11	230.3	81.91	99.68	ERW	W	451
		XS	0.500	17.000	4.712	4.450	27.49	227.0	93.38	98.22	ERW	W	530
		40	0.562	16.876	4.712	4.418	30.79	223.7	104.59	96.80	ERW	W	607
20	20.00	20 ST	0.375	19.250	5.236	5.039	23.12	291.0	78.54	125.94	ERW	W	337
		30 XS	0.500	19.000	5.236	4.974	30.63	283.5	104.05	122.69	ERW	W	477
		40	0.593	18.814	5.236	4.925	36.15	278.0	122.82	120.30	ERW	W	581

[a] Numbers are schedule numbers per ASME *Standard* B36.10M; ST = Standard Weight; XS = Extra Strong.

[b] T = Thread; W = Weld

[c] Working pressures were calculated per ASME B31.9 using furnace butt-weld (continuous weld, CW) pipe through 4 in. and electric resistance weld (ERW) thereafter. The allowance A has been taken as

(1) 12.5% of t for mill tolerance on pipe wall thickness, *plus*
(2) An arbitrary corrosion allowance of 0.025 in. for pipe sizes through NPS 2 and 0.065 in. from NPS 2½ through 20, *plus*
(3) A thread cutting allowance for sizes through NPS 2.

Because the pipe wall thickness of threaded standard pipe is so small after deducting the allowance A, the mechanical strength of the pipe is impaired. It is good practice to limit standard weight threaded pipe pressure to 90 psig for steam and 125 psig for water.

Table 9-9 Copper Tube Data

(Table 3, Chapter 46, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Nominal Diameter, in.	Type	Wall Thickness *t*, in.	Diameter		Surface Area		Cross Section		Weight		Working Pressure[a,b,c] ASTM B88 to 250°F	
			Outside *D*, in.	Inside *d*, in.	Outside, ft^2/ft	Inside, ft^2/ft	Metal Area, in^2	Flow Area, in^2	Tube, lb/ft	Water, lb/ft	Annealed, psig	Drawn, psig
1/4	K	0.035	0.375	0.305	0.098	0.080	0.037	0.073	0.145	0.032	851	1596
	L	0.030	0.375	0.315	0.098	0.082	0.033	0.078	0.126	0.034	730	1368
3/8	K	0.049	0.500	0.402	0.131	0.105	0.069	0.127	0.269	0.055	894	1676
	L	0.035	0.500	0.430	0.131	0.113	0.051	0.145	0.198	0.063	638	1197
	M	0.025	0.500	0.450	0.131	0.118	0.037	0.159	0.145	0.069	456	855
1/2	K	0.049	0.625	0.527	0.164	0.138	0.089	0.218	0.344	0.094	715	1341
	L	0.040	0.625	0.545	0.164	0.143	0.074	0.233	0.285	0.101	584	1094
	M	0.028	0.625	0.569	0.164	0.149	0.053	0.254	0.203	0.110	409	766
5/8	K	0.049	0.750	0.652	0.196	0.171	0.108	0.334	0.418	0.144	596	1117
	L	0.042	0.750	0.666	0.196	0.174	0.093	0.348	0.362	0.151	511	958
3/4	K	0.065	0.875	0.745	0.229	0.195	0.165	0.436	0.641	0.189	677	1270
	L	0.045	0.875	0.785	0.229	0.206	0.117	0.484	0.455	0.209	469	879
	M	0.032	0.875	0.811	0.229	0.212	0.085	0.517	0.328	0.224	334	625
1	K	0.065	1.125	0.995	0.295	0.260	0.216	0.778	0.839	0.336	527	988
	L	0.050	1.125	1.025	0.295	0.268	0.169	0.825	0.654	0.357	405	760
	M	0.035	1.125	1.055	0.295	0.276	0.120	0.874	0.464	0.378	284	532
1-1/4	K	0.065	1.375	1.245	0.360	0.326	0.268	1.217	1.037	0.527	431	808
	L	0.055	1.375	1.265	0.360	0.331	0.228	1.257	0.884	0.544	365	684
	M	0.042	1.375	1.291	0.360	0.338	0.176	1.309	0.682	0.566	279	522
	DWV	0.040	1.375	1.295	0.360	0.339	0.168	1.317	0.650	0.570	265	497
1-1/2	K	0.072	1.625	1.481	0.425	0.388	0.351	1.723	1.361	0.745	404	758
	L	0.060	1.625	1.505	0.425	0.394	0.295	1.779	1.143	0.770	337	631
	M	0.049	1.625	1.527	0.425	0.400	0.243	1.831	0.940	0.792	275	516
	DWV	0.042	1.625	1.541	0.425	0.403	0.209	1.865	0.809	0.807	236	442
2	K	0.083	2.125	1.959	0.556	0.513	0.532	3.014	2.063	1.304	356	668
	L	0.070	2.125	1.985	0.556	0.520	0.452	3.095	1.751	1.339	300	573
	M	0.058	2.125	2.009	0.556	0.526	0.377	3.170	1.459	1.372	249	467
	DWV	0.042	2.125	2.041	0.556	0.534	0.275	3.272	1.065	1.416	180	338
2-1/2	K	0.095	2.625	2.435	0.687	0.637	0.755	4.657	2.926	2.015	330	619
	L	0.080	2.625	2.465	0.687	0.645	0.640	4.772	2.479	2.065	278	521
	M	0.065	2.625	2.495	0.687	0.653	0.523	4.889	2.026	2.116	226	423
3	K	0.109	3.125	2.907	0.818	0.761	1.033	6.637	4.002	2.872	318	596
	L	0.090	3.125	2.945	0.818	0.771	0.858	6.812	3.325	2.947	263	492
	M	0.072	3.125	2.981	0.818	0.780	0.691	6.979	2.676	3.020	210	394
	DWV	0.045	3.125	3.035	0.818	0.795	0.435	7.234	1.687	3.130	131	246
3-1/2	K	0.120	3.625	3.385	0.949	0.886	1.321	8.999	5.120	3.894	302	566
	L	0.100	3.625	3.425	0.949	0.897	1.107	9.213	4.291	3.987	252	472
	M	0.083	3.625	3.459	0.949	0.906	0.924	9.397	3.579	4.066	209	392
4	K	0.134	4.125	3.857	1.080	1.010	1.680	11.684	6.510	5.056	296	555
	L	0.110	4.125	3.905	1.080	1.022	1.387	11.977	5.377	5.182	243	456
	M	0.095	4.125	3.935	1.080	1.030	1.203	12.161	4.661	5.262	210	394
	DWV	0.058	4.125	4.009	1.080	1.050	0.741	12.623	2.872	5.462	128	240
5	K	0.160	5.125	4.805	1.342	1.258	2.496	18.133	9.671	7.846	285	534
	L	0.125	5.125	4.875	1.342	1.276	1.963	18.665	7.609	8.077	222	417
	M	0.109	5.125	4.907	1.342	1.285	1.718	18.911	6.656	8.183	194	364
	DWV	0.072	5.125	4.981	1.342	1.304	1.143	19.486	4.429	8.432	128	240
6	K	0.192	6.125	5.741	1.603	1.503	3.579	25.886	13.867	11.201	286	536
	L	0.140	6.125	5.845	1.603	1.530	2.632	26.832	10.200	11.610	208	391
	M	0.122	6.125	5.881	1.603	1.540	2.301	27.164	8.916	11.754	182	341
	DWV	0.083	6.125	5.959	1.603	1.560	1.575	27.889	6.105	12.068	124	232
8	K	0.271	8.125	7.583	2.127	1.985	6.687	45.162	25.911	19.542	304	570
	L	0.200	8.125	7.725	2.127	2.022	4.979	46.869	19.295	20.280	224	421
	M	0.170	8.125	7.785	2.127	2.038	4.249	47.600	16.463	20.597	191	358
	DWV	0.109	8.125	7.907	2.127	2.070	2.745	49.104	10.637	21.247	122	229
10	K	0.338	10.125	9.449	2.651	2.474	10.392	70.123	40.271	30.342	304	571
	L	0.250	10.125	9.625	2.651	2.520	7.756	72.760	30.054	31.483	225	422
	M	0.212	10.125	9.701	2.651	2.540	6.602	73.913	25.584	31.982	191	358
12	K	0.405	12.125	11.315	3.174	2.962	14.912	100.554	57.784	43.510	305	571
	L	0.280	12.125	11.565	3.174	3.028	10.419	105.046	40.375	45.454	211	395
	M	0.254	12.125	11.617	3.174	3.041	9.473	105.993	36.706	45.863	191	358

[a] When using soldered or brazed fittings, the joint determines the limiting pressure.

[b] Working pressures were calculated using ASME *Standard* B31.9 allowable stresses. A 5% mill tolerance has been used on the wall thickness. Higher tube ratings can be calculated using the allowable stress for lower temperatures.

[c] If soldered or brazed fittings are used on hard drawn tubing, use the annealed ratings. Full-tube allowable pressures can be used with suitably rated flare or compression-type fittings.

Table 9-10 Application of Pipe, Fitting and Valves for Heating and Air-Conditioning

(Table 5, Chapter 46, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

				Fitting		System	
Application	**Pipe Material**	**Weight**	**Joint Type**	**Class**	**Material**	**Temperature, °F**	**Maximum Pressure at Temperature[a], psig**
Recirculating Water	Steel (CW)	Standard	Thread	125	Cast iron	250	125
2 in. and smaller	Copper, hard	Type L	Braze or silver solder[b]		Wrought copper	250	150
	PVC	Sch 80	Solvent	Sch 80	PVC	75	
	CPVC	Sch 80	Solvent	Sch 80	CPVC	150	
	PB	SDR-11	Heat fusion		PB	160	
			Insert crimp		Metal	160	
2.5 to 12 in.	A53 B ERW Steel	Standard	Weld	Standard	Wrought steel	250	400
			Flange	150	Wrought steel	250	250
			Flange	125	Cast iron	250	175
			Flange	250	Cast iron	250	400
			Groove		MI or ductile iron	230	300
	PB	SDR-11	Heat fusion		PB	160	
Steam and Condensate	Steel (CW)	Standard[c]	Thread	125	Cast iron		90
2 in. and smaller			Thread	150	Malleable iron		90
	A53 B ERW Steel	Standard[c]	Thread	125	Cast iron		100
			Thread	150	Malleable iron		125
	A53 B ERW Steel	XS	Thread	250	Cast iron		200
			Thread	300	Malleable iron		250
2.5 to 12 in.	Steel	Standard	Weld	Standard	Wrought steel		250
			Flange	150	Wrought steel		200
			Flange	125	Cast iron		100
	A53 B ERW Steel	XS	Weld	XS	Wrought steel		700
			Flange	300	Wrought steel		500
			Flange	250	Cast iron		200
Refrigerant	Copper, hard	Type L or K	Braze		Wrought copper		
	A53 B SML Steel	Standard	Weld		Wrought steel		
Underground Water							
Through 12 in.	Copper, hard	Type K	Braze or silver solder[b]		Wrought copper	75	350
Through 6 in.	Ductile iron	Class 50	MJ	MJ	Cast iron	75	250
	PB	SDR 9, 11	Heat fusion		PB	75	
		SDR 7, 11.5	Insert crimp		Metal	75	
Potable Water, Inside Building	Copper, hard	Type L	Braze or silver solder[b]		Wrought copper	75	350
	Steel, galvanized	Standard	Thread	125	Galv. cast iron	75	125
				150	Galv. mall. iron	75	125
	PB	SDR-11	Heat fusion		PB	75	
			Insert crimp		Metal	75	

[a] Maximum allowable working pressures have been derated in this table. Higher system pressures can be used for lower temperatures and smaller pipe sizes. Pipe, fittings, joints, and valves must all be considered.

[b] Lead- and antimony-based solders should not be used for potable water systems. Brazing and silver solders should be employed.

[c] Extra strong pipe is recommended for all threaded condensate piping to allow for corrosion.

corrosion, scale, thermal or mechanical fatigue, and metallurgical instability at high temperatures. Copper and red brass pipe have always been used in heating, ventilating, refrigeration, and water supply installations because of their corrosion resistance. However, copper and brass are not compatible with ammonia and should not be used in ammonia refrigeration systems.

Plastic pipe has become popular in heating and air-conditioning installations because of its flexibility in handling, lower labor costs, and resistance to corrosive fluids. Pressure and temperature are the basic limitations to be considered. Basically, two kinds of plastics are used:

1. **Thermoplastic**, which melts under heat, hardens when cooled, and can be melted and reworked again and again.
2. **Thermoset**, which hardens and fuses under heat and pressure into a permanent shape and cannot be remelted.

Some thermoplastics remain flexible and have moderate strength for water service where temperature and pressures are moderate. Thermoplastics may be joined by solvent welding and hot-gas welding. Both pipe and fittings are available in IPS sizes.

Epoxy or polyester resins, usually reinforced with glass fibers, are the major ingredient of thermosetting piping. They are available in commercial pipe sizes, joined by either

standard screwed or socket-type fittings. Commercial polyester pipe is suitable for temperatures up to 250°F (120°C).

9.4.3 Pipe Fittings

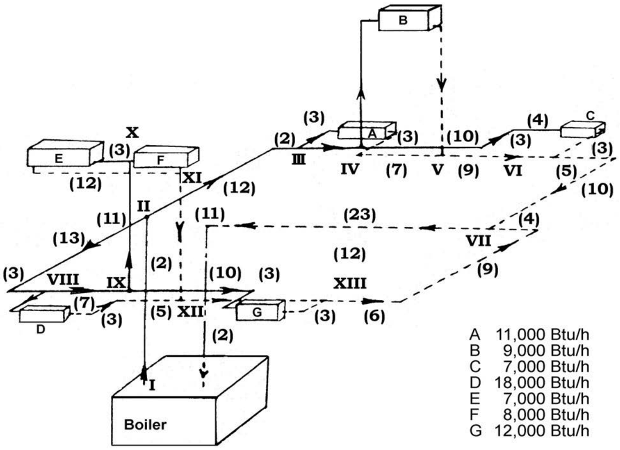
Metal pipe is joined by welding one length to another or by using fittings. Pipe fittings are made in such forms as elbows, tees, couplings, reducers, and unions; they may be screwed, flanged, or welded. The type of fitting used is determined by the pressure of the fluid being carried or by the intended use of the pipeline. The materials generally used are steel, cast iron, malleable iron (heat treated cast iron), copper, brass, stainless steel, alloy steel, or bronze. The material used also depends upon the pressure and fluid characteristics.

Fittings are designated and sized in accordance with ANSI specifications and are identified by their nominal pipe sizes. In the case of reducing tees, crosses, and Y-branches (laterals), the size of the largest run opening is given first, followed by the size of the opening at the opposite end of the run. Where the fitting is a tee or Y-branch (lateral), the size of the outlet is given last. Where the fitting is a cross, the largest side outlet opening is the third dimension given followed by the opening opposite. Where an external thread is wanted, the word male follows the size of that opening; female indicates an internal thread.

9.4.4 Pipe Sizing

Specific procedures for sizing various fluid flow systems using piping or tubing are given in Chapter 22 of the 2017 *ASHRAE Handbook—Fundamentals* and are required for solution.

Example 9.4. Select the design pipe sizes for the two-pipe, forced circulation hot water system shown in the figure below. A 20°F drop in the water temperature has been selected and the water leaves the boiler at 190°F. The numbers in parentheses are the measured linear runs of pipe. The dashed line is the return line pipe. The piping in the system is to be Type L copper tube.



The pressure loss through each convector is given as:

A 1600 milli-inches	E 1425 milli-inches
B 1500 milli-inches	F 1450 milli-inches
C 1450 milli-inches	G 1550 milli-inches
D 1600 milli-inches	

Solution:

First, calculate the gallons per minute circulated through each circuit of the system. For the ABC circuit, the total heat delivered is

$$11{,}000 + 9000 + 7000 = 27{,}000 \text{ Btu/h}$$

The relation between heat flow and flow rate is

$$q = 60Wc_p(t_{\text{in}} - t_{\text{out}}), \text{ or}$$

$$q = 490Q(t_{\text{in}} - t_{\text{out}})$$

where

q = output of the convectors, Btu/h

Q = flow rate, gpm

$(t_{\text{in}} - t_{\text{out}})$ = temperature drop of the water across convector

For this circuit

$$Q = 27{,}000/(490 \times 20) = 2.76 \text{ gpm}$$

For circuit DEFG

$$q = 45{,}000 \text{ Btu/h}$$

and

$$Q = 45{,}000/(490 \times 20) = 4.59 \text{ gpm}$$

Next, select a design friction loss for the system. Assume a design value of 2.5 ft/100 ft (0.3 in./ft). Now the pipe can be sized for each branch in the system, using Figure 15, Chapter 22 of the 2017 *ASHRAE Handbook—Fundamentals*, on the supply side:

for I-II	at 7.34 gpm (72,000 Btu/h), use 1-1/4 in.
II-III	at 2.75 gpm (27,000 Btu/h), use 3/4 in.
III-IV	at 1.63 gpm (16,000 Btu/h), use 5/8 in.
Convector A	at 1.12 gpm (11,000 Btu/h), use 5/8 in.
Convector B	at 0.92 gpm (0,000 Btu/h), use 1/2 in.
Convector C	at 0.71 gpm (7,000 Btu/h), use 1/2 in.
II-VIII	at 4.59 gpm (45,000 Btu/h), use 1 in.
VIII-IX	at 2.75 gpm (27,000 Btu/h), use 3/4 in.
IX-X	at 1.53 gpm (15,000 Btu/h), use 5/8 in.
Convector D	at 1.84 gpm (18,000 Btu/h), use 3/4 in.
Convector E	at 0.71 gpm (7,000 Btu/h), use 1/2 in.
Convector F	at 0.82 gpm (8,000 Btu/h), use 1/2 in.
Convector G	at 1.22 gpm (12,000 Btu/h), use 5/8 in.

and on the return side

for V-VI	at 2.04 gpm (20,000 Btu/h), use 3/4 in.
VI-VII	at 2.75 gpm (27,000 Btu/h), use 3/4 in.
XI-XII	at 1.53 gpm (15,000 Btu/h), use 5/8 in.
XII-XIII	at 3.34 gpm (33,000 Btu/h), use 1 in.
XIII-VII	at 4.59 gpm (45,000 Btu/h), use 1 in.
VII-I	at 7.34 gpm (72,000 Btu/h), use 1-1/4 in.

Next, find the longest run in the system. Apparently, this is the circuit from I-II-IX-X-Convector E-XI-XII-I. The other possible longest run could be one that includes

convector F or one that includes convector B. Both should be checked. Assume here that the convector E run is the longest. The pressure or head loss is found from Figure 15, Chapter 22 of the 2017 *ASHRAE Handbook—Fundamentals*. For example, for 7.34 gpm in a 1-1/4 in. pipe, the friction loss is 1.46 ft/100 ft.

For section I-II the loss is 2 × 1.46 = 2.92 ft/100 ft. The friction losses in the other straight sections are found in a similar manner:

I-II	2.92 ft/100 ft
II-VIII	24.66 ft/100 ft
VII-IX	17.50 ft/100 ft
IX-X	22.92 ft/100 ft
X-Convector	4.37 ft/100 ft
Convector-XI	17.50 ft/100 ft
XI-XII	23.83 ft/100 ft
XII-XIII	12.0 ft/100 ft
XIII-VII	15.0 ft/100 ft
VII-I	33.5 ft/100 ft
Total =	174.2 ft/100 ft

The friction losses for the fittings are as follows:

Elbow (using Figure 15 and Table 10, Chapter 22 of the 2017 *ASHRAE Handbook—Fundamentals*)

2.5 × 1.66 = 4.17 ft/100 ft
2 × 1.2 × 1.46 = 3.50 ft/100 ft
2 × 2.5 × 1.66 = 8.3 ft/100 ft
1.0 × 3.3 × 1.46 = 4.83 ft/100 ft
Total = 20.8 ft/100 ft

Tees (using Figure 15, Figure 7, and Table 17, Chapter 22 of the 2017 *ASHRAE Handbook—Fundamentals*) at

II	8.3 ft/100 ft
IX	8.17 ft/100 ft
X	3.50 ft/100 ft
XI	8.83 ft/100 ft
XII	4.83 ft/100 ft
XIII	8.0 ft/100 ft
VII	10.4 ft/100 ft
Total	= 57.5 ft/100 ft

Convector loss = 1425 milli-inch (1.425 in. of water per foot of pipe or 11.87 ft/100 ft)

Total fitting losses

$$= 20.8 + 57.5 + 11.87 = 90.2 \text{ ft/100 ft}$$

The total friction loss for the longest run is then

$$174.2 + 90.2 = 264.4 \text{ ft/100 ft}$$

Select a boiler and pump combination that will supply 72,000 Btu/h and circulate 7.62 gpm with a developed head of 264.4 ft/100 ft (friction loss).

Notes:

(a) Some friction loss will occur through the boiler, which must be added to the total friction loss of the system.

(b Balancing cocks will need to be placed in the various circuits to balance the flow to each convector. Include the friction loss for these balancing cocks (valves) in the calculation.

(c) Air vent valves, as well as drain valves, should also be put into the system. Also include the friction loss due to these fittings.

Table 9-11 Affinity Laws for Pumps

Impeller Diameter	Speed	Density	To Correct For	Multiply By
Constant	Variable	Constant	Flow	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)$
			Head	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)^2$
			Power	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)^3$
Variable	Constant	Constant	Flow	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)$
			Head	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)^2$
			Power	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)^3$
Constant	Constant	Variable	Power	$\left(\frac{\text{New Density}}{\text{Old Density}}\right)$

9.5 Pumps

Pumps can be classified into three broad categories: (1) reciprocating, (2) rotary, and (3) centrifugal. Both reciprocating and rotary pumps are **positive displacement** pumps. They discharge a fixed amount of liquid for a given speed. The primary moving element of a reciprocating pump is a piston or plunger, while for a rotary pump, it is a rotor. Rotary pumps include gear pumps, vane pumps, lobe pumps, screw pumps, and cam action pumps.

Centrifugal pumps can be classified by the style of impeller—single or double suction, closed or open, radial, Francis, axial—as well as the casing—volute, diffuser, and concentric.

In pumps with radial or Francis impellers, centrifugal force develops the pressure energy and is a function of the impeller peripheral velocity. The liquid enters at the eye of the impeller and energy is added to the liquid by the impeller. The casing collects the liquid as it leaves the impeller and guides it out the discharge of the pump.

In the heating, ventilating, and air-conditioning industry, the most frequently used pump has a radial or Francis enclosed impeller and a volute casing and is single stage.

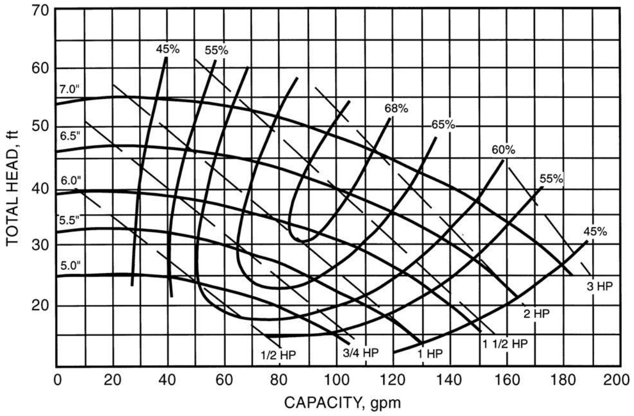


Fig. 9-13 Typical Pump Performance Curves
(Figure 13, Chapter 44, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

9.5.1 Pump Laws

Approximate centrifugal pump performance comparisons of flow, head, and power, depending on impeller diameter, speed, and liquid density are given in Table 9-11.

9.5.2 Pump Performance Curves

Performance of a pump is commonly shown graphically, as in Figure 9-13, which relates the flow (gpm and L/s), the pressure produced (ft and Pa), the power required (bhp and kW), the hydraulic efficiency in percent, the shaft speed (rpm and r/s), and the net positive suction head (ft and Pa, absolute) required for normal operation of the pump with various impeller diameters. Pump curves present the average results obtained from testing several pumps of the same design under standardized test conditions. Manufacturers should be consulted for pump applications that differ considerably from ordinary practice.

The maximum efficiency of a pump is not always the most important feature in making a selection. If the pump is to operate at its peak efficiency, system resistance must also be considered. Systems with no corrosion and systems protected against corrosion should not be designed from pipe friction loss tables containing high corrosion allowances. Such tables show excessive pressure losses, which require selecting a larger pump than necessary. Pumps for systems using liquids for heating and cooling should be hydraulically and mechanically designed for quiet operation, durability, simple service, minimum maintenance, and minimum suction requirements rather than for minimum cost or size.

9.5.3 Hydronic System Characteristics

Hydronic systems in HVAC are all of the loop type (open or closed); the water is circulated through the system and returned to the pumps. No appreciable amount of water is lost from the system, except in the cooling tower, where evaporative cooling occurs.

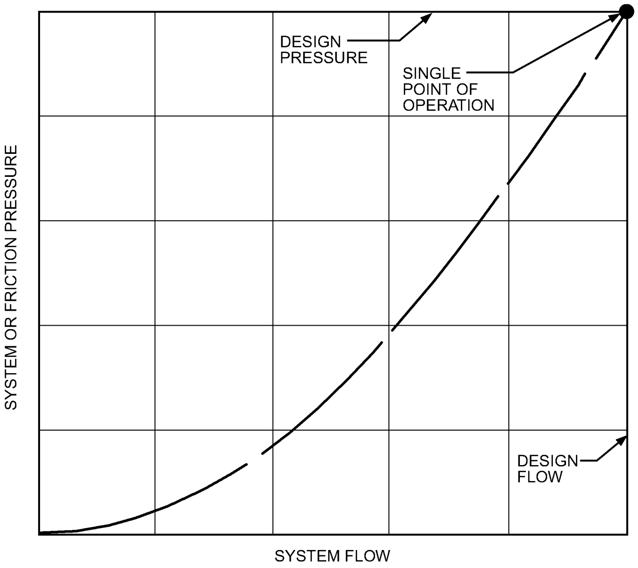


Fig. 9-14 Typical System Curve
(Figure 17, Chapter 44, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Hydronic systems are either full flow or throttling flow. Full flow systems are usually found on residential or small commercial systems where pump motors are small and the energy waste caused by constant flow is not appreciable. Larger systems use flow control valves. These valves control flow in the system in accordance with the heating or cooling load imposed on the system. Hydronic systems can also operate continuously or intermittently. Most hot water or chilled water systems are in continuous operation as long as a heating or cooling load exists on the system. Condensate and boiler feed pumps are often of the intermittent type, starting and stopping as the water level changes in condensate tanks or boilers.

Pumps move water through hydronic systems, overcoming the friction caused by liquid flow through equipment, piping, fittings, and valves. The system head curve graphically describes the flow-head relationship for a hydronic system from minimum to maximum flow (Figure 9-14). The total head consists of the independent head and the system friction head. The independent head is unaffected by the total flow in the system. Typical **independent heads** are: (1) static rise to the top of a cooling tower in condenser water systems; (2) boiler pressure in condensate and boiler feed systems; and (3) control valve and heating or cooling coil friction in hot and chilled water systems.

The head loss of a control valve and its coil is independent of total flow in a system because these head losses can occur at any time, regardless of the total flow on the system; that is, a particular coil may require its maximum flow and, therefore, maximum head loss for the coil and its valve, even though the total flow in the entire system is at a minimum with minimum system friction head.

The other component, **system friction head**, depends on total system flow, increasing or decreasing with total system flow.

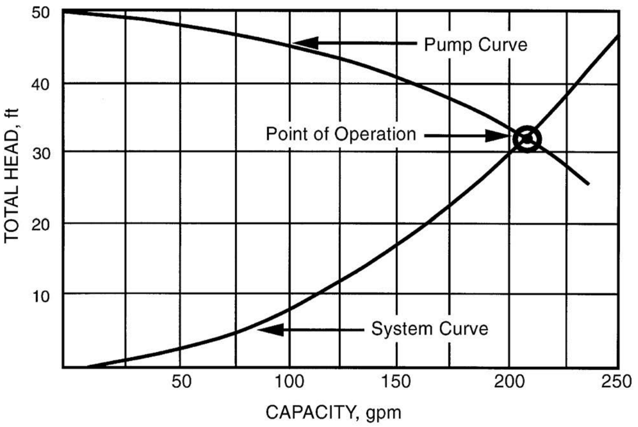


Fig. 9-15 System and Pump Curves
(Figure 19, Chapter 44, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Plotting the system head against the system flow produces a curve as shown in Figure 9-14. If no independent head exists in a hydronic system, the system friction curve becomes the system curve as shown in Figure 9-15. Domestic and small commercial heating and cooling systems without control valves are typical systems without independent head. In actual operation, since flow is not throttled, flow occurs at only one point in the system: at the intersection of the pump head-capacity curve and the system curve (Figure 9-15).

Additional data concerning pump design, selection, and operation can be found in Chapter 44 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

9.6 Problems

9.1 The air velocity in a human occupied zone should not exceed (a) 10 to 25 ft/min, (b) 25 to 40 ft/min, (c) 40 to 64 ft/min, (d) none of the above.

9.2 You are to select the type of outlets for a home to be constructed in Houston, Texas. Discuss your selection of outlets and locations for each of the following combinations: (a) Group A or Group C, (b) Group B or Group E.

9.3 What velocity of air is necessary at a location in a room such that most people will feel neither cool nor warm? Assume that the local temperature is equal to the control temperature of 24.4°C. [Ans: 0.152 m/s]

9.4 Solve the following problems.

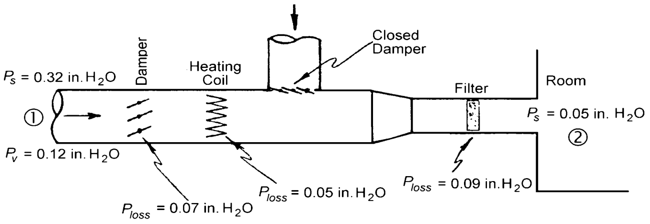
(a) Find the airflow through a 12 in. by 24 in. (305 mm by 610 mm) duct if the static pressure is measured at 0.5 in. of water (125 Pa) and total pressure is measured at 0.54 in. of water (135 Pa)

(b) The pressure difference available to a 60 ft (18.3 m) length of circular duct is 0.2 in. of water (50 Pa). The duct has an ID of 12 in. (305 mm). What rate of airflow is expected?

9.5 For a residential air-conditioning system, one branch duct must supply 207 cfm to one of the rooms. The branch duct has a run of 16 ft. (a) Determine the branch duct size and the pressure drop from the main duct to the room, and (b) specify the supply and return grille sizes for the room.

9.6 How large a duct is required to carry 20,000 cfm (9400 L/s) of air if the velocity is not to exceed 1600 fpm (8.1 m/s)? [Ans: 48 in. (1220 mm)]

9.7 Given the duct system shown below, plot p_v, Δp_s, and Δp_t for the flow through the system.



9.8 Solve the following.

(a) For *Problem 9.7*, find the frictional pressure loss between points (1) and (2).

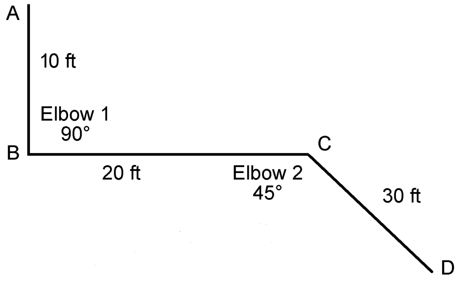
(b) How can the static pressure be increased in a duct system as the air moves away from the fan?

9.9 Determine the dynamic loss of total pressure that occurs in an abrupt expansion from a 1 ft^2 (0.093 m^2) duct to a 2 ft^2 (0.186 m^2) duct carrying 1000 cfm (470 L/s) of air.
[Ans: 0.02 in. of water (5 Pa)]

9.10 Determine the friction loss when circulating 20,000 cfm (9430 L/s) of air at 75°F (23.9°C) through 150 ft (45.7 m) of 36 in. (0.914 m) diameter galvanized steel duct.

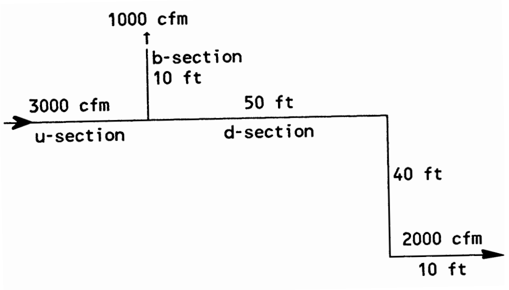
9.11 Find the equivalent rectangular duct for equal friction and capacity for the duct in: *Problem 9.6*, one side is 26 in.

9.12 Find the pressure loss between points A and D for the 12 in. by 12 in. duct shown below. Air at standard conditions is being supplied at the rate of 2000 cfm in galvanized duct of average construction. Elbows No. 1 and No. 2 have center line radii of 13 in. and 24 in., respectively. [Ans: 0.33 in. wg]



9.13 Analyze the air-handling system shown in the following diagram. Determine if a damper is needed in either section (d) or (b), and if so, in what section. There is a damper located in

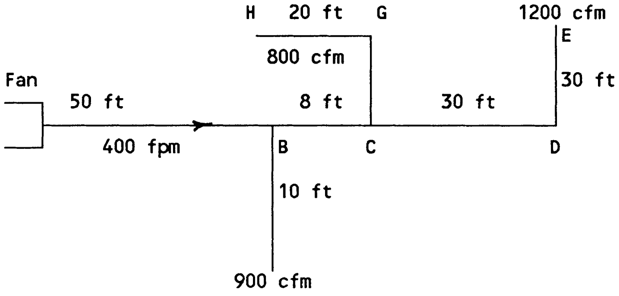
section (u) so that the proper static pressure can be maintained in section (u). If a damper is needed, what is the pressure loss across the damper?



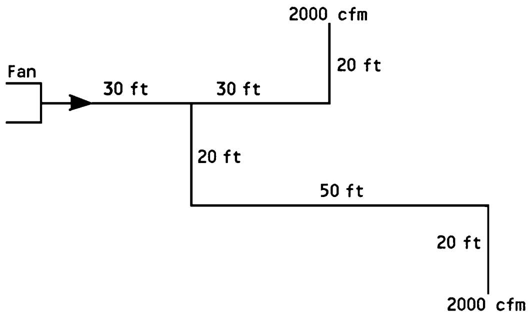
$\rho = 0.075$ lb/ft^3 $f = 0.02$
$D = 12$ in. $(R/D)_{\text{elbows}} = 1$

9.14 A 1 ft high by 3 ft wide main duct carries 2000 cfm of air to a branch where 1500 cfm continues in the 1 ft by 2 ft straight-through section and 500 cfm goes into the branch. Find the actual static pressure regain and the total pressure loss in the straight-through section if the static regain coefficient is 0.80. If the branch takeoff is a 45° cylindrical Y, find the static pressure loss in this section.

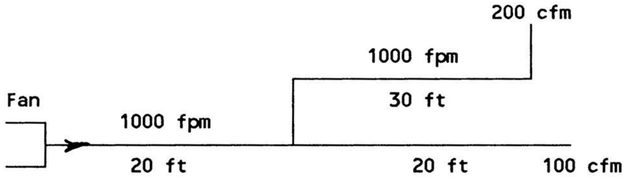
9.15 The supply ductwork for an office space is shown in the following diagram. Size the ductwork by the equal-friction method and calculate the pressure drop. Assume a maximum duct depth of 12 in. and that all duct take-offs are straight rectangular takeoffs. [Ans: $\Delta P = 0.211$ in. wg]



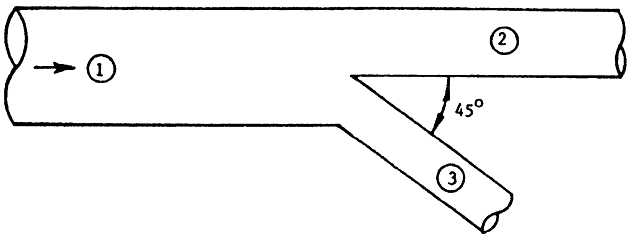
9.16 The following duct system contains circular, galvanized duct. The velocity in the ducts is to be 2000 fpm, and each outlet is to handle 2000 cfm. Each outlet grille has a pressure loss of 0.12 in. of water. Estimate the required pressure increase of the fan. [Ans: 0.97 in. of water]



9.17 Select a fan for the following system. The radius ratio of the elbows is 1.0 and the elbows are three-piece. The pipe is circular. Calculate the frictional pressure loss for the system and the total capacity required by the fan.



9.18 (a) Estimate the total pressure loss between points (1) and (2) and between (1) and (3) in the following take-off.



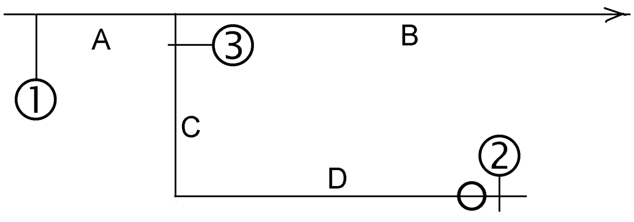
when:
$V_1 = 8.12$ m/s $Q_1 = 1510$ L/s
$V_2 = 6.1$ m/s $Q_2 = 1227$ L/s
$V_3 = 3.05$ m/s $Q_3 = 283$ L/s

The duct is rectangular, of commercial fabrication, and has mastic tape joints. [Ans: 0.0064 in. wg]

(b) Estimate the static pressure at (3) if the static pressure at (1) is 1.0 in. of water. [Ans: 1.05 in. wg]

9.19 Solve the following problems:

(a) What is the expected approximate frictional pressure from (1) to (2) in the following length of duct. Assume round ducts, clean sheet metal, and air at standard temperature and pressure.



Elbow radius = 36 in.
Grille loss = 0.1 in w.g. at 600 fpm

Duct	cfm	Vel., fpm	length
A	2000	1000	40 ft
B	1000	600	–
C	1000	600	30 ft
D	1000	600	30 ft

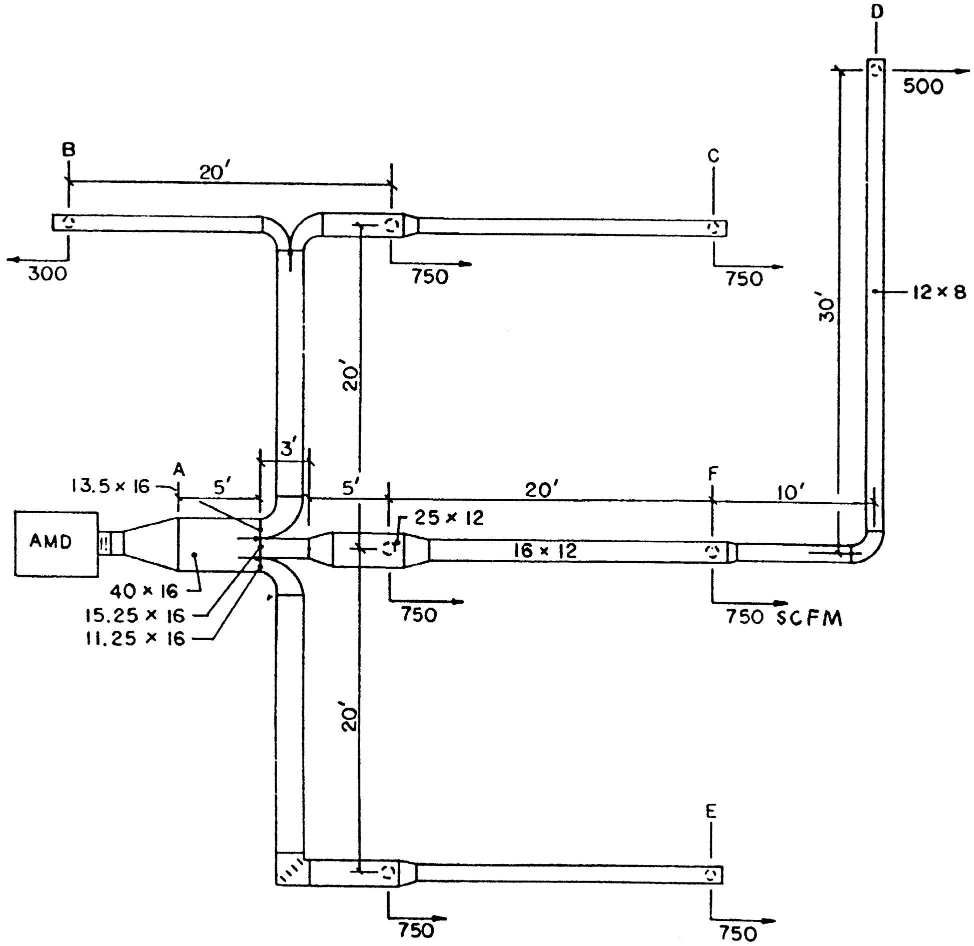


Diagram for Problem 9.20

(b) If the static pressure at (3) is 0.350 in. of water, what friction drop will be required of a damper at 68°F?

(c) What size duct would be required (for ducts C and D) if the damper is eliminated? What is the velocity in the line? Assume a temperature of 68°F. Assume that the static pressure at (3) is still 0.35 in. of water and that the *R*/*D* of the elbow is 2. Also assume that the grille loss is linear with velocity.

9.20 The pressure (energy) loss between A-B, A-C, A-D, and A-E must be equal if a proper air balance is to be achieved. The static pressure required in the duct at points B, C, D, and E to produce the proper flow from the air-diffusing terminal units (ceiling diffusers) is assumed to be uniform and to be 0.10 in. of water. Ignoring interference losses due to terminal unit takeoff,

1. Calculate the total pressure loss between A and D.
2. Size the ducts between A and B.
3. Size the ducts between A and C.
4. Size the ducts between A and E.
5. What would the static pressure be at F?

Note: In practical application, minor static pressure imbalances up to 0.05 in. of water can be absorbed by adjustment of dampers installed in air-diffusing terminal units and further minor adjustments of diverting dampers, in branch fittings. For greater imbalances, duct sizing must be modified and/or butterfly dampers installed in the branch ducts.

9.21 Determine the equivalent feet of pipe for a 2 in. (50 mm) open gate valve and a 2 in. (50 mm) open globe valve at a flow velocity of 5 fps (1.5 m/s). [Ans: 2.95 ft, 70.8 ft]

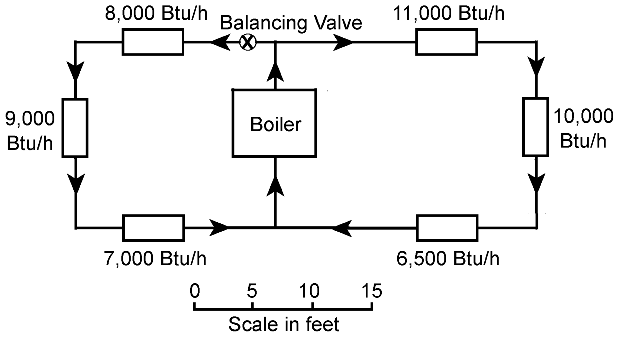
9.22 A convector unit is rated at 1.8 gpm (0.113 L/s) and has a 3.4 ft (10 kPa) pressure loss at rated flow. Estimate the pressure loss with a flow of 2.3 gpm through the convector. [Ans: 6 ft (1.8 m)]

9.23 Size the system shown in Example 9.5 for a 10°F temperature drop.

9.24 Size the system shown in Example 9.5 for 30°F temperature drop.

9.25 Size the system shown for iron pipe. The water leaves the boiler at 200°F and has a 20°F temperature drop. The convectors have a loss given by the equation: Loss (milli-inches) = 0.3 (Btu/h output). Note: 1 milli-inch = 0.001 in. of water.

Assume a 3 ft rise is needed to get to the convectors and then a 3 ft drop to return to the boiler. What head must be developed by the pump and what flow rate (gpm) is required?



9.26 Rework *Problem 9.25* using Type L copper tubing.

9.27 A steam system requires 15,000 lb/h of steam at an initial pressure of 120 psig. The design pressure drop is to be 6 psi per 100 ft. Determine the size of schedule 40 pipe required and the velocity in the steam pipe. [Ans: 3.5 in.]

9.28 Determine the pipe sizes for the refrigeration systems shown in the following figure.

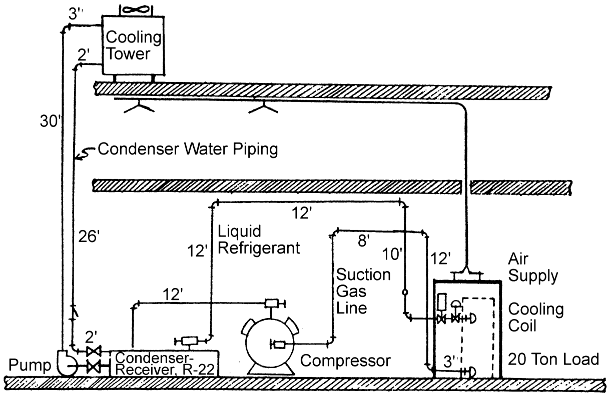


Diagram for Problem 9.28

(a) Refrigerant lines using R-22
(b) Condenser water lines

9.29 A gas appliance has an input rating of 80,000 Btu/h and is operated with natural gas (specific gravity = 0.60) having a heating value of 1050 Btu/ft^3. What size of supply pipe is necessary when the equivalent length is 70 ft and a pressure loss of 0.3 in. of water is allowable?

9.30 A fan operating at 1200 rpm has been delivering 6500 cfm against a static head of 3.25 in. of water and a total head of 5.25 in. of water. The air temperature is 130°F, gage temperature, 90°F, and the input power is 6.1 kW. During a time of power difficulty, the operator notices that the static head is now 2.36 in. of water. There has been no change in the system. Find:

(a) New capacity in cfm
(b) New power input in kW
(c) Original efficiency of the fan (%)
[Ans: 5570 cfm, 3.82 kw, 65.5%]

9.31 A certain damper design introduces a head loss of 0.5 velocity heads when wide open. A damper of this design is to be installed in a 12 in. by 30 in. duct that handles 3000 cfm. The pressure drop in the undampered system is 1.5 in. of water. If the pressure drop through the damper when wide open is to be 5% of the total system resistance, how much cross-sectional area in the duct should the damper occupy?

9.32 What effect on the following parameters does a variation in air density have for a fan operating in a system:
(a) flow rate
(b) developed head
(c) horsepower

9.33 A fan delivers 1500 cfm (708 L/s) of dry air at 65°F (18.3°C) against a static pressure of 0.20 in. of water (50 Pa) and requires 0.10 bhp. Find the volume circulated, the static pressure, and the bhp required to deliver the same weight of air when the air temperature is increased to 165°F (73.9°C). (Note: Atmospheric pressure is constant.) [Ans: 0.168 in. wg, 0.084 hp]

9.34 Should fans be placed before or after air heaters? Why?

9.35 A 40 in. by 24 in. rectangular duct conveying 12,000 cfm of standard air divides into 3 branches (see figure). Branch A carries 6000 cfm for 100 ft, B carries 4000 cfm for 150 ft, and C carries 2000 for 35 ft.

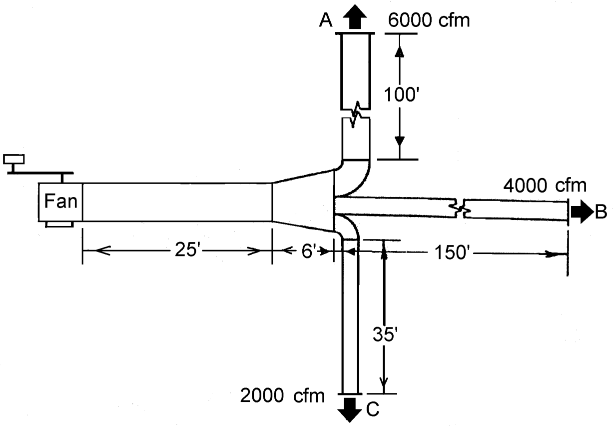


Diagram for Problem 9.35

(a) Size each branch for equal total friction of 0.15 in. of water. Do not exceed upper velocity limit of 2000 fpm.
(b) What is the total friction loss if the same quantity of air, 12,000 cfm of air at 150°F and 14.0 psia, is passed through the same system as part (a)?
(c) For a fan selected for part (a), at what percentage of the speed in part (a) must the fan run to satisfy part (b)?

9.36 A centrifugal fan operating at 2400 rpm delivers 20,000 cfm of air through a 32 in. diameter duct against a static pressure of 4.8

in. of water. The air is 40°F. The barometer is 29.0 in. Hg. Determine the horsepower input if the efficiency is 70%. If the fan size, gas density, and duct system remain the same, calculate the horsepower required if operated at 3200 rpm. [Ans: 60 hp]

9.37 Compute the efficiency of Fan 303 (Fig. 9-11b) when delivering 15,500 cfm at 4 in. static pressure (SP).

9.38 Develop and explain the following relations for fan performance:
(a) $HP = CFM \times \Delta P/6350\eta_f$
(b) $kWH = HP\ (0.746)\ hours/\eta_m$
(c) $\Delta t_f = \Delta P(0.371)/\eta_f$
(d) $HP \sim CFM^3$

9.39 A water pump develops a total head of 200 ft. The pump efficiency is 80% and the motor efficiency is 87.5%. If the power rate is 1.5 cents per kilowatt-hour, what is the power cost for pumping 1000 gallons? [Ans: 1.35 cents in 1 h]

9.40 For a certain system it is required to select a pump that will deliver 2400 gpm (150 L/s) at a total head of 360 ft (110 m), and a pump shaft speed of 2400 rpm. What type of pump would you suggest? [Ans: centrifugal]

9.41 A pump delivers 1400 gpm of water. The inlet pipe is 4 in. nominal and the outlet pipe is 2 in. nominal standard pipe. The water temperature is 40°F. The surface of the inlet supply is 40 ft higher than the pump centerline. The discharge gage, which is 22 ft above the pump centerline, reads 180 psi. If the pump and motor combined efficiency is 60%, calculate the necessary input to the motor in kilowatts. [Ans: 290 kW]

9.42 A pump is required to force 9250 lb/h (4200 kg/h) of water at 165°F (74°C) through a heating system against a total resistance of 82,300 milli-inches of water (20.5 kPa). If the mechanical efficiency of the pump is 65%, find the required horsepower input.

9.43 How many horsepower are required to pump 66 gpm (4.16 L/s) against 60 ft (18.3 m) of head assuming 75% efficiency?

9.44 Solve the following problems:

(a) A certain system is found to have losses due to frictional effects according to the equation $H = 0.001\ (\text{gpm})^2$ where H is in ft of water. The system is handling water at 160°F. For a design capacity of 300 gpm, what is the head developed by the pump and the bhp if the pump efficiency is 80%?

(b) What would be the theoretical maximum length of suction in order to prevent cavitation if the level of the supply tank is below the centerline of the pump? Assume atmospheric pressure to be 14.7 psi.

(c) If a capacity of 400 gpm is desired, what would be the speed ratio n_2/n_1 for the same pump, density of fluid, and system?

(d) Should a backward- or forward-curved blade pump be chosen? Would you make arrangements for a priming system for the pump? [Ans: 90 ft, 8.4 hp, 23.6 ft, 1.33]

9.7 References

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SI Figures and Tables

Table 9-1B SI Equivalent Flat Oval Duct Dimensions

(Table 4, Chapter 21, 2017 ASHRAE Handbook—Fundamentals, SI version)

Circular Duct Diameter, mm	Minor Axis *a*, mm																
	70	**100**	**125**	**150**	**175**	**200**	**250**	**275**	**300**	**325**	**350**	**375**	**400**	**450**	**500**	**550**	**600**
	Major Axis *A*, mm																
125	205																
140	265	180															
160	360	235	190														
180	475	300	235	200													
200		380	290	245	215												
224		490	375	305	—	240											
250			475	385	325	290											
280				485	410	360	—	285									
315				635	525	—	—	345	325								
355				840	—	580	460	425	395	375							
400				1115	—	760	—	530	490	460	435						
450				1490	—	995	—	675	—	570	535	505					
500						1275	—	845	—	700	655	615	580				
560						1680	—	1085	—	890	820	765	720				
630								1425	—	1150	1050	970	905	810			
710										1505	1370	1260	1165	1025			
800											1800	1645	1515	1315	1170	1065	
900												2165	1985	1705	1500	1350	
1000														2170	1895	1690	
1120															2455	2170	1950
1250																2795	2495

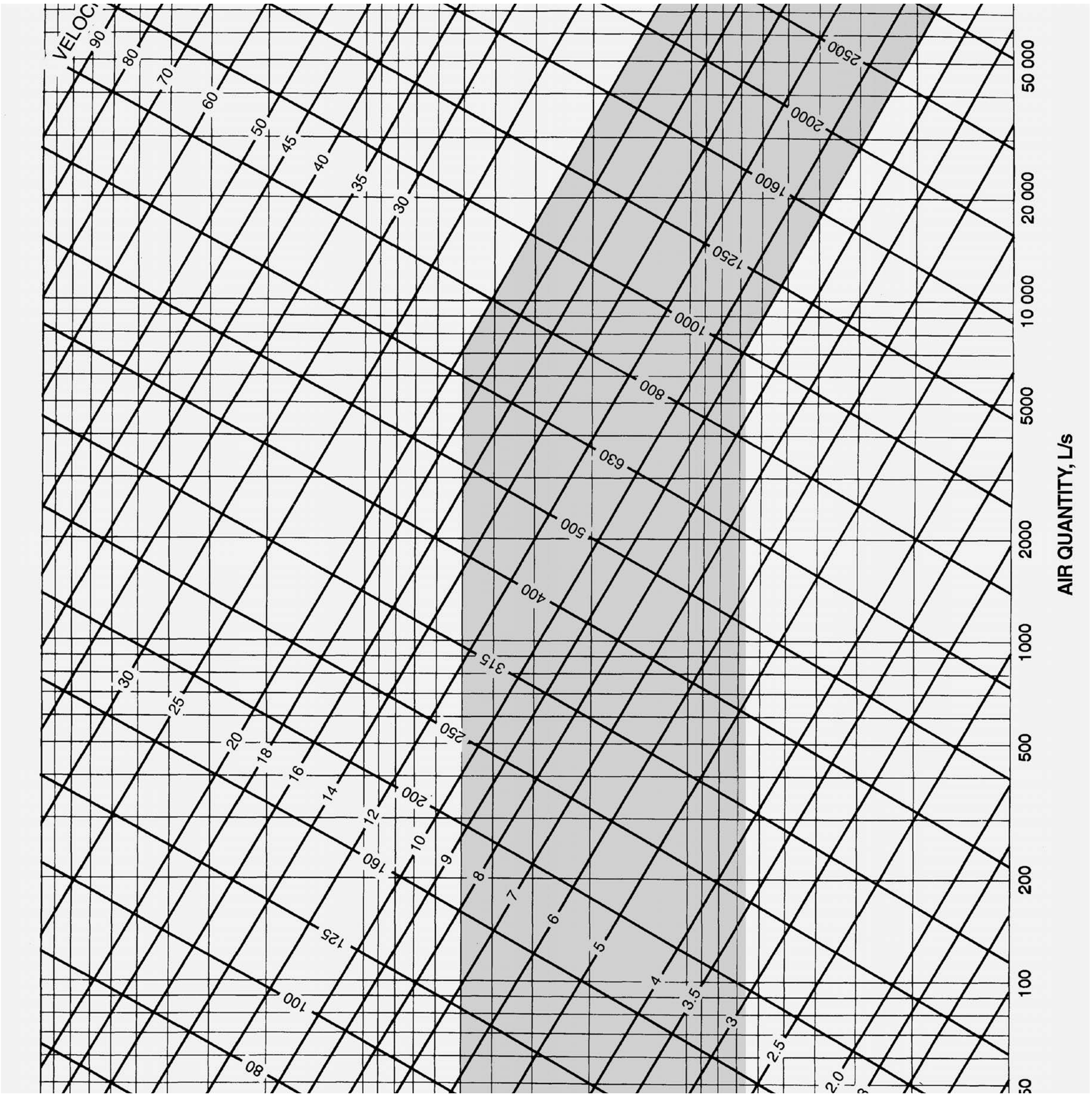

Fig. 9-2 SI Friction of Air in Straight Ducts

Shaded Area is Normal Design Region.

(Figure 10, Chapter 21, 2017 ASHRAE Handbook—Fundamentals [SI])

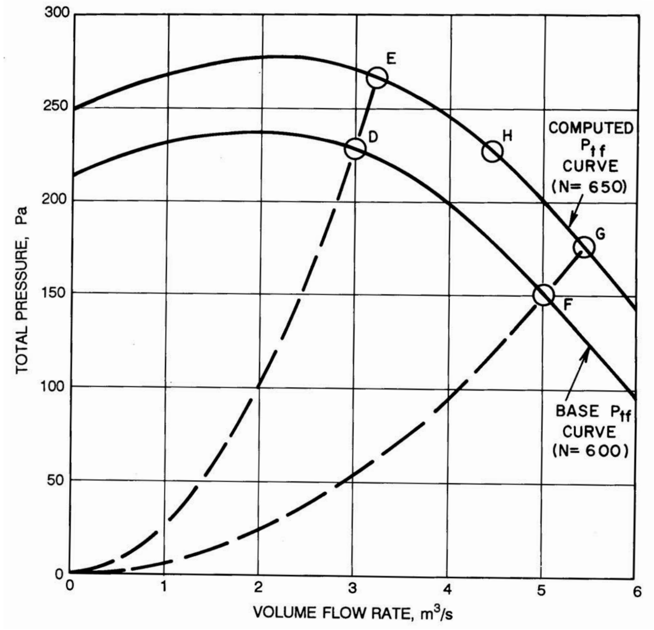

Fig. 9-6 SI Application of fan laws
(Chapter 21, 2016 ASHRAE Handbook—HVAC Systems and Equipment, SI version)

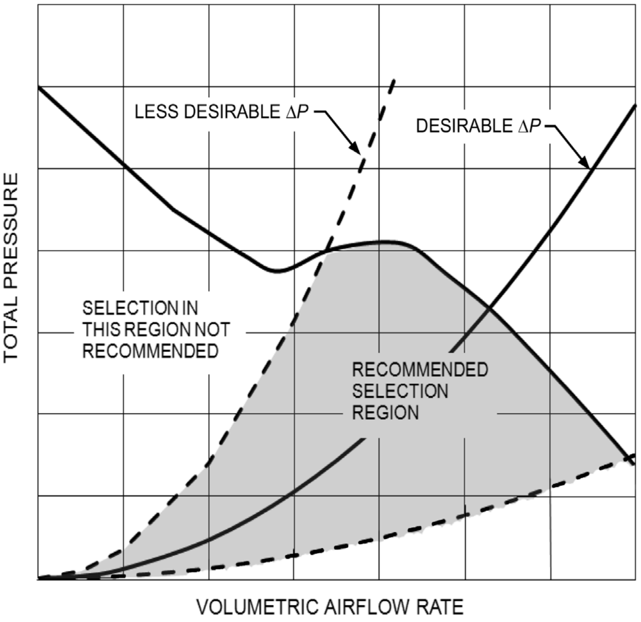

Fig. 9-12 SI Desirable Combination of p_{tf} *and* Δp *Curves*
(Chapter 21, 2016 ASHRAE Handbook—HVAC Systems and Equipment, SI version)

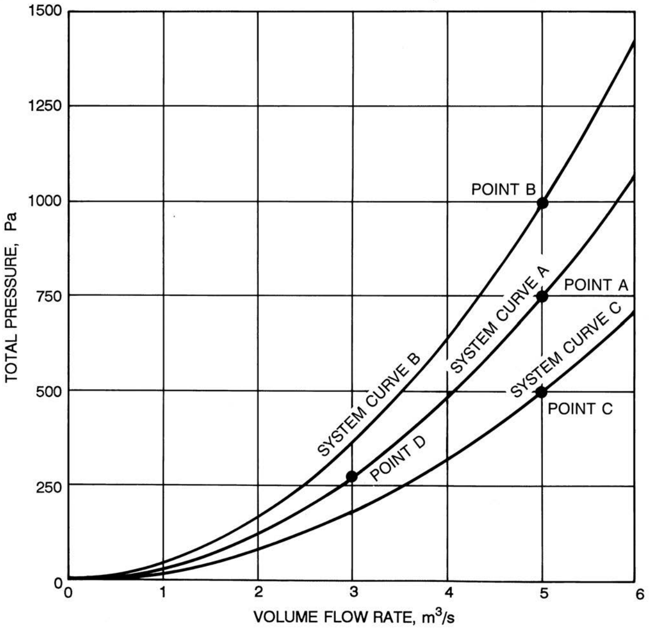

Fig. 9-8 SI System Total Pressure Loss (Δp) Curves
(Chapter 21, 2016 ASHRAE Handbook—HVAC Systems and Equipment, SI version)

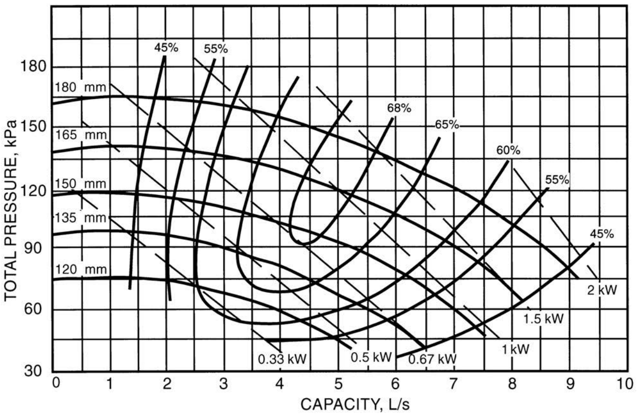

Fig. 9-13 SI Pump Performance Curves
(Chapter 44, *2016 ASHRAE Handbook—HVAC Systems and Equipment, SI version*)

Table 9-8 SI Steel Pipe Data

(Chapter 46, 2016 ASHRAE Handbook—HVAC Systems and Equipment, SI version)

					Surface Area		Cross Section		Mass		Working Pressure[c] ASTM A53 B to 200°C		
U.S. Nominal Size, in.	Nominal Size, mm	Schedule[a]	Wall Thickness *t*, mm	Inside Diameter *d*, mm	Outside, m^2/m	Inside, m^2/m	Metal Area, mm^2	Flow Area, mm^2	Pipe, kg/m	Water, kg/m	Mfr. Process	Joint Type[b]	kPa (gage)
1/4	8	40 ST	2.24	9.25	0.043	0.029	80.6	67.1	0.631	0.067	CW	T	1296
		80 XS	3.02	7.67	0.043	0.024	101.5	46.2	0.796	0.046	CW	T	6006
3/8	10	40 ST	2.31	12.52	0.054	0.039	107.7	123.2	0.844	0.123	CW	T	1400
		80 XS	3.20	10.74	0.054	0.034	140.2	90.7	1.098	0.091	CW	T	5654
1/2	15	40 ST	2.77	15.80	0.067	0.050	161.5	196.0	1.265	0.196	CW	T	1476
		80 XS	3.73	13.87	0.067	0.044	206.5	151.1	1.618	0.151	CW	T	5192
3/4	20	40 ST	2.87	20.93	0.084	0.066	214.6	344.0	1.68	0.344	CW	T	1496
		80 XS	3.91	18.85	0.084	0.059	279.7	279.0	2.19	0.279	CW	T	4695
1	25	40 ST	3.38	26.64	0.105	0.084	318.6	557.6	2.50	0.558	CW	T	1558
		80 XS	4.55	24.31	0.105	0.076	412.1	464.1	3.23	0.464	CW	T	4427
1 1/4	32	40 ST	3.56	35.05	0.132	0.110	431.3	965.0	3.38	0.965	CW	T	1579
		80 XS	4.85	32.46	0.132	0.102	568.7	827.6	4.45	0.828	CW	T	4096
1 1/2	40	40 ST	3.68	40.89	0.152	0.128	515.5	1 313	4.05	1.313	CW	T	1593
		80 XS	5.08	38.10	0.152	0.120	689.0	1 140	5.40	1.140	CW	T	3972
2	50	40 ST	3.91	52.50	0.190	0.165	690.3	2 165	5.43	2.165	CW	T	1586
		80 XS	5.54	49.25	0.190	0.155	953	1 905	7.47	1.905	CW	T	3799
2 1/2	65	40 ST	5.16	62.71	0.229	0.197	1 099	3 089	8.62	3.089	CW	W	3675
		80 XS	7.01	59.00	0.229	0.185	1 454	2 734	11.40	2.734	CW	W	5757
3	80	40 ST	5.49	77.93	0.279	0.245	1 438	4 769	11.27	4.769	CW	W	3323
		80 XS	7.62	73.66	0.279	0.231	1 946	4 261	15.25	4.261	CW	W	5288
4	100	40 ST	6.02	102.26	0.359	0.321	2 048	8 213	16.04	8.213	CW	W	2965
		80 XS	8.56	97.18	0.359	0.305	2 844	7 417	22.28	7.417	CW	W	4792
6	150	40 ST	7.11	154.05	0.529	0.484	3 601	18 639	28.22	18.64	ERW	W	4799
		80 XS	10.97	146.33	0.529	0.460	5 423	16 817	42.49	16.82	ERW	W	8336
8	200	30	7.04	205.0	0.688	0.644	4 687	33 000	36.73	33.01	ERW	W	3627
		40 ST	8.18	202.7	0.688	0.637	5 419	32 280	42.46	32.28	ERW	W	4433
		80 XS	12.70	193.7	0.688	0.608	8 234	29 460	64.51	29.46	ERW	W	7626
10	250	30	7.80	257.5	0.858	0.809	6 498	52 060	50.91	52.06	ERW	W	3344
		40 ST	9.27	254.5	0.858	0.800	7 683	50 870	60.20	50.87	ERW	W	4178
		XS	12.70	247.7	0.858	0.778	10 388	48 170	81.39	48.17	ERW	W	6116
		80	15.06	242.9	0.858	0.763	12 208	46 350	95.66	46.35	ERW	W	7453
12	300	30	8.38	307.1	1.017	0.965	8 307	74 060	65.09	74.06	ERW	W	3096
		ST	9.53	304.8	1.017	0.958	9 406	72 970	73.70	72.97	ERW	W	3641
		40	10.31	303.2	1.017	0.953	10 158	72 190	79.59	72.21	ERW	W	4020
		XS	12.70	298.5	1.017	0.938	12 414	69 940	97.28	69.96	ERW	W	5157
		80	17.45	289.0	1.017	0.908	16 797	65 550	131.62	65.57	ERW	W	7419
14	350	30 ST	9.53	336.6	1.117	1.057	10 356	88 970	81.15	88.96	ERW	W	3316
		40	11.10	333.4	1.117	1.047	12 013	87 290	94.13	87.30	ERW	W	3999
		XS	12.70	330.2	1.117	1.037	13 681	85 610	107.21	85.63	ERW	W	4695
		80	19.05	317.5	1.117	0.997	20 142	79 160	157.82	79.17	ERW	W	7453
16	400	30 ST	9.53	387.4	1.277	1.217	11 876	117 800	93.06	117.8	ERW	W	2903
		40 XS	12.70	381.0	1.277	1.197	15 708	114 000	123.09	114.0	ERW	W	4109
18	450	ST	9.53	438.2	1.436	1.376	13 396	150 800	104.98	150.8	ERW	W	2579
		30	11.10	435.0	1.436	1.367	15 556	148 600	121.90	148.6	ERW	W	3110
		XS	12.70	431.8	1.436	1.357	17 735	146 450	138.97	146.4	ERW	W	3654
		40	14.27	428.7	1.436	1.347	19 863	144 300	155.65	144.3	ERW	W	4185
20	500	20 ST	9.53	489.0	1.596	1.536	14 916	187 700	116.88	187.4	ERW	W	2324
		30 XS	12.70	482.6	1.596	1.516	19 762	182 900	154.85	182.9	ERW	W	3289
		40	15.06	477.9	1.596	1.501	23 325	179 400	182.78	179.4	ERW	W	4006

[a] Numbers are schedule numbers per ASME *Standard* B36.10M; ST = Standard; XS = Extra Strong.

[b] T = Thread; W = Weld

[c] Working pressures were calculated per ASME B31.9 using furnace butt-weld (continuous weld, CW) pipe through 100 mm and electric resistance weld (ERW) thereafter. The allowance A has been taken as

(1) 12.5% of *t* for mill tolerance on pipe wall thickness, *plus*

(2) An arbitrary corrosion allowance of 0.64 mm for pipe sizes through NPS 2 and 1.65 mm from NPS 2 1/2 through 20, *plus*

(3) A thread cutting allowance for sizes through NPS 2.

Because the pipe wall thickness of threaded standard pipe is so small after deducting allowance A, the mechanical strength of the pipe is impaired. It is good practice to limit standard threaded pipe pressure to 620 kPa (gage) for steam and 860 kPa (gage) for water.

Chapter 10

ECONOMIC ANALYSES AND LIFE-CYCLE COSTS

This chapter presents the fundamentals for doing a simple engineering economic analysis for heating and air-conditioning systems and refrigeration installations. This information is reproduced from Chapter 37 of the 2015 *ASHRAE Handbook—HVAC Applications*.

10.1 Introduction

Owning and operating cost information for the HVAC system should be part of the investment plan of a facility. This information can be used for preparing annual budgets, managing assets, and selecting design options. Table 10-1 shows a representative form that summarizes these costs.

A properly engineered system must also be economical, but this is difficult to assess because of the complexities surrounding effective money management and the inherent difficulty of predicting future operating and maintenance expenses. Complex tax structures and the time value of money can affect the final engineering decision. This does not imply use of either the cheapest or the most expensive system; instead, it demands intelligent analysis of financial objectives and the owner's requirements.

Certain tangible and intangible costs or benefits must also be considered when assessing owning and operating costs. Local codes may require highly skilled or certified operators for specific types of equipment. This could be a significant cost over the life of the system. Similarly, intangible items such as aesthetics, acoustics, comfort, safety, security, flexibility, and environmental impact may vary by location and be important to a particular building or facility.

10.2 Owning Costs

The following elements must be established to calculate annual owning costs: (1) initial cost, (2) analysis or study period, (3) interest or discount rate, and (4) other periodic costs such as insurance, property taxes, refurbishment, or disposal fees. Once established, these elements are coupled with operating costs to develop an economic analysis, which may be a simple payback evaluation or an in-depth analysis such as outlined in the section on Economic Analysis Techniques.

10.2.1 Initial Cost

Major decisions affecting annual owning and operating costs for the life of the building must generally be made before completing contract drawings and specifications. To achieve the best performance and economics, alternative methods of solving the engineering problems peculiar to each project should be compared in the early stages of design. Oversimplified estimates can lead to substantial errors in evaluating the system.

The evaluation should lead to a thorough understanding of installation costs and accessory requirements for the system(s) under consideration. Detailed lists of materials, controls, space and structural requirements, services, installation labor, and so forth can be prepared to increase accuracy in preliminary cost estimates. A reasonable estimate of capital cost of components may be derived from cost records of recent installations of comparable design or from quotations submitted by manufacturers and contractors, or by consulting commercially available cost-estimating guides and software. Table 10-2 shows a representative checklist for initial costs.

10.2.2 Analysis Period

The time frame over which an economic analysis is performed greatly affects the results. The analysis period is usually determined by specific objectives, such as length of planned ownership or loan repayment period. However, as the length of time in the analysis period increases, there is a diminishing effect on net present-value calculations. The chosen analysis period is often unrelated to the equipment depreciation period or service life, although these factors may be important in the analysis.

10.3 Service Life

For many years, the Owning and Operating Costs chapter of *ASHRAE Handbook—HVAC Applications* included estimates of service lives for various HVAC system components, based on a survey conducted in 1976 under ASHRAE research project RP-186 (Akalin 1978). These estimates have been useful to a generation of practitioners, but changes in technology, materials, manufacturing techniques, and maintenance practices now call into question the continued validity of the original estimates. Consequently, ASHRAE research project TRP-1237 developed an Internet-based data collection tool

Table 10-1 Owning and Operating Cost Data and Summary

OWNING COSTS

I. Initial Cost of System _______

II. Periodic Costs
  A. Income taxes _______
  B. Property taxes _______
  C. Insurance _______
  D. Rent _______
  E. Other periodic costs _______
    Total Periodic Costs _______

III. Replacement Cost _______

IV. Salvage Value _______
    Total Owning Costs _______

OPERATING COSTS

V. Annual Utility, Fuel, Water, etc., Costs
  A. Utilities
    1. Electricity _______
    2. Natural gas _______
    3. Water/sewer _______
    4. Purchased steam _______
    5. Purchased hot/chilled water _______
  B. Fuels
    1. Propane _______
    2. Fuel oil _______
    3. Diesel _______
    4. Coal _______
  C. On-site generation of electricity _______
  D. Other utility, fuel, water, etc., costs _______
  Total _______

VI. Annual Maintenance Allowances/Costs
  A. In-house labor _______
  B. Contracted maintenance service _______
  C. In-house materials _______
  D. Other maintenance allowances/costs _______
    (e.g., water treatment)
  Total _______

VII. Annual Administration Costs _______
    Total Annual Operating Costs _______

TOTAL ANNUAL OWNING AND OPERATING COSTS _______

and database on HVAC equipment service life and maintenance costs, to allow equipment owning and operating cost data to be continually updated and current. The database was seeded with information gathered from a sample of 163 commercial office buildings located in major metropolitan areas across the United States. Abramson et al. (2005) provide details on the distribution of building size, age, and other characteristics. Table 10-3 presents estimates of median service life for various HVAC components in this sample.

Table 10-2 Initial Cost Checklist

Energy and Fuel Service Costs

Fuel service, storage, handling, piping, and distribution costs
Electrical service entrance and distribution equipment costs
Total energy plant

Heat-Producing Equipment

Boilers and furnaces
Steam-water converters
Heat pumps or resistance heaters
Makeup air heaters
Heat-producing equipment auxiliaries

Refrigeration Equipment

Compressors, chillers, or absorption units
Cooling towers, condensers, well water supplies
Refrigeration equipment auxiliaries

Heat Distribution Equipment

Pumps, reducing valves, piping, piping insulation, etc.
Terminal units or devices

Cooling Distribution Equipment

Pumps, piping, piping insulation, condensate drains, etc.
Terminal units, mixing boxes, diffusers, grilles, etc.

Air Treatment and Distribution Equipment

Air heaters, humidifiers, dehumidifiers, filters, etc.
Fans, ducts, duct insulation, dampers, etc.
Exhaust and return systems
Heat recovery systems

System and Controls Automation

Terminal or zone controls
System program control
Alarms and indicator system
Energy management system

Building Construction and Alteration

Mechanical and electric space
Chimneys and flues
Building insulation
Solar radiation controls
Acoustical and vibration treatment
Distribution shafts, machinery foundations, furring

Table 10-3 Median Service Life

Equipment Type	Median Service Life, Years	Total No. of Units	No. of Units Replaced
DX air distribution equipment	>24	1907	284
Chillers, centrifugal	>25	234	34
Cooling towers, metal	>22	170	24
Boilers, hot-water, steel gas-fired	>22	117	24
Controls, pneumatic	>18	101	25
electronic	>7	68	6
Potable hot-water heaters, electric	>21	304	36

Median service life in Table 10-3 is based on analysis of survival curves, which take into account the units still in service and the units replaced at each age (Hiller 2000). Conditional and total survival rates are calculated for each age, and the percent survival over time is plotted. Units still in service are included up to the point where the age is equal to their current age at the time of the study. After that point, these units are censored (removed from the population). Median service life in this table indicates the highest age at which the survival rate remains at or above 50% while the sample size is 30 or more.

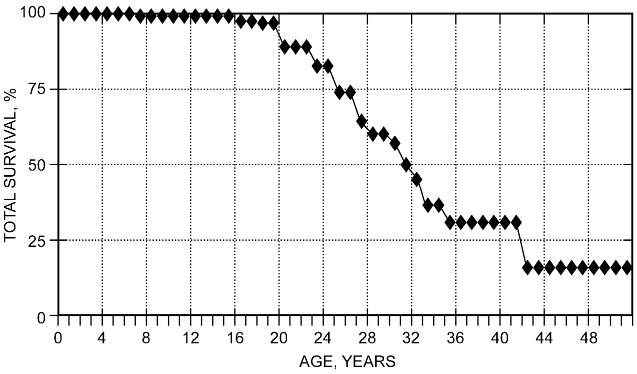

Fig. 10-1 Survival Curve for Centrifugal Chillers

[Based on data in Abramson et al. (2005)]

There is no hard-and-fast rule about the number of units needed in a sample before it is considered statistically large enough to be representative, but usually the number should be larger than 25 to 30 (Lovvorn and Hiller 2002). This rule-of-thumb is used because each unit removal represents greater than a 3% change in survival rate as the sample size drops below 30, and that percentage increases rapidly as the sample size gets even smaller.

The database initially developed and seeded under research project TRP-1237 (Abramson et al. 2005) is now available online, providing engineers with equipment service life and annual maintenance costs for a variety of building types and HVAC systems. The database can be accessed at www.ashrae.org/database.

As of the end of 2009 this database contained more than 300 building types, with service life data on more than 38,000 pieces of equipment.

The database allows users to access up-to-date information to determine a range of statistical values for equipment owning and operating costs. Users are encouraged to contribute their own service life and maintenance cost data, further expanding the utility of this tool. Over time, this input will provide sufficient service life and maintenance cost data to allow comparative analysis of many different HVAC systems types in a broad variety of applications. Data can be entered by logging into the database and registering, which is free. With this, ASHRAE is providing the necessary methods and information to assist in using life-cycle analysis techniques to help select the most appropriate HVAC system for a specific application. This system of collecting data also greatly reduces the time between data collection and when users can access the information.

Figure 10-1 presents the survival curve for centrifugal chillers, based on data in Abramson et al. (2005). The point at which survival rate drops to 50% based on all data in the survey is 31 years. However, because the sample size drops below the statistically relevant number of 30 units at 25 years, the median service life of centrifugal chillers can only be stated with confidence as >25 years.

Table 10-4 compares the estimates of median service life in Abramson et al. (2005) with those developed with those in Akalin (1978). Most differences are on the order of one to five years.

Estimated service life of new equipment or components of systems not listed in Table 10-3 or 10-4 may be obtained from manufacturers, associations, consortia, or governmental agencies. Because of the proprietary nature of information from some of these sources, the variety of criteria used in compiling the data, and the diverse objectives in disseminating them, extreme care is necessary in comparing service life from different sources. Designs, materials, and components of equipment listed in Tables 10-3 and 10-4 have changed over time and may have altered the estimated service lives of those equipment categories. Therefore, establishing equivalent comparisons of service life is important.

As noted, service life is a function of the time when equipment is replaced. Replacement may be for any reason, including, but not limited to, failure, general obsolescence, reduced reliability, excessive maintenance cost, and changed system requirements (e.g., building characteristics, energy prices, environmental considerations). Service lives shown in the tables are based on the age of the equipment when it was replaced, regardless of the reason it was replaced.

Locations in potentially corrosive environments and unique maintenance variables affect service life. Examples include the following:

- **Coastal and marine environments**, especially in tropical locations, are characterized by abundant sodium chloride (salt) that is carried by sea spray, mist, or fog.
- Many owners require equipment specifications stating that HVAC equipment located along coastal waters will have corrosion-resistant materials or coatings. Design criteria for systems installed under these conditions should be carefully considered.
- **Industrial** applications provide many challenges to the HVAC designer. It is very important to know if emissions from the industrial plant contain products of combustion from coal, fuel oils, or releases of sulfur oxides (SO_2, SO_3) and nitrogen oxides (NO_x) into the atmosphere. These gases typically accumulate and return to the ground in the form of acid rain or dew.

  Not only is it important to know the products being emitted from the industrial plant being designed, but also the adjacent upwind or downwind facilities. HVAC system design for a plant located downwind from a paper mill requires extraordinary corrosion protection or recognition of a reduced service life of the HVAC equipment.
- **Urban** areas generally have high levels of automotive emissions as well as abundant combustion by-products. Both of these contain elevated sulfur oxide and nitrogen oxide concentrations.
- **Maintenance** factors also affect life expectancy. The HVAC designer should temper the service life expectancy of equipment with a maintenance factor. To achieve the estimated service life values in Table 10-3, HVAC equipment must be maintained properly, including good filter-changing practices

Table 10-4 Comparison of Service Life Estimates

	Median Service Life, Years			Median Service Life, Years			Median Service Life, Years	
Equipment Item	**Abramson et al. (2005)**	**Akalin (1978)**	**Equipment Item**	**Abramson et al. (2005)**	**Akalin (1978)**	**Equipment Item**	**Abramson et al. (2005)**	**Akalin (1978)**
Air Conditioners			**Air Terminals**			**Condensers**		
Window unit	N/A*	10	Diffusers, grilles, and registers	N/A*	27	Air-cooled	N/A	20
Residential single or split package	N/A*	15	Induction and fan-coil units	N/A*	20	Evaporative	N/A*	20
Commercial through-the-wall	N/A*	15	VAV and double-duct boxes	N/A*	20	**Insulation**		
Water-cooled package	>24	15	**Air washers**	N/A*	17	Molded	N/A*	20
Heat pumps			**Ductwork**	N/A*	30	Blanket	N/A*	24
Residential air-to-air	N/A*	15[b]	**Dampers**	N/A*	20	**Pumps**		
Commercial air-to-air	N/A*	15	**Fans**	N/A*		Base-mounted	N/A*	20
Commercial water-to-air	>24	19	Centrifugal	N/A*	25	Pipe-mounted	N/A*	10
Roof-top air conditioners			Axial	N/A*	20	Sump and well	N/A*	10
Single-zone	N/A*	15	Propeller	N/A*	15	Condensate	N/A*	15
Multizone	N/A*	15	Ventilating roof-mounted	N/A*	20	**Reciprocating engines**	N/A*	20
Boilers, Hot-Water (Steam)			**Coils**			**Steam turbines**	N/A*	30
Steel water-tube	>22	24 (30)	DX, water, or steam	N/A*	20	**Electric motors**	N/A*	18
Steel fire-tube		25 (25)	Electric	N/A*	15	**Motor starters**	N/A*	17
Cast iron	N/A*	35 (30)	**Heat Exchangers**			**Electric transformers**	N/A*	30
Electric	N/A*	15	Shell-and-tube	N/A*	24	**Controls**		
Burners	N/A*	21	**Reciprocating compressors**	N/A*	20	Pneumatic	N/A*	20
Furnaces			**Packaged Chillers**			Electric	N/A*	16
Gas- or oil-fired	N/A*	18	Reciprocating	N/A*	20	Electronic	N/A*	15
Unit heaters			Centrifugal	>25	23	**Valve actuators**		
Gas or electric	N/A*	13	Absorption	N/A*	23	Hydraulic	N/A*	15
Hot-water or steam	N/A*	20	**Cooling Towers**			Pneumatic	N/A*	20
Radiant heaters			Galvanized metal	>22	20	Self-contained		10
Electric	N/A*	10	Wood	N/A*	20			
Hot-water or steam	N/A*	25	Ceramic	N/A*	34			

*N/A: Not enough data yet in Abramson et al. (2005). Note that data from Akalin (1978) for these categories may be outdated and not statistically relevant. Use these data with caution until enough updated data are accumulated in Abramson et al.

and good maintenance procedures. For example, chilled-water coils with more than four rows and close fin spacing are virtually impossible to clean even using extraordinary methods; they are often replaced with multiple coils in series, with a maximum of four rows and lighter fin spacing.

10.4 Depreciation

Depreciation periods are usually set by federal, state, or local tax laws, which change periodically. Applicable tax laws should be consulted for information on depreciation.

10.5 Interest or Discount Rate

Most major economic analyses consider the opportunity cost of borrowing money, inflation, and the time value of money. **Opportunity cost** of money reflects the earnings that investing (or lending) the money can produce. **Inflation** (price escalation) decreases the purchasing or investing power (value) of future money because it can buy less in the future. **Time value** of money reflects the fact that money received today is more useful than the same amount received a year from now, even with zero inflation, because the money is available earlier for reinvestment.

The cost or value of money must also be considered. When borrowing money, a percentage fee or interest rate must normally be paid. However, the interest rate may not necessarily be the correct cost of money to use in an economic analysis. Another factor, called the **discount rate**, is more commonly used to reflect the true cost of money [see Fuller and Petersen (1996) for detailed discussions]. Discount rates used for analyses vary depending on individual investment, profit, and other opportunities. Interest rates, in contrast, tend to be more centrally fixed by lending institutions.

To minimize the confusion caused by the vague definition and variable nature of discount rates, the U.S. government has specified particular discount rates to be used in economic analyses relating to federal expenditures. These discount rates are updated annually (Rushing et al. 2010) but may not be appropriate for private-sector economic analyses.

10.6 Periodic Costs

Regularly or periodically recurring costs include insurance, property taxes, income taxes, rent, refurbishment expenses, disposal fees (e.g., refrigerant recycling costs), occasional major repair costs, and decommissioning expenses.

Insurance. Insurance reimburses a property owner for a financial loss so that equipment can be repaired or replaced. Insurance often indemnifies the owner from liability, as well.

Financial recovery may include replacing income, rents, or profits lost because of property damage or machinery failure.

Some of the principal factors that influence the total annual insurance premium are building size, construction materials, amount and size of mechanical equipment, geographic location, and policy deductibles. Some regulations set minimum required insurance coverage and premiums that may be charged for various forms of insurable property.

Property Taxes. Property taxes differ widely and may be collected by one or more agencies, such as state, county, or local governments or special assessment districts. Furthermore, property taxes may apply to both real (land, buildings) and personal (everything else) property. Property taxes are most often calculated as a percentage of assessed value, but are also determined in other ways, such as fixed fees, license fees, registration fees, etc. Moreover, definitions of assessed value vary widely in different geographic areas. Tax experts should be consulted for applicable practices in a given area.

Income Taxes. Taxes are generally imposed in proportion to net income, after allowance for expenses, depreciation, and numerous other factors. Special tax treatment is often granted to encourage certain investments. Income tax professionals can provide up-to-date information on income tax treatments.

Other Periodic Costs. Examples of other costs include changes in regulations that require unscheduled equipment refurbishment to eliminate use of hazardous substances, and disposal costs for such substances.

Replacement Costs and Salvage Value. Replacement costs and salvage value should be evaluated when calculating owning cost. Replacement cost is the cost to remove existing equipment and install new equipment. Salvage value is the value of equipment or its components for recycling or other uses. Equipment's salvage value may be negative when removal, disposal, or decommissioning costs are considered.

10.7 Operating Costs

Operating costs are those incurred by the actual operation of the system. They include costs of fuel and electricity, wages, supplies, water, material, and maintenance parts and services. Energy is a large part of total operating costs. Chapter 19 of the 2017 *ASHRAE Handbook—Fundamentals* outlines how fuel and electrical requirements are estimated. Because most energy management activities are dictated by economics, the design engineer must understand the utility rates that apply. Electric rates are usually more complex than gas or water rates. In addition to general commercial or institutional electric rates, special rates may exist such as time of day, interruptible service, on-peak/off-peak, summer/winter, and peak demand. Electric rate schedules vary widely in North America. The design engineer should work with local utility companies to identify the most favorable rates and to understand how to qualify for them.

10.7.1 Electrical Energy

The total cost of electricity is determined by a rate schedule and is usually a combination of several components: consumption (kilowatt-hours), demand (kilowatts) fuel adjustment charges, special allowances or other adjustments, and applicable taxes. Of these, consumption and demand are the major cost components and the ones the owner or facility manager may be able to affect.

Electricity Consumption Charges. Most electric rates have step-rate schedules for consumption, and the cost of the last unit consumed may be substantially different from that of the first. The last unit is usually cheaper than the first because the fixed costs to the utility may already have been recovered from earlier consumption costs. Because of this, the energy analysis cannot use average costs to accurately predict savings from implementation of energy conservation measures. Average costs will overstate the savings possible between alternative equipment or systems; instead, marginal (or incremental) costs must be used.

To reflect time-varying operating costs or to encourage peak shifting, electric utilities may charge different rates for consumption according to the time of use and season, with higher costs occurring during the peak period of use.

Fuel Adjustment Charge. Because of substantial variations in fuel prices, electric utilities may apply a fuel adjustment charge to recover costs. This adjustment may not be reflected in the rate schedule. The fuel adjustment is usually a charge per unit of consumption and may be positive or negative, depending on how much of the actual fuel cost is recovered in the energy consumption rate. The charge may vary monthly or seasonally.

Allowances or Adjustments. Special discounts or rates may be available for customers who can receive power at higher voltages or for those who own transformers or similar equipment. Special rates or riders may be available for specific interruptible loads such as domestic water heaters.

Certain facility electrical systems may produce a low power factor [i.e., ratio of real (active) kilowatt power to apparent (reactive) kVA power], which means that the utility must supply more current on an intermittent basis, thus increasing their costs. These costs may be passed on as an adjustment to the utility bill if the power factor is below a level established by the utility.

When calculating power bills, utilities should be asked to provide detailed cost estimates for various consumption levels. The final calculation should include any applicable special rates, allowances, taxes, and fuel adjustment charges.

Demand Charges. Electric rates may also have demand charges based on the customer's peak kilowatt demand. Whereas consumption charges typically cover the utility's operating costs, demand charges typically cover the owning costs.

Demand charges may be formulated in a variety of ways:

- *Straight charge.* Cost per kilowatt per month, charged for the peak demand of the month.

Table 10-5 Electricity Data Consumption and Demand for Atlanta Example Building, 2003 to 2004

	Billing Days	Consumption, kWh	Actual Demand, kW	Billing Demand, kW	Total Cost, US$
Jan. 2003	29	57,120	178	185	4,118
Feb. 2003	31	61,920	145	185	4,251
Mar. 2003	29	60,060	140	185	4,199
Apr. 2003	29	62,640	154	185	4,271
May. 2003	33	73,440	161	185	4,569
Jun. 2003	26	53,100	171	185	4,007
Jul. 2003	32	67,320	180	185	4,400
Aug. 2003	30	66,000	170	185	4,364
Sep. 2003	32	63,960	149	171	4,127
Oct. 2003	30	55,260	122	171	3,865
Nov. 2003	27	46,020	140	171	3,613
Dec. 2003	34	61,260	141	171	4,028
Total 2003	362	670,980			49,812
Jan. 2004	31	59,040	145	171	3,967
Feb. 2004	29	54,240	159	171	3,837
Mar. 2004	20	37,080	122	171	2,584
Apr. 2004	12	22,140	133	171	1,547
May. 2004	34	64,260	148	171	4,110
Jun. 2004	29	63,720	148	171	4,321
Jul. 2004	30	69,120	169	169	4,458
Aug. 2004	32	73,800	170	170	4,605
Sep. 2004	29	64,500	166	166	4,281
Oct. 2004	30	60,060	152	161	3,866
Nov. 2004	32	65,760	128	161	4,018
Dec. 2004	31	51,960	132	161	3,646
Total 2004	339	685,680			45,240

- *Excess charge*. Cost per kilowatt above a base demand, (e.g., 50 kW) which may be established each month.
- *Maximum demand (ratchet)*. Cost per kilowatt for maximum annual demand, which may be reset only once a year. This established demand may either benefit or penalize the owner.
- *Combination demand*. Cost per hour of operation of demand. In addition to a basic demand charge, utilities may include further demand charges as demand-related consumption charges.

The actual demand represents the peak energy use averaged over a specific period, usually 15, 30, or 60 min. Accordingly, high electrical loads of only a few minutes' duration may never be recorded at the full instantaneous value. Alternatively, peak demand is recorded as the average of several consecutive short periods (i.e., 5 min out of each hour).

The particular method of demand metering and billing is important when load shedding or shifting devices are considered. The portion of the total bill attributed to demand may vary greatly, from 0% to as high as 70%.

- *Real-time* or *time-of-day rates*. Cost of electricity at time of use. An increasing number of utilities offer these rates. End users who can shift operations or install electric load-shifting equipment, such as thermal storage, can take advantage of such rates. Because these rates usually reflect a utility's overall load profile and possibly the availability of specific generating resources, contact with the supplying utility is essential to determine whether these rates are a reasonable option for a specific application.

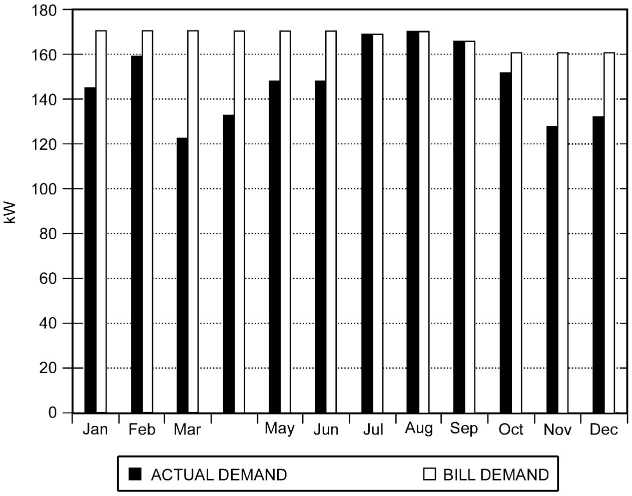

Fig. 10-2 Bill Demand and Actual Demand for Atlanta Example Building, 2004

Understanding Electric Rates. To illustrate a typical commercial electric rate with a ratchet, electricity consumption and demand data for an example building are presented in Table 10-5.

The example building in Table 10-5 is on a ratcheted rate, and bill demand is determined as a percentage of actual demand in the summer. How the ratchet operates is illustrated in Figure10-2.

Table 10-5 shows that the actual demand in the first six months of 2004 had no effect on the billing demand, and therefore no effect on the dollar amount of the bill. The same is true for the last three months of the year. Because of the ratchet, the billing demand in the first half of 2004 was set the previous summer. Likewise, billing demand for the last half of 2004 and first half of 2005 was set by the peak actual demand of 180 kW in July 2003. This tells the facility manager to pay attention to demand in the summer months (June to September) and that demand is not a factor in the winter (October to May) months for this particular rate. (Note that Atlanta's climate is hot and humid; in other climates, winter electric demand is an important determinant of costs.) Consumption must be monitored all year long.

Understanding the electric rates is key when evaluating the economics of energy conservation projects. Some projects save electrical demand but not consumption; others save mostly consumption but have little effect on demand. Electric rates must be correctly applied for economic analyses to be accurate. Chapter 56 in the 2015 *ASHRAE Handbook—Applications* contains a thorough discussion of various electric rates.

10.7.2 Natural Gas

Rates. Conventional natural gas rates are usually a combination of two main components: (1) utility rate or base

charges for gas consumption and (2) purchased gas adjustment (PGA) charges.

Although gas is usually metered by volume, it is often sold by energy content. The utility rate is the amount the local distribution company charges per unit of energy to deliver the gas to a particular location. This rate may be graduated in steps; the first 100 units of gas consumed may not be the same price as the last 100 units. The PGA is an adjustment for the cost of the gas per unit of energy to the local utility. It is similar to the electric fuel adjustment charge. The total cost per unit is then the sum of the appropriate utility rate and the PGA, plus taxes and other adjustments.

Interruptible Gas Rates and Contract/Transport Gas. Large industrial plants usually have the ability to burn alternative fuels and can qualify for special interruptible gas rates. During peak periods of severe cold weather, these customers' supply may be curtailed by the gas utility, and they may have to switch to propane, fuel oil, or some other back-up fuel. The utility rate and PGA are usually considerably cheaper for these interruptible customers than they are for firm-rate (noninterruptible) customers.

Deregulation of the natural gas industry allows end users to negotiate for gas supplies on the open market. The customer actually contracts with a gas producer or broker and pays for the gas at the source. Transport fees must be negotiated with the pipeline companies carrying the gas to the customer's local gas utility. This can be a very complicated administrative process and is usually economically feasible only for large gas users. Some local utilities have special rates for delivering contract gas volumes through their system; others simply charge a standard utility fee (PGA is not applied because the customer has already negotiated with the supplier for the cost of the fuel itself).

When calculating natural gas bills, be sure to determine which utility rate and PGA and/or contract gas price is appropriate for the particular interruptible or firm-rate customer. As with electric bills, the final calculation should include any taxes, prompt payment discounts, or other applicable adjustments.

10.7.3 Other Fossil Fuels

Propane, fuel oil, and diesel are examples of other fossil fuels in widespread use. Calculating the cost of these fuels is usually much simpler than calculating typical utility rates.

The cost of the fuel itself is usually a simple charge per unit volume or per unit mass. The customer is free to negotiate for the best price. However, trucking or delivery fees must also be included in final calculations. Some customers may have their own transport trucks, but most seek the best delivered price. If storage tanks are not customer-owned, rental fees must be considered. Periodic replacement of diesel-type fuels may be necessary because of storage or shelf-life limitations and must also be considered. The final fuel cost calculation should include any of these costs that are applicable, as well as appropriate taxes.

It is usually difficult, however, to relate usage of stored fossil fuels (e.g., fuel oil) with their operating costs. This is because propane or fuel oil is bought in bulk and stored until needed, and normally not metered or measured as it is consumed, whereas natural gas and electricity are utilities and are billed for as they are used.

Energy Source Choices. In planning for a new facility, the designer may undertake energy master planning. One component of energy master planning is choice of fuels. Typical necessary decisions include, for example, whether the building should be heated by electricity or natural gas, how service hot water should be produced, whether a hybrid heating plant (i.e., a combination of both electric and gas boilers) should be considered, and whether emergency generators should be fueled by diesel or natural gas.

Decision-makers should consider histories or forecasts of price volatility when selecting energy sources. In addition to national trending, local energy price trends from energy suppliers can be informative. These evaluations are particularly important where relative operating costs parity exists between various fuel options, or where selecting more efficient equipment may help mitigate utility price concerns.

Many sources of historic and projected energy costs are available for reference. In addition to federal projections, utility and energy supplier annual reports and accompanying financial data may provide insight into future energy costs. Indicators such as constrained or declining energy supply or production may be key factors in projecting future energy pricing trends. Pricing patterns that suggest unusual levels of energy price volatility should be carefully analyzed and tested at extreme predicted price levels to assess potential effects on system operating costs.

10.7.4 Water and Sewer Costs

Water and sewer costs should not be overlooked in economic analyses. Fortunately, these rates are usually very simple and straightforward: commonly, a charge per hundred cubic feet (CCF) for water and a different charge per CCF for sewer. Because water consumption is metered and sewage is not, most rates use the water consumption quantity to compute the sewer charge. If an owner uses water that is not returned to the sewer, there may be an opportunity to receive a credit or refund. Owners frequently use irrigation meters for watering grounds when the water authority has a special irrigation rate with no sewer charge. Another opportunity that is sometimes overlooked is to separately meter makeup water for cooling towers. This can be done with an irrigation meter if the costs of setting the meter can be justified; alternatively, it may be done by installing an in-line water meter for the cooling tower, in which case the owner reports the usage annually and applies for a credit or refund.

Because of rising costs of water and sewer, water recycling and reclamation is becoming more cost effective. For example, it may now be cost effective in some circumstances to capture cooling coil condensate and pump it to a cooling tower for makeup water.

Table 10-6 Comparison of Maintenance Costs Between Studies

Survey	Cost per ft^2, as Reported Mean	Median	Consumer Price Index	Cost per ft^2, 2004 Dollars Mean	Median
Dohrmann and Alereza (1983)	\$0.32	\$0.24	99.6	\$0.61	\$0.46
Abramson et al. (2005)	\$0.47	\$0.44	188.9	\$0.47	\$0.44

10.7.5 Maintenance Costs

The quality of maintenance and maintenance supervision can be a major factor in overall life-cycle cost of a mechanical system. The maintenance cost of mechanical systems varies widely depending upon configuration, equipment locations, accessibility, system complexity, service duty, geography, and system reliability requirements. Maintenance costs can be difficult to predict, because each system or facility is unique.

Dohrmann and Alereza (1986) obtained maintenance costs and HVAC system information from 342 buildings located in 35 states in the United States. In 1983 U.S. dollars, data collected showed a mean HVAC system maintenance cost of \$0.32/ft^2 per year, with a median cost of \$0.24/ft^2 per year. Building age has a statistically significant but minor effect on HVAC maintenance costs. Analysis also indicated that building size is not statistically significant in explaining cost variation. The type of maintenance program or service agency that building management chooses can also have a significant effect on total HVAC maintenance costs. Although extensive or thorough routine and preventive maintenance programs cost more to administer, they usually extend equipment life; improve reliability; and reduce system downtime, energy costs, and overall life-cycle costs.

Some maintenance cost data are available, both in the public domain and from proprietary sources used by various commercial service providers. These sources may include equipment manufacturers, independent service providers, insurers, government agencies (e.g., the U.S. General Services Administration), and industry-related organizations [e.g., the Building Owners and Managers Association (BOMA)] and service industry publications. More traditional, widely used products and components are likely to have statistically reliable records. However, design changes or modifications necessitated by industry changes, such as alternative refrigerants, may make historical data less relevant.

Newer HVAC products, components, system configurations, control systems and protocols, and upgraded or revised system applications present an additional challenge. Care is required when using data not drawn from broad experience or field reports. In many cases, maintenance information is proprietary or was sponsored by a particular entity or group. Particular care should be taken when using such data. It is the user's responsibility to obtain these data and to determine their appropriateness and suitability for the application being considered.

ASHRAE research project TRP-1237 (Abramson et al. 2005) developed a standardized Internet-based data collection tool and database on HVAC equipment service life and maintenance costs. The database was seeded with data on 163 buildings from around the country. Maintenance cost data were gathered for total HVAC system maintenance costs from 100 facilities. In 2004 dollars, the mean HVAC maintenance cost from these data was \$0.47/ft^2, and the median cost was \$0.44/ft^2. Table 10-6 compares these figures with estimates reported by Dohrmann and Alereza (1983), both in terms of contemporary dollars, and in 2004 dollars, and shows that the cost per square foot varies widely between studies.

Estimating Maintenance Costs. Total HVAC maintenance cost for new and existing buildings with various types of equipment may be estimated several ways, using several resources. Equipment maintenance requirements can be obtained from the equipment manufacturers for large or custom pieces of equipment. Estimating in-house labor requirements can be difficult; BOMA (2003) provides guidance on this topic. Many independent mechanical service companies provide preventative maintenance contracts. These firms typically have proprietary estimating programs developed through their experience, and often provide generalized maintenance costs to engineers and owners upon request, without obligation.

When evaluating various HVAC systems during design or retrofit, the absolute magnitude of maintenance costs may not be as important as the relative costs. Whichever estimating method or resource is selected, it should be used consistently throughout any evaluation. Mixing information from different resources in an evaluation may provide erroneous results.

Applying simple costs per unit of building floor area for maintenance is highly discouraged. Maintenance costs can be generalized by system types. When projecting maintenance costs for different HVAC systems, the major system components need to be identified with a required level of maintenance. The potential long-term costs of environmental issues on maintenance costs should also be considered.

Factors Affecting Maintenance Costs. Maintenance costs are primarily a measure of labor activity. System design, layout, and configuration can significantly affect the amount of time and effort required for maintenance and, therefore, the maintenance cost. Factors to consider when evaluating maintenance costs include the following:

- **Quantity and type of equipment.** Each piece of equipment requires a core amount of maintenance and time, regardless of its size or capacity. A greater number of similar pieces of equipment are generally more expensive to maintain than larger but fewer units. For example, one manufacturer suggests the annual maintenance for centrifugal chillers is 24 h for a nominal 1000 ton chiller and 16 h for a nominal 500 ton chiller. Therefore, the total maintenance labor for a 1000 ton chiller plant with two 500 ton chillers would be 32 h, or 1/3 more than a single 1000 ton chiller.

- **Equipment location and access.** The ability to maintain equipment in a repeatable and cost-effective manner is significantly affected by the equipment's location and accessibility. Equipment that is difficult to access increases the amount of time required to maintain it, and therefore increases maintenance cost. Equipment maintenance requiring erection of ladders and scaffolding or hydraulic lifts increases maintenance costs while likely reducing the quantity and quality of maintenance performed. Equipment location may also dictate an unusual working condition that could require more service personnel than normal. For example, maintenance performed in a confined space (per OSHA definitions) requires an additional person to be present, for safety reasons.
- **System run time.** The number of hours of operation for a HVAC system affects maintenance costs. Many maintenance tasks are dictated by equipment run time. The greater the run time, the more often these tasks need to be performed.
- **Critical systems.** High-reliability systems require more maintenance to ensure uninterrupted system operation. Critical system maintenance is also usually performed with stringent shutdown and failsafe procedures that tend to increase the amount of time required to service equipment. An office building system can be turned off for a short time with little effect on occupants, allowing maintenance almost any time. Shutdown of a hospital operating room or pharmaceutical manufacturing HVAC system, on the other hand, must be coordinated closely with the operation of the facility to eliminate risk to patients or product. Maintenance on critical systems may sometimes incur labor premiums because of unusual shutdown requirements.
- **System complexity.** More complex systems tend to involve more equipment and sophisticated controls. Highly sophisticated systems may require highly skilled service personnel, who tend to be more costly.
- **Service environment.** HVAC systems subjected to harsh operating conditions (e.g., coastal and marine environments) or environments like industrial operations may require more frequent and/or additional maintenance.
- **Local conditions.** The physical location of the facility may require additional maintenance. Equipment in dusty or dirty areas or exposed to seasonal conditions (e.g., high pollen, leaves) may require more frequent or more difficult cleaning of equipment and filters. Additional maintenance tasks may be needed.
- **Geographical location.** Maintenance costs for remote locations must consider the cost of getting to and from the locations. Labor costs for the number of anticipated trips and their duration for either in-house or outsourced service personnel to travel to and from the site must be added to the maintenance cost to properly estimate the total maintenance cost.
- **Equipment age.** The effect of age on equipment repair costs varies significantly by type of HVAC equipment. Technologies in equipment design and application have changed significantly, affecting maintenance costs.
- **Available infrastructure.** Maintenance costs are affected by the availability of an infrastructure that can maintain equipment, components, and systems. Available infrastructure varies on a national, regional, and local basis and is an important consideration in the HVAC system selection process.

10.8 Economic Analysis Techniques

Analysis of overall owning and operating costs and comparisons of alternatives require an understanding of the cost of lost opportunities, inflation, and the time value of money. This process of economic analysis of alternatives falls into two general categories: simple payback analysis and detailed economic analyses (life-cycle cost analyses).

A simple payback analysis reveals options that have short versus long paybacks. Often, however, alternatives are similar and have similar paybacks. For a more accurate comparison, a more comprehensive economic analysis is warranted. Many times it is appropriate to have both a simple payback analysis and a detailed economic analysis. The simple payback analysis shows which options should not be considered further, and the detailed economic analysis determines which of the viable options are the strongest. The strongest options can be accepted or further analyzed if they include competing alternatives.

10.8.1 Simple Payback

In the simple payback technique, a projection of the revenue stream, cost savings, and other factors is estimated and compared to the initial capital outlay. This simple technique ignores the cost of borrowing money (interest) and lost opportunity costs. It also ignores inflation and the time value of money.

Example 10-1 Equipment item 1 costs $10,000 and will save $2000 per year in operating costs; equipment item 2 costs $12,000 and saves $3000 per year. Which item has the best simple payback?

Item 1$10,000($2000/yr) = 5-year simple payback

Item 2$12,000/($3000/yr) = 4-year simple payback

Because analysis of equipment for the duration of its realistic life can produce a very different result, the simple payback technique should be used with caution.

10.8.2 More Sophisticated Economic Analysis Methods

Economic analysis should consider details of both positive and negative costs over the analysis period, such as varying inflation rates, capital and interest costs, salvage costs, replacement costs, interest deductions, depreciation allowances, taxes, tax credits, mortgage payments, and all other costs associated with a particular system. See the section on Symbols at the end of this chapter for definitions of variables.

Present-Value (Present Worth) Analysis. All sophisticated economic analysis methods use the basic principles of present value analysis to account for the time value of money.

The total present value (present worth) for any analysis is determined by summing the present worths of all individual items under consideration, both future single-payment items and series of equal future payments. The scenario with the highest present value is the preferred alternative.

Single-Payment Present-Value Analysis. The cost or value of money is a function of the available interest rate and inflation rate. The future value F of a present sum of money P over n periods with compound interest rate i per period is

$$F = P(1 + i)^n \tag{10-1}$$

Conversely, the present value or present worth P of a future sum of money F is given by

$$P = F/(1 + i)^n \tag{10-2}$$

or

$$P = F \times \mathrm{PWF}(i,n)_{\mathrm{sgl}} \tag{10-3}$$

where the single-payment present-worth factor $\mathrm{PWF}(i,n)_{\mathrm{sgl}}$ is defined as

$$\mathrm{PWF}(i,n)_{\mathrm{sgl}} = 1/(1 + i)^n \tag{10-4}$$

Example 10-2 Calculate the value in 10 years at 10% per year interest of a system presently valued at $10,000

$$F = P(1 + i)^n = \$10{,}000\ (1 + 0.1)^{10} = \$25{,}937.42$$

Example 10-3 Using the present-worth factor for 10% per year interest and an analysis period of 10 years, calculate the present value of a future sum of money valued at $10,000. (Stated another way, determine what sum of money must be invested today at 10% per year interest to yield $10,000 10 years from now.)

$$P = F \times \mathrm{PWF}(i,n)_{\mathrm{sgl}}$$

$$P = \$10{,}000 \times 1/(1 + 0.1)^{10}$$

$$= \$3855.43$$

Series of Equal Payments. The present-worth factor for a series of future equal payments (e.g., operating costs) is given by

$$\mathrm{PWF}(i,n)_{\mathrm{ser}} = \frac{(1+i)^n - 1}{i(1+i)^n} \tag{10-5}$$

The present value P of those future equal payments (PMT) is then the product of the present-worth factor and the payment [i.e., $P = \mathrm{PWF}(i,n)_{\mathrm{ser}} \times \mathrm{PMT}$].

The number of future equal payments to repay a present value of money is determined by the capital recovery factor (CRF), which is the reciprocal of the present-worth factor for a series of equal payments:

$$\mathrm{CRF} = \mathrm{PMT}/P \tag{10-6}$$

$$\mathrm{CRF}(i,n)_r = \frac{i(1+i)^n}{(1+i)^n - 1} = \frac{i}{1-(1+i)^{-n}} \tag{10-7}$$

The CRF is often used to describe periodic uniform mortgage or loan payments.

Note that when payment periods other than annual are to be studied, the interest rate must be expressed per appropriate period. For example, if monthly payments or return on investment are being analyzed, then interest must be expressed per month, not per year, and n must be expressed in months.

Example 10-4 Determine the present value of an annual operating cost of $1000 per year over 10 years, assuming 10% per year interest rate.

$$\mathrm{PWF}(i,n)_{\mathrm{ser}} = [(1+0.1)^{10} - 1]/[0.1(1+0.1)^{10}] = 6.14$$

$$P = \$1000(6.14) = \$6140$$

Example 10-5 Determine the uniform monthly mortgage payments for a loan of $100,000 to be repaid over 30 years at 10% per year interest. Because the payment period is monthly, the payback duration is 30(12) = 360 monthly periods, and the interest rate per period is 0.1/12 = 0.00833 per month.

$$\begin{aligned}
\mathrm{CRF}(i,n) &= 0.00833(1+0.00833)^{360}/[(1+0.00833)^{360} - 1] \\
&= 0.008773 \\
\mathrm{PMT} &= P(\mathrm{CRF}) \\
&= \$100{,}000(0.008773) \\
&= \$877.30 \text{ per month}
\end{aligned}$$

Improved Payback Analysis. This somewhat more sophisticated payback approach is similar to the simple payback method, except that the cost of money (interest rate, discount rate, etc.) is considered. Solving Equation (10-7) for n yields the following:

$$n = \frac{\ln[\mathrm{CRF}/(\mathrm{CRF} - i)]}{\ln(1+i)} \tag{10-8}$$

Given known investment amounts and earnings, CRFs can be calculated for the alternative investments. Subsequently, the number of periods until payback has been achieved can be calculated using Equation (10-8).

Example 10-6 Compare the years to payback of the same items described in Example 10-1 if the value of money is 10% per year.

Item 1
cost = $10,000
savings = $2000/year
CRF = $2000/$10,000 = 0.2
n = ln[0.2/(0.2 – 0.1)]/ln(1 + 0.1) = 7.3 years

Item 2
cost = $12,000
savings = $3000/year
CRF = $3000/$12,000 = 0.25
n = ln[0.25/(0.25 – 0.1)]/ln(1 + 0.1) = 5.4 years

If years to payback is the sole criteria for comparison, Item 2 is preferable because the investment is repaid in a shorter period of time.

Accounting for Inflation. Different economic goods may inflate at different rates. Inflation reflects the rise in the real

cost of a commodity over time and is separate from the time value of money. Inflation must often be accounted for in an economic evaluation. One way to account for inflation is to substitute effective interest rates that account for inflation into the equations given in this chapter.

The effective interest rate i' , sometimes called the real rate, accounts for inflation rate j and interest rate i or discount rate i_d; it can be expressed as follows (Kreider and Kreith 1982):

$$i' = \frac{1+i}{1+j} - 1 = \frac{i-j}{1+j} \tag{10-9}$$

Different effective interest rates can be applied to individual components of cost. Projections for future fuel and energy prices are available in the *Annual Supplement to NIST Handbook* 135 (Rushing et al. 2010).

Example 10-7 Determine the present worth P of an annual operating cost of $1000 over 10 years, given a discount rate of 10% per year and an inflation rate of 5% per year.

$$i' = (0.1 - 0.05)/(1 + 0.05) = 0.0476$$

$$\text{PWF}(i',n)_{\text{ser}} = \frac{(1+0.0476)^{10} - 1}{0.0476(1+0.0476)^{10}} = 7.813$$

$$P = \$1000(7.813) = \$7813$$

The following are three common methods of present-value analysis that include life-cycle cost factors (life of equipment, analysis period, discount rate, energy escalation rates, maintenance cost, etc., as shown in Table 10-1). These comparison techniques rely on the same assumptions and economic analysis theories but display the results in different forms. They also use the same definition of each term. All can be displayed as a single calculation or as a cash flow table using a series of calculations for each year of the analysis period.

Internal Rate of Return. The internal rate of return (IRR) method calculates a return on investment over the defined analysis period. The annual savings and costs are not discounted, and a cash flow is established for each year of the analysis period, to be used with an initial cost (or value of the loan). Annual recurring and special (nonannual) savings and costs can be used. The cash flow is then discounted until a calculated discount rate is found that yields a net present value of zero. This method assumes savings are reinvested at the same calculated rate of return; therefore, the calculated rates of return can be overstated compared to the actual rates of return.

Another version of this is the **modified** or **adjusted internal rate of return (MIRR or AIRR)**. In this version, reinvested savings are assumed to have a given rate of return on investment, and the financed moneys a given interest rate. The cash flow is then discounted until a calculated discount rate is found that yields a net present value of zero. This method gives a more realistic indication of expected return on investment, but the difference between alternatives can be small.

The most straightforward method of calculating the AIRR requires that the SIR for a project (relative to its base case) be calculated first. Then the AIRR can be computed easily using the following equation:

$$\text{AIRR} = (1 + r)(\text{SIR})^{1/N} - 1 \tag{10-10}$$

where r is the reinvestment rate and N is the number of years in the study period. Using the SIR of 12.6 from Equation (10-10) and a reinvestment rate of 3% [the minimum acceptable rate of return (MARR)], the AIRR is found as follows:

$$\text{AIRR}_{\text{A:BC}} = (1 + 0.03)(12.6)^{1/20} - 1 = 0.1691$$

Because an AIRR of 16.9% for the alternative is greater than the MARR, which in this example is the FEMP discount rate of 3%, the project alternative is considered to be cost effective in this application.

Life-Cycle Costs. This method of analysis compares the cumulative total of implementation, operating, and maintenance costs. The total costs are discounted over the life of the system or over the loan repayment period. The costs and investments are both discounted and displayed as a total combined life-cycle cost at the end of the analysis period. The options are compared to determine which has the lowest total cost over the anticipated project life.

Example 10-8 A municipality is evaluating two different methods of providing chilled water for cooling a government office building: purchasing chilled water from a central chilled-water utility service in the area, or installing a conventional chiller plant. Because the municipality is not a tax-paying entity, the evaluation does not need to consider taxes, allowing for either a current or constant dollar analysis.

The first-year price of the chilled-water utility service contract is $65,250 per year, and is expected to increase at a rate of 2.5% per year.

The chiller and cooling tower would cost $220,000, with an expected life of 20 years. A major overhaul ($90,000) of the chiller is expected to occur in year ten. Annual costs for preventative maintenance ($1400), labor ($10,000), water ($2000) and chemical treatments ($1800) are all expected to keep pace with inflation, which is estimated to average 3% annually over the study period. The annual electric cost ($18,750) is expected to increase at a rate of 5% per year. The municipality uses a discount rate of 8% to evaluate financial decisions.

Which option has the lowest life-cycle cost?

Solution:

Table 10-7 compares the two alternatives. For the values provided, alternative 1 has a 20-year life cycle cost of $769,283 and alternative 2 has a 20-year life cycle cost of $717,100. If LCC is the only basis for the decision, alternative 2 is preferable because it has the lower life-cycle cost.

Computer Analysis. Many computer programs are available that incorporate economic analysis methods. These range from simple calculations developed for popular spreadsheet applications to more comprehensive, menu-driven computer programs. Commonly used examples of the latter include Building Life-Cycle Cost (BLCC) and PC-ECONPACK.

BLCC was developed by the National Institute of Standards and Technology (NIST) for the U.S. Department of Energy (DOE). The program follows criteria established

Table 10-7 Two Alternative LCC Examples for Example 10-8

Alternative 1: Purchase Chilled Water from Utility

	Year										
	0	1	2	3	4	5	6	7	8	9	10
First costs		—	—	—	—	—	—	—	—	—	—
Chilled-water costs		$65,250	$66,881	$68,553	$70,267	$72,024	$73,824	$75,670	$77,562	$79,501	$81,488
Replacement costs		—	—	—	—	—	—	—	—	—	—
Maintenance costs		—	—	—	—	—	—	—	—	—	—
Net annual cash flow		65,250	66,881	68,553	70,267	72,024	73,824	75,670	77,501	79,501	81,488
Present value of cash flow		60,417	57,340	54,420	51,648	49,018	46,522	44,153	41,904	39,770	37,745

		Year									
		11	12	13	14	15	16	17	18	19	20
Financing annual payments		—	—	—	—	—	—	—	—	—	—
Chilled-water costs		$83,526	$85,614	$87,754	$89,948	$92,197	$94,501	$96,864	$99,286	$101,768	$104,312
Replacement costs		—	—	—	—	—	—	—	—	—	—
Maintenance costs		—	—	—	—	—	—	—	—	—	—
Net annual cash flow		83,526	85,614	87,754	89,948	92,197	94,501	96,864	99,286	101,768	104,312
Present value of cash flow		35,823	33,998	32,267	30,624	29,064	27,584	26,179	24,846	23,581	22,380
20-year life-cycle cost	$769,823										

Alternative 2: Install Chiller and Tower

	Year										
	0	1	2	3	4	5	6	7	8	9	10
First costs	$220,000	—	—	—	—	—	—	—	—	—	—
Energy costs		$18,750	$19,688	$20,672	$21,705	$22,791	$23,930	$25,127	$26,383	$27,702	$29,087
Replacement costs		—	—	—	—	—	—	—	—	—	90,000
Maintenance costs		15,200	15,656	16,126	16,609	17,108	17,621	18,150	18,694	19,255	19,833
Net annual cash flow	220,000	33,950	35,344	36,798	38,315	39,898	41,551	43,276	45,077	46,957	138,920
Present value of cash flow	220,000	31,435	30,301	29,211	28,163	27,154	26,184	25,251	24,354	23,490	64,347

		Year									
		11	12	13	14	15	16	17	18	19	20
Financing annual payments		—	—	—	—	—	—	—	—	—	—
Energy costs		$30,542	$32,069	$33,672	$35,356	$37,124	$38,980	$40,929	$42,975	$45,124	$47,380
Replacement costs		—	—	—	—	—	—	—	—	—	—
Maintenance costs		20,428	21,040	21,672	22,322	22,991	23,681	24,392	25,123	25,877	26,653
Net annual cash flow		50,969	53,109	55,344	57,678	60,115	62,661	65,320	68,099	71,001	74,034
Present value of cash flow		21,860	21,090	20,350	19,637	18,951	18,290	17,654	17,042	16,452	15,884
20-year life-cycle cost	$717,100										

by the Federal Energy Management Program (FEMP) and the Office of Management and Budget (OMB). It is intended for evaluation of energy conservation investments in nonmilitary government buildings; however, it is also appropriate for similar evaluations of commercial facilities.

PC-ECONPACK, developed by the U.S. Army Corps of Engineers for use by the DOD, uses economic criteria established by the OMB. The program performs standardized life-cycle cost calculations such as net present value, equivalent uniform annual cost, SIR, and discounted payback period.

10.9 Reference Equations

Table 10-8 lists commonly used discount formulas as addressed by NIST. Refer to NIST *Handbook* 135 (Fuller and Petersen 1996) for detailed discussions.

10.10 Problems

10.1 If $1000 is invested at 8% interest, determine the value of this money in 10 years.

Table 10-8 Commonly Used Discount Formulas

Name	Algebraic Form[a,b]	Name	Algebraic Form[a,b]
Single compound-amount (SCA) equation	$F = P \cdot [(1+d)^n]$	Uniform compound-amount (UCA) equation	$F = A \cdot \left[\frac{(1+d)^n - 1}{d}\right]$
Single present-value (SPV) equation	$P = F \cdot \left[\frac{1}{(1+d)^n}\right]$	Uniform present-value (UPV) equation	$P = A \cdot \left[\frac{(1+d)^n - 1}{d(1+d)^n}\right]$
Uniform sinking-fund (USF) equation	$A = F \cdot \left[\frac{d}{(1+d)^n - 1}\right]$	Modified uniform present-value (UPV*) equation	$P = A_0 \cdot \left(\frac{1+e}{d-e}\right) \cdot \left[1 - \left(\frac{1+e}{1+d}\right)^n\right]$
Uniform capital recovery (UCR) equation	$A = P \cdot \left[\frac{d(1+d)^n}{(1+d)^n - 1}\right]$		

where

A = end-of-period payment (or receipt) in a uniform series of payments (or receipts) over n periods at d interest or discount rate

A_0 = initial value of a periodic payment (receipt) evaluated at beginning of study period

A_t = $A_0(1+e)^t$, where $t = 1, \ldots, n$

d = interest or discount rate

e = price escalation rate per period

Source: NIST *Handbook* 135 (Fuller and Petersen 1996).

[a] Note that the USF, UCR, UCA, and UPV equations yield undefined answers when $d = 0$. The correct algebraic forms for this special case would be as follows: USF formula, $A = F/N$; UCR formula, $A = P/N$; UCA formula, $F = An$. The UPV* equation also yields an undefined answer when $e = d$. In this case, $P = A_0 \cdot n$.

[b] The terms by which known values are multiplied are formulas for the factors found in discount factor tables. Using acronyms to represent the factor formulas, the discounting equations can also be written as $F = P \times$ SCA, $P = F \times$ SPV, $A = F \times$ USF, $A = P \times$ UCR, $F =$ UCA, $P = A \times$ UPV, and $P = A_0 \times$ UPV*.

10.2 Find the present worth of money that will have a value of $35,000 in 3 years with an interest rate of 9%.

10.3 $1000 is invested at the end of each year for 10 years. Interest is 11%. Find the amount accumulated. [$16,722]

10.4 If $100,000 is invested at 8% interest, find the yearly withdrawal that will use up the money in 20 years.

10.5 The cost of a new heat pump system is $3000 with an expected lifetime of 20 years. Neglect energy and maintenance costs. Find the annual cost if the salvage value is $0 and the interest rate is 8%.

10.6 A new heating system has a cost of $15,000 and a salvage value of $5000, independent of age. The new system saves $1400 per year in fuel cost. Calculate the breakeven point if $i = 9\%$. Neglect maintenance costs.

10.7 A new high-efficiency cooling system costs $60,000 and saves $7500 in energy costs each year. The system has a salvage value of $10,000 in 20 years. Compute the rate of return. Neglect maintenance costs. [11.25%]

10.8 The costs of two small heat pump units A and B are $1000 and $1200 and the annual operating costs are $110 and $100, respectively. The interest rate is 8% and the amortization is selected as 20 years. Compare the systems on the basis of present worth.

10.9 Compare the units in Problem 10.8 on the basis of uniform annual costs.

10.10 An installation is going to require a 500 ton chiller. An annual energy analysis for this office building application shows that the required ton-hours over the year will be 2,100,000. The economic data are given below.

	Chiller A	Chiller B
Average chiller efficiency	0.73 kW/ton	0.63 kW/ton
Initial cost	$221,500	$240,500
Installation cost	$19,000	$19,000
Electricity cost	6¢/kWh	5.9¢/kWh
Maintenance costs	$9,500	$10,000
Estimated life	20 years	20 years

Perform a simple payback analysis for this option. [1.4 yrs]

10.11 Symbols

AIRR = modified or adjusted internal rate of return (MIRR or AIRR)

c = cooling system adjustment factor

C = total annual building HVAC maintenance cost

C_e = annual operating cost for energy

$C_{s,\mathrm{assess}}$ = assessed system value

$C_{s,\mathrm{init}}$ = initial system cost

$C_{s,\mathrm{salv}}$ = system salvage value at end of study period

C_y = uniform annualized mechanical system owning, operating, and maintenance costs

CRF = capital recovery factor

CRF(i,n) = capital recovery factor for interest rate i and analysis period n

CRF(i',n) = capital recovery factor for interest rate i' for items other than fuel and analysis period n

CRF(i'',n) = capital recovery factor for fuel interest rate i'' and analysis period n

$\text{CRF}(i_m,n)$ = capital recovery factor for loan or mortgage rate i_m and analysis period n
d = distribution system adjustment factor
D_k = depreciation during period k
$D_{k,\text{SL}}$ = depreciation during period k from straight-line depreciation method
$D_{k,\text{SD}}$ = depreciation during period k from sum-of-digits depreciation method
F = future value of sum of money
h = heating system adjustment factor
i = compound interest rate per period
i_d = discount rate per period
i_m = market mortgage rate
i' = effective interest rate for all but fuel
i'' = effective interest rate for fuel
I = insurance cost per period
ITC = investment tax credit
j = inflation rate per period
j_e = fuel inflation rate per period
k = end of period(s) during which replacement(s), repair(s), depreciation, or interest are calculated
M = maintenance cost per period
n = number of periods under analysis
P = present value of a sum of money
P_k = outstanding principle on loan at end of period k
PMT = future equal payments
PWF = present worth factor
$\text{PWF}(i_d,k)$ = present worth factor for discount rate i_d at end of period k
$\text{PWF}(i',k)$ = present worth factor for effective interest rate i' at end of period k
$\text{PWF}(i,n)_{\text{sgl}}$ = single payment present worth factor
$\text{PWF}(i,n)_{\text{ser}}$ = present worth factor for a series of future equal payments
R_k = net replacement, repair, or disposal costs at end of period k
SIR = savings-to-investment ratio
T_{inc} = net income tax rate
T_{prop} = property tax rate
T_{salv} = tax rate applicable to salvage value of system

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Chapter 11

AIR-CONDITIONING SYSTEM CONCEPTS

An air-conditioning system maintains desired environmental conditions within a space. In almost every application there are a myriad of options available to the designer to satisfy the basic goal. It is in the selection and combination of these options that the engineer must consider all the parameters relating to the project.

Air-conditioning systems are categorized by how they control temperature and humidity in the conditioned area. They are also segregated to accomplish specific purposes by special equipment arrangement. This chapter considers elements that constitute the system. Chapter 12 describes those systems that are used to solve the psychrometric problem and control the environmental conditions of the occupied space, and Chapter 13 discusses the design of hot water and chilled water systems. Chapters 14 through 19 describe the various components of the air-conditioning system.

11.1 System Objectives and Categories

The two fundamental objectives of the air-conditioning system are to control the quality of the air in the conditioned space and the thermodynamic properties of the air. Additionally, in conjunction with other aspects of the space and system design, they become the major element in providing for the thermal comfort in the space.

11.1.1 Air Quality

Air quality is usually obtained through a process of ventilation combined with filtration. Ventilation consists of the removal of contaminants from the indoor air by replacing the contaminated air with uncontaminated air from the outdoors. This replacement is generally achieved through a process of dilution or, in some cases, displacement. The amount of outdoor air required is determined by the rate and type of contaminant generation, generally as prescribed by ASHRAE Standard 62.

In some systems, the outdoor air is mixed with the return air from the space at the inlet to the air-conditioning system and the mixture, then, conditioned to the thermodynamic state necessary to control the space temperature and humidity. However, this method need not be employed, and often it is preferable to provide total thermodynamic conditioning of the outdoor ventilation air separate and apart from the space conditioning unit or directly to the space via a separate ventilating supply air system. The separate ventilating system, properly applied, is preferable when using variable air volume systems because of the assurance of adequate ventilation at all conditions of load and improved humidity control in humid climates with most temperature control systems.

Another benefit of the separate ventilation air-conditioning unit is that it can be used in conjunction with in-room devices designed to control the room temperature (such as fan-coil units, radiant panels, or induction coil devices). These separate ventilation air-conditioning systems are commonly called dedicated outdoor air systems (DOAS).

11.1.2 Thermodynamic Conditioning

The thermodynamic conditioning of the air is the process of simultaneously cooling and dehumidifying the air or heating (and sometimes humidifying) it to maintain comfort conditions within the space. In most systems designed for human comfort, the system control responds to dry-bulb temperature, with the dehumidification being provided as a noncontrolled by-product of the cooling process.

The fundamental equation describing a sensible cooling or heating process is

$$q_s = 1.1Q\,\Delta t \tag{11-1}$$

where

q_s = heating or sensible cooling load or capacity, Btu/h
Q = flow quantity of supply air, ft^3/min or cfm
1.1 = constant for standard air, Btu·min/h·°F·ft^3
Δt = difference between room temp. and supply air temp.

For design conditions, i.e., the airflow and temperature differential required at the design system load (maximum system capacity), the sensible design load is calculated and Equation (11-1) is usually solved for Q with the designer selecting a Δt. For cooling, the Δt is equal to $t_r - t_s$, where t_s is the supply air temperature, usually selected as the dewpoint temperature necessary to provide the latent cooling requirement at design conditions.

Then, where the myriad of different types of air-conditioning systems arise from are the different methods that are employed to change the capacity as the load reduces. Considering Equation (11-1), if the load q_s reduces (or changes), the only methods by which the equation can be kept in balance is by reducing either the Q or the Δt or some combination of the two. Thermodynamically, four different methods are used to achieve these changes:

- Heat-Cool-Off
- Dual Stream
- Reheat
- Variable Air Volume

11.1.3 Heat-Cool-Off Systems

A heat-cool-off system is any system that responds to the need for changes in capacity by varying the temperature of the air supplied by the unit, usually while maintaining a constant airflow. There are many configurations of heat-cool-off systems, both two-position and modulating, and although it is not common, they can be both variable flow rate Q or variable Δt but are usually the latter. The essential feature that defines a heat-cool-off system is that it is limited to a *single* control zone. Examples of heat-cool-off systems are single-zone air-handling units, residential heating/cooling units, and room fan-coil units.

11.1.4 Dual-Stream Systems

Dual-stream systems are applicable to cases in which multiple control zones are to be served from a single air-conditioning unit. They are constant flow (Q), variable Δt systems. They achieve the variation in supply air temperature (from full cooling load to full heating load) by mixing a high-temperature stream with a low-temperature stream. The two most common dual-stream systems are double-duct and multi-zone systems. The induction system is another special configuration of a dual-stream system.

11.1.5 Reheat Systems

Reheat systems also apply to cases in which multiple control zones are served from a single air-conditioning unit. However, single-zone units that function as heat-cool-off *can* be thermodynamically configured as reheat systems when applied to controlled dehumidifying requirements. This system is a constant flow, variable Δt system. In the multiple zoned reheat system (often called **terminal reheat systems**) the central air-handling system generally conditions the air to a fixed or controlled dew-point temperature and the cool air is reheated to the temperature required to satisfy the space sensible load for each control zone.)

11.1.6 Variable Air Volume (VAV) Systems

These systems, like the dual-stream and terminal reheat systems, are usually applied in cases in which multiple control zones are served from a single air-conditioning unit. The main difference between variable air volume systems and other multiple-zone systems is that the VAV system varies the flow of the supply air rather than the temperature. Two characteristics of the VAV system become immediately evident:

- The VAV system can either heat or cool at any given time, but it cannot do both at the same time. In order to do both, it must be combined with one of the other systems (i.e., VAV-reheat or VAV-dual stream).
- Conceptually, as the load reduces the air flow rate reduces and as the air flow rate reduces the fan power reduces, ideally as the cube of the flow.

Since, in most applications, the supply fan systems consume more energy annually than the refrigeration systems, the use of VAV systems offer significant potential benefits in energy conservation.

Most systems employ a combination of two or more of the above, such as dual stream/reheat, VAV/dual stream, VAV/reheat, heat-cool-off/reheat, etc. The following discussions will subdivide the system types into air systems and water systems, consistent with the types discussed in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. And, although the physical description and common name of the systems more closely describes their physical configuration, the designer is encouraged to always relate the system for purposes of modeling, diagnostics, or analyses to its generic thermodynamic type as described above.

11.2 System Selection and Design

A "system" is that device or assembly of devices that provides the environmental conditions that the engineer determines are required in the space. It could range from a simple window air-conditioning unit and portable space heater to a fully integrated environmental control system designed to maintain all of the ideal comfort conditions in numerous control zones of a major building complex.

The most important and significant contribution that the design engineer provides in the process of designing an HVAC system is in the "selection" of the system. That selection affects virtually all aspects of the building for the life of the building. The ingredients that go into the system selection include not only the rigid and quantifiable engineering aspects but the economic, psychological, and social aspects involved in constructing, running, operating, and using the building.

This chapter and those that follow relating to system selections are intended to prepare the student for this fascinating task.

11.3 Design Parameters

The design engineer is responsible for considering various systems and selecting the one that will best satisfy all of the design parameters. It is imperative that the designer and the owner collaborate on identifying the goals of the design. Some of the parameters that should be considered are

- Load dynamics
- Performance requirements
- Availability of equipment
- Capacity
- Spatial requirements
- First cost
- Energy consumption
- Operating cost
- Simplicity
- Reliability
- Flexibility (short range and long range)

- Operations requirements
- Serviceability
- Maintainability
- Availability of service
- Availability of replacement components
- Environmental requirements of space
- Environmental requirements of the community

The degree of success of the design of any system is directly related to the accuracy with which the designer (1) identifies the design parameters and (2) achieves them.

Because these factors are interrelated, the owner and system designer must consider how each factor affects the others. The relative impact of these parameters differs with different owners and often changes from one project to another for the same owner.

11.4 Performance Requirements

In addition to goals for providing the desired environment for human comfort, the designer must be aware of and account for other goals the owner may require. These goals may include

- Supporting a process, such as the operation of computer equipment
- Promoting an aseptic environment
- Increasing sales
- Increasing net rental income
- Increasing the salability of a property

Typical concerns of owners include first cost compared to operating cost, the extent and frequency of maintenance and whether that maintenance requires entering the occupied space, the expected frequency of failure of a system, the impact of a failure, and the time required to correct the failure. Each of these concerns has a different priority, depending on the owner's goals.

The owner can only make appropriate value judgments if the designer provides complete information on the advantages and disadvantages (i.e., the impact) of each option. Just as the owner does not usually know the relative advantages and disadvantages of different systems, the designer rarely knows all the owner's financial and functional goals. Hence, it is important to involve the owner in the selection of a system.

11.5 Focusing on System Options

Following the establishment of the design parameters, including the performance requirements, there are four fundamental features and constraints of the system that will inevitably assist the designer in focusing on the type of system. They are (1) the cooling and heating loads, (2) the zoning requirements, (3) the need for heating and ventilation, and (4) the architectural constraints.

11.5.1 Cooling Load

Establishing the cooling and heating loads often narrows the choice to systems that will satisfy all of the part-load requirements, fit within the available space, and are compatible with the building architecture. Chapter 7 covers determination of the magnitude and characteristics of the cooling and heating loads and their variation with time and operating conditions. By establishing the capacity requirement, the equipment size can be estimated. Then, the choice may be narrowed to those systems that work well on projects within a certain size range.

11.5.2 Zoning Requirements

Loads vary over time due to changes in the occupancy, weather, activities, and solar exposure. Each space with a different load dynamic requires a different control zone to maintain reasonably constant thermal conditions. Some areas with special requirements in dry-bulb or wet-bulb control points may need individual control or individual systems, independent of the rest of the building. Variations in indoor conditions that are acceptable in one space may be unacceptable in other areas of the same building. The extent of zoning, the degree of control required in each zone, and the space required for individual controlled spaces also narrow the system choices.

No matter how efficiently a particular system operates or how economical it may be to install, it cannot be considered if it (1) cannot maintain the desired interior environment within an acceptable tolerance under all conditions and occupant activities and (2) does not physically fit into the building without being objectionable and non-maintainable.

11.5.3 Heating and Ventilation

Cooling and humidity control are often the basis of sizing air-conditioning components and subsystems, but the system may also provide other functions, such as heating and ventilation. For example, if the system provides large quantities of outdoor air for ventilation or to replace air exhausted from the building, only systems that transport large air volumes need to be considered. In this situation, the ventilation system requires a large air-handling and duct distribution system, which may eliminate some systems.

Effective delivery of heat to an area may be an equally important factor in system selection. A distribution system that offers high efficiency and comfort for cooling may be a poor choice for heating. This performance compromise may be small for one application in one climate, but it may be unacceptable in another that has more demanding heating requirements.

11.5.4 Architectural Constraints

Air-conditioning systems and the associated distribution systems often occupy a significant amount of space. Major components may also require special support from the structure. The size and appearance of terminal devices (whether they are diffusers, fan-coil units, or radiant panels) have an

effect on the architectural design because they are visible from the occupied space.

Other factors that limit the selection of a system include (1) acceptable noise levels, (2) the space available to house equipment and its location relative to the occupied space, (3) the space available for distribution pipes and ducts, and (4) the acceptability of components obtruding into the occupied space, both physically and visually.

11.6 Narrowing the Choice

Chapter 12 covers the types of systems categorized by all-air systems, air-and-water systems, all-water systems, and unitary refrigerant-based systems. Each section includes an evaluation component, which briefly summarizes the advantages and disadvantages of various systems. Comparing the features against the list of design parameters and their relative importance usually allows identification of two or three systems that best satisfy the project criteria. In making subjective choices, it is helpful to keep notes on all systems considered and the reasons for eliminating those that are unacceptable.

In most cases, two or three systems must be selected: a **secondary system** (or distribution system), which delivers heating or cooling to the occupied space, and a **primary system**, which converts energy from fuel or electricity, and in some cases, an intermediate system, which conveys energy between the primary system and the secondary system. Chapter 13 discusses the most common type of intermediate systems—water systems. The systems are, to a great extent, independent, so that several secondary systems may work with different primary and intermediate systems. In some cases, however, only one secondary system may be suitable for a particular primary system.

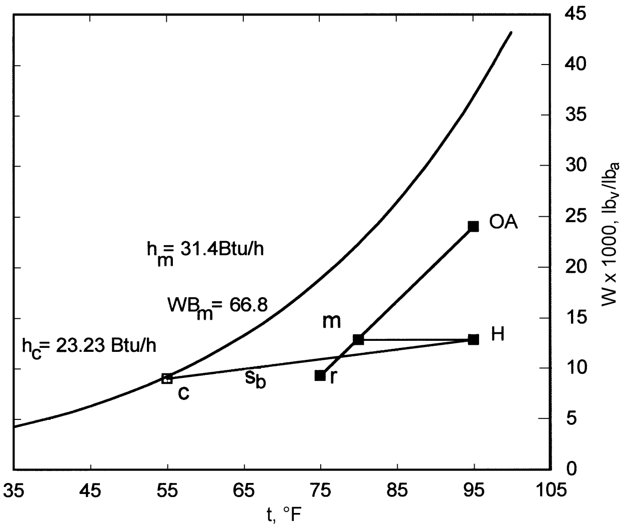
Once subjective analysis has identified two or three systems—and sometimes only one choice may remain—detailed quantitative evaluations of each system must be made. All systems considered should provide satisfactory performance to meet the design parameter goals. The owner or designated decision maker then needs specific data on each system to make an informed choice. Chapter 8 outlines methods for estimating annual energy costs, and Chapter 10 describes economic analyses and life-cycle costing, which can be used to compare the overall economics of systems.

Example 11-1 A dual-stream cooling system is a double-duct system with 55°F saturated air in the cold duct and 95°F db/60°F dewpoint air in the hot duct. The air in the cold duct is conditioned by cooling a mixed air stream (i.e., return air and outdoor air from 80°F db/66.8°F wb to 55°F saturated. The air in the hot duct is conditioned by heating the same mixed air stream to 95°F db with no addition or removal of water vapor. The room or space conditions are 75°F db/50% rh.

(a) A given zone has a sensible design cooling load of 12,960 Btu/h.What airflow (in cfm) must be supplied?

Solution:

From Equation (11-1)



$$q_s = 1.1 Q \, \Delta t$$

$$Q = \frac{q}{1.1(t_r - t_s)} = \frac{12960}{1.1(75-55)}$$

$$Q = 590 \text{ cfm}$$

(b) The load reduces, and when the space sensible load is 50% of design or 6480 Btu/h, what is the condition of the supply air (dry-bulb temperature and wet-bulb temperature)?

Solution:

Dry-bulb temperature

From Equation (11-1)

$$t_{sdb} = t_r - \frac{q}{1.1Q} = 75 - \frac{6480}{1.1(600)}$$

$$t_s = 65\,°\text{F dry bulb}$$

Wet-bulb temperature

The wet-bulb temperature is determined where the mixing line between H (air at "m" heated to 95°F) and C on the psychrometric chart shown in the figure cross the 65°F db line. Thus

$$t_{swb} = 59.7°\text{F wet bulb}$$

(c) At the half-load condition, approximately what airflow from the cold duct and the hot duct will be required?

Solution:

From the following equation for mixed air

$$t_m = t_c + \frac{m_H}{m_C + m_H}(t_H - t_C)$$

Solve for

$$\frac{Q_H}{Q_C} = \frac{m_H}{m_C + m_H} = \frac{t_m - t_C}{t_H - t_C} = \frac{65-55}{95-55} = 0.25$$

Then $Q_H = 0.25Q = 0.25(590) = 148$ cfm

$Q_C = 590 - 148 = 442$ cfm

(d) At the half sensible load condition, approximately what is the load on the cooling coil (in Btu/h) related to this zone?

Solution:

From the following energy equation

$$q_C = \dot{m}(h_m - h_C) = 60\rho_a Q(h_m - h_C)$$
$$q_C = 60(0.075)Q(31.4 - 23.23) = 4.5(442)(8.17)$$
$$= 16{,}250 \text{ Btu/h}$$

(e) At the half sensible load condition, approximately what is the load on the heating coil related to this zone?

Solution:

From Equation (11-1)

$$q_H = 1.1(148)(95 - 65) = 4880 \text{ Btu/h}$$

(f) At full-load conditions, approximately what is the load on the cooling coil related to this zone? The heating coil?

Solution:

$$q_C = 4.5Q(h_m - h_C)$$
$$= 4.5(590)(31.4 - 23.23)$$
$$= 22690 \text{ Btu/h}$$
$$q_H = 0$$

11.7 Energy Considerations of Air Systems

Those systems that move energy from place to place within a building, such as the air distribution systems and liquid fluid circulation systems, are called the energy transport systems. In first selecting and then designing these systems, the power and energy required are seen to be two of the basic selection parameters.

The impact of the power is twofold. First, if the power is less, the first cost will be less because the machinery is smaller. Second, if the power is less, the energy consumption will be less (other things being equal) because the energy consumed is simply the product of the power and the operating hours.

11.7.1 Air System Power and Energy

Equation (11-1), the heat capacity equation, is the first consideration in the design of energy effective systems. It becomes evident that if the designer can do anything to reduce the load, which is a power term (Btu/h or tons of cooling), the system and machinery will be smaller, thus will cost less and use less energy when operated.

The fan power equation is

$$P = \frac{Q\,\Delta p_t}{6350\eta_f} \tag{11-2}$$

where

P = fan power, hp
Q = air circulation rate, ft^3/min
Δp_t = fan total pressure rise, in. w.g.
η_f = fan efficiency, decimal

The equation shows that reducing the flow rate or the system pressure or increasing the fan efficiency will reduce the power. The flow rate can be reduced by (1) reducing the load [Equation (11-1)], (2) by making a careful load analysis that does not include excessive or hidden uncertainty or "safety factors," or (3) by increasing the temperature difference between the supply and room air. [i.e., $(t_r - t_s)$].

The fan pressure equals the pressure loss requirement of the conditioner and the distribution system, a term that can be controlled by the system designer. Friction losses in the distribution system can be expressed by the Darcy-Weisbach equation,

$$\Delta h_f = Cf\frac{L}{D}\frac{\rho V^2}{2g_c} \tag{11-3}$$

where

Δh_f = friction loss, in. w.g.
f = friction factor, dimensionless
L = length of duct, ft
D = diameter of duct, ft
V = velocity of air in duct, ft/s
g_c = units conversion constant, lb$_m$·ft/s^2·lb$_\text{f}$
C = unit conversion factor, 0.1923, $\frac{\text{in. w.g. ft}^2}{\text{lb}_f}$
ρ = density of air, lb$_m$/ft^3

The methods, then, to reduce the fan pressure requirement (fluid head loss) are:

- Limit the length of duct runs to the minimum possible
- Increase the diameter (or equivalent diameter)
- Reduce the velocity
- Reduce the roughness of interior surfaces, which reduces the friction factor

Additionally, the duct fittings, in most systems, create a significant amount of the pressure losses (see Chapter 9). All fittings, in both the supply and return ductwork, should be designed for minimum pressure losses, which are a function of the fitting construction and the velocity head $V^2/2g$. (Also see Chapter 21, 2017 *ASHRAE Handbook—Fundamentals*.) To ensure continued operation with low pressure drops, any device such as a damper, coil, turning vanes, etc., which could result in a blockage to air flow, should be provided with an inspection and access port.

An additional consideration regarding the pressure is the temperature rise across the fan. The temperature rise is expressed by the equation:

$$\Delta t_f = \frac{0.371\,\Delta p_t}{\eta_f} \tag{11-4}$$

where

Δt_f = temperature rise across fan, °F
Δp_t = total pressure rise across fan, in. w.g.
η_f = fan efficiency, decimal

Equation (11-4) shows that if the fan efficiency is 74%, the temperature rise in °F would equal one-half the pressure rise in in. w.g. For example, a 74% efficient fan producing 4 in. w.g. pressure would raise the air temperature 2°F. In most systems, this temperature rise becomes part of the cooling load, thus requiring the use of yet more energy.

Regarding fan efficiencies, fans should always be selected at the maximum efficiency point on their curves, and it is highly recommended that a designer always use a fan curve when selecting a fan so that the anticipated range of operation can be analyzed.

The fan energy equation (11-5) is the power equation multiplied by the hours of operation and with the appropriate terms for motor efficiency and conversion of horsepower to kilowatts.

$$q_{fan} = \frac{Q \Delta p_t \theta}{8512 \eta_f \eta_m} \qquad (11\text{-}5)$$

where

q_{fan} = fan energy, kWh
Q = air circulation rate, ft^3/min
Δp_t = total fan pressure, in. w.g.
θ = time of operation, hours
η_f = fan efficiency
η_m = motor efficiency

The only variables that have been incorporated in this equation that were not included in the power equation (11-2) are the hours of operation and the motor efficiency. In selecting and designing a system, accommodation should always be made to maintain an unoccupied building under controlled conditions while shutting down the major energy consuming devices or operating them at a low energy consumption idle mode.

Regarding motor efficiencies, the use of high-efficiency motors is always recommended for fan or pump drives.

Another form of the fan energy equation for *a fixed or given system* (i.e., the system curve is constant) is:

$$P_{fan} = \frac{Q^3}{6350 \eta_f C_s^2} \qquad (11\text{-}6)$$

$$q_{fan} = \frac{Q^3 \theta}{8512 \eta_f \eta_m C_s^2} \qquad (11\text{-}7)$$

where C_s is the system constant in $\text{ft}^3/[\text{min (in w.g.)}^{0.5}]$.

This equation shows that, if the air delivery rate can be reduced for a given system, the energy consumption is reduced as the cube of the flow rate (e.g., 20% flow reduction equates to 49% energy reduction, 50% flow reduction equates to 87.5% energy reduction, etc.). This characteristic explains one of the major benefits of using a variable flow (VAV) system instead of a variable Δt system.

11.7.2 Water System Power and Energy

The relationships of power and energy consumption in water systems are similar to those in air systems, except for the change in the constants and dimensions. The fundamental power and energy equations for pumps in chilled and heating water systems are:

$$P_p = \frac{Q \Delta h}{3960 \eta_p} \qquad (11\text{-}8)$$

$$q_p = \frac{Q \Delta h \theta}{5308 \eta_p \eta_m} \qquad (11\text{-}9)$$

where

P_p = pump horsepower
q_p = pump energy, kWh
Q = water circulating rate, gallons/minute (gpm)
Δh = pump head, feet of water
θ = time of operation, hours
η_p = pump efficiency, decimal
η_m = motor efficiency, decimal

The principles of energy effective design discussed above for air systems apply equally to water systems.

Example 11-2 An air system is designed to handle 30,000 cfm at a total pressure of 6 in. w.g. If the ideal fan horsepower can be calculated by the use of Equation (11-2) with the fan efficiency term set at 100% efficiency,

(a) What is the ideal fan horsepower?

$$P = \frac{29630(6)}{6350} = 28.0 \text{ hp}$$

(b) If the fan were 70% efficient, what would be the power requirement?

$$P = \frac{28.0}{0.70} = 40.0 \text{ hp}$$

(c) If the pressure required was reduced to 4 in. w.g. by modifying the distribution system, what power would be required?

$$P = \frac{29630(4)}{6350(0.70)} = 26.7 \text{ hp}$$

(d) If after installing the system, it was decided the capacity could be reduced 20% by reducing the fan speed, what would the power requirement be?

$$P = P_c \left(\frac{\text{New flow rate}}{\text{Original flow rate}} \right)^3$$

$$= 26.7(0.8)^3 = 13.7 \text{ hp}$$

11.8 Basic Central Air-Conditioning and Distribution System

The basic secondary system is an all-air, single-zone, air-conditioning system. It may be designed to supply a constant air volume or a variable air volume and for low-,

medium-, or high-velocity air distribution. Normally, the equipment is located outside the conditioned area, in a basement, penthouse, service area, or outdoors on the roof. It can, however, be installed within the area if conditions permit. The equipment can be adjacent to the primary heating and refrigeration equipment or at considerable distance from it by circulating refrigerant, chilled water, hot water, or steam for energy transfer.

11.8.1 Applications

Some central system applications are (1) spaces with uniform loads, (2) small spaces requiring precision control, (3) multiple systems for large areas, (4) systems for complete environmental control, and (5) primary sources of conditioned air for other subsystems.

11.8.2 Spaces with Uniform Loads

Spaces with uniform loads are generally those with relatively large open areas and small external loads, such as theaters, auditoriums, department stores, and the public spaces of many buildings. Adjustment for minor variations in the air-conditioning load in parts of the space can be made by supplying more or less air in the original design and balance of the system.

In office buildings, the interior areas generally meet these criteria as long as local areas of relatively intense and variable heat sources, such as computer rooms and conference rooms, are treated separately. In these applications, dwarf partitions allow wider diffusion of the conditioned air and equalization of temperatures. These areas usually require year-round cooling, and any isolated spaces with limited occupancy may require special evaluation, as discussed in Chapter 3 of the 2015 *ASHRAE Handbook—HVAC Applications.*

In most single-room commercial applications, temperature variations up to 4°F at the outside walls are usually considered acceptable for tenancy requirements. However, these variations should be carefully determined and limited during design. If people sit or work near the outside walls or if they are isolated by partitions, supplementary heating equipment may be required at the walls, depending on the outdoor design temperature in winter and the thermal characteristics of the wall. If the wall surface temperature is calculated to be more than 10°F below the room temperature, some special consideration must be given to the placing of heat at the perimeter.

11.8.3 Spaces Requiring Precise Control

These spaces are usually isolated rooms within a larger building (such as a computer room, auditorium, etc.) and have stringent requirements for such things as cleanliness, humidity, temperature control, and/or air distribution. Central system components should be selected and assembled to meet the exact requirements of the area.

11.8.4 Multiple Systems for Large Areas

In large buildings such as hangars, factories, large stores, office buildings, and hospitals, practical considerations sometimes require installation of multiple central single-zone systems. The size of the individual system is determined by an evaluation of the relative design parameters.

11.8.5 Systems for Environmental Control

ASHRAE Standard 62 specifies ventilation rates for acceptable indoor air quality (IAQ). All-air systems often provide the necessary air supply to dilute the contaminants in the air in controlled spaces. These applications consist of combinations of supply, return, and exhaust systems that circulate the diluting air through the space. The designer must consider the terminal systems used because establishing adequate dilution volumes is closely related to design criteria, occupancy type, air delivery, and scavenging methods. Indoor air quality is an essential consideration in the design of all systems.

Cleanliness of the air supply also relates directly to the level of environmental control desired. Suitable air filtration should be incorporated in the central system upstream from the air-moving and temperature control equipment. Some applications, such as hospitals, require downstream filtration as well.

Chapter 29 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* has information on air cleaners.

11.8.6 Primary Sources for Other Systems

Systems for separate control of multiple zones are described in Chapter 12. These secondary systems may move a constant or variable supply of conditioned air for ventilation and temperature control and handle some or all of the air-conditioning load. Use of a secondary system may reduce the amount of conditioned air required to be delivered by the central system, thereby reducing the size of and the space required for ductwork. Ductwork size can be reduced further either by moving air at high velocities or by delivering air at reduced temperatures. However, high-velocity system design must consider the resultant high pressure, sound levels, and power and energy requirements. System design for low-temperature air delivery must consider the minimum required ventilation rates and insulation and condensation problems associated with lower temperatures as well as the additional refrigeration power.

11.8.7 Central System Performance

Figure 11-1 shows a typical draw-through central system that supplies conditioned air to a single zone or to multiple zones. A blow-through configuration may also be used if space or other conditions dictate or require. The quantity and quality of supplied air are fixed by space requirements and determined as indicated in Chapters 6 and 7. Air gains and loses heat by contacting heat transfer surfaces and by mixing with air of another condition. Some of this mixing is intentional, as at the outdoor air intake; for others, mixing is the result of the physical characteristics of a particular component, as when untreated air passes without contacting the fins of a coil (bypass factor).

All treated and untreated air must be thoroughly mixed for maximum performance of heat transfer surfaces and for uniform temperatures in the airstream. Stratified, parallel paths of treated and untreated air must be avoided, particularly in the

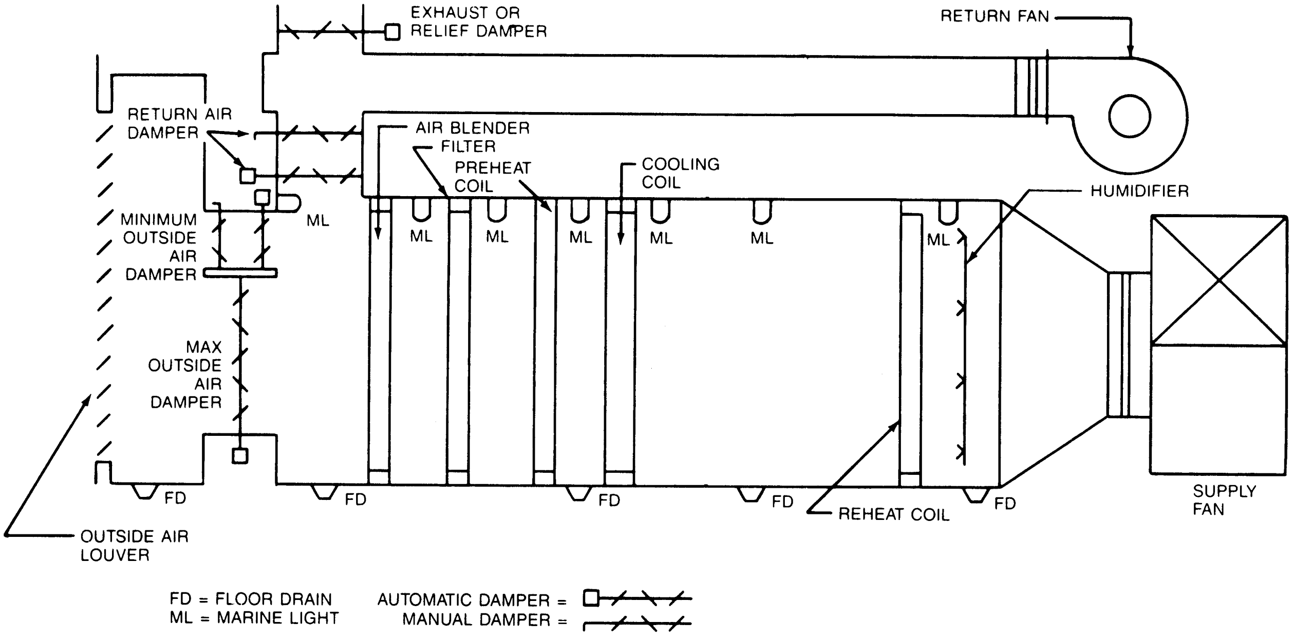


Fig. 11-1 Equipment Arrangement for Central System Draw-Through Unit

vertical plane of systems using double-inlet or multiple-wheel fans. Because these fans do not completely mix the air, different temperatures can occur in branches coming from opposite sides of the supply duct.

11.9 Smoke Management

Air-conditioning systems are often used for smoke control during fires. Controlled air flow can provide smoke-free areas for occupant evacuation and fire fighter access. Space pressurization attempts to create a low-pressure area at the smoke source, surrounding it with high-pressure spaces. For more information, see Chapter 53 of the 2015 *ASHRAE Handbook—HVAC Applications.* The ASHRAE publication *Principles of Smoke Management Systems* (Klote and Milke 2002) has detailed information.

11.10 Components

11.10.1 Air-Conditioning Units

To determine a system's air-handling requirement, the designer must consider the function and physical characteristics of the space to be conditioned and the air volume and thermal exchange capacities required. Then, the various components may be selected and arranged.

Further, the designer should consider economics in component selection. Both initial cost and operating costs affect design decisions. The designer should not arbitrarily design for a 500 fpm face velocity, which has been common for selection of cooling coils and other components. Filter and coil selection at 300 to 400 fpm, with resultant lower pressure loss, could produce a substantial payback in constant volume systems.

Figure 11-1 shows a general arrangement of the components for a single-zone, all-air central system suitable for year-round air conditioning. With this arrangement, close control of temperature and humidity are possible. All these components are seldom used simultaneously in a comfort application. Although Figure 11-1 indicates a built-up system, most of the components are available from many manufacturers completely assembled or in subassembled sections that can be bolted together in the field. These units are called **air-handling units**.

11.10.2 Return and Relief Air Fans

A return air fan is optional but is essential for the proper operation of a so-called air economizer or free cooling systems unless the excess intake air can be relieved directly from the space. It provides a positive return and exhaust from the conditioned area, preventing overpressurization of the space when mixing dampers provide cooling with outdoor air at times when the outdoor air temperature is at or lower than the required supply air temperature.

The return air fan prevents excess ambient pressure in the conditioned space(s) and reduces the static pressure the supply fan has to work against.

The supply fan(s) must be carefully matched with the return fans. The return air fan should handle a smaller amount of air to account for the building ventilation air requirements and to ensure a slight positive pressure in the conditioned space.

In many situations, a relief (or exhaust) air fan may be used instead of a return fan. A relief air fan relieves ventilation air introduced during air economizer operation and operates only when this control cycle is in effect.

When a relief air fan is used, the supply fan must be designed for the total supply and return static pressure in the system and to operate without the relief air fan during the non-economizer mode of operation. During the economizer mode of operation for a constant volume system, the relief fan must be controlled to exhaust at a rate that tracks the quantity of outdoor air introduced, to ensure a slight positive pressure in the conditioned space, as with the return air fan system.

When a mixing chamber and economizer system are utilized with a variable air volume system, special precautions must be taken in the control of the return air or relief air fans to ensure that adequate ventilation air is provided at all times. These systems become so complex that the use of an outdoor air-return air mixing chamber is not recommended for air-handling units serving VAV systems. For these systems, ventilation provided by a dedicated outdoor air system (DOAS) is preferable (see Chapter 12 for more discussion).

11.10.3 Automatic Dampers

Opposed blade dampers for the outdoor, return, and relief airstreams provides the highest degree of control. Section 11.10.7 on mixing plenums covers the conditions that dictate the use of parallel blade dampers.

Pressure relationships between various sections must be considered to ensure that automatic dampers are properly sized for wide open and modulating pressure drops.

11.10.4 Relief Openings

Relief openings in large buildings should be constructed similarly to outdoor air intakes, but they may require motorized or self-acting backdraft dampers to prevent high wind pressure or stack action from causing the airflow to reverse when the automatic dampers are open. The pressure loss through relief openings should be 0.10 in. w.g. or less unless they are provided with a relief fan. Low-leakage dampers, such as those for outdoor intakes, should always be used.

Relief dampers sized for the same air velocity as the maximum outdoor air dampers facilitate control when an air economizer cycle is used. The relief air opening should be located so that the exhaust air does not short-circuit to the outdoor air intake.

11.10.5 Return Air Dampers

The negative pressure in the outdoor air intake plenum is a function of the resistance or static pressure loss through the outdoor air louvers, damper, and duct. The positive pressure in the relief air plenum is, likewise, a function of the static pressure loss through the exhaust or relief damper, the exhaust duct between the plenum and outside, and the relief louver. The pressure drop through the return air damper must accommodate the pressure difference between the positive-pressure relief air plenum and the negative-pressure outdoor air intake plenum. Proper sizing of this damper facilitates both air balancing and mixing.

11.10.6 Outdoor Air Intakes

Resistance through outdoor intakes varies widely, depending on construction. Frequently, architectural considerations dictate the type and style of louver. The designer should ensure that the louvers selected offer minimum pressure loss, preferable not more than 0.10 in. w.g. High-efficiency, low-pressure louvers that effectively limit carryover of rain or snow are available. Flashing installed at the outside wall and weep holes or a floor drain will carry away rain and melted snow entering the intake. Cold regions may require a snow baffle to direct fine snow particles to a low-velocity area below the dampers. Outdoor dampers should be low-leakage types with special gasketed edges and special end treatment. **When mixing chambers are employed, separate damper sections with separate damper operators are strongly recommended for the minimum outdoor air needed for ventilation.** The maximum outdoor air needed for economizer cycles is then drawn through the outdoor air economizer damper.

11.10.7 Mixing Plenums

To achieve effective mixing, many designers prefer to mix in the ductwork some distance upstream of the unit inlet. If a mixing plenum is employed, careful consideration must be given to the objective of total mixing with no stratification under **all** conditions of operation.

If the equipment is alongside outdoor louvers in a wall, the minimum outdoor air damper should be located as close as possible to the return damper connection. An outdoor air damper sized for 1500 fpm usually provides acceptable control. The pressure difference between the relief plenum and outdoor intake plenum must be taken through the return damper section. A higher velocity through the return air damper—high enough to cause this loss at its full open position—enhances air balance and provides for better mixing. To create maximum turbulence and mixing, return air dampers should be set so that any deflection of air is toward the outdoor air. Parallel blade dampers may aid mixing.

When using a plenum or chamber, mixing dampers should be placed across the full width of the unit, even though the location of the return duct makes it more convenient to return air through the side. When return dampers are placed at one side, return air passes through one side of a double-inlet fan, and cold outdoor air passes through the other. If the air return must enter the side, some form of air blender or mixing device is necessary.

Although opposed blade dampers offer better control, properly proportioned parallel blade dampers are more effective for mixing airstreams of different temperatures. If parallel blades are used, each damper should be mounted so that its partially opened blades direct the airstreams toward the other damper for maximum mixing.

Baffles that direct the two airstreams to impinge on each other at right angles and in multiple jets create the turbulence required to mix the air thoroughly. In some instances, unit heaters or propeller fans have been used for mixing, regardless

of the final type and configuration of dampers. Otherwise, the preheat coil will waste heat, or the cooling coil may freeze. Low-leakage outdoor air dampers minimize leakage during shutdown.

Coil freezing can be a serious problem with chilled water, heating water, or steam coils. Full flow circulation of water during freezing weather, or even reduced flow with a small recirculating pump, discourages coil freezing. Further, it can provide a source of off-season chilled water in air-and-water systems. Antifreeze solutions or complete coil draining also prevent coil freezing. However, because it is difficult, if not impossible, to drain most cooling coils completely, caution should be exercised if this option is considered (see Chapter 13).

11.10.8 Filter Section

Control of air cleanliness depends heavily on the filter. Unless the filter is regularly maintained, system resistance is increased and airflow diminishes. Accessibility is a primary consideration in filter selection and location. In a built-up system, there should be a minimum of 3 ft between the upstream face of the filter bank and any obstruction. Other requirements for filters can be found in Chapters 29 and 30 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, and ASHRAE Standard 52.2.

Good mixing of outdoor and return air is also necessary for good filter performance. A poorly placed outdoor air duct or a poor duct connection to the mixing plenum can cause uneven loading of the filter and poor distribution of air through the coil section.

11.10.9 Preheat Coil

The preheat coil should have relatively wide fin spacing, be accessible for easy cleaning, and be protected by filters. If the preheat coil is located in the minimum outdoor airstream rather than in the mixed airstream as shown in Figure 11-1, it should not heat the air to an exit temperature above 35 to 45°F; preferably, it should become inoperative at outdoor temperatures of 45°F. Inner distributing tube or integral face and bypass coils are preferable with steam, and full steam pressure should be applied at entering air temperatures below 40°F. Hot water preheat coils should have a constant flow circulating pump and should be piped for parallel heat flow so that the coldest air will contact the warmest coil surface first.

11.10.10 Cooling Coil

In this section, sensible and latent heat are removed from the air. In many finned coils, some air passes through without contacting (i.e., being thermally affected by) the fins or tubes. The amount of this bypass can vary from 30% for a four-row coil at 700 fpm to less than 2% for an eight-row coil at 300 fpm.

The dew point of the air mixture leaving a four-row coil might satisfy a comfort installation with 25% or less outdoor air, a small internal latent load, and sensible temperature control only. For close control of room conditions for precision work, a deeper coil is required.

A central station unit that is the primary source of conditioned ventilation air for other subsystems, such as in an air-and-water system or for a VAV system, does not need to supply as much air to a space. In this case, the primary air furnishes outdoor air for ventilation and handles space dehumidification and some sensible cooling. This application normally requires deeper, cleanable coils with flat fins and sometimes utilizes sprays. (See DOAS units in Chapter 12.) To prevent water carryover and allow proper cleaning and dew-point temperature control, fin spacing should be 0.125 in. minimum with flat fins and a minimum depth of six rows.

11.10.11 Reheat Coil Section

Reheat is discouraged, unless it is required to satisfy humidity control requirements. For energy conservation purposes, consideration may be given to using some form of recovered energy for a reheat source.

Heating coils located in the reheat position, as shown in Figure 11-1, in addition to being used for temperature control purposes, are frequently used for warm-up.

Hot water heating coils provide the highest degree of control. Oversized coils, particularly steam, can stratify the airflow; thus, where cost-effective, inner distributing coils are preferable for steam applications. Electric coils may also be used.

11.10.12 Humidifiers

For comfort installations not requiring close control, moisture can be added to the air by mechanical atomizers or point-of-use electric or ultrasonic humidifiers. Proper location of this equipment prevents stratification of moist air in the system.

Steam grid humidifiers with dew-point control often are used for humidity control. In this application, the heat of evaporation should be replaced by heating the recirculated water rather than by increasing the size of the preheat coil. It is not possible, of course, to add moisture to saturated air, even with a steam grid humidifier. Air in a laboratory or other application that requires close humidity control must be reheated after leaving a dehumidifier coil before moisture can be added. This requires reconsideration of air discharge temperatures and quantities.

The capacity of the humidifying equipment should not exceed the expected peak load by more than 10%. If the humidity is controlled from a sensor in the room or the return air, a limiting humidistat and fan interlock may be needed in the supply duct. This prevents condensation and mold or mildew growth within the ductwork when temperature controls call for cooler air. Many humidifiers add some sensible heat that should be accounted for in the psychrometric analysis.

It is quite difficult to prevent a steam humidifier from supersaturating the airstream, which, of course, can result in moisture in the air-handling unit, ductwork, or the space. This moisture can then become a source of mold or mildew. A preferred

method of adding moisture to the air is with an evaporating matt or a sprayed coil in the outdoor airstream, downstream of a preheat coil (see Chapter 2).

11.10.13 Supply Air Fan

Axial flow, centrifugal, or plug fans may be chosen as supply air fans for straight-through flow applications. In factory-fabricated units, more than one centrifugal fan may be tied to the same shaft. If headroom permits, a single-inlet fan should be chosen when air enters at right angles to the flow of air through the equipment. This permits a more gradual transition from the fan to the duct and increases the static regain in the velocity pressure conversion.

To minimize inlet losses, the distance between the casing walls and the fan inlet should be at least equal to the diameter of the fan wheel. With a single-inlet fan, the length of the transition section should be at least half the width or height of the casing, whichever is longer.

If fans blow through the equipment, the air distribution through the downstream components needs analyzing, and baffles should be used to ensure uniform air distribution. See Chapter 21 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

11.11 Air Distribution

11.11.1 Ductwork

Ductwork should deliver conditioned air to an area as directly, quietly, and economically as possible. Structural features of the building generally require some compromise and often limit depth. Chapter 9 describes ductwork design in detail and gives several methods of sizing duct systems.

It is imperative that the designer coordinate duct design with the structure. In commercially developed projects, it is common that great effort is made to reduce floor-to-floor dimensions. The resultant decrease in the available interstitial space left for ductwork is a major design challenge.

11.11.2 Room Terminals

In some instances, such as in single-zone, all-air systems, the air may enter from the supply air ductwork directly into the conditioned space through a grille or diffuser.

In multiple zoned air systems, an intermediate device controls air temperature and/or volume. Various devices are available, including (1) an air-water induction terminal, which includes a coil or coils in the induced airstream to condition the return air before it mixes with the primary air and enters the space; (2) an all-air induction terminal, which controls the volume of primary air, induces ceiling plenum air, and distributes the mixture through low-velocity ductwork to the space; (3) a fan-powered mixing box, which uses a fan to accomplish the mixing rather than depending on the induction principle; (4) a VAV box, which varies the amount of air delivered with no induction; (5) a VAV reheat terminal; (6) a variable volume, constant velocity diffuser; (7) a double-duct mixing box; and (8) a terminal reheat device.

11.11.3 Insulation

In all new construction (except low-rise residential buildings), air-handling duct and plenums installed as part of an HVAC air distribution system should be thermally insulated in accordance with ASHRAE Standard 90.1. See Chapter 23 of the 2017 *ASHRAE Handbook—Fundamentals* for further discussion and calculation methodology.

11.11.4 Ceiling Plenums

The space between a hung ceiling and the floor slab above is used frequently as a return plenum to reduce sheet metal work and remove heat from the plenum. Local and national codes should be consulted before using this approach in new design, because most codes prohibit combustible material in a return air ceiling plenum.

Entrance lobby ceilings with lay-in panels do not work well as return plenums where negative pressures from high-rise elevators or stack effects of high-rise buildings may occur. If the plenum leaks to the low-pressure area, the tiles may lift and drop out when the outside door is opened and closed. Return plenums directly below a roof deck have substantially higher return air heat gain or losses than a ducted return, which has the advantage of reducing the heat gain to or loss from the space.

11.11.5 Controls

Controls should be automatic and *simple* for best operating and maintenance effectiveness. Operations should follow a natural sequence—depending on the space need, one controlling thermostat closes a normally open heating valve, opens the outdoor air mixing dampers, or opens the cooling valve. In some applications, an enthalpy controller, which compares the heat content of outdoor air to that of return air, then opens the outdoor air damper when enthalpy of the outdoors is less than return air or space enthalpy and thus reduces the refrigeration load. On other systems, a dry-bulb control saves the cost of the enthalpy control and approaches these savings when an optimum changeover temperature, near the design dew point, is established.

A minimum outdoor air damper with separate motor, selected for a velocity of 1500 fpm, is preferred to one large outdoor air damper with minimum stops. A separate damper simplifies air balancing.

For control system fundamentals see Chapter 7, 2017 *ASHRAE Handbook—Fundamentals*, and Chapter 47, 2015 *ASHRAE Handbook—HVAC Applications*.

11.11.6 Vibration Isolation

Vibration and sound isolation equipment is required for most central system fan installations. Standard mountings of fiberglass, ribbed rubber, neoprene mounts, and springs are available for both fans and prefabricated units. If the fan manufacturer supplies the vibration isolators, it is common practice to supply neoprene pads for speeds above 1200 rpm,

rubber-in-shear isolators (0.5 in. deflection) for speeds between 700 and 1200 rpm, and springs with 1 in. deflection for speeds below 700 rpm.

Special consideration is required in the isolation of rotating machinery such as fans and pumps when they are equipped with variable-speed drives because of the relationship between frequency (i.e., rotational speed) and the effectiveness of vibration isolation devices.

The transmissibility of vibration isolators is defined as the ratio of the transmitted force to the impressed force, or

$$\mathrm{TR} = \frac{\text{Transmitted force}}{\text{Impressed force}} \tag{11-10}$$

Thus, the smaller the transmissibility, the more effective is the isolator. It is common practice to select vibration isolators with a transmissibility between 0.10 and 0.05.

The relationship of the transmissibility to the system frequency is

$$\mathrm{TR} = \frac{1}{(f/f_n)^2 - 1} \tag{11-11}$$

where

TR = transmissibility
f = frequency of the rotating mass, Hz
f_n = natural frequency of isolator system, Hz

Then, the frequency of the rotating mass in Hz is

$$f = N/60 \tag{11-12}$$

where

N = rotational speed, rpm
60 = seconds per minute

and

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{y}} \tag{11-13}$$

where

g = gravitational constant, 386 in./s^2
y = static deflection of isolator, in.

The solution to Equation (11-11) for rotating fan speeds of 0 to 1000 rpm and static deflections of 0 to 3 in. is plotted in Figure 11-2 for transmissibility of 0.05, 0.10, 1, and ∞ The graph demonstrates that for a given static deflection as the speed is decreased, the transmissibility increases to the point where the isolator loses all of its effectiveness when $f/f_n = \sqrt{2}$, below which the isolator becomes an amplifier. Thus, extra care should be taken when selecting and designing isolators for a variable-speed device. One method of ensuring effective vibration control is to design the isolator at the minimum speed at which the system will operate rather than at the design or maximum speed.

Ductwork connections should be made with fireproof fiber cloth sleeves having considerable slack, but without offset between the fan outlet and rigid duct. Misalignment between the duct and fan outlet can cause turbulence, generate noise, and reduce system efficiency. Electrical and piping connections to vibration-isolated equipment should be made with flexible conduit and flexible pipe connections. Special considerations are required in seismic zones.

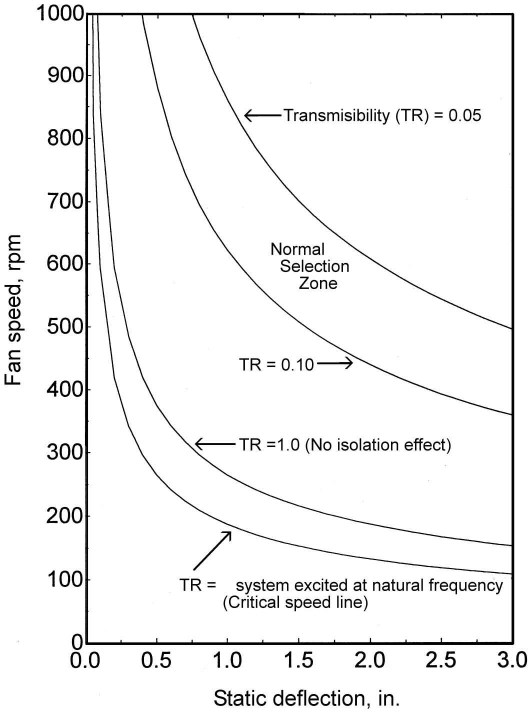


Fig. 11-2 Static Deflection of Isolator Versus Fan Speed

Equipment noise transmitted through the ductwork can be reduced by sound-absorbing units, acoustical lining, and other means of attenuation. Sound transmitted through the return and relief ducts should not be overlooked. Acoustical lining sufficient to adequately attenuate any objectionable system noise or locally generated noise should be considered. Chapter 48 of the 2015 *ASHRAE Handbook—HVAC Applications*, Chapter 8 of the 2017 *ASHRAE Handbook—Fundamentals*, and ASHRAE Standard 68 have further information on sound and vibration control.

The designer must account for seismic restraint requirements for the seismic zone in which the particular project is located.

Example 11-3 A fan is selected to be operated at 720 rpm at design conditions and is supplied with a spring vibration isolator with 1 in. static deflection.

(a) What will be the transmissibility of the fan with the selected isolator mount?

Solution:

$$f = N/60 = 720/60 = 12\ \text{Hz}$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{y}} = \frac{1}{2\pi}\sqrt{\frac{386}{1}} = 3.13\ \text{Hz}$$

$$\text{TR} = \frac{1}{(f/f_n)^2 - 1} = \frac{1}{(12/3.13)^2 - 1} = 0.07$$

(b) If the fan is driven by a variable-speed drive controlled to reduce the speed to 37% of the design value, what will the transmissibility be at the minimum speed?

$$N_b = 0.37(720) = 266.4 \text{ rpm}$$

$$f = 266.4/60 = 4.44 \text{ Hz}$$

$$\text{TR} = \frac{1}{(4.44/3.13)^2 - 1} = 1$$

The isolator would provide no isolation.

11.12 Space Heating

Although steam is an acceptable medium for central system preheat or reheat coils, low-temperature hot water provides a much more uniform means of perimeter and other heating devices located within the space. Individual automatic control of each terminal provides the ideal space comfort. A control system that varies the water temperature inversely with the change in outdoor air temperature provides water temperatures that produce acceptable results without individual room or space control in limited applications. To produce the best results, the most satisfactory ratio of indoor air temperature to that outdoors can be set after the installation is completed and actual operating conditions are ascertained.

Multiple perimeter spaces on one exposure served by a central system may be heated by supplying warm air from the central system. Areas that have heat gain from lights and occupants and no heat loss require cooling in winter as well as in summer, as do perimeter areas in which solar gains in winter combined with internal gains can exceed transmission losses.

In systems with mixing chambers, little or no heating of the return and outdoor air is required when the space is occupied. Local codes dictate the amount of outdoor air required, which is generally based on the requirements of ASHRAE Standard 62. For example, with return air at 75°F and outdoor air at 0°F, the temperature of a 25% outdoor/75% return air mixture would be 56°F, which is close to the temperature of the air supplied to cool such a space in summer. In this instance, a preheat coil installed in the minimum outdoor airstream to warm the outdoor air can produce overheating, unless it is sized as previously recommended. Assuming good mixing, a preheat coil located in the mixed airstream prevents this problem. The outdoor air damper should be kept closed until room temperatures are reached during warm-up. A return air thermostat can terminate the warm-up period.

When a central air-handling unit supplies both perimeter and interior spaces, the supply air must be cool to handle the interior zones. Additional control is needed to heat the perimeter spaces properly. Reheating the air is the simplest solution, but it is not acceptable by some energy codes. An acceptable solution is to vary the volume of air to the perimeter and combine it with a terminal heating coil or a separate perimeter heating system, either baseboard, overhead air heating, or a fan-powered mixing box with supplemental heat. The perimeter heating should be individually controlled and integrated with the cooling control.

11.13 Primary Systems

The type of central heating and cooling equipment used for air-conditioning systems in large buildings depends chiefly on economic factors, once the total required capacity has been determined. Component choice depends on such factors as the type of fuel available, environmental protection required, structural support, and available space.

Rising energy costs have fostered many designs to recover the internal heat from lights, people, and equipment to reduce the size of the heating plant. Chapters 9 and 26 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* describe several heat recovery devices. Also, see Chapter 16 of this book.

The search for energy savings has extended to cogeneration or total energy systems in which on-site power generation has been added to the heating and air-conditioning project. The economics of this function is determined by gas and electric rate differentials, investment cost of the plant, and by the ratio and coincidence of electric to heat demands for the project. In these systems, reject heat from the prime movers can be put into the heating system and the cooling equipment, either to drive the turbines of centrifugal compressors or to serve absorption chillers. Chapter 16 of this book and Chapter 7 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* present further details on cogeneration or total energy systems.

Among the largest installations of central mechanical equipment are the central cooling and heating plants serving groups of large buildings. These plants provide higher diversity and sometimes operate more efficiently and with lower maintenance and labor costs than individual plants.

The economics of these systems requires extensive analysis. Boilers, gas and steam turbine-driven centrifugals, and absorption chillers may be installed in combination in one plant. In large buildings with core areas that require cooling while perimeter areas require heating, one of several heat reclaim systems could heat the perimeter to save energy. Chapter 7 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* gives details of these combinations, and Chapters 11, 12, and 15 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* give design details of central plants. Also see Chapter 13.

Most large buildings, however, have their own central heating and cooling plant in which the choice of equipment depends on the following:

- Required capacity and type of usage
- Costs and types of energy available
- Location of the equipment room
- Type of air distribution system(s)
- Owning and operating costs

Many electric utilities impose severe penalties for peak summertime power use or, alternatively, offer incentives for off-peak use. This policy has renewed interest in both water and ice thermal storage systems. The storage capacity installed for summertime load leveling may also be available for use in the winter, making heat reclaim and storage a more viable option. With ice-storage, the low-temperature ice water can provide colder air than that available from a conventional system. Use of high water temperature rise and lower temperature air results in lower pumping and fan energy and, in some instances, offsets the energy penalty due to the lower refrigeration temperature required to make ice.

11.13.1 Heating Equipment

Steam and hot water boilers for heating are manufactured for high or low pressure and use gas, oil, or electricity and, sometimes, coal or waste material for fuel. Low-pressure boilers are rated for a working pressure of 15 psig for steam and 160 psig for water, with a maximum temperature limitation of 250°F. Package boilers, with all components and controls assembled as a unit, are available. Electrode or resistance-type electric boilers are available for either hot water or steam generation. Boilers and furnaces designed for higher efficiencies have the ability to condense the water vapor in the combustion chamber, but the boilers must operate with water temperatures much lower than noncondensing boilers, usually limited to about 125°F. For further information, see Chapter 19.

Where steam or hot water is supplied from a central plant, as on some university campuses and in downtown areas of large cities, the service entrance to the building must conform to utility standards. The utility should be contacted at the beginning of the project to determine availability, cost, and the specific requirements of the service.

11.13.2 Fuels

Chapter 19 gives fuel types, properties, and proper combustion factors and includes information for the design, selection, and operation of fuel selection and automatic fuel-burning equipment.

11.13.3 Refrigeration Equipment

The major types of refrigeration equipment used in large systems are reciprocating compressors, helical rotary compressors, screw rotary compressors, centrifugal compressors, and absorption chillers. See Chapter 18 for further discussion of refrigeration equipment, including the general size ranges of available equipment.

Reciprocating, helical or screw rotary, and centrifugal compressors are usually driven by electric motors; however, they can, and sometimes are, driven by natural gas or diesel engines and gas and steam turbines. The compressors may be purchased as part of a refrigeration chiller consisting of compressor, drive, chiller-evaporator, condenser, and necessary safety and operating controls.

Reciprocating and helical or screw rotary compressor units are frequently used in packaged as well as in field-assembled systems, with air-cooled or evaporative condensers arranged for remote installation. Centrifugal compressors are used usually only in packaged chillers.

Absorption chillers are water cooled. Most use a lithium bromide/water or water/ammonia cycle and are generally available in the following four configurations: (1) direct fired, (2) indirect fired by low-pressure steam or hot water, (3) indirect fired by high-pressure steam or hot water, and (4) fired by hot exhaust gas. Small direct-fired chillers are single-effect machines with capacities of 3.5 to 25 tons. Larger indirect-fired, single-effect and double-effect chillers in the 100 to 2000 ton capacity range are available.

Low-pressure steam at 12 psig or hot water heats the generator of single-effect absorption chillers with capacities from 50 to 1600 tons. Double-effect machines use higher pressure steam up to 150 psig or hot water at an equivalent temperature (365°F). Absorption chillers of this type are available from 350 to 2000 tons.

The absorption chiller is sometimes combined with steam turbine-driven centrifugal compressors in large installations. Steam from the noncondensing turbine is piped to the generator of the absorption machine. When a centrifugal chiller is driven by a gas turbine or an engine, an absorption machine generator may be fed with steam or hot water from the jacket and/or an exhaust gas heat exchanger.

11.13.4 Cooling Towers

Water is usually cooled by the atmosphere for use in water-cooled condensers. Either natural draft or mechanical draft cooling towers or spray ponds are used for the cooling. Of these, the mechanical draft tower, which may be of the forced draft, induced draft, or ejector type, is used for most designs because it does not depend on the wind. Air-conditioning systems use towers ranging from small package units of 5 to 500 tons or field-erected towers with multiple cells in virtually unlimited sizes.

The tower must be winterized if required for operation at ambient outdoor dry-bulb temperatures below 35°F. Winterizing includes the capability of bypassing water directly into the tower return line (either automatically or manually, depending on the installation) and of heating the tower pan water to a temperature above freezing. (See Chapter 14 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.)

Heat may be added by steam or hot water coils or electric resistance heaters in the tower basin. Also, it is often necessary to provide an electric heating cable on the condenser water and makeup water pipes and to insulate these heat-traced sections to prevent the pipes from freezing. Special controls are required when it is necessary to operate the cooling tower at or near freezing conditions.

Where the cooling tower will not operate in freezing weather, provisions for draining the tower and piping are necessary. Draining is the most effective way to prevent the tower and piping from freezing.

Careful attention must also be given to water treatment to keep maintenance required in the refrigeration machine absorbers and/or condensers to a minimum.

Cooling towers may also be used as a cooling source for the building in low-temperature seasons by filtering and directly circulating the condenser water through the chilled water circuit, by cooling the chilled water with the cooling tower water in a separate heat exchanger, or by using the heat exchangers in the refrigeration equipment to produce thermal cooling. Towers are usually selected in multiples so that they may be run at reduced capacity and shut down for maintenance in cool weather. Chapter 18 includes further design and application details.

11.13.5 Air-Cooled Condensers

Air-cooled condensers pass outdoor air over a dry coil to condense the refrigerant. This results in a higher condensing temperature and, thus, a larger power input at peak condition; however, over the year this peak time may be relatively short. The air-cooled condenser is popular in small reciprocating and helical or screw rotary systems because of its low first cost and lower cost maintenance requirements.

Recent emphasis on energy efficiency and water conservation have led to the concept of hybrid air/water-cooled refrigeration plants that use water-cooled chillers for peak loads in warm-weather months, and smaller air-cooled units for cold-weather operation.

11.13.6 Evaporative Condensers

Evaporative condensers pass air over refrigerant condensing coils sprayed with water, thus taking advantage of adiabatic saturation to lower the condensing temperature. As with the cooling tower, freeze prevention and close control of water treatment are required for successful operation. The lower power consumption of the refrigeration system and the much smaller footprint from the use of the evaporative versus the air-cooled condenser are gained at the expense of the cost of the water used and increased maintenance costs.

11.13.7 Pumps

Pumps used in heating and air-conditioning systems are usually centrifugal pumps. Pump configurations include horizontal split case with a double-suction impeller, or end suction pumps, either close-coupled or flexibly connected.

Major applications for pumps in the equipment room are as follows:

- Chilled water
- Heating water
- Condenser water
- Steam condensate pumps
- Boiler feed pumps
- Fuel oil

When the pumps handle hot liquids or have high inlet pressure drops, the required net positive suction head (NPSH) must not exceed the NPSH available at the pump. It is common practice to provide spare pumps or spare critical components to maintain system continuity in case of a pump failure. See Chapter 13 and Chapters 14 and 44 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, for more design information on pumps and pumping systems. Chapters 9 in this book and 13 also discuss pump selection.

11.13.8 Piping

Air-conditioning piping systems can be divided into two parts: the piping in the main equipment room and the piping required to deliver heat or chilled water to the air-handling equipment and terminal devices throughout the building. The air-handling system piping follows procedures detailed in Chapters 9 and 13.

The major piping in the main equipment room includes fuel lines, refrigerant piping, and steam and water connections. Chapter 11 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* includes more information on piping for steam systems, and Chapter 13 of the same volume has information on heating and chilled and dual-temperature water systems, as does Chapter 14. Chapter 22 in the 2017 *ASHRAE Handbook—Fundamentals* presents data on sizing steam, hydronic, fuel oil, natural gas, and service water piping.

11.13.9 Instrumentation

All equipment must have adequate gages, thermometers, flow-meters, balancing devices, and dampers for effective system testing, balancing, monitoring, commissioning, and operations. In addition, capped thermometer wells, gage cocks, plugged duct openings, volume dampers, and access/inspection, maintenance doors or ports should be installed at strategic points for inspection service and system balancing. Chapter 38 of the 2015 *ASHRAE Handbook—HVAC Applications* indicates the locations and types of fittings required.

A central control console or personal computer to monitor the many system points should be considered for any large, air-conditioning system. A computer terminal permits a single operator or building manager to monitor and perform functions at any point in the building to increase occupant comfort and to free maintenance staff for other duties. Chapter 47 of the 2015 *ASHRAE Handbook—HVAC Applications* describes these systems in detail.

11.14 Space Requirements

In the initial phases of building design, the engineer seldom has sufficient information to finalize the designs. Therefore, most experienced engineers have evaluation criteria to estimate the building space needed.

The air-conditioning system selected, the building configuration, and other variables govern the space required for the mechanical system. The final design is usually a compromise between what the engineer recommends and what the architect can accommodate. Where designers cannot negotiate a space large enough to suit the installation, the judgment of the owners should be called upon.

Although few buildings are identical in design and concept, some basic criteria are applicable to most buildings and help allocate space that approximates the final requirements.

These space requirements are often expressed as a percentage of the total building floor area.

11.14.1 Mechanical, Electrical, and Plumbing Facilities

The total mechanical and electrical space requirements range from 4 to 9% of the gross building area, the majority of buildings falling within the 6 to 9% range.

Most of the facilities should be centrally located to minimize long duct, pipe, and conduit runs and sizes; simplify shaft layouts; and centralize maintenance and operation. A central location also reduces pump and fan motor power, which may reduce building operating costs. But for many reasons, it is often impossible to centrally locate all the mechanical, electrical, and plumbing facilities within the building. In any case, the equipment should be located to minimize space requirements, centralize maintenance and operation, and simplify the electrical system.

Equipment rooms generally require clear ceiling height ranging from 12 to 20 ft, depending on equipment sizes and the complexity of ductwork, piping, and conduit.

The main electrical transformer and switchgear rooms should be located as close to the incoming electrical service as practical. If there is an emergency generator, it should be located considering (1) proximity to emergency electrical loads and sources of combustion and cooling air, (2) ease of properly venting exhaust gases to the outdoors, and (3) provisions for noise control.

The main plumbing equipment room usually contains gas and domestic water meters, the domestic hot water system, the fire protection system, and various other elements such as compressed air, special gases, and vacuum, ejector, and sump pump systems. Some water and gas utilities require a remote outdoor meter location.

The heating and air-conditioning equipment room houses the boiler or pressure-reducing station, or both; the refrigeration machines, including the chilled water and condensing water pumps; converters for furnishing hot or cold water for air conditioning; control air compressors; steam condensate pumps; and other miscellaneous equipment. Local codes and ASHRAE *Standard 15* should be consulted for special equipment room requirements.

In high-rise buildings, it is often economical to locate the refrigeration plant at the top or intermediate floors, or in a roof-level penthouse. The electrical service and structural costs will rise, but these may be offset by reducing costs for condenser and chilled water piping, energy consumption, and equipment because of the lower operating pressure. The boiler plant may also be placed at roof level, which eliminates the need for a chimney through the building.

Gas fuel may be more desirable because oil storage and pumping present added design and operating problems, and the cost of oil leak detection and prevention may be substantial. Heat recovery systems, in conjunction with the refrigeration equipment, can appreciably reduce the heating plant size in buildings with large core areas.

Additional space may be needed for a telephone terminal room, a pneumatic tube equipment room, an incinerator, or other equipment.

Many buildings, especially larger ones, need cooling towers, which often present problems. If the cooling tower is on the ground, it should be at least 100 ft away from the building for two reasons: (1) to reduce tower noise in the building and (2) to keep cool season discharge air from fogging the building's windows. Towers must also be located so as to prevent any possibility of tower discharge being mixed into the ventilation air intakes, to prevent the possibility of *Legionella* entering the building. Towers should be kept the same distance from parking lots to avoid staining car finishes with water treatment chemicals. When the tower is on the roof, its vibration and noise must be isolated from the building. Some towers are less noisy than others, and some have attenuation housings to reduce noise levels. These options should be explored before selecting a tower.

The bottom of roof-mounted towers, especially larger ones, must be set on a steel frame 4 to 5 ft above the roof to allow room for piping and proper tower and roof maintenance. Pumps below the tower should be designed for adequate net positive suction head, but they must be installed to prevent the draining of the piping on shutdown.

11.14.2 Fan Rooms and Rooftop Units

Fan rooms should preferably be located as close as possible to the space to be conditioned to reduce both installation cost, fan power, and fan energy.

The number of fan rooms depends largely on the geometry of the buildings and use schedules of the spaces. Buildings with large floor areas often have multiple fan rooms on each floor. Many high-rise buildings, however, may have one fan room serving 10 to 20 floors—one serving the lower floors, one serving the middle of the building, and one at the roof level serving the top portion of the building.

Life safety is a very important factor in fan room location. Chapter 53 of the 2015 *ASHRAE Handbook—HVAC Applications* discusses principles of fire spread and smoke management.

It is very common, particularly on smaller and low-rise buildings, to locate air-handling units or packaged HVAC equipment on the roof instead of in a fan room or mechanical equipment room. Many units are designed for rooftop installation and are furnished with a roof curb for mounting. However, it is the responsibility of the system designer to select machinery that will withstand the severity of the climate and that can be easily and conveniently accessed for normal maintenance and emergency service and to provide wearing surfaces on the roof for the maintenance and service personnel, tools, and replacement components.

11.14.3 Interior Shafts

In tall buildings, interior shaft space is necessary to accommodate return and exhaust air; interior supply air and hot, chilled, and condenser water piping; steam and return piping;

electrical closets; telephone closets; plumbing piping; and, possibly, pneumatic tubes and conveyer systems.

The shafts must be clear of stairs, elevators, and structural beams on at least two sides to allow maximum headroom when the pipes and ducts come out above the ceiling. In general, duct shafts having an aspect ratio of 2:1 to 4:1 are easier to develop than large square shafts. The rectangular shape also makes it easier to go from the equipment in the fan rooms to the shafts.

The size, number, and location of shafts is important in multi-story buildings. Vertical duct distribution systems with little horizontal branch ductwork are desirable because they are usually less costly; easier to balance; create less conflict with pipes, beams, and lights; and enable the architect to design lower floor-to-floor heights.

The number of shafts is a function of building geometry, but, in larger buildings, it is usually more economical in cost and space to have several smaller shafts rather than one large shaft. Separate supply and exhaust duct shafts may be desired to reduce the number of duct crossovers. **From 10% to 15% additional shaft space should be allowed for future expansion and modifications.** The additional space also reduces the initial installation cost.

11.14.4 Equipment Access

Properly designed mechanical equipment rooms must allow for the movement of large, heavy equipment in, out, and within the building. Equipment replacement and maintenance can be very costly if access is not planned properly. Many designers (and some machinery codes) require a minimum of 36 to 48 in. clear aisles between all machines and equipment.

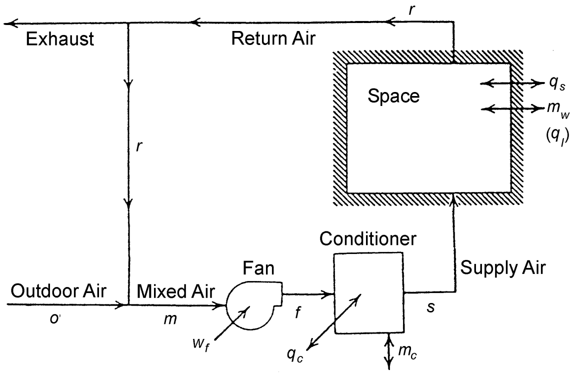
Because systems vary greatly, it is difficult to estimate space requirements for refrigeration and boiler rooms without making block layouts of the system selected. Block layouts allow the engineer to develop the most efficient arrangement of the equipment, with adequate access and serviceability. Block layouts can also be used in preliminary discussions with the owner and architect. Only then can the engineer obtain verification of the estimates and provide a workable and economical design.

Example 11-4 A new law office building is being designed for St. Louis, Missouri, to use the HVAC system shown in the sketch below. Minimum outdoor air of 15 cfm/person for meeting the ventilation requirements of the anticipated 125 occupants will be maintained throughout the year. During summer operation, the cooling coil supplies air to the conditioned space at 55°F, 90% relative humidity. The space summer design loads and designs conditions are

323,000 Btu/h sensible (gain);
89,000 Btu/h latent (gain)
Outdoor: 94°F db, 75°F wb; Indoor: 78°F

The 55% efficient fan must produce a 3 in. w.g. pressure increase to overcome the friction in the duct system. Determine:

(a) Size of cooling coil, ft^2 of face area
(b) Rating of chiller unit, Btu/h
(c) Fan motor size, hp
(d) Outdoor air damper face area, ft^2



Solution:

$$m_w = 89000/1100 = 80.9 \text{ lb/h}$$

$$m_a = \frac{323000}{0.244(78-55)} = 57555 \text{ lb/h}$$

$$m_o = \frac{125 \times 15 \times 60}{13.33} = 8440 \text{ lb/h}$$

$$m_r = 57555 - 8440 = 49115 \text{ lb/h}$$

$$W_r = W_s + \frac{m_w}{m_s} = 0.0083 + \frac{80.9}{57555} = 0.00971$$

$$W_m = \frac{8440(0.0144) + 49115(0.00971)}{57555} = 0.0104 \text{ lb/lb}$$

$$h_m = \frac{8440(38.5) + 49115(29.4)}{57555} = 30.7 \text{ Btu/lb}$$

$$t_m = 80.4°\text{F};\ \phi = 48\%$$

$$h_f = h_m + w_f = 30.7 + 0.49 = 31.2 \text{ Btu/lb}$$

$$W_f = 0.0104 \text{ lb/lb}$$

$$Q = \frac{57555(13.33)}{60} = 12800 \text{ cfm at std. conditions}$$

Coil Size (select 400 fpm face velocity)

$$A_F = 12800/400 = 32 \text{ ft}^2$$

Chiller Size

$$m_a[h_f - h_s - (W_f - W_s)h_c] + Q_c = 0$$

$$57555[31.2 - 22.3 - (0.0104 - 0.0083)23] + Q_c = 0$$

$$Q_c = -509{,}000 \text{ Btu/h}$$

$$Q_{\text{chiller}} \approx Q_{\text{coil}} = 509{,}000 \text{ Btu/h}$$

Fan Motor Size

$$P_f = \frac{57555(0.49)}{2545} = 11 \text{ hp}$$

Outdoor Air Damper Size

$$\text{OA} = 125 \times 15 = 1875 \text{ cfm}$$

If the recommended face velocity is 1500 fpm:

$$A_D = 1875/1500 = 1.25 \text{ ft}^2$$

11.15 Problems

11.1 A room is to be cooled to a temperature of 75°F and a relative humidity of 50%. If there is negligible latent load within the space, what is the highest temperature at which the conditioned air can be supplied (t_s)? Why?

11.2 A room has a total space cooling load of 20 tons and a sensible heat ratio of 0.90. If the conditioned air is to be supplied at 20°F less than the room temperature, how much air must be circulated?

11.3 What are the four generic types of air systems expressed by thermodynamic methods?

11.4 What are the 18 fundamental parameters that must be addressed in the selection and design of an HVAC system?

11.5 Before designing a system, the cooling and heating load for each room in a building must be calculated? Why?

11.6 If outdoor air at 95°F dry bulb and 78°F wet bulb is cooled to 75°F dry bulb without any dehumidification, what will the relative humidity be?

11.7 In the air-handling unit of Figure 11-1, under design conditions, the outdoor air temperature is 95°F dry bulb and 78°F wet bulb and the space temperature is 75°F and 50% rh. The supply fan handles 60,000 cfm of air at 55°F saturated (entering the fan).

If the minimum outdoor air dampers are sized for 6000 cfm of ventilation air, what is the state point (dry-bulb and wet-bulb temperatures) of the mixed air?

11.8 A constant-flow air-handling system is designed to circulate 60,000 cfm of air at a total fan pressure rise of 6 in. w.g. The system is designed to operate continuously. The fan efficiency is 70% and the motor efficiency is 90%.

(a) How much power (hp) is required to drive the fan?
(b) What will be the annual fan energy consumption?

11.9 If, in the above problem, the sensible space load were reduced by 25% by using a more energy effective building envelope and improved lighting system, and this change were accommodated by reducing the air flow rate at the same fan efficiencies, what would be the reduction in annual fan energy?

11.10 It is desired to transfer a given quantity of heat energy from one location to another location in a building. Two methods being considered are either by an air system operating at 4 in. w.g. total pressure or by a water system with a pump head of 40 ft. Calculate the ratio of fan power required for an air system to pump power required for a water system with the following system variables:

Fan efficiency	70%
Pump efficiency	80%
Air Δt	20°F
Water Δt	40°F

11.11 A fan with a variable-speed drive is selected to operate at 900 rpm, and it is installed on a spring isolator mount with 1 in. static deflection. Determine:

(a) Transmissibility of the isolator
(b) Minimum speed that the unit can be operated at before the transmissibility is 0.50

11.12 Specify typical temperatures for the following:

(a) Air leaving a gas-fired warm air furnace
(b) Air leaving a heat pump condenser
(c) Air leaving the cooling coil of a residential air conditioner
(d) Air leaving the cooling coil of a commercial air conditioner
(e) Hot water entering the convectors (radiators) of a hydronic system
(f) Hot water returning to the boiler from the convectors

11.13 An air-conditioned room has a sensible cooling load of 200,000 Btu/h, a latent cooling load of 50,000 Btu/h, an occupancy of 20 people, and is maintained at 76°F dry bulb and 64°F wet bulb. Twenty-five percent of the air entering the room is vented through cracks and hoods. Outdoor air is assumed to be at design conditions of 95°F dry bulb and 76°F wet bulb. Conditioned air leaves the apparatus and enters the room at 60°F dry bulb.

Use the following letters to designate state points:

A Outside design conditions
B Inside design conditions
C Air entering apparatus (mixed air temperature)
D Air entering room (supply air temperature)

(a) Complete the following table:

Point	Dry Bulb	Wet Bulb	h	W
A				
B				
C				
D				

(b) Calculate the room SHR.
(c) What air quantity must enter the room?
(d) What is the apparatus load in tons?
(e) What is the load of the outdoor air? In lb per hour? In cfm?
(f) Does the room load plus the outdoor air load equal the coil load?

11.14 A space has a sensible heat loss of 60,000 Btu/h and a latent loss of 20,000 Btu/h. The space is to be maintained at 70°F and 40% rh. The air that passes through the conditioner is 90% recirculated and 10% outdoor air at 40°F and 20% rh. The conditioner consists of an adiabatic saturator and a heating coil. Estimate the temperature and humidity ratio of the air entering the conditioned space. What is the flow rate in cfm? How much heat is added by the coil to the air in Btu/h? How much water is added to the air by the adiabatic saturator (lb/h)?

11.15 Air at 800 ft^3/min leaves a residential air conditioner at 65°F with 40% rh. The return air from the rooms has average dry- and wet-bulb temperatures of 75°F and 65°F, respectively. Determine:

(a) Size of the unit in tons (12,000 Btu/h = 1 ton)
(b) Rate of dehumidification
[Ans: 2.58 tons, 20.16 lb/h (9.1 kW, 2.54 g/s)]

11.16 In an air-conditioning unit 6000 cfm at 80°F dry bulb, 60% rh, and standard atmospheric pressure, enter the unit. The leaving condition of the air is 57°F dry bulb and 90% rh. Calculate:

(a) Cooling capacity of the air-conditioning unit, tons
(b) Rate of water removal from the unit, lb/h
(c) Sensible heat load on the conditioner, Btu/h
(d) Latent heat load on the conditioner, Btu/h
(e) Dew point of the air leaving the conditioner, °F

11.17 A space in an industrial building has a winter sensible heat loss of 200,000 Btu/h and a negligible latent heat load (latent losses to outside are made up by latent gains within the space). The space is to be maintained at 75°F and 50% rh. Due to the nature of the process, 100% outdoor air is required for ventilation. The outdoor air conditions can be taken as saturated air at 20°F. The amount of ventilation air required is 7000 cfm and the air is to be preheated, humidified with an adiabatic saturator, and then reheated. The temperature out of the adiabatic saturator is to be maintained at 60°F dry bulb. Calculate the following:

(a) Temperature of the air entering the preheater
(b) Temperature of the air entering the space to be heated
(c) Heat supplied to preheat coil, Btu/h
(d) Heat supplied to reheat coil, Btu/h
(e) Quantity of makeup water added to adiabatic saturator, gpm
(f) Temperature of the spray water
(g) Show the processes and label points on the psychrometric diagram

11.18 In winter, a meeting room with a large window is to be maintained at comfort conditions. The inside glass temperature on the design day is 40°F. Condensation on the window is highly undesirable. The room is to accommodate 18 adult males [250 Btu/h (sensible) and 200 Btu/h (latent) per person]. The heat loss through the walls, ceiling, and floor is 33,600 Btu/h. There are 640 watts of lights in the room.

(a) Determine the sensible heat loss or gain.
(b) Specify the desired interior dry-bulb temperature and relative humidity.
(c) If the heating system provides air at 95°F, determine the required airflow (cfm) and the maximum relative humidity permissible in the incoming air.
[Ans: 26,916 Btu/h; 75°F, 28%; 1246 cfm, 14%]

11.19 A zone in a building has a sensible load of 20.5 kW (70,000 Btu/h) and a latent load of 8.8 kW (30,000 Btu/h). The zone is to be maintained at 25°C (77°F) and 50% rh.

(a) Calculate the conditions (t and W) of the entering air to the zone if the air leaves the coil saturated.
(b) What flow rate is required in order to maintain the space temperatures?
(c) If a mixture of 50% return air and 50% outdoor air at 31.6°C (97°F) and 60% rh enters the air conditioner, what is the refrigeration load?

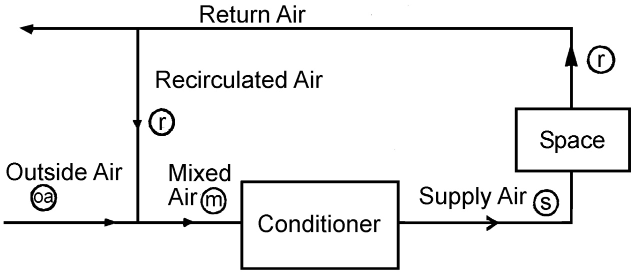
11.20 Sketch (with line diagrams) and list the advantages, disadvantages, and typical uses of the following systems:
(a) Fan-coil units
(b) Terminal reheat system
(c) Multizone system
(d) Double-duct system
(e) Variable-volume system
(f) Induction system

11.21 A general office building in St. Louis, Missouri, has a winter sensible space heating load of 1,150,000 Btu/h for design conditions of 75°F and –5°F. The heating system operates with 25% outdoor air mixed with return air.
(a) Schematically draw the flow diagram and label, including temperatures and flow rates at each location.
(b) Specify the necessary furnace size.

11.22 For the building of Problem 11.23, determine:
(a) Annual energy requirements for heating, Btu
(b) Annual fuel cost using No. 2 fuel oil at $1.60/gal

11.23 To provide comfort conditions for a general office building, 38 ft by 80 ft by 8 ft, an air-treating unit consisting of cooling coil, heating coil, and humidifier is provided for this space with the flow diagram as shown. Indoor design conditions are: summer, 78°F/60% rh; winter, 72°F/25% rh.

Ninety people are normally employed doing light work while seated. The building is in Kansas City, Missouri. Fan operation is constant all year long. Ventilation rate is 15 cfm/person.



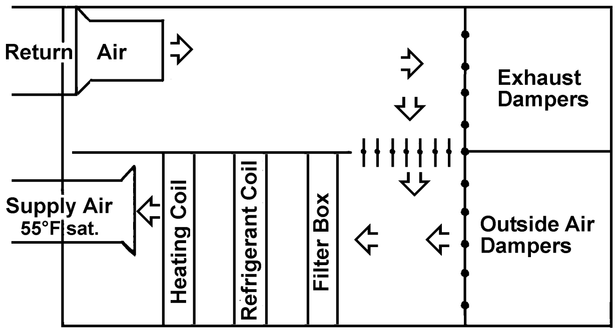
Winter: Sensible space heat loss is 189,000 Btu/h at design conditions, latent load is negligible. Maximum supply air temperature is 155°F.

Summer: Sensible space heat gain is 101,200 Btu/h at design conditions. Latent load is due entirely to the occupancy. The minimum supply air temperature from the cooling coil is 58°F.
(a) Determine the fan size (scfm) needed to provide sufficient air
(b) Size the heating unit needed, Btu/h
(c) Size the cooling coil needed, Btu/h
(d) Size the humidifier, gal/h

11.24 A view of the air-conditioning system for a building in Denver, Colorado (elevation = 5000 ft; barometric pressure = 12.23 psi), is given. Outdoor air at the rate of 2500 cfm is required for ventilation. Other conditions at summer design are

Space Loads
Sensible = 410,000 Btu/h
Latent = 220,000 Btu/h

Outdoor Air: 91°F, 30% rh



For an indoor design temperature of 78°F, determine

1. Supply airflow, lb/h
2. Supply airflow, cfm
3. Relative humidity at return, %
4. Size of cooling unit, Btu/h
5. Latent component of (4)
6. Sensible component of (4)
7. Sensible cooling load due to outdoor air, Btu/h

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Chapter 12

SYSTEM CONFIGURATIONS

In this chapter the various types of HVAC systems that provide cooling, dehumidifying, heating, and humidifying are described. The way the systems are identified with regard to the fluid used for carrying the energy and/or moisture is presented. Details about these systems are given in Chapters 1 through 6 in the 2016 *Handbook—HVAC Systems and Equipment.*

12.1 Introduction

It has been common practice for many years to classify systems as all-air systems, air and water systems, and all-water systems. However, the current publication of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* has subdivided systems into two categories: (1) decentralized cooling and heating (Chapter 2), and (2) central cooling and heating (Chapter 3). As high-performance systems evolve, though, there are a growing number of hybrid system options that combine central system technology with decentralized technology.

Decentralized Systems. In the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, Chapter 2, the decentralized systems are those that are generally located in or near the space to be conditioned and convert an available energy form to useful cooling, heating, or humidity control within that device. Most of these systems have historically been referred to as unitary systems. That chapter gives the following as examples of decentralized HVAC systems:

- Window air conditioners
- Through-the-wall room HVAC units
- Air-cooled heat pump systems
- Water-cooled heat pump systems
- Multiple-unit variable refrigerant flow systems
- Residential and light commercial split systems
- Self-contained (floor-by-floor) systems
- Outdoor package systems
- Packaged, special-procedure units (e.g., for computer rooms)

The outside package systems would include the very common host of rooftop units, generally located just above the space that they condition, and the multiple-unit systems would include multiple-evaporator minisplit systems, which have become quite popular.

Central Cooling and Heating. According to Chapter 3, Central Cooling and Heating, of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, central cooling and/or heating plants generate cooling and/or heating in one location for distribution to multiple locations in one building or an entire campus or neighborhood, and represent approximately 25% of HVAC system applications. Central cooling and heating systems are used in almost all classes of buildings, but particularly in large buildings and complexes or where there is a high density of energy use. They are especially suited to applications where maximizing equipment service life and using energy and operational workforce efficiently are important.

The following facility types are good candidates for central cooling and/or heating systems:

- Campus environments with distribution to several buildings
- High-rise facilities
- Large office buildings
- Large public assembly facilities, entertainment complexes, stadiums, arenas, and convention and exhibition centers
- Urban centers (e.g., city centers/districts)
- Shopping malls
- Large condominiums, hotels, and apartment complexes
- Educational facilities
- Hospitals and other health care facilities
- Industrial facilities (e.g., pharmaceutical, manufacturing)
- Large museums and similar institutions
- Locations where waste heat is readily available (result of power generation or industrial processes
- Larger systems where higher efficiency offsets the higher first cost of a chilled-water system)

The following are advantages and disadvantages of central cooling and heating systems:

Advantages

- Primary cooling and heating can be provided at all times, independent of the operation mode of equipment and systems outside the central plant.
- Using larger but fewer pieces of equipment generally reduces the facility's overall operation and maintenance cost. It also allows wider operating ranges and more flexible operating sequences.
- A centralized location minimizes restrictions on servicing accessibility.
- Energy-efficient design strategies, energy recovery, thermal storage, and energy management can be simpler and more cost-effective to implement.
- Multiple energy sources can be applied to the central plant, providing flexibility and leverage when purchasing fuel.

- Standardizing equipment can be beneficial for redundancy and stocking replacement parts. However, strategically selecting different-sized equipment for a central plant can provide better part-load capability and efficiency.
- Standby capabilities (for firm capacity/redundancy) and back-up fuel sources can easily be added to equipment and plant when planned in advance.
- Equipment operation can be staged to match load profile and taken offline for maintenance.
- A central plant and its distribution can be economically expanded to accommodate future growth (e.g., adding new buildings to the service group).
- Load diversity can substantially reduce the total equipment capacity requirement.
- Submetering secondary distribution can allow individual billing of cooling and heating users outside the central plant.
- Major vibration and noise-producing equipment can be grouped away from occupied spaces, making acoustic and vibration controls simpler. Acoustical treatment can be applied in a single location instead of many separate locations.
- Issues such as cooling tower plume and plant emissions are centralized, allowing a more economic solution.

Disadvantages

- Equipment may not be readily available, resulting in long lead-time for production and delivery.
- Equipment may be more complicated than decentralized equipment, and thus require a more knowledgeable equipment operator.
- A central location within or adjacent to the building is needed.
- Additional equipment room height may be needed.
- Depending on the fuel source, large underground or surface storage tanks may be required on site. If coal is used, space for storage bunker(s) will be needed.
- Access may be needed for large deliveries of fuel (oil or coal).
- Fossil-fuel heating plants require a chimney and possibly emission permits, monitoring, and treatments.
- Multiple equipment manufacturers are required when combining primary and ancillary equipment.
- System control logic may be complex.
- First costs can be higher. compared to alternatives with rooftop units (RTUs), water-source heat pumps (WSHPs), self-contained equipment, and other systems
- Special permitting may be required.
- Safety requirements are increased.
- A large pipe distribution system may be necessary (which may actually be an advantage for some applications).

12.2 Selecting the System

Willis Carrier defined air conditioning as follows:

> Air conditioning is the control of the humidity of the air by adding or removing moisture from the air, the control of the temperature of the air by heating or cooling the air, the control of the purity of the air by filtering or washing the air, and the control of air motion and ventilation.

Thus, in selecting a system to air-condition a building, the engineer must ensure that, for each space or room in the building, each of the five objectives (i.e., humidity, temperature, purity, air motion, and ventilation) are accomplished within the range necessary to achieve human comfort or satisfy an industrial parameter range.

Many times, it could be that, to achieve best control over all five properties and still satisfy all of the other design parameters (see section 11.3), more than a single system is required. As an example, control of ventilation, humidity, and air purity may best be provided with a dedicated outdoor air system, whereas control of the temperature and air motion may best be achieved with an in-space unit such as a fan-coil unit or unitary heat pump.

12.3 Multiple-Zone Control Systems

This discussion of central station air-handling systems serving multiple zones follows Chapter 4, "Air Handling and Distribution," of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment.*

12.3.1 Constant Volume, Variable Δ*t*

While maintaining constant airflow, constant-volume systems change the supply air temperature in response to the space load (Figure 12-1).

Single-Zone Systems. The simplest all-air system is a supply unit serving a single zone. The unit can be installed either in or remote from the space it serves, and may operate with or without distribution ductwork. Ideally, this system responds completely to the space needs, and well-designed control systems maintain temperature and humidity closely and efficiently. Single-zone systems often involve short ductwork with low pressure drop and thus low fan energy, and can be shut down when not required without affecting operation of adja-

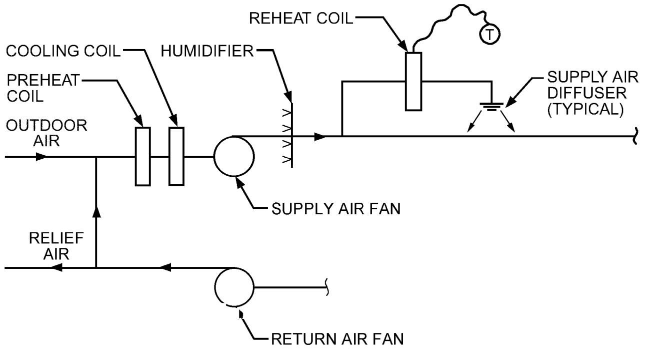


Fig. 12-1 Constant-Volume System with Terminal Reheat
(Figure 9, Chapter 4, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

cent areas, offering further energy savings. A return or relief fan may be needed, depending on system capacity and whether 100% outdoor air is used for cooling as part of an economizer cycle. Relief fans can be eliminated if overpressurization can be relieved by other means, such as gravity dampers.

Multiple-Zone Terminal Reheat Systems. Multiple-zone reheat is a modification of the single-zone system (Figure 12-1). It provides (1) zone or space control for areas of unequal loading, (2) simultaneous heating or cooling of perimeter areas with different exposures, and (3) close control for temperature, humidity, and space pressure in process or comfort applications. As the word *reheat* implies, heat is added as a secondary simultaneous process to either preconditioned (cooled, humidified, etc.) primary air or recirculated room air. Relatively small low-pressure systems place reheat coils in the ductwork at each zone. More complex designs include high-pressure primary distribution ducts to reduce their size and cost, and pressure reduction devices to maintain a constant volume for each reheat zone.

The system uses conditioned air from a central unit, generally at a fixed cold-air temperature that is low enough to meet the maximum cooling load. Thus, all supply air is always cooled the maximum amount, regardless of the current load. Heat is added to the airstream in each zone to avoid overcooling that zone, for every zone except the zone experiencing peak cooling demand. The result is very high energy use, and therefore use of this system is restricted by ASHRAE *Standard* 90.1. However, the supply air temperature from the unit can be varied, with proper control, to reduce the amount of reheat required and associated energy consumption. Care must be taken to avoid high internal humidity when the temperature of air leaving the cooling coil is allowed to rise during cooling. Constant-volume reheat can ensure close control of room humidity and/or space pressure.

In cold weather, when a reheat system heats a space with an exterior exposure, the reheat coil must not only replace the heat lost from the space, but also must offset the cooling of the supply air (enough cooling to meet the peak load for the space), further increasing energy consumption. If a constant-volume system is oversized, reheat energy becomes excessive.

In commercial applications, use of a constant-volume reheat system is generally discouraged in favor of variable-volume or other systems. Constant-volume reheat systems may continue to be applied in hospitals, laboratories, and other critical applications where variable airflow may be detrimental to proper pressure relationships (e.g., for infection control).

Dual-Duct Systems. A dual-duct system conditions all the air in a central apparatus and distributes it to conditioned spaces through two ducts, one carrying cold air and the other carrying warm air. In each conditioned zone, air valve terminals mix warm and cold air in proper proportion to satisfy the space temperature and pressure control (Figure 12-2). Dual-duct systems may be designed as constant volume or variable air volume; a dual-duct, constant-volume system generally uses more energy than a single-duct VAV system. As with other VAV systems, certain primary-air configurations can cause high relative humidity in the space during the cooling season.

Dual-duct, constant-volume systems using a single supply fan were common through the mid-1980s, and were used frequently as an alternative to constant-volume reheat systems. Today, dual-fan, dual-duct are preferred over the former, based on energy performance. There are two types of dual-duct, single-fan application: with reheat, and without.

Single Fan With Reheat. There are two major differences between this and a conventional terminal reheat system: (1) reheat is applied at a central point in the fan unit hot deck instead of at individual zones (Figure 12-2), and (2) only part of the supply air is cooled by the cooling coil (except at peak cooling demand); the rest of the supply is heated by the hot-deck coil during most hours of operation. This uses less heating and cooling energy than the terminal reheat system where all the air is cooled to full cooling capacity for more operating hours, and then all of it is reheated as required to match the space load. Fan energy is constant because airflow is constant.

Single Fan Without Reheat. This system has no heating coil in the fan unit hot deck and simply pushes a mixture of outside and recirculated air through the hot deck. A problem occurs during periods of high outside humidity and low internal heat load, causing the space humidity to rise rapidly unless reheat is added. This system has limited use in most modern buildings because they are not capable of maintaining comfort conditions in many climatic conditions. A single-fan, no-reheat dual-duct system does not use any extra energy for reheat, but fan energy is constant regardless of space load.

12.3.2 Variable Volume (VAV), Constant or Variable Δt

A VAV system (Figure 12-3) controls temperature in a space by varying the quantity of supply air rather than varying the supply air temperature. A VAV terminal unit at the zone varies the quantity of supply air to the space. The supply air temperature is held relatively constant. Although supply air temperature can be moderately reset depending on the season, it must always be low enough to meet the cooling load in the most demanding zone and to maintain appropriate humidity. VAV systems can be applied to interior or perimeter zones,

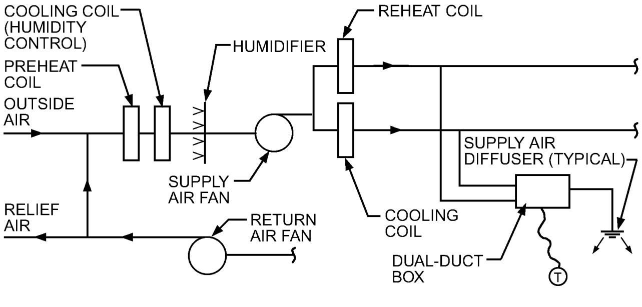


Fig. 12-2 Single-Fan, Dual-Duct System
(Figure 11, Chapter 4, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

with common or separate fans, with common or separate air temperature control, and with or without auxiliary heating devices. The greatest energy saving associated with VAV occurs at the perimeter zones, where variations in solar load and outside temperature allow the supply air quantity to be reduced.

Humidity control is a potential problem with VAV systems. If humidity is critical, as in certain laboratories, process work, etc., constant-volume airflow may be required.

Other measures must also maintain enough air circulation through the room to achieve acceptable ventilation and air movement. The human body is more sensitive to elevated air temperatures when there is little air movement. Minimum air circulation can be maintained during reduced load by (1) raising the supply air temperature of the entire system, which could increase space humidity, or supplying reheat on a zone-by-zone basis; (2) providing auxiliary heat in each room independent of the air system; (3) using individual-zone recirculation and blending varying amounts of supply and room air or supply and ceiling plenum air with fan-powered VAV terminal units, or, if design permits, at the air-handling unit; (4) recirculating air with a VAV induction unit; or (5) providing a dedicated recirculation fan to increase airflow.

VAV reheat can ensure close room space pressure control with the supply terminal functioning in sync with associated room exhaust. A typical application might be a fume hood VAV exhaust with constant open sash velocity (e.g., 85 or 100 fpm) or occupied/ unoccupied room hood exhaust (e.g., 100 fpm at sash in occupied periods and 60 fpm in unoccupied periods).

Dual-Conduit. This method is an extension of the single-duct VAV system: one supply duct offsets exterior transmission cooling or heating loads by its terminal unit with or without auxiliary heat, and the other supply air path provides cooling throughout the year. The first airstream (primary air) operates as a constant-volume system, and the air temperature is varied to offset transmission only (i.e., it is warm in winter and cool in summer). Often, however, the primary-air fan is limited to operating only during peak heating and cooling periods to further reduce energy use. When calculating this system's heating requirements, the cooling effect of secondary air must be included, even though the secondary system operates at minimum flow. The other airstream, or secondary air, is cool year-round and varies in volume to match the load from solar heating, lights, power, and occupants. It serves both perimeter and interior spaces.

Variable Diffuser. The discharge aperture of this diffuser is reduced to keep discharge velocity relatively constant while reducing conditioned supply airflow. Under these conditions, the induction effect of the diffuser is kept high, cold air mixes in the space, and the room air distribution pattern is more nearly maintained at reduced loads. These devices are of two basic types: one has a flexible bladder that expands to reduce the aperture, and the other has a diffuser plate that moves. Both devices are pressure-dependent, which must be considered in duct-distribution system design. They are either powered by the system or pneumatically or electrically driven.

Since the single-duct VAV system cannot provide heat, it is often combined with some configuration of reheat or dual-stream system, devices that are available in many configurations, including the following (see Figure 12-3).

VAV Reheat. This simple VAV system integrates heating at the terminal unit. It is applied to systems requiring full heating and cooling flexibility in interior and exterior zones. The terminal units are set to maintain a predetermined minimum throttling ratio, which is established as the lowest air quantity necessary to (1) offset the heating load, (2) limit the maximum humidity, (3) provide reasonable air movement within the space, and (4) provide required ventilation air. Note, (2) and (4) do not apply if a separate ventilation system is used. (See section 12.4.)

Variable-air-volume with reheat permits airflow to be reduced as the first step in control; heat is then initiated as the second step. Compared to constant-volume reheat, this procedure reduces energy consumption because the amount of primary air to be cooled and secondary air to be heated is reduced in additon to the reduction in fan energy.

A feature can be provided to isolate the availability of reheat during the summer, except in situations where low air-

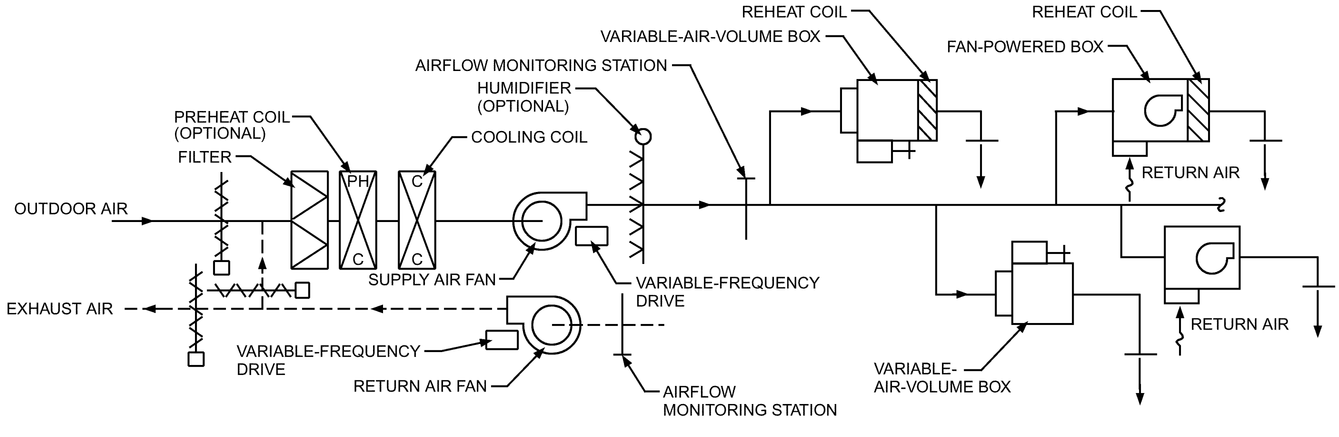


Fig. 12-3 Variable-Air-Volume System with Reheat and Induction and Fan-Powered Devices
(Figure 10, Chapter 4, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

flow should be avoided or where an increase in humidity causes discomfort (e.g., in conference rooms when the lights are turned off).

VAV Induction. The VAV induction system uses a terminal unit to reduce cooling capacity by simultaneously reducing primary air and inducing room or ceiling air to maintain a relatively constant circulating air volume to the room. The primary air quantity decreases with load, retaining the savings of reduced fan power, while quantity of the the air supplied to the space is kept relatively constant to avoid the effect of low-velocity "dumping" or low air movement.

The terminal device is usually located in the ceiling cavity to recover heat from lights. This allows the induction box to be used without reheat coils in internal spaces. In cold climates, provisions must be made for morning warm-up and night heating. Also, interior spaces with a roof load must have heat supplied either separately in the ceiling cavity or at the terminal.

Fan-Powered. Fan-powered systems are available in either parallel or series airflow. In parallel flow units, the fan is located outside the primary airstream to allow intermittent fan operation. In series units, the fan is located within the primary airstream and runs continuously when the zone is occupied. Fan-powered systems, both series and parallel, are often selected because they maintain higher air circulation through a room at low loads while still retaining some of the energy advantages of VAV systems.

As the cold primary air valve modulates from maximum to minimum (or closed), the unit recirculates more plenum air. In a perimeter zone, a hot water heating coil, electric heater, baseboard heater, or remote radiant heater can be sequenced with the cooling to offset external heat losses. Between heating and cooling operations, a dead band, in which the fan recirculates ceiling air only, is provided. This operation permits heat from lights to be used for space heating for improved energy efficiency. During unoccupied periods in cold climates, the main supply air-handling unit remains off and individual fan-powered heating zone terminals are cycled to maintain required space temperature, thereby reducing energy consumption.

Both parallel and series systems use the heat from lights in the ceiling plenum, and both may be provided with filters.

Parallel Arrangement—Intermittent Fan. In this device, primary air is modulated in response to cooling demand and energizes an integral fan at a predetermined reduced primary flow to deliver ceiling air to offset heating demand. These devices are primarily used in perimeter zones where auxiliary hot water or electric heating is required. The induction fan operating range normally overlaps the range of the primary air valve. A back-draft damper on the terminal fan prevents conditioned air from backflowing into the return air plenum when the terminal fan is off.

Series Arrangement—Constant Fan. A constant-volume (series) fan-powered box mixes primary air with air from the ceiling space using a continuously operating fan; this provides a relatively constant volume to the space. These devices are used for interior or perimeter zones and are supplied with or without an auxiliary heating coil. They can be used to mix primary and return air to raise the temperature of the air supplied to the space such as with low-temperature air systems.

12.4 Ventilation and Dedicated Outdoor Air Systems (DOAS)

The major source of water vapor in building spaces in warm, humid climates is the outdoor air. Since the control of both ventilation and humidity in those climates is a major requirement for acceptable indoor air quality and human comfort, the use of a dedicated unit to introduce and condition the outdoor ventilation air source is gaining favor over the use of mixing chambers, which depend on the temperature control psychrometric system to provide both the temperature and humidity controls.

Separating the control of the ventilation air from the room temperature control requires the use of a 100% outdoor air unit, sized to provide only the required ventilation air and designed to filter contaminants from the outdoor air, then provide the humidity control for the entire building space served by the unit and the temperature control for the ventilation air only. A typical unit of this type is shown diagrammatically in Figure 12-4.

The amount of air supplied by the ventilation air conditioning (VAC) unit must be no less than that required by either ASHRAE Standard 62.1 or that necessary to make up for all of the building exhaust plus some additional for building pressurization, whichever is greater.

The outdoor air enters the unit through intake louvers and a two-position intake damper (open-close), then through a filter section designed to remove undesirable particulate and/or chemical impurities from the outdoor air. The heating coil is designed to heat the air from winter outdoor temperatures up to a desired supply air temperature. This coil is not necessary in climates that do not experience winter temperatures below 70°F. The other feature of this coil is that it must be designed to prevent freezing of the heating fluid in below-freezing climates. With water coils, this is best accomplished with vari-

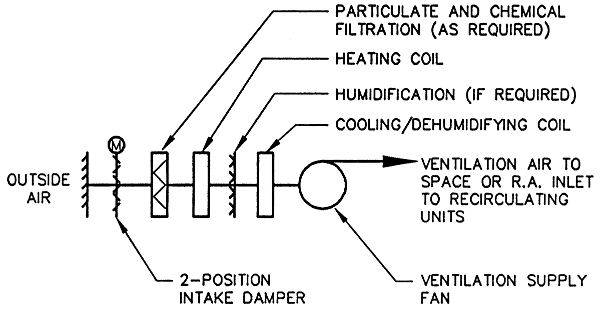


Fig. 12-4 Dedicated Outdoor Air System (DOAS) Unit

able-temperature constant-flow design, and with steam coils it is best achieved with some form of face and bypass control, most desirably integrated face-and-bypass design. Needless to say, low-temperature freeze protection is always required, which should have its sensor on the leaving face of the coil and, on a signal of approaching freezing temperature leaving the coil, should close the intake dampers, shut down the fan, and signal an alarm.

The cooling-dehumidifying coil can be either a chilled water coil or a refrigerant direct expansion coil. It should be sized for 400 fpm maximum velocity, have wide fin spacing (8 fins per inch maximum) to prevent condensate bridging and carryover and should have flat plate fins to allow ready inspection of fouling and related cleaning. Since all of the building dehumidification is achieved at this point, the coil should be close to 100% efficient, which will usually require a deep coil with a minimum of eight rows. The leaving dew-point temperature should be equal to the dew point required in the space less whatever is required to absorb the latent space load.

Winter humidification is generally not recommended or required for human comfort. If, however, it is required for process or safety purposes (such as in hospital operating rooms, museums, or rare book libraries), the humidifier should be placed between the heating coil and the cooling coil.

In systems wherein critical humidity control is required on both the humidifying and dehumidifying cycles, this can be achieved quite effectively by using a sprayed coil and simply sequencing the cooling coil valve with the heating coil valve and controlling with a discharge air controller. (The dry-bulb temperature will be equal to the dew-point temperature since the air leaving a sprayed coil is saturated.) Certain precautions must be taken in designing sprayed-coil units: (1) the pan and coil must be readily accessible for cleaning, and (2) high-temperature shutdown and alarm controls must be provided to protect against microbial contamination.

The discharge air from the VAC unit can be supplied directly into the space through a separate ventilation distribution system or can be supplied into the return air side of the recirculating air-handling system(s).

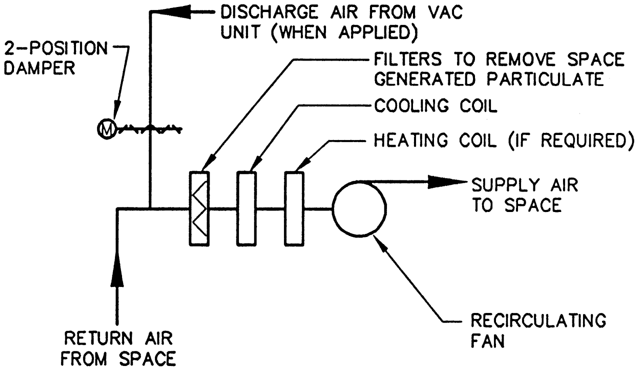


Fig. 12-5 Recirculating Unit

When a VAC unit is used, the space temperature and air circulation control can be provided either with a constant-volume or variable-volume recirculating air unit or with a unit or system located within the space.

12.5 All-Air System with DOAS Unit

Under these conditions, with the ventilation requirements and the humidity control handled by the DOAS unit, the air-handling unit becomes a simple sensible cooling and heating unit, as shown in Figure 12-5. It can be configured to handle any kind of terminal control, and the supply air temperature can be increased or decreased as desired without reheat since it is not necessary that the cooling coil provide any humidity control.

12.6 Air-and-Water Systems with DOAS Unit

If control of the space temperature is achieved by heating and cooling devices located within the space, such as fan-coil units or radiant panels, the ventilation air can be introduced directly into the space. Generally this air can be introduced at the dew-point temperature when the system is in a dehumidifying mode. If the load dynamics dictate the need for a higher supply air temperature, it is necessary to continue to cool the air to the design dew point and then reheat it to the supply temperature required.

Many earlier systems concepts combined large air circulation systems with in-space heating-cooling devices in some integrated fashion. These systems were generally categorized as air-water systems. The most common type of air-water system was the so-called induction system in which the air was introduced through high-velocity nozzles, inducing a stream of room air across a cold or warm coil to provide the ultimate control of room temperature. A later configuration of the induction system is the chilled beam, in which the induction heating and cooling coils are above the ceiling. (See Chapter 20 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and section 12.8 of this chapter.)

Following the publication of ASHRAE Standard 90-1975, the improvement in the thermal quality of building envelopes changed the dynamics of system requirements. The benefits of the air-and-water systems gave way to the all-air systems. But with the improved envelope designs came another issue, the need for a reliable and predictable supply of ventilation air, which was addressed by ASHRAE Standard 62. The result of these two events has led to another generation of the concept of air-water systems. These are the systems that utilize a dedicated outdoor air system (DOAS; see section 12.5) to provide the ventilation air and humidity control (and often the ambient air motion) and a heating-cooling device located within the space, such as a fan-coil unit, radiant panels, or a chilled-beam device to provide the space temperature control.

12.7 In-Space Temperature Control Systems

In-space temperature control systems generally utilize either hot or chilled water for the space conditioning or devices that generate the heating or cooling directly from electricity, such as resistance heating or unitary refrigeration and/or heat pumps. Such devices could include, but are not limited to, the following:

Heating only:
- Baseboard radiation
- Radiators and convectors
- Finned tube radiation
- Unit heaters (cabinet or “propeller”)
- Radiant panels

Cooling only:
- Fan-coil units (two-pipe)
- Radiant panels
- Mini-split evaporators
- Variable refrigerant volume terminals
- Package terminal air conditioners (PTAC units)
- Water-cooled package units
- Chilled-beam induction units

Heating and Cooling:
- Fan-coil units (two-pipe and four-pipe)
- Radiant panels (two-pipe and four-pipe)
- Mini-split evaporators with electric heat
- PTAC heat pump
- Water-source heat pump
- Hot/chilled-beam induction units

A complete description of each of these devices is included in the equipment chapters that follow or in the appropriate chapter(s) of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. Those devices that require the support of an infrastructure system are discussed below. Suffice it to say that none of these in-space temperature control systems has the capability to control humidity (except as a by-product) or provide controlled quantities of ventilation air.

12.7.1 Fan-Coil Units

Basic elements of fan-coil units are a finned-tube coil, filter, and fan section (Figure 12-6). The fan recirculates air continuously from the space through the coil, which contains either hot or chilled water. The unit may contain an additional electric resistance, steam, or hot water heating coil.

A cleanable or replaceable 35% efficiency filter, located upstream of the fan, prevents clogging the coil with dirt or lint entrained in the recirculated air. It also protects the motor and fan and reduces the level of airborne contaminants within the conditioned space. The fan-coil unit is equipped with an insulated drain pan. The fan and motor assembly should be arranged for quick removal for servicing and cleaning the coil. Most manufacturers furnish units with cooling performance that is AHRI certified. The prototypes of the units should be tested and labeled by Underwriters’ Laboratories (UL), or Engineering Testing Laboratories (ETL), as required by some codes.

Fan-coil units are for recirculation heating and cooling only. If ventilation is provided to or through a fan-coil unit, it must be provided by a DOAS unit (see section 12.5).

Room fan-coil units are generally available in nominal sizes of 200, 300, 400, 600, 800, and 1200 cfm, usually with multispeed fan motors. Ventilation should always be provided through a DOAS that engages each room or space.

Basic Components. Room fan-coil units are available in many configurations. Figure 12-7 shows several vertical units. Low vertical units are available for use under windows with low sills; however, in some cases, the low silhouette is achieved by compromising such features as filter area, serviceability, and cabinet style.

Floor-to-ceiling, chase-enclosed units are available in which the water and condensate drain risers are part of the factory-furnished unit. Supply and return air systems must be isolated from each other to prevent air and sound interchange between rooms.

Horizontal overhead units may be fitted with ductwork on the discharge to supply several outlets. A single unit may serve several rooms (e.g., in an apartment unit where individual room control is not essential and a common air return is feasible). High static pressure units with larger fan motors handle the higher pressure drops of units with ductwork.

Central ventilation air from the DOAS unit may be connected to the inlet plenums of the units or introduced directly into the space. If this is done, provisions should be made to ensure that this air is pretreated and held at a temperature to not cause occupant discomfort when the fan-coil unit is off. Coil selection must be based on the temperature of the entering mixture of primary and recirculated air, and the air leaving the coil must satisfy the room sensible cooling and heating requirements. Horizontal models conserve floor space and usually cost less, but when located overhead in furred ceilings, they create problems such as condensate collection and disposal,

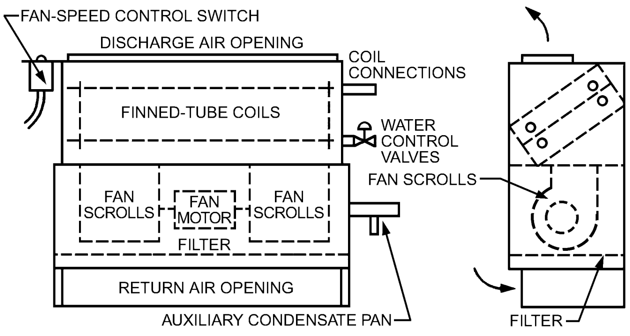
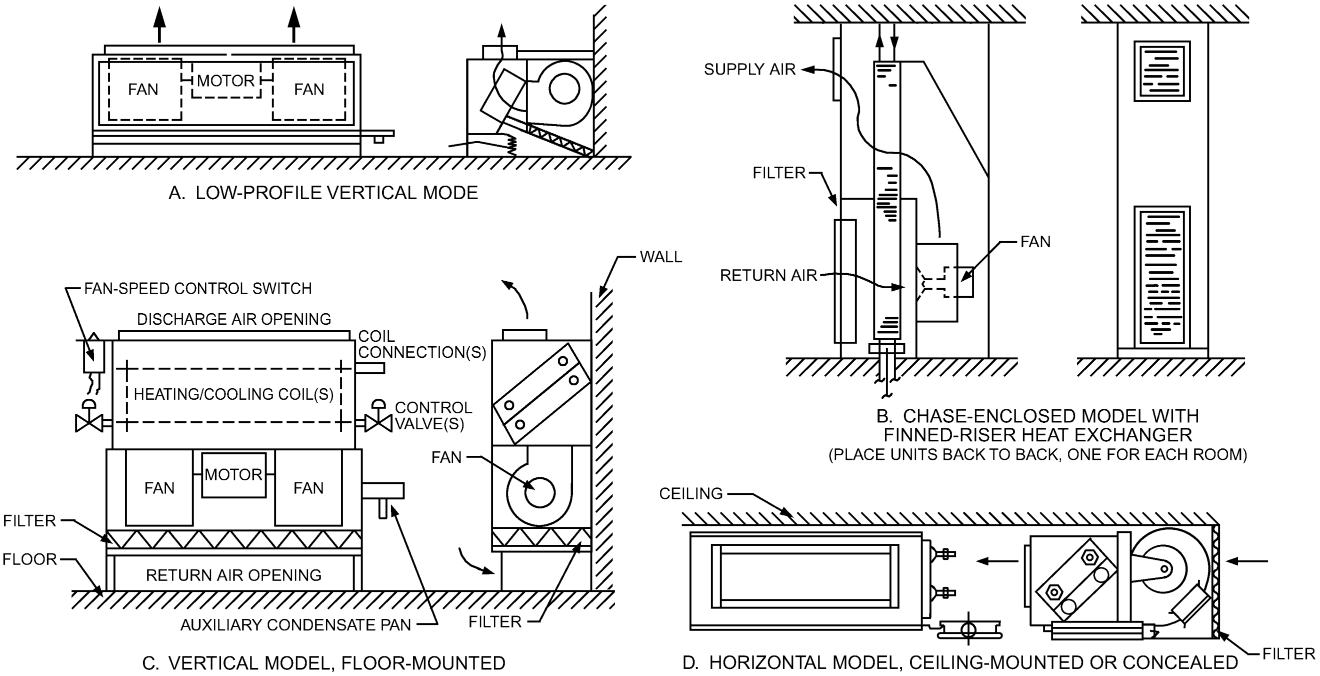


Fig. 12-6 Typical Fan-Coil Unit
(Figure 1, Chapter 5, 2016 ASHRAE Handbook—HVAC Systems and Equipment)



A. LOW-PROFILE VERTICAL MODE

B. CHASE-ENCLOSED MODEL WITH FINNED-RISER HEAT EXCHANGER (PLACE UNITS BACK TO BACK, ONE FOR EACH ROOM)

C. VERTICAL MODEL, FLOOR-MOUNTED

D. HORIZONTAL MODEL, CEILING-MOUNTED OR CONCEALED

Fig. 12-7 Typical Fan-Coil Unit Arrangements
(Figure 1, Chapter 5, 2008 *ASHRAE Handbook—HVAC Systems and Equipment*)

mixing of return air from other rooms, leakage of pans causing damage to ceilings, difficulty of access for maintenance and service, and related IAQ concerns.

Vertical models give better results in climates of extremely cold temperatures, since heating is enhanced by under-window or exterior wall locations. Vertical units can be operated as convectors with the fans turned off during winter night operation.

Selection. Some designers size fan-coil units for nominal cooling at the medium-speed setting when a three-speed control switch is provided. This method ensures quieter operation within the space and adds a safety factor in that capacity can be increased by operating at high speed. Sound power ratings are available from some manufacturers, and, as with any in-the-room unit, sound is a very important design parameter.

Only the sensible space heating and cooling loads need to be handled by the terminal fan-coil units when outdoor air is pretreated by a dedicated outdoor air system. If the ventilation air from the DOAS is supplied at or below dew-point temperature in design cooling weather, the sensible cooling capacity thereby provided can be deducted from the needed capacity of the fan-coil unit.

Wiring. Fan-coil conditioner fans are driven by small motors generally of the shaded pole or capacitor type, with internal overload protection. Operating wattage of even the largest sizes rarely exceeds 300 W at the high-speed setting. Running current rarely exceeds 2.5 A.

Almost all motors on units sold in the United States are wired for 120 V, single-phase, 60 Hz current, and they provide multiple (usually three) fan speeds and an off position or continual speed variation. Other voltages and power characteristics may be encountered, depending on the location, and should be investigated before selecting the fan motor characteristics. Many manufacturers are providing variable speed electronically commutated motors (ECM), which tend to be extremely quiet, and have many control system benefits.

In planning the wiring circuit, local and national electrical codes must be followed. Wiring methods generally provide separate electrical circuits for fan-coil units and do not connect them into the lighting circuit.

Separate electrical circuits connected to a central panel allow the building control system to turn off unit fans from a central point whenever is desired.

Condensate Removal. Even when outdoor air is pretreated, a condensate removal system should be installed on the terminal units. This precaution ensures that moisture condensed from air from an unexpected open window that bypasses the ventilation system is carried away. Drain pans should be an integral feature of all units. Condensate drain lines should be oversized to avoid clogging with dirt and other materials, and provision should be made for periodic cleaning of the condensate drain system. Condensation may occur on the outside of the drain piping, which requires that these pipes be insulated.

Capacity Control. Fan-coil unit capacity can be controlled by coil water flow, fan speed, or a combination of these. The units can be thermostatically controlled by either return air or wall thermostats.

Room thermostats are preferred where fan speed control is used. Return air thermostats do not give a reliable index of room temperature when the fan is off.

Maintenance. Room fan-coil units are equipped with either cleanable or disposable filters that should be cleaned or replaced when dirty. Good filter maintenance improves sanitation and provides full airflow, ensuring full capacity. The frequency of cleaning varies with the application. Applications in apartments, hotels, and hospitals usually require more frequent filter service because of lint. The condensate drain pan and drain system must be cleaned or flushed periodically to prevent overflow and microbiological buildup.

12.7.2 Water Distribution

Chilled and hot water must run to the fan-coil units. The piping arrangement determines the quality of performance, ease of operation, and initial cost of the system.

Two-Pipe Changeover. This method has low initial cost and supplies either chilled water or hot water through the same piping system (see Chapter 13). The fan-coil unit has a single coil, and room temperature controls reverse their action, depending on whether hot or cold water is available at the unit coil.

This system works well in warm weather when all rooms need cooling and in cold weather when all rooms need heat. The two-pipe system **does not have the simultaneous heating or cooling capability** that is required for most facilities during intermediate seasons when some rooms need cooling and others need heat. This problem can be especially troublesome if a single piping zone supplies the entire building. This difficulty may be partly overcome by dividing the piping into zones based on solar exposure. Then each zone may be operated to heat or cool, independent of the others. However, one room may still require cooling while another room on the same solar exposure requires heating—particularly if the building is partially shaded by an adjacent building.

Another difficulty of the two-pipe changeover system is the need for frequent changeover from heating to cooling, which complicates the operation and increases energy consumption to the extent that it may become impractical. For example, two-pipe changeover system hydraulics must consider the water expansion (and relief) that occurs during the cycling from cooling to heating.

The designer should consider the disadvantages of the two-pipe system carefully; many installations of this type waste energy and have been unsatisfactory in climates where frequent changeover is required and where interior loads require cooling simultaneously as exterior spaces require heat. Furthermore, most building occupants demand the ability to select either heating or cooling at any time as the thermal conditions *or* their personal metabolism dictates.

Any two-pipe system must be carefully analyzed. In any case, *they are not recommended for commercial buildings*.

Two-pipe changeover with partial electric resistance heat. This arrangement provides simultaneous heating and cooling in intermediate seasons by using a small electric resistance heater in the fan-coil unit. The unit can handle heating requirements in mild weather, typically down to 40° F, while continuing to circulate chilled water to handle any cooling requirements. When the outdoor temperature drops sufficiently to require heating in excess of the electric heater capacity, the water system must be changed over to hot water.

Four-Pipe Distribution. The four-pipe system with separate heating and coolng coils provides the best fan-coil system performance. It provides (1) all-season availability of heating and cooling at each unit, (2) no summer/winter changeover requirement, (3) simpler operation, and (4) use of any heating fuel, heat recovery, or solar heat. In addition, it can be controlled to maintain a “dead band” between heating and cooling so that there is no possibility of simultaneous heating and cooling with the same unit.

Central Equipment. Central equipment size is based on the block load of the entire building at the time of building peak load, not on the sum of individual fan-coil unit peak loads. Cooling load should include appropriate diversity factors for lighting and occupant loads. Heating load is based on maintaining the unoccupied building at design temperature, plus an additional allowance for pickup capacity if the building temperature is set back at night.

If water supply temperature or quantities are to be reset at times other than at peak load, the adjusted settings must be adequate for the most heavily loaded space in the building. An analysis of individual room load variations is required.

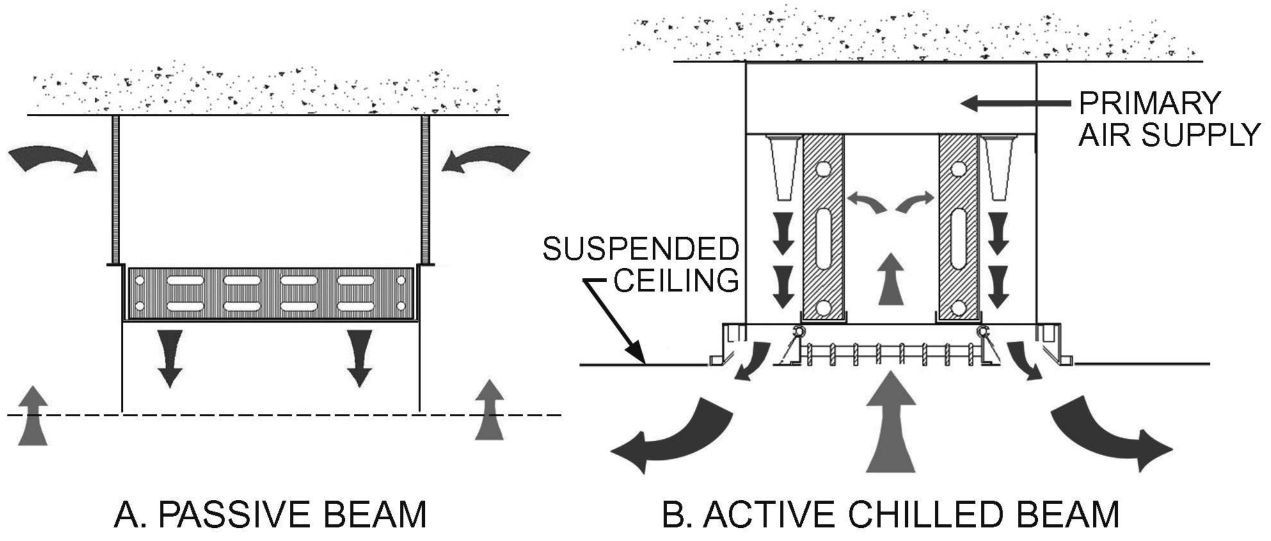
If the side exposed to the sun or interior zone loads require chilled water in cold weather, the use of condenser water with a water-to-water interchanger may be considered. Varying refrigeration loads requires the water chiller to operate satisfactorily under all conditions.

Ventilation requires a dedicated outdoor air unit complete with heating and cooling coils, filters, and fans to handle the ventilation load. An additional advantage of the DOAS unit is that, if it is sized for the internal latent load, the terminal cooling coils remain dry.

12.8 Chilled-Beam Systems

Chilled beams are an evolution of chilled ceiling panels. Reports of energy savings over variable-air-volume (VAV) systems, especially in spaces with high concentrations of sensible loads (e.g., laboratories), have been touted in Europe and Australia. Applications such as health care, data centers, and some office areas may be well suited to chilled-beam systems.

Two types of chilled beams, passive and active, are in use (Figure 12-8). Passive chilled beams consist of a chilled-water coil mounted inside a cabinet. Chilled water is piped to the convective coil at between 58°F and 60°F. Passive beams use convection currents to cool the space. As air that has been cooled by the beam's chilled water coil falls into the space, warmer air is displaced, rises into the coil, and is cooled. Passive beams can provide approximately 400 Btu·h/ft of cooling and, to ensure proper dehumidification and effective ventilation to the spaces, require a separate system to provide tempered, dehumidified air. Heat can be provided by finned-tube



A. PASSIVE BEAM

B. ACTIVE CHILLED BEAM

Fig. 12-8 Passive and Active Chilled Beam Operation
(Figure 2, Chapter 5, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

radiation along the space perimeter. Overcooling must be avoided during cooling seasons to prevent discomfort, condensation, and microbial growth in spaces. Active chilled beams can provide up to approximately 800 Btu·h/ft of cooling. They operate with induction nozzles that entrain room air and mix it with the primary or ventilation air that is ducted to the beam. Chilled water is piped to the coil at between 55°F and 60°F. Primary air should be ducted to the beam at 55°F or lower to provide proper dehumidification. The primary air is then mixed with inducted room air at a ratio of 1:2. For example, 50 cfm of primary air at 55°F may be mixed with 100 cfm of recirculated room air, and the active beam would distribute 150 cfm at around 65°F. If the low-temperature primary air alone will overcool spaces during any time of the year, there must be provision for reheat. Active beams can have either a two- or four-pipe distribution system. The two-pipe system may be cooling only or two-pipe changeover. Active beams can be designed to heat and cool the occupied space, but finned-tube radiation is still commonly used to provide heating in a space that is cooled with active beams. Both active and passive beams are designed to operate dry, without condensate.

See Chapter 5 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for additional details.

Example 12-1 A two-zone building in St. Louis, Missouri, has the hourly heating and cooling loads given in the following tables. Assuming that the January day is the coldest for the year, and the August day is the warmest for the year, size the basic components and sketch the equipment arrangement for each of the following types of systems:

1. Separate 4-pipe fan-coil units using chilled water supplied at 45°F for cooling and hot water supplied at 190°F for heating.
2. Variable volume with reheat (turndown to 50% of design airflow) with cooling coil discharge at 58°F all year long.
3. Double-duct or multizone with design cold and hot deck temperatures of 58°F and 130°F, respectively (winter), and 58°F and 85°F, respectively (summer).

Space Design Conditions

Summer:	78°F db
Winter:	72°F db, 30% rh

Outdoor Design Conditions

Summer:	94°F/75°F wb
Winter:	6°F

Ventilation (Outside) Air Requirements

Zone 1:	550 cfm
Zone 2:	400 cfm

Design Pressure Drop for Duct System
2.8 in. w.g. for Systems 2, 3
1.0 in. w.g. for System 1

Solution:
St. Louis
Winter: t_o = 6°F, 100% rh $\rightarrow h_o$ = 2.61, W_o = 0.0011
t_i = 72°F, 30% $\rightarrow h_i$ = 22.86, W_i = 0.0051
Summer: t_o = 94°F/75°F $\rightarrow h_o$ = 38.85, W_o = 0.0148
t_i = 78°F

Cold Deck: 58°F (if ϕ = 100%); h_s = 25.36, W_s = 0.0105

System No. 1: Four-Pipe Fan-Coil Units

(Δp = 1.0 in. w.g.)

Zone 1

$$Q_1 = \frac{54945}{1.10(78-58)} = 2498 \text{ cfm} \qquad Q_o = 550 \text{ cfm}$$

$$m_1 = \frac{2498(60)}{13.33} = 11,200 \text{ lb/h}$$

Summer:

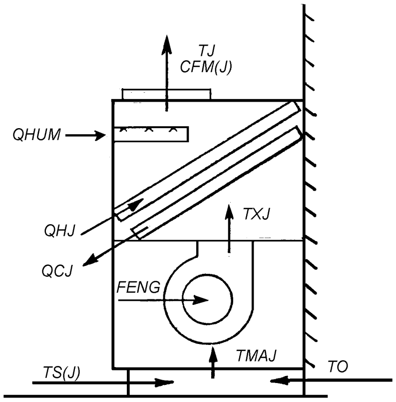
$$W_r = 0.0105 + \frac{1500/1100}{2498 \times 60/13.33} = 0.0106$$

JANUARY

Hour	Sensible Load		Latent Load	
	Zone 1 Btu/Hr	Zone 2 Btu/Hr	Zone 1 Btu/Hr	Zone 2 Btu/Hr
1	-32985.	-32485.	1500.	1800.
2	-33568.	-33068.	1500.	1800.
3	-33918.	-33418.	1500.	1800.
4	-34034.	-33534.	1500.	1800.
5	-34337.	-33837.	1500.	1800.
6	-34477.	-33977.	1500.	1800.
7	-34360.	-33860.	1500.	1800.
8	-26830.	-23173.	1500.	1800.
9	-13237.	-13728.	1500.	1800.
10	- 5232.	-14699.	1500.	1800.
11	2636.	-18162.	1500.	1800.
12	7092.	-21509.	1500.	1800.
13	11608.	-20380.	1500.	1800.
14	16073.	-20729.	1500.	1800.
15	13690.	-21782.	1500.	1800.
16	192.	-23353.	1500.	1800.
17	-27088.	-26588.	1500.	1800.
18	-28183.	-27683.	1500.	1800.
19	-29115.	-28615.	1500.	1800.
20	-30188.	-29688.	1500.	1800.
21	-30980.	-30480.	1500.	1800.
22	-31587.	-31087.	1500.	1800.
23	-32309.	-31809.	1500.	1800.
24	-32752.	-32252.	1500.	1800.

AUGUST

Hour	Sensible Load		Latent Load	
	Zone 1 Btu/Hr	Zone 2 Btu/Hr	Zone 1 Btu/Hr	Zone 2 Btu/Hr
1	1027.	527.	1500.	1800.
2	1820.	1320.	1500.	1800.
3	2426.	1926.	1500.	1800.
4	2659.	2159.	1500.	1800.
5	3195.	2695.	1500.	1800.
6	416.	7621.	1500.	1800.
7	3263.	22148.	1500.	1800.
8	10136.	29670.	1500.	1800.
9	18660.	31112.	1500.	1800.
10	31591.	34088.	1500.	1800.
11	38869.	27952.	1500.	1800.
12	42982.	23170.	1500.	1800.
13	48478.	23717.	1500.	1800.
14	54945.	22592.	1500.	1611.
15	45420.	17379.	1500.	1471.
16	41390.	16062.	1500.	1564.
17	36070.	13277.	1500.	1800.
18	21612.	9042.	1500.	1800.
19	6175.	6675.	1500.	1800.
20	4264.	4764.	1500.	1800.
21	2632.	3132.	1500.	1800.
22	1187.	1687.	1500.	1800.
23	185.	685.	1500.	1800.
24	515.	15.	1500.	1800.



System Sketch

$t_r = 78 \quad h_r = 30.34$

$$h_m = \frac{1948(30.34) + 550(38.85)}{2498} = 32.21$$

$$W_m = \frac{1948(0.0106) + 550(0.0148)}{2498} = 0.0115$$

$$W_{\text{fan}1} = \frac{Q\Delta p}{6350} = \frac{2498(1)}{6350} = 0.4 \text{ hp}$$

(2500 cfm, 1 in. w.g.)

$$h_f = 32.21 + \frac{0.4(2545)}{11,200} = 32.3$$

$$q_{cc} = 11{,}200[32.3 - 25.36 - (0.0115 - 0.0105)26]$$
$$= 77{,}400 \text{ Btu/h}$$

Winter:

$$t_s = 72 + \frac{34477}{1.10(2498)} = 84.5°\text{F}$$

$$W_s = 0.0051 - \frac{1500/1100}{11{,}200} = 0.00498$$

$$W_m = \frac{1948(0.0051) + 550(0.0011)}{2498} = 0.00422$$

$$t_m = \frac{1948(72) + 550(6)}{2498} = 57.5°\text{F}$$

$$t_f = 57.5 + \frac{0.4(2545)}{11{,}200(0.244)} = 57.9°\text{F}$$

$q_{hc} = 2498(1.1)(84.5 - 57.9) = 73{,}000$ Btu/h heaters

$m_w = 11{,}200(0.00498 - 0.00422) = 8.5$ lb/h humidifiers

Zone 2

$$Q_2 = \frac{34088}{1.10(78-58)} = 1549 \text{ cfm} \quad Q_0 = 400 \text{ cfm}$$

$$m_2 = \frac{1549(60)}{13.33} = 6970 \text{ lb/h}$$

Summer:

$$W_r = 0.015 + \frac{(1800/1100)}{6970} = 0.01073, \ h_r = 30.5$$

$$W_m = \frac{1139(0.01073) + 400(0.0148)}{1549} = 0.0117$$

$h_m = 32.45$

$$W_{\text{fan}2} = \frac{1549 \times 1}{6350} = 0.244 \text{ hp}, \ h_f = 32.53$$

$$q_{cc} = \frac{1549 \times 60}{13.33}[32.53–25.36–(0.0117 - 0.0105)26]$$

$$= 49,800 \text{ Btu/h}$$

Winter:

$$t_o = 72 + \frac{33977}{1.10(1549)} = 91.9°\text{F}$$

$$W_s = 0.0051 - \frac{1800/1100}{6970} = 0.00486$$

$$W_m = \frac{1149(0.0051) + 400(0.0011)}{1549} = 0.00407$$

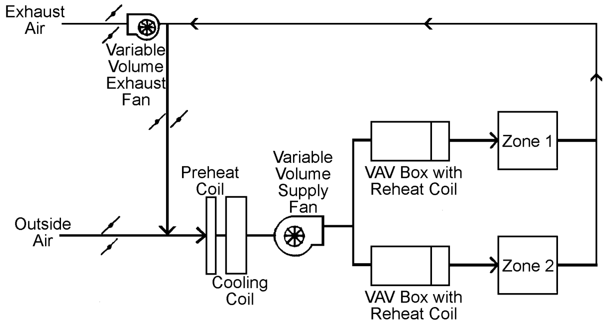
$$t_m = \frac{1149(72) + 400(6)}{1549} = 55.0°\text{F}$$

$$t_f = 55.0 + \frac{0.24(2545)}{6970(0.244)} = 55.4°\text{F}$$

$q_{Hc} = 1549(1.1)(91.9 - 55.4) = 52{,}200$ Btu/h

$m_w = 6970(0.00486 - 0.00407) = 5.5$ lb/h

System No. 2: VAV with Reheat



System Sketch

Design Peak Cooling = 77,537 = 1.10 Q_T (78 – 58)

$Q_T = 3524$ cfm $m = 15{,}860$ lb/h

Space 1 airflow: $54{,}945 = 1.10Q(78 - 58)$
$Q_1 = 2498$ cfm (max), 1249 cfm (min)

Space 2 airflow: $34{,}088 = 1.10Q(78 - 58)$
$Q_2 = 1549$ cfm (max), 775 cfm (min)

$$W_{\text{fan}} = \frac{3524(2.8)(62.4)(60)}{12(778)(2545)} = 1.6 \text{ hp}$$

$$\Delta h_f = \frac{1.6(2545)}{15{,}860} = 0.26 \quad \Delta t_f = \frac{\Delta h_f}{c_p} = 1.1°\text{F}$$

Fan: 3500 scfm; $\Delta p = 2.8$ in. w.g.; 1.6 hp motor

$$W_r = W_s + \frac{m_{w1} + m_{w2}}{m_a}$$

$$= 0.0105 + \frac{3111}{15{,}860\,(1100)} = 0.01068$$

$t_r = 78°$F $h_r = 30.42$

$$h_m = \frac{2574(30.42) + 950(38.85)}{3524} = 32.69 \quad h_f = 32.95$$

$$W_m = \frac{2574(0.01068) + 950(0.0148)}{3524} = 0.0118 = W_f$$

$q_{cc} = 15{,}860[32.95 - 25.36 - (0.0118 - 0.0105)26]$
$= 119{,}800$ Btu/h

Winter:

#1 $q_1 = 34477 = 1.10(1249)(_{t1} - 72) \quad t_1 = 97°\text{F}$
#2 $q_2 = 33977 = 1.10(775)(_{t2} - 72) \quad t_2 = 112°\text{F}$

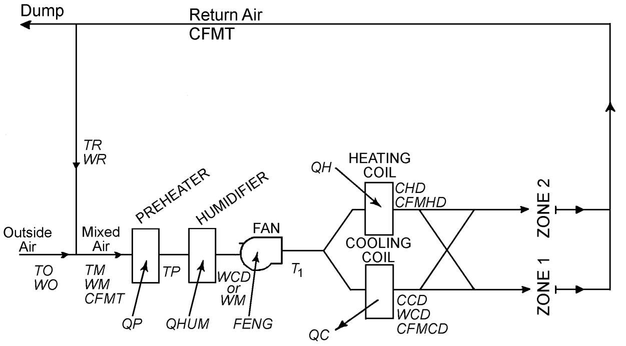
$q_{rh1} = 1.10(1249)(97 - 58) = 53{,}600$ Btu/h
$q_{rh2} = 1.10(775)(112 - 58) = 46{,}000$ Btu/h

$$W_m = \frac{950(0.0011) + 1074(0.0051)}{(950 + 1074)} = 0.00322$$

$$W_s = 0.0051 - \frac{3300/1100}{15{,}860/2} = 0.00472$$

$$m_w = \frac{15860}{2}(0.00472 - 0.00322) = 11.9 \text{ lb/h}$$

System No. 3: Multizone



System Sketch

$$Q_{1,\,s\mathrm{MAX}} = \frac{54{,}945}{1.10(78-58)} = 2498\ \text{cfm}$$

$$Q_{1,\,w\mathrm{MAX}} = \frac{34{,}477}{1.10(130-72)} = 540\ \text{cfm (use 2498 cfm)}$$

$$Q_{2,\,s\mathrm{MAX}} = \frac{34{,}088}{1.10(78-58)} = 1549\ \text{cfm}$$

$$Q_{2,\,w\mathrm{MAX}} = \frac{33{,}977}{1.10(130-72)} = 533\ \text{cfm (use 1549 cfm)}$$

$$m = 4047 \times 60/13.33 = 18{,}200$$

Total cfm = 2498 + 1549 = 4047

Fan

$$W_f = \frac{4047(2.8)}{6350} = 1.8\ \text{hp},\quad 4047\ \text{cfm},\quad 2.8\ \text{in. w.g.}$$

$$\Delta h_f = \frac{1.8(2545)}{18{,}200} = 0.25 \qquad \Delta t = 0.25/0.244 \approx 1°\text{F}$$

Summer:

$$W_r = 0.0105 + \frac{3111/1100}{18{,}200} = 0.0106 \qquad t_r = 78°\text{F}$$

$$h_r = 30.34$$

$$h_m = \frac{3097(30.34) + 950(38.85)}{4047} = 32.34$$

$$W_m = \frac{3097(0.0106) + 950(0.0148)}{4047} = 0.01158$$

$$h_f = 32.34 + 0.25 = 32.59$$

$$1.10Q_{c2}(78-58) = 22{,}592 + 1.10(1549 - Q_{c2})(85-78)$$

$$Q_{c2} = 1162\ \text{cfm};\ Q_{cc,\,\max} = 2498 + 1162 = 3660\ \text{cfm}$$

$$m_{cc} = \frac{3660(60)}{13.33} = 16{,}474\ \text{lb/h}$$

$$\begin{aligned} q_{cc} &= 16{,}474[32.59 - 25.36 - (0.01158 + 0.0105)26] \\ &= 118{,}640\ \text{Btu/h} \end{aligned}$$

Winter:

$$\begin{aligned} m_w &= \frac{950(60)}{13.33}(0.005 - 0.0011) - \frac{3300}{1100} \\ &= 13.67\ \text{lb/h} \end{aligned}$$

#1 $1.10Q_{H1}(130-72) = 34{,}477 + 1.10(2498 - Q_{H1})(72-58)$
$Q_{H1} = 921$ cfm
#2 $1.10Q_{H2}(130-72) = 33{,}977 + 1.10(1549 - Q_{H2})(72-58)$
$Q_{H2} = 730$ cfm

$Q_H = 921 + 730 = 1651$cfm

$$t_m = \frac{3097(72) + 950(6)}{4047} = 54.5°\text{F} \qquad t_f = 55.5°\text{F}$$

$q_{Hc} = 1.10(1651)(130 - 55.5) = 135{,}300$ Btu/h

> The humidity quantities are different in the three systems (14.0, 11.9, and 13.7 lb/h). The following could be reasons for this. Sometimes the humidity ratio *W* was rounded, and when differences are taken errors can occur. In the VAV system, all the air goes through the cooling coil and is dehumidified, while in the multi-zone, some of the air goes through the cooling coil, but not all. Also, in the calculations, the correct value for the specific volume (v) at each point was not always used for convenience reasons. For mixing of the air streams, the volume flow rate was used for convenience rather than the mass flow rate. Also, with a temperature control only thermostat being used, the relative humidity in some spaces for some systems may float rather than stay at the design value. The procedures used in the example are typical in the industry and with the various assumptions in the equation development, it is not unusual to see these kinds of small differences. HVAC is not an exact science.

All of the W_{fan} terms for each system in Example 12.1 are for the air horsepower. The motor efficiency would need to be applied to size the fan motor.

12.9 Problems

12.1 From an energy consumption perspective, list the four fundamental psychrometric system types from least consumption to most consumption.

12.2 In a VAV system with series fan powered terminals, why must all of the terminal fans be running prior to turning on the system fan?

12.3 What is the advantage of a parallel fan-powered terminal over a series fan-powered terminal?

12.4 What is the purpose of using a fan-powered terminal in a variable-air-volume system?

12.5

(a) Why do some VAV systems also use dual-duct or reheat features?

(b) In your own words, describe the operating sequence of the zone or terminal control of

(1) A VAV system
(2) A VAV reheat system
(3) A dual-duct VAV system

12.6 What is the primary advantage of a dedicated outdoor air system (DOAS)?

12.7 Why is a high-pressure primary system fan required with an induction system?

12.8 Are fan-coil units with direct connections to the outdoors recommended as an acceptable method for providing ventilation air? Why?

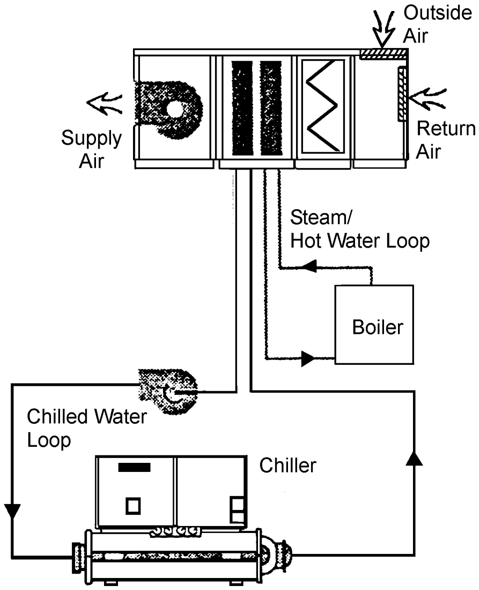
12.9 Size the basic components and sketch the equipment arrangement if the HVAC system now under consideration for

the building of Example 12-1 is a triple-deck multizone (hot, cold, and neutral decks).

12.10 Size the basic components and sketch the equipment arrangement if the HVAC system now under consideration for the building of Example 12-1 is a variable volume, dual fan, dual duct.

12.11 A small single-zone classroom building is being designed for Knoxville, Tennessee, to use the HVAC system shown in the sketch. Minimum outdoor air for meeting the ventilation requirements of the anticipated 550 occupants will be maintained throughout the year. Fan speed will be changed between summer and winter. The duct system will be designed so that at summer air flow rate the pressure drop does not exceed 3.75 in. w.g. At winter design conditions, the air is heated to 130°F at which temperature it is supplied to the conditioned space. The winter conditioning unit includes both a heating coil and a humidifier supplied with city water at 60°F. The humidistat in the return airstream maintains the design relative humidity of 30% in winter. During summer operation, the cooling coil supplied air to the conditioned space at 58°F. The space design loads are

Summer:	423,000 Btu/h sensible (gain) 139,000 Btu/h latent (gain)
Winter:	645,000 Btu/h sensible (loss) negligible latent



Size the following system components:

(a) Cooling coil, Btu/h and ft² of face area
(b) Chiller unit, Btu/h
(c) Heating coil, Btu/h and ft² of face area
(d) Boiler, Btu/h
(e) Humidifier, gph

Select an appropriate air handler from the following data.

Unit Physical Data (Approximate)

Unit Size	Design cfm	Unit Coil Face Area*, ft^2	Max Unit Wt., lb
3	1,660	2.34 – 3.32	≤3,600
6	2,930	4.31 – 5.86	
8	3,770	5.49 – 7.54	
10	4,820	7.01 – 9.64	
12	6,150	9.46 – 12.3	
14	7,110	10.2 – 14.2	
17	8,400	12.3 – 16.8	
21	10,390	15.0 – 20.8	
25	12,190	17.8 – 24.4	
30	14,505	21.2 – 29.0	
35	17,050	26.72 – 34.10	≤ 4, 500
40	19,650	30.78 – 39.30	
50	24,715	34.22 – 49.43	
66	32,815	48.13 – 65.63	≤6,000 (all modules)
80	39,375	56.88 – 78.75	
100	50,180	73.44 – 100.4	

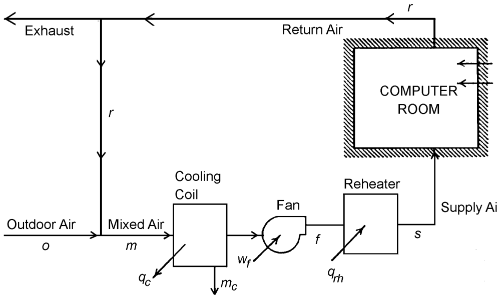
* Actual face area varies with unit coil type.

12.12 A double-duct system is to be used for air conditioning of a two-zone building. At winter design outdoor temperature of 0°F, exterior SPACE 1 has a design sensible heat *loss* of 112,000 Btu/h while interior SPACE 2 has a net sensible heat *gain* of 23,500 Btu/h. At summer design outdoor conditions of 95°F db and 75°F wb, SPACE 1 has a design sensible heat gain of 67,000 Btu/h while SPACE 2 experiences a design sensible heat gain of 49,000 Btu/h. Interior design temperatures of both spaces is 75°F, all year long. Duct pressure drop is 3.1 in. water. Outdoor air requirement is 1400 cfm.

Calculate the size of

(a) Fan (scfm, pressure, motor horsepower)
(b) Heating coil (Btu/h).

12.13 To maintain necessary close control of humidity and temperature required for a computer room, the reheat air conditioning system shown in the sketch is used. Space loads for the computer room include a heat load of 85,000 Btu/h and a moisture load of 42 lb/h. The return air conditions from the space must be exactly 50% relative humidity and 78°F. After mixing of the outside ventilation air with return air, the mixed air is at 80°F dry bulb with a relative humidity of 0.0114 lb/lb. The air is then cooled to saturation at 50°F by the cooling coil. There is a 2°F temperature rise across the fan. Air flow is controlled by a humidistat in the return air duct. The thermostat controls the temperature leaving the reheater.

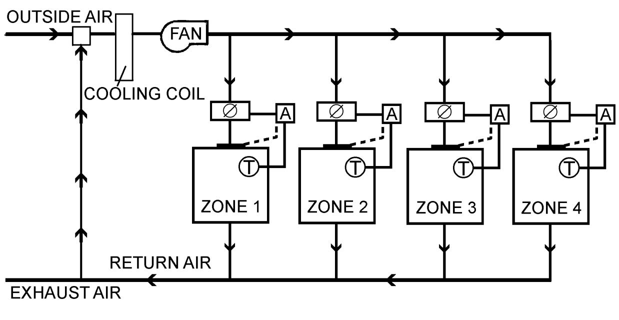


Sketch for Problem 12.13

Size the reheater (kW) and the cooling coil (Btu/h). From a manufacturer's catalog, select an appropriate electric resistance reheater coil. From a manufacturer's catalog, select an appropriate chilled water cooling coil.

12.14 A small commercial building located in St. Louis, Missouri is to be conditioned using a variable-air-volume (VAV) system with reheat, as shown in the following sketch. At this stage of the process, preliminary sizing of the central cooling unit, of the reheaters, and of the fan (scfm) is to take place. There are four zones (separately thermostated spaces) in the building. Supply air from the cooling coil is maintained at 55°F during the summer and 58°F during the winter. Relative humidity off the coil is approximately 90% in both cases. Minimum outdoor air of 4000 scfm is maintained at all times (just don't ask how). The VAV boxes are not to be cut back beyond 50% of rated flow. The design conditions and calculated design load for each zone are as follow:

Zone 1
  Winter inside temperature = 72°F
  Winter design heat loss = –55,000 Btu/h (a loss)
  Summer inside temperature = 78°F
  Summer design heat gains = 124,000 Btu/h (sensible) and 31,000 Btu/h (latent)
Zone 2 (an interior space)
  Winter inside temperature = 78°F
  Winter design heat loss = 40,000 Btu/h (a gain)
  Summer inside temperature = 78°F
  Summer design heat gains = 220,000 Btu/h (sensible) and 71,000 Btu/h (latent)
Zone 3 (an interior space)
  Winter inside temperature = 78°F
  Winter design heat loss = 115,000 Btu/h (a gain)
  Summer inside temperature = 78°F
  Summer design heat gains = 140,000 Btu/h (sensible) and 42,000 Btu/h (latent)
Zone 4
  Winter inside temperature = 72°F
  Winter design heat loss = –180,000 Btu/h (a loss)
  Summer inside temperature = 78°F
  Summer design heat gains = 210,000 Btu/h (sensible) and 52,500 Btu/h (latent)



Sketch for Problem 12.14

12.15 A commercial three-zone office building is being designed for St. Louis, Missouri where summer outdoor design conditions are 94°F db and 75°F wb and winter outdoor design conditions are 3°F and 100% rh. Each zone is to contain 10,000 ft^2 of floor space. A blow-through multizone unit will be used with cold deck temperature maintained at 58°F all year long and with hot deck temperature varying from a maximum of 130°F at winter design to 85°F during the summer. The amount of outdoor air is to equal the recommended 20 cfm per person. Design occupancy is to be 10 people per 1000 ft^2 of floor area. The duct system will be designed so that the pressure drop does not exceed 2.0 in. w.g. Fan efficiency is estimated at 65%. In winter, the control humidistat in the common return air duct is set at 30% rh. Due to the building orientation and internal zoning, all spaces will experience their peak loads at the same time. The space design loads at indoor design temperatures of 78°F summer and 72°F winter are

Summer
Zone 1: 116,000 Btu/h sensible, 43,000 Btu/h latent (gains)
Zone 2: 290,000 Btu/h Sensible, 59,000 Btu/h Latent (gains)
Zone 3: 190,000 Btu/h sensible, 39,000 Btu/h latent (gains)

Winter
Zone 1: –215,000 Btu/h sensible (loss), negligible latent
Zone 2: 110,000 Btu/h sensible (gain), negligible latent
Zone 3: –171,000 Btu/h sensible (loss), negligible latent

Conduct the preliminary sizing of the fan (scfm and horsepower), cooling coil (scfm and Btu/h), heating coil (scfm and Btu/h), and humidifier (gal/h). Provide a completely labeled sketch of the system.

12.16 An air-conditioning unit takes in 2000 cfm of outdoor air at 95°F dry bulb and 76°F wet bulb, and 6000 cfm of return air at 78°F dry bulb and 50% rh. The conditioned air leaves the chilled water coil at 52°F dry bulb and 90% rh.

(a) What is the refrigeration load on the chiller in tons?
(b) Assume the conditioned air were reheated to 58.5°F dry bulb with electric heaters. What would be the operating cost of these heaters at 2.5 cents per kWh?

12.17 In Problem 12.16, assume 2000 cfm of return air bypasses the chilled water coil and is used for reheat.

(a) How does the final condition of the air compare with the reheated air in part (b) of 12.16? [Ans: 58°F, $W = 0.0081$]

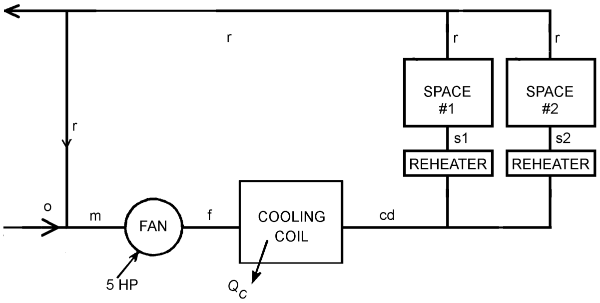
(b) Comment on the ability of the leaving air to absorb latent load in the conditioned space. [Ans: Less than in Example 12-1]

12.18 For the building and reheat system shown below, determine:

(a) Fan rating, scfm
(b) Return air relative humidity at summer design conditions, %
(c) Size cooling coil, Btu/h

(d) Size reheat coils, Btu/h and scfm for each

Winter: Outside 6°F, W = 0.001; indoor 72°F, no humidity control.

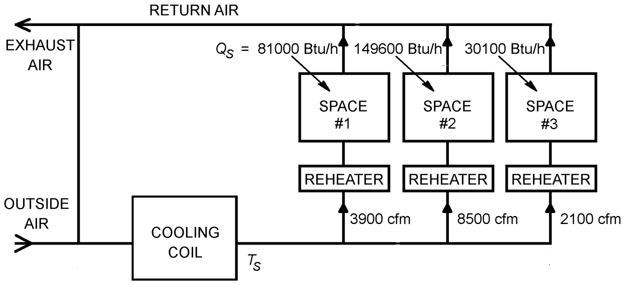


Sensible design heating loads
Space 1: 162,000 Btu/h
Space 2: 143,000 Btu/h

Summer: Outdoor 95°F dry bulb, 78°F wet bulb; indoor 78°F.
Sensible design cooling loads
Space 1: 64,500 Btu/h
Space 2: 55,000 Btu/h
Latent design loads (moisture produced)
Space 1: 38 lb/h
Space 2: 26 lb/h

Year-round: 10% by mass outdoor air required for ventilation.
Conditions of cooling coil: 58°F, 90% rh.

12.19 A basic reheat system has been retrofitted with an improved control system. For the operating conditions shown in the sketch below and with all thermostats set at 78°F, for what cooling coil discharge temperature T should the logic system of the controller be calling if there is no humidity override?



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Chapter 13

HYDRONIC HEATING AND COOLING SYSTEM DESIGN

This chapter provides information useful for the design of hydronic heating and cooling systems. It provides information on the classification of systems, system descriptions, and design procedures. Additional information can be obtained from the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* and the 2015 *ASHRAE Handbook—HVAC Applications*.

13.1 Introduction

Water systems that convey heat to or from a conditioned space or process with hot or chilled water are called hydronic systems. The water flows through piping that connects a boiler, water heater, or chiller to suitable terminal heat transfer units, usually located near or at the space or process.

Water systems can be classified by (1) operating temperature, (2) flow generation, (3) pressurization, (4) piping arrangement, and (5) pumping arrangement.

Classified by flow generation, hydronic heating systems may be (1) gravity systems, which use the difference in density between the supply and return water columns of a circuit or system to circulate water, or (2) forced systems, in which a pump, usually driven by an electric motor, maintains the flow. Gravity systems are only used for heating and are seldom used today.

Water systems can be either once-through or recirculating systems. This chapter describes forced recirculating systems.

13.1.1 Principles

The design of effective and economical water systems is affected by complex relationships between the various system components. The design water temperature, flow rate, piping layout, pump selection, terminal unit selection, and control method are all interrelated. The size and complexity of the system determines the importance of these relationships to the total system operating success. In the United States, present hydronic heating system design practice originated in residential heating applications, where a temperature drop (Δt) of 20°F was used to determine flow rate. Besides producing satisfactory operation and economy in small systems, this Δt enabled simple calculations because 1 gpm conveys approximately 10,000 Btu/h. However, almost universal use of hydronic systems for both heating and cooling of large buildings and building complexes has rendered this simplified approach obsolete.

13.1.2 Temperature Classifications

Water systems can be classified by operating temperature as follows.

Low-temperature water (LTW) system. This hydronic heating system operates within the pressure and temperature limits of the ASME Boiler and Pressure Vessel Code for low-pressure boilers. The maximum allowable working pressure for low-pressure boilers is 160 psig, with a maximum temperature limitation of 250°F. The usual maximum working pressure for boilers for LTW systems is 30 psig, although boilers specifically designed, tested, and stamped for higher pressures are frequently used. Steam-to-water or water-to-water heat exchangers are also used for heating low-temperature water. Low-temperature water systems are used in buildings ranging from small, single dwellings to very large and complex structures.

13.1.3 Condensing Systems

Condensing systems are a special class of low-temperature water systems in which the water is heated in a boiler that is designed to condense the water vapor contained in the flue gases. The condensing boilers are fueled with natural gas (methane), from which the vapor condenses at approximately 130°F. In order to remove the latent heat from the combustion products, the heating water must enter the boiler below that temperature and be of adequate flow to remain below that temperature while absorbing the latent heat in that section of the heat exchanger. (See section 19.5.)

Medium-temperature water (MTW) and high-temperature water (HTW) systems. MTW systems operate at temperatures between 250 and 350°F, with pressures not exceeding 160 psig. The usual design supply temperature is approximately 250 to 325°F, with a usual pressure rating of 150 psig for boilers and equipment.

HTW systems operate at temperatures over 350°F and usually at pressures of about 300 psig. The maximum design supply water temperature is usually about 400°F, with a pressure rating for boilers and equipment of about 300 psig. The pressure-temperature rating of each component must be checked against the system's design characteristics. The use of MTW and HTW systems is usually limited to large campus or district-type distribution systems.

Chilled water (CW) system. This hydronic cooling system normally operates with a design supply water temperature of 40 to 55°F, usually 44 or 45°F, and at a pressure of up to 120 psig. Antifreeze or brine solutions may be used for applications (usually process or low-dew-point applications) that require temperatures below 40°F or for coil

freeze protection. Direct well water systems can use supply temperatures of 60°F or higher.

Dual-temperature water (DTW) system. This hydronic combination heating and cooling system circulates hot and/or chilled water through common piping and terminal heat transfer apparatus. These systems operate within the pressure and temperature limits of LTW systems, with usual winter design supply water temperatures of about 100 to 150°F and summer supply water temperatures of 40 to 45°F.

The designer of the dual-temperature water system must be aware of the danger of combining different pipe materials or materials not suitable for cold/hot expansion ratios with considerable water temperature swings. For example, plastic ABS cannot be used in shared heating and cooling designs or leaks will develop after the first change of seasons. Many other materials are incompatible for the same reasons.

Terminal heat transfer units include convectors, cast-iron radiators, baseboard and commercial finned-tube units, fan-coil units, chilled beams, radiant panels, snow-melting panels, and air-handling unit coils. A large storage tank may be included in the system to store energy to use when such heat input devices as the boiler or a solar energy collector are not supplying adequate energy.

13.2 Closed Water Systems

Because most hot and chilled water systems are closed, this chapter addresses only closed systems. The fundamental difference between a closed and an open water system is the interface of the water with a compressible gas (such as air) or an elastic surface (such as a diaphragm). A closed water system is defined as one with no more than one point of interface with a compressible gas or surface. This definition is fundamental to understanding the hydraulic dynamics of these systems. Earlier literature referred to a system with an open or vented expansion tank as an "open" system, but such a system is actually a closed system; the atmospheric interface of the tank simply establishes the system pressure at that point.

An open system, on the other hand, has more than one such interface. For example, a cooling tower system has at least two points of interface: the tower basin and the discharge pipe or nozzles entering the tower. One of the major differences in hydraulics between open and closed systems is that certain hydraulic characteristics of open systems cannot occur in closed systems. For example, in contrast to the hydraulics of an open system, in a closed system (1) flow cannot be motivated by static head differences, (2) pumps do not provide static lift, and (3) the entire piping system is always filled with water.

13.2.1 Basic System

Figure 13-1 shows the fundamental components of a closed hydronic system. Actual systems generally have additional components such as valves, vents, regulators, etc., but they are not essential to the basic principles underlying the system.

These fundamental components are

- Loads
- Source
- Expansion chamber
- Pump
- Distribution system

Theoretically, a hydronic system could operate with only these five components.

The components are subdivided into two groups: thermal and hydraulic. The thermal components consist of the load, the source, and the expansion chamber. The hydraulic components consist of the distribution system, the pump, and the expansion chamber. The expansion chamber is the only component that serves both a thermal and a hydraulic function.

13.2.2 Thermal Components

Loads. The load is the point where heat flows out of or into the system to or from the space or process; it is the independent variable to which the remainder of the system must respond. Outward heat flow characterizes a heating system, and inward heat flow characterizes a cooling system. The quantity of heating or cooling is calculated by one of the following means.

Sensible heating or cooling. The rate of heat entering or leaving an airstream is expressed as follows:

$$q = 60 Q_a \rho_a c_p \Delta t \tag{13-1}$$

where

q = heat transfer rate to or from air, Btu/h
Q_a = airflow rate, cfm
ρ_a = density of air, lb/ft^3
c_p = specific heat of air, Btu/lb·°F
Δt = temperature increase or decrease of air, °F

For standard air with a density of 0.075 lb/ft^3 and a specific heat of 0.244 Btu/lb·°F, Equation (13-1) becomes

$$q = 1.1 Q_a \Delta t \tag{13-2}$$

The heat exchanger or coil must then transfer this heat from or to the water. The rate of sensible heat transfer to or from the heated or cooled medium in a specific heat

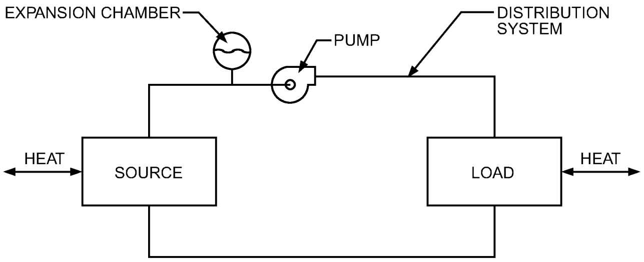


Fig. 13-1 Hydronic System—Fundamental Components
(Figure 1, Chapter 13, 2012 ASHRAE Handbook—HVAC Systems and Equipment)

exchanger is a function of the heat transfer surface area, the mean temperature difference between the water and the medium, and the overall heat transfer coefficient, which itself is a function of the fluid velocities, properties of the medium, geometry of the heat transfer surfaces, and other factors. The rate of heat transfer may be expressed by

$$q = UA(\text{LMTD}) \tag{13-3}$$

where

q = heat transfer rate through heat exchanger, Btu/h
U = overall coefficient of heat transfer, Btu/h·ft^2·°F
A = heat transfer surface area, ft^2
LMTD= logarithmic mean temperature difference, heated or cooled medium to water, °F

Cooling and dehumidification. The rate of heat removal from the cooled medium when both sensible cooling and dehumidification are present is expressed by

$$q_t = w\Delta h \tag{13-4}$$

where

q_t = total heat transfer rate from cooled medium, Btu/h
w = mass flow rate of cooled medium, lb/h
Δh = enthalpy difference between entering and leaving conditions of cooled medium, Btu/lb

Expressed for an air-cooling coil, this equation becomes

$$q_t = 60Q_a\rho_a\Delta h \tag{13-5}$$

which, for standard air with a density of 0.075 lb/ft^3, reduces to

$$q_t = 4.5Q_a\Delta h \tag{13-6}$$

Heat transferred to or from water. The rate of heat transfer to or from the water is a function of the flow rate, the specific heat, and the temperature rise or drop of the water as it passes through the heat exchanger. The heat transferred to or from the water is expressed by

$$q_w = mc_p\Delta t \tag{13-7}$$

where

q_w = heat transfer rate to or from water, Btu/h
m = mass flow rate of water, lb/h
c_p = specific heat of water, Btu/lb·°F
Δt = water temperature increase or decrease across unit, °F

With water systems, it is common to express the flow rate as volumetric flow, in which case Equation (13-7) becomes

$$q_w = 8.02\rho_w c_p Q_w\Delta t \tag{13-8}$$

where

Q_w = water flow rate, gpm
ρ_w = density of water, lb/ft^3

For standard conditions in which the density is 62.4 lb/ft^3 and the specific heat is 1 Btu/lb·°F, Equation (13-8) becomes

$$q_w = 500Q_w\Delta t \tag{13-9}$$

Equation (13-8) or (13-9) can be used to express the heat transfer across a single load or source device, or any quantity of such devices connected across a piping system. In the design or diagnosis of a system, the load side may be balanced with the source side using these equations.

Heat carrying capacity of piping. Equations (13-8) and (13-9) are also used to express the heat carrying capacity of the piping or distribution system or any portion thereof. When the existing temperature differential Δt, sometimes called the temperature range, is identified for any flow rate Q_w through the piping, q_w is called the heat carrying capacity.

Load systems can be any system in which heat is conveyed to or from the water for heating or cooling the space or process. Most load systems are basically a water-to-air finned-coil heat exchanger or a water-to-water exchanger. The specific configuration is usually used to describe the load device. The most common configurations include the following:

Heating load devices
Preheat coils in central units
Heating coils in central units
Zone or central unit reheat coils
Finned-tube radiation
Baseboard radiation
Convectors
Unit heaters
Fan-coil units
Induction unit and chilled beam coils
Water-to-water heat exchangers
Radiant heating panels
Snow-melting panels

Cooling load devices
Coils in central units
Fan-coil units
Induction unit and chilled beam coils
Radiant cooling panels
Water-to-water heat exchangers

Source. The source is the point where heat is added to (heating) or removed from (cooling) the system. Ideally, the amount of energy entering or leaving the source equals the amount entering or leaving through the load system(s). Under steady-state conditions, the load energy and source energy are equal and opposite. Also, when properly measured or calculated, temperature differentials and flow rates across the source and loads are all equal. Equations (13-8) and (13-9) are used to express the source capacities as well as the load capacities.

Any device that can be used to heat or cool water under controlled conditions can be used as a source device. The most common source devices for heating and cooling systems are the following:

Heating source devices
Hot water generator or boiler
Steam-to-water heat exchanger

Water-to-water heat exchanger
Solar heating panels
Heat recovery or salvage heat device, (e.g., water jacket of an internal combustion engine)
Exhaust gas heat exchanger
Incinerator heat exchanger
Heat pump condenser
Air-to-water heat exchanger (heat recovery coil)

Cooling source devices

Vapor compression chiller
Thermal absorption chiller
Heat pump evaporator
Air-to-water heat exchanger (heat recovery coil)
Water-to-water heat exchanger

The two primary considerations in selecting a source device are the design capacity and the part-load capability, sometimes called the turndown ratio. The turndown ratio, expressed in percent of design capacity, is

$$\text{Turndown ratio} = 100\,\frac{\text{Minimum capacity}}{\text{Design capacity}} \qquad (13\text{-}10)$$

The reciprocal of the turndown ratio is sometimes used (for example, a turndown ratio of 25% may also be expressed as a turndown ratio of 4:1).

The turndown ratio has a significant effect on performance; not considering the source system's part-load capability has been responsible for many systems that either do not function properly or do so at the expense of excess energy consumption. The turndown ratio has a significant impact on the ultimate equipment and/or system design selection.

System Temperatures. Design temperatures and temperature ranges are selected by considering the performance requirements and the economics of the components. For a cooling system that must maintain 50% rh at 75°F, the dew-point temperature is 55°F, which sets the maximum return water temperature at about 55°F (60°F maximum); on the other hand, the lowest reasonable temperature for refrigeration, considering the freezing point, energy consumption, and economics, is about 40°F. This temperature spread then sets constraints for a chilled water system.

For a heating system, the maximum hot water temperature is normally established by the ASME low-pressure code as 250°F, and with space temperature requirements of little over 75°F, the actual operating supply temperatures and the temperature ranges are set by the design of the load devices. The most economical systems related to distribution and pumping favor the use of the maximum possible temperature range (Δt). However, for better boiler fuel efficiency, the condensing boiler temperature limits may prevail. For systems with condensing boilers, the maximum temperature is around 130°F.

Expansion Chamber. The expansion chamber (also called an expansion or compression tank) serves both a thermal function and a hydraulic function. In its thermal function the tank provides a space into which the noncompressible liquid can expand or from which it can contract as the liquid undergoes volumetric changes with changes in temperature. To allow for this expansion or contraction, the expansion tank provides an interface point between the system fluid and a compressible gas. Note: By definition, a closed system can have only one such interface; thus, a system designed to function as a closed system should have only one expansion chamber.

Expansion tanks are of three basic configurations: (1) a closed tank, which contains a captured volume of compressed air and water, with an air water interface (sometimes called a plain steel tank); (2) an open tank (i.e., a tank open to the atmosphere); and (3) a diaphragm tank, in which a flexible membrane is inserted between the air and the water (another configuration of a diaphragm tank is the bladder tank).

In the plain steel tank and the open tank, gases can enter the water through the interface and can adversely affect performance. Thus, current design practice normally uses diaphragm or bladder tanks.

Sizing the tank is a primary thermal concern when designing the system. However, prior to sizing the tank, the control or elimination of air must be considered. The amount of air that will be absorbed and can be held in solution with the water is expressed by Henry's equation (Pompei 1981):

$$x = p/H \qquad (13\text{-}11)$$

where

x = solubility of air in water (% by volume)
p = absolute pressure
H = Henry's constant

Henry's constant, however, is constant only for a given temperature (Figure 13-2). Combining the data of Figure 13-2 (Himmelblau 1960) with Equation (13-11) results in the solubility diagram of Figure 13-3. With that diagram, the solubility can be determined if the temperature and pressure are known.

If the water is not saturated with air, it will absorb air at the air/water interface until the point of saturation has been reached. Once absorbed, the air moves through the water either by mass migration or by molecular diffusion until the water is uniformly saturated.

If the air/water solution changes to a state that reduces solubility, the excess air will be released as a gas. For example, if the air/water interface is at a high pressure, the water will absorb air to its limit of solubility at that point; if at another point the pressure is less, some of the dissolved air will be released. In the design of systems with open or plain steel expansion tanks, the tank is commonly used as the major air control or release point in the system.

The following equations are used to size the three common configurations of expansion tanks (Coad 1980a):

For closed tanks with air/water interface,

$$V_t = V_s \frac{[(v_2 / v_1) - 1] - 3\alpha\Delta t}{(P_a / P_1) - (P_a - P_2)} \qquad (13\text{-}12)$$

For open tanks with air/water interface,

$$V_t = 2\{V_s[(v_2 / v_1) - 1] - 3\alpha\Delta t\} \tag{13-13}$$

For diaphragm tanks,

$$V_t = V_s \frac{[(v_2/v_1) - 1] - 3\alpha\Delta t}{1 - (P_1/P_2)} \tag{13-14}$$

where

V_t = volume of expansion tank, gal
V_s = volume of water in system, gal
t_1 = lower temperature, °F
t_2 = higher temperature, °F
P_a = atmospheric pressure, psia
P_1 = pressure at lower temperature, psia
$P2$ = pressure at higher temperature, psia
v_1 = specific volume of water at lower temperature, ft^3/lb
v_2 = specific volume of water at higher temperature, ft^3/lb
α = linear coefficient of thermal expansion, in/in °F
= 6.5×10^{-6} in./in. °F for steel
= 9.5×10^{-6} in./in. °F for copper
Δt = $(t_2 - t_1)$, °F

As an example, the lower temperature for a heating system is usually normal ambient temperature at fill conditions (e.g., 50°F) and the higher temperature is the operating supply water temperature for the system. For a chilled water system, the lower temperature is usually the design chilled water supply temperature, and the higher temperature is ambient temperature (e.g., 95°F). However, in very large central systems that remain at operating temperatures the Δt is quite small since it is the average system water temperature, which remains almost constant except for central variance. For a dual-temperature hot/chilled system, the lower temperature is the chilled water design supply temperature, and the higher temperature is the heating water design supply temperature.

For specific volume and saturation pressure of water at various temperatures, see Table 3 in Chapter 1 of the 2017 *ASHRAE Handbook—Fundamentals*, or any other comprehensive steam table.

At the tank connection point, the pressure in closed tank systems increases as the water temperature increases. Pressures at the expansion tank are generally set by the following parameters:

- The pressure at the lower temperature is usually selected to hold a positive pressure at the highest point in the system (usually about 10 psig).
- The pressure at the higher temperature is normally set by the maximum pressure allowable at the location of the safety relief valve(s) without opening them.

Other considerations are to ensure that (1) the pressure at no point in the system will ever drop below the saturation pressure at the operating system temperature and (2) all pumps have sufficient net positive suction head (NPSH) available to prevent cavitation.

Example 13-1 Size an expansion tank for a water heating system that will operate at 180 to 220°F. The minimum pressure at the tank is 10 psig (24.7 psia) and the maximum pressure is 25 psig (39.7 psia). (Atmospheric pressure is

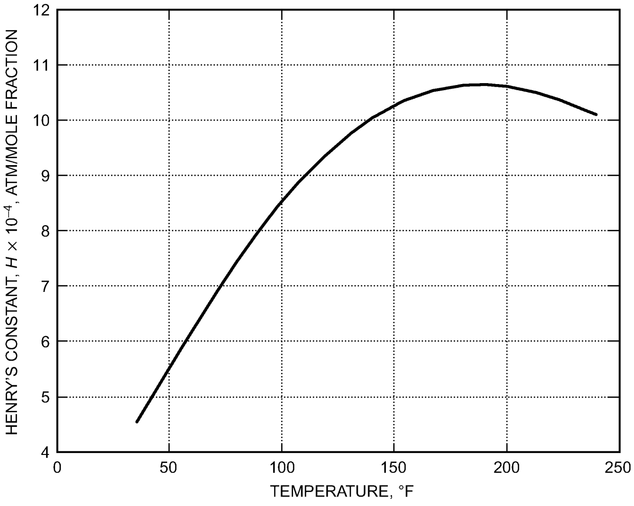


Fig. 13-2 Henry's Constant Versus Temperature for Air and Water
(Coad 1980a)
(Figure 2, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

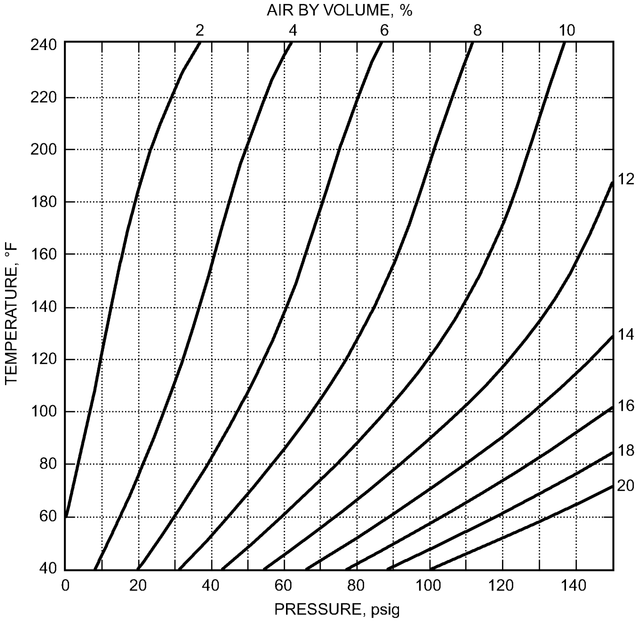


Fig. 13-3 Solubility Versus Temperature and Pressure for Air/Water Solutions
(Figure 3, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

14.7 psia.) The volume of water is 3000 gal. The piping is steel.

1. Calculate the required size for a closed tank with an air/water interface.

Solution: For lower temperature t_1, use 40°F.

From Table 3 in Chapter 1 of the 2017 *ASHRAE Handbook—Fundamentals*,

$$v_1 \text{ (at 40°F)} = 0.01602 \text{ ft}^3\text{/lb and } v_2 \text{ (at 220°F)} = 0.01677 \text{ ft}^3\text{/lb}$$

Using Equation (13-12),

$$V_t = 3000\frac{[(0.01677/0.01602)-1]-3(6.5\times10^{-6})(220-40)}{(14.7/24.7)-(14.7/39.7)} = 578 \text{ gal}$$

If a diaphragm tank were to be used in lieu of the plain steel tank, what tank size would be required?

Solution: Using Equation (13-14),

$$V_t = 3000\frac{[(0.01677/0.01602)-1]-3(6.5\times10^{-6})(220-40)}{1-(24.7/39.7)} = 344 \text{ gal}$$

13.2.3 Hydraulic Components

Distribution System. The distribution system is the piping system connecting the various other components of the system. The primary considerations in designing this system are (1) sizing the piping to handle the heating or cooling capacity required, (2) arranging the piping to ensure flow in the quantities required at design conditions and at all other loads, and (3) assuring that the air is purged from the system so the water can flow freely.

The flow requirement of the pipe is determined by Equation (13-8) or (13-9). After Δt is established based on the thermal requirements, either of these equations (as applicable) can be used to determine the flow rate. First-cost economics and energy consumption make it advisable to design for the greatest practical Δt because the flow rate is inversely proportional to Δt; that is, if Δt doubles, the flow rate is reduced by half.

The three related variables in sizing the pipe are flow rate, pipe size, and pressure drop. The primary consideration in selecting a design pressure drop is the relationship between the economics of first cost and energy costs.

Once the distribution system is designed, the pressure loss at design flow is calculated by the methods discussed in Chapter 9 in this book or in Chapter 22 of the 2017 *ASHRAE Handbook—Fundamentals*. The relationship between flow rate and pressure loss can be expressed by

$$Q = C_v\sqrt{\Delta p} \qquad (13\text{-}15)$$

where

Q = system flow rate, gpm
Δp = pressure drop in system, psi
C_v = system constant (sometimes called valve coefficient, which is discussed in Chapter 47 in 2016 *ASHRAE Handbook—HVAC Systems and Equipment*)

Equation (13-15) may be modified as follows:

$$Q = C_s\sqrt{\Delta h} \qquad (13\text{-}16)$$

where

Δh = system head loss, ft of fluid [$\Delta h = \Delta p/\rho$]
Cs = system constant
[$C_s = 0.67\ C_v$ for water with a density = 62.4 lb/ft^3]

Equations (13-15) and (13-16) are the system constant form of the Darcy-Weisbach equation. If the flow rate and head loss are known for a system, Equation (13-16) may be used to calculate the system constant C_s. From this calculation, the pressure loss can be determined at any other flow rate. Equation (13-16) can be graphed as a system curve (Figure 13-4).

The system curve changes if anything occurs that changes the flow/pressure drop characteristics. Examples of this are a strainer that starts to block or a control valve closing, either of which increases the head loss at any given flow rate, thus changing the system curve in a direction from curve A to curve B in Figure 13-4.

Pump or Pumping System. Centrifugal pumps are the type most commonly used in hydronic systems (see Chapter 44 in 2016 *ASHRAE Handbook—HVAC Systems and Equipment* or Chapter 9 in this book). Circulating pumps used in water systems can vary in size from small in-line circulators delivering 5 gpm at 6 or 7 ft head to base-mounted or vertical pumps handling hundreds or thousands of gallons per minute, with pressures limited only by the characteristics of the system. Pump operating characteristics must be carefully matched to system operating requirements.

Pump Curves and Water Temperature. Performance characteristics of centrifugal pumps are described by pump curves, which plot flow versus head or pressure, as well as by efficiency and power information. The point at which a pump operates is the point at which the pump curve intersects the system curve (Figure 13-5).

A complete piping system follows the same water flow/pressure drop relationships as any component of the system [see Equation (13-16)]. Thus, the pressure required for any proposed flow rate through the system may be determined and a system curve constructed. A pump may be selected by using the calculated system pressure at the design flow rate as the base point value.

Figure 13-6 illustrates how a shift of the system curve to the right affects system flow rate. This shift can be caused by incorrectly calculating the system pressure drop by using arbitrary safety factors or overstated pressure drop charts. Variable system flow caused by control valve operation or

improperly balanced systems (subcircuits having substantially lower pressure drops than the longest circuit) can also cause a shift to the right.

Pumps for closed-loop piping systems should have a flat pressure characteristic and should operate slightly to the left of the peak efficiency point on their curves. This characteristic permits the system curve to shift to the right without causing undesirable pump operation, overloading, or reduction in available pressure across circuits with large pressure drops.

Many dual-temperature systems are designed so that the chillers are bypassed during the winter months. The chiller pressure drop, which may be quite high, is thus eliminated from the system pressure drop, and the pump operating point shift to the right may be quite large. For such systems, system curve analysis should be used to check winter operating points.

Operating points may be highly variable, depending on (1) load conditions, (2) the types of control valves used, and (3) the piping circuitry and heat transfer elements. In general, the best selection will be:

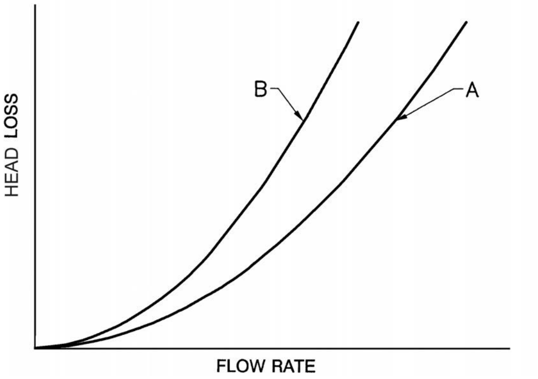


Fig. 13-4 Typical System Curves for Closed System

- For design flow rates calculated using pressure drop charts that illustrate actual closed-loop hydronic system piping pressure drops
- To the left of the maximum efficiency point of the pump curve to allow shifts to the right caused by system circuit unbalance, direct-return circuitry applications, and modulating three-way valve applications
- A pump with a flat curve to compensate for unbalanced circuitry and to provide a minimum pressure differential increase across two-way control valves

Parallel Pumping. When pumps are applied in parallel, each pump operates at the same head, and provides its share of the system flow at that pressure (Figure 13-7). Generally, pumps of equal size are used, and the parallel pump curve is established by doubling the flow of the single pump curve (with identical pumps).

Plotting a system curve across the parallel pump curve shows the operating points for both single and parallel pump operation (Figure 13-7). Note that single-pump operation does not yield 50% flow. The system curve crosses the single pump curve considerably to the right of its operating point when both pumps are running. This leads to two important concerns: (1) the pumps must be powered to prevent overloading during single-pump operation, and (2) a single pump can provide standby service of up to 80% of design flow; the actual amount depends on the specific pump curve and system curve.

Series Pumping. When pumps are operated in series, each pump operates at the same flow rate and provides its share of the total pressure at that flow. A system curve plotted across the series pump curve shows the operating points for both single and series pump operation (Figure 13-8). Note that the single pump can provide up to 80% flow for standby and at a lower power requirement.

Series pump installations are often used in two-pipe heating and cooling systems so that both pumps operate during the cooling season to provide maximum flow and head, while

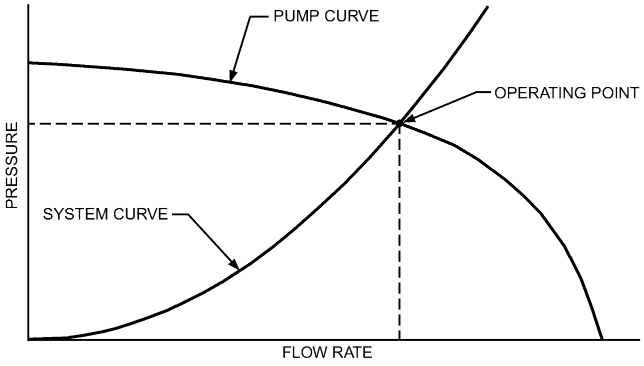


Fig. 13-5 Pump Curve and System Curve
(Figure 5, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

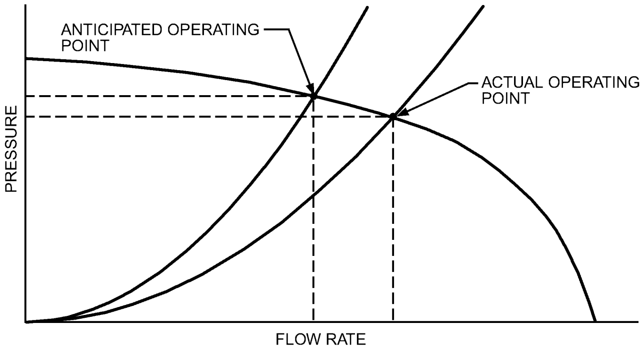


Fig. 13-6 Shift of System Curve due to Circuit Unbalance
(Figure 6, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

only a single pump operates during the heating season. Note that both parallel and series pump applications require that the actual pump operating points be used to accurately determine the pumping point. Adding artificial safety factor head, using improper pressure drop charts, or incorrectly calculating pressure drops may lead to an unwise selection.

Multiple-Pump Systems. Care must be taken in designing systems with multiple pumps to ensure that if pumps ever operate in either parallel or series, such operation is fully understood and considered by the designer. Pumps performing unexpectedly in series or parallel have been the cause of performance problems in hydronic systems. Typical problems resulting from pumps functioning in parallel and series when not anticipated by the designer are the following.

Parallel. With pumps of unequal pressures, one pump may create a pressure across the other pump in excess of its cutoff pressure, causing flow through the second pump to diminish significantly or to cease. This phenomenon can cause flow problems or pump damage.

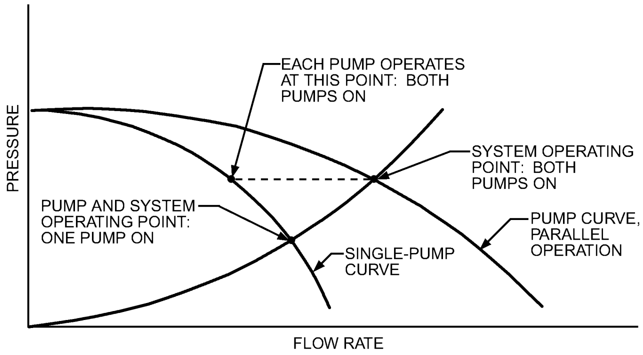


Fig. 13-7 Operating Conditions for Parallel Pump Installation

(Figure 8, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

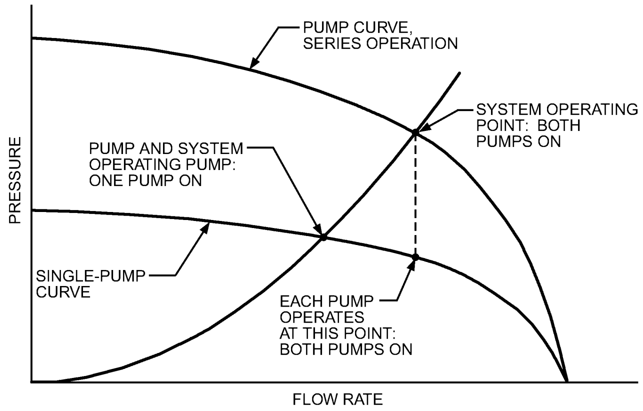


Fig. 13-8 Operating Conditions for Series Pump Installation

(Figure 9, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Series. With pumps of different flow capacities, the pump of greater capacity may overflow the pump of lesser capacity, which could cause damaging cavitation in the smaller pump and could actually cause a pressure drop rather than a pressure rise across that pump. In other circumstances, unexpected series operation can cause excessively high or low pressures that can damage system components.

Standby Pump Provision. If total flow standby capacity is required, a properly valved standby pump of equal capacity is installed to operate when the normal pump is inoperable. A single standby may be provided for several similarly sized pumps. Parallel or series pump installation can provide up to 80% standby, as stated above, which is often sufficient.

Compound Pumping. In larger systems, compound pumping, also known as primary-secondary pumping, is often employed to provide system advantages that would not be available with a single pumping system. The concept of compound pumping is illustrated in Figure 13-9.

In Figure 13-9, Pump No. 1 can be referred to as the source or primary pump and Pump No. 2 as the distribution or secondary pump. The short section of pipe between A and B is called the common pipe because it is common to both the source and distribution circuits. Other terms used for the common pipe are the decoupling line and the neutral bridge. In the design of compound systems, the common pipe should be kept as short and as large in diameter as practical to minimize the pressure loss between those two points. Care must be taken, however, to ensure adequate length in the common pipe to prevent recirculation from entry or exit turbulence. There should never be a valve or check valve in the common pipe. If these conditions are met and the pressure loss in the common pipe can be assumed to be zero, then neither pump will affect the other. Then, except for the system static pressure at any given point, the circuits can be designed and analyzed and will function dynamically independently of one another.

In Figure 13-9, if Pump No. 1 has the same flow capacity in its circuit as Pump No. 2 has in its circuit, all of the flow entering Point A from Pump No. 1 will leave in the branch supplying Pump No. 2, and no water will flow in the common

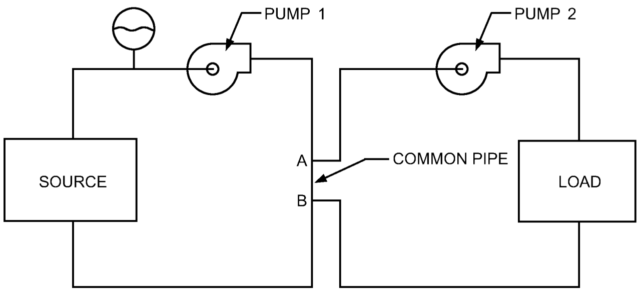


Fig. 13-9 Compound Pumpimg (Primary-Secondary Pumping)

(Figure 10, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

pipe. Under this condition, the water entering the load will be at the same temperature as that leaving the source.

If the flow capacity of Pump No. 1 exceeds that of Pump No. 2, some water will flow downward in the common pipe. Under this condition, Tee A is a diverting tee, and Tee B becomes a mixing tee. Again, the temperature of the fluid entering the load is the same as that leaving the source. However, because of the mixing taking place at Point B, the temperature of the water returning to the source is between the source supply temperature and the load return temperature.

On the other hand, if the flow capacity of Pump No. 1 is less than that of Pump No. 2, then Point A becomes a mixing point because some water must recirculate upward in the common pipe from Point B. Under this condition, the temperature of the water entering the load is between the supply water temperature from the source and the return water temperature from the load.

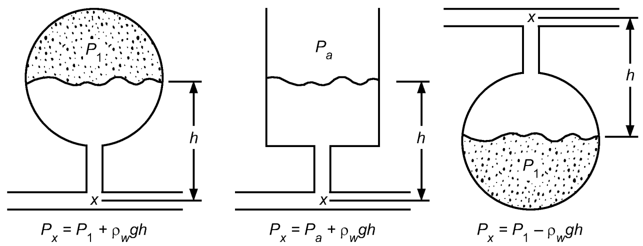
For example, if Pump No. 1 circulates 25 gpm of water leaving the source at 200°F, and Pump No. 2 circulates 50 gpm of water leaving the load at 100°F, then the water temperature entering the load is

$$t_{\text{load}} = 200 - (25/50)(200 - 100) = 150°\text{F}$$

The following are some advantages of compound circuits:

1. They enable the designer to achieve different water temperatures and temperature ranges in different elements of the system.
2. They decouple the circuits hydraulically, thereby making the control, operation, and analysis of large systems much less complex. Hydraulic decoupling also prevents unwanted series or parallel operation.
3. Circuits can be designed for different flow characteristics. For example, a chilled water load system can be designed with two-way valves for better control and energy conservation while the source system operates at constant flow to protect the water in the evaporator from freezing.

Expansion Chamber. As a hydraulic device, the expansion tank serves as the reference pressure point in the system, analogous to a ground in an electrical system (Lockhart and Carlson 1953). Where the tank connects to the piping, the pressure equals the pressure of the air in the tank plus or minus any fluid pressure due to the elevation difference between the tank liquid surface and the pipe (Figure 13-10).



A. CLOSED TANK AIR/ WATER INTERFACE B. OPEN TANK C. DIAPHRAGM TANK

Fig. 13-10 Tank Pressure Related to "System" Pressure
(Figure 15, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

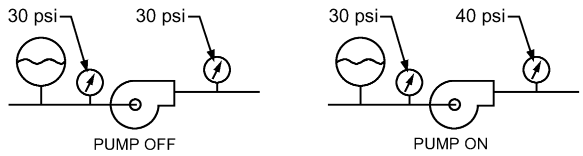
As previously stated, a closed system should have only one expansion chamber. The presence of more than one chamber or of excessive amounts of undissolved air in a piping system can cause the closed system to behave in unexpected (but understandable) ways, causing extensive damage from shock waves or water hammer.

With a single chamber on a system, assuming isothermal conditions for the air, the air pressure can change only as a result of displacement by the water. The only thing that can cause the water to move into or out of the tank (assuming no water is being added to or removed from the system) is expansion or contraction of the water in the system. Thus, in sizing the tank, thermal expansion is related to the pressure extremes of the air in the tank [Equations (13-12), (13-13), and (13-14)].

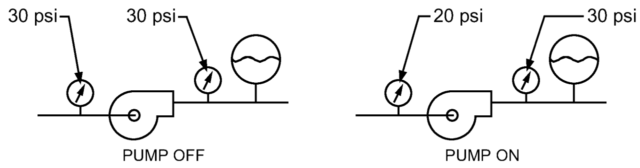
The point of connection of the tank should be based on the pressure requirements of the system, remembering that the pressure at the tank connection will not change as the pump is turned on or off. For example, consider a system containing an expansion tank at 30 psig and a pump with a pump head of 23.1 ft (10 psig). Figure 13-11 shows alternative locations for connecting the expansion tank; in either case, with the pump off, the pressure will be 30 psig on both the pump suction and discharge. With the tank on the pump suction side, when the pump is turned on, the pressure increases on the discharge side by an amount equal to the pump pressure (Figure 13-11A). With the tank connected on the discharge side of the pump, the pressure decreases on the suction side by the same amount (Figure 13-11B).

Other considerations relating to the tank connection include the following:

- A tank open to the atmosphere must be located above the highest point in the system.



A. TANK ON PUMP SUCTION SIDE



B. TANK ON PUMP DISCHARGE SIDE

Fig. 13-11 Effect of Expansion Tank Location with Respect to Pump Pressure
(Figure 16, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

- A tank with an air/water interface is generally used with an air control system that continually revents the air into the tank. For this reason, it should be connected at a point where air can best be released.
- Within reason, the lower the pressure in a tank, the smaller is the tank [see Equations (13-12) and (13-14)]. Thus, in a vertical system, the higher the tank is placed, the smaller it can be.

13.3 Design Considerations

13.3.1 Piping Circuits

Hydronic systems are designed with many different configurations of piping circuits. In addition to simple preference by the design engineer, the method of arranging the circuiting can be dictated by such factors as the shape or configuration of the building, the economics of installation, energy economics, the nature of the load, part-load capabilities or requirements, and others.

Each piping system is a network; the more extensive the network, the more complicated it is to understand, analyze, or control. Thus, a major design objective is to maintain the highest degree of simplicity.

Load distribution circuits are of four general types:

- Full series
- Diverting series
- Parallel direct return
- Parallel reverse return

Series Circuit. A simple series circuit is shown in Figure 13-12. Series loads generally have the advantage of both lower piping costs and higher temperature drops that result in smaller pipe sizes and lower energy consumption. A disadvantage is that the different circuits cannot be controlled separately. Simple series circuits are generally limited to residential and small commercial standing radiation heating systems. Figure 13-13 shows a typical layout of such a system with two zones for residential or small commercial heating.

Diverting Series. The simplest diverting series circuit diverts some of the flow from the main piping circuit through a special diverting tee to a load device (usually standing radiation) that has a low pressure drop. This system is generally limited to heating systems in residential or small commercial applications.

Figure 13-14 illustrates a typical one-pipe diverting tee circuit. For each terminal unit, a supply and a return tee are installed on the main. One of the two tees is a special diverting tee that creates a pressure drop in the main flow to divert part of the flow to the unit. One (return) diverting tee is usually sufficient for upfeed (units above the main) systems. Two special fittings (supply and return tees) are usually required to overcome thermal pressure in downfeed units. Special tees are proprietary; therefore, manufacturer's literature must be consulted for flow rates and pressure drop data on these devices. Unit selection can be only approximate without these data.

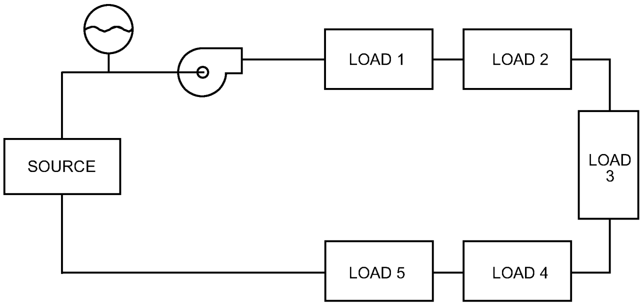


Fig. 13-12 Flow Diagram of Simple Series Circuit
(Figure 17, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

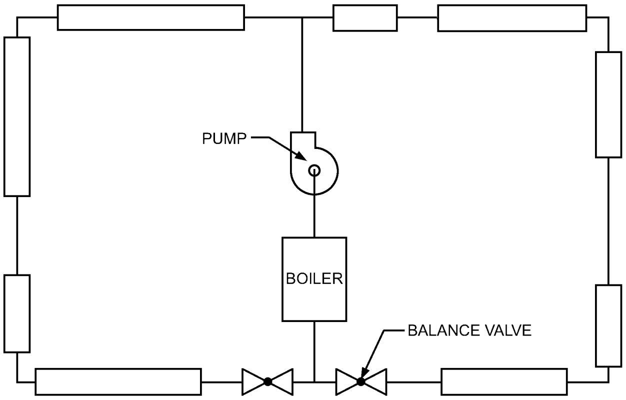


Fig. 13-13 Series Loop System
(Figure 18, Chapter 13, 2016ASHRAE Handbook—HVAC Systems and Equipment)

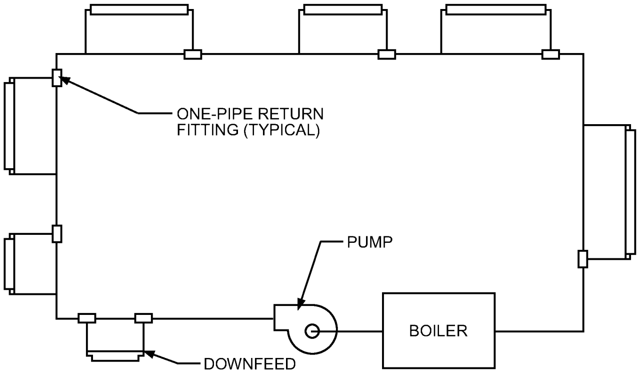


Fig. 13-14 One-Pipe Diverting Tee System
(Figure 19, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

One-pipe diverting series circuits allow manual or automatic control of flow to individual heating units. On-off rather than flow modulation control is preferred because of the relatively low pressure drop allowable through the control valve in the diverted flow circuit. This system is likely to cost more than the series loop because extra branch pipe and fittings, including special tees, are required. Each unit usually requires a manual air vent because of the low water velocity through the unit. The length and load imposed on a one-pipe circuit are usually small because of these limitations.

Because only a fraction of the main flow is diverted in a one-pipe circuit, the flow rate and pressure drop are less variable. When two or more one-pipe circuits are connected to the same two-pipe mains, the circuit flow will need to be mechanically balanced. After balancing, sufficient flow must be maintained in each one-pipe circuit to ensure adequate flow diversion to the loads.

When coupled with compound pumping systems, series circuits can be applied to multiple control zones on larger commercial or institutional systems (Figure 13-15). Note that in the series circuit with compound pumping, the load pumps need not be equal in capacity to the system pump. If, for example, load pump LP-1 circulates less flow (Q_{LP1}) than system pump SP-1 (Q_{SP1}), the temperature difference across Load 1 would be greater than the circuit temperature difference between A and B (i.e., water would flow in the common pipe from A to B). If, on the other hand, the load pump LP-2 is equal in flow capacity to the system pump SP-1, the temperature differentials across Load 2 and across the system from C to D would be equal and no water would flow in the common pipe. If Q_{LP3} exceeds Q_{SP1}, mixing occurs at Point E and, in a heating system, the temperature of the water entering pump LP-3 would be lower than that available from the system leaving load connection D.

Thus, a series circuit using compound or load pumps offers many design options. Each of the loads shown in Figure 13-15 could also be a complete piping circuit or network.

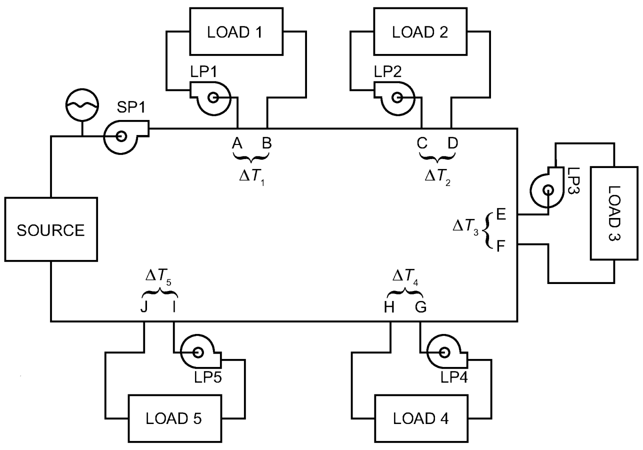


Fig. 13-15 Series Circuit with Load Pumps
(Figure 20, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Parallel Piping. These networks are the most commonly used in hydronic systems because they allow the same temperature water to be available to all loads. The two types of parallel networks are direct return and reverse return (Figure 13-16).

In the direct-return system, the length of supply and return piping through the subcircuits is unequal, which may cause unbalanced flow rates and require careful balancing to provide each subcircuit with design flow. Ideally, the reverse-return system provides nearly equal total lengths for all terminal circuits.

Direct-return piping has been successfully applied where the designer has guarded against major flow unbalance by:

1. Providing for pressure drops in the subcircuits or terminals that are significant percentages of the total, usually establishing pressure drops for close subcircuits at higher values than those for the far subcircuits by use of balancing valves.
2. Minimizing distribution piping pressure drop (in the limit, if the distribution piping loss is zero and the loads are of equal flow resistance, the system is inherently balanced).
3. Including balancing devices and some means of measuring flow at each terminal or branch circuit
4. Using control valves with a high head loss at the terminals

13.3.2 Capacity Control of Load System

The two alternatives for controlling the capacity of hydronic systems are on-off control and variable-flow or modulating control. The on-off option is generally limited to smaller systems or components (e.g., residential or small commercial) and individual components of larger systems. In smaller systems where the entire building is a single zone, control is accomplished by cycling the source device (the boiler or chiller) on and off. Usually a space thermostat allows the chiller or boiler to run, then a water temperature thermostat (aquastat) controls the capacity of the chiller(s) or

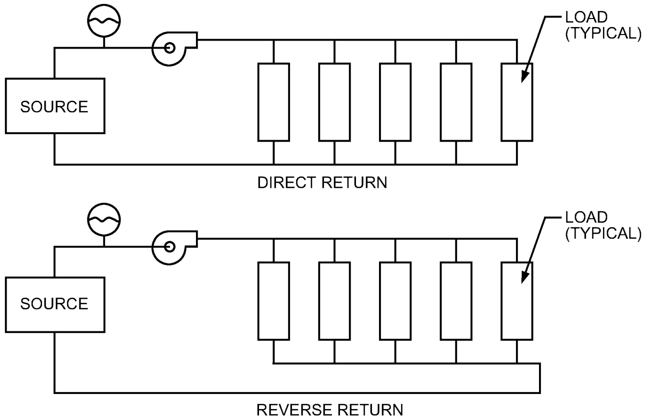


Fig. 13-16 Direct- and Reverse-Return Two-Pipe Systems
(Figure 21, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

boiler(s) as a function of supply or return water temperature. The pump can be either cycled with the load device (usually the case in a residential heating system) or left running (usually done in commercial hot or chilled water systems).

In these single-zone applications, the piping design requires no special consideration for control. Where multiple zones of control are required, the various load devices are controlled first, then the source system capacity is controlled to follow the capacity requirement of the loads.

Control valves are commonly used to control loads. These valves control the capacity of each load by varying the amount of water flow through the load device when load pumps are not used. Control valves for hydronic systems are straight-through (two-way) valves and three-way valves (Figure 13-17). The effect of either valve is to vary the amount of water flowing through the load device.

With a two-way valve (Figure 13-17A), as the valve strokes from full-open to full-closed, the quantity of water flowing through the load gradually decreases from design flow to no flow. With a three-way mixing valve (Figure 13-17B) in one position, the valve is open from Port A to AB, with Port B closed off. In that position, all the flow is through the load. As the valve moves from the A-AB position to the B-AB position, some of the water bypasses the load by flowing through the bypass line, thus decreasing flow through the load. At the end of the stroke, Port A is closed, and all of the fluid will flow from B to AB with no flow through the load. Thus, the three-way mixing valve has the same effect on the load as the two-way valve—as the load reduces, the quantity of water flowing through the load decreases.

The effect on load control with the three-way diverting valve (Figure 13-17C) is the same as with the mixing valve in a closed system—the flow is either directed through the load or through the bypass in proportion to the load. Because of the dynamics of valve operation, diverting valves are more complex in design and are thus more expensive than mixing valves; because they accomplish the same function as the simpler mixing valve, they are seldom used in closed hydronic systems.

In terms of load control, a two-way valve and a three-way valve perform identical functions—they both vary the flow through the load as the load changes. The fundamental difference between the two-way valve and the three-way valve is that as the source or distribution system sees the load, the two-way valve provides a variable flow load response and the three-way valve provides a constant flow load response.

According to Equation (13-9), the load q is proportional to the product of Q and Δt. Ideally, as the load changes, Q changes, while Δt remains fixed. However, as the system sees it, as the load changes with the two-way valve, Q varies and Δt is fixed; whereas with a three-way valve, Δt varies and Q is fixed. This principle is illustrated in Figure 13-18. An understanding of this concept is fundamental to the design or analysis of hydronic systems.

The flow characteristics of two-way and three-way valve ports are described in Chapter 47, "Design and Application of Controls," of the 2015 *ASHRAE Handbook—HVAC Applications* and must be understood. The equal percentage characteristic is recommended for proportional control of the load flow for two-way and three-way valves; the bypass flow port of three-way valves should have the linear characteristic to maintain a uniform flow during part-load operation.

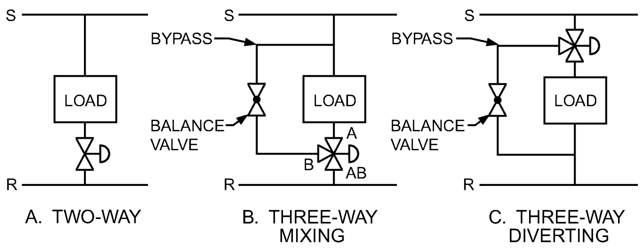
13.3.3 Sizing Control Valves

For stable control, the pressure drop in the control valve at the full-open position should be no less than one-half the pressure drop in the branch. For example, in Figure 13-18 the pressure drop at full-open position for the two-way valve should equal one-half the pressure drop from A to B, and for the three-way valve, the full-open pressure drop should be half that from C to D. The pressure drop in the bypass balancing valve in the three-way valve circuit should be set to equal that in the coil (load).

Control valves should be sized on the basis of the valve coefficient C_V. For more information, see the section on control valve sizing in Chapter 47 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

If a system is to be designed with multiple zones of control such that load response is to be by constant flow through the load and variable Δt, control cannot be achieved by valve control alone; a load pump is required.

Several control arrangements of load pump and control valve configurations are shown in Figure 13-19. Note that in all three configurations the common pipe has no restriction or check valve. In all configurations there is no difference in



A. TWO-WAY

B. THREE-WAY MIXING

C. THREE-WAY DIVERTING

Fig. 13-17 Load Control Valves

(Figure 22, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

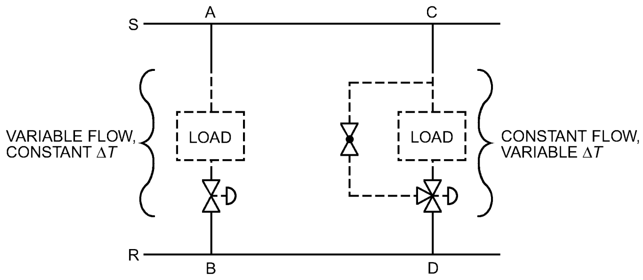


Fig. 13-18 System Flow with Two-Way and Three-Way Valves

(Figure 23, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

control as seen by the load. However, the basic differences in control are:

1. With the two-way modulation valve configuration (Figure 13-19A), the distribution system sees a variable flow and a constant Δt, whereas with both three-way configurations, the distribution system sees a constant flow and a variable Δt.
2. Configuration B differs from C in that the pressure required through the three-way valve in Figure 13-19B is provided by the load pump, while in Figure 13-19C it is provided by the distribution pump(s).

13.3.4 Low-Temperature Heating Systems

These systems are used for heating spaces or processes directly, as with standing radiation and process heat exchangers, or indirectly, through air-handling unit coils for preheating, for reheating, or in hot water unit heaters. These systems have generally been designed with supply water temperatures from 180 to 240°F and temperature drops from 20 to 100°F. With increasing use of condensing boilers, systems with temperature ranges between 110°F and 130°F are becoming more commonly used.

In the United States, hot water heating systems were historically designed for a 200°F supply water temperature and a 20°F temperature drop. This practice evolved from earlier gravity system designs and provides convenient design relationships for heat transfer coefficients related to coil tubing and finned-tube radiation and for calculations (1 gallon per minute conveys approximately 10,000 Btu/h at 20°F Δt). Because many terminal devices still require these flow rates, it is important to recognize this relationship in selecting devices and designing systems.

However, the greater the temperature range (and related lower flow rate) that can be applied, the less costly the system is to install and operate. A lower flow rate requires smaller and less expensive piping, less secondary building space, and smaller pumps. Also, smaller pumps require less electrical energy.

Nonresidential Heating Systems. Possible approaches to enhancing the economics of large heating systems include (1) higher supply temperatures, (2) compound pumping, and (3) terminal equipment designed for smaller flow rates. The three techniques may be used either singly or in combination.

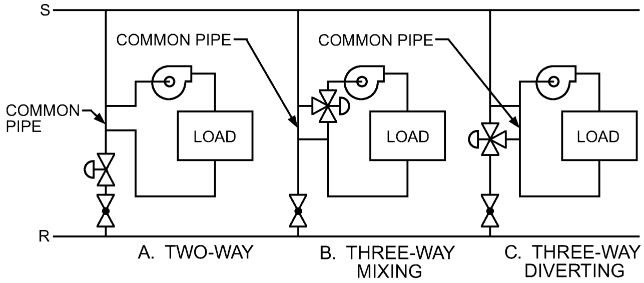


Fig. 13-19 Load Pumps with Valve Control
(Figure 29, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Using higher supply water temperatures achieves higher temperature drops and smaller flow rates. Terminal units with a reduced heating surface can be used. These smaller terminals are not necessarily less expensive, however, because their required operating temperatures and pressures may increase manufacturing costs and the problems of pressurization, corrosion, expansion, and control. System components may not increase in cost uniformly with temperature but rather in steps conforming to the three major temperature classifications. Within each classification, the most economical design uses the highest temperature in that classification.

Primary-secondary or compound pumping reduces the size and cost of the distribution system and also may use larger flows and lower temperatures in the terminal or secondary circuits. A primary pump circulates water in the primary distribution system while one or more secondary pumps circulate the terminal circuits. The connection between primary and secondary circuits provides complete hydraulic isolation of both circuits and permits a controlled interchange of water between the two. Thus, a high supply water temperature can be used in the primary circuit at a low flow rate and high temperature drop, while a lower temperature and conventional temperature drop can be used in the secondary circuit(s).

For example, a system could be designed with primary-secondary pumping in which the supply temperature from the boiler was 240°F, the supply temperature in the secondary was 200°F, and the return temperature was 180°F. This design results in a conventional 20°F Δt in the secondary zones but permits the primary circuit to be sized on the basis of a 60°F drop. This primary-secondary pumping arrangement is most advantageous with terminal units such as convectors and finned radiation, which are generally unsuited for small flow rate design.

A fourth technique is to put certain loads in series utilizing a combination of control valves and compound pumping (Figure 13-20). In the system illustrated, the capacity of the boiler or heat exchanger is 2×10^6 Btu/h, and each of the four loads is 0.5×10^6 Btu/h. Under design conditions, the system is designated for an 80°F water temperature drop, and the loads each provide 20°F of the total Δt. The loads in these systems, as well as the smaller or simpler systems in residential or commercial applications, can be connected in a direct-return or a reverse-return piping system. The different features of each load are as follows:

1. The domestic hot water heat exchanger has a two-way valve and is thus arranged for variable flow (while the main distribution circuit provides constant flow for the boiler circuit).
2. The finned-tube radiation circuit is a 20°F Δt circuit with the design entering water temperature reduced to and controlled at 200°F.

3. The reheat coil circuit takes a 100°F temperature drop for a very low flow rate.
4. The preheat coil circuit provides constant flow through the coil to keep it from freezing.

When loads such as water-to-air heating coils in low-temperature water systems are valve controlled (flow varies), they have a heating characteristic of flow versus capacity as shown in Figure 13-21 for 20°F and 60°F temperature drops, respectively. For a 20°F Δt coil, 50% flow provides approximately 90% capacity; valve control will tend to be unstable. For this reason, proportional temperature control is required, and equal percentage characteristic two-way valves should be selected such that 10% flow is achieved with 50% valve lift. This combination of the valve characteristic and the heat transfer characteristic of the coil makes the control linear with respect to the control signal. This type of control can be obtained only with equal percentage two-way valves and can be further enhanced if piped with a secondary pump arrangement as shown in Figure 13-19A. See Chapter 47 of the 2015 *ASHRAE Handbook—HVAC Applications* for further information on automatic controls.

13.3.5 Chilled-Water Systems

Designers have less latitude in selecting supply water temperatures for cooling applications because there is only a narrow range of water temperatures low enough to provide adequate dehumidification and high enough to avoid chiller freeze-up. Circulated water quantities can be reduced by selecting proper air quantities and heat transfer surface at the terminals. Terminals suited for a 12°F rise rather than an 8°F rise reduce circulated water quantity and pump power by one-third and increase chiller efficiency.

A proposed system should be evaluated for the desired balance between installation cost, operating cost, and energy efficiency. Table 13-1 shows the effect of coil circuiting and chilled water temperature on water flow and temperature rise. The coil rows, fin spacing, air-side performance, and cost are identical for all selections. Morabito (1960) showed how such changes in coil circuiting affect the overall system. Considering the investment cost of piping and insulation versus the operating cost of refrigeration and pumping motors, higher temperature rises (e.g., 16 to 24°F temperature rise at about 1.0 to 1.5 gpm per ton of cooling) can be applied on chilled-water systems with long distribution piping runs; larger flow rates should be used only where reasonable in close-coupled systems.

For the most economical design, the minimum flow rate to each terminal heat exchanger is calculated. For example, if one terminal can be designed for an 18°F rise, another for 14°F, and others for 12°F, the highest rise to each terminal should be used, rather than designing the system for an overall temperature rise based on the smallest capability.

Table 13-1 Chilled-Water Coil Performance

(Table 1, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Coil Circuiting	Chilled-Water Inlet Temp., °F	Coil Pressure Drop, psi	Chilled-Water Flow gpm/ton	Chilled-Water Temp. Rise, °F
Full[a]	45	1.0	2.2	10.9
Half[b]	45	5.5	1.7	14.9
Full[a]	40	0.5	1.4	17.1
Half[b]	40	2.5	1.1	21.8

Note: Table is based on cooling air from 81°F dry bulb, 67°F wet bulb to 58°F dry bulb, 56°F wet bulb.

[a] Full circuiting (also called single circuit). Water at the inlet temperature flows simultaneously through all tubes in a plane transverse to airflow; it then flows simultaneously through all tubes, in unison, in successive planes (i.e., rows) of the coil.

[b] Half circuiting. Tube connections are arranged so there are half as many circuits as there are tubes in each plane (row) thereby using higher water velocities through the tubes. This circuiting is used with small water quantities.

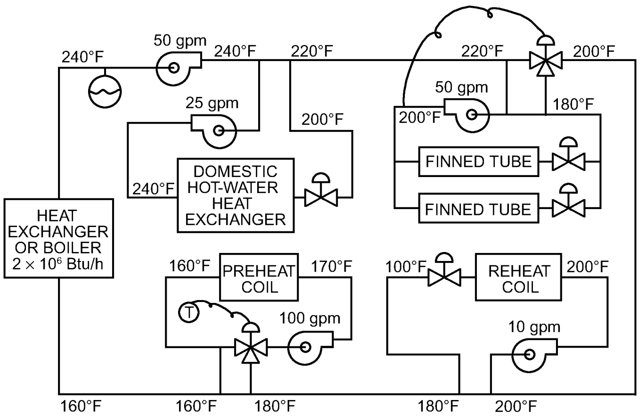


Fig. 13-20 Example of Series-Connected Loading

(Figure 31, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

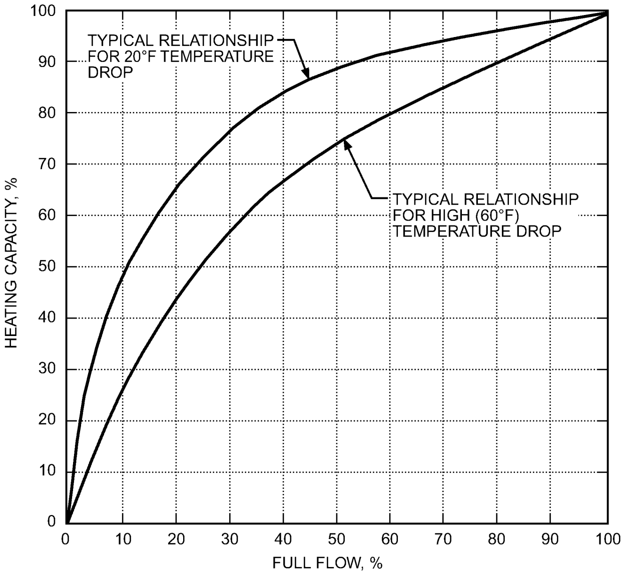


Fig. 13-21 Heat Emission Versus Flow Characteristic of Typical Hot Water Heating Coil

(Figure 32, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

The control system selected also influences the design water flow. For systems using multiple terminal units, diversity factors can be applied to flow quantities before sizing pump and piping mains if exposure or use prevents the unit design loads from occurring simultaneously and if two-way valves are used for water flow control. If air-side control (e.g., face-and-bypass or fan cycling) or three-way valves on the water side are used, diversity should not be a consideration in pump and piping design, although it should be considered in the chiller selection.

A primary consideration with chilled-water system design is the control of the source systems at reduced loads. The constraints on the temperature parameters are (1) a water freezing temperature of 32°F, (2) economics of the refrigeration system in generating chilled water, and (3) the dew-point temperature of the air at nominal indoor comfort conditions (55°F dew point at 75°F and 50% rh). These parameters have led to the common practice of designing for a supply chilled-water temperature of 44 to 45°F and a return water temperature between 55 and 64°F.

Historically, most chilled-water systems have used three-way control valves to achieve constant water flow through the chillers. However, as systems have become larger, as designers have turned to multiple chillers for reliability and controllability, and as energy efficiency has become an increasing concern, the use of two-way valves and source pumps for the chillers has greatly increased.

A typical configuration of a small chilled-water system using two parallel chillers and loads with three-way valves is illustrated in Figure 13-22. Note that the flow is essentially constant. A simple energy balance [Equation (13-9)] dictates that with a constant flow rate, at one-half of design load, the water temperature differential drops to one-half of design. At this load, if one of the chillers is turned off, the return water circulating through the off chiller mixes with the supply water. This mixing raises the temperature of the supply chilled water and can cause a loss of control if the designer does not consider this operating mode.

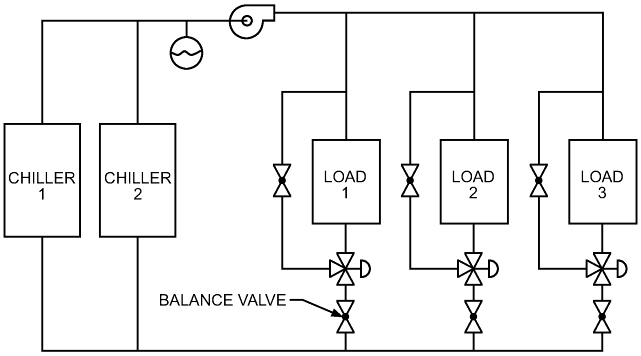


Fig. 13-22 Constant-Flow Chilled-Water System
(Figure 35, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

A typical configuration of a large chilled-water system with multiple chillers and loads and compound piping is shown in Figure 13-23. This system provides variable flow, essentially constant supply temperature chilled water, multiple chillers, more stable two-way control valves, and the advantage of adding chilled-water storage with little additional complexity.

One design issue illustrated in Figure 13-23 is the placement of the common pipe for the chillers. With the common pipe as shown on the left side of the chillers, the chillers will unload from left to right. With the common pipe in the alternate location shown, the chillers will unload equally in proportion to their capacity (i.e., equal percentage).

13.3.6 Dual-Temperature Systems

Dual-temperature systems are used when the same load devices and distribution systems are used for both heating and cooling (e.g., fan-coil units and central station air-handling unit coils). In the design of dual-temperature systems, the cooling cycle design usually dictates the requirements of the load heat exchangers and distribution systems. Basically, dual-temperature systems are of three different configurations, each requiring different design techniques:

1. Two-pipe systems
2. Four-pipe common load systems
3. Four-pipe independent load systems

Two-Pipe Systems. In a two-pipe system, the load devices and the distribution system circulate chilled water when cooling is required and hot water when heating is required (Figure 13-24). Design considerations for these systems include the following:

- Loads must all require cooling or heating coincidentally; that is, if cooling is required for some loads and heating for other loads at a given time, this type of system should not be used.
- When designing the system, the flow and temperature requirements for both the cooling and the heating media

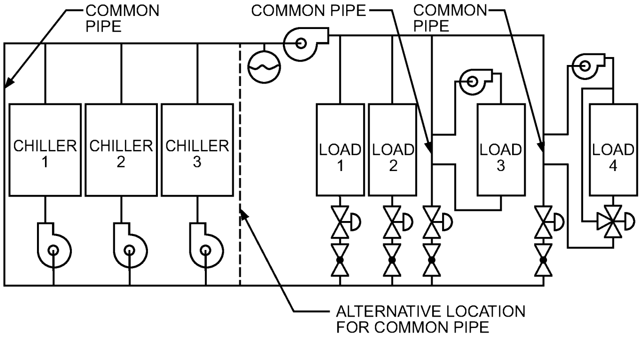


Fig. 13-23 Variable-Flow Chilled-Water System
(Figure 36, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

must be calculated first. The load and distribution system should be designed for the more stringent, and the water temperatures and temperature differential should be dictated by the other mode.

- The changeover procedure should be designed such that the chiller evaporator is not exposed to damaging high water temperatures and the boiler is not subjected to damaging low water temperatures. To accommodate these limiting requirements, the changeover of a system from one mode to the other requires considerable time. If rapid load swings are anticipated, a two-pipe system should not be selected, even though it is the least costly of the three options.

Four-Pipe Common Load Systems. In the four-pipe common load system, load devices are used for both heating and cooling, as in the two-pipe system. The four-pipe common load system differs from the two-pipe system in that both heating and cooling are available to each load device, and the changeover from one mode to the other takes place at each individual load device, or grouping of load devices, rather than at the source. Thus, some of the load systems can be in the cooling mode while others are in the heating mode. Figure 13-25 is a flow diagram of a four-pipe common load system, with multiple loads and a single boiler and chiller.

Although many of these systems have been installed, many have not performed successfully due to problems in implementing the design concepts.

One problem that must be addressed is the expansion tank connection(s). Many four-pipe systems were designed with two expansion tanks—one for the cooling circuit and one for the heating circuit. However, with multiple loads, these circuits become hydraulically interconnected, thus creating a system with two expansion chambers. The preferred method of handling the expansion tank connection sets the point of reference pressure equal in both circuits (Figure 13-25).

Another potential problem is the mixing of hot and chilled water. At each load connection, two three-way valves are required: a mixing valve on the inlet and a diverting valve on the outlet. These valves operate in unison in just two positions—opening either Port B to AB or Port A to AB. If, for example, the valve on the outlet does not seat tightly and Load 1 is indexed to cooling and Load 2 is indexed to heating, return heating water from Load 2 will flow into the chilled-water circuit, and return chilled water from Load 1 will flow into the heating-water circuit. The probability of this occurring increases as the number of loads increases because the number of control valves increases.

Another disadvantage of this system is that the loads have no individual capacity control as far as the water system is concerned. That is, each valve must be positioned to either full heating or full cooling with no control in between.

Because of these disadvantages, four-pipe common load systems should be limited to those applications in which there cannot be independent load circuits, such as radiant ceiling panels, chilled beams, or induction unit coils.

Four-Pipe Independent Load Systems. The four-pipe independent load system is preferred for those hydronic applications in which some of the loads are in the heating mode while others are in the cooling mode. Control is simpler and more reliable than for the common load systems, and in many applications, the four-pipe independent load system is more reliable and less costly to install. Also, the flow through the individual loads can be modulated, providing both the control capability for variable capacity and the opportunity for variable flow in either or both circuits.

A simplified example of a four-pipe independent load system with two loads, one boiler, and two chillers is shown in Figure 13-26. Note that both hydronic circuits are essentially independent, so that each can be designed with disregard for the other system. Although both circuits in the figure are shown as variable-flow distribution systems, they could be constant flow (three-way valves) or one variable flow and one constant flow. Generally, the control modulates the two load valves in sequence with a dead band at the control midpoint.

This type of system offers additional flexibility when some selective loads are arranged for heating only or cooling only, such as unit heaters or preheat coils. Then, central station systems can be designed for humidity control with reheat through

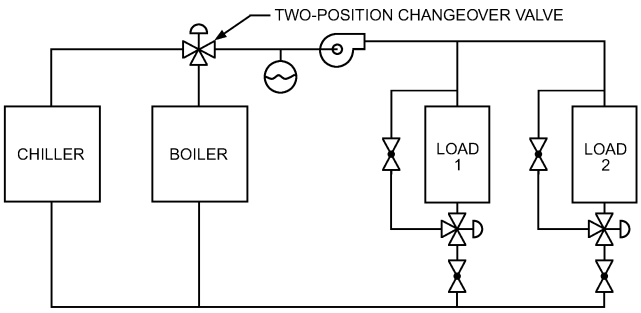


Fig. 13-24 Simplified Diagram of Two-Pipe System
(Figure 37, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

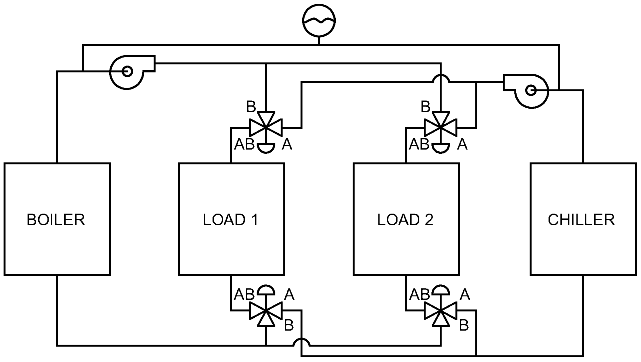


Fig. 13-25 Four-Pipe Common Load System
(Figure 38, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

configuration at the coil locations and with proper control sequences.

13.3.7 Other Design Considerations

Makeup and Fill Water Systems. Generally, a hydronic system is filled with water through a valved connection to a domestic water source, with a service valve, a backflow preventer, and a pressure gage. (The domestic water source pressure must exceed the system fill pressure.)

Because the expansion chamber is the reference pressure point in the system, the water makeup point is usually located at or near the expansion chamber.

Many designers prefer to install automatic makeup valves, which consist of a pressure-regulating valve in the makeup line. However, the quantity of water being made up must be monitored to identify leakage, which causes scaling, oxygen corrosion, gaseous air, and related problems in the system.

Safety Relief Valves. Safety relief valves should be installed at any point at which pressures can be expected to exceed the safe limits of the system components. Causes of excessive pressures include:

- Overpressurization from fill system
- Pressure increases due to thermal expansion
- Surges caused by momentum changes (shock or water hammer)

Overpressurization from the fill system could occur due to an accident in filling the system or due to the failure of an automatic fill regulator. To prevent this, a safety relief valve is usually installed at the fill location. Figure 13-27 shows a typical piping configuration for a system with a plain steel or air/water interface expansion tank. Note that no valves are installed between the hydronic system piping and the safety relief valve. This is a mandatory design requirement if the valve in this location is also to serve as a protection against pressure increases due to thermal expansion.

As previously stated, the expansion chamber is installed in a hydronic system to allow for the volumetric changes that accompany water temperature changes. However, if any part of the system is configured such that it can be isolated from the expansion tank and its temperature can increase while it is isolated, then overpressure relief should be provided.

The relationship between pressure change due to temperature change and the temperature change in a piping system is expressed by the following equation:

$$\Delta p = \frac{(\beta - 3\alpha)\Delta t}{(5/4)(D/E\Delta r) + \gamma} \qquad (13\text{-}17)$$

where

Δp = pressure increase, psi
β = volumetric coefficient of thermal expansion of water, 1/°F
α = linear coefficient of thermal expansion for piping material, 1/°F
Δt = water temperature increase, °F
D = pipe diameter, in.
E = modulus of elasticity of piping material, psi
γ = volumetric compressibility of water, in.^{2}/lb
Δr = thickness of pipe wall, in.

Figure 13-28 shows a solution to Equation (13-17) demonstrating the pressure increase caused by any given temperature increase for 1 in. and 10 in. steel piping. If the temperature in a chilled water system with piping spanning sizes between 1 and 10 in. were to increase by 15°F, the pressure would increase between 340 and 420 psi, depending on the average pipe size in the system.

Safety relief should be provided to protect boilers, heat exchangers, cooling coils, chillers, and the entire system when the expansion tank is isolated for air charging or other service. As a minimum, the ASME Boiler Code requires that a dedicated safety relief valve be installed on each boiler and that isolating or service valves be provided on the supply and return connections to each boiler.

Potential forces caused by shock waves or water hammer should also be considered in design. Chapter 22 of the 2017

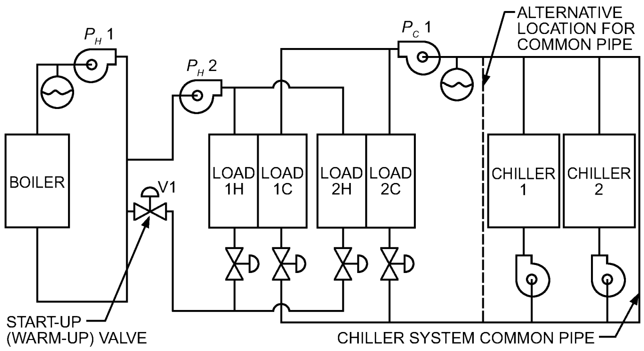


Fig. 13-26 Four-Pipe Independent Load System

(Figure 39, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

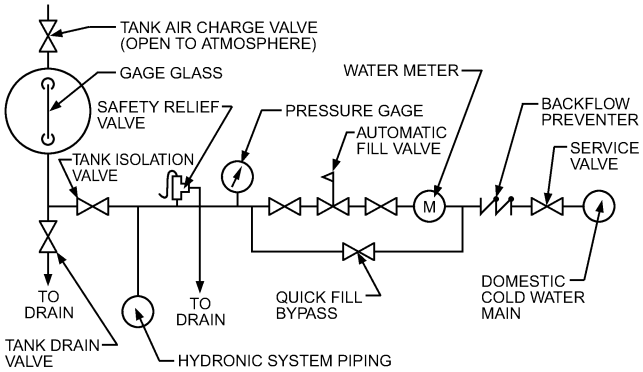


Fig. 13-27 Typical Makeup Water and Expansion Tank Piping Configuration for Plain Steel Expansion Tank

(Figure 40, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

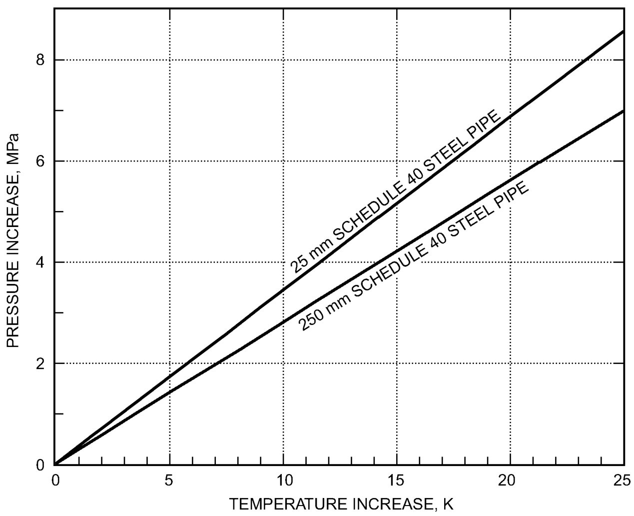


Fig. 13-28 Pressure Increase Resulting from Thermal Expansion as Function of Temperature Increase
(Figure 41, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

ASHRAE Handbook—Fundamentals discusses the causes of shock forces and the methodology for calculating the magnitude of these forces.

Air Elimination. If air and other gases are not eliminated from the flow circuit, they may cause binding in the terminal heat transfer elements, corrosion, noise, reduced pumping capacity or flow in a circuit, and loss of hydraulic stability. A closed tank without a diaphragm can be installed at the point of the lowest solubility of air in water (see Figure 13-3). When a diaphragm tank is used, air in the system can be removed by an air separator and air elimination valve installed at the point of lowest solubility. Manual vents should be installed at high points to remove all trapped air during initial operation and to ensure that the system is tight. Shutoff valves should be installed on any automatic air removal device to permit servicing without draining the system.

Drain and Shutoff. All low points should have drains. Separate shutoff and draining of individual equipment and circuits should be possible so that the entire system does not have to be drained to service a particular item. Whenever a device or section of the system is isolated, and the water in that section or device could increase in temperature following isolation, overpressure safety relief protection as discussed above must be provided.

Balance Fittings. Balance fittings or valves and a means of measuring flow quantity should be applied as needed to permit balancing of individual terminals and subcircuits.

Pitch. Piping need not pitch but can run level, providing that flow velocities exceeding 1.5 fps are maintained or a diaphragm tank is used.

Strainers. Strainers should be used where necessary to protect system elements. Strainers in the pump suction must be checked regularly to prevent cavitation. Large separating chambers can serve as main air venting points. Automatic control valves or other devices operating with small clearances require protection from pipe scale, gravel, and welding slag, which may readily pass through the pump and its protective separator. Individual fine mesh strainers may therefore be required ahead of each such device.

Thermometers. Thermometers or thermometer wells for temperature sensing and control calibration should be installed to assist the system operator in routine operation and troubleshooting. Permanent temperature sensors or thermometers, with the correct scale range and separate sockets, should be used at all points where temperature readings are regularly needed. Thermometer wells should be installed where readings will be needed only during start-up and infrequent trouble-shooting. If a central monitoring system is provided, a calibration well should be installed adjacent to each sensing point in insulated piping systems.

Flexible Connectors and Expansion Compensation. Flexible connectors are sometimes installed at pumps and machinery to reduce pipe stress. See Chapter 48 of the 2015 *ASHRAE Handbook—HVAC Applications* for vibration isolation information. Expansion, flexibility, and hanger and support information is in Chapter 46 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment Handbook.*

Gage Cocks. Gage cocks or quick-disconnect test ports should be installed at points requiring pressure readings. Gages permanently installed in the system will deteriorate because of vibration and pulsation and will, therefore, be unreliable. It is good practice to install gage cocks and provide the operator with several quality gages for diagnostic purposes.

Insulation. Insulation should be applied to minimize pipe thermal loss and to prevent condensation during chilled water operation (see Chapter 23 of the 2017 *ASHRAE Handbook—Fundamentals*). On chilled-water systems, special rigid metal sleeves or shields should be installed at all hanger and support points, and all valves should be provided with extended bonnets to allow for the full insulation thickness without interference with the valve operation.

Condensate Drains. Condensate drains from dehumidifying coils should be trapped and piped to an open-sight plumbing drain. Traps should be deep enough to overcome the air pressure differential between drain inlet and room, which ordinarily will not exceed 2 in. of water. Pipe should be noncorrosive and insulated to prevent moisture condensation. Depending on the quantity and temperature of condensate, plumbing drain lines may require insulation to prevent sweating.

Common Pipe. In compound (primary-secondary) pumping systems, the common pipe is used to dynamically decouple the two pumping circuits. Ideally, there is no pressure drop in this section of piping; however, in actual systems, it is recommended that this section of piping be a minimum of 10 diameters in length to reduce the likelihood of unwanted mixing resulting from velocity (kinetic) energy or turbulence.

13.4 Design Procedures

13.4.1 Preliminary Equipment Layout

Flows in Mains and Laterals. Regardless of the method used to determine the flow through each item of terminal equipment, the desired result should be listed in terms of mass flow on the preliminary plans or in a schedule of flow rates for the piping system. (In the design of small systems and chilled-water systems, the determination may be made in terms of volumetric flow).

In an equipment schedule or on the plans, starting from the most remote terminal and working toward the pump, progressively list the cumulative flow in each of the mains and branch circuits in the distribution system.

Preliminary Pipe Sizing. For each portion of the piping circuit, select a tentative pipe size from the unified flow chart in Chapter 22 of the 2017 *ASHRAE Handbook—Fundamentals*, using a value of pipe friction loss ranging from 0.75 to 4 ft per 100 ft (approximately 0.1 to 0.5 in./ft).

Residential piping size is often based on pump preselection using pipe sizing tables, which are available from the Hydronics Institute or from manufacturers.

Preliminary Pressure Drop. Using the preliminary pipe sizing indicated above, determine the pressure drop through each portion of the piping. The total pressure drop in the longest or highest head loss circuit determines the maximum pressure drop through the piping, including the terminals and control valves, that must be made available by the pump.

Preliminary Pump Selection. The preliminary selection should be based on the pump's ability to fulfill the determined capacity requirements. It should be selected at a point left of center on the pump curve and should not overload the motor. Because pressure drop in a flow system varies as the square of the flow rate, the flow variation between the nearest size of stock pump and an exact point selection will be relatively minor.

13.4.2 Final Pipe Sizing and Pressure Drop Determination

Final Piping Layout. Examine the overall piping layout to determine whether pipe sizes in some areas need to be readjusted. Several principal circuits should have approximately equal pressure drops so that excessive pressures are not needed to serve a small portion of the building.

Consider both the initial costs of the pump and piping system and the pump's power and energy requirement when determining final system friction loss. Lower heads and larger piping are more energy-efficient and are generally more economical when longer amortization periods are considered, especially in larger systems. However, in small systems such as in residences, it may be most economical to select the pump first and design the piping system to meet the available pressure. In all cases, adjust the piping system design and pump selection until the optimum design is found.

Final Pressure Drop. When the final piping layout has been established, determine the friction loss for each section of the piping system from the pressure drop charts (Chapter 22 of the 2017 *ASHRAE Handbook—Fundamentals*) for the mass flow rate in each portion of the piping system.

After calculating the friction loss at design flow for all sections of the piping system and all fittings, terminal units, and control valves, sum them for several of the longest piping circuits to determine the pressure against which the pump must operate at design flow.

Final Pump Selection. After completing the final pressure drop calculations, select the pump by plotting a system curve and pump curve and selecting the pump or pump assembly that operates closest to the calculated design point.

13.4.3 Freeze Prevention

All circulating water systems require precautions to prevent freezing, particularly in makeup air applications in cold climates (1) where coils are exposed to outdoor air at below-freezing temperatures, (2) where undrained chilled water coils are in the winter airstream, or (3) where piping passes through unheated spaces. Freezing will not occur as long as flow is maintained and the water is at least warm. Unfortunately, during extremely cold weather or in the event of a power failure, water flow and temperature cannot be guaranteed. Additionally, continuous pumping can be energy-intensive and cause system wear. The following are precautions to avoid flow stoppage or damage from freezing:

1. Select all load devices (such as preheat coils) that are subjected to outdoor air temperatures for constant flow, variable Δt control.
2. Position the coil valves of all cooling coils with valve control that are dormant in winter months to the full-open position at those times.
3. If intermittent pump operation is used as an economy measure, use an automatic override to operate both chilled water and heating water pumps in below-freezing weather.
4. Select pump starters that automatically restart after power failure (i.e., maintain contact control).
5. Select non-overloading pumps.
6. Instruct operating personnel never to shut down pumps in subfreezing weather.
7. Do not use aquastats, which can stop a pump, in boiler circuits.
8. Avoid sluggish circulation, which may cause air binding or dirt deposit. Properly balance and clean systems. Provide proper air control or means to eliminate air.
9. Install low-temperature-detection thermostats that have phase-change capillaries wound in a serpentine pattern across the leaving face of the upstream coil.

In fan equipment handling outdoor air, take precautions to avoid stratification of air entering the coil. The best methods for proper mixing of indoor and outdoor air are the following:

1. Select dampers for pressure drops adequate to provide stable control of mixing, preferably with dampers installed

several equivalent diameters upstream of the air-handling unit.

2. Design intake and approach duct systems to promote natural mixing.
3. Select heating coils with circuiting to allow parallel flow of air and water.

Freeze-up may still occur with any of these precautions. If an antifreeze solution is not used, water should circulate at all times. Valve-controlled elements should have low-limit thermostats, and sensing elements should be located to ensure accurate air temperature readings. Primary-secondary pumping of coils with three-way valve injection (as in Figure 13-19) is advantageous.

13.4.4 Antifreeze Solutions

In systems in danger of freeze-up, water solutions of ethylene glycol and propylene glycol are commonly used. Freeze protection may be needed (1) in snow-melting applications (see Chapter 51 of the 2015 *ASHRAE Handbook—HVAC Applications*); (2) in systems subjected to 100% outdoor air, where the methods outlined above may not provide absolute antifreeze protection; (3) in isolated parts or zones of a heating system where intermittent operation or long runs of exposed piping increase the danger of freezing; and (4) in process cooling applications requiring temperatures below 40°F. Although using ethylene glycol or propylene glycol is comparatively expensive and tends to create corrosion problems unless suitable inhibitors are used, it may be the only practical solution in many cases.

Solutions of triethylene glycol, as well as certain other heat transfer fluids, may also be used. However, ethylene glycol and propylene glycol are the most common substances used in hydronic systems because they are less costly and provide the most effective heat transfer.

Heat Transfer and Flow. Chapter 31 of the 2017 *ASHRAE Handbook—Fundamentals* presents density, specific heat, thermal conductivity, and viscosity of various aqueous solutions of ethylene glycol and propylene glycol.

System heat transfer rate is affected by relative density and specific heat according to the following equation:

$$q_w = 500\, Q(\rho / \rho_w) c_p \Delta t \qquad (13\text{-}18)$$

where

q_w = total heat transfer rate, Btu/h
Q = flow rate, gpm
ρ = fluid density, lb/ft^3
ρ_w = density of water at 60°F, lb/ft^3
c_p = specific heat of fluid, Btu/lb·°F
Δt = temperature increase or decrease, °F

Effect on Heat Source or Chiller. Generally, ethylene glycol solutions should not be used directly in a boiler because of the danger of chemical corrosion caused by glycol breakdown on direct heating surfaces. However, properly inhibited glycol solutions can be used in low-temperature water systems directly in the heating boiler if proper operation can be ensured. Automobile antifreeze solutions are not recommended because the silicate inhibitor can cause fouling, pump seal wear, fluid gelation, and reduced heat transfer. The area or zone requiring the antifreeze protection can be isolated with a separate heat exchanger or converter. Glycol solutions are used directly in water chillers in many cases.

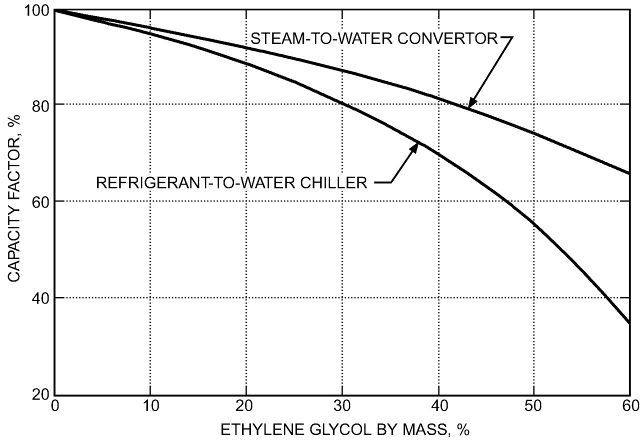


Fig. 13-29 Example of Effect of Aqueous Ethylene Glycol Solutions on Heat Exchanger Output

(Figure 42, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Glycol solutions affect the output of a heat exchanger by changing the film coefficient of the surface contacting the solution. This change in film coefficient is caused primarily by viscosity changes. Figure 13-29 illustrates typical changes in output for two types of heat exchangers, Curve A for a steam-to-liquid converter and Curve B for a refrigerant-to-liquid chiller. The curves are plotted for one set of operating conditions only and reflect the change in ethylene glycol concentration as the only variable. Propylene glycol has a similar effect on heat exchanger output.

Because many other variables, such as liquid velocity, steam or refrigerant loading, temperature difference, and unit construction, affect the overall coefficient of a heat exchanger, designers should consult manufacturers' ratings when selecting such equipment. The curves indicate only the relative magnitude of these output changes.

Effect on Terminal Units. Because the effect of glycol on the capacity of terminal units may vary widely with temperature, the manufacturer's rating data should be consulted when selecting heating or cooling units in glycol systems.

Effect on Pump Performance. Centrifugal pump characteristics are affected to some degree by glycol solutions because of viscosity changes. Figure 13-30 shows these effects on pump capacity, head, and efficiency. Figures in Chapter 31 of the 2017 *ASHRAE Handbook—Fundamentals* plot the viscosity of aqueous ethylene glycol and propylene glycol. Centrifugal pump performance is normally cataloged for water at 60 to 80°F. Hence, absolute viscosity effects

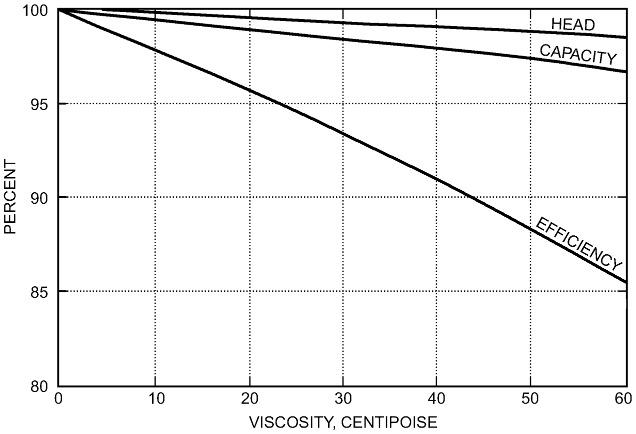


Fig. 13-30 Effect of Viscosity on Pump Characteristics

(Figure 43, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

below 1.1 centipoise can safely be ignored as far as pump performance is concerned. In intermittently operated systems, such as snow-melting applications, viscosity effects at start-up may decrease flow enough to slow pickup.

Effect on Piping Pressure Loss. The friction loss in piping also varies with viscosity changes. Figure 13-31 gives correction factors for various ethylene glycol and propylene glycol solutions. These factors are applied to the calculated pressure loss for water. No correction is needed for ethylene glycol and propylene glycol solutions above 160°F.

Installation and Maintenance. Because glycol solutions are comparatively expensive, the smallest possible concentrations to produce the desired antifreeze properties should be used. The total water content of the system should be calculated carefully to determine the required amount of glycol (Craig et al. 1993). The solution can be mixed outside the system in drums or barrels and then pumped in. Air vents should be watched during filling to prevent loss of solution. The system and the cold water supply should not be permanently connected, so automatic fill valves are usually not used.

Ethylene glycol and propylene glycol must include an inhibitor to help prevent corrosion. Solutions should be checked regularly using a suitable refractometer to determine glycol concentration. The following precautions regarding the use of inhibited glycol solutions should be taken to extend their service life and to preserve equipment:

1. Before injecting the glycol solution, thoroughly clean and flush the system.
2. Use waters that are soft and low in chloride and sulfate ions to prepare the solution whenever possible.
3. Limit the maximum operating temperature to 250°F in a closed hydronic system. In a heat exchanger, limit glycol film temperatures to 300 to 350°F (steam pressures 120 psi or less) to prevent deterioration of the solution.
4. Check the concentration of inhibitor regularly, following procedures recommended by the glycol manufacturer.

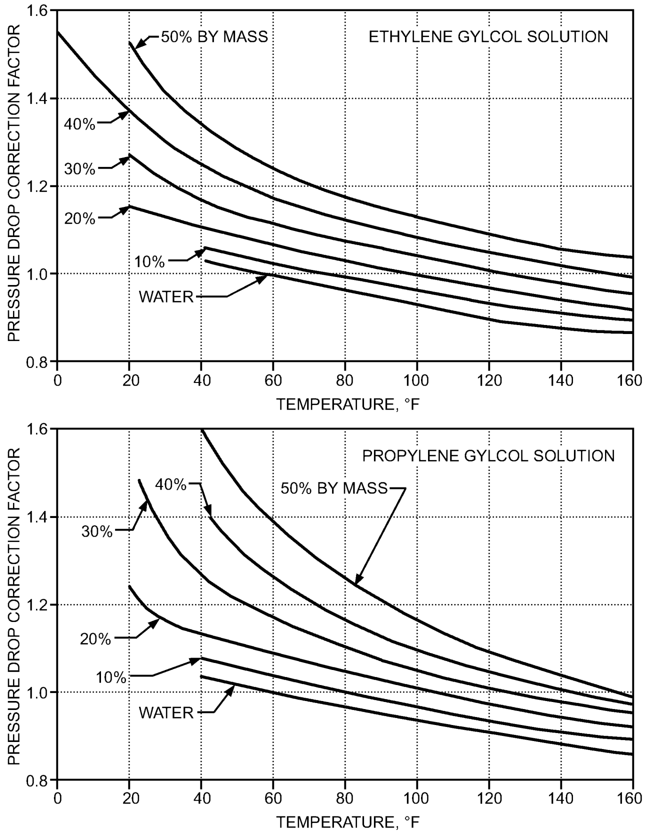


Fig. 13-31 Pressure Drop Correction for Glycol Solutions

(Figure 44, Chapter 13, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

13.5 Problems

13.1 What is the maximum temperature at which a heating water system can be operated if the boiler (hot water generator) is rated as low pressure by the ASME Boiler and Pressure Vessel Code?

13.2 Sketch the fundamental components for a chilled water system with a single load and source and a capacity of 100 tons of cooling.

(a) What is the water circulation rate (gpm) required if the temperature range of the water is 12°F.

(b) If the head loss in the system is 60 feet, and the pump is 80% efficient, what is the pump horsepower? Motor size?

(c) If the motor is 90% efficient and it operates for one-third of the total hours in the year, what is the annual energy consumption of the pump?

13.3 Calculate the size of the expansion tank for a hot water heating system of 1,200,000 Btu/h heating capacity if the tank is a closed tank with an air/water interface and the following system parameters are known:

Supply water temperature	210°F

Ambient temperature	60°F
Fill pressure (at tank)	30 psig
Maximum operating pressure (at tank)	35 psig
System water volume	6,000 gallons
Steel piping system material	

13.4 What size diaphragm tank would be required for the above system?

13.5 In a given chilled water system, the pump head required at 640 gpm is 80 ft.

(a) What is the system constant, C_s?

(b) Plot the system curve from 0 to 800 gpm.

13.6 In a chilled water system, the pump is located in a basement equipment room with the expansion tank connected to the pump suction. The pump is the lowest point in the system and the highest point is a pipe in the penthouse, which is 115 feet above the pump. The dynamic head losses in the system are:

Piping and fittings	30 ft
Chiller	20 ft
Control valve	10 ft
Cooling coil	10 ft

When the system is filled (at 95°F ambient temperature) it is desired to have a pressure of 10 psig at the highest point in the system, which will reduce to 5 psig when the water temperature reduces to 45°F.

(a) What operating pressures (p_1, p_2) should the expansion tank be designed for?

(b) What pump head is required?

(c) With the pump off and a cold (45°F) system, what is the pressure at the pump suction? The pump discharge?

(d) With the pump on and a cold (45°F) system, what is the pressure at the pump suction? The pump discharge?

13.7 In your own words, explain the difference between a three-way control valve and a two-way control valve as they affect the hydraulics of the system.

13.8 A control valve is to be sized for a cooling coil with a capacity of 30 tons of cooling. The water temperature entering the coil is at 44°F with a 12°F Δt. It is determined that the valve should have a pressure drop of 5 psi. What is the required C_v of the valve?

13.9 A section of 1in. steel pipe in a chilled water system at 50 psig is in a pipe chase and is located between two service valves. With a cold system (45°F) the section is isolated by closing off the two service valves. If the chase is at a temperature of 95°F and the pipe reaches thermal equilibrium with the chase, what will the final pressure in the pipe be?

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Chapter 14

UNITARY AND ROOM AIR CONDITIONERS

This chapter discusses the availability of the various types of unitary units and room air conditioners. Additional details on this equipment can be found in Chapters 49 and 50 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment.*

14.1 Unitary Air Conditioners

Unitary air-conditioning equipment is an assembly of factory-matched refrigerant cycle devices for inclusion as components in field-designed air-conditioning systems. Some of the many types of unitary air conditioners available include the following characteristics:

Arrangement: Single package or split system (i.e., an indoor evaporator and blower and a separate, usually outdoor, compressor and condenser unit).

Heat rejection: Air cooled, water cooled, evaporative condenser.

Unit exterior: Decorative for in-space applications, functional for equipment room and ducts, weatherproofed for outdoors.

Placement: Floor standing, wall mounted, ceiling suspended, roof mounted.

Indoor air: Vertical upflow or downflow, horizontal flow, 90° and 180° turns, with fan, or for use with forced-air furnaces.

Locations: *Indoor*—exposed with plenums or furred in ductwork concealed in closets, attic, crawlspaces, basements, garages, utility rooms, or equipment rooms. Wall, window, or transom mounted.

Outdoor—rooftop, wall mounted, or on ground

Heat: May be combined with electric heat, gas heat, hot water, or steam coil.

Unitary air conditioners, in contrast to room air conditioners, include fans capable of operating with ductwork, although some units may be applied with supply air plenums. Heat pumps are also offered in many of the same types and capacities as unitary air conditioners. Packaged reciprocating and centrifugal water chillers are considered to be unitary air conditioners, particularly when applied with unitary chilled-water blower coil units.

Single-package air conditioners are depicted in Figures 14-1 through 14-3. Split systems and condensing units with coils and with blower coil units are shown in Figures 14-4 through 14-6.

The many combinations of coil configurations, evaporator temperatures, air-handling arrangements, refrigerating capacities, and variations thereof that are available in central systems are seldom possible with unitary systems. Consequently, a higher level of design ingenuity and performance is required to develop acceptable system performance from unitary equipment.

Unitary equipment tends to serve zoned systems, with each zone served by its own unit. The room conditioner or packaged terminal air conditioner (PTAC) carries this concept to relatively small rooms.

For large single spaces where central systems are at their best advantage, multiple central systems are often advantageous because as load sources move within the larger space,

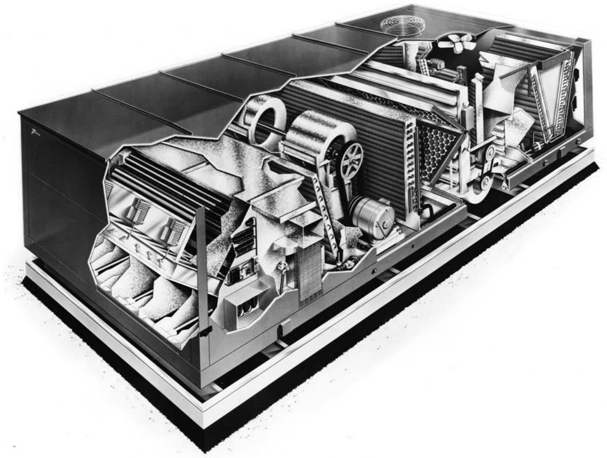


Fig. 14-1 Typical Rooftop Air-Cooled Single-Package Air Conditioner
(Figure 1, Chapter 49, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

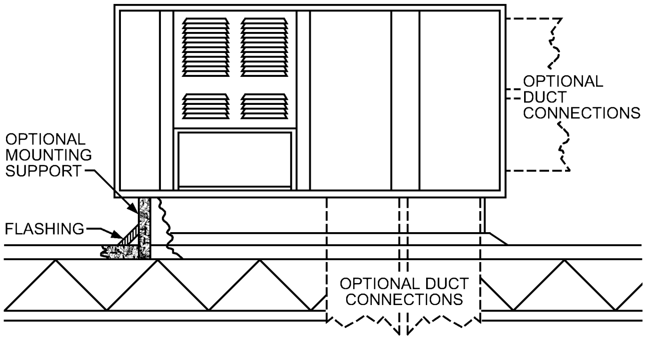


Fig. 14-2 Rooftop Installation of a Single-Package Unit
(Figure 5, Chapter 49, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

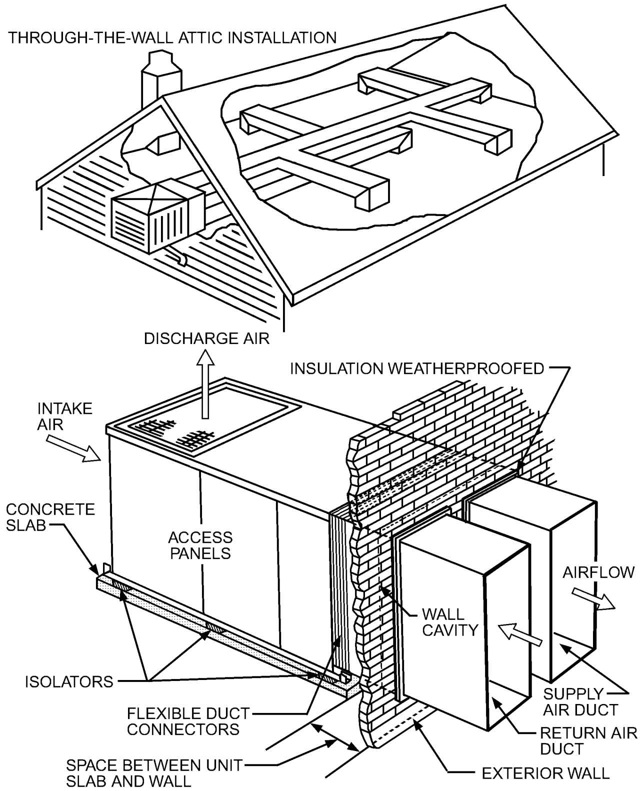


Fig. 14-3 Typical Through-the-Wall Air-Cooled Single-Package Unit

(Figure 7, Chapter 49, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

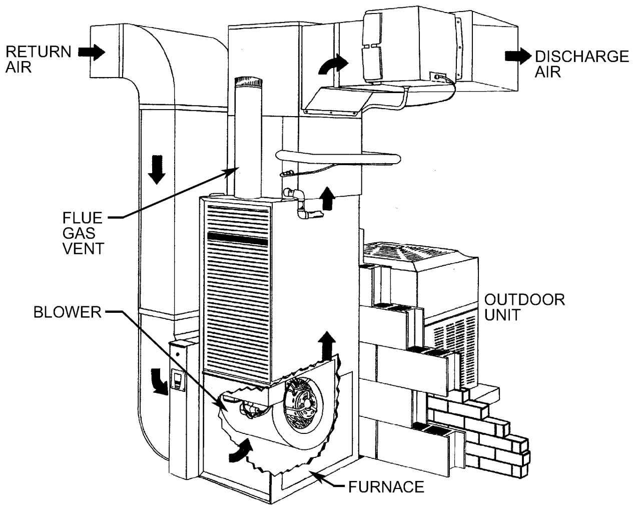


Fig. 14-4 Residential Installation of Split-System Air-Cooled Condensing Unit with Indoor Coil and Upflow Furnace

(Figure 8, Chapter 49, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

the many smaller interlocked and independent systems have more flexibility than one central system.

However, rooms with less than 0.5 ton (2 kW) or more than 25 ton (100 kW) cooling loads are seldom conditioned by their own single unit in multiple-unit systems. Multiple-unit systems may provide the following advantages over central system alternatives:

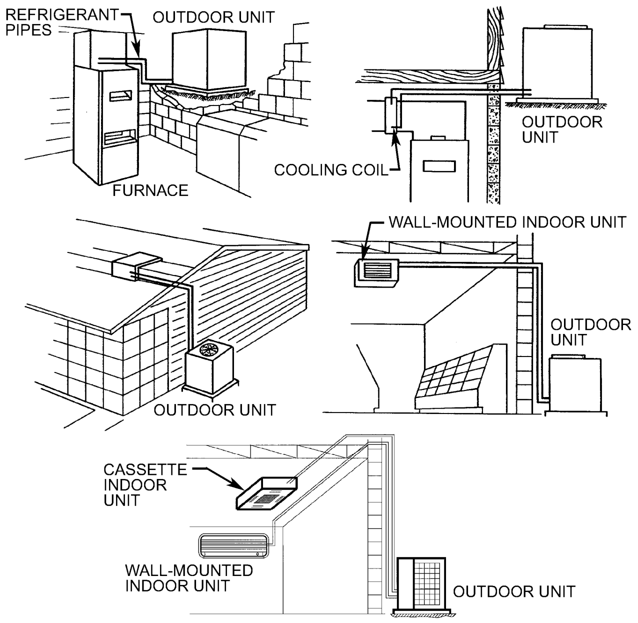


Fig. 14-5 Outdoor Installations of Split-System Air-Cooled Condensing Units with Coil and Upflow Furnace or with Indoor Blower Coils

(Figure 9, Chapter 49, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

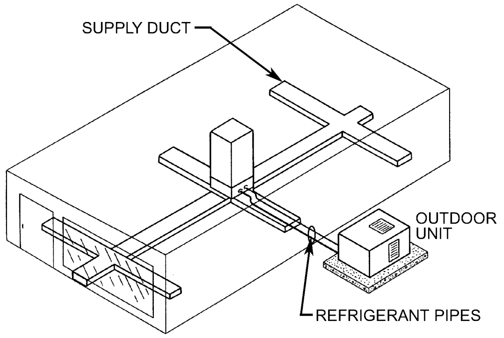


Fig. 14-6 Outdoor Installation of Split-System Air-Cooled Condensing Unit with Indoor Coil and Downflow Furnace

(Figure 10, Chapter 49, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

- Simple and inexpensive individual room control
- Individual air distribution for each room, usually with convenient and simple adjustment
- Heating and cooling capability at all times, independent of the mode of operation of other spaces in the building
- Consistent performance assured by manufacturer-matched components
- Generally have published certified ratings and performance data

- Single source of accountability because manufacturer assembles components
- Manufacturer instructions and multiple-unit nature simplify and systematize installation through repetition of tasks
- Only one terminal zone or conditioner is affected in the event of equipment malfunction
- Often saves some space
- Usually quick availability and installation are possible
- Often lower initial cost
- Responsibility for performance of complete package(s) rests with one manufacturer and its agents who provide information on application, installation, maintenance, and service
- Equipment serving spaces that become vacant can be turned off locally or from a central point without affecting occupied spaces

Multiple-unit systems may have the following disadvantages:

- Limited performance options are available because airflow and cooling coil and condenser sizing are fixed.
- Not generally suited for effective humidity control, except when using special purpose equipment such as packaged units for a computer room. Poor humidity control can result in mold and mildew growth within the space.
- Energy use may be greater than for central systems if efficiency of the unitary equipment is less than that of the combined central system components.
- Winter cooling by outdoor air economizers is not always available.
- Air distribution control may be limited.
- Operating sound levels can be high.
- Ventilation capabilities are limited by equipment design.
- Engineered ventilation and humidity control must be provided by a supplementary system, usually a dedicated outdoor air system (DOAS).
- Overall appearance can be unappealing.
- Air filtration options are limited.
- Maintenance may be difficult because of the many pieces of equipment and their locations.

14.2 Combined Unitary and Dedicated Outdoor Air Systems

Combining some type of unitary system with a dedicated outdoor air system (DOAS) can provide very good comfort conditions at a reasonable cost and low energy consumption (see Chapter 12). The DOAS is designed to provide the ventilation, humidity control, and a high level of filtration of the outdoor air, and control of room temperature and air motion is assigned to the unitary room unit. In some cases, with a well-designed supply of DOAS air, that unit can also provide the room air motion, or at least supplement it so the fan of the unitary system can be cycled with the need for cooling or heating.

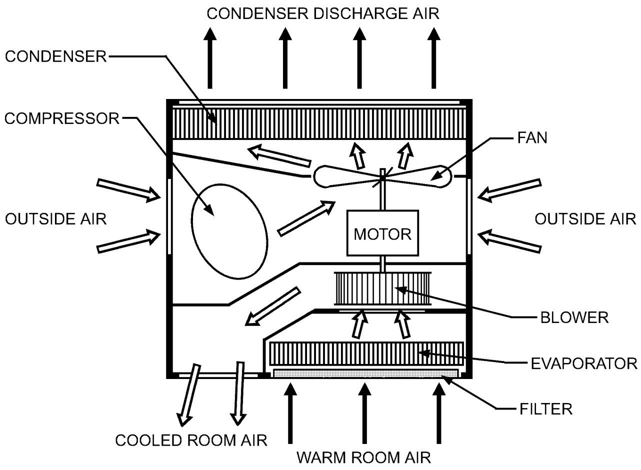


Fig. 14-7 Schematic View of Window Air Conditioner
(Figure 1, Chapter 50, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

14.3 Window Air Conditioners

A window air conditioner is an encased assembly designed as a unit primarily for mounting in a window. These units are designed for comfort cooling and provide delivery of conditioned air to the room either without ducts or with very short ducts up to a maximum of about 48 in. (1.2 m).

A window air conditioner cools, dehumidifies, filters or cleans, and circulates room air. Ventilation may also be provided by introducing outdoor air into the room and/or by exhausting room air to the outside. Some conditioners provide heating by reverse cycle (heat pump) operation or by electric resistance elements.

A typical window air conditioner is shown diagrammatically in Figure 14-7. Warm room air passes over the cooling coil, giving up its sensible and latent heat. The conditioned air is then circulated in the room by a fan or blower.

The cooling and heating capacities of window air conditioners are always measured and stated in terms of Btu/h (W). A wide range of capacities is available [from approximately 4000 to 36,000 Btu/h (1.2 to 10.5 kW)].

The design of a window air conditioner is usually based on one or more of the following criteria, any one of which automatically limits the freedom of the designer in overall system design:

- Lowest initial cost
- Lowest operating cost (highest efficiency)
- Low sound level
- Physical chassis size
- An unusual chassis shape (minimal depth, height, etc.)
- An amperage limitation (7.5 A, 12 A, etc.)
- Weight

The basic design is a carefully selected group of components consisting of an evaporator, a condenser, a compressor, one or more fan motors, blower wheels for evaporator and condenser airflow, and an expansion device, usually consisting of one or more capillary tubes.

14.4 Through-the-Wall Conditioner System

A through-the-wall system is an air-cooled room air conditioner designed for mounting through the wall and normally capable of providing both heating and cooling. Design and manufacturing specifications range from appliance grade to heavy-duty commercial grade. The latter is called *packaged terminal air conditioner* (PTAC) and is defined as such by AHRI in the "Packaged Terminal Air Conditioners" subsection of *Air Conditioning Heat Transfer Products*; all others are covered by AHAM *Standard* CN-l.

System Concept and Description. The through-the- wall concept incorporates a complete air-cooled refrigeration and air-handling system in an individual package, using space normally occupied by the building wall for equipment, with the remainder projecting inside the room.

Each packaged terminal air conditioner has a self-contained, direct-expansion cooling system, heating coil (electric, hot water, or steam), and packaged controls. Two general configurations used consist of (1) wall box, heat section, room cabinet, outdoor louver, and cooling chassis (Figure 14-8) and (2) a combination wall sleeve and room cabinet, combination heating and cooling chassis, and outdoor louver (Figure 14-9).

The exterior louver is installed flush (or nearly so) with the outside wall of the building and receives a variety of architectural treatments, such as emphasizing the existing louvers and using them as aesthetic highlights, hiding them completely with solid or pierced wall coverings, or designing the building so that the louvers blend into the overall exterior.

Advantages. The initial cost of the through-the-wall system in multiroom applications is considerably less than central systems adapted to simultaneously heat or cool each room under control of the room occupants. The cost differences may run as high as 20% to 50%, depending on the design and components.

Because a through-the-wall system has no system auxiliaries, the energy consumption may be lower than for central systems. For this reason, economical comparisons between central systems and through-the-wall conditioners should include all system components, including fans, pumps, and heat rejection devices. Keep in mind that fan and motor efficiencies will be significantly lower for the through-the-wall air conditioner.

A through-the-wall system requires less space because it requires no ductwork or equipment rooms. This space savings can range from 5% to 15% of the total building space when the savings are considered on a floor-by-floor basis.

Limitations. The through-the-wall system is limited to multizone systems and generally cannot be used economically in large spaces requiring more than three units per zone. However, where the packaged terminal air-conditioner system is coordinated with a well-designed core system, it can be economically used for large office areas while allowing for maximum flexibility in moving partitions. Through-the-wall units are more likely to be limited by their ability to throw air across the room.

Current products do not include individual conditioner humidifier systems. Humidification and controlled dehumidification can be achieved with a through-the-wall system but must be done through a separate dedicated outdoor air system. The through-the-wall system should not be used where there are high sensible load requirements coming from a concentration of heat-producing equipment, such as is used in computer server rooms or radio and television stations.

In commercial or public buildings through-the-wall or PTAC units should not be used for ventilation. Ventilation should be provided with a supplementary DOAS.

Applications. Through-the-wall systems are often applied in multiple zone applications. The system lends itself to both low- and high-rise buildings. The system is most generally applied in

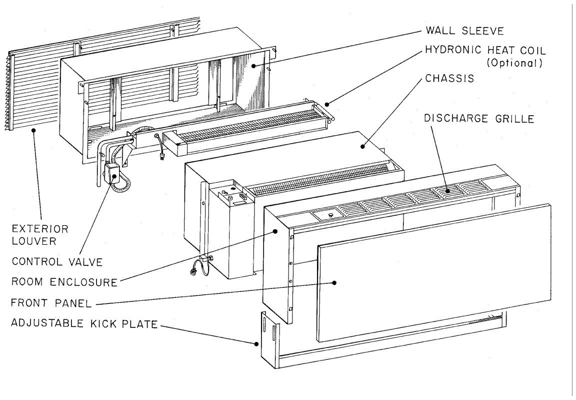


Fig. 14-8 Packaged Terminal Air Conditioner with Separate Heat Section and Cooling Chassis

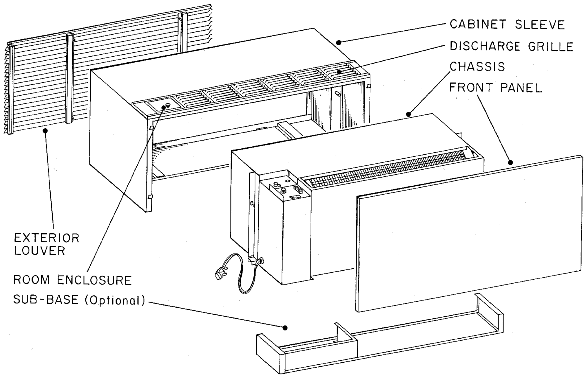


Fig. 14-9 Packaged Terminal Air Conditioner with Combination Heating and Cooling Chassis

- Office buildings
- Motels and hotels
- Apartments and dormitories
- Schools and other educational buildings
- Nursing homes

This system is also applicable for renovating existing buildings because all or part of the existing heating system can be used. This equipment could cause less disruption and construction-forced sacrifice of rentable space than alternative systems. The result is an automatically controlled heating system and a self-contained cooling system, except for the companion DOAS. The major disadvantage of most PTAC systems is the relatively high sound level in the occupied space.

14.5 Typical Performance

Specially constructed equipment cannot be justified for small commercial and residential applications. Furthermore, these applications generally have a higher sensible heat factor (SHF), so dehumidification is not as critical as in large commercial buildings. Therefore, the equipment is manufactured to operate at or near one set of conditions.

For example, typical residential and light commercial cooling equipment operates with a coil SHF of 0.75 to 0.8 with the air entering the coil at about 80°F (27°C) dry-bulb and 67°F (19°C) wet-bulb temperature. This equipment usually has a capacity of less than 10 tons (35 kW). When the peak cooling load and latent heat requirements are appropriate, this less expensive type of equipment can be considered.

Selected equipment should be within the range of 95 to 115% of the peak cooling load. The air quantity is specified by the manufacturer for each unit and is about 400 cfm/ton (50 L/s per kW). The total air quantity is then divided among the various rooms according to the cooling load of each room. Typical performance data for residential and light commercial cooling equipment are listed in Table 14-1.

Example 14.1 An air-conditioning unit selected for a residence has a rated total cooling capacity of 36,000 Btu/h and a sensible cooling capacity of 27,000 Btu/h. The manufacturer also lists a SEER of 13 for this unit. The unit is expected to operate for 1900 hours during each cooling season.

Table 14-1 Typical Residential or Light Commercial Cooling Coil Performance Data (Split System)

Capacity		Airflow		Coil Pressure Loss	
Btu/h	kW	cfm	m^3/s	in. of water	Pa
18,000	5.3	600	0.28	0.18	45
24,000	7.0	800	0.38	0.30	75
30,000	8.8	1050	0.50	0.13	32
41,000	10.3	1270	0.60	0.20	50
48,000	14.1	1750	0.83	0.25	62
59,000	17.3	2140	1.01	0.30	75

a) Determine the latent cooling capacity.
b) What is the sensible heat ratio for the unit?
c) What is the expected energy use per cooling season?

Solution:

a) $Q_{\text{latent}} = Q_{\text{Total}} - Q_{\text{sensible}}$

$$Q_{\text{latent}} = 36,000 - 27,000 = 9000 \text{ Btu/h}$$

b) $\text{Sensible Heat Ratio (SHR)} = \dfrac{Q_{\text{sensible}}}{Q_{\text{Total}}}$

$$\text{SHR} = \frac{27{,}000}{36{,}000} = 0.75$$

c) $$\frac{36{,}000 \text{ Btu/h} \times 1900 \text{ h}}{13 \text{ Btu/h} \cdot \text{W} \times 1000 \text{ W/kW}} = 5262 \text{ kWh}$$

14.6 Minisplits, Multisplits, and Variable-Refrigerant-Flow (VRF) Systems

A minisplit is a packaged air-conditioning (cooling) system that is supplied as two components. The indoor component includes and evaporator coil, a blower or fan, and an expansion device. It is generally finished for installing in the finished space, and is of small capacity (0.5 to 1 ton [1.75 to 3.5 kW]). The outdoor unit is matched in capacity to the indoor unit, and contains the compressor, condenser, and control package.

A multisplit system uses indoor evaporator-blower units similar to the minisplit, but will connect several of the indoor units to one outdoor condensing (compressor/condenser) unit, with various capacity ranges up to about 8 tons (28 kW).

A variable-refrigerant-flow (VRF) system typically consists of a condensing section housing compressor(s) and condenser heat exchanger interconnected by a single set of refrigerant piping to multiple indoor direct-expansion (DX) evaporator fan-coil units. Thirty or more DX fan coil units can be connected to a single condensing section, depending on system design, and with capacity ranging from 0.5 to 8 tons (1.75 to 28 kW).

The DX fan coils are constant air volume, but use variable refrigerant flow through an electronic expansion valve. The electronic expansion valve reacts to several temperature-sensing devices such as return air, inlet and outlet refrigerant temperatures, or suction pressure. The electronic expansion valve modulates to maintain the desired set point.

14.6.1 Application

VRF systems are most commonly air-to-air, but are also available in a water-source (water-to-refrigerant) configuration. They can be configured for simultaneous heating and cooling operation, i.e., operating on a heat pump cycle with liquid, suction, and hot gas lines to each unit that contains the changeover valve assembly with some indoor fan coil units operating in heating and some in cooling, depending on requirements of each building zone.

Indoor units are typically direct-expansion evaporators using individual electronic expansion devices and dedicated microprocessor controls for individual control. Each indoor unit can be controlled by individual thermostat. The

outdoor unit may connect several indoor evaporator or heat pump units with capacities 130% or more than the outdoor condensing unit capacity.

14.6.2 Categories

VRF equipment is divided into three general categories: residential, light commercial, and applied. Residential equipment is single-phase unitary equipment with a cooling capacity of 65,000 Btu/h (19 kW) or less. Light commercial equipment is generally three-phase, with cooling capacity greater than 65,000 Btu/h, (19 kW) and is designed for small businesses and commercial properties. Applied equipment has cooling capacity higher than 135,000 Btu/h (40 kW) and is designed for large commercial buildings.

14.6.3 Refrigerant Circuit and Components

VRF heat pump systems use a two-pipe (liquid and suction gas) system; simultaneous heat and cool systems use the same system, as well as a hot gas line and flow device that determines the proper routing of refrigerant gas to a particular indoor unit.

VRF systems use a sophisticated refrigerant circuit that monitors mass flow, oil flow, and balance to ensure optimum performance. This is accomplished in unison with variable-speed compressors and condenser fan motors. Both of these components adjust their frequency in reaction to changing mass flow conditions and refrigerant operating pressures and temperatures. A dedicated microprocessor continuously monitors and controls these key components to ensure proper refrigerant is delivered to each indoor unit in cooling or heating.

14.6.4 Heating and Defrost Operation

In heating mode, VRF systems typically must defrost like any mechanical heat pump, using reverse cycle valves to temporarily operate the outdoor coil in cooling mode. Oil return and balance with the refrigerant circuit is managed by the microprocessor to ensure that any oil entrained in the low side of the system is brought back to the high side by increasing the refrigerant velocity using a high-frequency operation performed automatically based on hours of operation.

14.7 Water-Source Heat Pumps

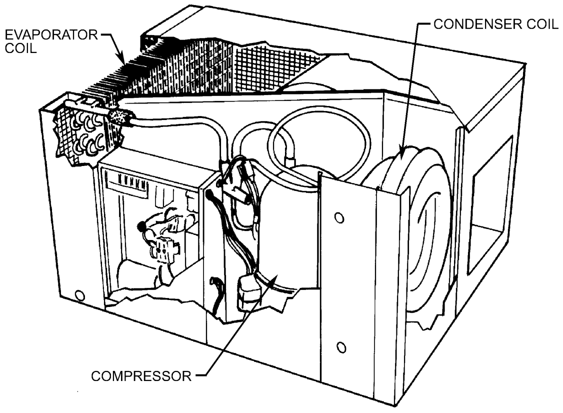
A water-source heat pump (WSHP) is a single-package reverse-cycle heat pump that uses water as the heat source for heating and as the heat sink for cooling. The water supply may be a recirculating closed loop, a well, a lake, or a stream. Water for closed-loop heat pumps is usually circulated at 2 to 3 gpm per ton (0.04 to 0.05 L/s per kW) of cooling capacity. **A groundwater heat pump (GWHP)** can operate with considerably less water flow. The main components of a WSHP refrigeration system are a compressor, refrigerant-to-water heat exchanger, refrigerant-to-air heat exchanger, refrigerant expansion devices, and refrigerant-reversing valve.

Designs of packaged WSHPs range from horizontal units located primarily above the ceiling or on the roof, to vertical units usually located in basements or equipment rooms, to console units located in the conditioned space. Figure 14-10 illustrates typical designs.

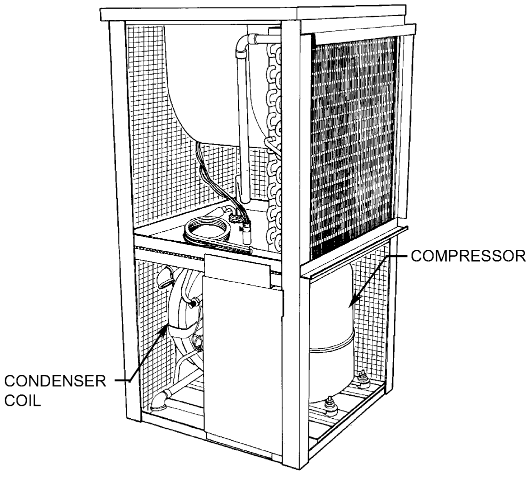
Systems. WHSPs are used in a variety of systems, such as:

- Water-loop heat pump systems
- Groundwater heat pump systems
- Closed-loop surface-water heat pump systems
- Surface-water heat pump systems
- Ground-coupled heat pump systems

A **water-loop heat pump (WLHP)** uses a circulating water loop as the heat source and heat sink. When loop water temperature exceeds a certain level during cooling, a cooling tower dissipates heat from the water loop into the atmosphere. When loop water temperature drops below a prescribed level during heating, heat is added to the circulating loop water,



Horizontal



Vertical

Fig. 14-10 Typical Arrangements of Water-Source Heat Pump
(Figures 14 and 15, Chapter 49, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

usually with a boiler. In multiple-unit installations, some heat pumps may operate in cooling mode while others operate in heating, and controls are needed to keep loop water temperature within the prescribed limits. In commercial applications water-loop heat pumps should be used in conjunction with a dedicated outdoor air system, which provides the ventilation and humidity control.

A **groundwater heat pump (GWHP)**, sometimes referred to as a **geothermal heat pump**, should more accurately be referred to as a ground-coupled heat pump. When installing water pipes or refrigerant piping to serve as a heat exchanger with the ground above the water table, on the warm cycle (building cooling), the earth tends to shrink away from the warm pipes, forming an air space between the pipes and the earth. The ground-coupled heat pump utilizes a deep water well (usually 200 to 400 ft deep). Into this well is inserted a pipe loop: a supply and return line with a U-bend at the bottom. The piping is then encased in a heat transfer grout through which it transfers heat to or from the ground as the season requires.

Installing this type of system requires detailed knowledge of the climate; site; soil temperature, moisture content, and thermal characteristics; and performance, design, and installation of water-to-earth heat exchangers.

Entering Water Temperatures. These various water sources provide a wide range of entering water temperatures to WSHPs. Entering water temperatures vary not only by water source but also by climate and time of year. Because of the wide range of entering water or brine temperatures encountered, it is not feasible to design a universal packaged product that can handle the full range of possibilities effectively. Therefore, WSHPs are rated for performance at a number of standard rating conditions.

14.8 Problems

14.1 An air-cooled packaged air conditioning unit with a hot water heating coil is to be used to condition a small office suite in a high-rise office building. The unit has a total cooling capacity of three tons of refrigeration, and the power requirement to the compressor is 1 kW per ton of cooling.

How many cfm of air must be brought into the condenser from an ambient outdoor temperature of 95°F db and 78°F wb if the condensing temperature is to be 115°F with a 10°F approach to the leaving air temperature?

14.2 If the ductwork supplying the air to and from the condenser section in Problem 14.1 were sized for a velocity of 800 ft/min, what would be the cross-sectional area of the ductwork?

(a) From the outdoors to the condenser?

(b) From the condenser back to the outdoors?

14.3 In passing through the condenser coil, the air would be heated a a constant humidity ratio. Air at 95°F db and 78°F wb (w = 117.49 gr/lb) heated to 115°F db has a final specific volume (v) of 14.85 ft^3/lb.

14.4 If the packaged air-conditioning unit of Problem 14.1 were provided with a water-cooled condenser instead of an air-cooled unit, and 1) the water was supplied at 85°F, 2) the leaving water temperature was 95°F, and 3) the condensing temperature was 105°F, what would be

(a) The Carnot COP between 40°F suction temperature and the 90°F condensing temperature?

(b) The Carnot COP between the 40°F suction temperature and the 105°F condensing temperature of the Problem 14.1 air-cooled unit?

14.5 Assuming that the actual power requirement for the cooling cycles of Problems 14.1 and 14.3 were proportioned in the same relationship as the Carnot COPs of Problem 14.3, what would be the kW per ton for the water-cooled unit of Problem 14.3?

14.6 How many gallons per minute of water would be required for the water-cooled unit of Problem 14.3?

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Chapter 15

PANEL HEATING AND COOLING SYSTEMS

This chapter discusses the principles and equipment available for panel heating and cooling systems. Additional details can be found in Chapter 6, "Radiant Heating and Cooling," in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment.*

15.1 General

Radiant heating and cooling systems combine temperature control of room surfaces with central air conditioning. Radiant surfaces may be located in the floor, walls, or ceiling, and the temperature is maintained by circulating water or air or by electric resistance. Where heating-cooling panel systems are used in commercial and institutional applications, they must be supplemented with a dedicated outdoor air system (DOAS) that must provide the ventilation, filtration, air motion, and all of the space humidity control. On the cooling cycle, the system must ensure that the room dew-point temperature is always above the lowest panel or chilled-water supply pipe temperature. A controlled-temperature surface is called a **radiant panel** if 50% or more of the heat transfer is by radiation to other surfaces.

Residential heating-only applications usually consist of pipe coils embedded in wood or masonry floors or plaster ceilings. This construction serves well where loads are relatively stable and where solar effects are minimized by building design. However, in buildings with large glass areas and rapid load changes, the slow response, lag, and override effect of concrete or masonry panels is unsatisfactory. Lightweight metal panel ceiling systems quickly respond to load changes and are used for cooling as well as heating in commercial and institutional applications.

Warm air and electric heating elements are used where local factors influence such use. In the warm air system, air is supplied to a cavity behind, under, or encapsulated in the panel surface. The air may leave the cavity through a normal diffuser and flow into the room. These systems are used as floor radiant panels in schools and in floors subject to extreme cold, such as over an overhang. Electric heating elements embedded in the floor or ceiling construction and unitized electric ceiling panels are used in residences, apartments, and various applications for local spot heating. Two factors to consider when using electric radiant panels are local electric codes and the relative difference between electric and fossil fuel heating costs.

The radiant panel is often located in the ceiling of a room. A ceiling is used because it sees all other surfaces and objects in the room; it is not subject to unpredictable coverings, as are floors; for heating, higher surface temperatures can be used; it is of smaller mass and therefore has quicker response to load changes; radiant cooling can be incorporated, and, in the case of the metal ceiling system, the piping is accessible for servicing.

Ceiling panel systems commonly used are an outgrowth of the perforated metal, suspended, acoustical ceiling. These radiant ceiling systems are usually designed into buildings where the features of the suspended acoustical ceiling can be combined with panel heating and cooling. The panels can be designed as small units to fit the building module and provide extensive flexibility for zoning and control, or, for maximum economy, the panels can be arranged as large continuous areas. Two types of metal ceiling systems are available. One type consists of lightweight aluminum panels, usually 12 in. by 24 in. (305 mm by 610 mm), that are attached in the field to 0.5 in. (15 mm) galvanized pipe coils. The second type consists of a copper coil metallurgically bonded to the aluminum face sheet forming a modular panel. Modular panels are available in sizes up to approximately 36 in. by 60 in. (910 mm by 1520 mm) and are held in position by various ceiling suspension systems.

The arrangement of components in radiant panel systems is similar to other air-water systems. Room temperature conditions are primarily maintained by a combination of direct transfer of radiant energy, and by convective heating and cooling. The room heating and cooling loads are calculated in the conventional manner. Manufacturers generally rate their equipment in the form of total performance, which can be applied directly to the calculated room load for heating and to the room sensible load for cooling.

These are the principal advantages of panel heating and cooling systems:

1. If they are properly designed; because of the low airflow quantities these systems can be very energy efficient.
2. Panel systems do not require any mechanical heat exchange equipment at the outside walls, thus simplifying the wall, floor, and structural systems.
3. All pumps, fans, filters, and so forth, are centrally located, thereby centralizing maintenance and operation.
4. Cooling or heating may be obtained during any season, without central zoning or seasonal changeover, when four-pipe systems are used.
5. Supply air quantities usually do not exceed those required for ventilation and dehumidification.
6. No mechanical equipment requiring maintenance or repair is placed within the occupied space, except possibly the control valves.
7. Draperies and curtains can be installed at the outside wall without interfering with heating and cooling systems.

8. The modular panel provides flexibility to meet changes in partitioning.
9. No space is required within the air-conditioned room for the mechanical equipment. This feature is especially valuable when compared to other conditioning methods for applications in existing buildings, hospital patient rooms, and other applications where space is at a premium and where maximum cleanliness is essential.
10. A common central air system for ventilation and dehumidification can serve both the interior and perimeter zones.
11. Wet surface cooling coils are eliminated from the occupied space, thus reducing the potential for septic contamination.

Other essential factors when considering the use of panel systems are as follows:

1. Evaluate early to plan an optimum physical arrangement of the building to take full advantage of the panel system.
2. Select recessed lighting fixtures, air diffusers, hung ceiling, and other ceiling devices to provide the maximum ceiling area possible for use as radiant panels.
3. The air-side design must maintain the room dew-point temperature below the lowest temperature of panel surface at all times to eliminate any possibility of condensation on the panels. The systems must be interlocked to shut down the chilled water to the panels if the dehumidifying system fails.
4. As with any hydronic system, design the piping system to avoid noises from entrained air, high velocity or high pressure drop devices, or from pump and pipe vibrations.
5. Anticipate thermal expansion of the ceiling and other devices in or adjacent to the ceiling.

15.2 Types

The most common forms of panels applied in panel heating and cooling systems are

- Metal ceiling panels
- Embedded piping in ceilings, walls, or floors (heat only)
- Air-heated floors
- Electrically heated ceilings or floors

Metal Ceiling Panels. Metal ceiling panels are often integrated into a system that both heats and cools. In such a system, a source of dehumidified ventilation air is required. This system must provide all of the ventilation air and all of the humidity control, as well as pressurize the building to avoid any significant infiltration. In such a system, various amounts of forced air are supplied year-round. (See section 12.4.)

A metal ceiling panel system using copper tubing metallurgically bonded to an aluminum panel is shown in Figure 15-1. This panel can be mounted into various ceiling suspension systems.

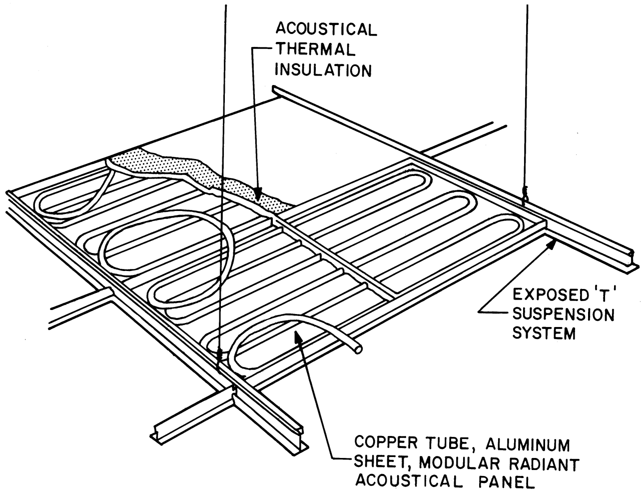


Fig. 15-1 Metal Ceiling Panels Metallurgically Bonded to Copper Tubing

(Figure 13, Chapter 6, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

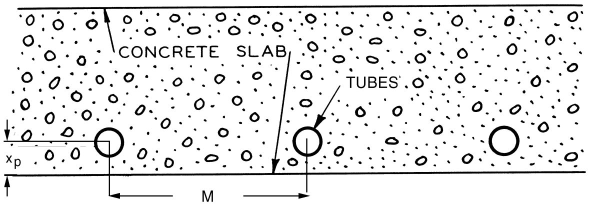


Fig. 15-2 Coils in Structural Concrete Slab

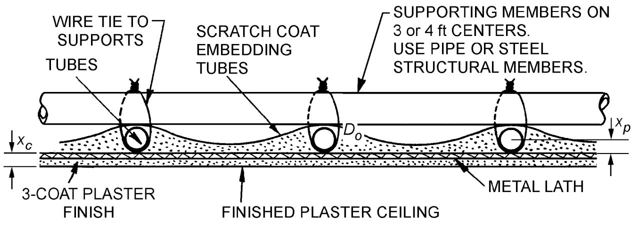
(Figure 16, Chapter 6, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Two-pipe and four-pipe distribution systems have been used successfully with metal ceiling panels. Common design practice calls for a 20°F (11°C) drop for heating across a given grid and a 5°F (3°C) rise for cooling, but higher temperature differentials may apply in some cases.

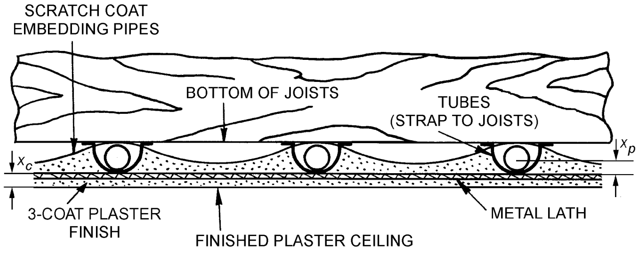
Some ceiling installations require that active grids cover only a part of the room, and consequently, compatible matching standard acoustical panels are normally used for the remaining ceiling area.

Embedded Piping in Ceilings, Walls, and Floors. When piping is embedded in ceilings, the construction used is generally one of the following:

1. Pipe or tube is embedded in the lower portion of a concrete slab, generally within an inch of its lower surface. If plaster is to be applied to the concrete, the piping may be placed directly on the wood forms. If the slab is to be used without plaster finish, then the piping should be installed not less than 0.75 in. (19 mm) above the undersurface of the slab. This method of construction is shown in Figure 15-2. The minimum coverage must be in compliance with the local building code requirements.
2. Pipe or tube is embedded in a metal lath and plaster ceiling. If the lath is suspended to form a hung ceiling, both the lath and the heating coils are securely wired to the supporting members in such a way that the lath is below, but in good



SUSPENDED PLASTER CEILING



PLASTER CEILING BELOW JOISTS

Fig. 15-3 Coils in Plaster Above Lath
(Figure 17, Chapter 6, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

contact with, the coils. Plaster is then applied to the metal lath, with care being taken to embed the coil.

3. Copper tube of the smaller diameters or cross-linked polyethylene (PEX) tubing is attached to the underside of a wire lath or gypsum lath. Plaster is then applied to the lath to embed the tube (Figure 15-3).
4. Other forms of ceiling construction are composition board, wood paneling, etc., with warm water piping, tube, or channels built into the panel sections.

Coils are usually of the sinuous type, although some header or grid-type coils have been used in ceilings. Coils may be of plastic (PEX), ferrous, or nonferrous pipe or tube, with coil pipes spaced from 4.5 to 9 in. (115 to 230 mm) on centers, depending on the required output, pipe or tube size, and other factors.

Although not so universally used as ceiling panels, wall panels may be constructed by any of the methods described for ceilings.

The construction for piping embedded in floors depends on whether (1) the floor is laid on grade or (2) the floor is above grade.

On-Grade Floor. Plastic (PEX), ferrous, and nonferrous pipe and tube are used in floor slabs which rest on grade. The coils are constructed as either sinuous, continuous pipe coils or arranged as heater coils with the pipes spaced from 6 to 18 in. (150 to 460 mm) on centers. The coils are generally installed with 1.5 to 4 in. (40 to 100 mm) of cover above the coils. Insulation should be used to reduce the perimeter and reverse side losses. Illustrated in Figure 15-4 is the application of pipe coils in slabs resting on grade. Coils should be embedded completely and should not rest on an interface. Any supports used for positioning the heating coils should be nonabsorbent and inorganic.

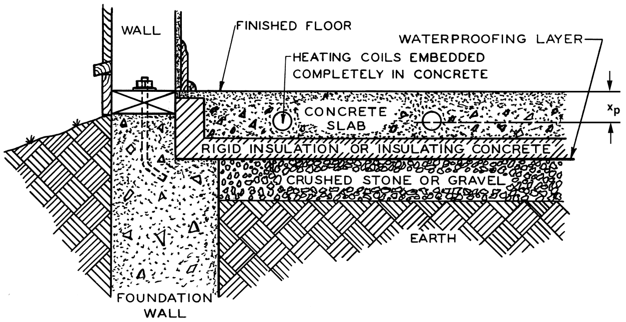


Fig. 15-4 Coils in Floor Slab on Grade
(Figure 19, Chapter 6, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

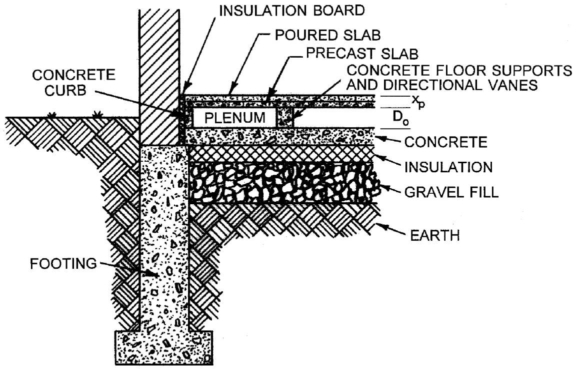


Fig. 15-5 Warm Air Floor Panel Construction
(Figure 27, Chapter 6, 2012 ASHRAE Handbook—HVAC Systems and Equipment)

Above-Grade Floor. Where the coils are embedded in structural load-supporting slabs above grade, construction codes may affect their position. Otherwise, the coil piping is installed in the same manner as described for slabs resting on grade. Except the pipes should be installed in the wearing (finish) concrete rather than in the structural concrete.

Air-Heated Floors. Several methods have been devised to warm interior room surfaces by circulating heated air through passages in the floor. In some cases, the heated air is recirculated in a closed system. In others, all or part of the air is passed through the room on its way back to the furnace or air-handling unit to provide supplementary heating and ventilation (Figure 15-5).

Electrically Heated Ceilings. Several types of electric resistance units are available for heating interior room surfaces. These include (1) electric heating cables that may be embedded in concrete or plaster or laminated in drywall ceiling construction; (2) prefabricated electric heating panels to be attached to room surfaces; and (3) electrically heated fabrics or other materials for application to, or incorporation into, finished room surfaces.

Ceiling Cables. The details of ceiling cable installation for plastered and drywall construction is shown in Figure 15-6.

Electric Heating Panels. A variety of prefabricated electric heating panels are used for either supplemental or full

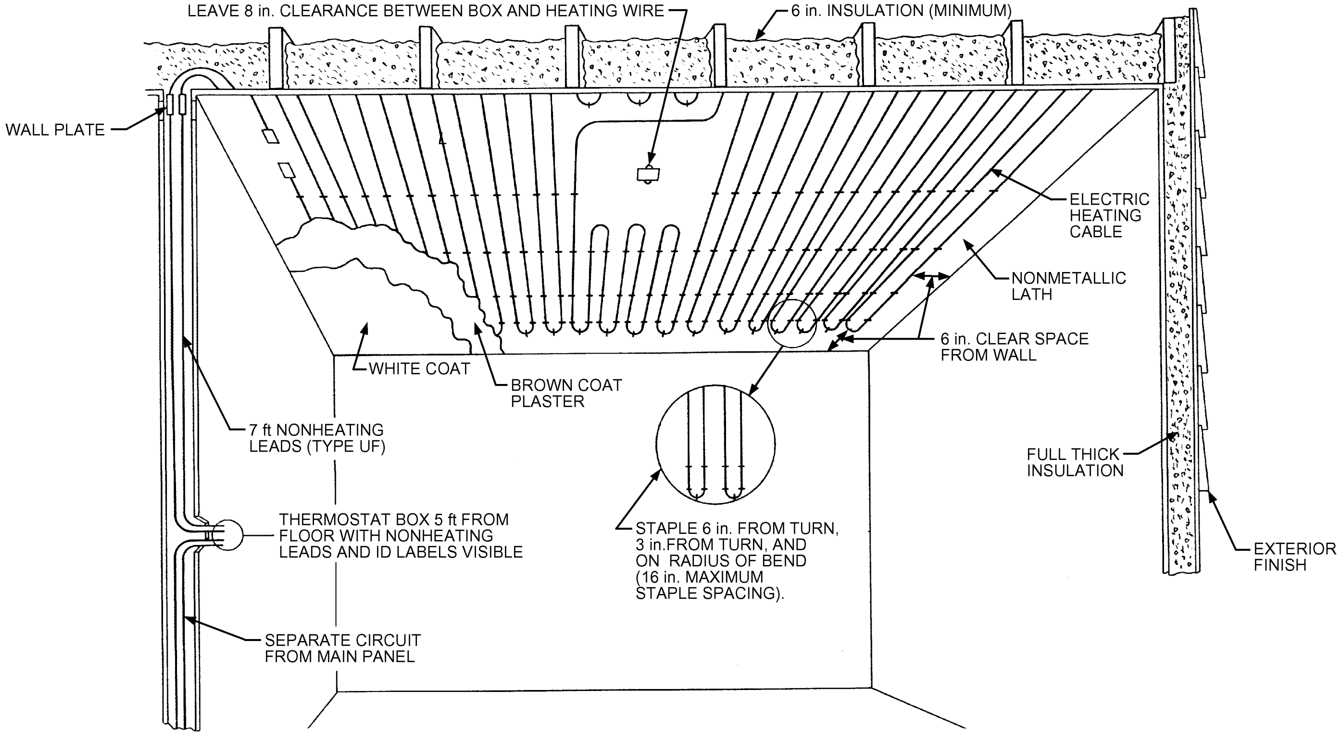


Fig. 15-6 Electric Heating Panel for Wet Plastered Ceiling
(Figure 24, Chapter 6, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

room heating. These panels are available in sizes from 2 ft by 4 ft to 6 ft by 12 ft (0.6 m by 1.2 m to 1.8 m by 3.6 m). They are constructed from a variety of materials such as gypsum board, glass, steel, and vinyl. Different panels have rated inputs varying from 10 to 95 W/ft^2 (108 to 1023 W/m^2) for 120, 208, and 240 V service. Maximum operating temperatures vary from about 100 to about 300°F (38 to 49°C) depending on watt density.

Panel heating elements may be embedded conductors, laminated conductive coatings, or printed circuits. Nonheating leads are connected and furnished as part of the panel.

Some panels may be cut to fit available space; others must be installed as received. Panels may be either flush or surface mounted. In some cases, they are finished as part of the ceiling. Rigid panels that are about 1 in. (25 mm) thick and weigh about 25 lb (11 kg) each are available to fit standard 2 ft by 4 ft (0.6 m by 1.2 m) modular tee-bar ceilings.

Cable embedded in walls, similar to ceiling construction, is occasionally found in Europe. Because of possible damage due to nails driven for hanging pictures or because of building alteration, most codes prohibit such panels in the United States.

Some of the prefabricated panels described in the preceding section are also used for wall panel heating.

Electric heating cable assemblies, such as those used for ceiling panels, are sometimes used for concrete floor heating systems.

15.3 Design Steps

Panel design requires specification of the following: panel area, size and location of the heating elements in the panel, insulation on the reverse side and edge of the panel, required input to panel, and temperature of the heating elements. Specific procedures are given in Chapter 6 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. The procedure is summarized as follows:

1. Calculate heat loss for each room.
2. Determine the available area for panels in each room.
3. Calculate the required unit panel output.
4. Determine the required panel surface temperature.
5. Select the means of heating the panel and the size and location of the heating elements.
6. Select insulation for the reverse side and edge of panel.
7. Determine panel heat loss and required input to the panel.
8. Determine the other temperatures that are required or developed.
9. Design the system for heating the panels in accordance with conventional practice and manufacturers recommendations.

In the steps outlined for design, the effect of each assumption or choice on comfort should be considered carefully. The following general rules should be followed:

1. Place panels near cold areas where heat losses occur.

2. Do not use high-temperature ceiling panels in very low ceilings.
3. Keep floor panels temperatures at or below 85°F (30°C).

Example 15-1 The living room in a home is occupied by adults in light clothing and engaged in sedentary activity. The room has a net outside wall area of 275 ft^2 with a surface temperature of 54°F, 45 ft^2 of glass with a surface temperature of 20°F; 540 ft^2 of ceiling with a surface temperature of 60°F; 670 ft^2 of partitions with a surface temperature of 70°F; and 540 ft^2 of floor with a surface temperature of 70°F. If the air movement is 20 fpm, determine the air temperature necessary for comfort.

Solution:

$$\text{Mean radiant temperature} = \text{MRT} =$$

$$\text{MRT} = \frac{275(54) + 45(20) + 540(60) + 670(70) + 540(70)}{275 + 45 + 540 + 670 + 540}$$

$$\text{MRT} = 64.1°\text{F}$$

for sedentary activity with light clothing at 20 fpm from Figure 4-3:

$$t_{\text{dry bulb}} = 90°\text{F for comfort}$$

15.4 Problems

15.1 A room has a net outside wall area of 300 ft^2 that has a surface temperature of 55°F; 50 ft^2 of glass with a surface temperature of 30°F; 560 ft^2 of ceiling with a surface temperature of 70°F; and 560 ft^2 with a surface temperature of 70°F. Estimate the average unheated surface temperature or the area-weighted mean radiant temperature. [Ans: 65.6°F]

15.2 For the room in Problem 15.1, estimate the following:

(a) radiant output for a 100 ft^2 heating panel with a panel surface temperature of 120°F

(b) natural convection output for the ceiling panel when the air temperature is 70°F

15.3 A room has 1500 ft^2 of surface area and 320 ft^2 is to be heated. The average unheated surface temperature in the room is 67°F. The air temperature in the room is 75°F. The room is occupied by adults in light clothing at a sedentary activity. Determine the surface temperature of the heated panel necessary to produce comfort if the air velocity is 20 fpm. [Ans: 131°F]

15.4 For Problem 15.3, determine the total heat transferred by the ceiling heating panel.

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Chapter 16

HEAT PUMP, COGENERATION, AND HEAT RECOVERY SYSTEMS

This chapter discusses applied heat pump systems, heat recovery systems, and cogeneration systems. Specific details on these subjects can be found in Chapters 7 and 9 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

16.1 General

As described in Chapter 2, a heat pump extracts heat from a source and transfers it to a sink at a higher temperature. According to this definition, all pieces of refrigeration equipment, including air conditioners and chillers with refrigeration cycles, are heat pumps. In engineering, however, the term *heat pump* is generally reserved for equipment that heats for beneficial purposes, rather than that which removes heat for cooling only. Dual-mode heat pumps alternately provide heating, cooling, or both simultaneously. Heat reclaim heat pumps provide heating only or simultaneous heating and cooling. An applied heat pump requires field engineering for the specific application, in contrast to the use of a manufacturer-designed unitary product. Applied heat pumps include built-up heat pumps (field- or custom-assembled from components) and industrial process heat pumps. Most current heat pumps use a vapor compression (modified Rankine) cycle or absorption cycle. Any of the other refrigeration cycles discussed in Chapter 2 of the 2017 *ASHRAE Handbook—Fundamentals* are also suitable. Although most heat pump compressors are powered by electric motors, use is also made of engine and turbine drives that can add engine coolant or "waste" heat to the heat generated. Applied heat pumps are most commonly used for heating and cooling buildings, but they are occasionally used for domestic and service water heating, pool heating, and industrial process heating.

Applied heat pumps with capacities from 24,000 to 150,000,000 Btu/h (7 to 45,000 kW) operate in many facilities. Some machines are capable of output water temperatures up to 220°F and steam pressures up to 60 psig [415 kPa (gage)].

Compressors in large systems vary from one or more reciprocating, scroll, or screw types to single- or multistaged centrifugal types. A single or central system is often used, but in some instances, multiple or unitary heat pump systems are used (Chapter 14) to facilitate zoning. Heat sources include the ground, well water, surface water, gray water, solar energy, the air, internal building heat, and a hydronic water circuit that is heated or cooled. Compression can be single-stage or multistage. Frequently, heating and cooling are supplied simultaneously to separate zones.

Decentralized systems with water loop heat pumps are common, using multiple water-source heat pumps connected to a common circulating water loop. They can also include ground coupling, heat rejectors (cooling towers and dry coolers), supplementary heaters (boilers and steam heat exchangers), loop reclaim heat pumps, solar collection devices, and thermal storage.

Community and district heating and cooling systems can utilize both centralized and distributed heat pump systems.

16.2 Types of Heat Pumps

Heat pumps are classified by (1) heat source and sink, (2) heating and cooling distribution fluid, (3) thermodynamic cycle, (4) building structure, (5) size and configuration, and (6) limitation of the source and sink. Table 16-1 shows the more common types of closed vapor-compression cycle heat pumps for heating and cooling service.

Air-to-Air Heat Pumps. This type of heat pump is quite common and is particularly suitable for factory-built unitary heat pumps. It is widely used in residential and commercial applications (see Chapter 14). The first diagram in Table 16-1 is a typical refrigeration circuit.

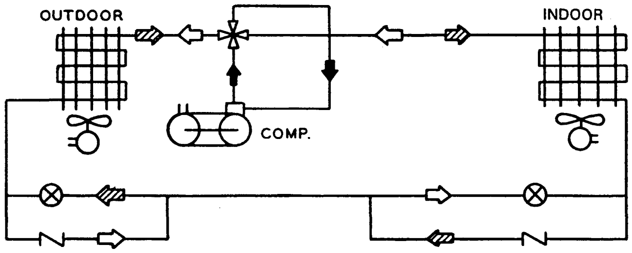
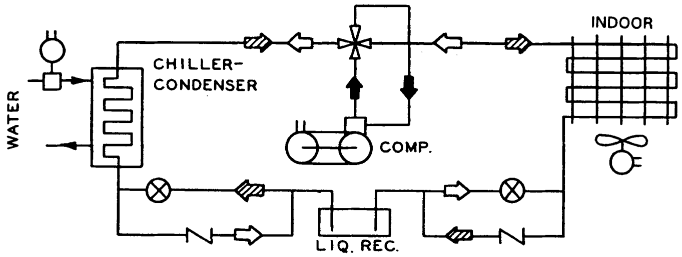
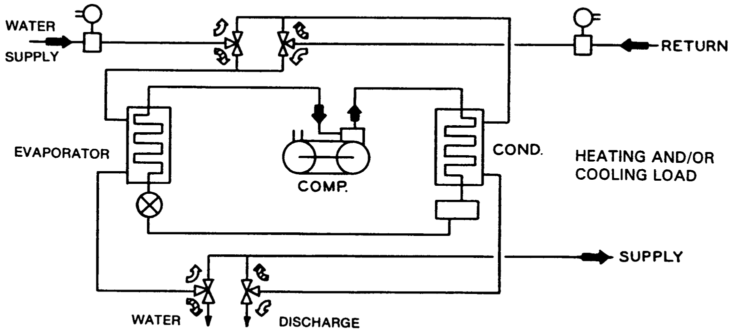
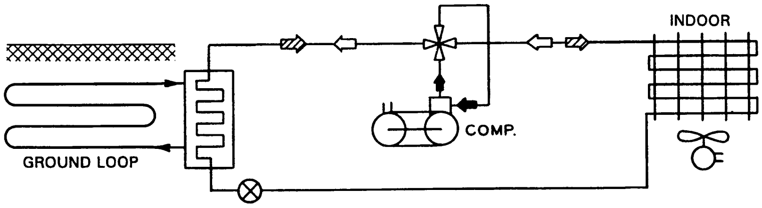
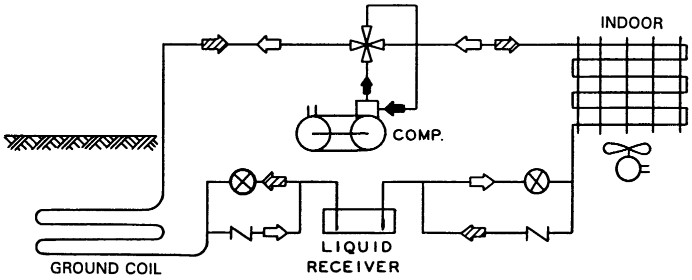
In other air-to-air heat pump systems, air circuits can be interchanged by motor-driven or manually operated dampers to obtain either heated or cooled air for the conditioned space. In that system, one heat exchanger coil is always the evaporator, and the other is always the condenser. Conditioned air passes over the evaporator during the cooling cycle, and outdoor air passes over the condenser. Damper positioning causes the change from cooling to heating.

Water-to-Air Heat Pumps. These heat pumps rely on water as the heat source and sink and use air to transmit heat to or from the conditioned space. (See the second diagram in Table 16-1.) They include the following:

- ***Groundwater heat pumps***, which use groundwater from wells as a heat source and/or sink. They can either circulate source water directly to the heat pump or use an intermediate fluid in a closed loop, similar to the ground-coupled heat pump.
- ***Surface water heat pumps***, which use surface water from a lake, pond, or stream as a heat source or sink. As with ground-coupled and groundwater heat pumps, these systems

Table 16-1 Common Types of Heat Pumps

(Figure 5, Chapter 9, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Heat Source and Sink	Distribution Fluid	Thermal Cycle	Diagram ▨ Heating ⇨ Cooling ➡ Heating and Cooling
Air	Air	Refrigerant changeover	
Water	Air	Refrigerant changeover	
Water	Water	Water changeover	
Ground-coupled (or Closed-loop ground-source)	Air	Refrigerant changeover	
Ground-source, Direct-expansion	Air	Refrigerant changeover	

can either circulate source water directly to the heat pump or use an intermediate fluid in a closed loop.

- ***Internal-source heat pumps***, which use high internal cooling load generated in buildings either directly or with storage. These include water-loop heat pumps and variable refrigerant flow heat pump systems.
- ***Solar-assisted heat pumps***, which rely on low-temperature solar energy as the heat source. Solar heat pumps may resemble water-to-air, or other types, depending on the form of solar heat collector and the type of heating and cooling distribution system.
- ***Wastewater-source heat pumps***, which use sanitary waste heat or laundry waste heat as a heat source. Waste fluid can be introduced directly into the heat pump evaporator after waste filtration, or it can be taken from a storage tank, depending on the application. An intermediate loop may also be used for heat transfer between the evaporator and the waste heat source.

Water-to-Water Heat Pumps. These heat pumps use water as the heat source and sink for cooling and heating. Heating/cooling changeover can be done in the refrigerant circuit, but it is often more convenient to perform the switching in the water circuits, as shown in the third diagram of Table 16-1. Although the diagram shows direct admittance of the water source to the evaporator, in some cases, it may be necessary to apply the water source indirectly through a heat exchanger (or double-wall evaporator) to avoid contaminating the closed chilled-water system, which is normally treated. Another configuration employs a closed-circuit condenser water system that is a water chiller of which the condenser water is a hydronic heating simultaneously with the chilled-water serving as a chilled water circuit.

Ground-Coupled Heat Pumps. These use the ground as a heat source and sink. A heat pump may have a refrigerant-to-water heat exchanger or may be direct-expansion (DX). Both types are shown in Table 16-1. In systems with refrigerant-to-water heat exchangers, a water or antifreeze solution is pumped through horizontal, vertical, or coiled pipes embedded in the ground. Direct-expansion ground-coupled heat pumps use refrigerant in direct-expansion, flooded, or recirculation evaporator circuits for the ground pipe coils.

A common configuration of ground-coupled heat pump employs a deep well (usually 6 in. diameter and several hundred feet deep). Into the well is inserted a supply and return water pipe loop of high-pressure plastic pipe with a U bend at the bottom, which serves as a heat exchanger to either reject heat to or obtain heat from the ground. After inserting the pipe, the well is filled with a heat transfer grout that holds the pipe in place and protects the water table from contamination by any undesirable surface materials. Depending upon the depth of the well, the depth of the water table, the heat transfer characteristics of the well construction, and the soil, the capacity of each well is usually between 3 and 5 tons (10 and 18 kW) of heat rejection capacity. They are often used singly for residential applications, and in multiple “fields” of wells, spaced 20 to 40 ft (6 to 12 m) apart, for commercial and institutional installations. These systems are sometimes called geothermal heat pumps.

Soil type, moisture content, composition, density, and uniformity close to the surrounding field areas affect the success of this method of heat exchange of any ground-coupled heat exchange. With some piping materials, the material of construction for the pipe and the corrosiveness of the local soil and underground water may affect the heat transfer and service life. In a variation of this cycle, all or part of the heat from the evaporator plus the heat of compression are transferred to a water-cooled condenser. This condenser heat is then available for uses such as heating air or domestic hot water.

Additional heat pump types include the following:

Air-to-Water Heat Pumps Without Changeover. These are commonly called *heat pump water heaters*.

Refrigerant-to-Water Heat Pumps. These condense a refrigerant by the cascade principle. Cascading pumps the heat to a higher temperature, where it is rejected to water or another liquid. This type of heat pump can also serve as a condensing unit to cool almost any fluid or process. More than one heat source can be used to offset those times when insufficient heat is available from the primary source.

16.3 Heat Sources and Sinks

Table 16-2 shows the principal media used as heat sources and sinks. Selecting a heat source and sink for an application is primarily influenced by geographic location, climate, initial cost, availability, and type of structure. Table 16-2 presents various factors to be considered for each medium.

Air. Outdoor air is a universal heat source and sink medium for heat pumps and is widely used in residential and light commercial systems. Extended-surface, forced-convection heat transfer coils transfer heat between the air and refrigerant. Typically, the surface area of outdoor coils is considerably larger than that of indoor coils. The volume of outdoor air handled is also greater than the volume of indoor air handled. During heating, the temperature of the evaporating refrigerant is generally 10°F to 20°F (5 to 10 °C) less than the outdoor air temperature.

When selecting or designing an air-source heat pump, two factors in particular must be considered: (1) the local outdoor air temperature and (2) frost formation.

As the outdoor temperature decreases, the heating capacity of an air-source heat pump decreases. This makes equipment selection for a given outdoor heating design temperature more critical for an air-source heat pump than for a fuel-fired system. Equipment must be sized for as low a balance point as is practical for heating, often requiring much more compressor capacity for heating than for cooling. Many heat pumps utilize auxillary heating in cold climates and only utilize the heat pump to the outdoor air limits of the air-conditioning compressor.

When the surface temperature of an outdoor air coil is 32°F (0°C) or less, with a corresponding outdoor air dry-bulb

Table 16-2 Heat Pump Sources and Sinks

(Table 1, Chapter 9, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

		Suitability		Availability		Cost		Temperature		Common Practice	
Medium	**Examples**	**Heat Source**	**Heat Sink**	**Location Relative to Need**	**Coincidence with Need**	**Installed**	**Operation and Maintenance**	**Level**	**Variation**	**Use**	**Limitations**
AIR											
Outdoor	Ambient air	Good, but efficiency and capacity in heating mode decrease with decreasing outdoor air temperature	Good, but efficiency and capacity in cooling mode decrease with increasing outdoor air temperature	Universal	Continuous	Low	Moderate	Variable	Generally extreme	Most common, many standard products	Defrosting and supplemental heat usually required
Exhaust	Building ventilation	Excellent	Fair	Excellent if planned for in building design	Excellent	Low to moderate	Low unless exhaust is laden with dirt or grease	Excellent	Very low	Excellent as energy-conservation measure	Insufficient for typical loads
WATER											
Well[*]	Ground-water well may also provide a potable water source	Excellent	Excellent	Poor to excellent, practical depth varies by location	Continuous	Low if existing well used or shallow wells suitable; can be high otherwise	Low, but periodic maintenance required	Generally excellent, varies by location	Extremely stable	Common	Water disposal and required permits may limit; may require double-wall exchangers; may foul or scale
Surface	Lakes, rivers, oceans	Excellent for large water bodies or high flow rates	Excellent for large water bodies or high flow rates	Limited, depends on proximity	Usually continuous	Depends on proximity and water quality	Depends on proximity and water quality	Usually satisfactory	Depends on source	Available, particularly for fresh water	Often regulated or prohibited; may clog, foul, or scale
Tap (city)	Municipal water supply	Excellent	Excellent	Excellent	Continuous	Low	Low energy cost, but water use and disposal may be costly	Excellent	Usually very low	Use is decreasing due to regulations	Use or disposal may be regulated or prohibited; may corrode or scale
Condensing	Cooling towers, refrigeration systems	Excellent	Poor to good	Varies	Varies with cooling loads	Usually low	Moderate	Favorable as heat source	Depends on source	Available	Suitable only if heating need is coincident with heat rejection
Closed loops	Building water-loop heat pump systems	Good, loop may need supplemental heat	Favorable, may need loop heat rejection	Excellent if designed as such	As needed	Low	Low to moderate	As designed	As designed	Very common	Most suitable for medium or large buildings
Waste	Raw or treated sewage, gray water	Fair to excellent	Fair, varies with source	Varies	Varies, may be adequate	Depends on proximity, high for raw sewage	Varies, may be high for raw sewage	Excellent	Usually low	Uncommon, practical only in large systems	Usually regulated; may clog, foul, scale, or corrode
GROUND[a]											
Ground-coupled	Buried or submerged fluid loops	Good if ground is moist, otherwise poor	Fair to good if ground is moist, otherwise poor	Depends on soil suitability	Continuous	High to moderate	Low	Usually good	Low, particularly for vertical systems	Rapidly increasing	High initial costs for ground loop
Direct-expansion	Refrigerant circulated in ground coil	Varies with soil conditions	Varies with soil conditions	Varies with soil conditions	Continuous	High	High	Varies by design	Generally low	Extremely limited	Leak repair very expensive; requires large refrigerant quantities
SOLAR ENERGY											
Direct or heated water	Solar collectors and panels	Fair	Poor, usually unacceptable	Universal	Highly intermittent, night use requires storage	Extremely high	Moderate to high	Varies	Extreme	Very limited	Supplemental source or storage required
INDUSTRIAL PROCESS											
Process heat or exhaust	Distillation, molding, refining, washing, drying	Fair to excellent	Varies, often impractical	Varies	Varies	Varies	Generally low	Varies	Varies	Varies	May be costly unless heat need is near rejected source

[a] Groundwater-source heat pumps are also considered ground-source heat pump systems.

temperature 4°F to 10°F (2°C to 5°C) higher, frost may form on the coil surface. If allowed to accumulate, frost inhibits heat transfer; therefore, the outdoor coil must be defrosted periodically. The number of defrosting operations is influenced by the climate, air-coil design, and the hours of operation. Experience shows that, generally, little defrosting is required when outdoor air conditions are below 17°F and 60% rh. However, under very humid conditions, when small suspended water droplets are present in the air, the rate of frost deposit may be about three times as great as predicted from psychrometric theory and the heat pump may require defrosting after as little as 20 minutes of operation. The loss of available heating capacity caused by frosting should be considered when sizing an air-source heat pump utilizing outdoor air.

Following commercial refrigeration practice, early designs of air-source heat pumps had relatively wide fin spacing of 4 to 5 fins/in., (5 to 6 mm apart) based on the theory that this would minimize defrosting frequency. However, experience has shown that effective hot-gas defrosting allows much closer fin spacing and reduces the system's size and bulk. In current practice, fin spacings of 10 to 20 fins/in. (1.5 to 2.5 mm apart) are sometimes used.

In many institutional and commercial buildings, some air must be continuously exhausted year-round. This exhaust air can be used as a heat source for some configurations of a heat recovery system.

High humidity caused by indoor swimming pools causes condensation on ceiling structural members, walls, windows, and floors and causes discomfort to spectators. Traditionally, outdoor air and dehumidification coils with reheat from a boiler that also heats the pool water are used. This is ideal for air-to-air and air-to-water heat pumps because energy costs can be reduced. Suitable materials must be chosen so that heat pump components are resistant to corrosion from chlorine and high humidity.

Water. Water can be a satisfactory heat source, subject to the considerations listed in Table 16-2. City water is seldom used because of cost and municipal restrictions. Groundwater (well water) is particularly attractive as a heat source because of its relatively high and nearly constant temperature. Water temperature depends on source depth and climate but in the United States generally ranges from 40°F (4.5°C) in northern areas to 70°F (21°C) in southern areas. Frequently, sufficient water is available from wells. In some locations, under strict regulations, water can be reinjected into the aquifer. This use is nonconsumptive and, with proper design, only the water temperature changes. Water quality should be analyzed, and the possibility of scale formation and corrosion should be considered. In some instances, it may be necessary to separate the well fluid from the equipment with an additional heat exchanger. Special consideration must also be given to filtering and settling ponds for specific fluids. Other considerations are the costs of drilling, piping, pumping, and a means for disposal of used water. Information on well water availability, temperature, and chemical and physical analysis is available from US Geological Survey offices in most locations.

Heat exchangers may also be submerged in open ponds, lakes, or streams. When surface or stream water is used as a source, the temperature drop across the evaporator in winter may need to be limited to prevent freeze-up.

In industrial applications, waste process water (e.g., spent warm water in laundries, plant effluent, and warm condenser water) may be a heat source for heat pump operation.

Sewage, which often has temperatures higher than that of surface or groundwater, may be an acceptable heat source. Secondary effluent (treated sewage) is usually preferred, but untreated sewage may be used successfully with proper heat exchanger design.

Use of water during cooling follows the conventional practice for water-cooled condensers.

Water-to-refrigerant heat exchangers are generally direct-expansion or flooded water coolers, usually shell-and-coil or shell-and-tube. Brazed-plate heat exchangers may also be used. In large applied heat pumps, the water is usually reversed instead of the refrigerant.

Ground. The ground is used extensively as a heat source and sink, with heat transfer through buried coils. Soil composition, which varies widely from wet clay to sandy soil, has a predominant effect on thermal properties and expected overall performance. The heat transfer process in soil depends on transient heat flow. Thermal diffusivity is a dominant factor and is difficult to determine without local soil data. Thermal diffusivity is the ratio of thermal conductivity to the product of density and specific heat. The soil's moisture content influences its thermal conductivity.

There are three primary types of ground-source heat pumps: (1) groundwater, which is discussed in the previous section; (2) direct-expansion, in which the ground-to-refrigerant heat exchanger is buried underground; and (3) ground-coupled (also called closed-loop ground-source and geothermal), in which a secondary loop (sometimes with a brine) connects the ground-to-water and water-to-refrigerant heat exchangers (see Table 16-1).

Ground loops can be placed either horizontally or vertically. A horizontal system consists of single or multiple serpentine heat exchanger pipes buried 3 to 6 ft (1 to 2 m) apart in a horizontal plane at a depth 3 to 6 ft (1 to 2 m) below grade. Pipes may be buried deeper, but excavation costs and temperature must be considered. Horizontal systems can also use coiled loops referred to as *slinky coils*. A vertical system uses a concentric tube or U-tube heat exchanger. The design of ground-coupled heat exchangers is covered in Chapter 34 of the 2015 *ASHRAE Handbook—HVAC Applications*.

Solar Energy. Solar energy may be used either as the primary heat source or in combination with other sources. Air, surface water, shallow groundwater, and shallow ground-source systems all use solar energy indirectly. The principal advantage of using solar energy directly is that, when available, it provides

heat at a higher temperature than the indirect sources, increasing the heating coefficient of performance. Compared to solar heating without a heat pump, the collector efficiency and capacity are increased because a lower collector temperature is required.

Research and development of solar-source heat pumps has been concerned with two basic types of systems: direct and indirect. The direct system places refrigerant evaporator tubes in a solar collector, usually a flat-plate type. Research shows that a collector without glass cover plates can also extract heat from the outdoor air. The same surface may then serve as a condenser using outdoor air as a heat sink for cooling

An indirect system circulates either water or air through the solar collector. When air is used, the collector may be controlled in such a way that (1) the collector can serve as an outdoor air preheater, (2) the outdoor air loop can be closed so that all source heat is derived from the sun, or (3) the collector can be disconnected from the outdoor air serving as the source or sink.

16.4 Cogeneration

Cogeneration designates on-site electrical generating systems that salvage byproduct or waste heat from the generating process. The magnitude, duration, and coincidence of electrical and thermal loads must be analyzed, and prime movers and waste heat recovery systems must be selected to determine system feasibility and design. The basic components of the cogeneration plant are (1) prime mover, (2) generator, (3) waste heat recovery systems, (4) control systems, and (5) connections to building mechanical and electrical services.

The normal prime movers are reciprocating internal combustion engines, combustion gas turbines, expansion turbines, and steam boiler-turbine combinations. These units convert the heat in the fuel (liquid, solid, gaseous, or nuclear) into rotating shaft energy. Figure 16-1 is an example of this heat recovery.

Use of the prime mover heat determines overall system efficiency and is one of the critical factors of economic feasibility. Two kinds of energy are available from the prime mover: (1) mechanical energy from the shaft and (2) heat energy remaining after the fuel or steam has acted on the shaft.

Shaft loads (generators, centrifugal chillers, compressors, process equipment) require a given amount of rotating mechanical energy. Once the prime mover has been selected to provide the required shaft output, it has a fixed relationship to system efficiency that is dependent upon the prime mover fuel versus load and the load versus heat balance curves.

Steam turbine drives can be arranged to extract steam at intermediate turbine stages. The waste heat value of a steam turbine is the enthalpy of the steam at the point it is extracted from the turbine or at the turbine's exhaust outlet. This steam, reduced in pressure and temperature by the amount of shaft work, can be fed to heat exchange equipment, absorption chillers, and steam turbine-driven centrifugal chillers.

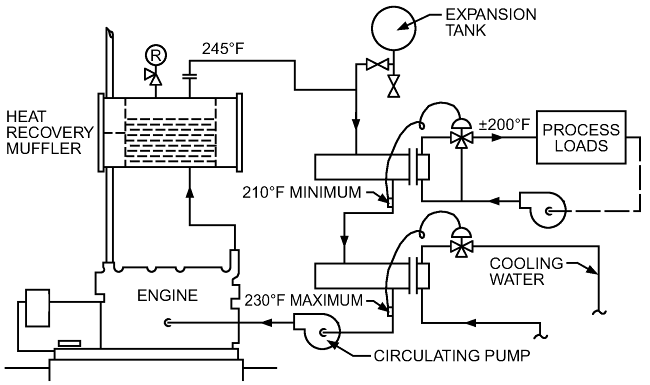


Fig. 16-1 Typical Reciprocating Engine Heat Recovery System

(Figure 46, Chapter 7, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

The gas turbine cycle has a thermal efficiency of approximately 20%, with the remainder of the fuel energy exhausted or radiated. A minimum exhaust temperature of approximately 300°F (150°C) is required to prevent condensation. The recoverable heat per unit of power is greater for a gas turbine than for a reciprocating engine because the power is less per unit of fuel input. Because gas turbine exhaust contains a large percentage of excess air, afterburners or boost burners may be installed in the exhaust to create a supplementary boiler system. This system can provide additional steam or level the steam production during reduced turbine loads.

In all reciprocating internal combustion engines except small air-cooled units, heat can be reclaimed from the lubricating system, jacket cooling water system, and the exhaust.

Coolant fluids and lubricating oil are generally circulated to remove excess heat conducted into the power train during combustion and heat from friction. Some engines are constructed to convert cooling water to steam within the engine.

The approximate distribution of input fuel energy is as follows:

Useful work	33%
Friction and radiation	7%
Rejected in jacket water	30%
Rejected in exhaust	30%

Of course, neither all of the exhaust heat nor all of the jacket heat can be recovered. Good design could result in the usable portion of the jacket heat and the exhaust heat being about 70% of that shown, or 21% of the input for each.

Depending on engine design, these amounts vary. However, they do indicate that overall cycle thermal efficiency can be greatly improved by waste heat recovery systems if there is a beneficial use for the recovered heat.

Example 16-1 A 200 kW internal combustion engine power unit produces 2570 lb/h (0.32 kg/s) of exhaust gas at 950°F

(510°C). The exhaust gas mixture has a specific heat of 0.252 Btu/lb·°F (1.06 kJ/(kg·K)]. Energy in the exhaust gas is to be used in a waste heat boiler to produce dry saturated steam at 280°F (138°C) from water supplied at 60°F (16°C). The exhaust gas is cooled during the process from 950°F to 400°F (510°C to 204°C). Determine the quantity of steam that can be produced, lb/h.

Solution:

From Table 2-1 (Chapter 2 of this book), Thermodynamic Properties of Water:

$$h_g \text{ [at 280°F (138°C)]} = 1173.94 \text{ Btu/lb (2730.88 kJ/kg)}$$

$$h_f \text{ [at 60°F (16°C)]} = 28.07 \text{ Btu/lb (67.16 kJ/kg)}$$

Equating the heat transfer rate from the exhaust gas to that for the water/steam, yields

$$m_g c_p\,(t_{g,\text{in}} - t_{g,\text{out}}) = m_s(h_g - h_f)$$

$$2570(0.252)(950 - 400) = m_s(1173.4 - 28.07)$$

$$m_s = 311 \text{ lb/h (0.039 kg/s)}$$

16.5 Heat Recovery Terminology and Concepts

The following definitions serve as an introduction to heat recovery systems.

Balanced heat recovery. Occurs when internal heat gain equals recovered heat and no external heat is introduced to the conditioned space. Maintaining balance may require raising the temperature of recovered heat.

Break-even temperature. The outdoor temperature at which total heat losses from conditioned spaces equal internally generated heat gains.

Changeover temperature. The outdoor temperature the designer selects as the point of changeover from cooling to heating by the HVAC system.

External heat. Heat generated from sources outside the conditioned area. This heat from gas, oil, steam, electricity, or solar sources supplements internal heat and internal process heat sources. Recovered internal heat can reduce the demand for external heat.

Internal heat. The total passive heat generated within the conditioned space. It includes heat generated by lighting, computers, business machines, occupants, and mechanical and electrical equipment such as fans, pumps, compressors, and transformers.

Internal process heat. Heat from industrial activities and sources such as wastewater, boiler flue gas, coolants, exhaust air, and some waste materials. This heat is normally wasted unless equipment is included to extract it for further use.

Pinch technology. An energy analysis tool that uses vector analysis to evaluate all heating and cooling utilities in a process. Composite curves created by adding the vectors allow identification of a "pinch" point, which is the best thermal location for a heat pump.

Recovered (or reclaimed) heat. Comes from internal heat sources. It is used for space heating, domestic or service water heating, air reheat in air conditioning, process heating in industrial applications, or other similar purposes. Recovered heat may be stored for later use.

Stored heat. Heat from external or recovered heat sources that is held in reserve for later use.

Usable temperature. The temperature or range of temperatures at which heat energy can be absorbed, rejected, or stored for use within the system.

Waste heat. Heat rejected from the building (or process) because its temperature is too low for economical recovery or direct use or storage capacity is not available.

16.5.1 Definition of Balanced Heat Recovery Systems

In an ideal heat recovery system, all components work year-round to recover all of the internal heat before adding external heat. Any excess heat is either stored or rejected. Such an idealized goal is identified as a balanced heat recovery system.

When the outdoor temperature drops significantly, or when the building is shut down (e.g., on nights and weekends), internal heat gain may be insufficient to meet the space heating requirements. Then, a balanced system provides heat from storage or an external source. When internal heat is again generated, the external heat is automatically reduced to maintain proper temperature in the space. There is a time delay before equilibrium is reached. The size of the equipment and the external heat source can be reduced in a balanced system that includes storage. Regardless of the system, a heat balance analysis establishes the merits of balanced heat recovery at various outdoor temperatures.

Outdoor air less than 55°F to 65°F (13 to 18°C) may be used to cool building spaces with an air economizer cycle. When considering this method of cooling, the space required by ducts, air shafts, and fans, as well as the increased filtering requirements to remove contaminants and the hazard of possible freeze-up of dampers and coils must be weighted against alternatives such as using deep row coils with antifreeze fluids and efficient heat exchange. Innovative use of heat pump principles may give considerable energy savings and more satisfactory human comfort than an air economizer. In any case, hot and cold air should not be mixed (if avoidable) to control zone temperatures because it wastes energy.

Many buildings, especially those with computers or large interior areas, generate more heat than can be used for most of the year. Operating cost is minimized when the system changes over from net heating to net cooling at the break-even outdoor temperature at which the building heat loss equals the internal heat load. If heat is unnecessarily rejected or added to the space, the changeover temperature varies from the natural break-even temperature, and operating costs increase. Heating costs can be reduced or eliminated if excess heat is stored for later distribution. The concept of ideal heat balance in an overall building project or a single space requires that one of the following takes place on demand:

- Heat must be removed.
- Heat must be added.
- Heat recovered must exactly balance the heat required, in which case heat should be neither added nor removed.

In small air-conditioning projects serving only one space, either cooling or heating satisfies the thermostat demand. If humidity control is not required, operation is simple. Assuming both heating and cooling are available, automatic controls will respond to the thermostat to supply either. A system should not heat and cool the same space simultaneously.

Multiroom buildings commonly require heating in some rooms and cooling in others. Optimum design considers the building as a whole and transfers excess internal heat from one area to another, as required, without introducing external heat that would require waste heat disposal at the same time. The heat balance concept is violated when this occurs.

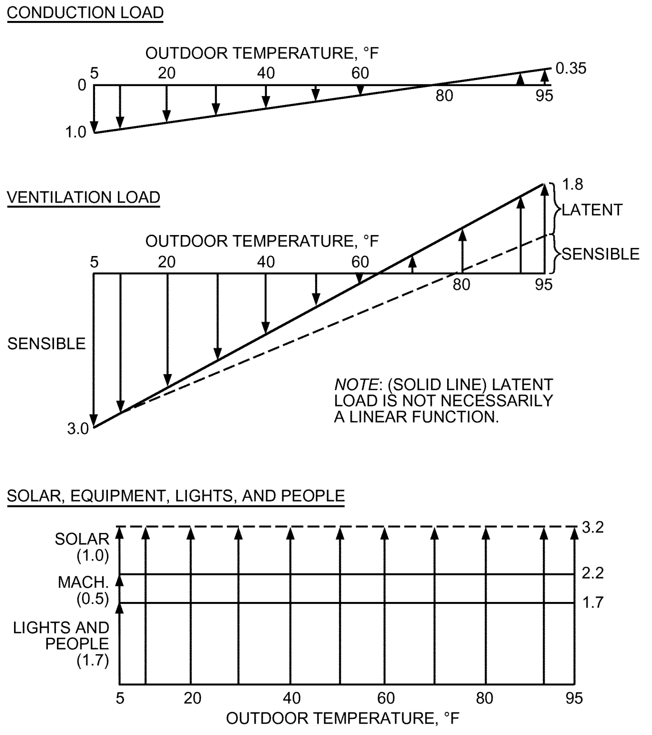
Humidity control must also be considered. Any system should add or remove only enough heat to maintain the desired temperature and control the humidity. Large percentages of outdoor air with high wet-bulb temperatures, as well as certain types of humidity control, may require reheat, which could upset the desirable balance. Usually, humidity control can be obtained without upsetting the balance. When reheat is unavoidable, internally transferred heat from heat recovery should always be used to the extent it is available before using an external heat source such as a boiler. However, the effect of the added reheat must be analyzed because it affects the heat balance and may have to be treated as a variable internal load.

When a building requires heat and the refrigeration plant is not in use, dehumidification is not usually required and the outdoor air is dry enough to compensate for any internal moisture gains. This should be carefully reviewed for each design.

Heat Balance Studies. The following examples illustrate situations that can occur in nonrecovery and unbalanced heat recovery situation. Figure 16-2 shows the major components of a building that comprise the total air-conditioning load. Values above the zero line are cooling loads, and values below the zero line are heating loads. On an individual basis, the ventilation and conduction loads cross the zero line, which indicated that these loads can be a heating or a cooling load, depending on outdoor temperature. Solar and internal loads are always a cooling load and are, therefore, above the zero line.

Figure 16-3 combines all of the loads shown in Figure 16-2. The graph is obtained by plotting the conduction load of a building at various outdoor temperatures and then adding or subtracting the other loads at each temperature. The project load lines, with and without solar effect, cross the zero line at 16°F (–9°C) and 30°F (–1°C), respectively. These are the outdoor temperatures for the plotted conditions when the naturally created internal load exactly balances the loss.

As plotted, this heat balance diagram includes only the building loads with no allowance for additional external heat from a boiler or other source. If external heat is necessary because of system design, the diagram should include the additional heat.



NOTE: ALL LOADS ARE IN 10^6 Btu/h

Fig. 16-2 Major Load Components
(Figure 34, Chapter 9, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Figure 16-4 illustrates what happens when heat recovery is not used. It is assumed that with a temperature of 70°F (21°C), heat from an external source is added to balance conduction through the building's skin in increasing amounts down to the minimum outdoor temperature winter design condition. Figure 16-4 also adds the heat required for the outdoor air intake. The outdoor air, comprising part or all of the supply air, must be heated from outdoor temperature to room temperature. Only the temperature range above the room temperature is effective for heating to balance the perimeter conduction loss.

These loads are plotted at the minimum outdoor winter design temperature, resulting in a new line passing through points A, D, and E. This line crosses the zero line at –35°F (–37°C), which becomes the artificially created break-even temperature rather than 30°F (–1°C), when not allowing for solar effect. When the sun shines, the added solar heat at the minimum design temperature would further drop the –35°F (–37°C) break-even temperature. Such a design adds more heat than the overall project requires and does not use balanced heat recovery to use the available internal heat. This problem is most evident during mild weather on systems not designed to take full advantage of internally generated heat year-round.

The following are two examples of situations that can be shown in a heat balance study:

1. As the outdoor air wet-bulb temperature drops, the total heat of the air falls. If a mixture of outdoor and recirculated

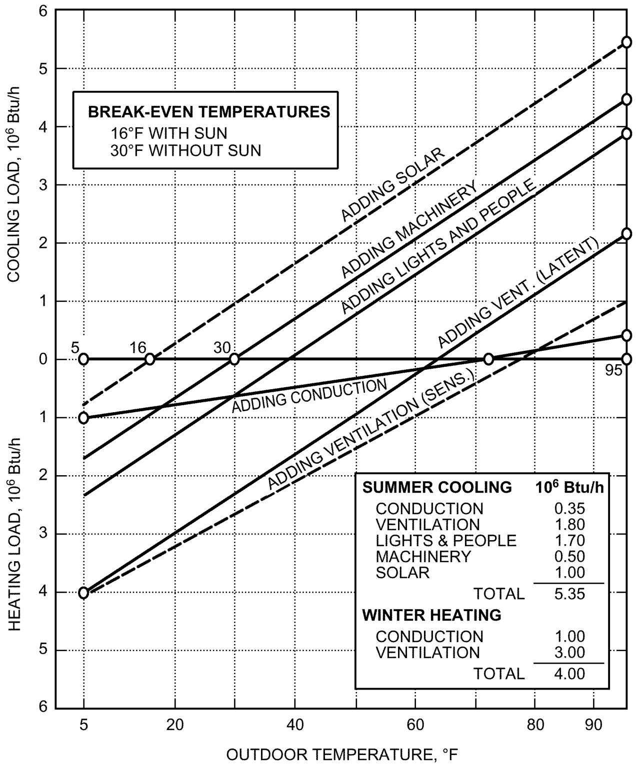


Fig. 16-3 Composite Plot of Loads in Fig. 16-2
(Figure 35, Chapter 9, 2016 ASHRAE Handbook— HVAC Systems and Equipment)

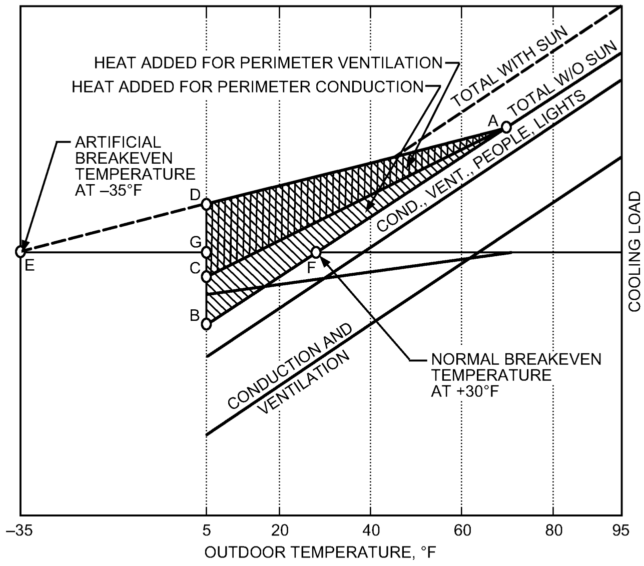


Fig. 16-4 Non-Heat Recovery System
(Figure 36, Chapter 9, 2016 ASHRAE Handbook— HVAC Systems and Equipment)

air is cooled to 55°F (13°C) in summer and the same dry-bulb temperature is supplied by an economizer cycle for interior space cooling in winter, there will be an entirely different result. As the outdoor wet-bulb temperature drops below 55°F (13°C), each unit volume of air introduced does more cooling. To make matters more difficult, this increased cooling is latent cooling, which requires adding latent heat to prevent too low a relative humidity, yet this air is intended to cool. The extent of this added external heat for free cooling is shown to be very large when plotted on a heat balance analysis at 0°F (–18°C) outside temperature.

Figure 16-4 is typical for many current non-heat-recovery systems. There may be a need for cooling, even at the minimum design temperature, but the need to add external heat for humidification can be eliminated by using available internal heat. When this asset is thrown away and external heat is added, operation is less efficient.

Some systems recover heat from exhaust air to heat the incoming air. When a system operates below its natural break-even temperature t_{be}, such as 30°F (–1°C) or 16°F (–9°C) (shown in Figure 16-3), the heat recovered from exhaust air is useful and beneficial. This assumes that only the available internal heat is used and that no supplementary heat is added at or above t_{be}. Above t_{be}, the internal heat is sufficient and any recovered heat would become excessive heat to be removed by more outdoor air or refrigeration.

If heat is added to a central system to create an artificial t_{be} of –35°F (–37°C) as in Figure 16-4, any recovered heat above –35°F (–37°C) requires an equivalent amount of heat removal elsewhere. If the project were in an area with a minimum design temperature of 0°F (–18°C), heat recovery from exhaust air could be a liability at all times for the conditions stipulated in Figure 16-3. This does not mean that the value of heat recovered from exhaust air should be forgotten. The emphasis should be on recovering heat from exhaust air rather than on adding external heat.

2. A heat balance shows that insulation, double glazing, and so forth can be extremely valuable in some situations. However, these practices may be undesirable in certain regions during the heating season, when excess heat must usually be removed from large buildings. For instance, for minimum winter design temperatures of approximately 35°F to 40°F (1.7°C to 4.5°C), it is improbable that the interior core of a large office building will ever reach its break-even temperature. The temperature lag for shutdown periods, such as nights and weekends, at minimum design conditions could never economically justify the added cost of double-pane windows. Therefore, double-pane windows merely require the amount of heat saved to be removed elsewhere. However, in cold climates the double-pane windows may be necessary to provide comfort.

16.6 Heat Recovery Systems

Figures 16-5 through 16-9 show several possible heat recovery/simultaneous heating-cooling systems. Figure 16-5 illustrates one method of using water as the heat source or sink and as the heating and cooling medium. The compressor, evaporator, condenser, refrigerant piping, and accessories are

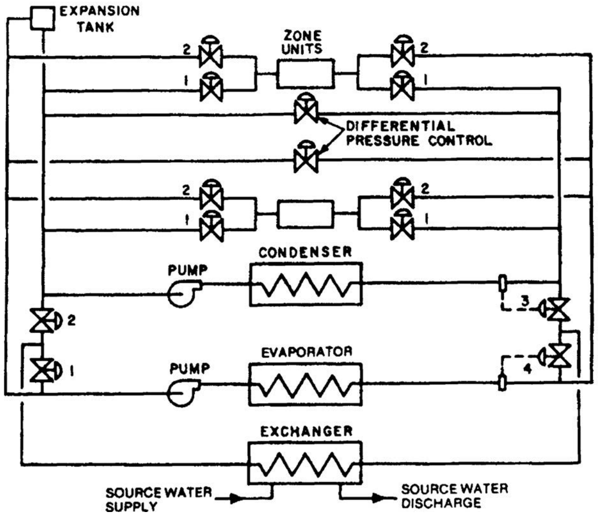


Fig. 16-5 Water-to-Water Heat Pump Cycle

essentially standard and are available as a factory-packaged water-to-water heat pumps (see Chapters 14 and 15).

The cycle is flexible, and the heating or cooling medium is instantly available at all times. Heating can be provided exclusively to the zone conditioners by closing valves 2 and 3 and opening valves 1 and 4. With the valves in these positions, the water is divided into two separate circuits. The warm water circuit consists of the condenser (where the heat is supplied by the high-temperature refrigerant), valve 1, zone conditioners, and a circulating pump. The cold water circuit consists of the evaporator (where heat is taken from the water by the low-temperature refrigerant), valve 4, a heat exchanger (where heat is taken from the source water), valve 1, and a circulating pump. The refrigerating compressor operates to maintain the desired leaving water temperature from the condenser.

Similarly, cooling can be exclusively obtained in the cycle of Figure 16-5 by opening valves 2 and 3 and closing valves 1 and 4. With this arrangement, the cold water circuit consists of the evaporator (where heat is removed from the water by the low-temperature refrigerant), valve 2, zone conditioners, and a circulating pump. The warm water circuit consists of the condenser (which receives the heat from the refrigerant), valve 3, a heat exchanger (where heat is rejected to the source water), valve 2, and a circulating pump. The refrigerating compressor operates to maintain the desired water temperature leaving the evaporator.

During the intermediate season, simultaneous heating and cooling can be provided by the cycle shown in Figure 16-5. Valves 3 and 4 are modulated when valves 1 and 2 are open. Valve 3 is adjusted to maintain 85 to 140°F (29 to 60°C) water in the condenser circuit and valve 4 to maintain 45 to 50°F (7 to 10°C) in the evaporator circuit. The excess heating or cooling effect is discharged to the exchanger, which passes it on to the source water.

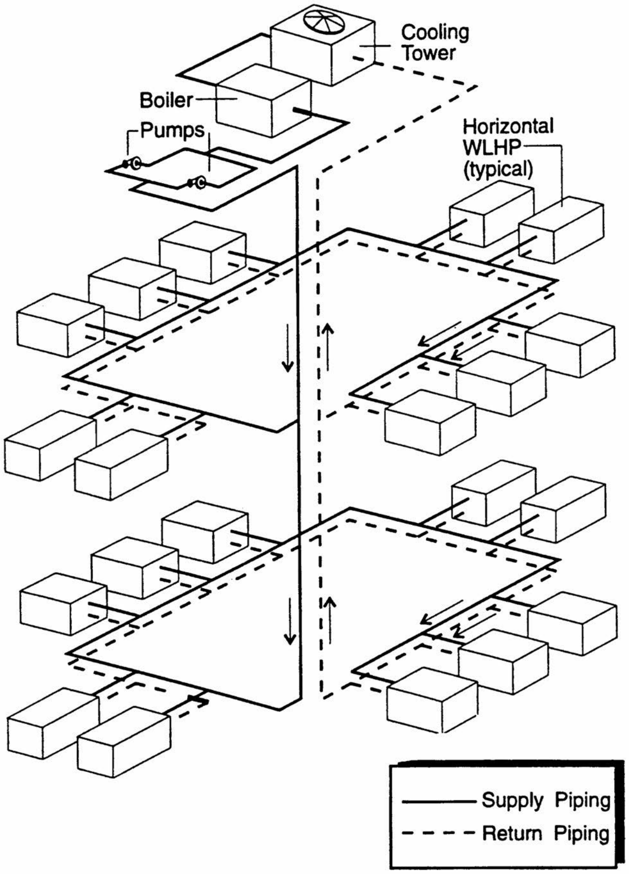


Fig. 16-6 Heat Recovery System Using Water-to-Air Heat Pump in Closed Loop
(Figure 30, Chapter 9, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

The source or sink water, if of suitable quality, can be supplied directly to the condenser and evaporator instead of using an exchanger (Figure 16-5). This eliminates one heat transfer surface and its performance penalty.

A water loop heat pump (WLHP) cycle that combines load transfer characteristics with water-to-air heat pump units is illustrated in Figure 16-6. Each module or space has one or more water-to-air heat pumps. The units in both the building core and perimeter areas are connected hydronically with a common two-pipe system. Each unit cools conventionally, supplying air to the individual module and rejecting the heat removed to the two-pipe system through its integral condenser. The total heat gathered by the two-pipe system is expelled through a common heat rejection device. This device often includes a closed circuit evaporative cooler with an integral spray pump. If and when some of the modules, particularly on the northern side, require heat, the individual units switch (by means of four-way refrigerant valves) into the heating cycle. The units derive their heat source from the two-pipe water loop, basically obtaining heat from a relatively high temperature source, that is, the condenser water of the other units. When only heating is required, all units are in the heating cycle and, consequently, an external heat input source is needed to provide heating capability. The heat of compression contrib-

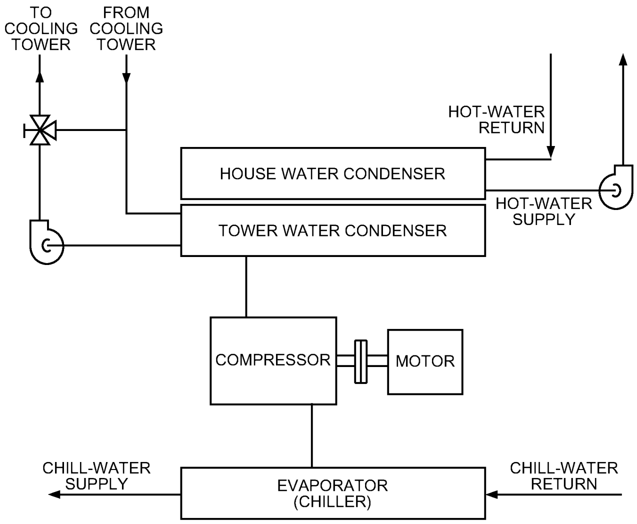


Fig. 16-7 Heat Transfer Heat Pump with Double-Bundle Condenser
(Figure 23, Chapter 9, 2012 ASHRAE Handbook—HVAC Systems and Equipment)

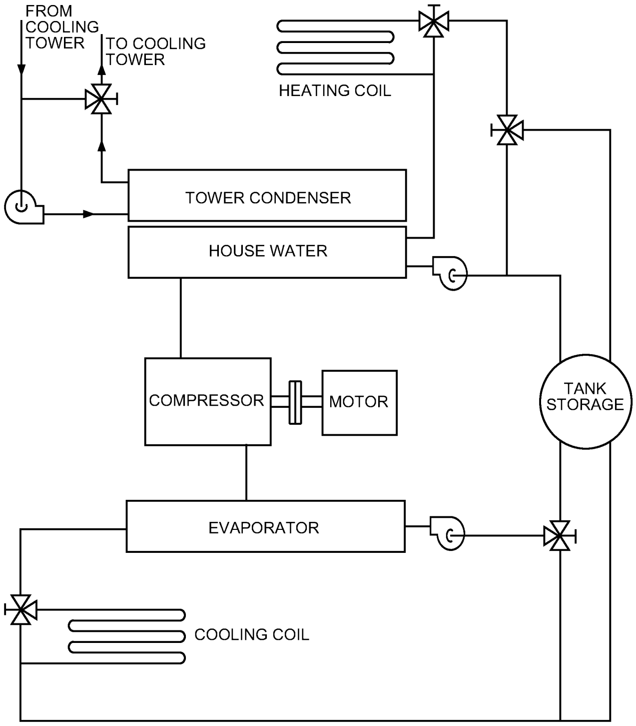


Fig. 16-8 Heat Transfer System with Storage Tank
(Figure 24, Chapter 9, 2012 ASHRAE Handbook—HVAC Systems and Equipment)

utes to the heat source. The water loop is usually 60 to 90°F (15 to 32°C) and, therefore, seldom requires piping insulation.

A water-to-water heat pump can be added in the closed water loop before the heat rejection device for further heat reclaim. This heat pump reuses the heat and can provide domestic hot water or elevate water temperatures in a storage tank to be bled back into the loop.

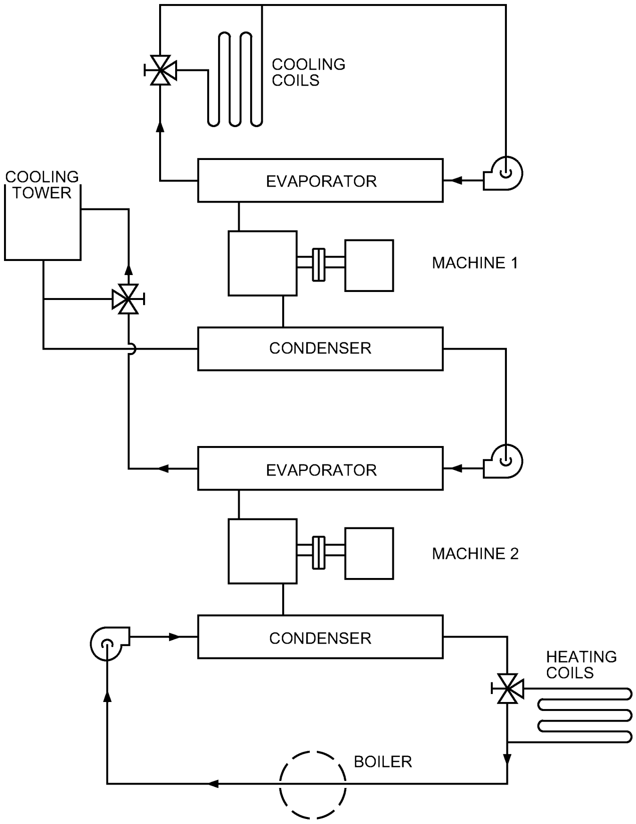


Fig. 16-9 Multistage (Cascade) Heat Transfer System
(Figure 25, Chapter 9, 2012 ASHRAE Handbook—HVAC Systems and Equipment)

In many large buildings, internal heat gains require year-round chiller operation. This internal heat is often discharged through a cooling tower. Prudent design may dictate cascade systems with chillers in parallel or series. Manufacturers can assist with custom components to meet a wide range of load and temperature requirements. The double-bundle condenser working with a reciprocating or centrifugal compressor is most often used in this application. Figure 16-7 shows the basic configuration of this system, which makes heat available in the range of 100 to 130°F (38 to 54°C). The warm water is supplied as a secondary function of the heat pump and represents recovered heat.

Figure 16-8 shows a similar cycle, except that a storage tank has been added, enabling the system to store heat during occupied hours by raising the temperature of the water in the tank. During unoccupied hours, water from the tank is gradually fed to the evaporator providing load for the compressor and condenser that heats the building during off hours.

Figure 16-9 is another transfer system capable of generating 130 to 140°F (54 to 60°C) or warmer water whenever there is a cooling load by cascading two compressors hydronically. In this configuration, one chiller can be considered as a chiller only and the second unit as a heating-only heat pump.

16.7 Problems

16.1 A heat pump is used in place of a furnace for heating a house. In winter, when the outdoor air temperature is 15°F, the heat loss from the house is 100,000 Btu/h if the inside is maintained at 70°F. (a) Determine the minimum electric power (Carnot) required to operate the heat pump. (b) Determine the actual electric power to operate the heat pump with a heating COP of 3.

16.2 An air-source heat pump is to be used for both air conditioning and heating of a residence, maintaining the interior at 80°F in summer with an outdoor air temperature of 95°F and a cooling load of 36,000 Btu/h. As a heat pump, it is to maintain 70°F in winter with an outdoor air temperature of 2°F and a heating load of 52,000 Btu/h.

Select a heat pump from the table in Problem 8.13, sized for cooling. What size resistance heater is required at the winter design condition?

16.3 A 100,000 ft^2 building design has a design electrical load of 5 W/ft^2. A reciprocating natural gas engine cogeneration plant is to serve the building. The engine-generator is sized for the electrical load, with salvaged heat being used for heating and for driving a single-effect absorption chiller. The design heating load is 3,000,000 Btu/h. The design cooling load is 250 tons; the absorber requires 20,000 Btu/ton·h input.

Calculate hourly design operating costs for heating and cooling. Any shortfall in heating from recovered heat must be made up by a boiler. Any shortfall in cooling by the absorber with recovered heat must be made up by the boiler as input to the absorber.

Compare design operating costs with hourly design operating costs using conventional equipment (purchased electricity for the building and for cooling with an electric chiller at 1.0 kW/ton, purchased gas for a boiler for heating). Use $1.00 per therm, boiler efficiency of 80% for fuel cost, $0.10/kWh for purchased electricity cost.

16.8 Bibliography

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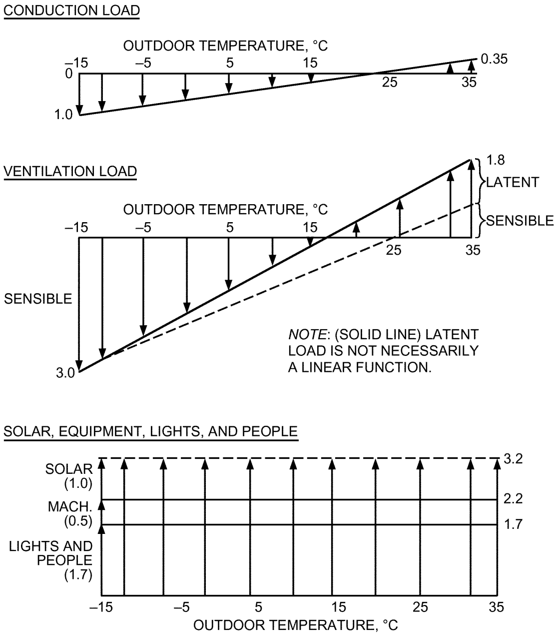
SI Figures



NOTE: ALL LOADS ARE IN MEGAWATTS

Fig. 16-2 SI Major Load Components
(Figure 31, Chapter 12, 2012 ASHRAE Handbook—HVAC Systems and Equipment)

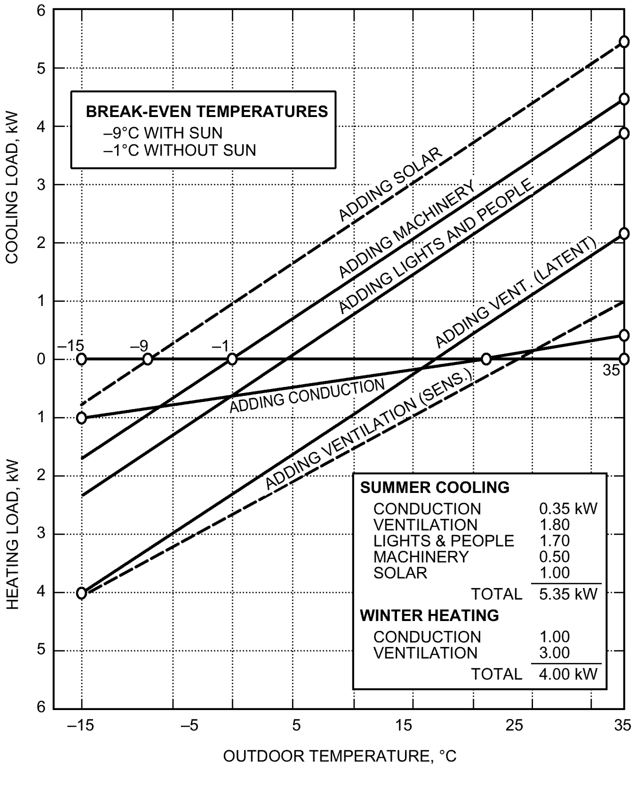


Fig. 16-3 SI Composite Plot of Loads in Fig. 16-2
(Figure 32, Chapter 9, 2012 ASHRAE Handbook—HVAC Systems and Equipment)

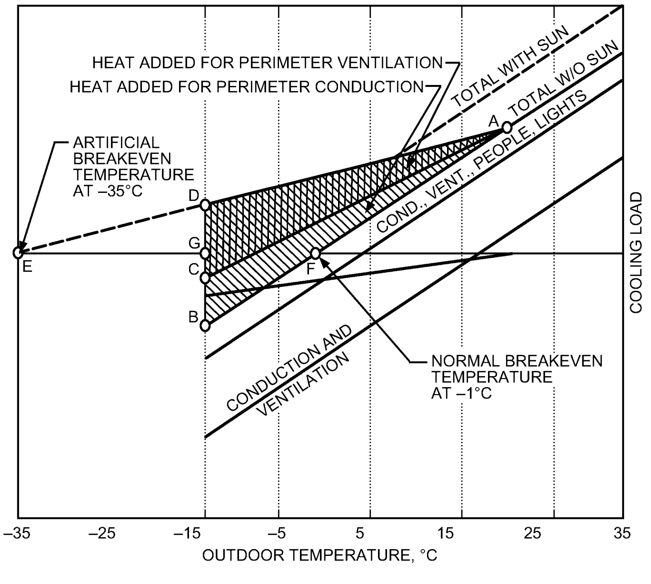


Fig. 16-3 SI Nonheat Recovery-System
(Figure 33, Chapter 9, 2012 ASHRAE Handbook—HVAC Systems and Equipment)

Chapter 17

AIR-PROCESSING EQUIPMENT

Cooling, heating, humidifying, dehumidifying, and cleaning of air are some of the processes for which mechanical equipment is needed in HVAC systems. These kinds of air-processing devices are examined in this chapter, along with air-to-air energy recovery equipment. For additional information on these topics, the reader is referred to Chapters 21 through 29, and 41 of the *2016 ASHRAE Handbook—HVAC Systems and Equipment*. Chapters 3 and 4 of the *2017 ASHRAE Handbook—Fundamentals* provide the heat transfer background for much of this equipment.

17.1 Air-Handling Equipment

Air-handling units consist of the equipment that filter, heat, humidify, cool, dehumidify, and move the air. The three major types of air handlers are

- Factory-fabricated units
- Built-up (field-erected) units
- Customized units

Each category contains both single-zone and multizone units.

Factory-fabricated units usually contain the fan and cooling and/or heating coils and filter assembled in a cabinet, usually of sheet metal. Mixing boxes are also included as desired. Standard arrangements and options available with factory-assembled units include

- Draw-through or blow-through
- Horizontal or vertical assembly
- Chilled water or refrigerant cooling coils
- Hot water, steam, or electric heating coils
- Forward-curved, backward-curved, airfoil, axial, or plug fans
- Flat or “V” bank filter sections
- Mixing boxes with damper assemblies

Factory-fabricated units are usually less expensive than field-erected built-up systems and the delivery time is usually shorter than for the purchase and assembly of a built-up unit.

Rooftop units may be considered a special type of factory-fabricated unit, being similar to such indoor units, except that they are located on the roof, saving valuable space that would otherwise be needed for the mechanical equipment room. Packaged rooftop units are sometimes completely self-contained including the condensing unit and usually either a gas-fired or electric heating section. On the negative side, rooftop units generally require higher maintenance and have a shorter service life than units protected in mechanical equipment rooms.

Built-up air-handling units consist of separate casings enclosing fans, coils, filters, mixing boxes, and plenums. Built-up unit casings may be factory made or fabricated by a local sheet metal contractor. Dimensions of built-up units can often be varied for the floor area available, being limited only by the coil and fan selections. There is almost no limit to the capacity of built-up units.

Customized air-handling units are normally selected to provide either a higher level of quality or special component configurations for a special application.

17.2 Cooling Coils

Ceiling coils are used to cool an airstream under forced convection. Such equipment may consist of a single coil section or several individual coil sections built into banks. Coils are also used extensively as components in central station air-handling units, room terminals, and in factory-assembled, self-contained air conditioners.

Chilled water, brines, or volatile refrigerants are the usual cooling media. Cooling coils that use relatively high temperature water usually do not dehumidify the air; however, most coil equipment is designed to remove sensible heat and dehumidify simultaneously. The coil assembly should include a means to protect the coil from dirt accumulation and to keep dust and foreign matter out of the conditioned space.

17.2.1 Coil Construction and Arrangement

Coils include bare tubes or pipe, and those with extended or finned surfaces. The design of coils with extended surfaces on the air side considers the materials, fin size and spacing, ratio of extended surface area to that of the tube area, tube nesting center dimensions, staggered or in-line tube arrangement, and use of turbulators.

Staggered tubes increase the total heat transfer over that of the in-line arrangement, and turbulators may also be used to enhance total heat transfer efficiency. The surface arrangement has a great effect on the air-film heat transfer resistance and associated air-side pressure drop. Several arrangements are illustrated in Figure 17-1.

In fin or extended surface coils, the external surface of the tubes is **primary**, and the fin surface is **secondary**. The primary surface consists of rows of round tubes or pipes, which may be staggered or placed in line with respect to the airflow. Some tubes are flattened or have nonround internal passageways. The inside surface of the tubes is usually smooth and plain, but some designs have internal fins or

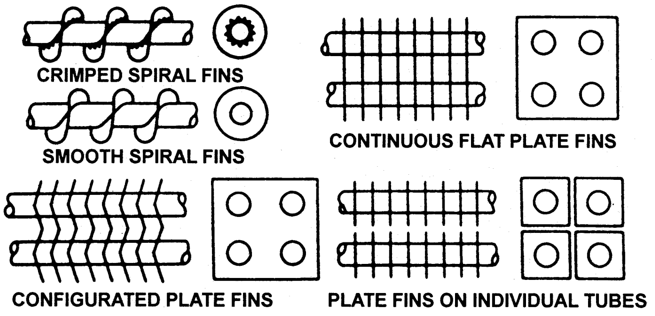


Fig. 17-1 Types of Fin-Coil Arrangements

turbulence promoters (either fabricated or extruded) to provide turbulence and additional inside surface area for enhancing performance. Individual tubes are generally interconnected by return bends, sometimes together with hairpin tubes to form the required serpentine arrangement of multipass tube circuits. Some flattened tubes are folded into continuous serpentine circuits with fins metallurgically bonded between adjacent tube passes. Numerous fin arrangements are used; the most common are smooth spiral, crimped spiral, flat plate, and configurated plate. A good thermal bond between the tube and the fin must be maintained permanently to ensure low resistance to heat transfer from fin to tube.

In some coils, fins are wound under pressure onto the tubes in order to upset the metal slightly at the fin root. They are then coated with solder while the fin and tube are still revolving to ensure a uniform solder coating. In other types of coils, the spiral fin may be knurled into a shallow groove on the exterior of the tube. The tube may be expanded after the fins are assembled, or the tube-hole flanges of a flat or configurated fin may be made to override those in the preceding fin and so compress them on the tube. Some construction techniques even form the fin from the material of the tube itself.

Cooling coils for water or for volatile refrigerants frequently have aluminum fins and copper tubes, although copper fins on copper tubes are also used and the combination of aluminum fins on aluminum tubes is finding use. Adhesive bonding is sometimes used in making header connections, return bends, and fin-tube joints, particularly for aluminum-to-aluminum joints. Many types of lightweight, extended-surface cooling coils are made for both heating and cooling. Tube outside diameters are commonly 1/4, 3/16, 3/8, 5/8, 3/4, and 1 in. (6.4, 9.5 12.7, 15.9, 19.1, and 25 mm), and fins are spaced from 3 to 14 per inch (1.8 to 8.5 mm apart). The tube spacing varies from about 5/8 to 2 1/2 in. (16 to 64 mm) on centers, depending on the width of individual fins and other performance considerations. Fin spacing should be chosen for the duty to be performed, with special attention paid to air friction, prevention of water carryover, possible lint accumulation, and, especially at lower temperatures, frost accumulation.

17.2.2 Water Coils

Water coil performance depends on eliminating air from the water circuit and properly distributing the water. Unless the system is vented, air may accumulate in the coil circuits, which reduces thermal performance and may cause noise or vibration. Air vent connections are usually provided on the coil water headers. Depending on performance requirements, the water velocity inside tubes usually ranges from approximately 1 to 8 ft/s (0.3 to 2.4 m/s), and the design water pressure drop across coils varies from about 5 to 50 ft of water (15 to 150 kPa).

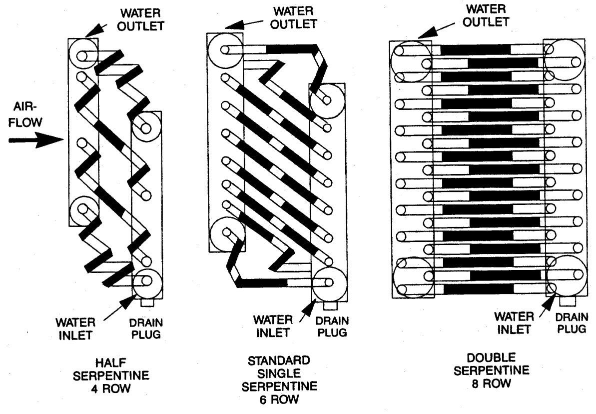


Fig. 17-2 Typical Water Circuit Arrangements

A variety of circuit arrangements, and combinations thereof, for varying the number of parallel water flow passes within the tube core are usually available. Some typical arrangements are shown in Figure 17-2.

17.2.3 Direct-Expansion Refrigerant Coils

Direct-expansion coils present more complex problems of cooling fluid distribution than water or brine coils. The coil should be effectively and uniformly cooled throughout, and the compressor must be protected from entrained, unevaporated refrigerant. Direct-expansion coils are used on both flooded and dry-expansion refrigeration systems.

The **flooded system** is mainly used for low-temperature applications where a small temperature difference between the air and refrigerant is advantageous. However, a relatively large volume of refrigerant is required, together with extra components such as a surge tank and interconnecting piping. Other applications of direct-expansion coils generally use dry expansion. For **dry-expansion systems**, the most commonly used refrigerant liquid metering devices are the capillary tube and the thermostatic expansion valve.

The **capillary tube system** is applied on evaporator coils in factory-assembled, self-contained air conditioners up to approximately 10 tons (35 kW) capacity and is used extensively on the smaller capacity models such as window or room units. In this system, the bore and length of a capillary tube are sized so that at full load, just enough refrigerant liquid is metered from the condenser to the evaporator coil to be completely evaporated.

The **thermostatic expansion valve system** is common for all dry-expansion coil applications, particularly for field-assembled coil sections in central air-handling units, as well as for the larger factory-assembled hermetic air conditioners.

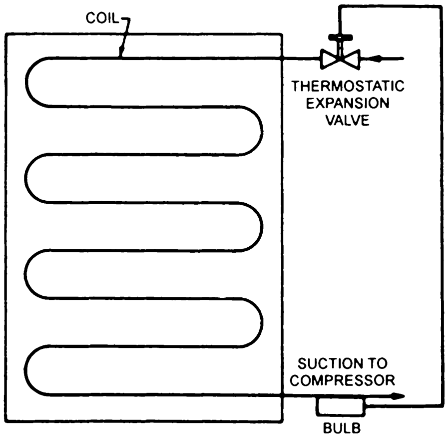


Fig. 17-3 Dry Expansion Coil with Thermostatic Expansion Valve

A schematic typical of a coil and thermostatic expansion valve assembly is shown in Figure 17-3. The thermostatic expansion valve automatically regulates the rate of refrigerant liquid flow to the coil in direct proportion to the rate of evaporation of refrigerant liquid in the coil, thereby maintaining the superheat at the coil suction outlet within the usual predetermined limits of 6 to 10°F (3 to 6°C).

17.2.4 Coil Selection

When selecting a coil, the following factors must be considered:

- Job requirements: cooling, dehumidifying, and the capacity required to maintain balance with other system components, for example, compressor equipment in the case of direct-expansion coils.
- Temperature of entering air: dry bulb only if there is no dehumidification; dry and wet bulb if moisture is to be removed.
- Temperature of entering, chilled water or evaporating pressure of refrigerant and type of refrigerant.
- Temperature of leaving air dry bulb and wet bulb or dewpoint.
- Available cooling media and its operating temperature and quantity.
- Space and dimensional limitations.
- Air quantity limitations.
- Allowable frictional resistances in cooling media piping system (including coils).
- Allowable frictional resistances in air circuit (including coils).
- Characteristics of individual coil designs.
- Individual installation requirements, such as the type of automatic control to be used or the presence of a corrosive atmosphere.
- Coil air face velocity.
- To prevent water carryover from dehumidifying coils, face velocities should not exceed 500 fpm (2.54 m/s), and fin density should be no greater than 8 fins per inch (fins 3 mm apart).

Coil ratings and selection procedures are usually presented in one of two ways:

Basic Data Method. Coil performance parameters are published in the form of tables or charts from which the coil row depth is calculated after determining the required coil sensible and total heat capacities and other design variables from the job conditions. The initial selection generally indicates a nonintegral row depth requirement. It is frequently necessary to recheck and reselect the coil to match more closely the required air and cooling fluid conditions with the integral row depth actually installed, particularly for dehumidifying coils. The method is generally used in selecting coils or coil banks for field assembly, since there is a vast number of size and row depth combinations available.

Unit Rating Method. Performance for specific combinations of coil face area and row depth are presented in tables or charts. This method provides a direct selection of specific coils to match the required capacity under the job conditions. This method is frequently used in selecting coils for central station air-handling units and also in determining performance for factory-assembled, self-contained air conditioners.

With either method, various combinations of coil face area, row depth, air velocity, and air quantity may be chosen to do the same job. Coil selection requires understanding each case, and selection should be based on an economic analysis of the plant as a whole.

Most coil manufacturers and some commercial software providers have programs to assist in the selection and diagnosis of cooling coils.

Table 17-1 provides an illustration of coil performance data as may be found in a typical cooling coil catalog.

Coils that operate wet, particularly those with enhanced (configurated) fins tend to build up dirt on the fin and tube surfaces so they must be capable of being, first, visually inspected for fouling and, second, cleaned. Both of these are difficult without sufficient space on each side of the coil. Fin spacing closer than 1/8 of an inch or enhanced fins are not recommended.

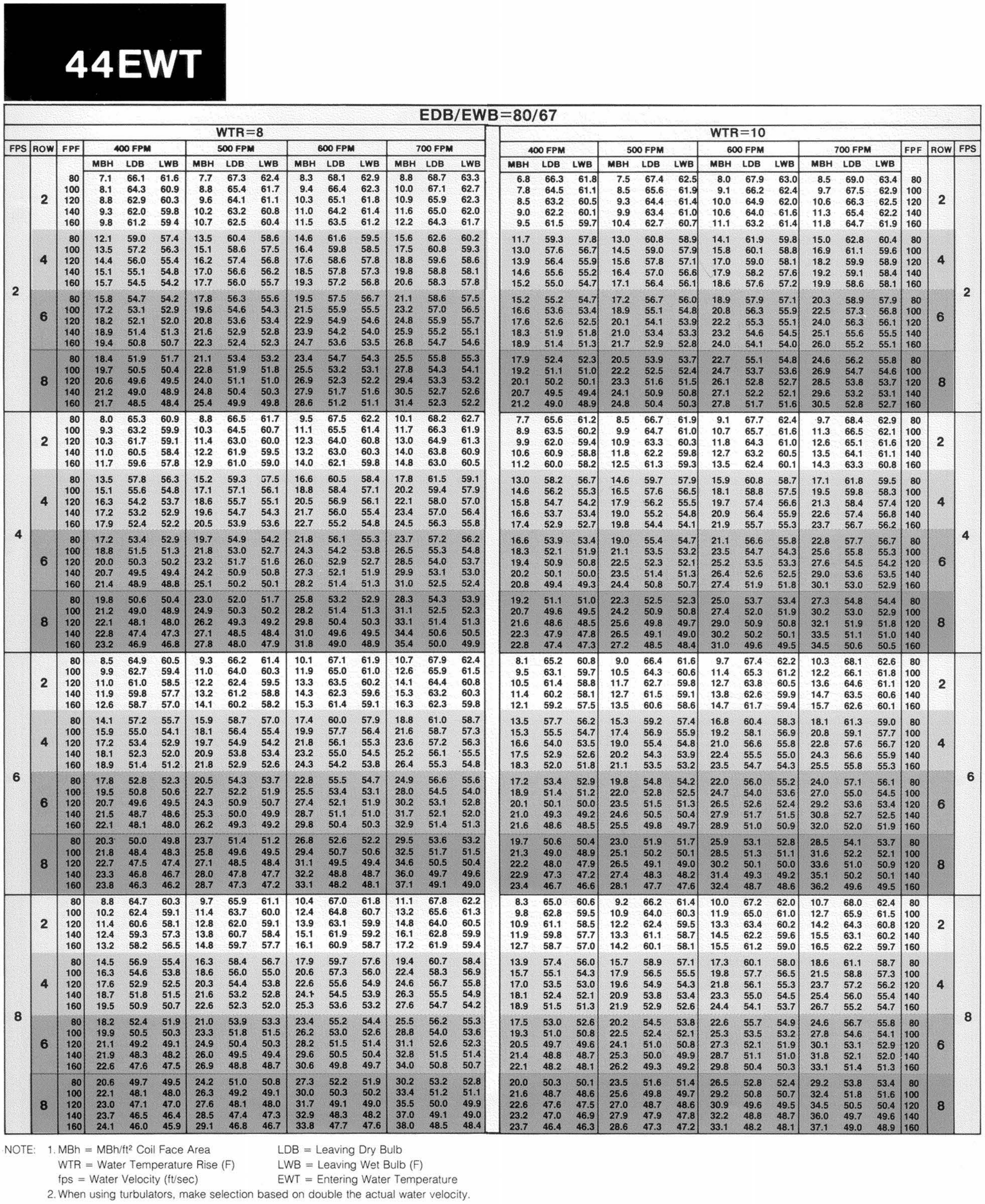
Improper selection of the cooling coil is the most common cause of performance failures in air-conditioning systems.

17.2.5 Application Range

Based on information in AHRI Standard 410, dry surface (sensible cooling) coils and dehumidifying coils (which both cool and dehumidify), particularly for field-assembled coil banks, factory-assembled coil banks, or factory-assembled central station air conditioners using different combinations of coils, are usually rated within the following limits:

Entering air temperature	65 to 100°F (18 to 38°C)
Entering air wet bulb	60 to 85°F (15 to 30°C)
Air face velocity	300 to 800 fpm (1.5 to 4.0 m/s) (sometimes as low as 200 and as high as 1500 fpm)
Refrigerant saturated	30 to 55°F (–1 to 13°C) at temperature coil suction outlet [refrigerant vapor superheat at coil suction outlet is 6°F (3.3°C) or higher

Table 17-1 Example of Cooling Coil Catalog Data



44EWT

EDB/EWB=80/67

			WTR=8												WTR=10														
FPS	ROW	FPF	400 FPM			500 FPM			600 FPM			700 FPM			400 FPM			500 FPM			600 FPM			700 FPM			FPF	ROW	FPS
			MBH	LDB	LWB	MBH	LDB	LWB	MBH	LDB	LWB	MBH	LDB	LWB	MBH	LDB	LWB	MBH	LDB	LWB	MBH	LDB	LWB	MBH	LDB	LWB			
2	2	80	7.1	66.1	61.6	7.7	67.3	62.4	8.3	68.1	62.9	8.8	68.7	63.3	6.8	66.3	61.8	7.5	67.4	62.5	8.0	67.9	63.0	8.5	69.0	63.4	80	2	2
		100	8.1	64.3	60.9	8.8	65.4	61.7	9.4	66.4	62.3	10.0	67.1	62.7	7.8	64.5	61.1	8.5	65.6	61.9	9.1	66.2	62.4	9.7	67.5	62.9	100		
		120	8.8	62.9	60.3	9.6	64.1	61.1	10.3	65.1	61.8	10.9	65.9	62.3	8.5	63.2	60.5	9.3	64.4	61.4	10.0	64.9	62.0	10.6	66.3	62.5	120		
		140	9.3	62.0	59.8	10.2	63.2	60.8	11.0	64.2	61.4	11.6	65.0	62.0	9.0	62.2	60.1	9.9	63.4	61.0	10.6	64.0	61.6	11.3	65.4	62.2	140		
		160	9.8	61.2	59.4	10.7	62.5	60.4	11.5	63.5	61.2	12.2	64.3	61.7	9.5	61.5	59.7	10.4	62.7	60.7	11.1	63.2	61.4	11.8	64.7	61.9	160		
	4	80	12.1	59.0	57.4	13.5	60.4	58.6	14.6	61.6	59.5	15.6	62.6	60.2	11.7	59.3	57.8	13.0	60.8	58.9	14.1	61.9	59.8	15.0	62.8	60.4	80	4	
		100	13.5	57.2	56.3	15.1	58.6	57.5	16.4	59.8	58.5	17.5	60.8	59.3	13.0	57.6	56.7	14.5	59.0	57.9	15.8	60.1	58.8	16.9	61.1	59.6	100		
		120	14.4	56.0	55.4	16.2	57.4	56.8	17.6	58.6	57.8	18.8	59.6	58.6	13.9	56.4	55.9	15.6	57.8	57.1	17.0	59.0	58.1	18.2	59.9	58.9	120		
		140	15.1	55.1	54.8	17.0	56.6	56.2	18.5	57.8	57.3	19.8	58.8	58.1	14.6	55.6	55.2	16.4	57.0	56.6	17.9	58.2	57.6	19.2	59.1	58.4	140		
		160	15.7	54.5	54.2	17.7	56.0	55.7	19.3	57.2	56.8	20.6	58.3	57.8	15.2	55.0	54.7	17.1	56.4	56.1	18.6	57.6	57.2	19.9	58.6	58.1	160		
	6	80	15.8	54.7	54.2	17.8	56.3	55.6	19.5	57.5	56.7	21.1	58.6	57.5	15.2	55.2	54.7	17.2	56.7	56.0	18.9	57.9	57.1	20.3	58.9	57.9	80	6	
		100	17.2	53.1	52.9	19.6	54.6	54.3	21.5	55.9	55.5	23.2	57.0	56.5	16.6	53.6	53.4	18.9	55.1	54.8	20.8	56.3	55.9	22.5	57.3	56.8	100		
		120	18.2	52.1	52.0	20.8	53.6	53.4	22.9	54.9	54.6	24.8	55.9	55.7	17.6	52.6	52.5	20.1	54.1	53.9	22.2	55.3	55.1	24.0	56.3	56.1	120		
		140	18.9	51.4	51.3	21.6	52.9	52.8	23.9	54.2	54.0	25.9	55.2	55.1	18.3	51.9	51.8	21.0	53.4	53.3	23.2	54.6	54.5	25.1	55.6	55.5	140		
		160	19.4	50.8	50.7	22.3	52.4	52.3	24.7	53.6	53.5	26.8	54.7	54.6	18.9	51.4	51.3	21.7	52.9	52.8	24.0	54.1	54.0	26.0	55.2	55.1	160		
	8	80	18.4	51.9	51.7	21.1	53.4	53.2	23.4	54.7	54.3	25.5	55.8	55.3	17.9	52.4	52.3	20.5	53.9	53.7	22.7	55.1	54.8	24.6	56.2	55.8	80	8	
		100	19.7	50.5	50.4	22.8	51.9	51.8	25.5	53.2	53.1	27.8	54.3	54.1	19.2	51.1	51.0	22.2	52.5	52.4	24.7	53.7	53.6	26.9	54.7	54.6	100		
		120	20.6	49.6	49.5	24.0	51.1	51.0	26.9	52.3	52.2	29.4	53.3	53.2	20.1	50.2	50.1	23.3	51.6	51.5	26.1	52.8	52.7	28.5	53.8	53.7	120		
		140	21.2	49.0	48.9	24.8	50.4	50.3	27.9	51.7	51.6	30.5	52.7	52.6	20.7	49.5	49.4	24.1	50.9	50.8	27.1	52.2	52.1	29.6	53.2	53.1	140		
		160	21.7	48.5	48.4	25.4	49.9	49.8	28.6	51.2	51.1	31.4	52.3	52.2	21.2	49.0	48.9	24.8	50.4	50.3	27.8	51.7	51.6	30.5	52.8	52.7	160		
4	2	80	8.0	65.3	60.9	8.8	66.5	61.7	9.5	67.5	62.2	10.1	68.2	62.7	7.7	65.6	61.2	8.5	66.7	61.9	9.1	67.7	62.4	9.7	68.4	62.9	80	2	4
		100	9.3	63.2	59.9	10.3	64.5	60.7	11.1	65.5	61.4	11.7	66.3	61.9	8.9	63.5	60.2	9.9	64.7	61.0	10.7	65.7	61.6	11.3	66.5	62.1	100		
		120	10.3	61.7	59.1	11.4	63.0	60.0	12.3	64.0	60.8	13.0	64.9	61.3	9.9	62.0	59.4	10.9	63.3	60.3	11.8	64.3	61.0	12.6	65.1	61.6	120		
		140	11.0	60.5	58.4	12.2	61.9	59.5	13.2	63.0	60.3	14.0	63.8	60.9	10.6	60.9	58.8	11.8	62.2	59.8	12.7	63.2	60.5	13.5	64.1	61.1	140		
		160	11.7	59.6	57.8	12.9	61.0	59.0	14.0	62.1	59.8	14.8	63.0	60.5	11.2	60.0	58.2	12.5	61.3	59.3	13.5	62.4	60.1	14.3	63.3	60.8	160		
	4	80	13.5	57.8	56.3	15.2	59.3	57.5	16.6	60.5	58.4	17.8	61.5	59.1	13.0	58.2	56.7	14.6	59.7	57.9	15.9	60.8	58.7	17.1	61.8	59.5	80	4	
		100	15.1	55.6	54.8	17.1	57.1	56.1	18.8	58.4	57.1	20.2	59.4	57.9	14.6	56.2	55.3	16.5	57.6	56.5	18.1	58.8	57.5	19.5	59.8	58.3	100		
		120	16.3	54.2	53.7	18.6	55.7	55.1	20.5	56.9	56.1	22.1	58.0	57.0	15.8	54.7	54.2	17.9	56.2	55.5	19.7	57.4	56.6	21.3	58.4	57.4	120		
		140	17.2	53.2	52.9	19.6	54.7	54.3	21.7	56.0	55.4	23.4	57.0	56.4	16.6	53.7	53.4	19.0	55.2	54.8	20.9	56.4	55.9	22.6	57.4	56.8	140		
		160	17.9	52.4	52.2	20.5	53.9	53.6	22.7	55.2	54.8	24.5	56.3	55.8	17.4	52.9	52.7	19.8	54.4	54.1	21.9	55.7	55.3	23.7	56.7	56.2	160		
	6	80	17.2	53.4	52.9	19.7	54.9	54.2	21.8	56.1	55.3	23.7	57.2	56.2	16.6	53.9	53.4	19.0	55.4	54.7	21.1	56.6	55.8	22.8	57.7	56.7	80	6	
		100	18.8	51.5	51.3	21.8	53.0	52.7	24.3	54.2	53.8	26.5	55.3	54.8	18.3	52.1	51.9	21.1	53.5	53.2	23.5	54.7	54.3	25.6	55.8	55.3	100		
		120	20.0	50.3	50.2	23.2	51.7	51.6	26.0	52.9	52.7	28.5	54.0	53.7	19.4	50.9	50.8	22.5	52.3	52.1	25.2	53.5	53.3	27.6	54.5	54.2	120		
		140	20.7	49.5	49.4	24.2	50.9	50.8	27.3	52.1	51.9	29.9	53.1	53.0	20.2	50.1	50.0	23.5	51.4	51.3	26.4	52.6	52.5	29.0	53.6	53.5	140		
		160	21.4	48.9	48.8	25.1	50.2	50.1	28.2	51.4	51.3	31.0	52.5	52.4	20.8	49.4	49.3	24.4	50.8	50.7	27.4	51.9	51.8	30.1	53.0	52.9	160		
	8	80	19.8	50.6	50.4	23.0	52.0	51.7	25.8	53.2	52.9	28.3	54.3	53.9	19.2	51.1	51.0	22.3	52.5	52.3	25.0	53.7	53.4	27.3	54.8	54.4	80	8	
		100	21.2	49.0	48.9	24.9	50.3	50.2	28.2	51.4	51.3	31.1	52.5	52.3	20.7	49.6	49.5	24.2	50.9	50.8	27.4	52.0	51.9	30.2	53.0	52.9	100		
		120	22.1	48.1	48.0	26.2	49.3	49.2	29.8	50.4	50.3	33.1	51.4	51.3	21.6	48.6	48.5	25.6	49.8	49.7	29.0	50.9	50.8	32.1	51.9	51.8	120		
		140	22.8	47.4	47.3	27.1	48.5	48.4	31.0	49.6	49.5	34.4	50.6	50.5	22.3	47.9	47.8	26.5	49.1	49.0	30.2	50.2	50.1	33.5	51.1	51.0	140		
		160	23.2	46.9	46.8	27.8	48.0	47.9	31.8	49.0	48.9	35.4	50.0	49.9	22.8	47.4	47.3	27.2	48.5	48.4	31.0	49.6	49.5	34.5	50.6	50.5	160		
6	2	80	8.5	64.9	60.5	9.3	66.2	61.4	10.1	67.1	61.9	10.7	67.9	62.4	8.1	65.2	60.8	9.0	66.4	61.6	9.7	67.4	62.2	10.3	68.1	62.6	80	2	6
		100	9.9	62.7	59.4	11.0	64.0	60.3	11.9	65.0	61.0	12.6	65.9	61.5	9.5	63.1	59.7	10.5	64.3	60.6	11.4	65.3	61.2	12.2	66.1	61.8	100		
		120	11.0	61.0	58.5	12.2	62.4	59.5	13.3	63.5	60.2	14.1	64.4	60.8	10.5	61.4	58.8	11.7	62.7	59.8	12.7	63.8	60.5	13.6	64.6	61.1	120		
		140	11.9	59.8	57.7	13.2	61.2	58.8	14.3	62.3	59.6	15.3	63.2	60.3	11.4	60.2	58.1	12.7	61.5	59.1	13.8	62.6	59.9	14.7	63.5	60.6	140		
		160	12.6	58.7	57.0	14.1	60.2	58.2	15.3	61.4	59.1	16.3	62.3	59.8	12.1	59.2	57.5	13.5	60.6	58.6	14.7	61.7	59.4	15.7	62.6	60.1	160		
	4	80	14.1	57.2	55.7	15.9	58.7	57.0	17.4	60.0	57.9	18.8	61.0	58.7	13.5	57.7	56.2	15.3	59.2	57.4	16.8	60.4	58.3	18.1	61.3	59.0	80	4	
		100	15.9	55.0	54.1	18.1	56.4	55.4	19.9	57.7	56.4	21.6	58.7	57.3	15.3	55.5	54.7	17.4	56.9	55.9	19.2	58.1	56.9	20.8	59.1	57.7	100		
		120	17.2	53.4	52.9	19.7	54.9	54.2	21.8	56.1	55.3	23.6	57.2	56.3	16.6	54.0	53.5	19.0	55.4	54.8	21.0	56.6	55.8	22.8	57.6	56.7	120		
		140	18.1	52.3	52.0	20.9	53.8	53.4	23.2	55.0	54.5	25.2	56.1	55.5	17.5	52.9	52.6	20.2	54.3	53.9	22.4	55.5	55.0	24.3	56.6	55.9	140		
		160	18.9	51.4	51.2	21.8	52.9	52.6	24.3	54.2	53.8	26.4	55.3	54.8	18.3	52.0	51.8	21.1	53.5	53.2	23.5	54.7	54.3	25.5	55.8	55.3	160		
	6	80	17.8	52.8	52.3	20.5	54.3	53.7	22.8	55.5	54.7	24.9	56.6	55.6	17.2	53.4	52.9	19.8	54.8	54.2	22.0	56.0	55.2	24.0	57.1	56.1	80	6	
		100	19.5	50.8	50.6	22.7	52.2	51.9	25.5	53.4	53.1	28.0	54.5	54.0	18.9	51.4	51.2	22.0	52.8	52.5	24.7	54.0	53.6	27.0	55.0	54.5	100		
		120	20.7	49.6	49.5	24.3	50.9	50.7	27.4	52.1	51.9	30.2	53.1	52.8	20.1	50.1	50.0	23.5	51.5	51.3	26.5	52.6	52.4	29.2	53.6	53.4	120		
		140	21.5	48.7	48.6	25.3	50.0	49.9	28.7	51.1	51.0	31.7	52.1	52.0	21.0	49.3	49.2	24.6	50.5	50.4	27.9	51.7	51.5	30.8	52.7	52.5	140		
		160	22.1	48.1	48.0	26.2	49.3	49.2	29.8	50.4	50.3	32.9	51.4	51.3	21.6	48.6	48.5	25.5	49.8	49.7	28.9	51.0	50.9	32.0	52.0	51.9	160		
	8	80	20.3	50.0	49.8	23.7	51.4	51.2	26.8	52.6	52.2	29.5	53.6	53.2	19.7	50.6	50.4	23.0	51.9	51.7	25.9	53.1	52.8	28.5	54.1	53.7	80	8	
		100	21.8	48.4	48.3	25.8	49.6	49.5	29.4	50.7	50.6	32.5	51.7	51.5	21.3	49.0	48.9	25.1	50.2	50.1	28.5	51.3	51.1	31.6	52.2	52.1	100		
		120	22.7	47.5	47.4	27.1	48.5	48.4	31.1	49.5	49.4	34.6	50.5	50.4	22.2	48.0	47.9	26.5	49.1	49.0	30.2	50.1	50.0	33.6	51.0	50.9	120		
		140	23.3	46.8	46.7	28.0	47.8	47.7	32.2	48.8	48.7	36.0	49.7	49.6	22.9	47.3	47.2	27.4	48.3	48.2	31.4	49.3	49.2	35.1	50.2	50.1	140		
		160	23.8	46.3	46.2	28.7	47.3	47.2	33.1	48.2	48.1	37.1	49.1	49.0	23.4	46.7	46.6	28.1	47.7	47.6	32.4	48.7	48.6	36.2	49.6	49.5	160		
8	2	80	8.8	64.7	60.3	9.7	65.9	61.1	10.4	67.0	61.8	11.1	67.8	62.2	8.3	65.0	60.6	9.2	66.2	61.4	10.0	67.2	62.0	10.7	68.0	62.4	80	2	8
		100	10.2	62.4	59.1	11.4	63.7	60.0	12.4	64.8	60.7	13.2	65.6	61.3	9.8	62.8	59.5	10.9	64.0	60.3	11.9	65.0	61.0	12.7	65.9	61.5	100		
		120	11.4	60.6	58.1	12.8	62.0	59.1	13.9	63.1	59.9	14.8	64.0	60.5	10.9	61.1	58.5	12.2	62.4	59.5	13.3	63.4	60.2	14.2	64.3	60.8	120		
		140	12.4	59.3	57.3	13.8	60.7	58.4	15.1	61.9	59.2	16.1	62.8	59.9	11.9	59.8	57.7	13.3	61.1	58.7	14.5	62.2	59.6	15.5	63.1	60.2	140		
		160	13.2	58.2	56.5	14.8	59.7	57.7	16.1	60.9	58.7	17.2	61.9	59.4	12.7	58.7	57.0	14.2	60.1	58.1	15.5	61.2	59.0	16.5	62.2	59.7	160		
	4	80	14.5	56.9	55.4	16.3	58.4	56.7	17.9	59.7	57.6	19.4	60.7	58.4	13.9	57.4	56.0	15.7	58.9	57.1	17.3	60.1	58.0	18.6	61.1	58.7	80	4	
		100	16.3	54.6	53.8	18.6	56.0	55.0	20.6	57.3	56.0	22.4	58.3	56.9	15.7	55.1	54.3	17.9	56.5	55.5	19.8	57.7	56.5	21.5	58.8	57.3	100		
		120	17.6	52.9	52.5	20.3	54.4	53.8	22.6	55.6	54.9	24.6	56.7	55.8	17.0	53.5	53.0	19.6	54.9	54.3	21.8	56.1	55.3	23.7	57.2	56.2	120		
		140	18.7	51.8	51.5	21.6	53.2	52.8	24.1	54.5	53.9	26.3	55.5	54.9	18.1	52.4	52.1	20.9	53.8	53.4	23.3	55.0	54.5	25.4	56.0	55.4	140		
		160	19.5	50.9	50.7	22.6	52.3	52.0	25.3	53.6	53.2	27.6	54.7	54.2	18.9	51.5	51.3	21.9	52.9	52.6	24.4	54.1	53.7	26.7	55.2	54.7	160		
	6	80	18.2	52.4	51.9	21.0	53.9	53.3	23.4	55.2	54.4	25.5	56.2	55.3	17.5	53.0	52.6	20.2	54.5	53.8	22.6	55.7	54.9	24.6	56.7	55.8	80	6	
		100	19.9	50.5	50.3	23.3	51.8	51.5	26.2	53.0	52.6	28.8	54.0	53.6	19.3	51.0	50.8	22.5	52.4	52.1	25.3	53.5	53.2	27.8	54.6	54.1	100		
		120	21.1	49.2	49.1	24.9	50.4	50.3	28.2	51.5	51.4	31.1	52.6	52.3	20.5	49.7	49.6	24.1	51.0	50.8	27.3	52.1	51.9	30.1	53.1	52.9	120		
		140	21.9	48.3	48.2	26.0	49.5	49.4	29.6	50.5	50.4	32.8	51.5	51.4	21.4	48.8	48.7	25.3	50.0	49.9	28.7	51.1	51.0	31.8	52.1	52.0	140		
		160	22.6	47.6	47.5	26.9	48.8	48.7	30.6	49.8	49.7	34.0	50.8	50.7	22.1	48.2	48.1	26.2	49.3	49.2	29.8	50.4	50.3	33.1	51.4	51.3	160		
	8	80	20.6	49.7	49.5	24.2	51.0	50.8	27.3	52.2	51.9	30.2	53.2	52.8	20.0	50.3	50.1	23.5	51.6	51.4	26.5	52.8	52.4	29.2	53.8	53.4	80	8	
		100	22.1	48.1	48.0	26.3	49.2	49.1	30.0	50.3	50.2	33.4	51.2	51.1	21.6	48.7	48.6	25.6	49.8	49.7	29.2	50.8	50.7	32.4	51.8	51.6	100		
		120	23.0	47.1	47.0	27.6	48.1	48.0	31.7	49.1	49.0	35.5	50.0	49.9	22.6	47.6	47.5	27.0	48.7	48.6	30.9	49.6	49.5	34.5	50.5	50.4	120		
		140	23.7	46.5	46.4	28.5	47.4	47.3	32.9	48.3	48.2	37.0	49.1	49.0	23.2	47.0	46.9	27.9	47.9	47.8	32.2	48.8	48.7	36.0	49.7	49.6	140		
		160	24.1	46.0	45.9	29.1	46.8	46.7	33.8	47.7	47.6	38.0	48.5	48.4	23.7	46.4	46.3	28.6	47.3	47.2	33.1	48.2	48.1	37.1	49.0	48.9	160		

NOTE: 1. MBh = MBh/ft² Coil Face Area LDB = Leaving Dry Bulb
WTR = Water Temperature Rise (F) LWB = Leaving Wet Bulb (F)
fps = Water Velocity (ft/sec) EWT = Entering Water Temperature
2. When using turbulators, make selection based on double the actual water velocity.

Note: 1. EWT=entering water temperature, °FFPS = water velocity, feet per secondLDB = leaving dry bulb, °F
MBH=1000 Btu/h per square foot of coil area fpm = air velocity, feet per minuteLWB = leaving wet bulb, °F
EDB=entering dry bulb, °FWTR = water temperature rise, °FFPF = fins per foot
EWB=entering wet bulb, °F
2. When using turbulators, make selection based on double the actual water velocity.

Entering liquid	35 to 65°F (2 to 18°C) temperature
Water flow rate	1.2 to 6 gpm per ton (0.02 to 0.1 L/s per kW) [equivalent to a water temperature rise of from 4 to 20°F (2 to 11°C)]
Water velocity	1 to 8 fps (0.3 to 2.4 m/s)

17.2.6 Determining Refrigeration Load

The following method of calculating refrigeration load shows a division of the true sensible and latent heat loss of the air which is accurate within the limitations of the data. This division does not correspond to load determination obtained from approximate factors or constants.

The total refrigeration load q_t of a cooling and dehumidifying coil (or air washer) per unit mass dry air is indicated in Figure 17-4 and consists of the following components:

1. The sensible heat q_s removed from the dry air and moisture in cooling from entering temperature t_1 to leaving temperature t_2.
2. The latent heat q_l removed to condense the moisture from W_1 to W_2.
3. The heat leaving the system as liquid condensate.

Items 1, 2 and 3 may be related by

$$q_t = q_s + q_l - q_w \tag{17-1}$$

If only the total heat value is desired, it may be computed by

$$q_t = (h_1 - h_2) - (W_1 - W_2)h_{f4} \tag{17-2}$$

where

h_1 and h_2 = enthalpy of moist air at points 1 and 2, respectively

W_1 and W_2= humidity ratio at points 1 and 2, respectively

h_{f4} = enthalpy of saturated liquid at final temperature

$W_1 - W_2$ = mass of water vapor condensed per unit mass of air

If a breakdown into latent and sensible heat components is desired, the following relations may be used. The latent heat may be found from

$$q_1 = (h_1 - h_3) - (W_1 - W_2)h_{f2} \tag{17-3}$$

where

h_{g1} = enthalpy of saturated water vapor at temperature t_1

h_{g3} = enthalpy defined by temperature T_1, and humidity ratio W_2

The sensible heat may be shown to be

$$q_s = (h_3 - h_2) + (W_1 + W_2)(h_{f2} - h_{f4}) \tag{17-4}$$

The final condensate temperature t_4 leaving the system is subject to substantial variations, depending on the method of coil installation, as affected by such factors as coil face orientation, airflow direction and air duct insulation. In practice, t_4 is frequently the same as the leaving wet-bulb temperature. Within the normal air-conditioning range, precise values of t_4 are not necessary since energy in the condensate removed from the air usually represents about 0.5 to 1.5% of the total refrigeration load. Values needed to calculate moist air properties can be found on the ASHRAE Psychrometric Chart (Chart 1) and Tables 2 and 3 of Chapter 1 of the 2017 *ASHRAE Handbook—Fundamentals*.

Example 17-1 Air enters a coil at 90°F dry bulb, 75°F wet bulb; it leaves at 61°F dry bulb, 58°F wet bulb; leaving liquid water is at 54°F.

Solution:

From the ASHRAE Psychrometric Chart, find the following:

h_1 = 38.37 Btu/lb$_m$ dry air

h_2 = 25.06 Btu/lb$_m$ dry air

h_3 = 32.14 Btu/lb$_m$ dry air

W_1 = 106.63 grains water vapor/lb$_m$ dry air

W_2 = 67.04 grains water vapor/lb$_m$ dry air

From Table 3, Chapter 1, of the 2017 *ASHRAE Handbook—Fundamentals*, find the following:

h_{f4} = 22.07 Btu/lb$_m$ liquid water

From Equation (17-2), the total heat is:

$$\begin{aligned} q_t &= (h_1 - h_2) - (W_1 - W_2)h_{f4} \\ &= (38.37 - 25.06) - \frac{(106.63 - 67.04)}{7000}(22.07) \\ &= 13.19 \text{ Btu/lb}_m \text{ dry air} \end{aligned}$$

From Equation (17-3), the latent heat is

$$\begin{aligned} q_1 &= (h_1 - h_3) - (W_1 - W_2)h_{f2} \\ &= (38.37 - 32.14) - \frac{(106.63 - 67.04)}{7000}29.08 \\ &= 6.07 \text{ Btu/lb}_m \text{ dry air} \end{aligned}$$

From Equation (17-4), the sensible heat is

$$\begin{aligned} q_s &= (h_3 - h_2) + (W_1 - W_2)(h_{f2} - h_{f4}) \\ &(32.14 - 25.06) + \frac{(106.63 - 67.04)}{7000}(29.08 - 22.07) \\ &7.12 \text{ Btu}_m \text{ dry air} \end{aligned}$$

The heat content of the leaving condensate is

$$\begin{aligned} q_w &= (W_1 - W_2)h_{f4} \\ &\frac{(106.63 - 67.04)}{7000}22.07 \\ &0.12 \text{ (Btu/lb}_m \text{ dry air)} \end{aligned}$$

17.3 Heating Coils

Generally, extended-surface coils are used for heating air with steam or hot water.

17.3.1 Steam Coils

For proper performance of steam heating coils, condensate and air or other noncondensables must be rapidly eliminated and the steam must be uniformly distributed to the individual tubes. Noncondensable gases (such as air or carbon dioxide) that remain in the coil cause chemical corrosion and result in early coil failure.

Steam is distributed uniformly by methods such as:

- Individual orifices in the tubes
- Distributing plates in the steam headers
- Perforated, small-diameter, inner steam-distributing tubes extending to the larger tubes of the primary surface

Coils with perforated inner tubes are constructed with different arrangements such as:

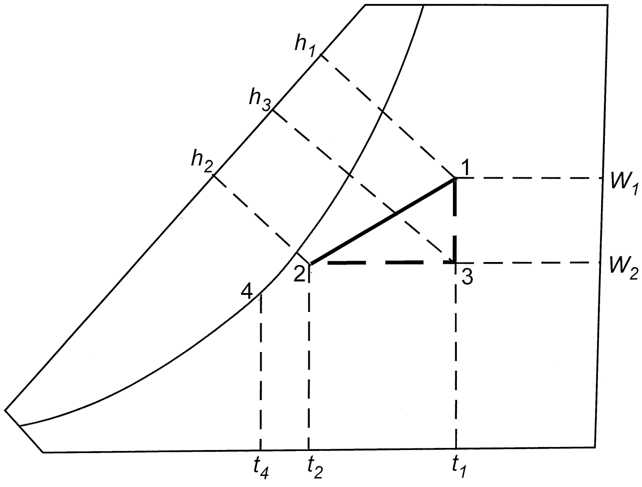
- Supply and return on one end, with the incoming steam used to heat the leaving condensate
- Supply and return on opposite ends
- Supply and return on one end and a supply on the opposite end

Properly designed and selected steam distribution coils distribute steam throughout the full length of all primary tubes, even when the leaving air temperature is controlled by modulating the steam supply through a steam-metering valve. Thus, more uniform leaving air temperatures are produced over the entire length and face of the coil than from a single-tube coil.

Piping, controls, and installation must be designed to protect the coils from freezing due to incomplete draining of condensate. When the entering air temperature is 32°F (0°C) or below, the steam supply to the coil should not be modulated. A series of coils in the airstream, with each coil sized to be on or completely off in a specific sequence depending on the entering air temperature, has less likelihood for a freeze-up. Bypass dampers are also common. When less than full load conditions occur, air is bypassed around the steam coil with full steam being kept on the coil. In this system, the high-velocity jets of low-temperature air must not impinge on the coil when the face dampers are in a partially closed position.

17.3.2 Water Coils

In order to produce the desired heating capacity, hot water comfort heating systems usually require no more than one or two rows of tubes in the direction of airflow. Various circuits are used to produce the most efficient capacity without excessive water pressure drop through the coil. The relative directions of fluid flows influence the performance of heat transfer surfaces. In air-heating coils with only one row of tubes, the air flows at right angles relative to the heating medium. In coils with more than one row of tubes in the direction of the airflow, the heating medium in the tubes may be circuited in various ways.



Psychrometric Performance of a Cooling and Dehumidifying Coil

Fig. 17-4 Psychrometric Performance of Cooling and Dehumidifying Coil

Although crossflow is common in steam-heating coils, parallel flow and counterflow arrangements are common in water coils. Counterflow is the preferred arrangement to obtain the highest possible mean temperature difference.

A single-tube serpentine circuit on small-size booster heaters requires small water quantities up to a maximum of approximately 4 or 5 gpm (0.25 or 0.32 L/s). With this arrangement, a single tube handles the entire water quantity, provided the tube is circuited such that it makes a number of passes across the airstream.

Commonly selected water circuits are illustrated in Figure 17-5. When entering air temperatures are below freezing (an antifreeze brine is sometimes used), piping the coil for parallel flow rather than counterflow should be considered, placing the highest water temperature on the entering air side. Coils piped for counterflow have water enter the coil in the tube row on the exit air side of the coil. Coils piped for parallel flow have water enter the tube row on the entering air side of the coil.

If air temperatures near or below freezing are anticipated, full design water flow rate should be ensured whenever the temperature approaches freezing. These coils are usually provided with a coil pump that provides constant flow through the pump with varying water temperature for reduced load control.

17.3.3 Electric Heating Coils

An electric heating coil consists of a length of resistance wire (commonly nickel/chromium) to which a voltage is applied. The resistance wire may be bare or sheathed. The

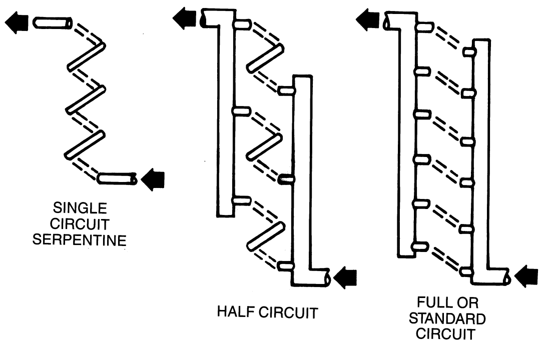


Fig. 17-5 Water Circuit Arrangements for Heating Coils

sheathed coil is a resistance wire encased by an electrically insulating layer such as magnesium oxide, which is encased in a finned steel tube. The sheathed coils are more expensive, have a higher air-side pressure drop, and require more space. The outer surface temperature of sheathed coils is lower, the coils are mechanically stronger, and electrical contact with body or housing is prevented.

17.3.4 Coil Ratings

Steam and hot water coils are usually rated within these limits, which may be exceeded for special applications:

Air face velocity	Between 200 and 1500 fpm (1 to 8 m/s), based on air at standard density of 0.075 lb/ft^3 (1.2 kg/m^3)
Entering air temperature	–20 to 100°F (–30 to 38°C) for steam coils; 0 to 100°F (–20 to 38°C) for hot water coils
Steam pressures	From 2 to 250 psia (14 to 1700 kPa) at the coil steam supply connection (pressure drop through the steam control valve must be considered)
Hot water	Between 120 and 250°F (50 and 120°C) temperature
Water velocities	From 0.5 to 8 fps (0.2 to 2.5 m/s)

Individual installations vary widely, but the following values can be used as a guide. The most common air face velocities used are between 500 and 1000 fpm (2.5 and 5 m/s). Delivered air temperatures vary from about 72°F (22°C) for ventilation only to about 150°F (66°C) for complete heating. Steam pressures vary from 2 to 15 psig [15 to 100 kPa (gage)], with 5 psig [35 kPa (gage)] being the most common. A minimum steam pressure of 5 psig [35 kPa (gage)] is recommended for systems with entering air temperatures below freezing. Water temperatures for comfort heating are commonly between 120 and 200°F (50 and 93°C), with water velocities between 4 and 6 fps (1.2 and 1.8 m/s). Water quantity is usually based on about 20 to 40°F (10 to 20°C) temperature drop through the coil. Air resistance is usually limited to 0.4 to 0.6 in. (100 to 150 Pa) of water for commercial buildings and to about 1 in. (250 Pa) for industrial buildings. High-temperature water systems have water temperatures commonly between 300 and 400°F (150 and 200°C), with up to 100°F (55°C) drops through the coil.

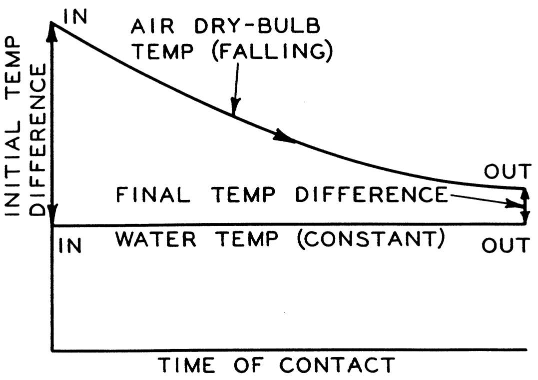


Fig. 17-6 Interaction of Air and Water in Evaporative Air Coolers

17.4 Evaporative Air-Cooling Equipment

In its broadest sense, the evaporative cooling principle applies to all equipment that exchanges sensible heat for latent heat. Cooling towers, evaporative condensers, vacuum cooling apparatus, air washers, spray coil dehumidifiers, and packaged evaporative coolers cool air by evaporation. This equipment falls into two general categories: (1) apparatus for air cooling and (2) apparatus for heat rejection.

Evaporative air cooling evaporates water into an airstream. Illustrated in Figure 17-6 are thermodynamic changes that take place between air and water that are in direct contact in the moving airstream. The continuously recirculated water has reached an equilibrium temperature that equals the entering air wet-bulb temperature. The heat and mass transfer process between the air and water lowers the air dry-bulb temperature and increases the humidity ratio at constant wet-bulb temperature.

The extent to which the leaving air temperature approaches the thermodynamic wet-bulb temperature of the entering air, or the extent to which complete saturation is approached, is conveniently expressed as cooling or saturation efficiency and is defined as

$$e_c = (t_1 - t_2)/(t_1 - t') \tag{17-5}$$

where

e_c = cooling or saturation efficiency
t_1 = dry-bulb temperature of the entering air
t_2 = dry-bulb temperature of the leaving air
t' = thermodynamic wet-bulb temperature of entering air

If warm or cold unrecirculated water is used, the air can be heated and humidified or cooled and dehumidified.

Evaporative air-cooling equipment can be placed in two general classes, direct and indirect. In the direct system, air is cooled by direct contact with the water, either from the wetted

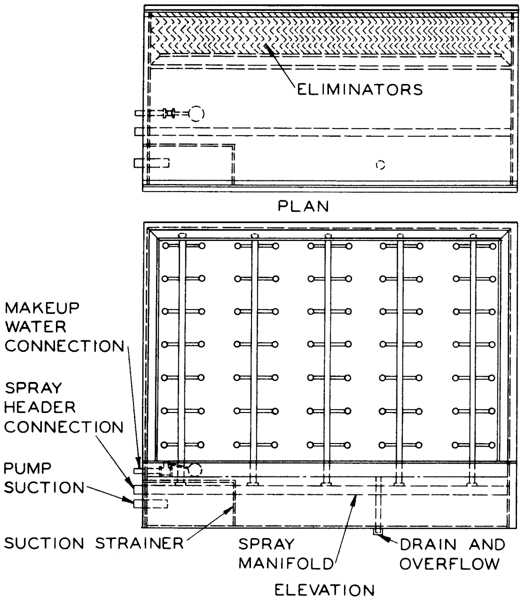


Fig. 17-7 Single-Bank Air Washer

surface of an extended-surface material (as in packaged air coolers) or with a series of sprays (as in an air washer). In the indirect system, air is cooled in a heat exchanger by a secondary stream of air and water that has been evaporatively cooled, such as by a cooling tower and cooling coil.

By applying recirculating, regenerating principles, temperatures below the initial wet bulb may be produced. However, the cost of these more complex devices has restricted their use.

17.5 Air Washers

A spray-type air washer consists of a chamber or casing containing a spray nozzle system, a tank for collecting the spray water as it falls, and an eliminator section at the discharge to remove the entrained drops of water from the air. A pump recirculates water at a rate in excess of the evaporation rate. Heat and mass transfer between the air and the water cools the water.

Construction features of conventional spray-type air washers are shown in Figure 17-7. Requirements of air washer operation are:

- Uniform distribution of air across the spray chamber
- Air velocities from 300 to 700 fpm (1.5 to 3.5 m/s) in the washer chamber
- Adequate spray water broken up into fine droplets at pressures from 20 to 40 psig [140 to 280 kPa (gage)]
- Good spray distribution across the airstream
- Sufficient length of travel through the spray and wetted surfaces
- Elimination of free moisture from the outlet air

Spray water requirements for spray-type air washers used for washing or evaporative cooling vary from 4 gpm per 1000 cfm with a single bank to 10 gpm per 1000 cfm for double banks (0.5 to 1.3 L/s per m^3/s). Pumping heads usually range from 55 to 150 ft (160 to 450 kPa) of water, depending on such factors as spray pressure, height of apparatus, and pressure losses in pipe and strainers.

17.6 Dehumidification

Dehumidification is the reduction of the water content of air, gases, or other fluids. Dehumidification is normally limited to equipment operating at essentially atmospheric pressures, built to standards similar to other types of air-handling equipment.

Drying of gases has become an increasingly important operation. Some commercial applications include the following:

- Lowering the relative humidity to facilitate manufacturing and handling of hygroscopic materials
- Air conditioning for comfort (in combination with cooling, under certain design conditions, such as high moisture load in comparison to sensible cooling load)
- Providing protective atmospheres for the heat treatment of metals
- Maintaining controlled humidity conditions in warehouses and caves for storage
- Preserving ships and other surplus equipment, which would otherwisedeteriorate
- Condensation and corrosion control
- Drying air for wind tunnels
- Drying natural gas
- Drying instrument air and plant air
- Drying of process and industrial gases
- Dehydration of liquids

17.6.1 Methods of Dehumidification

Dehumidification can be accomplished by compression, refrigeration, liquid sorption, solid sorption, or combinations of these systems.

Three methods by which sorbent materials or sorbent equipment dehumidify air are illustrated in the skeleton psychrometric chart shown in Figure 17-8. Air at point A can be dehumidified and cooled to point B by a liquid sorption system with intercooling directly. Alternatively, it may be dehumidified in a solid sorption unit by precooling and dehumidifying from point A to point C, desiccating from point C to point E, and finally cooling to point B. Air could also be dehumidified with solid sorption equipment by desiccating from point A to point D and then by refrigeration from point D to point B.

Compression Dehumidification. Compressing a gas to be dehumidified reduces its absolute moisture content, but it generally produces a saturated condition at the elevated pressure. This method is uneconomical but is of value for pressure systems since part of the moisture is removed by compression of the gas; the remaining moisture may be removed by cooling alone, with sorption, or both, depending on the final dew

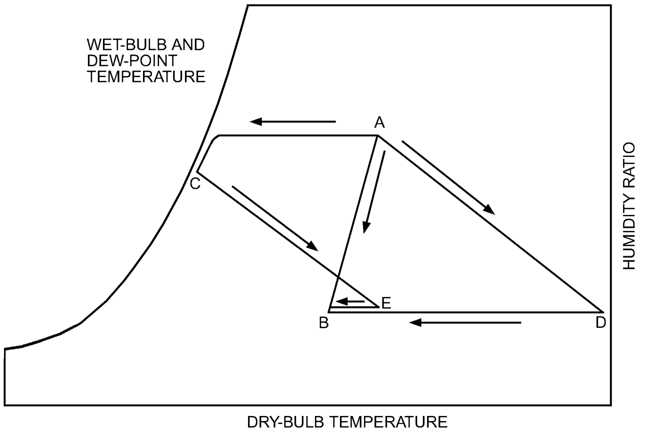


Fig. 17-8 Methods of Dehumidification
(Figure 1, Chapter 24, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

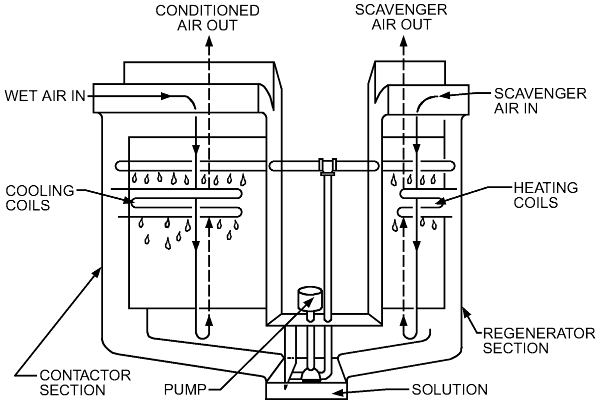


Fig. 17-9 Flow Diagram for Liquid-Absorbent Dehumidifier
(Figure 2, Chapter 24 2016 ASHRAE Handbook—HVAC Systems and Equipment)

point required. Expansion of high-pressure gas also lowers the dew point.

Refrigeration Dehumidification. Refrigerating gas below its dew point is the most common method of dehumidification. This method is advantageous when the gas (1) is comparatively warm, (2) has a high moisture content, and (3) requires an outlet dew point above 40°F (5°C). Frequently, refrigeration is used in combination with sorption dehumidifiers to obtain an extremely low dew point that is difficult to achieve with refrigeration alone.

Direct Sorption Dehumidification. Sorbent materials used in dehumidification equipment may either be liquids or solids. The performance of the different sorption dehumidification machines is, to some extent, a function of the sorbent used. Sorbents either retain water on their surface (adsorption) or chemically combine with water (absorption). In regenerative equipment, the water must be removed from the sorbent to regenerate it for further use.

Nonregenerative equipment uses hygroscopic salts such as calcium chloride, urea, or sodium chloride. In regenerative systems, the sorbent is usually a form of silica, alumina gel, activated alumina, molecular sieves, lithium chloride salt, lithium chloride solution, or glycol solution.

In liquid sorption dehumidification systems, the gas passes through sprays of a liquid sorbent such as lithium chloride or a glycol solution. The sorbent in active state has a vapor pressure below that of the gas to be dehumidified and absorbs moisture from the gas stream. The sorbent solution during the process of absorption becomes diluted with moisture, which during regeneration is given up to an outdoor airstream in which the solution is heated. A partial bleed-off of the solution is used for continuous reconcentration of the sorbent in a closed circuit between the spraying or contactor unit and the regenerator unit.

In solid sorption, the gas stream passes through or over granular beds of fixed desiccant structures. A number of commercially available desiccants may be used, depending on such factors as the character of the gas to be dried, inlet temperature, moisture levels, and required final dew point. Outdoor air is passed through beds or layers of the sorbent, which in its active state has a vapor pressure below that of the gas to be dehumidified and absorbs moisture from the gas stream. After becoming saturated with moisture, the desiccant is periodically reactivated to give up previously absorbed moisture to an outdoor air or gas stream.

17.6.2 Liquid Absorption Equipment

The flow diagram for a typical liquid absorption system with extended-surface contactor coils is shown in Figure 17-9. For dehumidifying operation, the strong absorbent solution is pumped from the sump of the unit and sprayed over the contactor coils.

Air to be conditioned passes over the contactor coils and comes in contact with the hygroscopic solution. Airflow can be either parallel with or counter to the sprayed solution flow, depending on the space and application requirements.

The degree of dehumidification depends on the concentration, temperature, and characteristics of the hygroscopic solution. Moisture is absorbed from the air by the solution due to the vapor pressure difference between the air and the liquid absorbent. The moisture content of the outlet air can be precisely maintained by varying the coolant flow in the coil to control the absorbent solution contact temperature. The absorbent solution is maintained at the proper concentration by continuous regeneration.

The heat generated in absorbing moisture from the air consists of the latent heat of condensation of water vapor and the heat of solution, or the heat of mixing, of the water and the absorbent. The heat of mixing varies with the liquid absorbent used and with the concentration and temperature of the absorbent. The total heat removal required by the conditioner coil consists of the heat of absorption, sensible heat removed from the air, and the residual heat load added by the regeneration process.

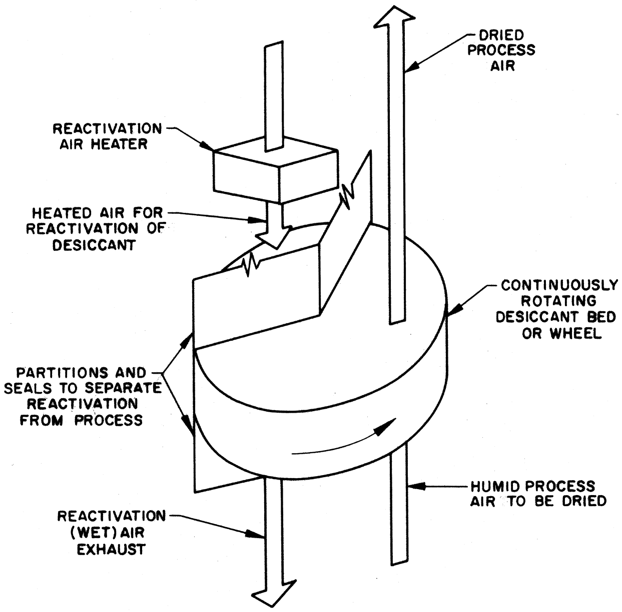


Fig. 17-10 Rotary Dehumidification Unit

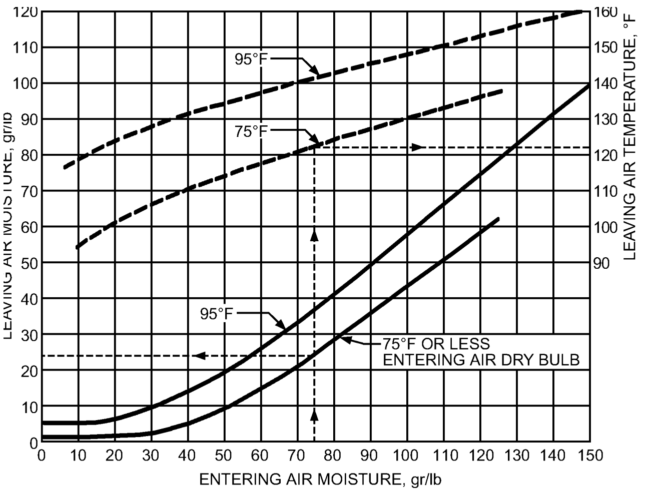


Fig. 17-11 Performance Data for Rotary Solid Absorption Dehumidifier

(Figure 13, Chapter 24, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

17.6.3 Solid Sorption

Dehumidification by use of a solid desiccant, such as silica gel, molecular sieves, activated alumina, or hygroscopic salts, may be performed under either static or dynamic conditions. In the static method, the air or gas is not forced through the desiccant. Instead, the air immediately surrounding the desiccant dries, and through convection and vapor diffusion, water vapor from surrounding areas reaches the desiccant where it is absorbed. In dynamic dehumidification, the air or gas being dried flows through the desiccant bed or structure. In the air-conditioning industry, which is primarily concerned with operation at atmospheric pressure, an air-moving device such as a fan forces the gas through the desiccant bed and a heater or other means periodically reactivates the desiccant.

17.6.4 Solid Absorption Equipment

The arrangement of major components of a typical solid absorption system is shown in Figure 17-10. To achieve dehumidification, moist air is passed through a desiccant structure, where the water vapor in the air is absorbed by the desiccant. The systems employ a desiccant structure that may be a disc, drum, or wheel, filled or saturated with an absorbent, such as lithium chloride. The desiccant structure rotates slowly through a heated stream of air where the desiccant is desorbed or reactivated. By continuous rotation, freshly reactivated portions of the desiccant are always available for drying.

The amount of drying depends on the temperature and absolute humidity of the air and the useful concentration of desiccant. The useful concentration of the desiccant is affected by:

- Quantity of desiccant in relation to the mass flow of air and water vapor
- Energy (amount of heat and temperature) for reactivation
- Frequency of reactivation (speed of rotation)

Performance typical of a solid absorption dehumidifier using a fixed desiccant structure is shown in Figure 17-11.

17.7 Humidification

When selecting humidification equipment, both the environmental conditions of the occupancy or process and the characteristics of the building enclosure must be considered. These may not always be compatible and a compromise solution may be necessary.

Air washers and evaporative coolers may be used as humidifiers but are usually selected to provide some additional function, such as air cooling or air cleaning, as discussed in the section on air washers.

Residential Humidifiers for Central Air Systems. This type of unit depends on airflow in the heating system for evaporation and distribution. General principles of operation and description of equipment are as follows:

Pan Type. The humidification rate varies with temperature, humidity, and velocity of the air in the system (Figure 17-12).

Wetted Elements. These units use an open textured, wetted media, through or over which air is circulated. The evaporating surface may take the form of a fixed pad, wetted by sprays or by water flowing through by gravity from a header at the top, or the pad may be a paddle wheel, drum, or belt rotating through a water reservoir. Figures 17-13 and 17-14 are examples of this type of humidifier.

Residential Humidifiers for Nonducted Applications. Many portable or room humidifiers are available for use in

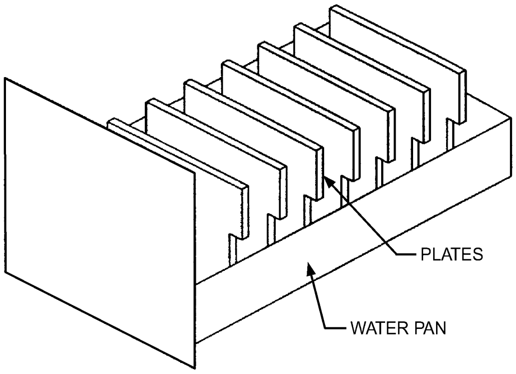


Fig. 17-12 Pan-Type Humidifier

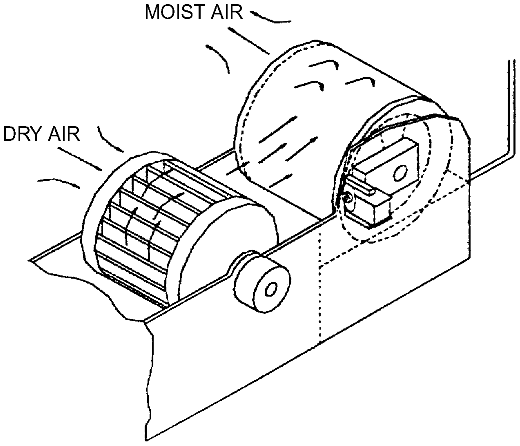


Fig. 17-13 Wetted-Drum Humidifier

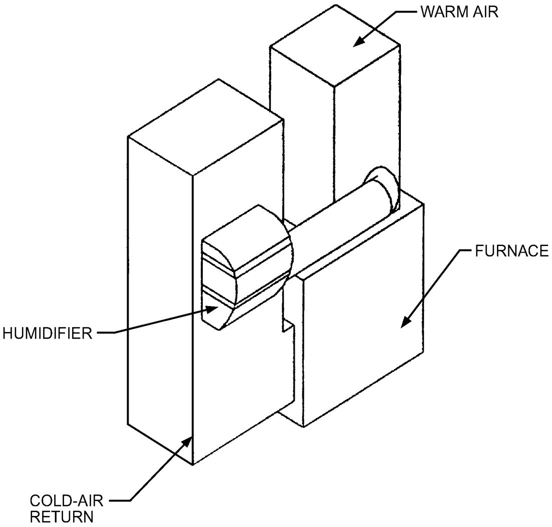


Fig. 17-14 Bypass Wetted-Element Humidifier

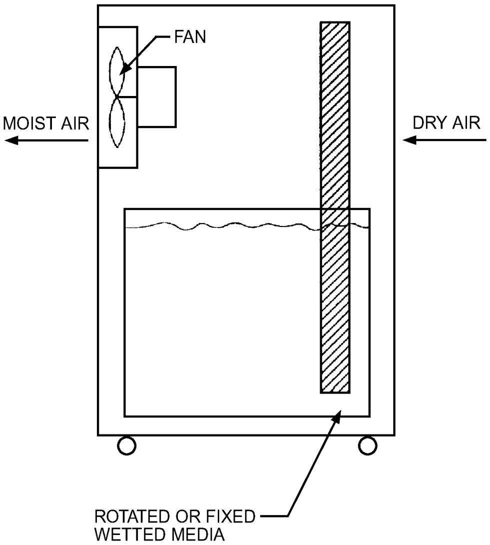


Fig. 17-15 Portable Humidifier

residences and apartments heated by nonducted systems, such as hydronic or electric, or where the occupant is prevented from making a permanent installation. An example of this type of humidifier is shown in Figure 17-15.

Industrial/Commercial Humidifiers for Central Air Systems. Humidification equipment commonly used in central air-handling systems incorporates a heated water pan, direct steam injection, or atomizers. Specific types are shown in Figure 17-16 and discussed briefly in the following paragraphs.

Heated Pan. These units offer a broad range of capacity and may be heated by an electric element, steam, or hot water coil (see Figure 17-16A). Some units are designed to be attached directly to the underside of a system duct. Others are provided with a fan or a steam hose, which allows them to be installed remote from the ductwork.

Steam. Direct steam injection humidifiers cover a wide range of designs and capacities. Since water vapor is steam at low pressure and temperature, the whole process can be simplified by introducing steam directly into the air to be humidified. This method is essentially an isothermal process because the temperature of the air remains constant as the moisture is added in vapor form. The steam control valve may be modulating or two-position in response to a humidity controller. The steam may be from an external source, as in the enclosed grid, cup, or jacketed dry steam humidifiers, or produced within the humidifier, as in the self-contained type.

- *Enclosed steam grid humidifier*s (Figure 17-16B) should be used on low steam pressures to prevent splashing of condensate in the duct.
- A *cup* or *pot-type humidifier* (Figure 17-16C) is usually attached under a system duct. Steam is attached tangentially to the inner periphery of the cup by one or more steam

inlets, depending on the capability of the unit. The steam supply line should have a suitable steam trap.

- A *jacketed steam humidifier* uses an integral steam valve with a steam-jacketed duct-traversing dispersing tube and condensate separator to prevent condensate from being introduced into the airstream (Figure 17-16D). An inverted bucket-type steam trap is required to drain the separating chamber.

The aforementioned humidifiers inject steam directly from the boiler into the space or duct system. Some boiler treatment chemicals can be discharged, which can affect indoor air quality.

- A *self-contained steam humidifier* converts tap water to steam by electrical energy using either the electrode boiler principle or resistance heating. This steam is injected into the duct system through a dispersion manifold (Figure 17-16E), or the humidifier may be freestanding for nonducted applications.

Atomizing humidifiers with optional filter eliminator (Figure 17-16F). Centrifugal atomizers use a high-speed disc that slings water through a fine comb to create a fine mist that is introduced directly into the air where it is evaporated. The ability of the air to absorb the moisture depends on temperature, air velocity, and moisture content. Where mineral fallout from hard water is a problem, optional filter eliminators may be added to remove mineral dust from humidified air, or water demineralizers may be installed. Additional atomizing methods use nozzles; one uses water pressure and the other uses both air and water, as shown in Figure 17-16G. Mixing air and water streams at combined pressures atomizes water into a fine mist, which is evaporated in the room or air duct.

Wetted-element humidifier. Wetted-element humidifiers have a wetted media, sometimes in modular configurations, through or over which air is circulated to evaporate water. This unit depends on airflow for evaporation; the rate varies with temperature, humidity, and velocity of the air.

17.8 Sprayed Coil Humidifiers/Dehumidifiers

A special adaptation of an air washer can be used very effectively to simultaneously control both the temperature and humidity of an outdoor airstream as in a dedicated outdoor air system (DOAS).

The cooling coil system provided with a deep cooling coil (usually 8 to 12 rows with flat [nonenhanced] fins spaced not less than 1/8 in. apart and with the entire coil assembly and housing made of a noncorrosive material such as stainless steel.) (Figure 17-17A)

The coil is located in a stainless steel section of the air-handling unit over a drain pan with an air spray and recirculating pump. The air spray sprays water from the pump into the airstream against the cooling coil. The water tends to approach the average temperature of the coil and the leaving airstream dew-point temperature. The coil/spray assembly essentially has a coil efficiency of 100% (by pass factor of 0), and the coil-leaving air is saturated at the temperature of the water.

Since the system also functions as an air washer, it very effectively removes particulate (and some chemical) contamination from the air.

Another benefit is that it serves as both a humidifier and a dehumidifier as necessary, depending upon the dew-point temperature or humidity ratio of the leaving air stream compared to the entering air stream Figure 17-17B is a skeleton psychometric chart illustrating the performance as a humidifier and as a dehumidifier. It is controlled by a single discharge temperature sensor set at the desired dew point temperature and operating the heating coil and chilled water coil in sequence. For freeze protection, the heating coil is usually equipped with a constant flow pump or an integrated face and bypass damper.

The sprayed coil DOAS is very effective in maintaining a fixed dew point temperature for critical applications such as art museums, archived storage, rare book libraries, critical humidity manufacturing processes, surgical suites, etc.

Since the sprayed coil and water system functions as an air washer, it requires scheduled blow down and cleaning to remove the particulate collected from the pan and the coil surfaces. Also, to prevent bacterial growth, the safety controls should shut the system down if the water temperature exceeds 58°F.

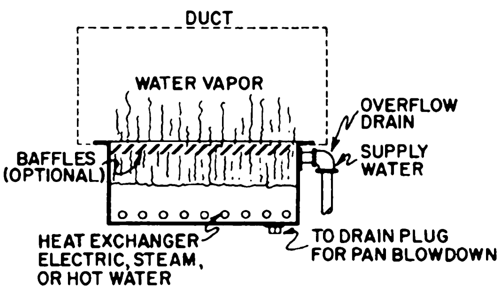
17.9 Air Cleaners

In buildings, dust content is usually less than 0.2 mg/m^3 (0.01×10^{-6} lb$_m$/ft^3) of air. This concentration is in contrast to exhaust gases from processes and flue gases where dust concentration typically ranges from 200 to 40 000 mg/m^3 (10×10^{-6} to 2000×10^{-6} lb$_m$/ft^3).

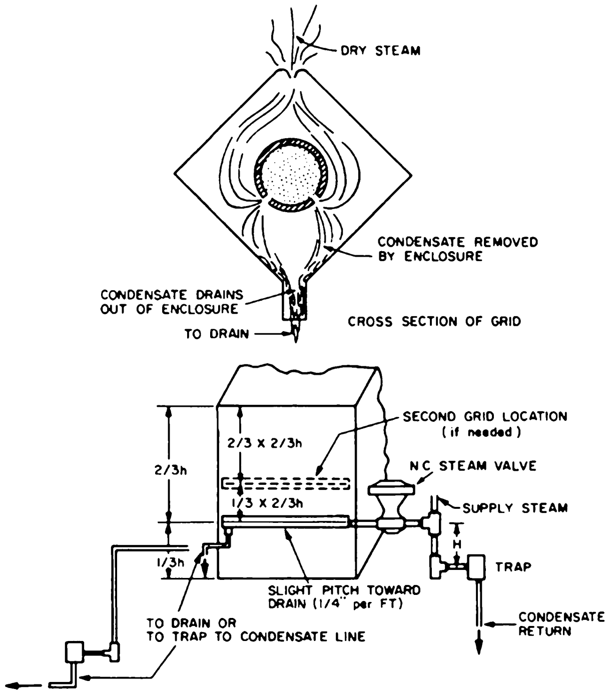
With certain exceptions, the air cleaners described in this section are not applicable to the cleaning of exhaust gases. Atmospheric dust is a complex mixture of smokes, mists, fumes, dry granular particles, and fibers (such particles, when suspended in a gas, are called aerosols). A sample of atmospheric dust gathered at any point generally contains materials common to that locality, together with other components that originated at a distance but have been transported by air currents or diffusion.

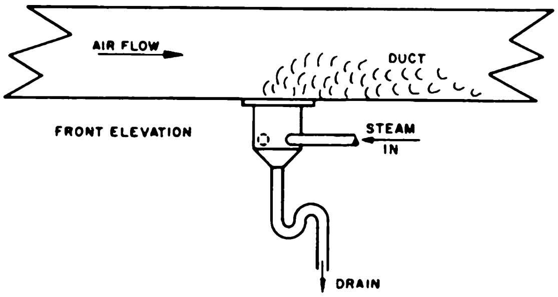
These components vary with the geography of the locality in question, the season of the year, the direction and strength of the wind, and the proximity of dust sources. A sample of atmospheric dust usually contains soot and smoke, silica, clay, decayed animal and vegetable matter, organic materials in the form of lint and plant fibers, and metallic fragments. It may also contain living organisms, such as mold spores, bacteria, and plant pollens, which may cause diseases or allergic responses. Chapter 11 of the 2017 *ASHRAE Handbook—Fundamentals* contains further information on atmospheric contaminants.

The particles in the atmosphere can range in size from less than 0.01 mm to dimensions of lint, leaves, and insects.



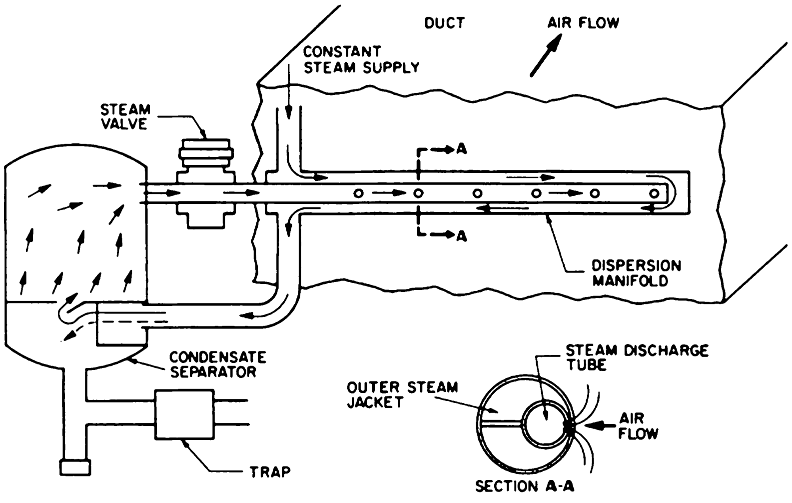
A. Heated Pan Humidifier

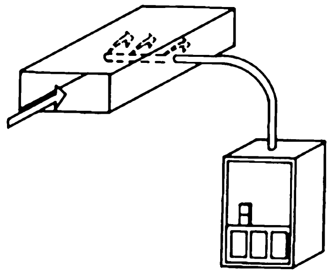




C. Cup Steam Humidifier

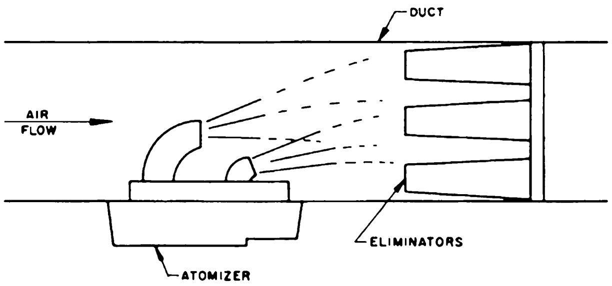
B. Enclosed Steam Grid Humdifier

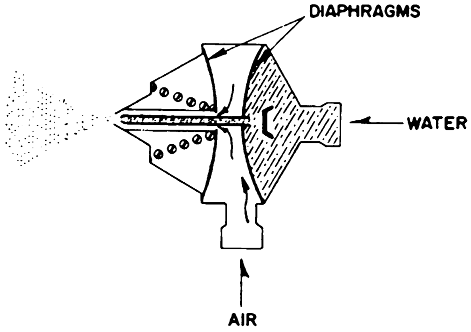




E. Self-contained Steam Humidifier

D. Jacketed Steam Humidifier





F. Atomizing Humidifier with Optional Filter Eliminator

G. Pneumatic Atomizing Humidifier

Fig. 17-16 Commercial Humidifiers

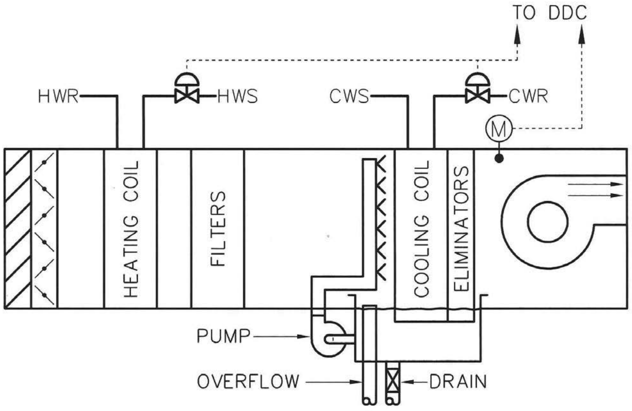


Fig. 17-17a Sprayed Coil DOAS

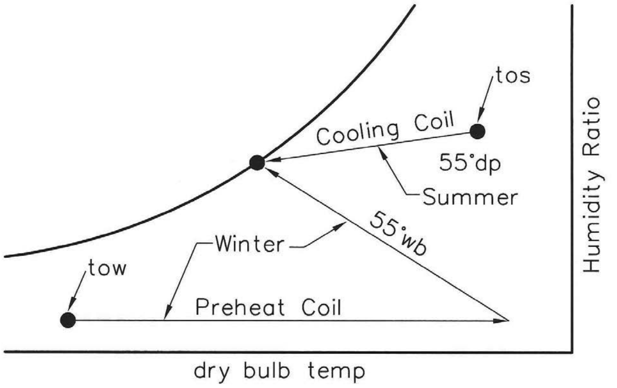


Fig. 17-17b Psychrometric Process for Sprayed Coil DOAS

Almost all conceivable shapes and sizes are represented. This wide variety makes it impossible to design one cleaner that is best for all applications. For example, in industrial ventilation, only the coarser dust particles may need to be removed from the airstream for cleanliness of the structure and protection of mechanical equipment. In other applications, surface discoloration must be prevented. Unfortunately, the smaller components of atmospheric dust are the worst offenders in smudging and discoloring building interiors. Electronic air cleaners or high-efficiency dry filters are required for small particle removal. In clean room applications or when radioactive or other dangerous particles are present, extremely high-efficiency mechanical filters should be used.

The most important characteristics of aerosols affecting the performance of an air filter include particle size and shape, density, and concentration. The most important of these is particle size. Data on the sizes and characteristics of airborne particulate matter and the wide range of particle size encountered are given in Figure 17-18. Cleaning efficiency is also affected by the velocity of the airstream. The degree of air cleanliness required is a major factor influencing filter design and selection. Removal of particles becomes progressively more difficult as particle size decreases.

Cost considerations (both in initial investment and maintenance), space requirements, and airflow resistance, in addition to wide-ranging criteria as to degree of air cleanliness, have resulted in a wide variety of commercial air cleaners.

To evaluate filters and air cleaners properly for a particular application, the following factors should be considered:

- Degree of air cleanliness required
- Disposal of dust after it is removed from the air
- Amount and type of dust in the air filtered
- Operating resistance to airflow (pressure drop)
- Space available for filtration equipment
- Cost of maintaining or replacing filters
- Initial cost of the system

Figure 17-19 shows typical locations at filters and lists some applications of filters classified according to their efficiencies and type.

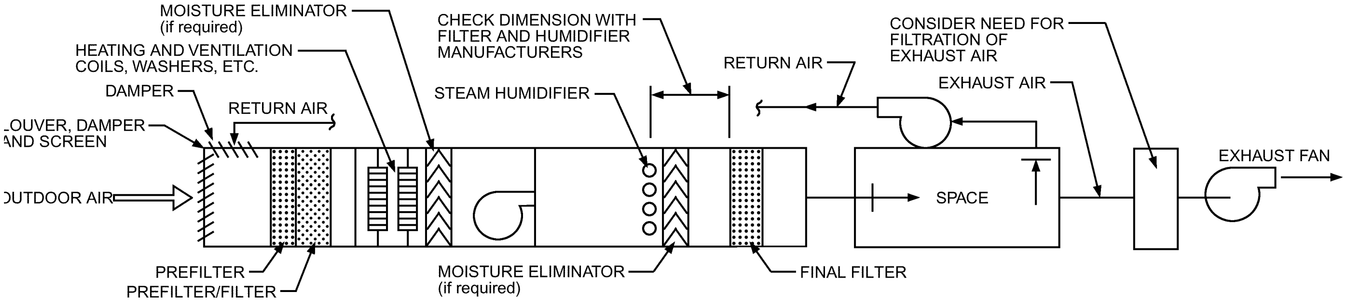
17.9.1 Types of Air Cleaners

Air cleaners fall into the following three categories:

1. *Fibrous media unit filters* in which accumulating dust load causes pressure drop to increase up to some maximum permissible value. During this period, efficiency normally increases; however, at high dust loads, dust may adhere poorly to filter fibers, and efficiency drops. Filters in such condition should be replaced or reconditioned, as should filters that have reached their terminal (maximum permissible) pressure drop. This category includes both viscous impingement and dry air filters.
2. *Renewable media filters* in which fresh media is introduced into the airstream as needed to maintain essentially constant resistance. This also maintains essentially constant efficiency.
3. *Electronic air cleaners*, which have essentially constant pressure drop and efficiency unless their precipitating elements become severely dust-loaded.

17.9.2 Fibrous Media Unit Filters

Viscous Impingement Filters. These flat panel filters consist of coarse fibers and have a high porosity. The filter media are coated with a viscous substance, such as oil, which acts as an adhesive for particles that impinge on the fibers. Design air velocity through the media is usually in the range of 250 to 700 fpm (1 to 4 m/s). These filters are characterized by low pressure drop, low cost, and good efficiency on lint, but low efficiency on normal atmospheric dust. They are commonly 1/2 to 4 in. (13 to 100 mm) thick; 1 to 2 in. (25 to 50 mm) nominal thickness is most popular. The thicker configurations have a high dust-holding capacity. Unit panels are available in standard and special sizes up to about 24 in. by 24 in. (610 mm by

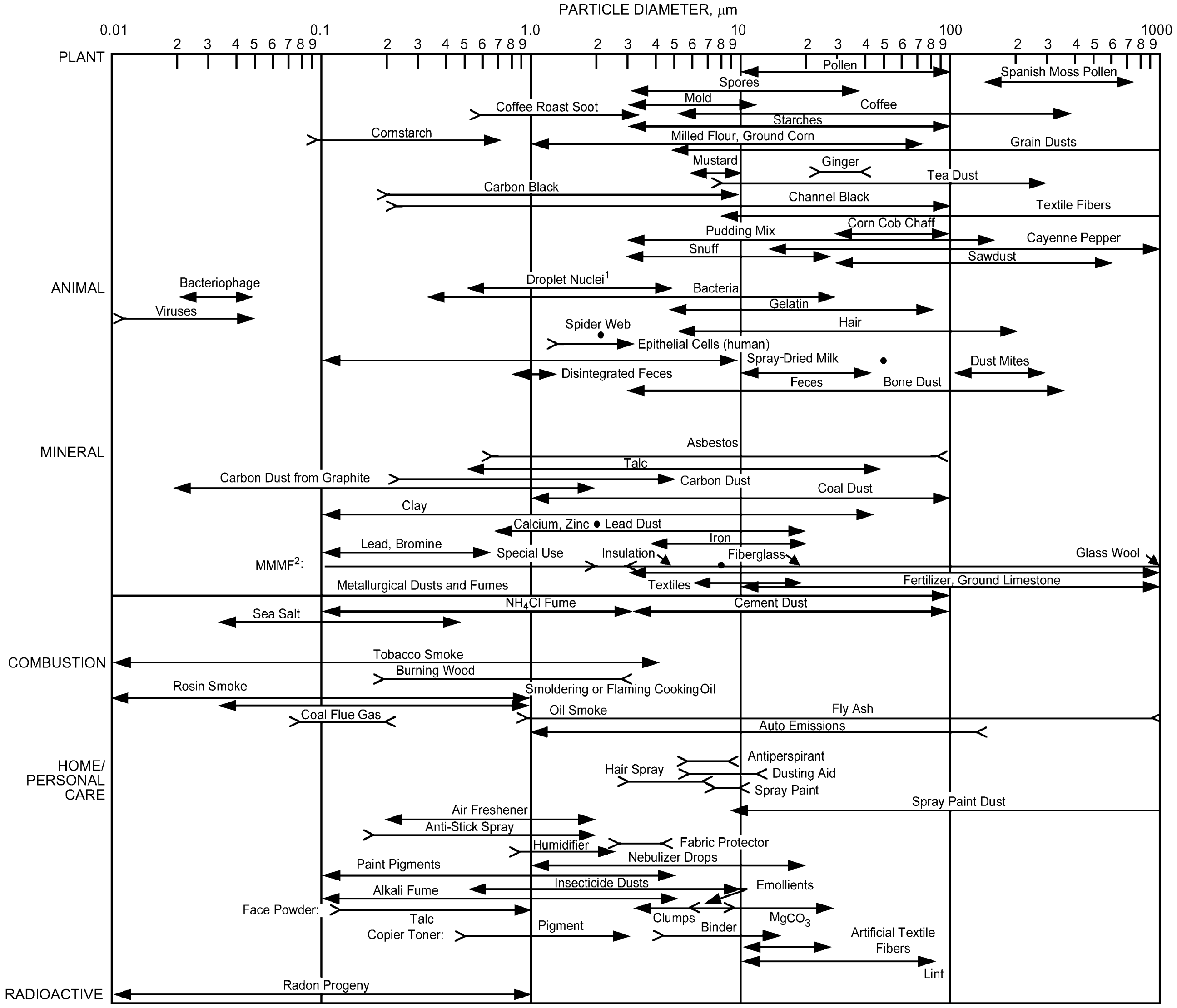


Standard 52.2 MERV	Intended Standard 52.1 Value	Arrestance Value	Example Range of Contaminants Controlled	Example Applications	Sample Air Cleaner Type(s)
HEPA filters					
MERV 20	N/A		0.12 to 0.5 µm particles: virus (unattached), carbon dust, sea salt, radon progeny, combustion smoke	Cleanroom, pharmaceutical manufacturing and exhaust, radioactive material handling and exhaust, orthopedic and organ transplant surgery, carcinogenic materials, welding fumes	SULPA >99.999% 0.1 to 0.2 µm IEST type F (ceiling panel)
MERV 19					ULPA >99.999% 0.3 µm IEST type D (ceiling panel)
MERV 18					HEPA > 99.99% 0.3 µm IEST type C (ceiling or up to 12 in.deep)
MERV 17					HEPA > 99.97% 0.3 µm IEST type A (box style 6 to 12 in. deep)
E-1 Range					
MERV 16	Intended to replace 70 to 98% dust-spot efficiency filters	>99%	0.3 to 1.0 µm size range: bacteria, smoke (ETS), paint pigments, face powder, some virus, droplet nuclei, insecticide dusts, soldering fumes	Day surgery, general surgery, hospital general ventilation, turbo equipment, compressors, welding/soldering air cleaners, prefilters to HEPAs, LEED for existing (EB) and new (NC) commercial buildings, smoking lounges	Box-style wet-laid or lofted fiberglass, box-style synthetic media, minipleated synthetic or fiberglass paper, depths from 4 to 12 in., Pocket filters of fiberglass or synthetic media 12 to 36 in.
MERV 15		>99%			
MERV 14		>98%			
MERV 13		>97%			
E-2 Range					
MERV 12	Intended to replace 50 to 80% dust-spot efficiency filters	>97%	1.0 to 3.0 µm size range: milled flour, lead dust, combustion soot, *Legionella*, coal dust, some bacteria, process grinding dust	Food processing facilities, air separation plants, commercial buildings, better residential, industrial air cleaning, prefiltration to higher-efficiency filters, schools, gymnasiums	Box-style wet-laid or lofted fiberglass, box-style synthetic media, minipleated synthetic or fiberglass paper, depths from 2 to 12 in. Pocket filters either rigid or flexible in synthetic or fiberglass, depths from 12 to 36 in.
MERV 11		>95%			
MERV 10		>95%			
MERV 9		>90%			
E-3 Range					
MERV 8	Intended to replace 20 to 60% dust-spot efficiency filters	>90%	3.0 to 10 µm size range: pollens, earth-origin dust, mold spores, cement dust, powdered milk, snuff, hair spray mist	General HVAC filtration, industrial equipment filtration, commercial property, schools, prefilter to high-efficiency filters, paint booth intakes, electrical/phone equipment protection	Wide range of pleated media, ring panels, cubes, pockets in synthetic or fiberglass, disposable panels, depths from 1 to 24 in.
MERV 7		>90%			
MERV 6		>85%			
MERV 5		>85%			
MERV 4	<20%	>70%	Arrestance method	Protection from blowing large particle dirt and debris, industrial environment ventilation air	Inertial separators
MERV 3	<20%	>70%			
MERV 2	<20%	>65%			
MERV 1	<20%	<65%			

Note: MERV for non-HEPA/ULPA filters also includes test airflow rate, but it is not shown here because it is of no significance for the purposes of this table.
N/A = not applicable.

Fig. 17-19 Typical Filter Locations and Cross Reference and Application Guidelines

(Figure 3 and Table 2, Chapter 29, 2016 ASHRAE Handbook—HVAC Systems and Equipment)



¹Liquid droplets containing bacteria, etc., sneezed, etc.

²Man-made mineral fibers

Fig. 17-18 Sizes of Indoor Particles
(Owen et al. 1992)
(Figure 3, Chapter 11, 2017 ASHRAE Handbook—Fundamentals)

610 mm). This type of filter is often used as a prefilter to higher efficiency filters.

Many different materials have been used as the filtering medium, including coarse (15 to 60 µm diameter) glass fibers, animal hair, vegetable fibers, synthetic fibers, metallic wools, expanded metals and foils, crimped screens, random-matted wire, and synthetic open-cell foams.

Although viscous impingement filters usually operate in the range of 300 to 600 fpm (1.5 to 3 m/s), they may operate at higher velocities. The limiting factor other than increased flow resistance is the danger of blowing off agglomerates of collected dust and the viscous coating on the filter.

The rate of filter loading depends on the type and concentration of the dirt being handled and on the operating cycle of the system. Manometers or draft gages are often installed to measure the pressure drop across the filter bank and thereby indicate when the filter needs servicing. The final allowable pressure drop may vary from one type of filter to another. The decline in filter efficiency that occurs when all the viscous coating has been absorbed by the collected dust, rather than the increased resistance due to dust load, may be the limiting factor in operating life.

Dry Air Filters. The media used in dry air filters are random fiber mats or blankets of varying thicknesses, fiber sizes, and densities. Media of bonded glass fiber, cellulose fibers, wool felt, synthetics, and other materials have been used commercially. The medium in filters of this class is frequently supported by a wire frame in the form of pockets or V-shaped pleats. In other designs, the media may be self-supporting because of inherent rigidity or because airflow inflates it into extended form. Pleating of the media provides a high ratio of media area to face area, thus allowing reason-

able pressure drop despite the density and fineness of the media.

The efficiency of dry air filters is usually higher than that of viscous impingement filters, and the variety of media available makes it possible to furnish almost any degree of cleaning efficiency desired. Most dry filter media and filter configurations also have higher dust-holding capacities than viscous impingement filters. Coarse prefilters placed ahead of high-efficiency dry filters may be economically justified by the even longer life they give the main filters.

Electronic Air Cleaners. These ionizing filters are efficient, low pressure drop devices for removing fine dust and smoke particles. Collector plates are often coated with a special oil as an adhesive. Cleaning is generally accomplished by washing the cells in place with hot water from a water hose or by a fixed or moving nozzle system.

Electrical forces drive most particles to the collecting surface but cannot hold them there. In fact, after a particle touches the collecting surface, the electrical force reverses and tends to pull it off, and the dust is held only by intermolecular adhesion forces. It is, therefore, very important with the washed type of electronic air cleaner to ensure that either the dust is naturally adherent or that the plates are always covered with adhesive.

Electronic air cleaners, however, are often used without any adhesive treatment on the plates. Under such conditions, the precipitator forms agglomerates that eventually blow off the plates. They must be followed downstream by a secondary filter or storage section. The dry agglomerates produced in the precipitator are allowed to blow off and be caught by the downstream filter. An automatic replaceable media filter for catching these agglomerates gives a combination with a high degree of cleaning efficiency and also the convenient maintenance associated with an automatic filter.

17.10 Air-to-Air Energy Recovery Equipment

Energy can be recovered from exhaust air, as well as transferred from one location to another. This can be done using rotary devices, heat pipe heat exchangers, coil heat recovery loops, twin tower heat recovery loops, fixed plate exchangers, and thermosiphon heat exchangers.

Table 17-2 provides comparative data for common types of air-to-air energy recovery devices. The following sections describe the construction, operation, and unique features of the various devices.

17.10.1 Performance Rating of Equipment

Performance of air-to-air heat exchangers is usually expressed in terms of their effectiveness in transferring (1) sensible energy (dry-bulb temperature), (2) latent energy (humidity ratio), or (3) total energy (enthalpy). The effectiveness ε of a heat exchanger is defined as follows:

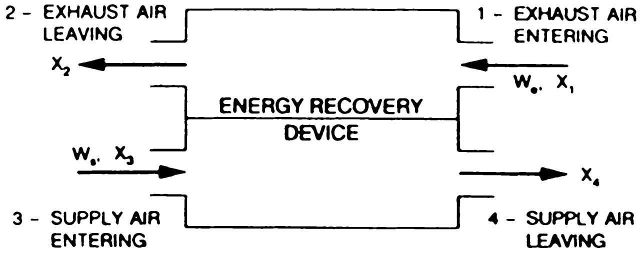


Fig. 17-20 Nomenclature for Effectiveness Evaluation

$$\varepsilon = \frac{\text{Actual transfer for given device}}{\text{Maximum possible transfer between airstreams}}$$

Referring to Figure 17-20,

$$\varepsilon = \frac{W_e(X_1 - X_2)}{W_{\min}(X_1 - X_3)} = \frac{W_s(X_4 - X_3)}{W_{\min}(X_1 - X_3)} \tag{17-6}$$

where

ε = sensible, latent, or total heat effectiveness
X = dry-bulb temperature, humidity ratio, or enthalpy at locations indicated in Figure 17-20

For the latent and total heat effectiveness,

W_s = mass flow rate of supply
W_e = mass flow rate of exhaust

For the sensible heat effectiveness,

W_s = (specific heat)(mass flow rate) for the supply
W_e = (specific heat)(mass flow rate) for the exhaust
W_{min} = smaller of W_s and W_e

The leaving supply air condition is then:

$$X_2 = X_1 - \varepsilon(X_1 - X_3)(W_{\min}/W_s) \tag{17-7}$$

The leaving exhaust air condition is:

$$X_4 = X_3 + \varepsilon(X_1 - X_3)(W_{\min}/W_e) \tag{17-8}$$

The effectiveness of a particular air-to-air energy recovery device is a function of several variables, including the supply and exhaust mass flow rates and the energy transfer characteristics of the device. Because of this combination, performance data must be established for each device.

17.10.2 Economics of Air-to-Air Energy Recovery

In analyzing the advisability of any air-to-air energy recovery application, one must usually consider both first and operating costs.

First Cost

- Energy recovery device
- Installing energy recovery device

Table 17-2 Comparison of Air-to-Air Energy Recovery Devices

(Table 3, Chapter 26, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

	Fixed Plate	Membrane Plate	Energy Wheel	Heat Wheel	Heat Pipe	Runaround Coil Loop	Thermosiphon	Liquid Desiccant
Airflow arrangements	Counterflow Cross flow[a]	Counterflow Cross flow[a]	Counterflow Parallel flow	Counterflow	Counterflow Parallel flow	—	Counterflow Parallel flow	—
Equipment size range, cfm	50 and up	50 and up	50 to 74,000 and up	50 to 74,000 and up	100 and up	100 and up	100 and up	—
Typical sensible effectiveness ($m_s = m_e$), %[c]	50 to 75	55 to 75	65 to 80	65 to 80	40 to 60[b]	45 to 65[b]	40 to 60	40 to 60[b]
Typical latent effectiveness,* %[c]	0	25 to 60	50 to 80	0	0	0	0	50 to 75[b,d]
Total effectiveness,* %[c]	20 to 50	35 to 70	55 to 80	25 to 60	15 to 35	—	—	40 to 75[d]
Face velocity, fpm	200 to 1000	200 to 600	500 to 1000	400 to 1000	400 to 800	300 to 600	400 to 800	300 to 450
Pressure drop, in. of water	0.4 to 4	0.4 to 2	0.4 to 1.2	0.4 to 1.2	0.6 to 2	0.6 to 2	0.6 to 2	0.7 to 1.2
EATR, %	0 to 2	0 to 5	0.5 to 10	0.5 to 10	0 to 1	0	0	0
OACF	0.97 to 1.06	0.97 to 1.06	0.99 to 1.1	1 to 1.2	0.99 to 1.01	1.0	1.0	1.0
Temperature range, °F	–75 to 1470	–40 to 140	–65 to 1470	–65 to 1470	–40 to 200	–50 to 930	–40 to 104	–40 to 115
Typical mode of purchase	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Coil only Complete system	Exchanger only Exchanger in case	Complete system
Advantages	No moving parts Low pressure drop Easily cleaned	No moving parts Low pressure drop Low air leakage Moisture/mass transfer	Moisture/mass transfer Compact large sizes Low pressure drop Available on all ventilation system platforms	Compact large sizes Low pressure drop Easily cleaned	No moving parts except tilt Fan location not critical Allowable pressure differential up to 2 psi	Exhaust airstream can be separated from supply air Fan location not critical	No moving parts Exhaust airstream can be separated from supply air Fan location not critical	Latent transfer from remote airstreams Efficient microbiological cleaning of both supply and exhaust airstreams
Limitations	Large size at higher flow rates	Few suppliers Long-term maintenance and performance unknown	Supply air may require some further cooling or heating Some EATR without purge	Some EATR with purge	Effectiveness limited by pressure drop and cost Few suppliers	Predicting performance requires accurate simulation model	Effectiveness may be limited by pressure drop and cost Few suppliers	Few suppliers Maintenance and performance unknown
Heat rate control (HRC) methods	Bypass dampers and ducting	Bypass dampers and ducting	Bypass dampers and wheel speed control	Bypass dampers and wheel speed control	Tilt angle down to 10% of maximum heat rate	Bypass valve or pump speed control	Control valve over full range	Control valve or pump speed control over full range

[a] Rated effectiveness values are for balanced flow conditions for cross flow. Effectiveness values increase slightly if flow rates of either or both airstreams are higher than flow rates at which testing is done.

[b] Data not based on third-party certified data.

[c] Data based on typical range of third-party certified data.

[d] Face velocity of 250 to 500 fpm.

EATR = exhaust air transfer ratio

OACF = outdoor air correction factor

- Additional cost of building space to accommodate apparatus
- Additional ductwork to accommodate device or cost of piping for liquid energy transfer
- Larger fans and/or motors to overcome air pressure loss of energy recovery device
- Additional air filtration required (if any)
- Capacity controls
- Any auxiliary heaters required for frost control
- Savings of boiler or heating equipment due to reduced design load
- Savings of heating coils and associated piping and pumps due to reduced design capacity
- Savings of chiller or cooling plant due to reduced design load
- Savings of cooling coils and associated piping and pumps due to reduced design capacity
- Savings of electric power requirement

Operating Cost

- Maintaining the energy recovery device
- Operating fans to overcome additional static pressure
- Maintaining additional filtration (if any)
- Operating energy recovery device drive, pumps, controls, and defrost heaters
- Savings of annual heating energy, based on weather data for the system location

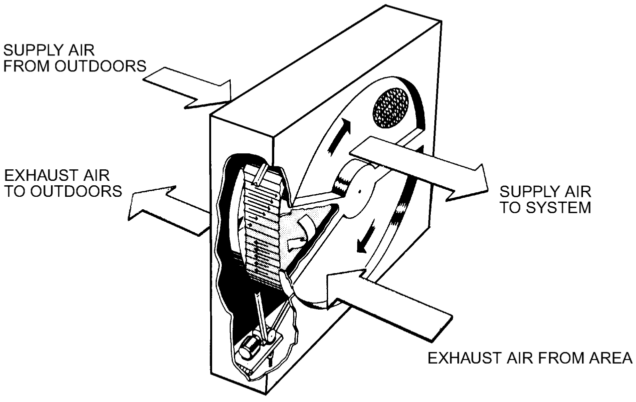


Fig. 17-21 Rotary Energy Wheel
(Figure 7, Chapter 26, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

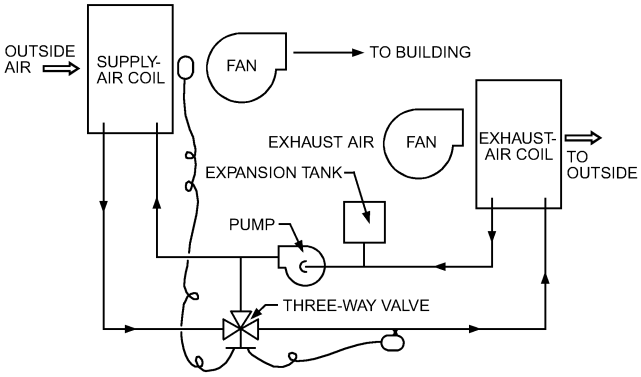


Fig. 17-22 Arrangement of Coil Energy Recovery Loop
(Figure 14, Chapter 26, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

- Savings of annual cooling energy, based on weather data for the system location

17.10.3 Rotary Air-to-Air Energy Exchangers

A rotary air-to-air energy exchanger, often called a *heat wheel*, is a revolving cylinder filled with an air-permeable medium with a large internal surface area for contact with the air passing through it. Adjacent supply and exhaust airstreams each flow through half the exchanger in a counterflow pattern (Figure 17-21). Media material may be selected to recover either sensible heat only or total heat (sensible heat plus latent heat). With total heat recovery, the unit is called an *energy wheel.*

Sensible heat is recovered (transferred) as the medium picks up and stores heat from the hot airstream and gives it up to the cold one. Latent heat is transferred as the medium (1) condenses moisture from the airstream having the higher humidity ratio (by means of absorption for liquid desiccants and adsorption for solid desiccants) with a simultaneous release of heat and (2) then releases the moisture through evaporation (and heat pickup) into the airstream with the lower humidity ratio. Thus, more moist air is dried while drier air is humidified. In total heat transfer, both sensible and latent heat recovery processes take place simultaneously.

Choice of structural material for the casing, rotor structure, and medium of a rotary energy exchanger is influenced by the contaminants, dew point, and temperature of the exhaust air, as well as by the properties of the supply air. Aluminum and steel are the usual casing and rotor materials for normal comfort ventilating systems. Exchanger media are fabricated from metal, mineral, or man-made materials and classified as providing either random flow or directionally oriented flow through their structures.

The performance of a rotary energy exchanger is defined by the exchanger's effectiveness and the media pressure drops. Face velocities for most energy recovery applications range from 500 to 800 fpm (2.5 to 4.0 m/s). Low face velocities give lower pressure drop, higher effectiveness, and lower operating costs, but require larger size units with higher capital costs and more installation space.

Typical pressure drops for various types of media at 500 fpm (2.5 m/s) vary from 0.4 to 0.7 in. (100 to 170 kPa) of water. (Consult the manufacturer's catalog in each case for actual data.) Average effectiveness values for sensible and total heat exchangers lie in the 70 to 85% range for equal supply and exhaust air mass flow rates and usual exchanger face velocities.

Rotary energy wheels are available in single units to 68,000 cfm (32 m^3/s) capacity; they are usually not larger than 14 ft (4.25 m) in diameter due to difficulty in shipping, erecting, and fitting into buildings. Multiple units may be used to provide greater single-system capacities. Units are available for temperatures from −70 to 1500°F (–60 to 800°C). When installed horizontally (vertical airflow), units greater than 8 ft (2.4 m) in diameter may require special structural considerations due to their size and weight.

17.10.4 Coil Energy Recovery Loops

The coil energy recovery or "runaround loop" system uses extended-surface, finned-tube water coils placed in the supply and exhaust airstreams of a building or process. The coils are connected in a closed loop via counterflow piping and an intermediate heat transfer fluid of water (typically) or a freeze-protected solution pumped through the coils.

This system allows energy to transfer from the warmer to the cooler airstream. In a typical comfort-to-comfort application, the system is seasonally reversible—the supply air preheats when outdoor air is cooler than the exhaust air and precools when the outdoor air is warmer. This system operates generally for sensible heat recovery only.

As with other air-to-air energy recovery equipment, measures must be taken to prevent potential freezing of exhaust air condensate. A dual-purpose, three-way temperature control valve is used to prevent exhaust coil frosting. The valve is controlled to maintain an entering solution temperature to the exhaust coil of not less than 30°F (–1°C). This is accomplished by bypassing some of the warmer solution from the supply air coil. The valve can also ensure that a prescribed leaving air temperature from the supply air coil is not exceeded for those applications where the energy recovered must be limited.

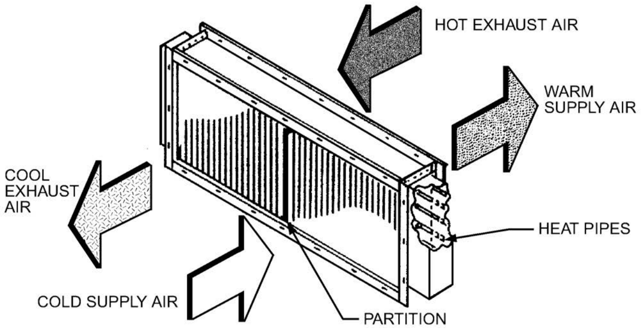


Fig. 17-23 Heat Pipe Assembly
(Figure 20, Chapter 26, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

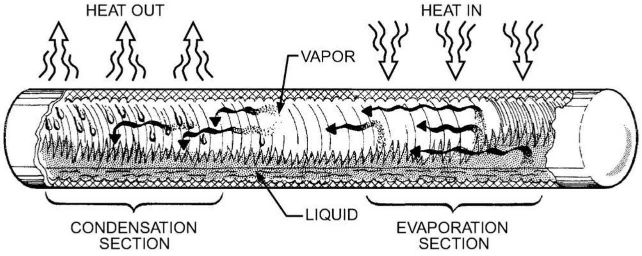


Fig. 17-24 Heat Pipe
(Figure 21, Chapter 26, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

This system affords a high degree of flexibility, which makes it well-suited for renovation and industrial applications. The system accommodates remotely located supply and exhaust ducts. It also allows simultaneous energy recovery from multiple supply and exhausts. A basic arrangement of the coil energy recovery loop is depicted in Figure 17-22.

17.10.5 Heat Pipe Heat Exchangers

A heat pipe heat exchanger has no moving parts; it is a passive energy recovery device. Although it appears similar to a standard steam or chilled water coil, it differs in two major aspects. As shown in Figure 17-23, each tube is an individual heat pipe, that operates independently and acts as a superconductor of heat. Secondly, the heat pipe heat exchanger is divided into two airflow paths. Hot air passes through one side of the exchanger and cold air through the other side in the opposite direction, that is, in a counterflow arrangement. Sensible energy from the hot air is transferred by the heat pipes to the other side of the exchanger, where it is captured by the cold air, thereby warming it. Although the heat pipes span the width of the unit, a sealed partition separates the two airstreams, preventing any cross-contamination between them.

Figure 17-24 is a schematic of a heat pipe. A heat pipe is a tube that is fabricated with a capillary wick structure, filled with a refrigerant or suitable change-of-phase heat transfer fluid, and permanently sealed.

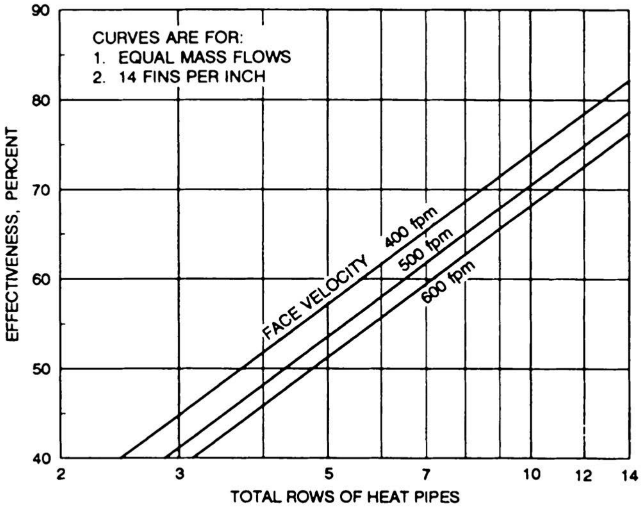


Fig. 17-25 Heat Pipe Exchanger Effectiveness
(Figure 22, Chapter 26, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Basically, the heat pipe operates on a condensation/evaporation cycle that is continuous as long as there is a temperature difference to drive the process. Energy transfer within a heat pipe is accomplished with a very small temperature drop; a heat pipe is essentially an isothermal device.

Thermal energy applied to either end of the pipe causes the fluid at the end to vaporize. The vapor then travels to the other end of the pipe where the removal of thermal energy causes the vapor to condense into liquid again, thus giving up the latent heat of condensation. The condensed liquid then flows back to the evaporator section (i.e., the hot end) to be reused, completing the cycle.

Heat pipes have a finite heat transfer capacity affected by such factors as wick design, tube diameter, working fluid, and tube orientation relative to horizontal. Design face velocities for heat pipe heat exchangers range from 400 to 800 fpm (2.0 to 4.1 m/s), with 450 to 550 fpm (2.3 to 2.8 m/s) most common. Design face velocities are generally established on allowable pressure drop rather than recovery performance.

Pressure drops at 60% effectiveness range from 0.4 to 0.7 in. of water at (100 to 170 Pa) 400 fpm (2.0 m/s) up to 1.5 to 2.0 in. of water (370 to 500 Pa) at 800 fpm (4.1 m/s). Recovery performance, or effectiveness, decreases with increasing velocity, but the effect is not as pronounced as with pressure drop.

Available fin designs include continuous corrugated plate fins, continuous flat plate fins, and spiral fins. These fin designs and tube spacing cause the variation in pressure drop, noted previously, at a given face velocity.

Figure 17-25 presents a typical effectiveness curve for various face velocities and rows of tubes. As the number of rows increases, effectiveness increases at a decreasing rate. For example, doubling the rows of tubes in a 60% effective heat exchanger increases the effectiveness to 75%.

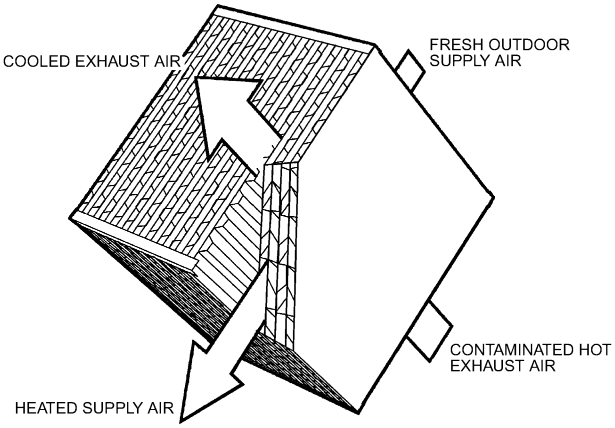


Fig. 17-26 Fixed Plate Cross-Flow Heat Exchanger
(Figure 4, Chapter 26, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

17.10.6 Fixed Plate Exchangers

The plate heat exchanger is a static device that has no leakage between airstreams. Since it uses no secondary heat transfer medium, such as water or refrigerant, its temperature range is the broadest of all air-to-air energy recovery equipment.

A fixed surface plate exchanger can be classified as (1) a **pure-plate heat exchanger**, consisting of only a primary heat transfer surface, or (2) a **plate-fin heat exchanger**, which is made up of alternate layers of separate plates and interconnecting fins. The pure-plate exchanger is usually a counterflow design, whereas the basic plate-fin exchanger is a crossflow design with combinations sometimes arranged to approach a counterflow unit. Counterflow provides the greatest temperature difference for maximum heat transfer, but crossflow can sometimes give more convenient air connections.

Fixed surface plate exchangers have no moving parts. Alternate layers of plates, separated and sealed (referred to as the heat exchanger core), form the exhaust and supply airstream passages (Figure 17-26). Plate spacings range from 0.1 to 0.5 in. (2.5 to 13 mm), depending on the design and application. Heat transfers directly from the warm airstreams through the separating plates into the cool airstreams.

Normally, both latent heat of condensation, from moisture condensed as the temperature of the warm (exhaust) airstream drops below its dew point, and sensible heat are conducted through the separating plates into the cool (supply) airstream. Thus, latent energy but not actual moisture may be transferred. Recovering upward of 80% of the available waste exhaust heat is not uncommon.

Fixed-plate heat exchangers can be made from permeable microporous membranes designed to maximize moisture and energy transfer between airstreams while minimizing air transfer. Suitable permeable microporous membranes for this emerging technology include cellulose, polymers, and other synthetic materials such as hydrophilic electrolyte. Hydrophilic electrolytes are made from sulphonation chemistry techniques and contain charged ions that attract polar water molecules; adsorption and desorption of water occur in vapor state.

Plate exchangers are of many proprietary designs, with weights, sizes, and flow patterns depending on the manufacturer (Figure 17-27). Most manufacturers of plate exchangers offer the equipment in modular design.

17.10.7 Thermosiphon Heat Exchangers

Thermosiphon heat exchangers use the natural gravity circulation of a boiling and condensing intermediate fluid to transfer energy between exhaust and supply airstreams. They may be classified as sealed tube thermosiphons and coil loop thermosiphons. These two types are illustrated in Figures 17-28 and 17-29.

The sealed tube type is similar to a heat pipe and is often given that name. The only distinction made between the two is that heat pipes are usually considered to use, if not solely rely on, capillary forces to cause the intermediate liquid to flow from the cold to the hot end of the tubes, whereas the thermosiphon tubes rely only on gravity. The coil loop type is similar in appearance to the coil energy recovery loop discussed previously. The most obvious difference is the absence of a circulating pump in the thermosiphon loop and the need for evaporator and condenser coils rather than single-phase liquid coils.

Example 17-2 Determine the leaving conditions for an energy wheel exchanger with the following conditions:

Cooling

1. Design conditions: outdoor air = 94°F (34.4°C) dry bulb, 77°F (25°C) wet bulb
2. Design conditions: space = 75°F (23.9°C) dry bulb, 62.5°F (16.9°C) wet bulb

Heating

1. Design conditions: outdoor air = 0°F (–17.8°) dry bulb, −3°F (–19.4°C) wet bulb
2. Design conditions: space = 75°F (23.9°C) dry bulb, 59.5°F (15.3°C) wet bulb

Cooling and Heating

Exchanger effectiveness = 80% on sensible heat and 80% on latent heat. Equal mass flow rates.

Solution:

Dry-bulb temperature and humidity ratio are calculated from Equation (17-8).

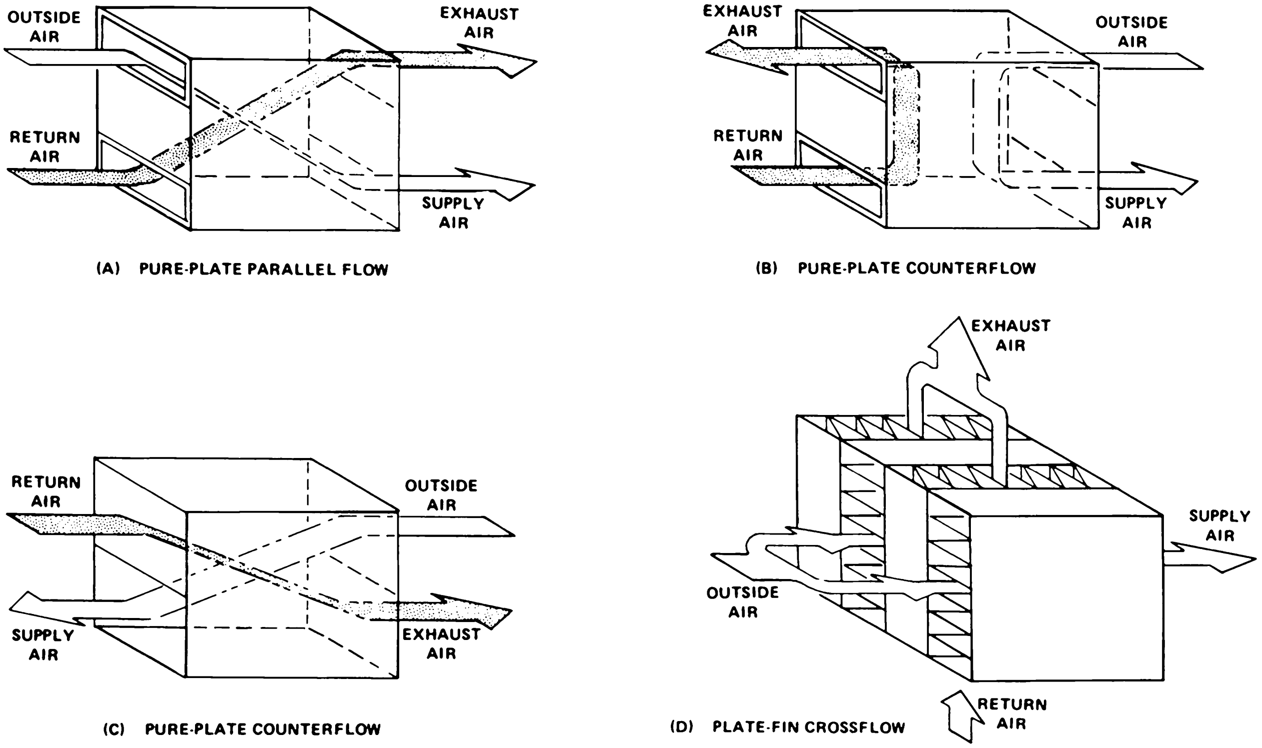
Summer dry-bulb temperature

$$X_2 = X_1 - \varepsilon(X_1 - X_2) = 94 - 0.8(94 - 75)$$
$$= 78.8°\text{F } (26.0°\text{C})$$

Summer humidity ratio

$$X_2 = X_1 - \varepsilon(X_1 - X_2) = 0.0162 - 0.8(0.0162 - 0.0092)$$
$$= 0.0106 \text{ lb/lb } (0.0106 \text{ kg/kg})$$

Winter dry-bulb temperature



(A) PURE-PLATE PARALLEL FLOW

(B) PURE-PLATE COUNTERFLOW

(C) PURE-PLATE COUNTERFLOW

(D) PLATE-FIN CROSSFLOW

Fig. 17-27 Pure-Plate and Plate-Fin Models

$$X_2 = X_1 - \varepsilon(X_1 - X_2) = 0 - 0.8(0 - 75)$$
$$= 60°\text{F dry bulb } (15.5°\text{C})$$

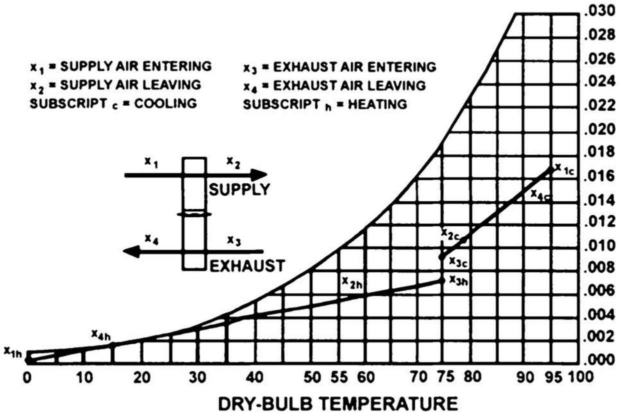
Winter humidity ratio

$$X_2 = X_1 - \varepsilon(X_1 - X_2) = 0.0001 - 0.8(0.0001 - 0.0074)$$
$$= 0.0059 \text{ lb/lb } (0.0059 \text{ kg/kg})$$

Example 17-3 Determine the leaving conditions for an energy wheel exchanger with the following comfort conditions:

Cooling

1. Design conditions. Outdoor air = 95°F (35°C) dry bulb, 78°F (25.6°C) wet bulb, and 41.3 Btu/lb (96.1 kJ/kg).
2. Design conditions. Space = 75°F (23.9°C) dry bulb, 62.5°F (17°C) wet bulb, and 28.3 Btu/lb (65.8 kJ/kg) of dry air.



Conditions for Example 17-2

Heating

1. Design conditions. Outdoor air = 20°F (–6.7°C) dry bulb and 5.0 Btu/lb (11.6 kJ/kg) of dry air.
2. Design conditions. Space = 75°F (23.9°C) dry bulb and 22.4 Btu/lb (52.1 kJ/kg) of dry air.

Cooling and Heating

Exchanger effectiveness = 80% on sensible heat and 65% on total heat. Equal mass flow rates.

Solution:

Dry-bulb temperature and enthalpy are calculated from Equation (17-8).

Summer dry-bulb temperature

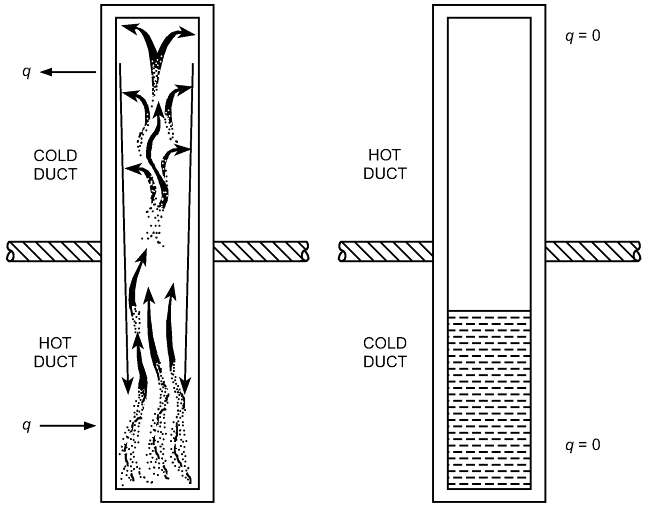
$$X_2 = X_1 - \varepsilon(X_1 - X_3) = 95 - 0.8(95 - 75)$$
$$= 79°\text{F } (26.1°\text{C})$$

Summer enthalpy

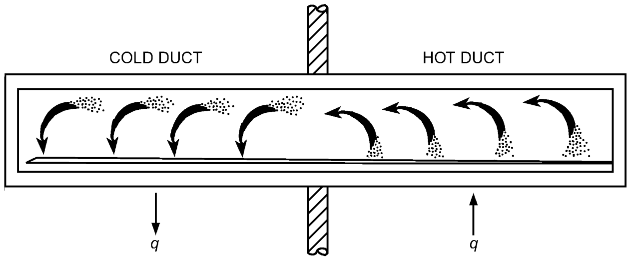
$$X_2 = X_1 - \varepsilon(X_1 - X_3) = 41.3 - 0.65(41.3 - 28.3)$$
$$= 32.9 \text{ Btu/lb } (76.5 \text{ kJ/kg})$$

Winter dry-bulb temperature

$$X_2 = X_1 - \varepsilon(X_1 - X_3) = 20 - 0.8(20 - 75)$$

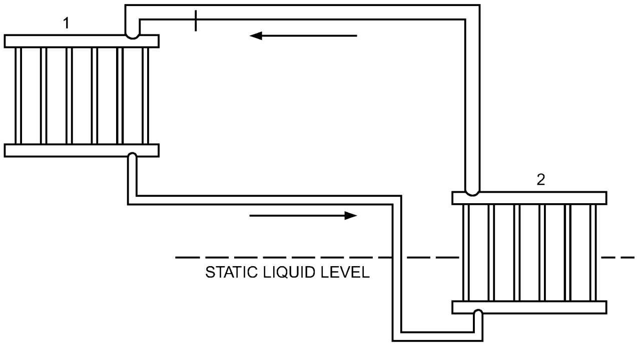


A. UNIDIRECTIONAL

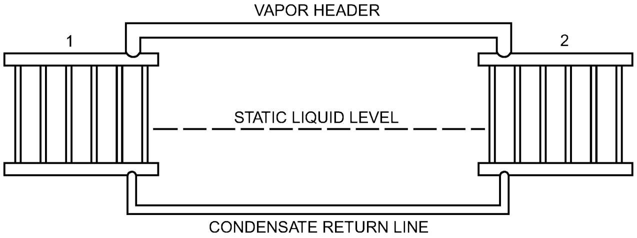


B. BIDIRECTIONAL
(TRANSFERS HEAT EQUALLY WELL IN EITHER DIRECTION)

Fig. 17-28 Sealed-Tube Thermosiphons



A. UNIDIRECTIONAL LOOP
(TRANSFERS ENERGY ONLY FROM 2 TO 1)



B. BIDIRECTIONAL LOOP
(TRANSFERS ENERGY IN EITHER DIRECTION)

Fig. 17-29 Coil-Type Thermosiphon Loops

= 64°F (17.8°C)

Heating enthalpy

$$X_2 = X_1 - \varepsilon(X_1 - X_3) = 0.5 - 0.65(5.0 - 22.4)$$
$$= 16.3 \text{ Btu/lb } (37.9 \text{ kJ/kg})$$

Example 17-4 Calculate recovered temperatures and volumes (measured at point 2), as well as energy savings, for a heat wheel exchanger handling 8000 cfm (3.8 m^3/s) of process exhaust at 300°F (149°C) measured at point 3, considered dry air. Assume an equal mass of makeup air at winter design conditions of 0°F (–17.8°C). Exchanger effectiveness is 80% at equal mass flows. Neglect cross leakage and purge volume.

Solution:

1. For conditions at 300°F (149°C)

$$8000 \text{ ft}^3\text{/min}/19.13 \text{ ft}^3\text{/lb} = 418 \text{ lb/min } (189.3 \text{ kg/min})$$

2. For conditions of 0°F (–17.8°C)

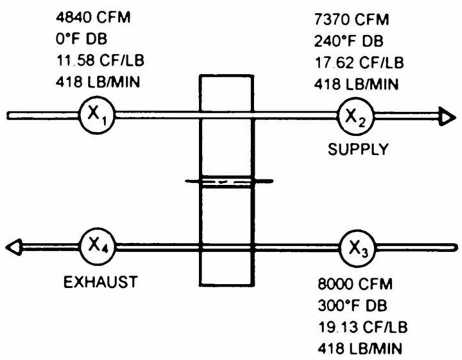
$$v_1 = 11.58 \text{ ft}^3\text{/lb}$$

$$418 \text{ lb/min} \times 11.58 \text{ ft}^3\text{/lb} = 4840 \text{ cfm at Point 1 } (2.28 \text{ m}^3\text{/s})$$

3. Temperature at Point 2 from Equation (17-8):

$$X_2 = X_1 - \varepsilon(X_1 - X_3) = 0 - 0.8(0 - 300)$$
$$= 240°\text{F dry bulb } (115.5°\text{C})$$

4. Specific volume at 240°F dry bulb (115.5°C)



Schematic and Conditions for Example 17-4

$$v_2 = 17.62 \text{ ft}^3\text{/lb}$$

$$418 \text{ lb/min} \times 17.62 \text{ ft}^3\text{/lb} = 7370 \text{ cfm } (3.5 \text{ m}^3\text{/s})$$

5. Savings at winter design conditions
[specific heat = 0.242 Btu/(lb·°F)]

$$418 \text{ lb/min } \times 60 \text{ min/h} \times 0.242 \text{ Btu/(lb·°F)} \times (240 - 0)$$
$$= 1{,}460{,}000 \text{ Btu/h } (428 \text{ kW})$$

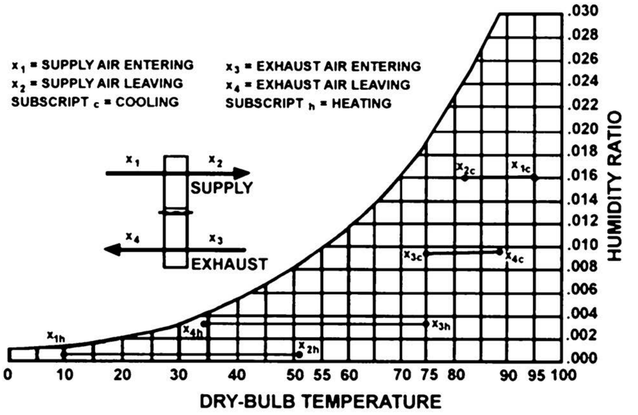
Example 17-5 Determine the leaving conditions for a heat pipe exchanger with the following comfort conditions:

Cooling

1. Design conditions, outdoor air = 95°F dry bulb (35°C)
2. Design conditions, space = 75°F dry bulb (23.9°C)

Heating

1. Design conditions, outdoors = 10°F dry bulb (–12.2°C)



Conditions for Example 17-5

2. Design conditions, space = 75°F dry bulb (23.9°C)

Cooling and Heating

Exchanger effectiveness = 65% on sensible heat at equal mass flows.

Solution:

Dry-bulb temperature is calculated from Equation (17-8).

Cooling dry-bulb temperature

$$X_2 = X_1 - \varepsilon(X_1 - X_3) = 95 - 0.65(95 - 75)$$
$$= 82°\text{F } (27.8°\text{C})$$

Heating dry-bulb temperature

$$X_2 = X_1 - \varepsilon(X_1 - X_3) = 10 - 0.65(10 - 75)$$
$$= 52.3°\text{F } (11.3°\text{C})$$

17.11 Economizers

An economizer uses outdoor air to reduce the refrigeration required to provide cooling for the building when the outdoor air dry-bulb temperature is low enough to provide for the sensible cooling needs or reduce the refrigeration requirements to do so. Either the air-side or the water-side economizer is an attractive option for reducing energy costs with self-contained HVAC systems. Although climate often dictates which economizer type is selected, either one can provide advantages.

The air-side economizer includes an outdoor air damper, relief damper, return air damper, filters, actuator, and linkage. Economizer controls are usually a factory-installed option. The air-side economizer takes advantage of cool outdoor air to either assist mechanical cooling or, if the outdoor air is sufficiently cool, provide total system cooling. However, if the building has significant simultaneous heating and cooling requirements, the interaction of the economizer with an installed heat recovery system must be thoroughly analyzed.

Self-contained units usually do not include return air fans. It is necessary to include a variable-volume relief fan unit when air-side economizers are employed. The relief fan volume is generally controlled with discharge dampers in response to building space pressure. The relief fan is off and discharge dampers are closed when the air-side economizer is inactive. However, the installed cost of an air-side economizer is generally higher than that for a water-side economizer.

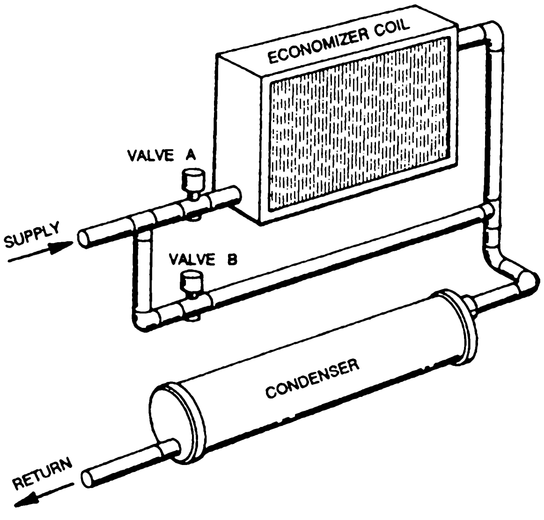


Fig. 17-30 Water-Side Economizer

Typically, in an air-side economizer, an enthalpy sensor or dry-bulb temperature probe energizes the unit to bring in outdoor air as the first stage of cooling. An outdoor air damper modulates the flow to meet a design temperature, and when outdoor air can no longer provide enough cooling, the refrigeration system is energized.

The water-side economizer consists of a water coil located upstream of the main cooling coil. All economizer control valves, piping between the economizer coil and the condenser, and economizer control wiring can be factory installed. The water-side economizer takes advantage of low cooling tower water temperature (approaching ambient wet-bulb temperature) to either precool the entering air, assist mechanical cooling, or, if the cooling water is cold enough, to provide total system cooling. If the economizer is unable to maintain the supply air setpoint for VAV units or zone set point for constant-volume units, factory-mounted controls integrate economizer and compressor operation to meet cooling requirements.

Cooling water flow rate is controlled by two valves (Figure 17-30), one at the economizer coil inlet (A) and one in the bypass loop to the condenser (B). Two control methods are common—constant water flow and variable water flow.

Standard modulating control allows constant condenser water flow during operation. The two control valves are wired for complementary operation, where one valve is drive open while the other is drive closed. Energy-saving modulating valve control allows variable condenser water flow during operation. The valve in the bypass loop (B) is

an on-off valve and is closed when the economizer is enabled. Water flow through the economizer coil is modulated by valve A. As the cooling load increases, valve A opens, increasing water flow through the economizer coil. If the economizer is unable to completely satisfy the cooling requirements, the control system integrate economizer and compressor operation. When the system is not in the cooling mode, both valves are closed.

17.12 Problems

17.1 Air enters a coil at 95°F dry-bulb and 78°F wet-bulb temperature and leaves at 62°F dry-bulb and 60°F wet-bulb temperature. The condensate is assumed to be at a temperature of 56°F. Find the total, latent, and sensible cooling loads on the coil with air at 14.7 psia.

17.2 Air enters a direct expansion coil at 85°F (29.4°C) dry bulb and 70°F (21.1°C) wet bulb and leaves at 62°F (16.7°C) dry bulb and 90% rh.

(a) How much sensible heat and how much latent heat is removed from the air by the coil?

(b) How much condensate drains off the coil?

17.3 Air enters a direct-expansion coil at 90°F (32.2°C) dry bulb 60% rh, and leaves the coil at 60°F (15.6°C) dry bulb, 95% rh. Find:

(a) heat removed from air

(b) moisture condensed from air

(c) SHR for the condition line

[Ans: (a) 19.2 Btu/lb (44.6 kJ/kg), (b) 0.008 lb/lb (0.008 kg/kg), (c) 0.45]

17.4 Water flowing at 60 lb/min and at 51°F is chilled in an evaporator to 40°F. The heat transfer area is 20 ft^2 and the heat exchanger has an overall heat transfer coefficient of 60 $Btu/h \cdot ft^2 \cdot °F$. The direct-expansion evaporator uses R-12 and operates at 35°F. Find the evaporator effectiveness.

17.5 Outdoor air at 35°F and 70% rh is supplied to an air-conditioning apparatus. Recirculated air is returned from the plant at 69°F dry bulb and 40% rh; 8100 cfm of outdoor air mixes with 18,900 cfm of recirculated air. The mixture is heated by a steam coil and humidified by a pan humidifier to final conditions of 115°F dry bulb and 20% rh.

(a) What steam flow, in pounds per hour, should be supplied to the heating coil?

(b) Estimate the steam consumption of the humidifier.

17.6 Outdoor air (8000 cfm) at 10°F dry bulb and 50% rh enters the central apparatus of a split heating system. It is tempered to 55°F dry bulb. Then, it flows through a spray humidifier where the leaving sump water is maintained at 50°F. The spray humidifier has a performance factor of 0.80. After leaving the humidifier, the air flows through a steam heating coil and is heated to 70°F dry bulb.

(a) What is the final relative humidity and humidity ratio of the air as it leaves the heating coil? [Ans: 40%]

(b) Assume steam at 2 psig and 90% quality is supplied to the tempering coil, the sump water heat exchanger, and the heating coil. How many pounds of steam per hour should be supplied to each? [Ans.: 170.7 lb]

17.7 The heat exchanger for the spray water in Problem 17.6 is out of service for maintenance. The split heating system is operating as specified except that the sump water is recirculated. Assume makeup water to the sump is 37°F and saturating effectiveness is equal to the performance factor.

(a) What is the final relative humidity and humidity ratio of the air leaving the heating coil?

(b) What is the steam rate (lb/h) for the tempering coils and for the heating coil?

17.8 Air at 105°F dry bulb (40.6°C) and 75% rh passes through a chilled water spray. Air leaves the spray chamber at 45°F dry bulb (7.2°C) saturated. How many grains of moisture per pound of entering air are condensed?

17.9 A building space is to be maintained at 70°F and 35% rh when outdoor design temperature is 10°F. Design heat losses from the space are 250,000 Btu/h, sensible, and 45,000 Btu/h, latent. Ventilation requires that 1500 cfm of outdoor air be used. Supply air is to be at 120°F. Determine:

(a) the amount of supply air required, lb/h and cfm

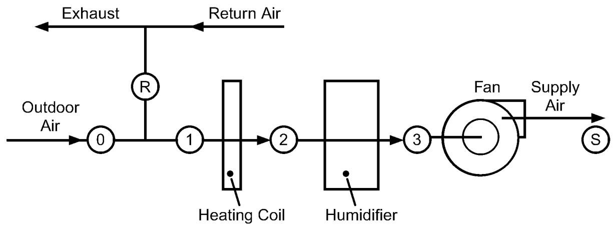
(b) the capacity of the heating coil, Btu/h, if:

1) the humidifier is a spray washer using recirculated spray water with makeup water provided at 60°F

2) the humidifier is a steam humidifier using dry, saturated steam at 17.2 psia

(c) the capacity of the humidifier, lb/h.

The conditioning equipment and nomenclature are shown in the following sketch.

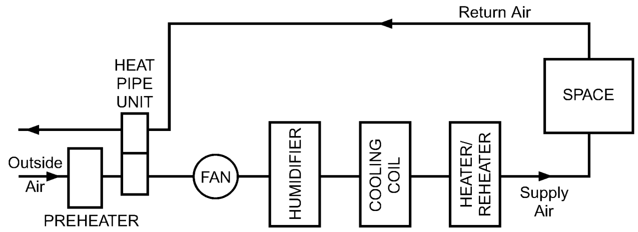


17.10 A spray-type air washer is to be used for humidification as well as cleaning of 9000 scfm of air. Inlet conditions to the washer are 75°F db and 48 F wb. Desired humidity ratio at outlet is 0.005 lb_w/lb_a. Determine (a) the necessary humidification efficiency of the washer, %, and (b) the makeup water requirements (humidifying capacity) of the unit, lb_w/h.

17.11 A heat pipe air-to-air energy recovery device is being considered for a system requiring 9000 scfm of outdoor air. Initially, a separate preheater was planned for bringing the outdoor air from its −2°F design ambient outdoor tempera-

ture to 40°F. Determine (a) the rating (Btu/h) and (b) the sensible effectiveness (%) to specify for the heat pipe unit if it is to eliminate the need for the air preheater. [Ans: 415,800 Btu/h, 57%]

17.12 The HVAC system for a hospital operating room, which requires 100% outdoor air, is shown in the following figure and includes an air-to-air heat pipe energy recovery unit having a sensible effectiveness of 73%. The air leaving the cooling coil is maintained at 58°F, 90% rh, all year long. During winter operation, air leaves the heater at 130°F. Fan speed is changed between summer and winter operation. Design duct system pressure drop (summer) is 3.25 in. water.



1. At winter design conditions (indoor: 72°F and 30% rh; outdoor: 5°F and 100% rh) the space load is 235,000 Btu/h (sensible) with negligible latent load. Determine (a) the necessary size of heating unit (Btu/h) both with and without the energy recovery unit and (b) the humidifier size (gal/day). Neglect fan effects.
2. At summer design conditions (indoor: 78°F; outdoor: 95°F db/76°F wb), the space cooling loads are 146,000 Btu/h (sensible) and 79,000 Btu/h (latent). Determine (a) fan size (hp and scfm), (b) sensible coil load, Btu/h, (c) latent coil load, Btu/h, and (d) necessary size of cooling unit, Btu/h, both with and without the energy recovery unit. Include fan effects.

17.13 Bibliography

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ASHRAE. 2016. *2016 ASHRAE Handbook—HVAC Systems and Equipment*.

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SI Table

Table 17-2 SI Comparison of Air-to-Air Energy Recovery Devices

(Table 3, Chapter 26, 2012 ASHRAE Handbook—HVAC Systems and Equipment, SI Version)

	Fixed Plate	Membrane Plate	Energy Wheel	Heat Wheel	Heat Pipe	Runaround Coil Loop	Thermosiphon	Twin Towers
Airflow arrangements	Counterflow Cross-flow	Counterflow Cross-flow	Counterflow Parallel flow	Counterflow	Counterflow Parallel flow	—	Counterflow Parallel flow	—
Equipment size range, cfm	50 and up	50 and up	50 to 74,000 and up	50 to 74,000 and up	100 and up	100 and up	100 and up	—
Typical sensible effectiveness ($m_s = m_e$), %	50 to 80	50 to 75	50 to 85	50 to 85	45 to 65	45 to 65	40 to 60	40 to 60
Typical latent effectiveness,* %	—	50 to 72	50 to 85	0	—	—	—	—
Total effectiveness,* %	—	50 to 73	50 to 85	—	—	—	—	—
Face velocity, fpm	200 to 1000	200 to 600	500 to 1000	400 to 1000	400 to 800	300 to 600	400 to 800	300 to 450
Pressure drop, in. of water	0.4 to 4	0.4 to 2	0.4 to 1.2	0.4 to 1.2	0.6 to 2	0.6 to 2	0.6 to 2	0.7 to 1.2
EATR, %	0 to 2	0 to 5	0.5 to 10	0.5 to 10	0 to 1	0	0	0
OACF	0.97 to 1.06	0.97 to 1.06	0.99 to 1.1	1 to 1.2	0.99 to 1.01	1.0	1.0	1.0
Temperature range, °F	–75 to 1470	–40 to 120	–65 to 1470	–65 to 1470	–40 to 104	–50 to 930	–40 to 104	–40 to 115
Typical mode of purchase	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Coil only Complete system	Exchanger only Exchanger in case	Complete system
Advantages	No moving parts Low pressure drop Easily cleaned	No moving parts Low pressure drop Low air leakage Moisture/mass transfer	Moisture/mass transfer Compact large sizes Low pressure drop Available on all ventilation system platforms	Compact large sizes Low pressure drop Easily cleaned	No moving parts except tilt Fan location not critical Allowable pressure differential up to 2 psi	Exhaust airstream can be separated from supply air Fan location not critical	No moving parts Exhaust airstream can be separated from supply air Fan location not critical	Latent transfer from remote airstreams Efficient microbiological cleaning of both supply and exhaust airstreams
Limitations	Large size at higher flow rates	Few suppliers Long-term maintenance and performance unknown	Supply air may require some further cooling or heating Some EATR without purge	Some EATR with purge	Effectiveness limited by pressure drop and cost Few suppliers	Predicting performance requires accurate simulation model	Effectiveness may be limited by pressure drop and cost Few suppliers	Few suppliers Maintenance and performance unknown
Heat rate control (HRC) methods	Bypass dampers and ducting	Bypass dampers and ducting	Bypass dampers and wheel speed control	Bypass dampers and wheel speed control	Tilt angle down to 10% of maximum heat rate	Bypass valve or pump speed control	Control valve over full range	Control valve or pump speed control over full range

*Rated effectiveness values are for balanced flow conditions. Effectiveness values increase slightly if flow rates of either or both airstreams are higher than flow rates at which testing is done.

EATR = exhaust air transfer ratio
OACF = outdoor air correction factor

Chapter 18

REFRIGERATION EQUIPMENT

This chapter provides a relatively brief treatment of the systems and components used for providing the cooling requirements of building HVAC systems. Primary topics are vapor compression refrigeration, absorption refrigeration, and cooling towers. Additional information can be obtained from the 2014 *ASHRAE Handbook—Refrigeration* and the 2016 *ASHRAE Handbook—HVAC Systems and Equipment.*

18.1 Mechanical Vapor Compression

The basic components of the mechanical vapor compression cycle are the compressor, condenser, expansion device, and evaporator (Figure 18-1). The basic principles of the vapor compression cycle are detailed in Chapter 2 of this book. Additional information is also provided in Chapter 2 of the 2017 *ASHRAE Handbook—Fundamentals*.

18.1.1 Compressors

The compressor is one of the essential parts of the compression refrigeration system and serves both to provide the necessary increase in pressure of the refrigerant vapor and as a refrigerant pump to circulate the refrigerant through the system in a continuous cycle.

There are two basic types of compressors: positive displacement and dynamic. Positive-displacement compressors increase the pressure of refrigerant vapor by reducing the volume of the compressor chamber through work applied to the compressor's mechanism. This class of compressor includes reciprocating, rolling piston, rotary vane, single screw, double screw, trochoidal, and scroll. Dynamic compressors increase the pressure of refrigerant vapor by a continuous transfer of angular momentum from the rotating member to the vapor followed by the conversion of this momentum into a pressure rise. Centrifugal compressors function based on these principles.

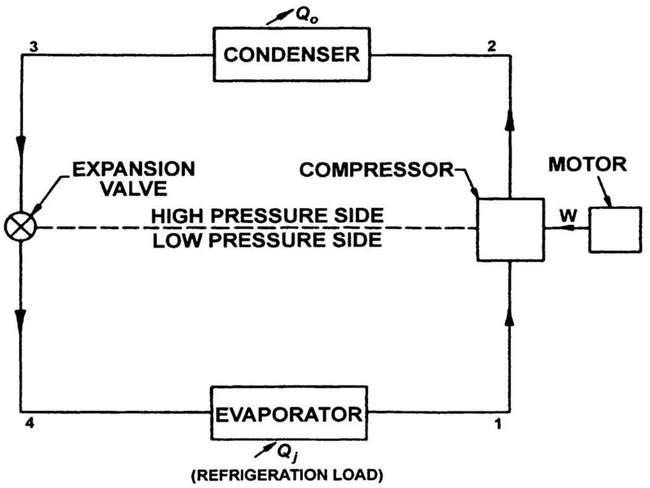


Fig. 18-1 Simplified Equipment Diagram for the Basic Vapor-Compression Cycle

Compressor performance is the result of design constraints involving physical limitations of the refrigerant, compressor, and motor, while attempting to provide the following:

- Greatest trouble-free life expectancy
- Most refrigeration effect for the least power input
- Lowest applied cost
- Wide range of operating conditions
- Acceptable vibration and sound level

Two useful measures of compressor performance are capacity (which is related to compressor volume displacement) and efficiency. Compressor refrigerating capacity is the rate of heat removal by the refrigerant pumped by the compressor in a refrigerating system at the evaporator. Capacity equals the product of the mass flow rate of refrigerant pumped by the compressor and the difference in specific enthalpies of the refrigerant when it leaves the evaporator and when it enters the evaporator.

Reciprocating Compressors. Most reciprocating compressors are single acting, using pistons driven directly through a pin and connecting rod from the crankshaft. Double-acting compressors are not extensively used.

The halocarbon compressor is the most widely used and is manufactured in three designs: (1) open, (2) semihermetic or bolted hermetic, and (3) welded-shell hermetic. Ammonia compressors are manufactured only in the open design, in which the driveshaft extends through a seal in the crankcase for an external drive.

In hermetic compressors, the motor and compressor are contained within the same pressure vessel; the motor shaft is integral with the compressor crankshaft, and the motor is in contact with the refrigerant. A hermetic compressor is shown in Figure 18-2. A semi-, bolted, accessible, or serviceable hermetic compressor is bolted together and may be repaired in the field. The motor compressor in a welded shell (sealed) hermetic compressor is mounted inside a steel shell, which in turn is sealed by welding. Table 18-1 shows combinations of common design features and Table 18-2 gives typical performance values for halocarbon refrigerant compressors.

Capacity data are given in Figure 18-3, which is a typical set of curves for a four-cylinder semi-hermetic compressor,

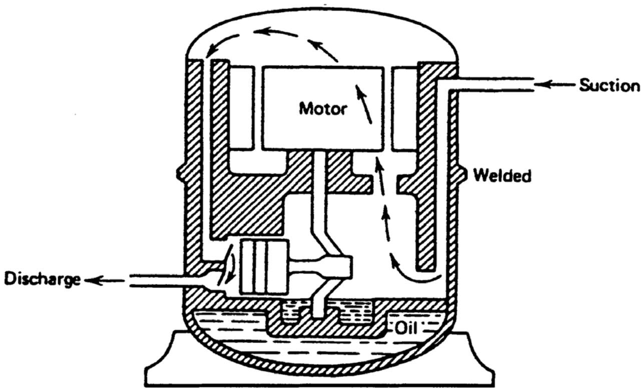


Fig. 18-2 Hermetic Compressor

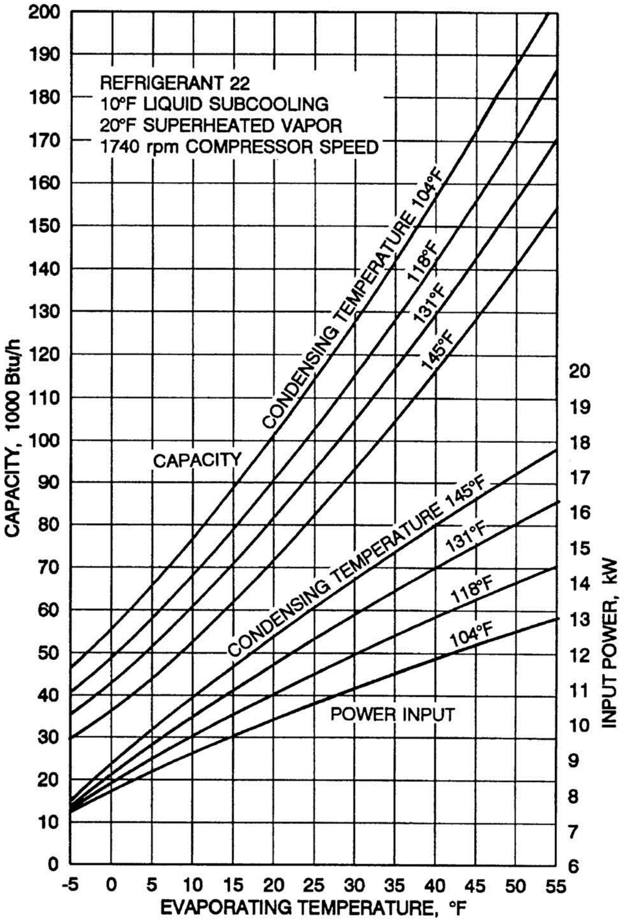


Fig. 18-3 Typical Capacity and Power for Reciprocating Compressor
(Figure 10, Chapter 38, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

2 3/8 in. (60 mm) bore, 1 3/4 in. (44 mm) stroke, 1740 rpm, operating with R-22. A set of power curves for the same compressor is also shown.

Reciprocating compressors are most commonly used for systems in the range of 0.5 to 100 tons (2 to 350 kW) and larger. They are used in unitary heat pumps and, in most cases, are either fully or accessibly hermetic.

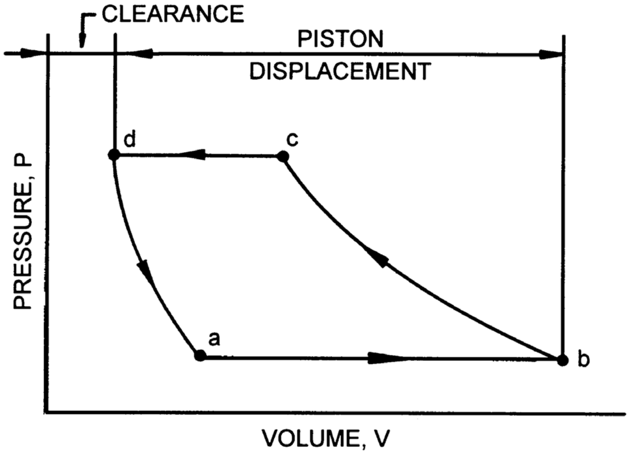


Fig. 18-4 Cycle for Idealized Piston Compressor

One of the important thermodynamic considerations for this compressor is the effect of the clearance volume (i.e., the volume occupied by the refrigerant within the compressor that is not displaced by the moving member). The effect is illustrated, in the case of the piston-type compressor, by considering the clearance volume between the piston and the cylinder head when the piston is at top dead-center position. The clearance gas remaining in this space after the compressed gas is discharged from the cylinder reexpands as the piston moves downward, preventing a fresh charge into the cylinder until the pressure falls to the inlet (suction) pressure (see Figure 18-4). As a consequence, the volume (and mass) of refrigerant entering the cylinder is less than the volume swept by the piston. This effect is quantitatively expressed by the volumetric efficiency e_v as

$$\frac{e_v}{100} = \frac{m_a}{m_i}$$

where

m_a = actual mass of new gas entering the compressor
m_i = theoretical mass, equal to piston displacement divided by specific volume of refrigerant vapor at suction conditions

The volumetric efficiency due only to reexpansion of the clearance volume gas can be calculated as follows:

$$\frac{e_v}{100} = 1 + C\left(\frac{v_s}{v_d}\right)$$

where

C = clearance ratio = $(V_b - V_a)/(V_b - V_d)$
v_s = specific volume of refrigerant at suction conditions
v_d = specific volume of refrigerant at discharge conditions
V_a, V_b, and V_d = the volumes at the locations given in Figure 18-4

Table 18-1 Typical Design Features of Reciprocating Compressors

	Refrigerant Type			
	Halo-, Fluoro-, or Hydrocarbon			Ammonia
Item	Open	Semi-hermetic	Welded Hermetic	Open
1. Number of cylinders: one to	16	12	6	16
2. Power range	0.17 hp125 W	0.50.35 to	0.170.12 to	10 hp7.5 kW
	and up	150 hp110 kW	25 hp20 kW	and up
3. Cylinder arrangement				
a. Vertical, V or W, radial	X	X		
b. Radial, horizontal opposed			X	
c. Horizontal, vertical V or W		X		X
4. Drive				
a. Electric motor		X	X	
b. Direct drive, V belt chain, gear, by electric motor or engine	X			X
5. Lubrication: splash or force feed, flooded	X	X	X	X
6. Suction and discharge valves: ring plate or ring or reed flexing	X	X	X	X
7. Suction and discharge valve arrangement				
a. Suction and discharge valves in head	X	X	X	X
b. Uniflow: suction valves in top of piston, suction gas entering through cylinder walls; discharge valves in head	X			X
8. Cylinder cooling				
a. Suction-gas-cooled	X	X	X	X
b. Water jacket cylinder wall, head, or cylinder wall and head	X			X
c. Air-cooled	X	X	X	X
d. Refrigerant-cooled heads	X			X
9. Cylinder head				
a. Spring-loaded	X	X	X	X
b. Bolted	X	X	X	X
10. Bearings				
a. Sleeve, antifriction	X	X	X	X
b. Tapered roller	X			X
11. Capacity control, if provided: manual or automatic				
a. Suction valve lifting	X	X	X	X
b. Bypass-cylinder heads to suction	X	X	X	X
c. Closing inlet	X	X		X
d. Adjustable clearance	X	X		X
e. Variable-speed	X	X	X	X
12. Materials				
Motor insulations and rubber materials must be compatible with refrigerant and lubricant mixtures; otherwise, no restrictions		X	X	
No copper or brass				X
13. Lubricant return				
a. Crankcase separated from suction manifolds, oil return check valves, equalizers, spinners, foam breakers	X	X		X
b. Crankcase common with suction manifold			X	
14. Synchronous fixed speeds, rpm	250 to 3600	1500 to 3600	1500 to 3600	250 to 1500
15. Pistons				
a. Aluminum or cast iron	X	X	X	X
b. Ringless	X	X	X	X
c. Compression and oil-control rings	X	X	X	X
16. Connecting rod				
Split rod with removable cap or solid eccentric strap	X	X	X	X
17. Mounting				
Internal spring mount		X	X	
External spring mount		X	X	
Rigidly mounted on base	X	X		X

The actual volumetric efficiency is affected by other factors such as cylinder wall heating due to friction and pressure drops through the inlet and discharge valves and is best obtained by actual laboratory measurements of the amount of refrigerant compressed and delivered by the compressor. The difference between actual and predicted volumetric efficiency, considering only clearance volume effects, is illustrated in Figure 18-5.

Rotary Compressors. Rotary compressors operate with a circular, or rotary, motion instead of reciprocating motion. Their positive-displacement compression process is nonreversing and either continuous or cyclical, depending on the type of mechanism. Most are direct drive machines.

The rolling piston rotary compressor is shown in Figure 18-6; the rotary vane type is shown in Figure 18-7. These two machines are similar in size, weight, thermodynamic

Table 18-2 Typical Performance Values

	Operating Conditions and Refrigerants			
	R-404a	R-134a	R-22	R-22
Compressor Size and Type	Evap. Temp. = –40°F Cond. Temp. = 105°F Suction Gas = 65°F Subcooling = 0°F	Evap. Temp. = 0°F Cond. Temp. = 110°F Suction Gas = 65°F Subcooling = 0°F	Evap. Temp. = 40°F Cond. Temp. = 105°F Suction Gas = 55°F Subcooling = 0°F	Evap. Temp. = 45°F Cond. Temp. = 130°F Suction Gas = 65°F Subcooling = 0°F
Large, over 25 hp				
Open	0.21 tons/hp	0.40 tons/hp	1.05 tons/hp	1.07 tons/hp
Hermetic	3.15 Btu/h per W	6.00 Btu/h per W	14.2 Btu/h per W	10.4 Btu/h per W
Medium, 5 to 25 hp				
Open	0.19 tons/hp	0.37 tons/hp	1.00 ton/hp	1.00 tons/hp
Hermetic	2.89 Btu/h per W	5.60 Btu/h per W	14.0 Btu/h per W	10.2 Btu/h per W
Small, under 5 hp				
Open	—	—	—	—
Hermetic	—	3.80 Btu/h per W	13.8 Btu/h per W	10.0 Btu/h per W

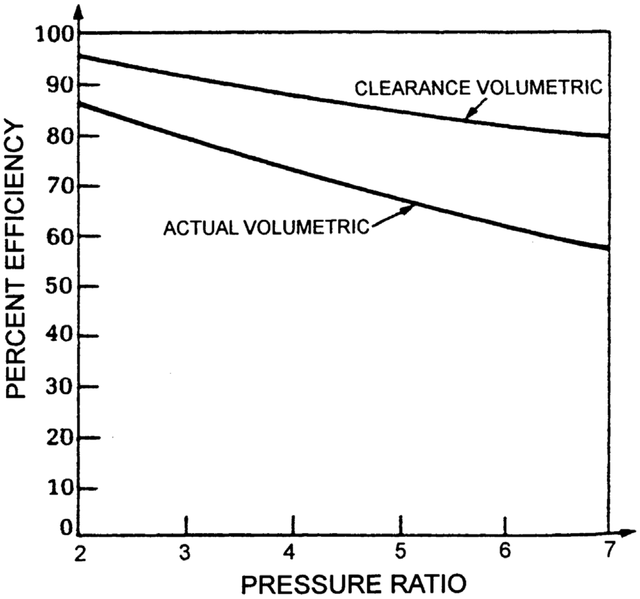


Fig. 18-5 Volumetric Efficiency

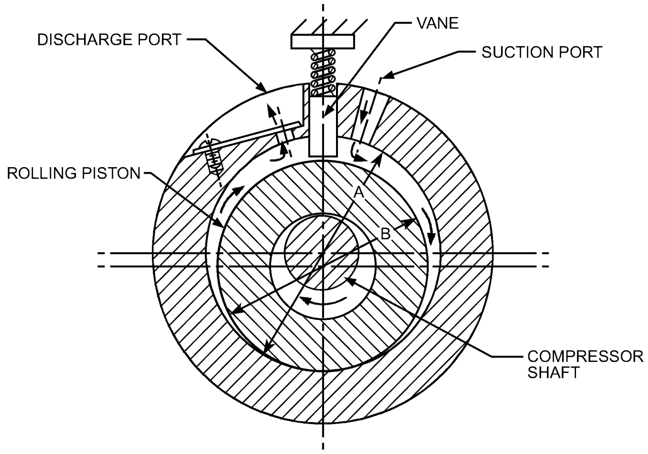


Fig. 18-6 Fixed-Vane Rolling Piston Rotary Compressor
(Figure 13, Chapter 38, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

performance, field of applications, range of capacities, durability, and sound level.

Internal leakage is controlled in rotary compressors through hydrodynamic sealing; thus, precision fits and optimum clearance are design requirements. The hydrodynamic sealing depends on clearances, surface speed, oil viscosity, and surface finish of the parts. Smoother finishes and closer clearances are used with low-viscosity oil in small machines. Larger machines have greater clearances and usually use a higher-viscosity oil.

Rotary compressor performance is characterized by high volumetric efficiency due to the small clearance volume and by correspondingly low reexpansion loss.

The **rolling piston compressor** uses a roller mounted on an eccentric shaft. A single vane or blade positioned in the nonrotating cylindrical housing reciprocates as the eccentrically moving roller turns. Rolling piston compressors are used in household refrigerators and air-conditioning units in sizes up to about 3 hp (2 kW).

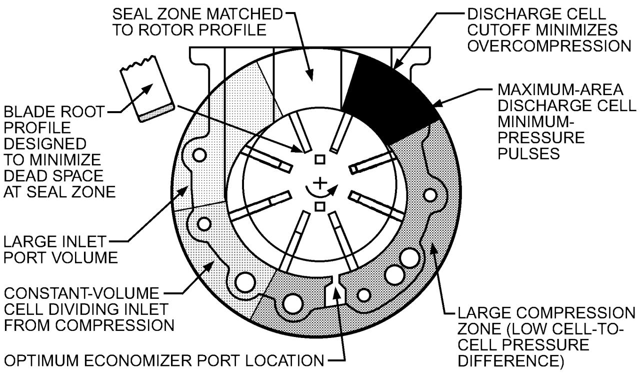


Fig. 18-7 Rotary Vane Compressor
(Figure 16, Chapter 38, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Displacement for this compressor can be calculated from the following equation:

$$V_d = \frac{\pi H(A^2 - B^2)}{4}$$

where

V_d = displacement
H = cylinder block height
A = cylinder diameter
B = roller diameter

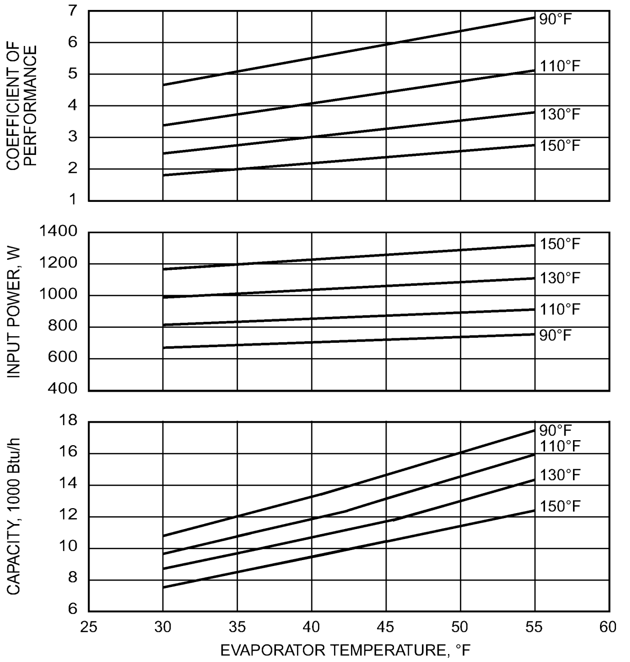
Suction gas is directly piped into the suction port of the compressor, and the compressed gas is discharged into the compressor housing shell. This high-side shell design is used because of the simplicity of its lubrication system and the absence of oiling and compressor cooling problems. Compressor performance is also improved because this arrangement minimizes heat transfer to the suction gas and reduces gas leakage areas.

The performance typical of rolling piston compressors is illustrated in Figure 18-8.

The **rotating vane compressor** has a rotor concentric with the shaft, with vanes in the rotor; this assembly is off center with respect to the cylindrical housing. An oval shaped bore produces a double lobe or a two-cylinder compressor. Rotary vane compressors have a low weight-to-displacement ratio, which, in combination with their compact size, makes them suitable for transport applications. Small compressors in the 3 to 50 hp (2 to 40 kW) range are single staged, for a saturated suction temperature range of –40 to 45°F (–40 to 7°C) at saturated condensing temperatures of up to 140°F (60°C). By employing a second stage, low-temperature applications down to –60°F (–50°C) are possible. Currently, R-22, R-502, and R-717 refrigerants are used.

Screw Compressors. The helical rotary compressor, or the screw compressor, belongs to the class of positive-displacement compressors. Screw compressors currently in production for refrigeration and air-conditioning applications comprise two distinct types: single screw and twin screw. Both are conventionally used in the fluid injection mode where sufficient fluid cools and seals the compressor. Single-screw compressors have the capability to operate at pressure ratios above 20:1 single stage. The capacity range currently available is from 20 to 1300 tons (70 to 4600 kW).

The single-screw compressor consists of a single cylindrical main rotor that works with a pair of gaterotors. Both the main rotor and gaterotors can vary widely in terms of form and mutual geometry. Figure 18-9 shows the design normally encountered in refrigeration.



R-22
3450 rpm 95°F AMBIENT 65°F SUCTION GAS 15°F LIQUID SUBCOOLING
CONDENSING TEMPERATURES SHOWN ON EACH CURVE

Fig. 18-8 Typical Rolling Piston Compressor Performance
(Figure 14, Chapter 38, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

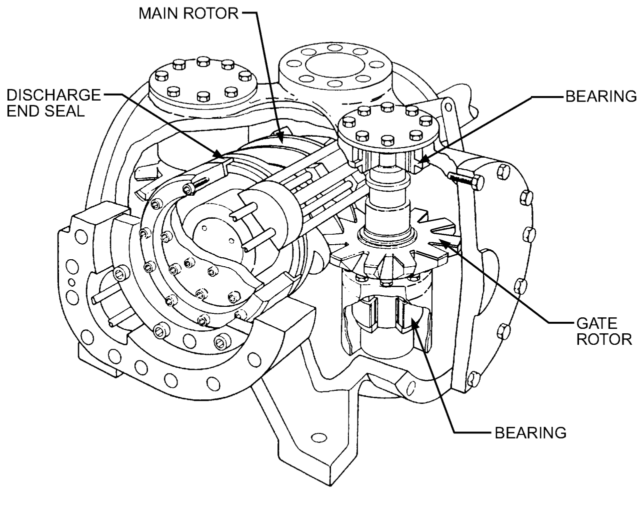


Fig. 18-9 Section of Single-Screw Compressor
(Figure 17, Chapter 38, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

The main rotor has six helical grooves, with a cylindrical periphery and a globoid (or hourglass shape) root profile. The two identical gaterotors each have 11 teeth and are located on opposite sides of the main rotor. The casing enclosing the main rotor has two slots, which allow the teeth of the gaterotors to pass through them. Two diametrically opposed discharge ports use a common discharge manifold located in the casting. The compressor is driven through the main rotor shaft, and the gaterotors follow by direct meshing action at 6:11 ratio of the main rotor speed. The geometry of the single-screw compressor is such that 100% of the gas compression power is transferred directly from the main rotor to the gas. No power (other than small frictional losses) is transferred across the meshing points to the gaterotors.

Compression is obtained by direct volume reduction with pure rotary motion as illustrated in Figure 18-10.

The four basic continuous phases of the working cycle are as follows:

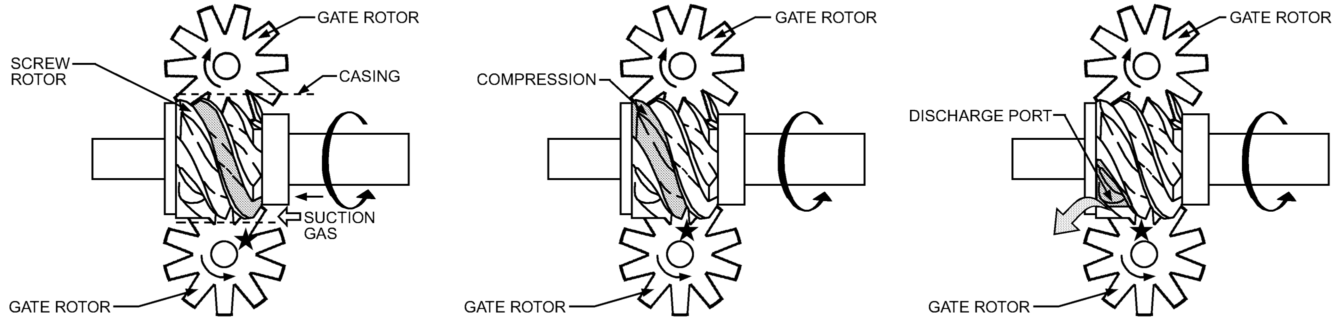
Suction. As a lobe of the male rotor begins to unmesh from an interlobe space in the female rotor, a void is created and gas is drawn in through the inlet port. As the rotors continue to turn, the interlobe space increases in size and gas flows continuously into the compressor. Prior to the point at which the interlobe space leaves the inlet port, the entire length of the interlobe space is completely filled with gas.

Transfer. As rotation continues, the trapped gas pocket in the interlobe space is moved circumferentially around the compressor housing at constant suction pressure.

Compression. Further rotation starts meshing of another male lobe with the female interlobe space on the suction end and progressively squeezes (compresses) the gas in the direction of the discharge port. Thus, the occupied volume of the trapped gas within the interlobe space is decreased and the gas pressure consequently increased.

Discharge. At a point determined by the design built-in volume ratio, the discharge port is uncovered and the compressed gas is discharged by further meshing of the lobe and interlobe space.

During the remeshing period of compression and discharge, a fresh charge is drawn through the inlet on the opposite side of the meshing point. With four male lobes rotating at 3600 rpm, four interlobe volumes are filled and discharged per revolution, providing 14,400 discharges per minute or



Suction. During rotation of the main rotor, a typical groove in open communication with the suction chamber gradually fills with suction gas. The tooth of the gaterotor in mesh with the groove acts as an aspirating piston.

Compression. As the main rotor turns, the groove engages a tooth on the gaterotor and is covered simultaneously by the cylindrical main rotor casing. The gas is trapped in the space formed by the three sides of the groove, the casing, and the gaterotor tooth. As rotation continues, the groove volume decreases and compression occurs.

Discharge. At the geometrically fixed point where the leading edge of the groove and the edge of the discharge port coincide, compression ceases, and the gas discharges into the delivery line until the groove volume has been reduced to zero.

Fig. 18-10 Sequence of Compression Process in Single-Screw Compressor
(Figure 18, Chapter 38, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

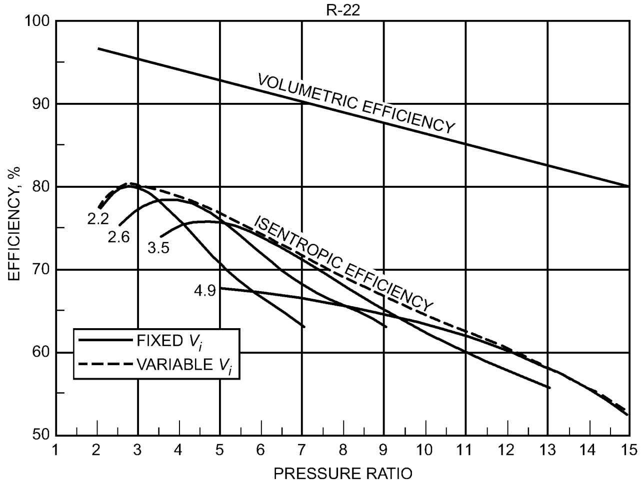


Fig. 18-11 Typical Screw Compressor Performance with R-22
(Figure 28, Chapter 38, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

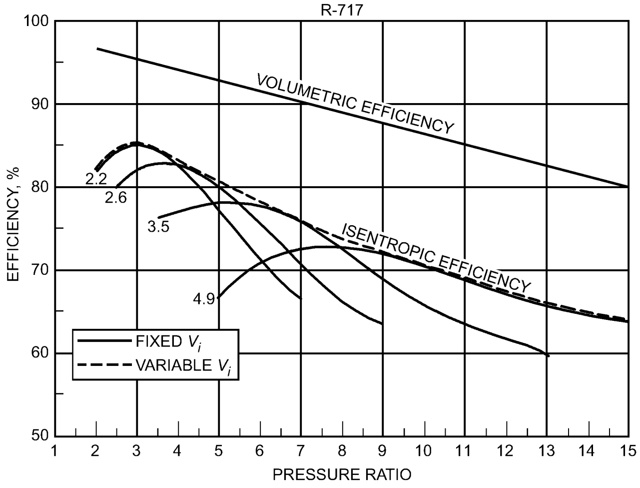


Fig. 18-12 Typical Screw Compressor Performance with R-717 (Ammonia)
(Figure 29, Chapter 38, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

240 per second. Since the intake and discharge cycles overlap effectively, a smooth, continuous flow of gas results.

Figures 18-11 and 18-12 show typical efficiencies of all single-screw compressor designs. High isentropic and volumetric efficiencies are the result of internal compression, the absence of suction and discharge valves and their losses, and extremely small clearance volumes. The curves show the importance of selecting the correct volume ratio in fixed volume ratio compressors.

Twin screw is a common designation for double helical rotary screw compressors. A twin-screw compressor consists of two mating helically grooved rotors—male (lobes) and female (flutes or gullies) in a stationary housing with inlet and outlet gas ports (Figure 18-13).

While operating, some twin-screw compressors adjust the volume ratio of the compressor to the most efficient ratio for whatever system pressures are encountered. The comparative efficiencies of fixed and variable volume ratio screw compressors are shown in Figure 18-14 for full-load operation on ammonia and R-22 refrigerants. The greater the change in either suction or condensing pressure a given system experiences, the more benefits are possible with a variable volume ratio. Efficiency improvements as high as 30% are possible, depending on the application, refrigerant, and system operating range. Hermetic screw compressors are commercially available through 400 tons (1.4 MW) of refrigeration using R-22.

Scroll compressors are rotary motion, positive-displacement machines that compress with two interfitting, spiral-shaped scroll members. They are currently used in residential and commercial air-conditioning and heat pump applications as well as in automotive air-conditioning systems. Capacities range from 10,000 to 170,000 Btu/h (3 to 50 kW). To function effectively, the scroll compressor requires close tolerance machining of the scroll members, which has become possible only recently due to current advances in manufacturing technology. This positive-displacement, rotary motion compressor includes performance features, such as high efficiency and low noise.

Scroll members are typically a geometrically matched pair, assembled 180° out of phase. Each scroll member is open on one end of the vane and bound by a base plate on the other. The two scrolls are fitted to form pockets between their respective base plates and various lines of contact between their vane walls. One scroll is held fixed, while the other moves in an orbital path with respect to the first. The flanks of the scrolls remain in contact, although the contact locations move progressively inward. Relative rotation between the pair is prevented by a coupling. An alternative approach creates relative orbital motion via two scrolls synchronously rotating about noncoincident axes.

Compression is accomplished by sealing suction gas in pockets of a given volume at the outer periphery of the scrolls and progressively reducing the size of these pockets as the scroll relative motion moves them inward toward the discharge port. Figure 18-15 shows the sequence of suction, compression, and discharge phases.

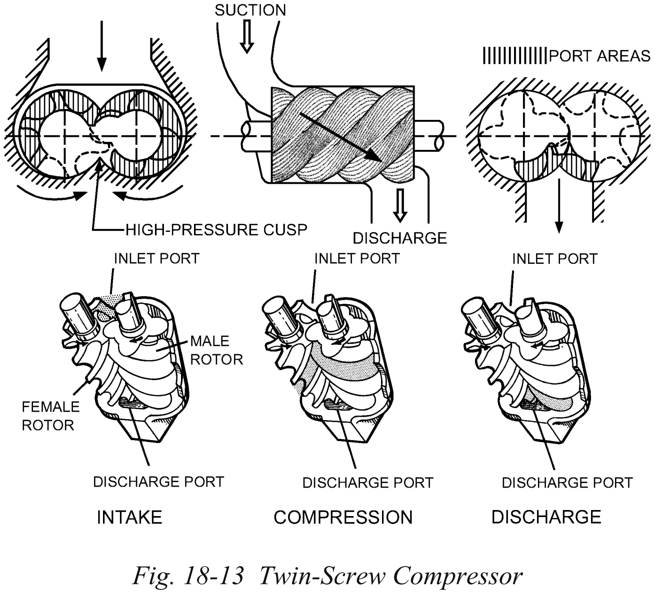


Fig. 18-13 Twin-Screw Compressor
(Adapted from Figures 32 and 33, Chapter 38, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

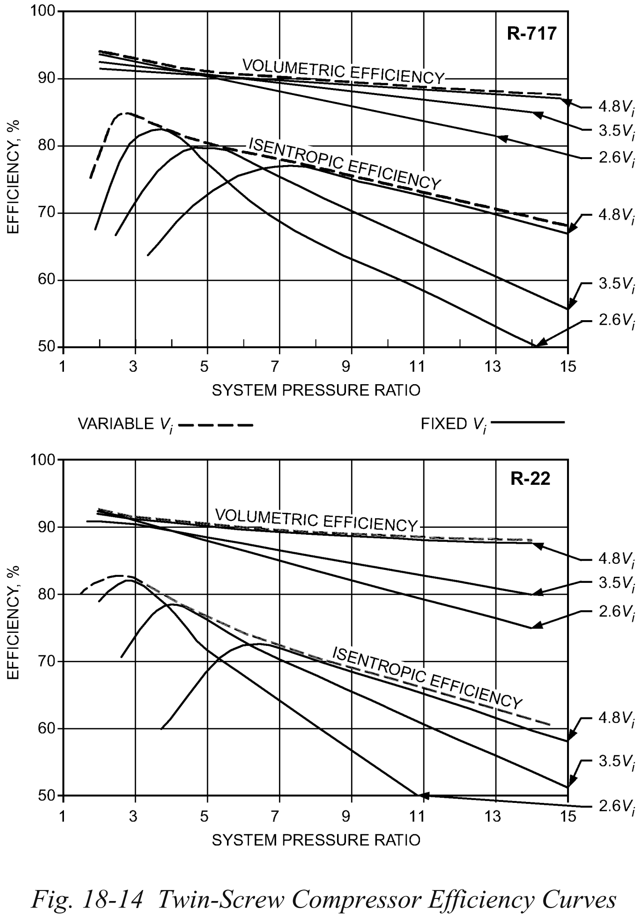


Fig. 18-14 Twin-Screw Compressor Efficiency Curves
(Figure 37, Chapter 38, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

As the outermost pockets are sealed off (Figure 18-15a), the trapped gas is at suction pressure and has just entered the compression process. At stages (b) through (f), orbiting motion moves the gas toward the center of the scroll pair, and pressure rises as pocket volumes are reduced. At stage (g), the gas reaches the central discharge port and begins to exit the scrolls. Stages (a) through (h) show that two distinct compression paths operate simultaneously in a scroll set. The discharge process is nearly continuous, since new pockets reach the discharge stage very shortly after the previous discharge pockets have been evacuated.

Both high-side and low-side shells are available. In the former, the entire compressor is at discharge pressure, except for the outer areas of the scroll set. Suction gas is introduced into the suction port of the scrolls through piping, which keeps it discrete from the rest of the compressor. Discharge gas is directed into the compressor shell, which acts as a plenum. In the low-side type, most of the shell is at suction pressure, and the discharge gas exiting the scrolls is routed outside the shell, sometimes through a discrete or integral plenum.

Scroll technology offers an advantage in performance for a number of reasons. Large suction and discharge ports reduce pressure losses incurred in the suction and discharge processes. Physical separation of these processes also reduces heat transfer to the suction gas. The absence of valves and reexpansion volumes and the continuous flow process results in high volumetric efficiency over a wide range of operation conditions. Figure 18-16 illustrates this effect.

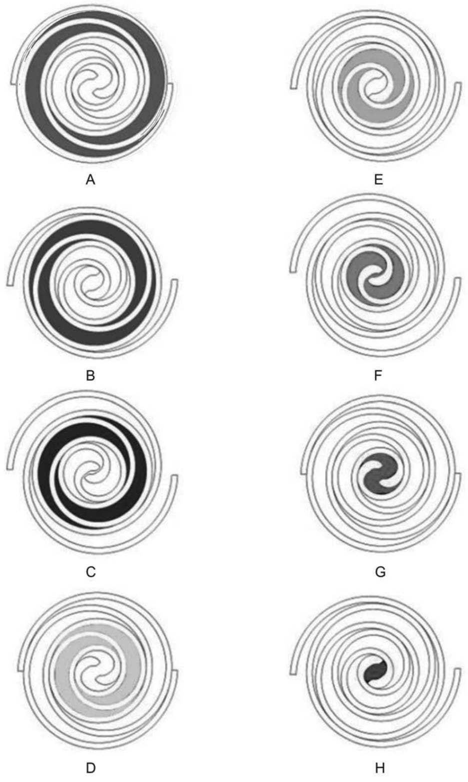


Fig. 18-15 Scroll Compression Process
(Figure 45, Chapter 38, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

The built-in volume ratio can be designed for lowest over- or undercompression at typical demand conditions (2.5 to 3.5 pressure ratio for air conditioning). Isentropic efficiency in the range of 70% is possible at such pressure ratios, and it remains quite close to the efficiency of other compressor types at high pressure ratios. Scroll compressors offer a flatter capacity versus outdoor ambient curve than reciprocating products, which means that they can more closely approach indoor requirements at high demand conditions. As a result, the heat pump mode requires less supplemental heating; the cooling mode is more comfortable because cycling is less as demand decreases. Scroll compressors available in the United States are typically specified as producing AHRI operating efficiencies (COP) in the range of 3.10 to 3.34.

Trochoidal compressors are small, rotary, positive-displacement compressors that can run at high speed up to 9000 rpm. They are manufactured in various configurations. Trochoidal curvatures can be produced by the rolling motion of one circle outside or inside the circumference of a basic circle, producing either epitrochoids or hypotrochoids, respectively. Both types of trochoids can be used either as a cylinder or piston form, so that four types of trochoidal machines can be designed (Figure 18-17).

In each case, the counterpart of the trochoid member always has one apex more than the trochoid itself. In the case of a trochoidal cylinder, the apexes of the piston show slipping along the inner cylinder surface; for trochoidal piston design, the piston shows a gear-like motion. As seen in Figure 18-17, a built-in theoretical pressure ratio disqualifies many configurations as valid concepts for refrigeration compressor design. Because of additional valve ports, clearances, etc., and the

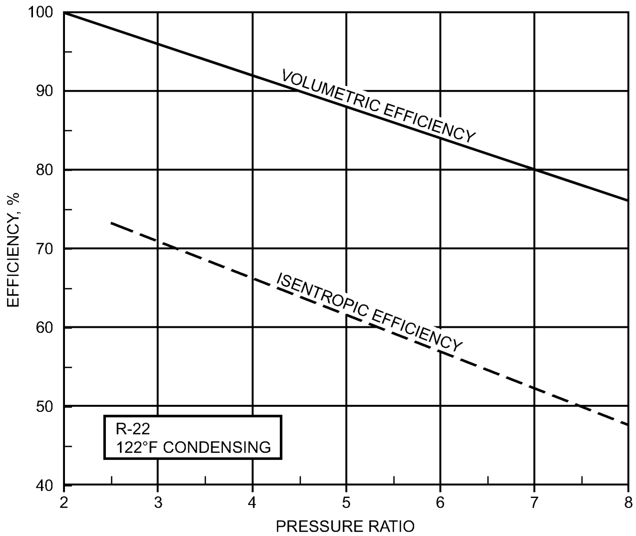


Fig. 18-16 Volumetric and Isentropic Efficiency versus Pressure Ratio for Scroll Compressor
(Figure 41, Chapter 38, 2012 ASHRAE Handbook—HVAC Systems and Equipment)

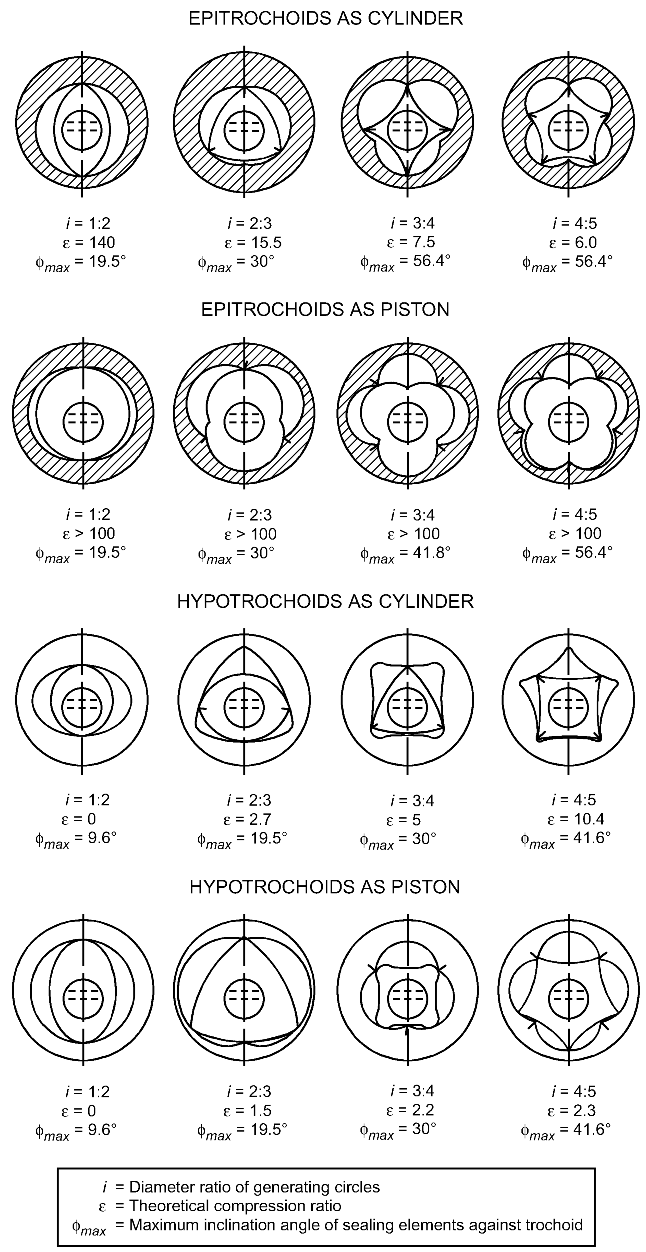


Fig. 18-17 Possible Versions of Epitrochoidal and Hypotrochoidal Machines
(Figure 53, Chapter 38, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

resulting decrease in the built-in maximum theoretical pressure ratio, only the first two types with epitrochoidal cylinders, and all candidates with epitrochoidal pistons, can be used for compressor technology. The latter, however, require sealing elements on the cylinder as well as on the side plates, which does not allow the design of a closed sealing borderline.

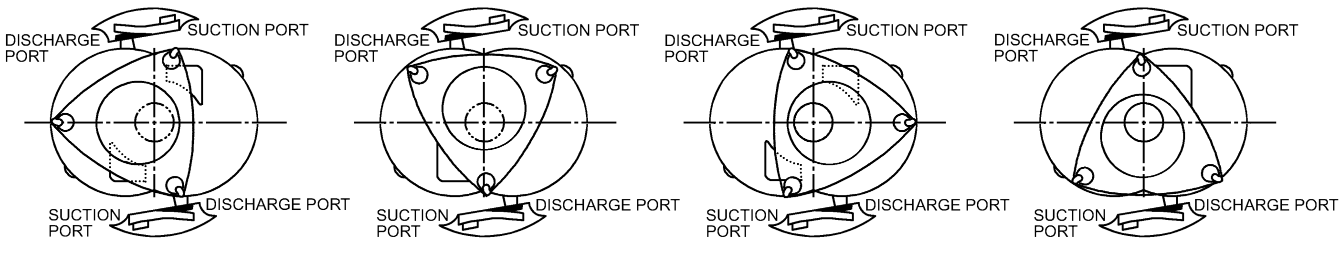
In the past, trochoidal machines were designed much like those of today. However, like other positive-displacement rotary concepts that could not tolerate oil injection, early trochoidal equipment failed because of sealing problems. The invention of a closed sealing border by Wankel changed this. Today, the Wankel trochoidal compressor with a three-sided epitorchoidal piston (motor) and two-envelope cylinder (casing) is built in capacities of up to 2 tons. The sequence of operation of a Wankel rotary compressor is illustrated in Figure 18-18.

Centrifugal compressors, or turbocompressors, are characterized by a continuous exchange of angular momentum between a rotating mechanical element and a steadily flowing fluid. Because their flows are continuous, turbomachines have greater volumetric capacities, size-for-size, than do positive-displacement devices. For effective momentum exchange, their rotative speeds must be higher, but little vibration or wear results because of the steadiness of the motion and the absence of contacting parts.

In centrifugal compressors, the suction flow enters the rotating element, or impeller, in the axial direction and is discharged radially at a higher velocity. This dynamic head is then converted to static head, or pressure, through a diffusion process, which generally begins within the impeller and ends in a radial diffuser and scroll outboard of the impeller.

Centrifugal compressors are used in a variety of refrigeration and air-conditioning installations, but primarily in packaged water chillers. Suction flow rates range between 60 and 30,000 cfm (0.03 and 14 m^3/s), with rotational speeds between 1800 and 90,000 rpm. However, the high angular velocity associated with a low volumetric flow establishes a minimum practical capacity for most centrifugal applications. The upper capacity limit is determined by physical size, a 30,000 cfm (14 m^3/s) compressor being about 6 or 7 ft (2 m) in diameter.

A centrifugal compressor can be single stage, having only one impeller, or it can be multistage, having two or more impellers mounted in the same casing as shown in



1. SUCTION 2. COMPRESSION 3. DISCHARGE 4. FINISHED DISCHARGE

Fig. 18-18 Sequence of Operation of Wankel Rotary Compressor
(Figure 46, Chapter 38, 2012 ASHRAE Handbook—Systems and Equipment)

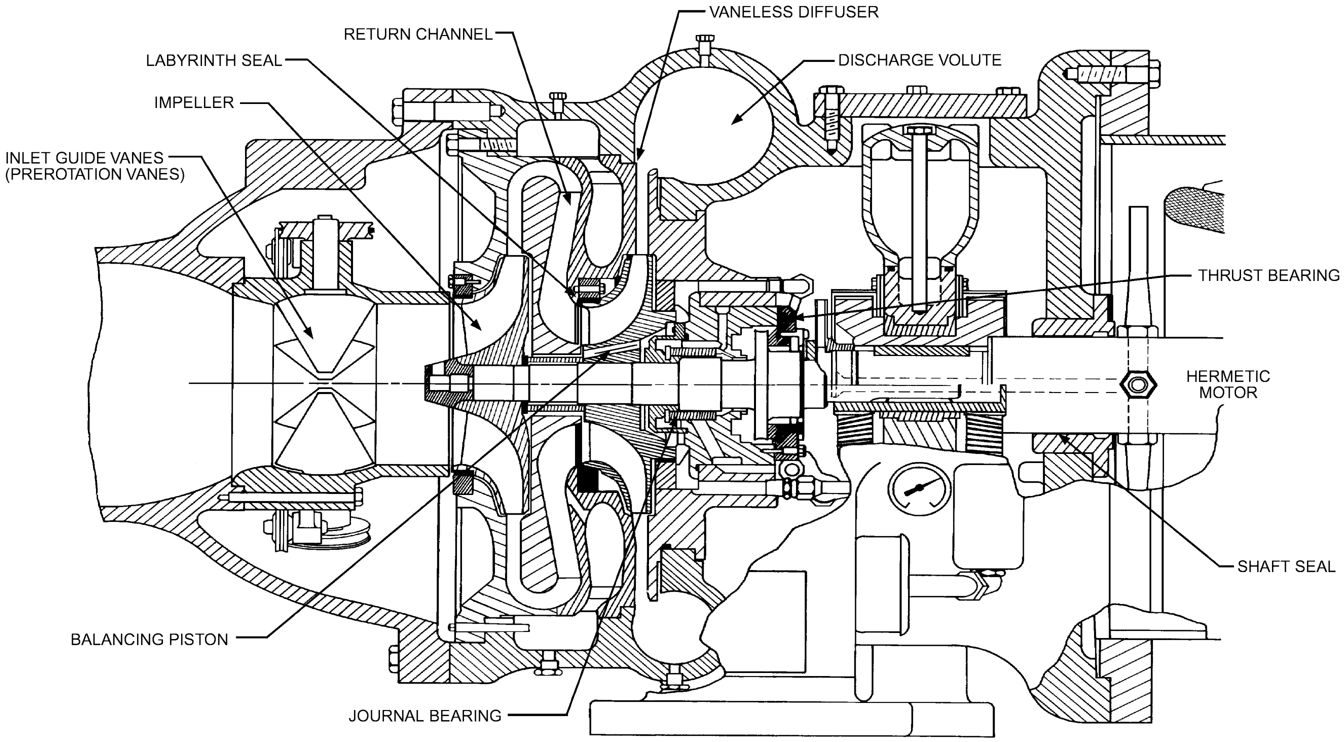


Fig. 18-19 Centrifugal Refrigeration Compressor
*(Figure 55, Chapter 38, 2016 ASHRAE Handbook—
HVAC Systems and Equipment)*

Figure 18-19. For process refrigeration applications, a compressor can have as many as ten stages.

The suction gas generally passes through a set of adjustable inlet guide vanes or an external suction damper before entering the impeller. The vanes (or suction damper) are used for capacity control.

Suction temperatures are usually between 50 and –150°F (10 and –100°C), with suction pressures between 2 and 100 psia (14 and 700 kPa) and discharge pressures up to 300 psia (2100 kPa). Pressure ratios range between 2 and 30. Almost any refrigerant can be used.

The momentum exchange, or energy transfer, between a centrifugal impeller and a flowing refrigerant is expressed by the following equation:

$$W_i = u_i\, c_u / g$$

where

W_i = impeller work input per unit mass of refrigerant, $\text{ft·lb}_f/\text{lb}_m$
u_i = impeller blade tip speed, ft/s
c_u = tangential component of refrigerant velocity leaving impeller blades, ft/s
g = gravitational constant, 32.17 $\text{lb}_m\text{·ft/lb}_f\text{s}^2$

These velocities are shown in Figure 18-20, where refrigerant flows out from between impeller blades with relative velocity b and absolute velocity c. The relative angle β is a few

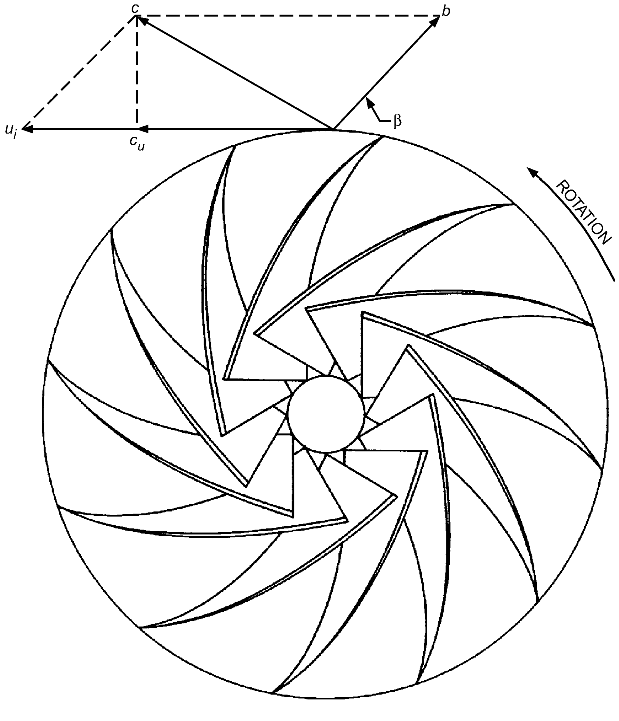


Fig. 18-20 Impeller Exit Velocity Diagram
*(Figure 59, Chapter 38, 2016 ASHRAE Handbook—
HVAC Systems and Equipment)*

degrees smaller than the blade angle because of a phenomenon known as slip. This equation assumes that the refrigerant enters the impeller without any tangential velocity component or swirl. This is generally the case at design flow conditions.

At least at low refrigerant flow rates, the tip speed of the impeller and the tangential velocity of the refrigerant are nearly identical. Thus, for a mass flow rate m, the ideal power can be estimated by

$$P = mc_u^2 = mu_i^2$$

Another expression for the ideal power input comes from the first law of thermodynamics:

$$P = m\Delta h_i$$

where Δh_i is the isentropic change in enthalpy across the compressor.

Equating the two expressions for power yields an order-of-magnitude estimate of the tip speed:

$$u_i^2 = \Delta h_i(g) \qquad \text{ft/s}$$

Example 18-1 Estimate the impeller tip speed needed to compress R-717 (ammonia) from saturated vapor at 20°F to a pressure corresponding to a condensing temperature of 100°F.

Solution*:*

From Figure 18, Chapter 30,2017 *ASHRAE Handbook—Fundamentals*:

$$\Delta h_i = 718 - 617 = 101 \text{ Btu/lb}$$

The tip speed is

$$u_i = [(32.2)(101)(778)]^{1/2} = 1591 \text{ ft/s}$$

Note: $g = 32.2$ ft/s^2 and 778 ft-lb/Btu is a conversion factor.

Some of the work done by the impeller increases the refrigerant pressure, while the remainder only increases its kinetic energy. The ratio of pressure-producing work to total work is known as the impeller reaction. Since this varies from about 0.4 to 0.7, an appreciable amount of kinetic energy leaves the impeller with magnitude $c^2/2g$. To convert this kinetic energy into additional pressure, a diffuser is located after the impeller. Radial vaneless diffusers are most common, but vaned, scroll, and conical diffusers are also used. In a multistage compressor, the flow leaving the first diffuser is guided to the inlet of the second impeller and so on through the machine. The total compression work input is the sum of the individual stage inputs provided that the mass flow rate is constant throughout the compressor:

$$W = \Delta W_i$$

18.1.2 Condensers

The condenser removes (from the refrigerant gas) the heat of compression and the heat absorbed by the refrigerant in the evaporator. The refrigerant is thereby converted back into the liquid phase at the condenser pressure and is available for reexpansion into the evaporator. The common forms of condensers may be classified on the basis of the cooling medium as (1) water-cooled, (2) air-cooled, and (3) evaporative (air and water) cooled.

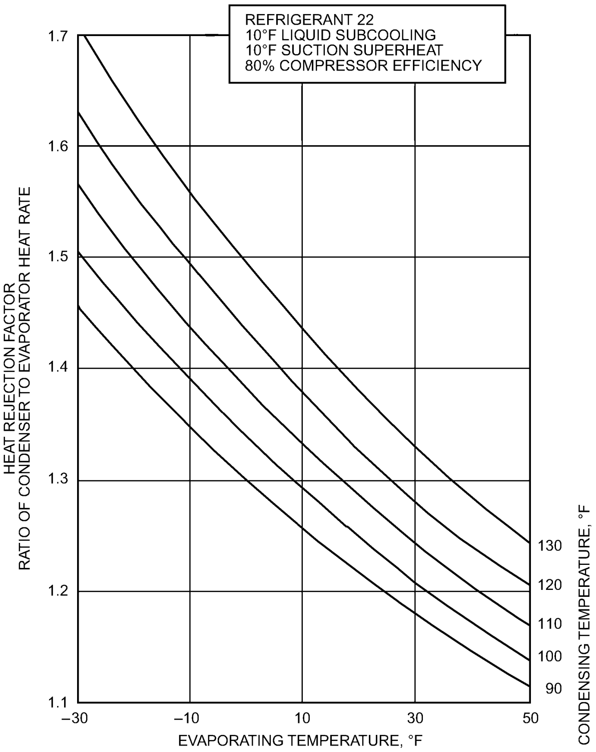


Fig. 18-21 Heat Removed in Condenser
(Figure 1, Chapter 39, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Water-cooled condensers consist of the following types:

- Shell-and-tube (vertical)
- Shell-and-tube (horizontal)
- Shell-and-coil (horizontal and vertical)
- Double pipe
- Atmospheric

The selection of a water-cooled condenser depends on the cooling load, the refrigerant used, the source and temperature of the available cooling water, the amount of water that can be circulated, the condenser location, the required operating pressures, and the maintainability.

The heat rejection rate of the condenser for each unit of refrigeration produced in the evaporator may be estimated from Figure 18-21. Similar plots can be prepared for other refrigerants from tables of thermodynamic properties. In practice, the heat removed is 5 to 10% higher than the theoretical values because of losses during compression.

An accurate determination of the heat rejection requirement q_o can usually be made from known values of evaporator load q_i and the heat equivalent of the actual power required q_w for compression:

$$q_o = q_i + q_w \qquad \text{Btu/h} \tag{18-1}$$

Note: q_w is reduced by any independent heat rejection processes such as oil cooling and motor cooling.

The volumetric flow rate Q of condensing water required may be found from the following equation:

$$Q = \frac{q_o}{\rho c_p(t_2 - t_1)} \quad \text{ft}^3/\text{h} \qquad (18\text{-}2)$$

where

q_o = heat rejection rate, Btu/h
ρ = density of water, lb/ft^3
t_1 = temperature of water entering condenser, °F
t_2 = temperature of water leaving condenser, °F
c_p = specific heat of water, Btu/lb·°F

The heat rejection rate may also be determined as:

$$q_o = UA\Delta t_m \quad \text{Btu/h} \qquad (18\text{-}3)$$

where

U = overall heat transfer coefficient, Btu/h·ft^2·°F
A = surface area associated with U, ft^2
Δt_m = mean temperature difference, °F

The computation of overall heat transfer in a water-cooled condenser with water inside the tubes may be made from calculated or test-derived heat transfer coefficients of the water and refrigerant sides, from physical measurements of the condenser tubes, and from a fouling factor on the water side, by using

$$U_o = \frac{1}{(S_r/h_w) + S_R r_{fw} + (X/k)(A_o/A_m) + 1/h_r \phi_w} \qquad (18\text{-}4)$$

where

U_o = overall heat transfer coefficient, based on the external surface and the log mean temperature difference, between the external and internal fluids, Btu/h·ft^2·°F [(W/(m^2·K)]
S_R = ratio of external to internal surface area
h_w = internal or water side film coefficient, Btu/h·ft^2·°F [W/(m^2·K)]
r_{fw} = fouling resistance on water side, ft^2·°F·h/Btu (m^2·K/W)
X = thickness of tube wall, ft (m)
k = thermal conductivity of tube material, Btu/h·ft·°F [W/(m·K)]
A_o/A_m = ratio of external surface to mean heat transfer area of metal wall
h_r = external, or refrigerant side coefficient, Btu/h·ft^2·°F [W/(m^2·K)]
ϕ_w = weighted fin efficiency (100% for bare tubes)

Values of the water-side coefficient may be calculated from equations in Chapter 4 of the 2017 *ASHRAE Handbook—Fundamentals*. For turbulent flow, at Reynolds numbers exceeding 10,000 in horizontal tubes and using average water temperatures, the general equation is

$$\frac{h_w D}{k} = 0.023\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c_p \mu}{k}\right)^{0.4} \qquad (18\text{-}5)$$

where

D = inside tube diameter, ft (m)
k = thermal conductivity of water, Btu/h·ft·°F [W/(m·K)]
G = mass velocity of water, lb/s·ft^2 (kg/s·m^2)
μ = viscosity of water, lb/ft·h (mPa·s)
c_p = specific heat of water at constant pressure, Btu/lb·°F [(kJ/kg·K)]

Note: The constant (0.023) in this equation applies only to tubes with plain inside diameters.

Because of its strong influence on the value of h_w, water velocity should be maintained as high as permitted by water pressure drop considerations. Maximum velocities with clean water of 6 to 10 ft/s are commonly used. A minimum velocity of 3 ft/s is considered good practice when the water quality is such that noticeable fouling or corrosion could result. With clean water, the velocity may be lower if dictated by conservation or low supply temperature considerations.

Factors that influence the value of h_r are

- Type of refrigerant being condensed
- Geometry of condensing surface (plain tube, outside diameter, finned tube, fin spacing, height, and cross-section profile)
- Condensing temperature
- Condensing rate, in terms of mass velocity or rate of heat transferred
- Arrangement of tubes in bundle
- Vapor distribution and flow rate
- Condensate drainage

Values of the refrigerant side coefficients may be estimated from correlations shown in Chapters 4 or 5 of the 2017 *ASHRAE Handbook—Fundamentals*.

Example 18-2 Estimate the volumetric flow rate of condensing water required for the condenser of an R-22 water chilling unit assumed to be operating at a condensing temperature of 100°F, and evaporating temperature of 40°F, an entering condensing water temperature of 86°F, a leaving condensing water temperature of 95°F, and a refrigeration load of 100 tons.

Solution:

From Figure 18-21, the heat rejection factor is found to be 1.17.

$$q_o = 100 \times 1.17 = 117 \text{ tons} = 1{,}404{,}000 \text{ Btu/h}$$

ρ = 62.2 lb/ft^3 at 90.5°F
c_p = 1 Btu/lb·°F

From Equation (18-2),

$$Q = 1{,}404{,}000/[62.2 \times 1(95 - 86)] = 2500 \text{ ft}^3/\text{h} = 310 \text{ gpm}$$

A typical horizontal closed shell-and-tube ammonia condenser is shown in Figure 18-22.

Air-Cooled Condensers. The heat transfer process in an air-cooled condenser has three main phases: (1) desuperheating, (2) condensing, and (3) subcooling. The changes of state of R-134a passing through the condenser coil and the corresponding temperature change of the cooling air as it passes through the coil are shown in Figure 18-23. Desuper-

heating, condensing, and subcooling zones vary 5 to 10%, depending on the entering gas temperature and the leaving liquid temperature, but Figure 18-23 is typical for most of the commonly used refrigerants.

Condensing occurs in approximately 85% of the condenser area at a substantially constant temperature. The drop in condensing temperature is due to the friction loss through the condenser coil.

Coils in air-cooled condensers are commonly constructed of copper, aluminum, or steel tubes, ranging from 1/4 to 3/4 in (8 to 20 mm). diameter. Copper is easy to use in manufacturing and requires no protection against corrosion. Aluminum requires exact manufacturing methods, and special protection must be provided if aluminum to copper joints are made. Steel tubing is used, but weather protection must be provided.

Fins are used to improve the air-side heat transfer. Fins are usually made of aluminum, but copper and steel are also used. The most common forms are plate fins making a coil bank, plate fins individually fastened to the tube, or a fin spirally wound onto the tube. Other forms such as plain tube-fin extrusions or tube extrusions with accordion type fins are also used. The number of fins per inch varies from 4 to 30 (0.8 to 6.4 mm fin spacing). The most common range is 8 to 18 (1.4 to 3.22 mm spacing).

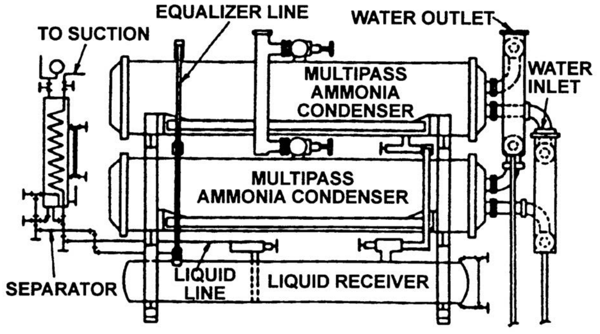


Fig. 18-22 Horizontal Shell-and-Tube Ammonia Condenser and Receiver

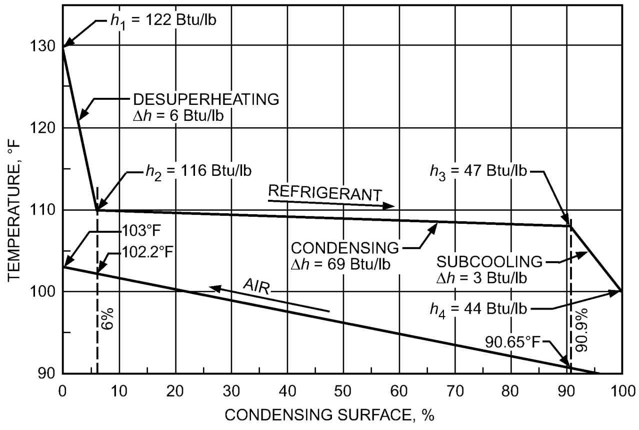
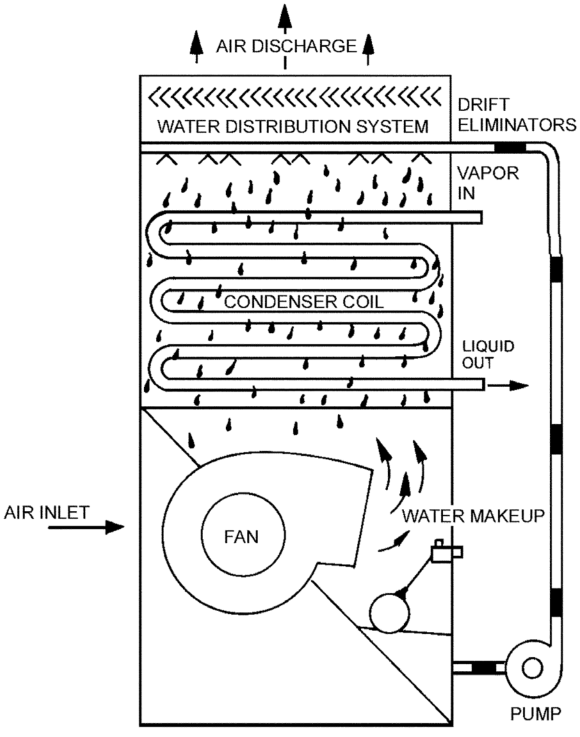
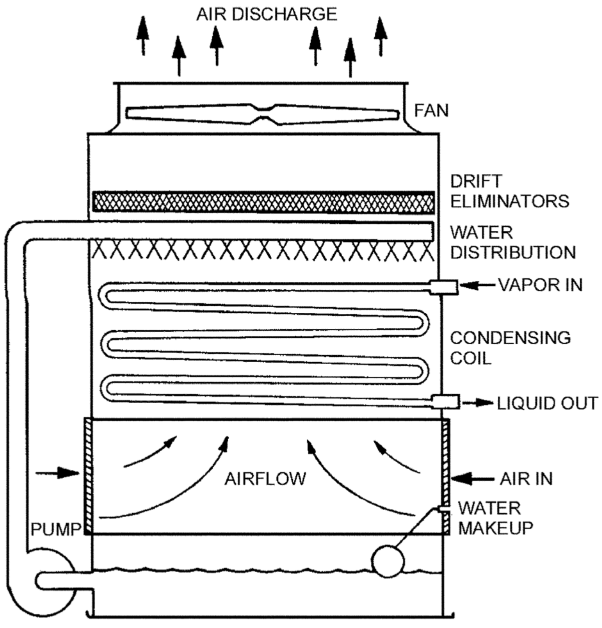


Fig. 18-23 Temperature and Enthalpy Changes in Air-Cooled Condenser with R-134a
(Figure 6, Chapter 39, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Evaporative Condensers. As with water-cooled and air-cooled condensers, evaporative condensers reject heat from a condensing vapor into the environment. In an evaporative condenser, hot high-pressure vapor from the compressor discharge circulates through a condensing coil that is continually wetted on the outside by a recirculating water system. As seen in Figure 18-24, air is simultaneously



BLOW-THROUGH TYPE



DRAW-THROUGH TYPE

Fig. 18-24 Functional View of Evaporative Condenser
(Adapted from Figure 10, Chapter 39, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

directed over the coil, causing a small portion of the recirculated water to evaporate. This evaporation removes heat from the coil, thus cooling and condensing the vapor.

Evaporative condensers reduce the water pumping and chemical treatment requirements associated with cooling tower/refrigerant condenser systems. In comparison with an air-cooled condenser, an evaporative condenser requires less coil surface and airflow to reject the same heat, or alternatively, greater operating efficiencies can be achieved by operating at a lower condensing temperature.

The evaporative condenser can operate at a lower condensing temperature than an air-cooled condenser because the air-cooled condenser is limited by the ambient dry-bulb temperature. In the evaporative condenser, heat rejection is limited by the ambient wet-bulb temperature, which is normally 14 to 24°F (8 to 13°C) lower than the ambient dry bulb. The evaporative condenser also provides lower condensing temperatures than the cooling tower/water-cooled condenser because the heat transfer/mass transfer steps are reduced from two (between the refrigerant and the cooling water and between the water and ambient air) to one step (refrigerant directly to ambient wet bulb). While both the water-cooled condenser/cooling tower combination and the evaporative condenser use evaporative heat rejection, the former has added a second step of nonevaporative heat transfer from the condensing refrigerant to the circulating water, requiring more surface area. Evaporative condensers are, therefore, the most compact for a given capacity.

18.1.3 Refrigerant Expansion and Control Devices

Any refrigeration system requires that the flow of refrigerant be controlled. Valves are used to start, stop, direct, and modulate the flow of refrigerant to satisfy load requirements. To ensure satisfactory performance, valves should be adequately protected from foreign material, excessive moisture, and corrosion. Such protection is accomplished by installing properly sized strainers and driers.

Thermostatic Expansion Valves. The thermostatic expansion valve controls the flow rate of liquid refrigerant entering the evaporator in response to the superheat of the refrigerant gas leaving the evaporator. It keeps the entire evaporator active, without permitting unevaporated refrigerant liquid to be returned through the suction line to the compressor. The thermostatic expansion valve does so by controlling the mass flow rate of refrigerant entering the evaporator so that it equals the rate at which the refrigerant can be completely vaporized in the evaporator by heat absorption. Since the thermostatic expansion valve is operated by the superheated refrigerant gas leaving the evaporator and is responsive to changes in superheat of this gas, a portion of the evaporator must be devoted to superheating the refrigerant gas.

Unlike the constant pressure expansion valve, the thermostatic expansion valve is not limited to constant load applications. It is used to control refrigerant flow to all types of direct-expansion evaporators in air-conditioning, commercial, low-temperature, and ultra-low-temperature refrigeration systems.

A schematic cross section of the thermostatic expansion valve, with the principal components identified, is shown in Figure 18-25. Three forces are shown that govern thermostatic expansion valve operation:

p_1 = vapor pressure of the thermostatic element (a function of the bulb temperature), which is applied to the top of the diaphragm and acts to open valve
p_2 = evaporator pressure, which is applied underneath the diaphragm through the equalizer passage, and acts in a closing direction
p_3 = pressure equivalent of the superheat spring force, which is applied underneath the diaphragm, and is also a closing force

At any constant operating condition, these forces are balanced and $p_1 = p_2 + p_3$.

An additional force is that arising from the unbalanced pressure across the valve port. It can affect thermostatic expansion valve operation to a degree. For the configuration shown in Figure 18-26, the force due to port unbalance is the product of the pressure drop across the port and the difference in area of the port and the stem, and it would be an opening force. In other designs, depending on the direction of flow through the valve, the port unbalance might result in a closing force.

The principal effect of port unbalance is on valve control stability. As with any modulating control, if the ratio of power element area to port area is kept large, the unbalanced

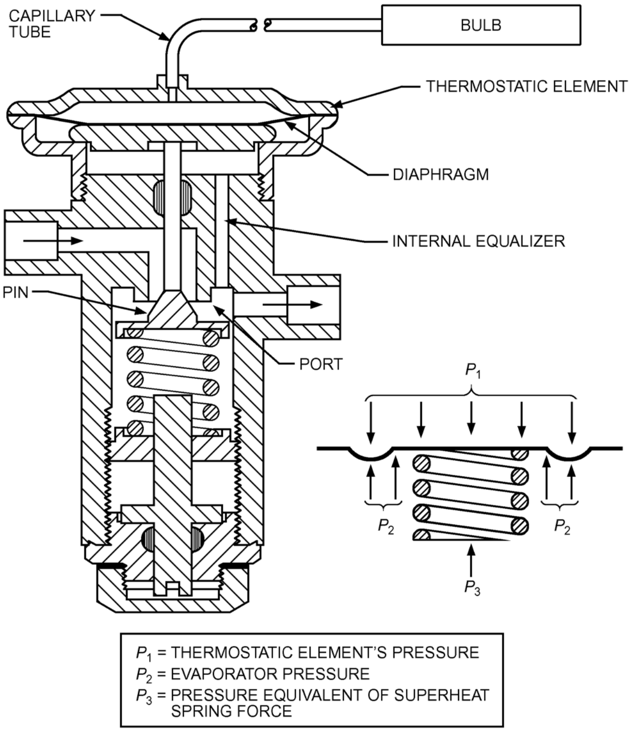


Fig. 18-25 Typical Thermostatic Expansion Valve
(Figure 10, Chapter 11, 2014 ASHRAE Handbook—Refrigeration)

port effect is minor. Large capacity valves are made with double-ported, or semibalanced, construction to minimize the effect of unbalanced pressure.

An evaporator using R-22 and operating at a saturation temperature of 40°F (4.4°C) and a pressure of 68.5 psig (472 kPa) is shown in Figure 18-26. Liquid refrigerant enters the expansion valve, is reduced in pressure and temperature at the valve port, and enters the evaporator at point A as a mixture of saturated liquid and vapor. As flow continues through the evaporator, more and more of the boiling refrigerant is evaporated. The refrigerant temperature remains at 40°F (4.4°C) until the liquid portion is completely evaporated by the absorption of heat at point B. From this point, additional heat absorption increases the temperature and superheats the refrigerant gas, while the pressure remains constant at 68.5 psig (472 kPa), until at point C (the outlet of the evaporator), the refrigerant gas temperature is 50°F (10°C). At this point, the superheat is 10°F (from 40 to 50°F) [5.6°C (from 4.4 to 10°C)].

An increase in the heat load on the evaporator increases the temperature of the refrigerant gas leaving the evaporator. The bulb of the thermostatic expansion valve senses this increase; the thermostatic charge pressure (p_1) increases and causes the valve to open wider. The increased flow rate results in a higher evaporator pressure (p_2) and a balanced control point is established again. Conversely, a decrease in the heat load on the evaporator decreases the temperature of the refrigerant gas leaving the evaporator and causes the thermostatic expansion valve pin to move in a closing direction.

External pressure equalizing thermostatic expansion valves are also used. A pressure line is connected between the valve and the suction side of the evaporator. This connection compensates for the frictional pressure loss in the evaporator. A common technique for this type of valve installation is illustrated in Figure 18-27.

Constant Pressure Expansion Valves. The constant pressure expansion valve is operated by the evaporator or valve outlet pressure to regulate the mass flow rate of liquid refrigerant entering the evaporator and thereby maintain this pressure at a constant value.

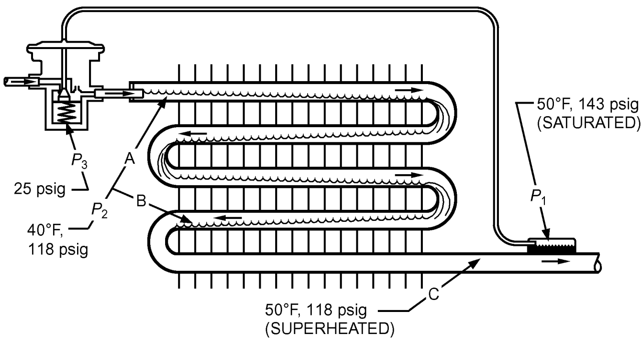


Fig. 18-26 Thermostatic Expansion Valve Controlling Flow of Liquid R-22 Entering Evaporator (Assuming R-22 Charge in Bulb)
(Figure 12, Chapter 11, 2014 ASHRAE Handbook—Refrigeration)

Figure 18-28 shows a schematic cross section of a constant pressure expansion valve. The valve has an adjustable spring that exerts its force on top of the diaphragm in an opening direction and a spring beneath the diaphragm that exerts its force in a closing direction. Evaporator pressure admitted beneath the diaphragm, through either the internal or external equalizer passage, combines with the closing spring to counterbalance the opening spring pressure.

With the valve set and feeding refrigerant at a given pressure, a small increase in the evaporator pressure forces the diaphragm upward and causes the valve pin to move in a closing direction, thereby restricting refrigerant flow and limiting evaporator pressure. When the evaporator pressure, because of a decrease in load, drops below the valve setting, the top spring pressure moves the valve pin in an opening direction, thereby increasing the refrigerant flow in an effort to raise the evaporator pressure to the balanced valve setting. This valve controls the evaporation of the liquid refrigerant in the evaporator at a constant temperature.

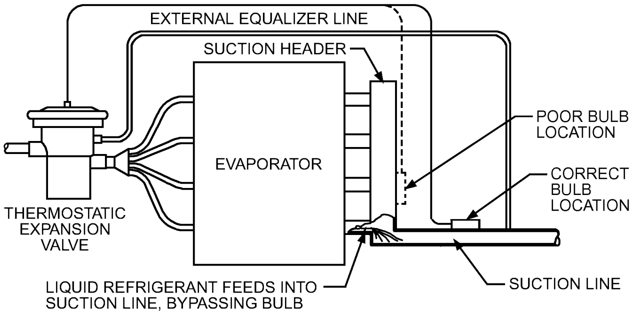
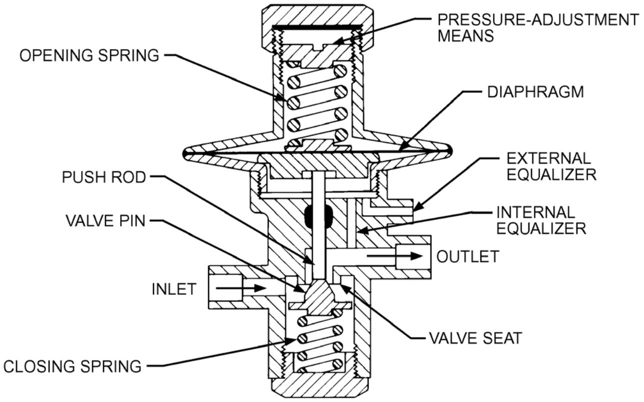


Fig. 18-27 Bulb Location for Thermostatic Expansion Valve
(Figure 16, Chapter 11, 2014 ASHRAE Handbook—Refrigeration)



Valve is used with either internal or external equalizer, but not with both.

Fig. 18-28 Constant Pressure Expansion Valve
(Figure 25, Chapter 11, 2010 ASHRAE Handbook—Refrigeration)

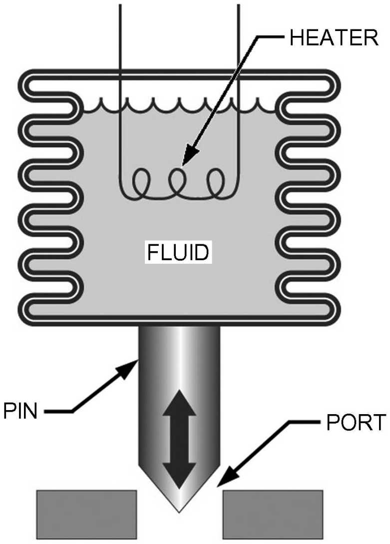


Fig. 18-29 Fluid-Filled Heat-Motor Valve
(Figure 20, Chapter 11, 2014 ASHRAE Handbook—Refrigeration)

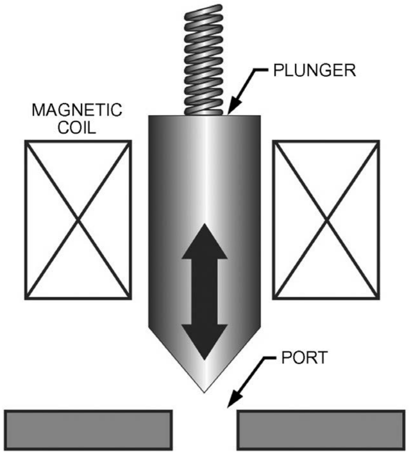


Fig. 18-30 Magnetically Modulated Valve
(Figure 21, Chapter 11, 2014 ASHRAE Handbook—Refrigeration)

Electric Expansion Valves. Application of an electric expansion valve requires a valve, controller, and control sensors. The control sensors may include pressure transducers, thermistors, resistance temperature devices (RTDs), or other pressure and temperature sensors. See Chapter 37 in the 2017 *ASHRAE Handbook—Fundamentals* for a discussion of instrumentation. Specific types should be discussed with the electric valve and electronic controller manufacturers to ensure compatibility of all components.

Electric valves typically have four basic types of actuation:

- Heat-motor operated
- Magnetically modulated
- Pulse-width-modulated (on/off type)
- Step-motor-driven

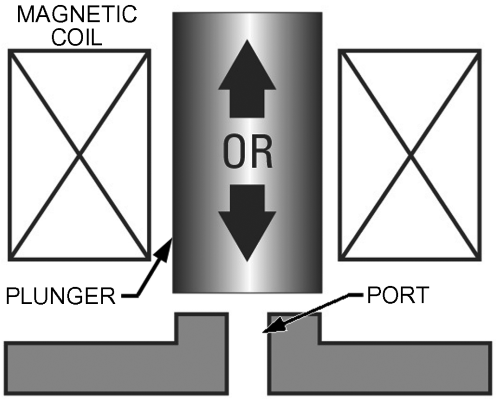


Fig. 18-31 Pulse-Width-Modulated Valve
(Figure 22, Chapter 11, 2014 ASHRAE Handbook—Refrigeration)

Heat-motor valves may be one of two types. In one type, one or more bimetallic elements are heated electrically, causing them to deflect. The bimetallic elements are linked mechanically to a valve pin or poppet; as the bimetallic element deflects, the valve pin or poppet follows the element movement. In the second type, a volatile fluid is contained within an electrically heated chamber so that the regulated temperature (and pressure) is controlled by electrical power input to the heater. The regulated pressure acts on a diaphragm or bellows, which is balanced against atmospheric air pressure or refrigerant pressure. The diaphragm is linked to a pin or poppet, as shown in Figure 18-29.

A **magnetically modulated** (analog) valve functions by modulation of an electromagnet; a solenoid armature compresses a spring progressively as a function of magnetic force (Figure 18-30). The modulating armature may be connected to a valve pin or poppet directly or may be used as the pilot element to operate a much larger valve. When the modulating armature operates a pin or poppet directly, the valve may be of a pressure-balanced port design so that pressure differential has little or no influence on valve opening.

The **pulse-width-modulated valve** is an on/off solenoid valve with special features that allow it to function as an expansion valve through a life of millions of cycles (Figure18-31). Although the valve is either fully opened or closed, it operates as a variable metering device by rapidly pulsing the valve open and closed. For example, if 50% flow is needed, the valve will be open 50% of the time and closed 50% of the time. The duration of each opening, or pulse, is regulated by the electronics.

A **step motor** is a multiphase motor designed to rotate in discrete fractions of a revolution, based on the number of signals or “steps” sent by the controller. The controller tracks the number of steps and can offer fine control of the valve position with a high level of repeatability. Step motors are used in instrument drives, plotters, and other applications where accurate positioning is required. When used to drive expansion valves, a lead screw changes the rotary motion of the rotor to a linear motion suitable for moving a valve pin or

poppet (Figure 18-32A). The lead screw may be driven directly from the rotor, or a reduction gearbox may be placed between the motor and lead screw. The motor may be hermetically sealed within the refrigerant environment, or the rotor may be enclosed in a thin-walled, nonmagnetic, pressuretight metal tube, similar to those used in solenoid valves, which is surrounded by the stator such that the rotor is in the refrigerant environment and the stator is outside the refrigerant environment. In some designs, the motor and gearbox can operate outside the refrigerant system with an appropriate stem seal (Figure 18-32B).

Electric expansion valves may be controlled by either digital or analog electronic circuits. Electronic control gives additional flexibility over traditional mechanical valves to consider control schemes that would otherwise be impossible, including stopped or full flow when required.

The electric expansion valve, with properly designed electronic controllers and sensors, offers a refrigerant flow control means that is not refrigerant specific, has a very wide load range, can often be set remotely, and can respond to a variety of input parameters.

Evaporator Pressure Regulators. The evaporator pressure regulator (back pressure regulator) regulates the evaporator pressure (pressure entering the regulator) at a constant value. It is used in the evaporator outlet or suction line to prevent frosting on the coil or to keep the leaving air temperature from lowering under light load conditions. These pressure regulators are commonly used on multiple evaporators served by a single compressor or when different suction pressures are required by multiple evaporator coils.

As illustrated in Figure 18-33, the inlet pressure acts on the bottom of the seat disk and is opposed by the adjusting spring. The outlet pressure acts on the underside of the bellows and the top of the seating disk, and, since the effective areas of the bellows and the port are equal, the two forces cancel and the valve is responsive to inlet pressure only. When the evaporator pressure rises above the force exerted by the spring, the valve moves in the opening direction. When the evaporator pressure drops below the force exerted by the spring, the valve moves in the closing direction. In actual operation, the valve assumes a throttling position to balance system load.

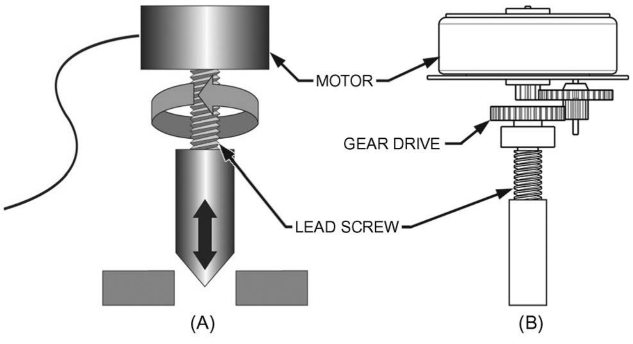


Fig. 18-32 Step Motor with (A) Lead Screw and (B) Gear Drive with Stem Seal
(Figure 23, Chapter 11, 2014 ASHRAE Handbook—Refrigeration)

Capillary Tubes. Every refrigerating unit requires a pressure-reducing device to meter the refrigerant flow to the low side in accordance with the system demands. The capillary tube is popular for smaller unitary hermetic equipment, such as household refrigerators and freezers, dehumidifiers, and room air conditioners. It is also used in larger units such as unitary air conditioners in sizes up to 10 tons (35 kW) capacity. The capillary operates on the principle that liquid passes through it more readily than does gas. It consists of a small diameter line that connects the outlet of the condenser to the inlet of the evaporator. It is sometimes soldered to the outer surface of the suction line for heat exchange purposes.

Assume that a condenser-to-evaporator capillary has been sized to permit the desired flow of refrigerant with a liquid seal at its inlet. If a system unbalance occurs so that some gas (uncondensed refrigerant) enters the capillary, this gas tends to considerably reduce the mass flow of refrigerant with little or no change in the system pressures. If the opposite type of unbalance occurs, liquid refrigerant backs up in the condenser. This condition tends to cause subcooling and increases the mass flow of refrigerant. Thus, a capillary properly sized for the application tends to automatically compensate for load and system variations and gives acceptable performance over a wide range of operating conditions.

A refrigerating system is operating at the **condition of capacity balance** when the resistance of the capillary is sufficient to maintain a liquid seal at its entrance without excess liquid accumulating in the high side of the system (Figure 18-34). Only one such capacity balance point exists for any given compressor discharge pressure.

18.1.4 Evaporators for Liquid Chillers

A liquid cooler (hereafter called a cooler) is a component of a refrigeration system in which the refrigerant is evaporated to produce a cooling effect on a fluid (usually water or

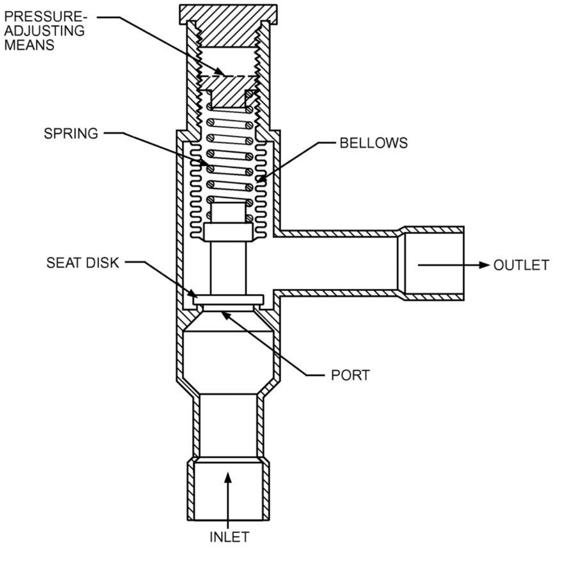


Fig. 18-33 Direct-Acting Evaporator Pressure Regulator
(Figure 25, Chapter 11, 2014 ASHRAE Handbook—Refrigeration)

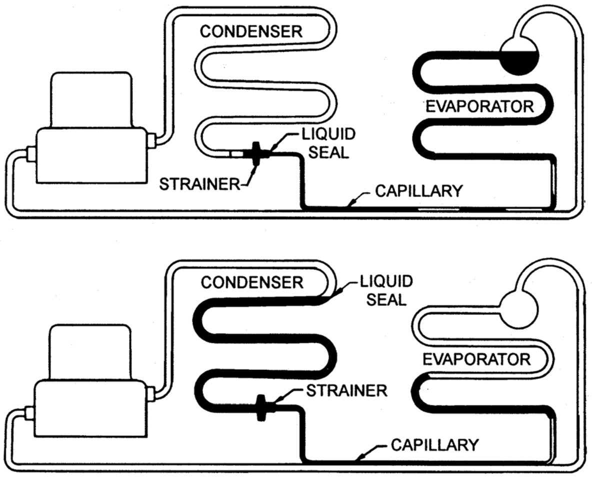
brine). Various types of water and brine coolers, as well as refrigerant flow control, capacity range, and refrigerants commonly used, are listed in Table 18-3.

In the **direct-expansion cooler**, the refrigerant is expanded into the inside of the tubes and vaporizes completely before leaving. The fluid being cooled is circulated on the outside of the tube surface within an enclosing shell. These coolers are usually used with positive-displacement compressors, such as reciprocating, rotary, or rotary screw compressors, to cool water or brine. Shell-and-tube is the most common arrangement, although tube-in-tube and brazed plate cooler are also available.

Figure 18-35 shows a typical shell-and-tube cooler. A series of baffles channels the fluid throughout the shell side. The baffles increase the velocity of the fluid, thereby increasing its heat transfer coefficient. The velocity of the fluid flowing perpendicular to the tubes should be at least 2 ft/s (0.6 m/s) to clean the tubes and less than 10 ft/s (3 m/s) to prevent erosion.

Distribution is critical in direct-expansion coolers. If some tubes are fed more refrigerant than others, they tend to bleed liquid refrigerant into the suction line. Since most direct-expansion coolers are controlled to a given suction super-heat, the remaining tubes must produce a higher superheat to evaporate the liquid bleeding through. This unbalance causes poor heat transfer. Uniform distribution is often achieved by a spray distributor. Most direct-expansion coolers are designed for horizontal mounting.

In a **flooded cooler**, the refrigerant vaporizes on the outside of tubes, which are submerged in liquid refrigerant within a closed shell. The fluid flows through the tubes as shown in Figure 18-36. Flooded coolers are usually used with rotary screw or centrifugal compressors to cool water or brine.



Notes:

1. Capillary selected for capacity balance conditions. Liquid seal at capillary inlet but no excess liquid in condenser. Compressor discharge and suction pressure normal. Evaporator properly charged.
2. Too much capillary resistance—liquid refrigerant backs up in condenser and causes evaporator to be undercharged. Compressor discharge pressure may be abnormally high. Suction pressure below normal. Bottom of condenser subcooled.

Fig. 18-34 Effect of Capillary Tube Selection on Refrigerant Distribution
(Figure 47, Chapter 11, 2014 ASHRAE Handbook—Refrigeration)

Refrigerant liquid/vapor mixture usually feeds into the bottom of the shell through a distributor that distributes the refrigerant vapor equally under the tubes. The relatively warm fluid in the tubes heats the refrigerant liquid surrounding the tubes, causing it to boil. As bubbles rise up through the space between tubes, the liquid surrounding the tubes becomes increasingly bubbly (or foamy, if much oil is present). The refrigerant vapor must be separated from the mist generated by the boiling refrigerant. The simplest separation method is provided by a dropout area between the top row of tubes and the suction connections. If this dropout area is insufficient, a coalescing filter may be required between the tubes and connectors.

The size of tubes, number of tubes, and number of passes should be determined to maintain the fluid velocity typically between 3 and 10 ft/s (1 and 3 m/s). In some cases, the minimum velocity may be determined by a lower Reynolds number limit to ensure turbulent flow. Flooded shell-and-tube coolers are generally unsuitable for other than horizontal orientation.

A **spray cooler** is similar to a flooded shell-and-tube cooler except that the refrigerant liquid is recirculated through spray nozzles located above the top tubes. None of the tubes is submerged in liquid.

A **shell-and-coil cooler** is a tank containing the fluid to be cooled with a simple coiled tube used to cool the fluid. This type of cooler has the advantage of cold fluid storage to offset peak loads. In some models, the tank can be opened for cleaning. Most applications are at low capacities (e.g., for bakeries, for photographic laboratories, and to cool drinking water).

The coiled tube containing the refrigerant can be either inside the tank (Figure 18-37) or attached to the outside of the tank in a way that permits heat transfer.

The rate at which heat is transferred in the evaporator is given by the following equation:

$$q = U_o A_o \Delta t_m \qquad \text{Btu/h} \qquad (18\text{-}6)$$

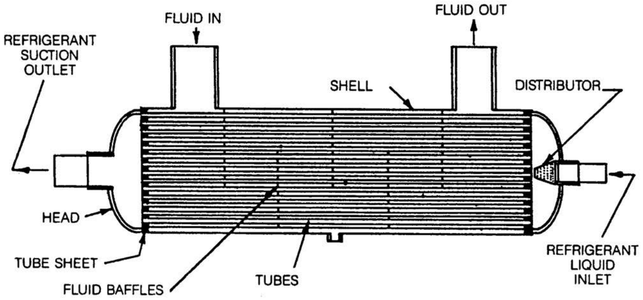


Fig. 18-35 Direct-Expansion Shell-and-Tube Cooler
(Figure 1, Chapter 42, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Table 18-3 Types of Coolers

(Table 1, Chapter 42, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Type of Cooler	Subtype	Usual Refrigerant Feed Device	Usual Capacity Range, tons	Commonly Used Refrigerants
Direct-expansion	Shell-and-tube	Thermal expansion valve	2 to 500	12, 22, 134a, 404A, 407C, 410A, 500, 502, 507A, 717
		Electronic modulation valve	2 to 500	
	Tube-in-tube	Thermal expansion valve	5 to 251	12, 22, 134a, 717
	Brazed-plate	Thermal expansion valve	0.6 to 200	12, 22, 134a, 404A 407C, 410A, 500, 502, 507A, 508B, 717, 744
	Semiwelded plate	Thermal expansion valve	50 to 1990	12, 22, 134a, 500, 502, 507A, 717, 744
Flooded	Shell-and-tube	Low-pressure float	25 to 2000	11, 12, 22, 113, 114
		High-pressure float	25 to 6000	123, 134a, 500, 502, 507A, 717
		Fixed orifice(s)	25 to 60000	
		Weir	25 to 6000	
	Spray shell-and-tube	Low-pressure float	50 to 10,000	11, 12, 13B1, 22
		High-pressure float	50 to 10,000	113, 114, 123, 134a
	Brazed-plate	Low-pressure float	0.6 to 200	12, 22, 134a, 500, 502, 507A, 717, 744
	Semiwelded plate	Low-pressure float	50 to 1990	12, 22, 134a, 500, 502, 507A, 717, 744
Baudelot	Flooded	Low-pressure float	10 to 100	22, 717
	Direct-expansion	Thermal expansion valve	5 to 25	12, 22, 134a, 717
Shell-and-coil	—	Thermal expansion valve	2 to 10	12, 22, 134a, 717

where

q = heat transfer rate, Btu/h (W)

U_o = overall heat transfer coefficient based on outside surface, Btu/h·ft^2·°F [W/(m^2·K)]

A_o = outside surface area, ft^2 (m^2)

Δt_m = logarithmic mean temperature difference, °F (°C)

Details on determining these quantities are given in Chapter 4 of the 2017 *ASHRAE Handbook—Fundamentals*. Listed in Table 18-4 are approximate minimum and maximum values for U_o.

18.1.5 Refrigerants

The choice of a refrigerant for a particular application often depends on properties not directly related to its ability to remove heat. Such properties are flammability, toxicity, density, viscosity, availability, and environmental acceptability. As a rule, the selection of a refrigerant is a compromise between conflicting desirable properties. For example, the pressure in the evaporator should be as high as possible, and at the same time, a low condensing pressure is desirable.

Tables 18-5 and 18-6 provide the ASHRAE standard designation of refrigerant and refrigerant blend data and safety classifications given in ANSI/ASHRAE Standard 34. Table 18-7 lists the basic physical properties of these refrigerants. Table 18-8 gives the comparative refrigerant performance per unit (ton) of refrigeration.

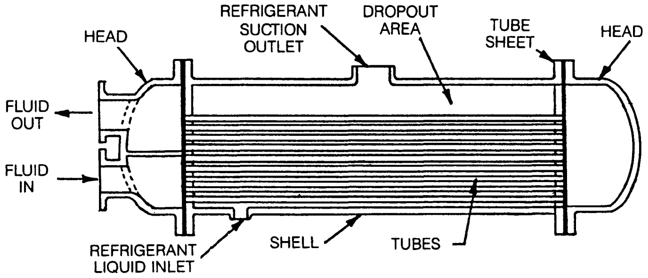


Fig. 18-36 Flooded Shell-and-Tube Cooler
(Figure 2, Chapter 42, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

A discussion of the properties of various refrigerants, as well as their relative performance characteristics, is presented in Chapter 29 of the 2017 *ASHRAE Handbook—Fundamentals*. Complete thermodynamic and thermophysical properties for the refrigerants may be found in Chapter 30 of the 2017 *ASHRAE Handbook—Fundamentals*.

18.2 Absorption Air-Conditioning and Refrigeration Equipment

Absorption refrigeration cycles are heat-operated cycles in which a secondary fluid, the **absorbent**, is used to absorb the **primary fluid**, a gaseous refrigerant, which has been vaporized in the evaporator. The basic absorption cycle is shown in Figure 18-38.

Chapter 2 of the 2017 *ASHRAE Handbook—Fundamentals* discusses operating principles and thermodynamics of

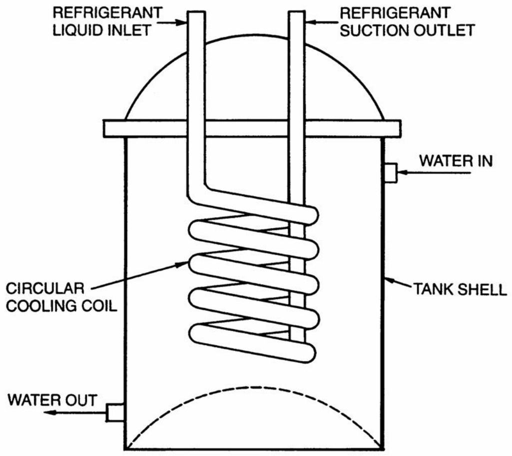


Fig. 18-37 Shell-and-Coil Cooler
(Figure 5, Chapter 42, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Table 18-4 Overall Heat Transfer Coefficients for Liquid Coolers

Type of Evaporator	Overall U, Btu/h·ft^2·°F (W/(m^2·K)) Minimum	Overall U, Btu/h·ft^2·°F (W/(m^2·K)) Maximum	Surface Side Basis for U
Flooded shell-and-plain-tube (water to Refrigerants 12, 22, and 717)	130 (740)	190 (1080)	Refrigerant
Flooded shell-and-finned-tube (water to Refrigerants 12, 22, or 500)	90 (510)	170 (970)	Refrigerant
Flooded shell-and-plain-tube (brine to Refrigerant 717)	45 (260)	100 (570)	Refrigerant
Flooded shell-and-plain-tube (brine to Refrigerants 12, 22, or 502)	30 (170)	90 (510)	Refrigerant
Direct-expansion, shell-and-plain-tube (water to Refrigerants 12, 22, and 717) (Refrigerant in Tubes)	80 (450)	220 (1250)	Liquid
Direct-expansion, shell-and-internal-finned-tubes (water to Refrigerants 12 or 22) (Refrigerant in Tubes)	160 (910)	250 (1420)	Liquid
Direct-expansion, shell-and-plain-tube (brine to Refrigerants 12, 22, 717, or 502) (Refrigerant in Tubes)	60 (340)	140 (790)	Liquid
Direct-expansion, shell-and-internal-finned-tubes (nonsalt brines to Refrigerants 12, 22, or 502)	60 (340)	170 (970)	Liquid
Shell-and-plain-tube coil (water in shell) (Refrigerant 12, 22, or 717 in coil)	10 (57)	25 (140)	Liquid
Baudelot cooler, flooded (Refrigerant 12 or 22 to water)	100 (570)	200 (1130)	Liquid
Baudelot cooler, direct expansion (Refrigerant 717 to water)	60 (340)	150 (850)	Liquid
Baudelot cooler, direct expansion (Refrigerant 12 or 22 to water)	60 (340)	120 (680)	Liquid
Double-pipe cooler (Refrigerant 717 to water)	50 (280)	150 (850)	Liquid
Double-pipe cooler (Refrigerant 717 to water)	50 (280)	125 (710)	Liquid
Tank-and-agitator, coil type water cooler (flooded, Refrigerant 717)	80 (450)	125 (710)	Liquid
Tank-and-agitator, coil type water cooler (flooded, Refrigerant 12, 22, or 500)	60 (340)	100 (570)	Liquid
Tank, ammonia (Refrigerant 717) to brine cooling, coils between cans in ice	15 (85)	40 (230)	Liquid
Tank-and-agitator, coil type water cooler (flooded, Refrigerant 717)	80 (450)	110 (620)	Liquid

Table 18-5 Refrigerant Data and Safety Classifications

Refrigerant Number	Chemical Name[a,b]	Chemical Formula[a]	Molecular Mass[a]	Normal Boiling Point,[a] °F°C	Safety Group
Methane Series					
11	Trichlorofluoromethane	CCl_3F	137.4	7524	A1
12	Dichlorodifluoromethane	CCl_2F_2	120.9	–22–30	A1
12B1	Bromochlorodifluoromethane	$CBrClF_2$	165.4	25–4	
13	Chlorotrifluoromethane	$CClF_3$	104.5	–115–81	A1
13B1	Bromotrifluoromethane	$CBrF_3$	148.9	–72–58	A1
14	Tetrafluoromethane (carbon tetrafluoride)	CF_4	88.0	–198–128	A1
21	Dichlorofluoromethane	$CHCl_2F$	102.9	489	B1
22	Chlorodifluoromethane	$CHClF_2$	86.5	–41	A1
23	Trifluoromethane	CHF_3	70.0	–116–82	A1
30	Dichloromethane (methylene chloride)	CH_2Cl_2	84.9	10440	B2
31	Chlorofluoromethane	CH_2ClF	68.5	16–9	
32	Difluoromethane (methylene fluoride)	CH_2F_2	52.0	–62–52	A2L
40	Chloromethane (methyl chloride)	CH_3Cl	50.4	–12–24	B2
41	Fluoromethane (methyl fluoride)	CH_3F	34.0	–109–78	
50	Methane	CH_4	16.0	–259–161	A3
Ethane Series					
113	1,1,2-trichloro-1,2,2-trifluoroethane	CCl_2FCClF_2	187.4	11848	A1
114	1,2-dichloro-1,1,2,2-tetrafluoroethane	$CClF_2CClF_2$	170.9	384	A1
115	Chloropentafluoroethane	$CClF_2CF_3$	154.5	–38–39	A1
116	Hexafluoroethane	CF_3CF_3	138.0	–109–78	A1
123	2,2-dichloro-1,1,1-trifluoroethane	$CHCl_2CF_3$	153.0	8127	B1
124	2-chloro-1,1,1,2-tetrafluoroethane	$CHClFCF_3$	136.5	10–12	A1
125	Pentafluoroethane	CHF_2CF_3	120.0	–55–48	A1
134a	1,1,1,2-tetrafluoroethane	CH_2FCF_3	102.0	–15–26	A1
141b	1,1-dichloro-1-fluoroethane	CH_3CCl_2F	117.0	9032	
142b	1-chloro-1,1-difluoroethane	CH_3CClF_2	100.5	14–10	A2
143a	1,1,1-trifluoroethane	CH_3CF_3	84.0	–53–47	A2L
152a	1,1-difluoroethane	CH_3CHF_2	66.0	–11–24	A2
170	Ethane	CH_3CH_3	30.0	–128–89	A3

Table 18-5 Refrigerant Data and Safety Classifications *(Continued)*

Refrigerant Number	Chemical Name[a,b]	Chemical Formula[a]	Molecular Mass[a]	Normal Boiling Point,[a] °F°C	Safety Group
Ethers					
E170	Dimethyl ether	CH_3OCH_3	46.0	–13–25	A3
Propane Series					
218	Octafluoropropane	$CF_3CF_2CF_3$	188.0	–35–37	A1
227ea	1,1,1,2,3,3,3-heptafluoropropane	CF_3CHFCF_3	170.0	3–16	A1
236fa	1,1,1,3,3,3-hexafluoropropane	$CF_3CH_2CF_3$	152.0	29–1	A1
245fa	1,1,1,3,3-pentafluoropropane	$CF_3CH_2CHF_2$	134.0	5915	B1
290	Propane	$CH_3CH_2CH_3$	44.0	–44–42	A3
Cyclic Organic Compounds (see Table 2 for blends)					
C318	Octafluorocyclobutane	$–(CF_2)_4–$	200.0	21–6	A1
Miscellaneous Organic Compounds					
Hydrocarbons					
600	Butane	$CH_3CH_2CH_2CH_3$	58.1	310	A3
600a	2-methylpropane (isobutane)	$CH(CH_3)_2CH_3$	58.1	11–12	A3
601	Pentane	$CH_3(CH_2)_3CH_3$	72.15	9736.1	A3
601a	2-methylbutane (isopentane)	$(CH_3)_2CHCH_2CH_3$	72.15	8227.8	A3
Oxygen Compounds					
610	Ethyl ether	$CH_3CH_2OCH_2CH_3$	74.1	9435	
611	Methyl formate	$HCOOCH_3$	60.0	8932	B2
Sulfur Compounds					
620	(Reserved for future assignment)				
Nitrogen Compounds					
630	Methanamine (methyl amine)	CH_3NH_2	31.1	20–7	
631	Ethanamine (ethyl amine)	$CH_3CH_2(NH_2)$	45.1	6217	
Inorganic Compounds					
702	Hydrogen	H_2	2.0	–423–253	A3
704	Helium	He	4.0	–452–269	A1
717	Ammonia	NH_3	17.0	–28–33	B2L
718	Water	H_2O	18.0	212100	A1
720	Neon	Ne	20.2	–411–246	A1
728	Nitrogen	N_2	28.1	–320–196	A1
732	Oxygen	O_2	32.0	–297–183	
740	Argon	Ar	39.9	–303–186	A1
744	Carbon dioxide	CO_2	44.0	–109–78[c]	A1
744A	Nitrous oxide	N_2O	44.0	–129–90	
764	Sulfur dioxide	SO_2	64.1	14–10	B1
Unsaturated Organic Compounds					
1150	Ethene (ethylene)	$CH_2=CH_2$	28.1	–155–104	A3
1234yf	2,3,3,3-tetrafluoro-1-propene	$CF_3CF=CH_2$	114.0	–20.9–29.4	A2L
1234ze(E)	Trans-1,3,3,3-tetrafluoro-1-propene	$CF_3CH=CHF$	114.0	–2.2–19.0	A2L
1270	Propene (propylene)	$CH_3CH=CH_2$	42.1	–54–48	A3

Source: ANSI/ASHRAE *Standard* 34-2010.

[a] Chemical name, chemical formula, molecular mass, and normal boiling point are not part of this standard.

[b] Preferred chemical name is followed by the popular name in parentheses.

[c] Sublimes.

Table 18-6 Data and Safety Classifications for Refrigerant Blends

(Table 2, Chapter 29, 2013 ASHRAE Handbook—Fundamentals)

Refrigerant Number	Composition (Mass %)	Composition Tolerances	Molecular Mass[a]	Normal Bubble Point, °F	Normal Dew Point, °F	Safety Group
Zeotropes						
400	R-12/114 (must be specified)					A1
401A	R-22/152a/124 (53.0/13.0/34.0)	(±2.0 /+0.5,–1.5/±1.0)	94.4	–29.9	–19.8	A1
401B	R-22/152a/124 (61.0/11.0/28.0)	(±2/+0.5,–1.5/±1.0)	92.8	–32.3	–23.4	A1
401C	R-22/152a/124 (33.0/15.0/52.0)	(±2/+0.5,1.5/±1.0)	101	–22.9	–10.8	A1
402A	R-125/290/22 (60.0/2.0/38.0)	(±2.0/+0.1,–1.0/±2.0)	101.6	–56.6	–52.6	A1
402B	R-125/290/22 (38.0/2.0/60.0)	(±2/+0.1,–1/±2)	94.7	–53.0	–48.8	A1

Table 18-6 Data and Safety Classifications for Refrigerant Blends ***(Continued)***

(Table 2, Chapter 29, 2013 ASHRAE Handbook—Fundamentals)

Refrigerant Number	Composition (Mass %)	Composition Tolerances	Molecular Mass[a]	Normal Bubble Point, °F	Normal Dew Point, °F	Safety Group
403A	R-290/22/218 (5.0/75.0/20.0)	(+0.2,–2/±2/±2)	92	–47.2	–44.1	A1
403B	R-290/22/218 (5.0/56.0/39.0)	(+0.2,–2/±2/±2)	103.3	–46.8	–44.1	A1
404A	R-125/143a/134a (44.0/52.0/4.0)	(±2/±1/±2)	97.6	–51.9	–50.4	A1
405A	R-22/152a/142b/C318 (45.0/7.0/5.5/42.5)	(±2/±1/±1 /±2) sum of R-152a and R-142b = (+0.0, –2.0)	111.9	–27.2	–12.1	
406A	R-22/600a/142b (55.0/4.0/41.0)	(±2/±1/±1)	89.9	–26.9	–10.3	A2
407A	R-32/125/134a (20.0/40.0/40.0)	(±2/±2/±2)	90.1	–49.4	–37.7	A1
407B	R-32/125/134a (10.0/70.0/20.0)	(±2/±2/±2)	102.9	–52.2	–44.3	A1
407C	R-32/125/134a (23.0/25.0/52.0)	(±2/±2/±2)	86.2	–46.8	–34.1	A1
407D	R-32/125/134a (15.0/15.0/70.0)	(±2/±2/±2)	91	–38.9	–26.9	A1
407E	R-32/125/134a (25.0/15.0/60.0)	(±2,±2,±2)	83.8	–45.0	–32.1	A1
407F	R-32/125/134a (30.0/30.0/40.0)	(±2,±2,±2)	82.1	–51.0	–39.5	A1
408A	R-125/143a/22 (7.0/46.0/47.0)	(±2/±1/±2)	87	–49.9	–49.0	A1
409A	R-22/124/142b (60.0/25.0/15.0)	(±2/±2/±1)	97.4	–31.7	–17.5	A1
409B	R-22/124/142b (65.0/25.0/10.0)	(±2/±2/±1)	96.7	–33.7	–21.5	A1
410A	R-32/125 (50.0/50.0)	(+0.5,–1.5/+1.5,–0.5)	72.6	–60.9	–60.7	A1
410B	R-32/125 (45.0/55.0)	(±1/±1)	75.6	–60.7	–60.5	A1
411A	R-1270/22/152a (1.5/87.5/11.0)	(+0,–1/+2,–0/+0,–1)	82.4	–39.5	–35.0	A2
411B	R-1270/22/152a (3.0/94.0/3.0)	(+0,–1/+2,–0/+0,–1)	83.1	–42.9	–42.3	A2
412A	R-22/218/142b (70.0/5.0/25.0)	(±2/±2/±1)	92.2	–33.5	–19.8	A2
413A	R-218/134a/600a (9.0/88.0/3.0)	(±1/±2/±0,–1)	104	–20.7	–17.7	A2
414A	R-22/124/600a/142b (51.0/28.5/4.0/16.5)	(±2/±2/±0.5/+0.5,–1)	96.9	–29.2	–14.4	A1
414B	R-22/124/600a/142b (50.0/39.0/1.5/9.5)	(±2/±2/±0.5/+0.5,–1)	101.6	–29.9	–15.0	A1
415A	R-22/152a (82.0/18.0)	(±1/±1)	81.9	–35.5	–30.5	A2
415B	R-22/152a (25.0/75.0)	(±1/±1)	70.2	–17.8	–15.2	A2
416A	R-134a/124/600 (59.0/39.5/1.5)	(+0.5,–1/+1,–0.5/+1,–0.2)	111.9	–10.1	–7.2	A1
417A	R-125/134a/600 (46.6/50.0/3.4)	(±1.1/±1/+0.1,0.4)	106.7	–36.4	–27.2	A1
417B	R-125/134a/600 (79.0/18.3/2.7)	(±1/±1/+0.1,–0.5)	113.1	–48.8	–42.7	A1
418A	R-290/22/152a (1.5/96.0/2.5)	(±0.5/±1/±0.5)	84.6	–42.2	–40.2	A2
419A	R-125/134a/E170 (77.0/19.0/4.0)	(±1/±1/±1)	109.3	–44.7	–32.8	A2
420A	R-134a/142b (88.0/12.0)	(±1,–0/+0,–1)	101.8	–13.0	–11.6	A1
421A	R-125/134a (58.0/42.0)	(±1/±1)	111.8	–41.5	–31.9	A1
421B	R-125/134a (85.0/15.0)	(±1/±1)	116.9	–50.2	–44.6	A1
422A	R-125/134a/600a (85.1/11.5/3.4)	(±1/±1/+0.1,–0.4)	113.6	–51.7	–47.4	A1
422B	R-125/134a/600a (55.0/42.0/3.0)	(±1/±1/+0.1,–0.5)	108.5	–40.9	–32.2	A1
422C	R-125/134a/600a (82.0/15.0/3.0)	(±1/±1/+0.1,–0.5)	116.3	–49.5	–44.2	A1
422D	R-125/134a/600a (65.1/31.5/3.4)	(+0.9,–1.1/±1/+0.1,–0.4)	109.9	–45.8	–37.1	A1
423A	R-134a/227ea (52.5/47.5)	(±1/±1)	126	–11.6	–10.3	A1
424A	R-125/134a/600a/600/601a (50.5/47.0/0.9/1.0/0.6)	(±1/±1/+0.1,–0.2/+0.1,–0.2/+0.1,–0.2)	108.4	–38.4	–27.9	A1
425A	R-32/134a/227ea (18.5/69.5/12.0)	(±0.5/±0.5/±0.5)	90.3	–36.6	–24.3	A1
426A[a]	R-125/134a/600a/601a (5.1/93.0/1.3/0.6)	(±1/±1/+0.1,–0.2/+0.1,–0.2)	101.6	–19.3	–16.1	A1
427A[a]	R-32/125/143a/134a (15.0/25.0/10.0/50.0)	(±2/±2/±2/±2)	90.4	–45.4	–33.3	A1
428A[a]	R-125/143a/290/600a (77.5/20.0/0.6/1.9)	(±1/±1/+0.1,–0.2/+0.1,–0.2)	107.5	–54.9	–53.5	A1
429A	R-E170/152a/600a (60.0/10.0/30.0)	(±1/±1/±1)	50.8	–14.8	–14.1	A3
430A	R-152a/600a (76.0/24.0)	(±1/±1)	64	–17.7	–17.3	A3
431A	R-290/152a (71.0/29.0)	(±1/±1)	48.8	–45.6	–45.6	A3
432A	R-1270/E170 (80.0/20.0)	(±1/±1)	42.8	–51.9	–50.1	A3
433A	R-1270/290 (30.0/70.0)	(±1/±1)	43.5	–48.3	–47.6	A3
433B	R-1270/290 (5.0/95.0)	(±1/±1)	44	–44.9	–44.5	A3
433C	R-1270/290 (25.0/75.0)	(±1/±1)	43.6	–47.7	–47.0	A3
434A	R-125/143a/134a/600a (63.2/18.0/16.0/2.8)	(±1/±1/±1/+0.1,–0.2)	105.7	–49.0	–44.1	A1
435A	R-E170/152a (80.0/20.0)	(±1/±1)	49.04	–15.0	–14.6	A3
436A	R-290/600a (56.0/44.0)	(±1/±1)	49.33	–29.7	–16.2	A3
436B	R-290/600a (52.0/48.0)	(±1/±1)	49.87	–28.1	–13.0	A3
437A	R-125/134a/600/601 (19.5/78.5/1.4/0.6)	(+0.5,–1.8/+1.5,–0.7/+0.1,–0.2/+0.1/–0.2)	103.7	–27.2	–20.6	A1
438A	R-32/125/134a/600/601a (8.5/45.0/44.2/1.7/0.6)	(+0.5,–1.5/±1.5/±1.5/+0.1,–0.2/+0.1/–0.2)	99.1	–45.4	–33.5	A1
439A	R-32/125/600a (50.0/47.0/3.0)	(±1/±1)	71.2	–61.6	–61.2	A2
440A	R-290/134a/152a (0.6/1.6/97.8)	(±0.1/±0.6/±0.5)	66.2	–13.9	–11.7	A2
441A	R-170/290/600a/600 (3.1/54.8/6.0/36.1)	(±0.3/±2/±0.6/±2)	48.2	–43.4	–4.7	A3
442A	R-32/125/134a/152a/227ea (31.0/31.0/30.0/3.0/5.0)	(±1.0/±1.0/±1.0/+0.5/±1.0)	81.77	–51.7	–39.8	A1

Table 18-6 Data and Safety Classifications for Refrigerant Blends *(Continued)*

(Table 2, Chapter 29, 2013 ASHRAE Handbook—Fundamentals)

Refrigerant Number	Composition (Mass %)	Composition Tolerances	Azeotropic Temperatures, °F	Molecular Mass[a]	Normal Boiling Point, °F	Safety Group
Azeotropes[b]						
500	R-12/152a (73.8/26.2)		32	99.3	–27	A1
501	R-22/12 (75.0/25.0)[c]		–42	93.1	–42	A1
502	R-22/115 (48.8/51.2)		66	112.0	–49	A1
503	R-23/13 (40.1/59.9)		–126	87.5	–126	
504	R-32/115 (48.2/51.8)		63	79.2	–71	
505	R-12/31 (78.0/22.0)[c]		239	103.5	–22	
506	R-31/114 (55.1/44.9)		64	93.7	10	
507A[d]	R-125/143a (50.0/50.0)		–40	98.9	–52.1	A1
508A[d]	R-23/116 (39.0/61.0)		–122	100.1	–122	A1
508B	R-23/116 (46.0/54.0)		–50.1	95.4	–126.9	A1
509A[d]	R-22/218 (44.0/56.0)		32	124.0	–53	A1
510A	R-E170/600a (88.0/12.0)	(±0.5/±0.5)	–13.4	47.24	–13.4	A3
511A	R-290/E170 (95.0/5.0)	(±1/±1)	–4 to 104	44.19	–43.7	A3
512A	R-134a/152a (5.0/95.0)	(±1/±1)	–4 to 104	67.24	–11.2	A2
513A	R-1234yf/134a (56.0/44.0)	(±1.0/±1.0)	81	108.4	–20.4	A1

Source: ANSI/ASHRAE *Standard* 34-2010.

[a] Molecular mass and normal boiling point are not part of this standard.

[b] Azeotropic refrigerants exhibit some segregation of components at conditions of temperature and pressure other than those at which they were formulated. Extent of segregation depends on the particular azeotrope and hardware system configuration.

[c] Exact composition of this azeotrope is in question, and additional experimental studies are needed.

[d] R-507, R-508, and R-509 are allowed designations for R-507A, R-508A, and R-509A because of a change in designations after assignment of R-500 through R-509. Corresponding changes were not made for R-500 through R-506.

Table 18-7 Physical Properties of Selected Refrigerants[a]

(Table 5, Chapter 29, 2013 ASHRAE Handbook—Fundamentals)

Refrigerant									
Number	Chemical Name or Composition (% by Mass)	Chemical Formula	Molecular Mass	Boiling Pt.[f] (NBP) at 14.696 psia, °F101.325 kPa, °C	Freezing Point, °F °C	Critical Temperature, °F °C	Critical Pressure, psik Pa	Critical Density, lb/ft^3 kg/m^3	Refractive Index of Liquid[b,c]
728	Nitrogen	N_2	28.013	–320.44–195.8	–346–210.0	–232.528–146.96	492.53395.8	19.56313.3	1.205 (83 K) 589.3 nm
729	Air	—	28.959	–317.65–194.25	—	–221.062–140.59	549.63789.6	20.97335.94	—
740	Argon	Ar	39.948	–302.53–185.85	–308.812–189.34	–188.428–122.46	705.34863.0	33.44535.6	1.233 (84 K) 589.3 nm
732	Oxygen	O_2	31.999	–297.328–182.96	–361.822–218.79	–181.426–118.57	731.45043.0	27.23436.14	1.221 (92 K) 589.3 nm
50	Methane	CH_4	16.043	–258.664–161.48	–296.428–182.46	–116.6548–82.586	667.14599.2	10.15162.66	—
14	Tetrafluoromethane	CF_4	88.005	–198.49–128.05	–298.498–183.61	–50.152–45.64	543.93750.0	39.06625.66	—
170	Ethane	C_2H_6	30.07	–127.4764–88.581	–297.01–182.8	89.92432.72	706.64872.2	12.87206.18	—
508A	R-23/116 (39/61)	—	100.1	–125.73–87.60	—	50.34610.192	529.53650.8	35.43567.58	—
508B	R-23/116 (46/54)	—	95.394	–125.68–87.6	—	52.17011.205	547.03771.6	35.49568.45	—
23	Trifluoromethane	CHF_3	70.014	–115.6324–82.018	–247.234–155.13	79.057426.143	700.84832	32.87526.5	—
13	Chlorotrifluoromethane	$CClF_3$	104.46	–114.664–81.48	–294.07–181.15	83.9328.85	562.63879	36.39582.88	1.146 (25)[2]
744	Carbon dioxide	CO_2	44.01	–109.12–78.4[d]	–69.8044–56.558[e]	87.760430.978	1070.07377.3	29.19467.6	1.195 (15)
504	R-32/115 (48.2/51.8)	—	79.249	–72.23–57.906	—	143.8562.138	642.34428.8	31.51504.68	—
32	Difluoromethane	CH_2F_2	52.024	–60.9718–51.651	–214.258–136.81	172.58978.105	838.65782.0	26.47424	—
410A	R-32/125 (50/50)	—	72.585	–60.5974–51.446	—	160.444471.358	711.14902.6	28.69459.53	—
125	Pentafluoroethane	C_2HF_5	120.02	–54.562–48.09	–149.134–100.63	150.841466.023	524.73617.7	35.81573.58	—
1270	Propylene	C_3H_6	42.08	–53.716–47.62	–301.35–185.2	195.9191.061	660.64554.8	14.36230.03	1.3640 (–50)[1]
143a	Trifluoroethane	CH_3CF_3	84.041	–53.0338–47.241	–169.258–111.81	162.872672.707	545.53761.0	26.91431.0	—
507A	R-125/143a (50/50)	—	98.859	–52.1338–46.741	—	159.110670.617	537.43705	30.64490.77	—
404A	R-125/143a/134a (44/52/4)	—	97.604	–51.1996–46.222	—	161.682872.046	540.83728.9	30.37486.53	—
502	R-22/115 (48.8/51.2)	—	111.63	–49.3132–45.174	—	178.7180.507	582.64016.8	35.50568.70	—
407C	R-32/125/134a (23/25/52)	—	86.204	–46.5286–43.627	—	186.861286.034	671.54629.8	30.23484.23	—
290	Propane	C_3H_8	44.096	–43.805–42.11	–305.72–187.62	206.1396.74	616.584251.2	13.76220.4	1.3397 (–42)
22	Chlorodifluoromethane	$CHClF_2$	86.468	–41.458–40.81	–251.356–157.42	205.06196.145	723.74990.0	32.70523.84	1.234 (25)[2]
115	Chloropentafluoroethane	$CClF_2CF_3$	154.47	–38.65–39.25	–146.92–99.39	175.9179.95	453.83129.0	38.38614.8	1.221 (25)[2]

Table 18-7 Physical Properties of Selected Refrigerants[a] *(Continued)*

(Table 5, Chapter 29, 2013 ASHRAE Handbook—Fundamentals)

Refrigerant		Chemical Formula	Molecular Mass	Boiling Pt.[f] (NBP) at 14.696 psia, °F101.325 kPa, °C	Freezing Point, °F °C	Critical Temperature, °F °C	Critical Pressure, psik Pa	Critical Density, lb/ft^3 kg/m^3	Refractive Index of Liquid[b,c]
Number	Chemical Name or Composition (% by Mass)								
500	R-12/152a (73.8/26.2)	—	99.303	–28.4854–33.603	—	215.762102.09	604.64168.6	30.91495.1	—
717	Ammonia	NH_3	17.03	–27.9886–33.327	–107.779–77.655	270.05132.25	1643.711 333.0	14.05225.0[d]	1.325 (16.5)
12	Dichlorodifluoromethane	CCl_2F_2	120.91	–21.5536–29.752	–250.69–157.05	233.546111.97	599.94136.1	35.27565.0	1.288 (25)[2]
R-1234yf	2,3,3,3-tetrafluoroprop-1-ene	$CF_3CF{=}CH_2$	114.04	–21.01–29.45		202.4694.7	490.553382.2	29.668475.55	
134a	Tetrafluoroethane	CF_3CH_2F	102.03	–14.9332–26.074	–153.94–103.3	213.908101.06	588.84059.3	31.96511.9	—
152a	Difluoroethane	CHF_2CH_3	66.051	–11.2414–24.023	–181.462–118.59	235.868113.26	655.14516.8	22.97368	—
R-1234ze (E)	Trans-1,3,3,3-tetrafluoropropene	$CF_3CH{=}CHF$	114.04	–2.11–18.95		228.87109.37	527.293636.3	30.542489.24	
124	Chlorotetrafluoroethane	$CHClFCF_3$	136.48	10.4666–11.963	–326.47–199.15	252.104122.28	525.73624.3	34.96560.0	—
600a	Isobutane	C_4H_{10}	58.122	10.852–11.75	–254.96–159.42	274.39134.66	526.343629.0	14.08225.5	1.3514 (–25)[1]
142b	Chlorodifluoroethane	$CClF_2CH_3$	100.5	15.53–9.15	–202.774–130.43	278.798137.11	590.34055.0	27.84466.0	—
C318	Octafluorocyclobutane	C_4F_8	200.03	21.245–5.975	–39.64–39.8	239.414115.23	402.82777.5	38.70619.97	
600	Butane	C_4H_{10}	58.122	31.118–0.49	–216.86–138.27	305.564151.98	550.63796.0	14.23227.94	1.3562 (–15)[1]
114	Dichlorotetrafluoroethane	$CClF_2CClF_2$	170.92	38.45483.586	–134.54–92.5	294.224145.68	472.43257.0	36.21579.97	1.294 (25)
11	Trichlorofluoromethane	CCl_3F	137.37	74.674423.708	–166.846–110.47	388.328197.96	639.34407.6	34.59554.0	1.362 (25)[2]
123	Dichlorotrifluoroethane	$CHCl_2CF_3$	152.93	82.0827.823	–160.87–107.15	362.624183.68	531.13661.8	34.34550.0	—
141b	Dichlorofluoroethane	CCl_2FCH_3	116.95	89.6932.05	–154.25–103.5	399.83204.4	610.94212.0	28.63458.6	—
113	Trichlorotrifluoroethane	CCl_2FCClF_2	187.38	117.65347.585	–33.196–36.22	417.308214.06	492.03392.2	34.96560.0	1.357 (25)[2]
718[3]	Water	H_2O	18.015	211.953299.974	32.0180.01	705.11373.95	3200.122 064.0	20.10322.0	—

Notes:

[a] Data from NIST (2010) REFPROP v. 9.0.

[b] Temperature of measurement (°C, unless kelvin is noted) shown in parentheses. Data from CRC (1987), unless otherwise noted.

[c] For the sodium D line.

[d] Sublimes.

[e] At 76.4 psi527 kPa.

[f] Bubble point used for blends

References:

[1] Kirk and Othmer (1956).

[2] *Bulletin* B-32A (DuPont).

[3] *Handbook of Chemistry* (1967).

the basic absorption cycle and other information on the thermodynamics of workable absorbent-refrigerant combinations. A complete thermodynamic analysis of the absorption cycle is complex. However, a detailed analysis is not necessary to understanding the operating principles of the cycle.

The absorption cycle and the mechanical compression cycle have in common the evaporation and condensation of a refrigerant liquid; these processes occur at two pressure levels within the unit. The two cycles differ in that the absorption cycle uses a pump and a heat-operated generator to produce the pressure differential, whereas the mechanical compression cycle uses a compressor; the absorption cycle substitutes physiochemical processes for the purely mechanical processes of the compression cycle. Both cycles require energy for operation: heat in the absorption cycle, mechanical energy in the compression cycle.

Of the many combinations that have been tried, only the lithium bromide-water and the ammonia-water cycles remain in common use for air-conditioning. In addition, ammonia-water absorption equipment has been used in large industrial applications requiring low temperatures for process work.

Figure 18-39 is a typical schematic diagram of machines available in the form of indirect-fired liquid chillers in capacities of 50 to 1500 tons (180 to 5300 kW).

Generators (concentrators) are tube bundles submerged in the solution, heated by steam or hot liquids.

Condensers are tube bundles located in the vapor space over the generator and shielded from carryover of salt by eliminators. Cooling water to the condenser first passes through the absorber.

Absorbers are tube bundles over which strong absorbent is sprayed. Refrigerant vapor is condensed into the absorbent, releasing heat to the cooling water passing through.

Evaporators (coolers) are tube bundles over which the refrigerant water is sprayed and evaporated. The liquid to be cooled passes inside the tubes.

Solution heat exchangers are of all steel shell-and-tube construction.

Table 18-8 Comparative Refrigerant Performance per Ton of Refrigeration

Refrigerant Number	Refrigerant Chemical Name or Composition (% by mass)	Evaporator Pressure, psia	Condenser Pressure, psia	Compression Ratio	Net Refrigerating Effect, Btu/lb	Refrigerant Circulated, lb/min	Liquid Circulated, gal/min	Specific Volume of Suction Gas, ft^3/lb	Compressor Displacement, ft^3/min	Power Consumption, hp	Coefficient of Performance	Compressor Discharge Temp., °F
Evaporator –25°F/Condenser 86°F												
744	Carbon dioxide	195.7	1046.2	5.35	56.8	3.52	0.711	0.457	1.61	2.779	1.698	196.3
170	Ethane	146.8	675.1	4.6	66.0	3.03	1.314	0.878	2.66	2.805	1.681	136.2
1270	Propylene	28.8	189.3	6.57	115.7	1.73	0.416	3.63	6.28	1.637	2.88	120.3
507A	R-125/143a (50/50)	28.8	211.7	7.34	43.5	4.60	0.54	1.52	6.98	1.833	2.573	100.6
404A	R-125/143a/134a (44/52/4)	27.6	206.1	7.46	45.1	4.44	0.521	1.61	7.13	1.817	2.595	102.1
502	R-22/115 (48.8/51.2)	26.5	189.2	7.14	42.1	4.76	0.48	1.48	7.06	1.722	2.739	106.3
22	Chlorodifluoromethane	22.1	172.9	7.81	66.8	3.00	0.307	2.32	6.95	1.589	2.967	149.8
717	Ammonia	16.0	169.3	10.61	463.9	0.43	0.087	16.7	7.19	1.569	3.007	285.6
Evaporator 20°F/Condenser 86°F												
744	Carbon dioxide	421.9	1046.2	2.48	55.7	3.59	0.726	0.203	0.73	1.342	3.514	142.3
170	Ethane	293.6	675.1	2.3	70.1	2.85	1.238	0.421	1.20	1.314	3.588	115.8
32	Difluoromethane	94.7	279.6	2.95	111.2	1.80	0.229	0.902	1.62	0.797	5.924	139.4
410A	R-32/125 (50/50)	93.2	273.6	2.94	73.5	2.72	0.316	0.651	1.77	0.815	5.78	115.8
507A	R-125/143a (50/50)	72.9	211.7	2.9	49.4	4.05	0.476	0.616	2.50	0.848	5.564	93.5
404A	R-125/143a/134a (44/52/4)	70.5	206.1	2.92	51.1	3.92	0.46	0.649	2.54	0.842	5.598	94.3
1270	Propylene	69.1	189.3	2.74	126.6	1.58	0.381	1.58	2.50	0.79	5.975	102.8
502	R-22/115 (48.8/51.2)	66.3	189.2	2.86	47.1	4.25	0.429	0.619	2.63	0.813	5.799	95.8
22	Chlorodifluoromethane	57.8	172.9	2.99	71.3	2.80	0.287	0.935	2.62	0.772	6.105	118.0
407C	R-32/125/134a (23/25/52)	57.5	183.7	3.19	71.9	2.78	0.296	0.942	2.62	0.795	5.93	111.0
290	Propane	55.8	156.5	2.8	124.1	1.61	0.399	1.89	3.05	0.787	5.987	94.8
717	Ammonia	48.2	169.3	3.51	478.5	0.42	0.084	5.91	2.47	0.754	6.254	179.8
1234yf	2,3,3,3-tetrafluoropropene*	36.3	113.6	3.13	51.8	3.86	0.43	1.15	4.44	0.809	5.835	86.0
134a	Tetrafluoroethane	33.1	111.7	3.37	65.8	3.04	0.307	1.41	4.28	0.778	6.063	94.7
1234ze(E)	Trans-1,3,3,3-tetrafluoropropene*	24.4	83.9	3.44	60.0	3.33	0.349	1.74	5.81	0.782	6.03	86.0
600a	Isobutane*	17.9	58.7	3.29	119.5	1.67	0.368	4.78	7.99	0.764	6.171	86.0
Evaporator 45°F/Condenser 86°F												
32	Difluoromethane	147.7	279.6	1.89	112.2	1.78	0.223	0.577	1.03	0.445	10.602	116.4
410A	R-32/125 (50/50)	145.0	273.6	1.89	75.2	2.66	0.308	0.416	1.11	0.455	10.379	103.7
502	R-22/115 (48.8/51.2)	102.0	189.2	1.85	49.6	4.03	0.407	0.404	1.63	0.451	10.474	91.8
407C	R-32/125/134a (23/25/52)	92.8	183.7	1.98	74.7	2.68	0.284	0.588	1.57	0.443	10.655	102.7
22	Chlorodifluoromethane	90.8	172.9	1.9	73.5	2.72	0.279	0.604	1.64	0.433	10.885	104.5
290	Propane	85.3	156.5	1.84	130.7	1.53	0.379	1.26	1.92	0.439	10.743	90.7
717	Ammonia	81.0	169.3	2.09	484.9	0.41	0.083	3.61	1.49	0.421	11.186	137.4
500	R-12/152a (73.8/26.2)	66.5	127.6	1.92	64.7	3.09	0.331	0.725	2.24	0.432	10.925	94.2
1234yf	2,3,3,3-tetrafluoropropene*	58.1	113.6	1.96	55.5	3.61	0.402	0.726	2.62	0.444	10.623	86.0
12	Dichlorodifluoromethane	56.3	107.9	1.92	54.6	3.67	0.34	0.719	2.64	0.429	11.004	91.6
134a	Tetrafluoroethane	54.7	111.7	2.04	69.2	2.89	0.292	0.868	2.51	0.433	10.903	90.6
1234ze(E)	Trans-1,3,3,3-tetrafluoropropene*	40.6	83.9	2.06	64.1	3.12	0.327	1.07	3.34	0.433	10.899	86.0
600a	Isobutane*	29.2	58.7	2.01	127.4	1.57	0.345	3.01	4.72	0.425	11.084	86.0
600	Butane*	19.5	41.1	2.11	140.5	1.42	0.301	4.57	6.50	0.42	11.226	86.0
123	Dichlorotrifluoroethane	6.5	15.9	2.44	66.9	2.99	0.246	5.3	15.85	0.414	11.397	86.0
113	Trichlorotrifluoroethane*	3.1	7.9	2.57	59.2	3.38	0.26	9.41	31.81	0.413	11.409	86.0

*Superheat required

Source: Data from NIST CYCLE_D 4.0, zero subcool, zero superheat unless noted, no line losses, 100% efficiencies, average temperatures.

Solution and evaporator pumps are generally electric-motor-driven centrifugal pumps of hermetic design that use the cycle fluids for cooling and lubrication.

Purgers are used to remove noncondensable gases. Noncondensable gases present in small quantities can raise the total pressure in the absorber sufficiently to significantly change the evaporator pressure. Small pressure increases cause appreciable change in the refrigerant evaporating temperature.

Expansion devices commonly used in absorption machines are usually an orifice or fixed restriction, which controls the flow of refrigerant liquid between the condenser and the evaporator.

Lithium bromide-water cycle absorption machines meet load variations and maintain chilled water temperature control by varying the rate of reconcentration of the absorbent solution. At any given constant load, the chilled water temperature is maintained by a temperature difference between refrigerant and chilled water. In turn, the refrigerant temperature is maintained by the absorber being supplied with a flow rate and concentration of solution, and by the absorber cooling water temperature.

Load changes are reflected by corresponding changes in chilled water temperature. A load reduction, for example, results in less temperature difference needed in the evaporator and a reduced requirement for solution flow or concentration. The resultant chilled water temperature drop is met by adjust-

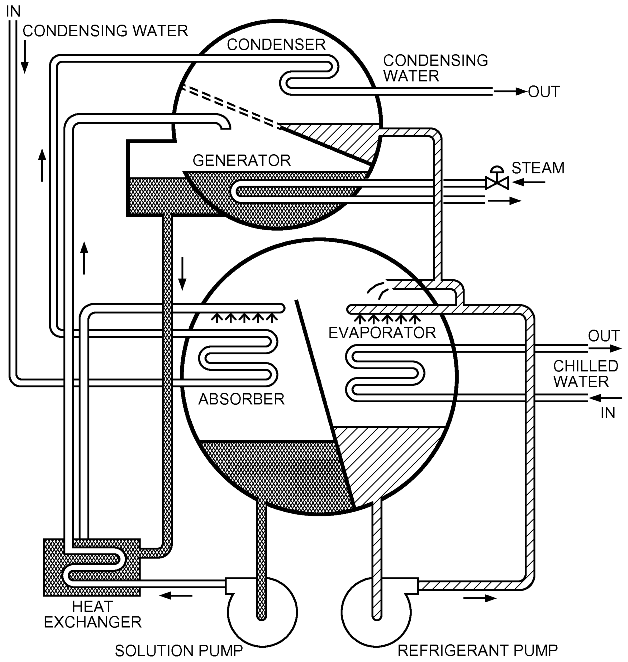


Fig. 18-38 Two-Shell Lithium Bromide Cycle Water Chiller
(Figure 2, Chapter 18, 2014 ASHRAE Handbook—Refrigeration)

ing the rate of reconcentration to match the reduced requirements of the absorber.

The coefficient of performance (COP) of a lithium bromide-water cycle absorption machine operating at 45° leaving chilled water temperature, 85° entering condenser water temperature and 12 psig steam pressure is typically in the range of 0.65 to 0.70. Whenever chilled water temperatures are above the nominal, or condensing water temperatures are below the nominal, a COP as high as 0.70 can be reached. Reversing the temperature conditions cited reduces the COP to below 0.60. A coefficient of performance of 0.68 corresponds approximately to a steam rate of 18 lb/h per ton of refrigeration (1.45 kW/kW).

Absorption machines can be made with a two-stage generator. Such a unit may be called **dual effect**. Figure 18-40 is a schematic diagram of a nominally single-shell design with a two-stage generator. The first-effect generator receives the external heat, which boils refrigerant from the weak absorbent. This hot refrigerant vapor then goes to a second generator and supplies heat for further refrigerant vaporization from the absorbent of intermediate concentration, which flows from the first generator and is cooled by passing through a first-stage heat economizer. Other than the generator, all components of the single-stage lithium bromide-water absorption units are common to the two-stage units. The advantage of the dual-effect unit is higher performance, with steam rates approximately two-thirds those of single-stage machines. Heat source temperature for the dual-effect unit is over 120°F (67°C) higher than for the single-effect unit, requiring higher steam pressures. Figure 18-41 illustrates performance characteristics of lithium-bromide-cycle water chillers.

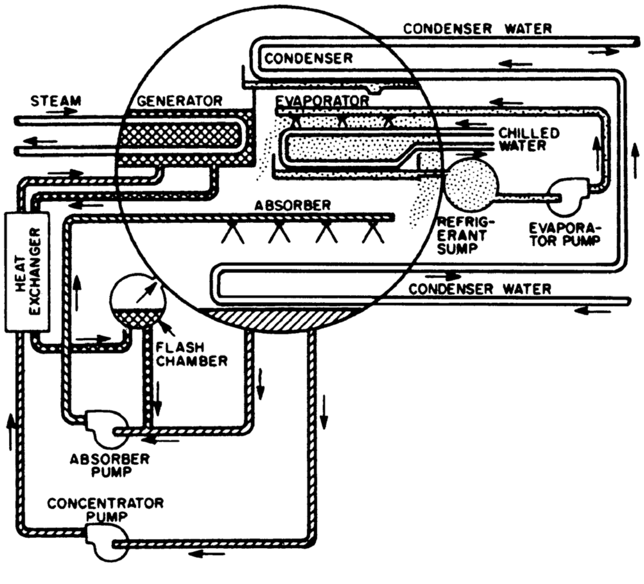


Fig. 18-39 Diagram of One-Shell Lithium Bromide Cycle Water Chiller

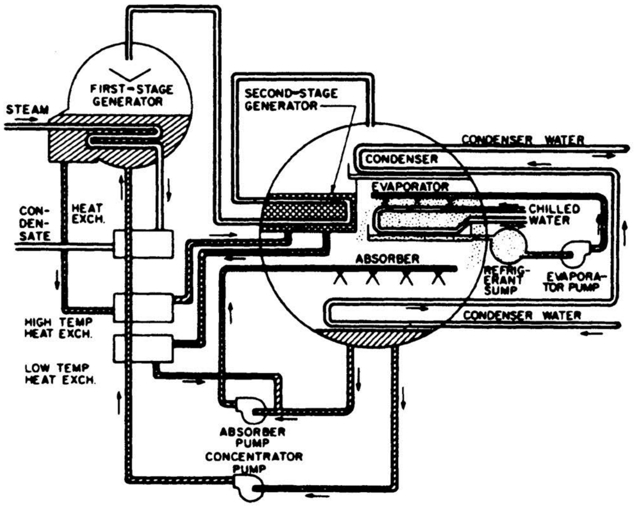


Fig. 18-40 Diagram of One-Shell Lithium Bromide Cycle Water Chiller with Two-Stage Generator

18.3 Cooling Towers

A cooling tower, through a combination of mass and energy transfer, cools water by exposing it as an extended surface to the atmosphere. Water to be cooled is distributed in the tower by spray nozzles, splash bars, or film-type fill, which exposes a very large water surface area to atmospheric

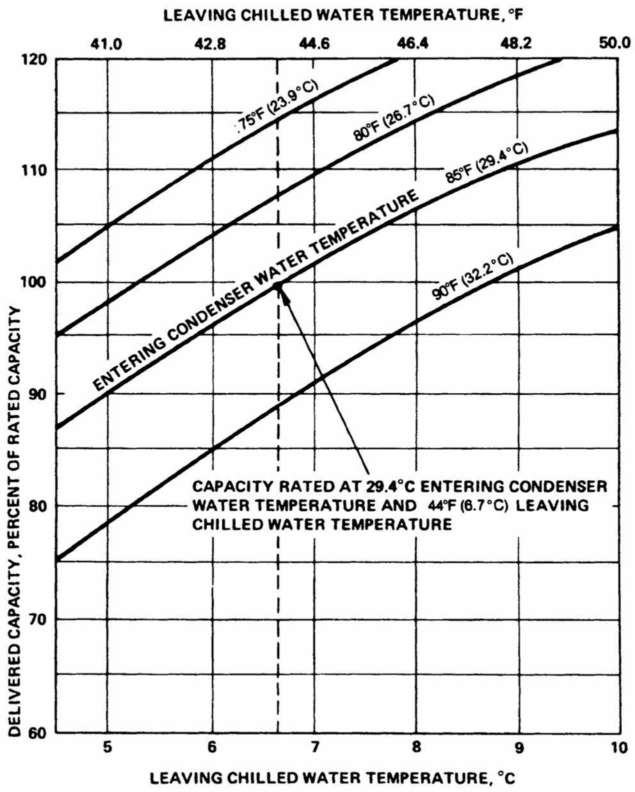


Fig. 18-41 Performance Characteristics of Lithium Bromide Cycle Water Chiller

air. The airflow may be caused by mechanical means, by convection currents due to variation in density, or by natural wind currents. The airflow is either crossflow or counterflow. *Crossflow* describes air flowing horizontally in the filled portion of the tower or normal to the water flow, whereas *counterflow* implies the airstream rises vertically or countercurrent to a falling stream of water.

Counterflow mechanical-draft towers are principally found in air-conditioning applications. The main advantage of counterflow is its adaptability to restrictive space limitations. Factory-assembled towers often use centrifugal blowers in forced-draft configurations. The field-erected designs are usually induced-draft units with axial flow fans.

Crossflow towers are widely used in air-conditioning, process, and industrial applications. Crossflow towers have: (1) low air-side pressure drop in relation to high transfer surface areas and (2) the inherent capability to obtain uniform distributional characteristics of both the air and water streams.

The thermal capability of any cooling tower may be defined by the following parameters:

- Entering and leaving water temperatures
- Entering air wet-bulb temperature
- Water flow rate

The variations in tower performance associated with changes in these parameters are discussed in Chapter 40 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

The thermal capability of cooling towers for air-conditioning applications is usually stated in terms of nominal refrigeration tonnage based on heat dissipation of 15,000 Btu/h (1.25 kW/kW) per ton and a water circulation rate of 3 gpm per ton (0.054 L/s per kW) cooled from 95 to 85°F (35 to 39.4°C) at 78°F (25.6°C) wet-bulb temperature. For industrial applications, nominal tonnage ratings are not used and the performance capability of the cooling tower is usually stated in terms of flow rate at specified operating conditions (entering and leaving water temperature and entering air wet-bulb temperature).

Fans in the mechanical-draft tower provide a positive and constant airflow. Since performance does not depend on the wind, mechanical-draft towers may be designed for exacting conditions. The fans may operate to provide forced or induced draft, depending on their location at the inlet or outlet of the tower. The tower may be crossflow (Figure 18-42) or counterflow (Figure 18-43). The addition of the fan makes it possible to design wider towers that are more compact than the tall, narrow atmospheric towers.

Factory-assembled cooling towers in both crossflow and counterflow designs are available. Water distribution is by gravity or low-pressure flume with crossflow design, and spray nozzles are used on counterflow units.

A major consideration in the selection of cooling towers is the power requirement per ton of refrigeration, since the tower is a parasitic energy burden.

18.3.1 Spray Ponds

Heat dissipates from the surface of a body of water by evaporation, radiation, and convection. A spray pond divides the water into small droplets, greatly extending the water surface and bringing it into contact with the air. Heat transfer is largely due to evaporative cooling. Temperature control, large space requirements, limited ability to approach the wet-bulb temperature, and winter operational difficulties have generally ruled out the spray pond in favor of more compact and more controllable mechanical-draft or hyperbolic towers.

18.4 Problems

18.1 A condenser used in a refrigeration system has a capacity of 10 tons at a 40°F evaporating temperature. When 20 gpm of cooling water enters at 75°F, the condensing temperature is 90°F. The manufacturer claims a U-factor of 95 Btu/h·ft^2·°F, with a heat transfer area of 83 ft^2. Are these claims reasonable? Why?

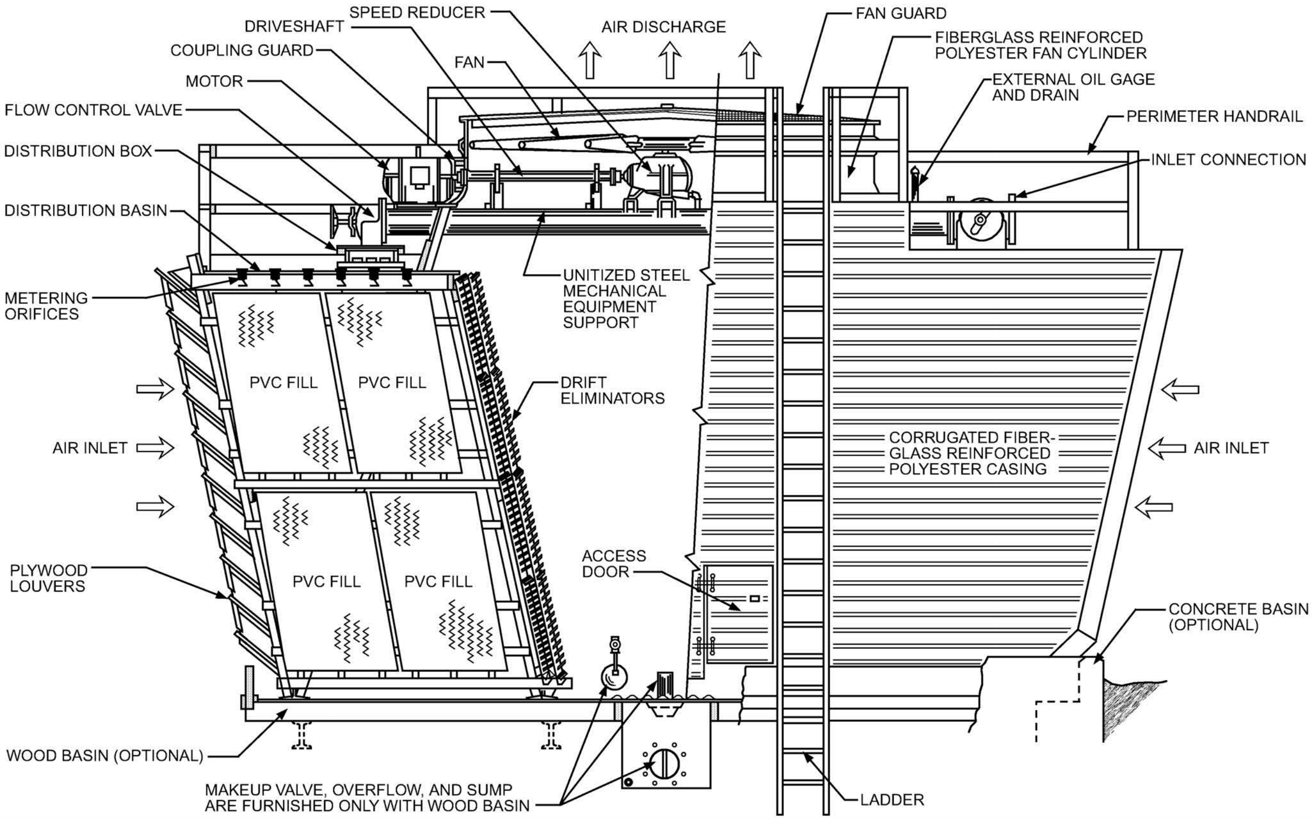


Fig. 18-42 Crossflow Induced-Draft Tower

(Figure 13, Chapter 40, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

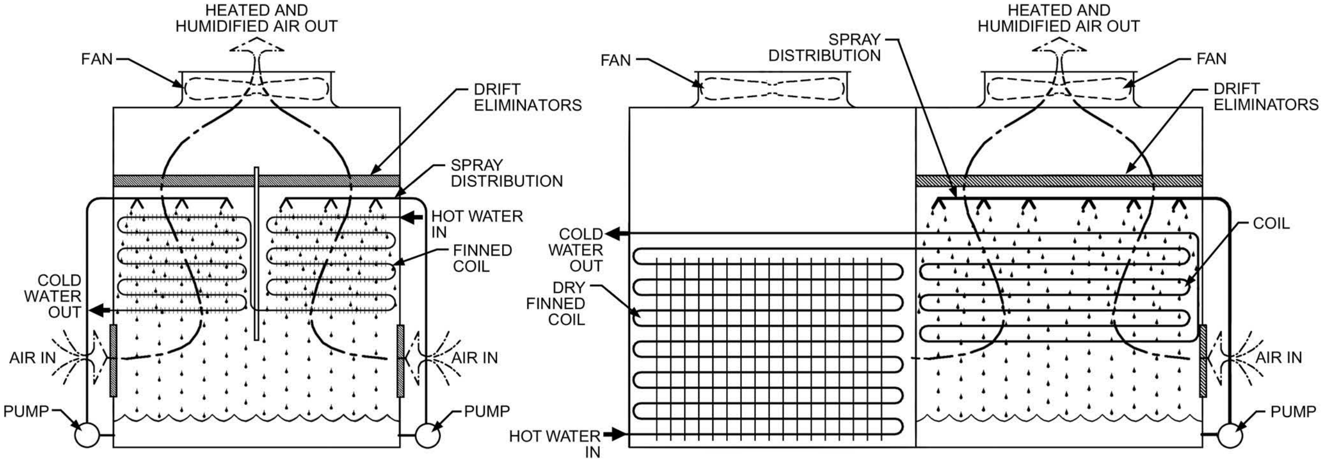


Fig. 18-43 Counterflow Forced-Draft Cooling Tower

(Figure 16, Chapter 40, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

18.2 Given a compressor using R-22 condensing at 80°F (26.7°C) and evaporating at 20°F (–6.7°C), find the enthalpy of the refrigerant when it enters the

(a) compressor

(b) condenser

(c) evaporator

Find the power required for the compressor.

[Ans: (a) 106.4, (b) 116.5, (c) 33.1 Btu/lb, and 0.65 HP/ton]

18.3 What is the maximum theoretical COP of a refrigeration device operating between 0°F and 75°F (–17.8°C and 23.9°C). Why is this theoretical limit difficult to obtain? [Ans: 6.14]

18.4 A reference book on refrigeration indicates that a compressor using R-22 requires a displacement of 40.59 cfm per ton for evaporation at –100°F and condensing at –30°F. Is this correct? Substantiate your answer with calculations based on knowledge of R-22 for these conditions. Also, verify the mass flow rate in lb per min.

18.5 An R-134a refrigerating system develops 10 tons of refrigeration when operating at 100°F condensing and +10°F evaporating, with no liquid subcooling or vapor superheating. Determine the volume of the refrigerant leaving the expansion valve in cubic feet per minute.

18.6 An expansion device has a mass flow rate for R-134a given by

$$m = 60 + 0.25\Delta p$$

where m = flow rate in lb/min and Δp = pressure drop across the valve in psi.

For an evaporator temperature of 0°F and a condenser temperature of 100°F, estimate the piston displacement required for a compressor if C = 0.04 and the polytropic compression coefficient n = 1.1 for the compression process. [Ans: 236 cfm]

18.7 A liquid-to-suction heat exchanger is installed in an R-134a system to cool liquid that comes from the condenser with vapor that flows from the evaporator. The evaporator generates 10 tons (35.17 kW) of refrigeration at 30°F (–1.1°C). Liquid leaves the condenser saturated at 100°F (37.8°C), vapor leaves the evaporator saturated, and vapor leaves the heat exchanger at a temperature of 50°F (10°C). What is the flow rate of the refrigerant?

18.8 An eight-cylinder ammonia compressor is designed to operate at 800 rpm and deliver 30 tons of refrigeration. The evaporator is to operate at 10°F with a condensing temperature of 100°F. The vapor enters the compressor at 30°F. The ammonia leaves the condenser as saturated liquid. If the average piston speed is to be 600 ft/min and the actual volumetric efficiency at this condition is 83%, find the bore of the compressor.

18.9 A condenser is to be selected for a system that generates 30 tons (105.5 kW) of refrigeration at 10°F (–12.2°C). The condenser is to operate at 110°F (43.3°C) and is cooled with 90 gpm (5.68 L/s) of water at 85°F (29.4°C). If the expected *U-factor* of the condenser is 130 Btu/h· ft^2·°F [738 W/(m^2·K)], calculate the condensing area required.

18.10 A cooling tower cools water by passing it through a stream of air. If 1000 cfm of air at 95°F dry bulb and 78°F wet bulb enters the tower and leaves saturated at 84°F, to what temperature can this air cool water that enters at 110°F with a flow of 80 lb/min? What is the makeup water rate?

18.5 Bibliography

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SI Tables

Table 18-3 SI Types of Coolers

Type of Cooler	Subtype	Usual Refrigerant Feed Device	Usual Capacity Range, kW	Commonly Used Refrigerants
Direct-expansion	Shell-and-tube	Thermal expansion valve Electronic modulation valve	7 to 1800 7 to 1800	12, 22, 134a, 404A, 407C, 410A, 500, 502, 507A, 717
	Tube-in-tube	Thermal expansion valve	18 to 90	12, 22, 134a, 717
	Brazed-plate	Thermal expansion valve	2 to 700	12, 22, 134a, 404A 407C, 410A, 500, 502, 507A, 508B, 717, 744
	Semiwelded plate	Thermal expansion valve	175 to 7000	12, 22, 134a, 500, 502, 507A, 717, 744
Flooded	Shell-and-tube	Low-pressure float	90 to 7000	11, 12, 22, 113, 114
		High-pressure float	90 to 21 100	123, 134a, 500, 502, 507A, 717
		Fixed orifice(s) Weir	90 to 21 100 90 to 21 100	
	Spray shell-and-tube	Low-pressure float	180 to 35 000	11, 12, 13B1, 22
		High-pressure float	180 to 35 000	113, 114, 123, 134a
	Brazed-plate	Low-pressure float	2 to 700	12, 22, 134a, 500, 502, 507A, 717, 744
	Semiwelded plate	Low-pressure float	175 to 7000	12, 22, 134a, 500, 502, 507A, 717, 744
Baudelot	Flooded	Low-pressure float	35 to 350	22, 717
	Direct-expansion	Thermal expansion valve	18 to 90	12, 22, 134a, 717
Shell-and-coil	—	Thermal expansion valve	7 to 35	12, 22, 134a, 717

Table 18-6 SI Physical Properties of Selected Refrigerants[a]

(Table 3, Chapter 19, *2005 ASHRAE Handbook—Fundamentals* SI)

Refrigerant		Chemical Formula	Molecular Mass	Boiling Pt. (NBP) at 101.325 kPa, °C	Freezing Point, °C	Critical Temperature, °C	Critical Pressure, kPa	Critical Density, kg/m^3	Refractive Index of Liquid[b,c]
No.	Chemical Name or Composition (% by Mass)								
728	Nitrogen	N_2	28.013	–195.8	–210.0	–146.96	3395.8	313.3	1.205 (83 K) 589.3 nm
729	Air	—	28.959	–194.25	—	–140.59	3789.6	335.94	—
740	Argon	Ar	39.948	–185.85	–189.34	–122.46	4863.0	535.6	1.233 (84 K) 589.3 nm
732	Oxygen	O_2	31.999	–182.96	–218.79	–118.57	5043.0	436.14	1.221 (92 K) 589.3 nm
50	Methane	CH_4	16.043	–161.48	–182.46	–82.586	4599.2	162.66	—
14	Tetrafluoromethane	CF_4	88.005	–128.05	–183.61	–45.64	3750.0	625.66	—
170	Ethane	C_2H_6	30.07	–88.598	–182.8	32.18	4871.8	206.58	—
503	R-23/13 (40.1/59.9)	—	87.247	–87.76	—	18.417	4280.5	565.68	—
508A[4]	R-23/116 (39/61)	—	100.1	–87.377	—	10.844	3668.2	570.62	—
508B[4]	R-23/116 (46/54)	—	95.394	–87.344	—	11.827	3789	572.13	—
23	Trifluoromethane	CHF_3	70.014	–82.018	–155.13	26.143	4832	526.5	—
13	Chlorotrifluoromethane	$CClF_3$	104.46	–81.48	–181.15	28.85	3879	582.88	1.146 (25)[2]
744	Carbon dioxide	CO_2	44.01	–78.4[d]	–56.558[e]	30.978	7377.3	467.6	1.195 (15)
504	R-32/115 (48.2/51.8)	—	79.249	–57.695	—	61.084	433.7	504.62	—
32	Difluoromethane	CH_2F_2	52.024	–51.651	–136.81	78.105	5782.0	424	—
410A	R-32/125 (50/50)	—	72.585	–51.443	—	71.358	4902.6	459.53	—
125	Pentafluoroethane	C_2HF_5	120.02	–48.09	–100.63	66.023	3617.7	573.58	—
1270	Propylene	C_3H_6	42.08	–47.69	–185.2	92.42	4664.6	223.39	1.3640 (–50)[1]
143a	Trifluoroethane	CH_3CF_3	84.041	–47.241	–111.81	72.707	3761.0	431.0	—
507A	R-125/143a (50/50)	—	98.859	–46.741	—	70.617	3705	490.77	—
404A	R-125/143a/134a (44/52/4)	—	97.604	–46.222	—	72.046	3728.9	486.53	—
502	R-22/115 (48.8/51.2)	—	111.63	–45.174	—	80.153	3917.6	566.03	—
407C	R-32/125/134a (23/25/52)	—	86.204	–43.627	—	86.034	4629.8	484.23	—
290	Propane	C_3H_8	44.096	–42.09	–187.67	96.675	4247.1	218.5	1.3397 (–42)
22	Chlorodifluoromethane	$CHClF_2$	86.468	–40.81	–157.42	96.145	4990.0	523.84	1.234 (25)[2]
115	Chloropentafluoroethane	$CClF_2CF_3$	154.47	–38.94	–99.39	79.95	3120.0	613.1	1.221 (25)[2]
500	R-12/152a (73.8/26.2)	—	99.303	–33.603	—	102.09	4168.6	495.1	—
717	Ammonia	NH_3	17.03	–33.327	–77.655	132.25	11333.0	225.0[d]	1.325 (16.5)
12	Dichlorodifluoromethane	CCl_2F_2	120.91	–29.752	–157.05	111.97	4136.1	565.0	1.288 (25)[2]
134a	Tetrafluoroethane	CF_3CH_2F	102.03	–26.074	–103.3	101.06	4059.3	511.9	—
152a	Difluoroethane	CHF_2CH_3	66.051	–24.023	–118.59	113.26	4516.8	368	—
124	Chlorotetrafluoroethane	$CHClFCF_3$	136.48	–11.963	–199.15	122.28	3624.3	560.0	—
600a	Isobutane	C_4H_{10}	58.122	–11.67	–159.59	134.67	3640.0	224.35	1.3514 –25)[1]
142b	Chlorodifluoroethane	$CClF_2CH_3$	100.5	–9.15	–130.43	137.11	4070.0	446.0	—
C318	Octafluorocyclobutane	C_4F_8	200.03	–5.975	–39.8	115.23	2777.5	619.97	
600	Butane	C_4H_{10}	58.122	–0.55	–138.28	151.98	3796.0	227.84	1.3562 (–15)[1]
114	Dichlorotetrafluoroethane	$CClF_2CClF_2$	170.92	3.586	–94.15	145.68	3257.0	579.97	1.294 (25)
11	Trichlorofluoromethane	CCl_3F	137.37	23.708	–110.47	197.96	4407.6	554.0	1.362 (25)[2]
123	Dichlorotrifluoroethane	$CHCl_2CF_3$	152.93	27.823	–107.15	183.68	3661.8	550.0	—
141b	Dichlorotrifluoroethane	CCl_2FCH_3	116.95	32.05	–103.3	206.81	4460.0	460.0	—
113	Trichlorotrifluoroethane	CCl_2FCClF_2	187.38	47.585	–36.22	214.06	3392.2	560.0	1.357 (25)[2]
718[3]	Water	H_2O	18.015	99.974	0.01	373.95	22 064.0	322.0	—

Note:

[a] Data from *ASHRAE Thermodynamic Properties of Refrigerants* (Stewart et al. 1986) or from Lemmon et al. (2002), unless otherwise noted.

[b] Temperature of measurement (°C, unless kelvin is noted) shown in parentheses. Data from *CRC Handbook of Chemistry and Physics* (CRC 1987), unless otherwise noted.

[c] For the sodium D line.

[d] Sublimes.

[e] At 527 kPa.

References:

[1] Kirk and Othmer (1956).

[2] *Bulletin* B-32A (DuPont).

[3] *Handbook of Chemistry* (1967).

[4] *NIST Standard Reference Database 23*, v.7.

Table 18-7 SI Comparative Refrigerant Performance per Kilowatt of Refrigeration

(Table 7, Chapter 19, *2005 ASHRAE Handbook—Fundamentals* SI)

Refrigerant												
No.	**Chemical Name or Composition (% by mass)**	**Evaporator Pressure, MPa**	**Condenser Pressure, MPa**	**Compression Ratio**	**Net Refrigerating Effect, kJ/kg**	**Refrigerant Circulated, g/s**	**Liquid Circulated, L/s**	**Specific Volume of Suction Gas, m^3/kg**	**Compressor Displacement, L/s**	**Power Consumption, kW**	**Coefficient of Performance**	**Compressor Discharge Temp., K**
170	Ethane	1.608	4.639	2.88	161.71	6.10	0.0219	0.0338	0.206	0.365	2.70	323
744	Carbon dioxide	2.254	7.18	3.19	133.23	3.88	0.0064	0.0168	0.065	0.192	2.69	343
1270	Propylene	0.358	1.304	3.64	286.17	3.46	0.0070	0.1299	0.449	0.220	4.50	315
290	Propane	0.286	1.075	3.76	277.90	3.53	0.0073	0.1562	0.551	0.218	4.50	309
502	R-22/115 (48.8/51.2)	0.343	1.312	3.83	105.95	9.43	0.0079	0.0508	0.479	0.228	4.38	311
507A	R-125/143a (50/50)	0.379	1.459	3.85	110.14	9.07	0.0089	0.0508	0.461	0.239	4.18	308
404A	R-125/143a/134a (44/52/4)	0.365	1.42	3.89	114.15	8.75	0.0086	0.0537	0.470	0.237	4.21	309
410A	R-32/125 (50/50)	0.478	1.872	3.92	167.89	5.84	0.0056	0.0545	0.318	0.222	4.41	324
125	Pentafluoroethane	0.403	1.561	3.87	85.30	11.41	0.0098	0.0394	0.449	0.244	3.99	304
22	Chlorodifluoro-methane	0.295	1.187	4.02	162.67	6.13	0.0052	0.0779	0.478	0.214	4.66	326
12	Dichlorodifluoro-methane	0.181	0.741	4.09	117.02	8.49	0.0066	0.0923	0.784	0.212	4.70	311
500	R-12/152a (73.8/26.2)	0.214	0.876	4.09	139.68	7.08	0.0063	0.0939	0.665	0.212	4.66	314
407C	R-32/125/134a (23/25/52)	0.288	1.26	4.38	163.27	6.11	0.0054	0.0805	0.492	0.222	4.50	321
600a	Isobutane*	0.088	0.403	4.58	263.91	3.76	0.0069	0.4073	1.533	0.215	4.62	303
134a	Tetrafluoroethane	0.163	0.767	4.71	148.03	6.71	0.0056	0.1214	0.814	0.216	4.60	310
124	Chlorotetrafluoro-ethane*	0.088	0.443	5.03	117.83	8.41	0.0063	0.1711	1.439	0.214	4.62	303
717	Ammonia	0.235	1.162	4.94	1103.14	0.90	0.0015	0.5117	0.463	0.210	4.76	372
600	Butane*	0.056	0.283	5.05	292.24	3.53	0.0062	0.6446	2.274	0.218	4.74	303
11	Trichlorofluoro-methane	0.02	0.125	6.25	155.95	6.36	0.0043	0.7689	4.891	0.197	5.02	316
123	Dichlorotrifluoro-ethane	0.016	0.109	6.81	142.28	7.02	0.0048	0.8914	6.259	0.204	4.90	306
113	Trichlorotrifluoro-ethane*	0.007	0.054	7.71	122.58	7.84	0.0051	1.6818	13.187	0.200	4.81	303

*Superheat required.

Chapter 19

HEATING EQUIPMENT

In this chapter the principles of combustion and data concerning various types of fuels common to HVAC systems are discussed. The information in this chapter has been extracted from Chapter 28 in the 2017 *ASHRAE Handbook—Fundamentals* and Chapters 28 and 31 through 37 in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

19.1 Fuels and Combustion

19.1.1 Principles of Combustion

Combustion is the chemical process in which an oxidant reacts rapidly with a fuel to liberate stored energy as thermal energy, generally in the form of high-temperature gases. The oxidant is usually oxygen in the air.

Conventional hydrocarbon fuels contain primarily hydrogen and carbon, in either elemental form or in various compounds. Their complete combustion produces mainly carbon dioxide and water; however, small quantities of carbon monoxide and partially reacted flue gas constituents may form. Most conventional fuels also contain small amounts of sulfur, oxidized to SO_2 or SO_3 during combustion, and noncombustible substances (mineral matter or ash, water, and inert gases). Flue gas is the product of complete or incomplete combustion, including excess air (if present) but not dilution air.

The rate of fuel combustion depends on (1) the chemical reaction rate of combustible fuel constituents with oxygen, (2) the rate at which oxygen is supplied to fuel (mixing of air and fuel), and (3) the temperature in the combustion region. The reaction rate is fixed by fuel selection. Increasing the mixing rate or temperature increases the combustion rate. In complete combustion of hydrocarbon fuels, all hydrogen and carbon in the fuel are oxidized to H_2O and CO_2.

For complete combustion, excess oxygen or excess air must generally be supplied beyond the amount theoretically required to oxidize the fuel; this is usually expressed as a percentage of the air required to completely oxidize the fuel. Incomplete combustion occurs when a fuel element is not completely oxidized in the combustion process. For example, a hydrocarbon may not completely oxidize to carbon dioxide and water, but it may form partially oxidized compounds such as carbon monoxide, aldehydes, and ketones. Conditions that promote incomplete combustion include

- Insufficient air and fuel mixing, causing local fuel-rich and fuel-lean zones.
- Insufficient air supply to the flame, providing less than the required quantity of oxygen.
- Insufficient reactant residence time in the flame, preventing completion of combustion reactions.
- Flame impingement on a cold surface, quenching combustion reactions.
- Too-low flame temperature, slowing combustion reactions.

Incomplete combustion uses fuel inefficiently, contributes to air pollution, and can be hazardous because of carbon monoxide production.

Combustion of oxygen with the combustible elements and compounds in fuels occurs according to fixed chemical principles, including:

- Chemical reaction equations.
- Law of matter conservation. The mass of each element in the reaction products must equal the mass of that element in the reactants.
- Law of combining mass. Chemical compounds are formed by elements combining in fixed mass relationships.
- Chemical reaction rates.

Oxygen for combustion is normally obtained from air, which is a physical mixture of nitrogen, oxygen, small amounts of water vapor, carbon dioxide, and inert gases. For practical combustion calculations, dry air consists of 20.95% oxygen and 79.05% inert gases, primarily nitrogen, by volume, or 23.15% oxygen and 76.85% inert gases by mass. For calculation purposes, nitrogen is assumed to pass through the combustion process unchanged (although small quantities of nitrogen oxides are known to form).

The quantity of heat generated by complete combustion of a unit of specific fuel is constant and is called the **heating value**, **heat of combustion**, or **caloric value** of that fuel. The heating value of a fuel can be determined by measuring the heat evolved during combustion of a known quantity of the fuel in a calorimeter, or it can be estimated from chemical analysis of the fuel and the heating value of the several chemical elements in the fuel.

Higher heating value, gross heating value, or total heating value is determined when water vapor in fuel combustion products is condensed and the latent heat of vaporization is included in the fuel's heating value. Conversely, lower heating value or net heating value is obtained when latent heat of vaporization is not included. When the heating value of a fuel is specified without designating higher or lower, it generally means the higher heating value in the United States. (lower heating value is mainly used for internal combustion engine fuels.)

Table 19-1 Heating Values of Selected Fuels

Substance	Molecular Symbol	Higher Heating Values,[a] Btu/lb	Lower Heating Values,[a] Btu/lb	Specific Volume,[b] ft^3/lb	Higher Heating Values,[c] MJ/kg	Lower Heating Values,[c] MJ/kg	Density,[d] kg/m^3
Carbon (to CO)	C	3950	3950		9.188	9.188	
Carbon (to CO_2)	C	14,093	14,093		32.788	32.780	
Carbon Monoxide	CO	4347	4347	13.5	10.111	10.111	1.187
Hydrogen	H_2	61,095	51,623	188.0	142.107	118.680	0.085
Methane	CH_4	23,875	21,495	23.6	55.533	49.997	0.679
Ethane	C_2H_6	22,323	20,418	12.5	51.922	47.492	1.28
Propane	C_3H_8	21,669	19,937	8.36	50.402	46.373	1.92
Butane	C_4H_{10}	21,321	19,678	6.32	49.593	45.771	2.53
Ethylene	C_2H_4	21,636	20,275		50.325	47.160	
Propylene	C_3H_6	21,048	19,687	9.01	48.958	45.792	1.78
Acetylene	C_2H_2	21,502	20,769	14.3	50.028	48.309	1.120
Sulfur (to SO_2)	S	3980	3980		9.257	9.257	
Sulfur (to SO_3)	S	5940	5940		13.816	13.816	1.456
Hydrogen Sulfide	H_2S	7097	6537	11.0	16.508	15.205	

a All values corrected to 60 °F, 30 in. Hg dry. For gases saturated with water vapor at 60°F, deduct 1.74% of the value to adjust for gas volume displaced by water vapor.
b At 32 °F and 29.92 in. Hg

c All values corrected to 16 °C, 101.4 kPa dry. For gases saturated with water vapor at 16°F, deduct 1.74% of the value to adjust for gas volume displaced by water vapor.
d At 32 °F and 29.92 in. Hg

Heating values are usually expressed in Btu/ft^3 (kJ/m^3) for gaseous fuels, Btu/gal (kJ/m^3) for liquid fuels, and Btu/lb (kJ/kg) for solid fuels. Heating values are always given in relation to a certain reference temperature and pressure, usually 60, 68, or 77°F and 14.696 psia (15, 20, or 25°C and 103.325 kPa), depending on the particular industry practice. Heating values of several substances and common fuels are listed in Table 19-1.

When combustion is incomplete, not all fuel is completely oxidized, and the heat released is less than the heating value of the fuel. Therefore, the quantity of heat produced per unit of fuel consumed decreases, implying lower combustion efficiency.

Not all heat released during combustion can be used effectively. The greatest heat loss is in the form of increased temperature (thermal energy) of hot exhaust gases above the temperature of incoming air and fuel. Other heat losses include radiative and convective heat transfer from outer walls of combustion equipment to the environment.

19.1.2 Fuels

Generally, hydrocarbon fuels are classified according to physical state (gaseous, liquid, or solid). Different types of combustion equipment are usually needed to burn fuels in different physical states. Gaseous fuels can be burned in premix or diffusion burners that take advantage of the gaseous state. Liquid fuel burners must include a means of atomizing or vaporizing fuel into small droplets or to a vapor for burning and must provide adequate mixing of fuel and air. Solid fuel combustion equipment must (1) heat fuel to vaporize sufficient volatiles to initiate and sustain combustion, (2) provide residence time to complete combustion, and (3) provide space for ash containment.

Principal uses of fuel include space heating and cooling of residential, commercial, industrial, and institutional buildings; service water heating; steam generation; and refrigeration. Fuels for these applications are natural and liquefied petroleum gases, fuel oils, diesel and gas turbine fuels (for total energy applications), and coal.

Gaseous Fuels. Heating and cooling applications are presently limited to natural and liquefied petroleum gases. Natural gas is a nearly odorless and colorless gas that accumulates in the upper parts of oil and gas wells. Raw natural gas is a mixture of methane (55 to 98%), higher hydrocarbons (primarily ethane), and noncombustible gases. Some constituents, principally water vapor, hydrogen sulfide, helium, liquefied petroleum gases, and gasoline, are removed prior to distribution.

Heating values of natural gases vary from 900 to 1200 Btu/ft^3 (34 to 45 MJ/m^3); the usual range is 1000 to 1050 Btu/ft^3 (37 to 39 MJ/m^3) at sea level. Three liquefied petroleum gases—butane, propane, and a mixture of the two—are commercially available.

Commercial propane consists primarily of propane but generally contains about 5 to 10% propylene. It has a heating value of about 21,560 Btu/lb (50.15 MJ/kg) or about 2500 Btu/ft^3 (93 MJ/m^3) of gas. At atmospheric pressure, commercial propane has a boiling point of about –40°F (–40°C). The low boiling point of propane makes it usable during winter in cold climates such as the northern United States. It is available in cylinders, bottles, tank trucks, or tank cars.

Commercial butane consists primarily of butane but may contain up to 5% butylene. It has a heating value of about 21,180 Btu/lb (49.26 MJ/kg) or 3200 Btu/ft^3 (119 MJ/m^3).

Commercial propane-butane mixtures with varying ratios of propane and butane are available. Their properties generally fall between those of the unmixed fuels. Propane-air and butane-air mixtures are used in place of natural gas in small communities and by natural gas companies at peak loads. Typical heating values are listed in Table 19-2.

Liquid Fuels. Liquid fuels, with few exceptions, are mixtures of hydrocarbons refined from crude petroleum. Fuel oils for heating are broadly classified as distillate fuel oils

Table 19-2 Heating Values of Gaseous Fuels

Gas	Btu/ft^3	MJ/m^3	Specific Gravity Air = 1.0
Natural	1030	38.4	0.60
Propane	2500	93.1	1.53
Butane	3175	118.3	2.00

Table 19-3 Typical API Gravity, Density, and Heating Value of Standard Grades of Fuel Oil

Grade No.	API Gravity	Density, lb/gal[a]	Heating Value, Btu/gal[b]
1	38 to 45	6.950 to 6.675	137,000 to 132,900
2	30 to 38	7.296 to 6.960	141,800 to 137,000
4	20 to 28	7.787 to 7.396	148,100 to 143,100
5L	17 to 22	7.940 to 7.686	150,000 to 146,800
5H	14 to 18	8.080 to 7.890	152,000 to 149,400
6	8 to 15	8.448 to 8.053	155,900 to 151,300

[a] 1 lb/gal = 120 kg/m^3
[b] 1 Btu/gal = 279 kJ/m^3

(lighter oils) or residual fuel oils (heavier oils). ASTM has established specifications for fuel oil properties that subdivide the oils into various grades. Grades No. 1 and 2 are distillate fuel oils. Grades 4,5 (Light), 5 (Heavy), and 6 are residual fuel oils. Specifications for the grades are based on required characteristics of fuel oils for use in different types of burners. Typical gravity and heating values of standard grades of fuel oil are shown in Table 19-3.

Solid Fuels. Solid fuels include coal, coke, wood, and waste products of industrial and agricultural operations. Of these, only coal is widely used. Chemically, coal consists of carbon, hydrogen, oxygen, nitrogen, sulfur, and a mineral residue, ash. Heating values may be reported on an as-received, dry, dry and mineral-matter-free, or moist and mineral-matter-free basis. Higher heating values of coals are frequently reported with their proximate analysis. When more specific data are lacking, the higher heating value of higher quality coals can be calculated by the Dulong Formula:

Higher Heating Value (Btu/lb)

$$= 14{,}544C + 62{,}028\,[H - (O/8)] + 4050S \tag{19-1a}$$

or Higher Heating Value (MJ/kg)

$$= 33.829C + 144.28\,[H - (O/8)] + 9.42S \tag{19-1b}$$

where C, H, O, and S are the mass fractions of carbon, hydrogen, oxygen, and sulfur in the coal. Typical values for coal are listed in Table 19-4.

19.1.3 Combustion Calculations

Calculations of the quantity of air required for combustion and quantity of flue gas products generated during combustion are frequently needed for sizing system components and as input to efficiency calculations. Other calculations, such as values for excess air and theoretical CO_2, are useful in estimating combustion system performance. Analysis of the flue gas products is often done using an Orsat analysis. This is a measure in percent by volume of the CO_2, CO, and O_2 in the dry products. The quantity of water vapor in the products is taken to be zero for this calculation. It is then assumed that the % N_2 in the products is 100% minus the % CO_2, % CO, and % O_2. Details on combustion calculations are given in Chapter 28 of the 2017 *ASHRAE Handbook—Fundamentals*.

Table 19-4 Heating Value of Coal

Rank	Heating Value As Received Btu/lb	MJ/kg
Anthracite	12,700	29.5
Semianthracite	13,600	31.6
Low-volatile bituminous	14,350	33.4
Medium-volatile bituminous	14,000	32.6
High-volatile bituminous A	13,800	32.1
High-volatile bituminous B	12,500	29.1
High-volatile bituminous C	11,000	25.6
Subbituminous B	9000	20.9
Subbituminous C	8500	19.8
Lignite	6900	16.0

Example 19-1 Carbon burns with 160% theoretical air. Combustion goes to completion. Determine: (a) the air/fuel ratio by mass and (b) the Orsat gas analysis and dew point of the products.

***Solution*:**

***Theoretical*:**

$$C + O_2 + 3.76N_2 \rightarrow CO_2 + 3.76N_2$$

***Actual*:**

$$C + 1.6O_2 + 1.6\,(3.76)\,N_2 \rightarrow CO_2 + 6.02N_2 + 0.6O_2$$

(a) A/F = [1.6(32) + 1.6(3.76)28]/12 = 18.3 lb_{air}/lb_{fuel}

$$\begin{array}{llll}
\text{(b)} & CO_2 & 1.0\text{ moles}/7.62 & = 13.1\% \\
 & O_2 & 0.6\text{ moles}/7.62 & = 7.9\%\text{ ORSAT} \\
 & N_2 & \underline{6.02\text{ moles}/7.6} & = \underline{79.0\%} \\
 & & 7.62\text{ moles} & = 100.0\%
\end{array}$$

No dew point because there is no water vapor in the products.

Note: Assuming air is 79% N_2, 21% O_2 by volume:

$$1\text{ mol }O_2 + 3.76\text{ mol }N_2 = 4.76\text{ mol air}$$

Example 19-2 The flue gas analysis of a hydrocarbon fuel on a percent by volume dry basis shows $CO_2 = 12.4\%$, $O_2 = 3.2\%$, CO = 0.1%, $H_2 = 0.2\%$, and $N_2 = 84.1\%$. Determine the air/fuel ratio by volume.

***Solution*:**

$$C_xH_y + (84.1/3.76)\,O_2 + 84.1\,N_2 \rightarrow 12.4\,CO_2 + 3.2\,O_2 + 0.1\,CO + 0.2\,H_2 + 84.1\,N_2 + z\,H_2O$$

$x = 12.4 + 0.1 = 12.5$ Carbon balance
$22.4 = 12.4 + 3.2 + 0.05 + z/2$ O_2 balance
$z = 13.5$
$y = 0.4 + 27 = 27.4$ Hydrogen balance

$$C_{12.5}H_{27.4} + 22.4O_2 + 84.1N_2 \rightarrow 12.4CO_2 + 3.2O_2 + 0.1CO + 0.2H_2 + 84.1N_2 + 13.5H_2O$$

$$A/F = (22.4 \text{ moles } O_2 + 84.1 \text{ moles } N_2)/\text{mole fuel}$$

$$A/F = 106.5/1 \text{ by volume}$$

19.2 Burners

When heating is required within a controlled environment, it is normally done using hot water, warm air, steam, radiant heat, or electric heat. Water, air, and steam can be heated using gas, oil, coal, or electric energy. In some instances, energy for heating can be provided by heat recovery techniques or solar energy.

19.2.1 Domestic Gas Burning Equipment

A gas burner is used to convey gas or a mixture of gas and air to the combustion zone. Burners are of the atmospheric injection, luminous flame, or power burner types.

Domestic gas burners may be classified as either those types designed for central heating plants or those designed for unit application. Gas-designed units and conversion burners are available for the several kinds of central systems and for other applications where the units are installed in the heated space. Central heating appliances include warm air furnaces and steam or hot water boilers.

Warm air furnaces are of two types, gravity and forced air. Forced warm-air furnaces use a motor-driven blower to circulate air over the heat exchanger and through the ducts. A draft hood is attached to the outlet of the furnace and replaces the barometric damper.

Gas-designed boilers for hot water or steam heating are available in cast-iron, steel, and nonferrous metals. Burners are located beneath the sections; the flue gases pass upward between the sections to the flue collector. Some boilers are designed to provide domestic hot water, using tankless or instantaneous heaters.

Some gas furnaces and boilers are available with sealed combustion chambers. These units have no draft hood and are called **direct vent appliances**. Combustion air is piped, usually through a side wall from outdoors, directly to the combustion chamber.

The air intake pipe terminates in the same location as the flue gas vent, sometimes as concentric pipes, with an appropriate terminal covering or cowl. No chimney or vertical flue is needed with such units.

Floor furnaces are used in mild climates for auxiliary heating, heating of single rooms, and, in some cases, heating of several rooms. Floor furnaces are not usually considered central heating systems. However, some are furnished with a circulating fan and as many as eight takeoff ducts. This type of floor furnace is a central heating system. In such a system, piping is below the floor as in other central warm air systems.

Space heaters are generally used for heating a single room or limited area. They differ from central heating equipment in the extent of distribution systems used. Both natural convection and forced-circulation systems distribute the warm air; however, no ductwork distributes heat beyond the immediate area occupied by the heater. Domestic, gas-fired space heaters include floor furnaces (without ductwork), vented wall furnaces, vented and unvented room heaters, radiant heaters, baseboard-sealed combustion system wall furnaces, wall heaters, and unit heaters. Some small room heaters are operated without vents, but they must be used with caution. Both manual and automatic controls are used. An automatic pilot or ignition system must be a part of any control system.

Under controlled conditions, an oil burner combines fuel oil and air for combustion. Fuel oil may be either **atomized** or **vaporized** for the combustion process. Air for combustion is supplied by natural or mechanical draft. Ignition is generally accomplished by an electric spark, gas pilot flame, oil pilot flame, or combination of these. Burners of different types operate with luminous or nonluminous flame. Operation may be continuous, modulating, or intermittent with high-low flame.

Residential oil burners are ordinarily used in the range of 0.5 to 3.5 gph (0.5 to 3.7 mL/s) fuel consumption rate. However, burners up to 7.0 gph (7.2 mL/s) sometimes fall in the residential classification because of similarities in controls and standards. Burner capacity of 7.0 gph (7.35 mL/s) and above is classified as commercial-industrial. Generally, No. 2 fuel oil is used, although burners in the residential size range can also operate on No. 1 fuel oil. In addition to boilers and furnaces for space heating, burners in the 0.5 to 1.0 gph (0.5 to 1.0 mL/s) size range are also used for separate tank type residential hot water heaters, infrared heaters, space heaters, and other commercial equipment.

Most burners manufactured (over 95%) are high-pressure, atomizing gun burners. While other types of burners are still in operation, only a few of these types are currently in production.

The high-pressure, atomizing gun burner illustrated in Figure 19-1 supplies oil to the atomizing nozzle at 100 to 300 psi (700 to 2100 kPa). A blower supplies air for combustion, and a damper or other device regulates the air supply at the burner. Ignition is usually accomplished by a high voltage electric spark, which may be intermittent, sometimes called either **constant ignition** ("on" when the burner motor is on) or **interrupted** (on only to start combustion). Typically, these burners fire into a combustion chamber in which negative draft is maintained.

Present high-pressure atomizing gun burner design includes retention heads and residential burner motors operating at 3450 rpm instead of 1725 rpm. The retention head assists combustion by providing better air-oil mixing, turbulence, and shear. Using the higher-speed 3450 rpm motors (often combined with a retention head design) results in a more compact burner having equal capacity to one operating at 1725 rpm, and one that has a wide tolerance for varying combustion chamber and draft conditions.

Vaporizing burners are designed for use with No. 1 fuel oil. Fuel is ignited by manual pilot or electric spark. The combustion process usually provides enough heat for oil vaporization.

Oil-designated boiler-burner units for hydronic (hot water or steam) heating are constructed of cast iron, steel, and nonferrous metals. The oil burner is usually located at, or near,

the base of the boiler; the flue gases pass upward and around the heat transfer sections and then out the chimney connector. Most oil-powered boiler burner units provide domestic hot water by incorporating a water heating coil (tankless heater) within the boiler.

Warm air furnaces with oil burner units are normally of forced-air design. A blower circulates air over the heat exchanger and through the duct. Upflow, counterflow, horizontal, or downflow furnaces are available. Most forced warm air furnaces make provisions for installing direct expansion air conditioning directly in the unit.

Additional details concerning commercial and industrial gas and/or oil burning equipment, as well as solid fuel burning equipment, can be found in Chapter 31 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*. Details about various types of steam and hot water boilers are in Chapter 32 of the above reference.

19.3 Residential Furnaces

19.3.1 Equipment Variations

Residential furnaces provide heated air through a system of ductwork into the space being heated. They are of two basic types:

Fuel-Burning Furnaces. Combustion takes place within a combustion chamber. Circulating air passes over the outside surfaces of the heat exchanger so that it does not contact the fuel or the products of combustion. The products of combustion are passed to the outside atmosphere through a vent.

Electric Furnaces. A resistance-type heating element heats the circulating air either directly or through a metal sheath enclosing the resistance element.

Residential furnaces may be further categorized by:

- Fuel type
- Mounting arrangement
- Airflow direction
- Combustion system
- Installation location

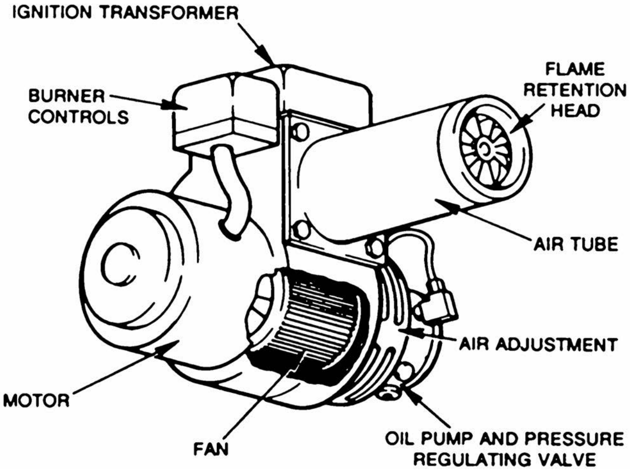


Fig. 19-1 High-Pressure Atomizing Oil Burner

A forced air furnace is a self-enclosed appliance used to heat air that is circulated through the enclosure and discharged directly into the space being heated or air that is conveyed through ducts to the space to be heated. An induced draft gas-fired unit is illustrated in Figure 19-2.

In fuel-burning furnaces, combustion occurs within a metal-walled heat exchanger. Air passes over the outside surfaces of the heat exchanger and the heat is transferred through the heat exchanger walls; the circulating air does not come in contact with the fuel or the products of combustion. The products of combustion are conveyed to the outside atmosphere through a flue or vent.

In an electric furnace, the heating element, heated by its electric resistance, heats the circulating air either directly or through a metal sheath enclosing the resistance element.

Residential forced air furnaces usually have a capacity under 250,000 Btu/h (74 kW) output. Residential furnaces are often used in commercial installations. The reverse is usually not true since more than one residential furnace would usually be used in an extremely large home rather than a furnace intended for commercial use.

The furnace may be designed and manufactured to combine components in a variety of ways. The relative positions of the components in the different types of furnaces are described as follows:

1. In a **horizontal furnace**, the blower is located beside the heat exchanger (Figure 19-3). The air enters one end and travels horizontally through the blower, over the heat

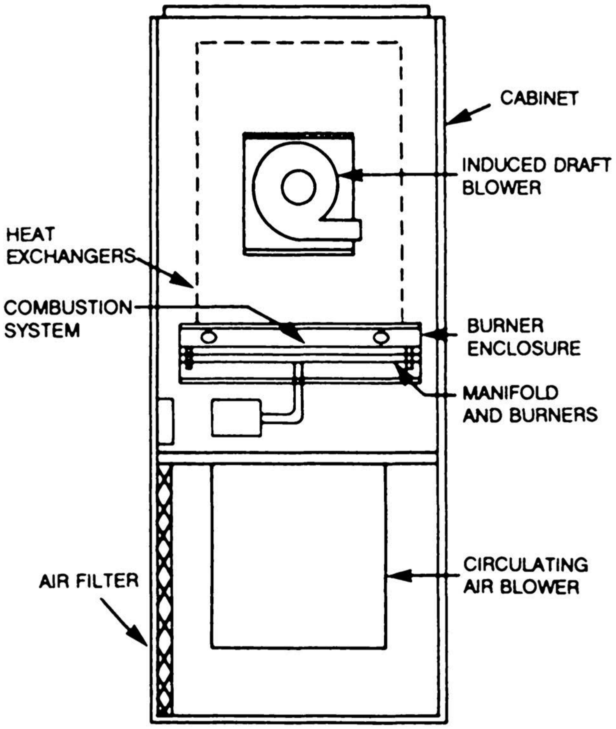


Fig. 19-2 Induced Draft Gas Furnace
(Figure 1, Chapter 33, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

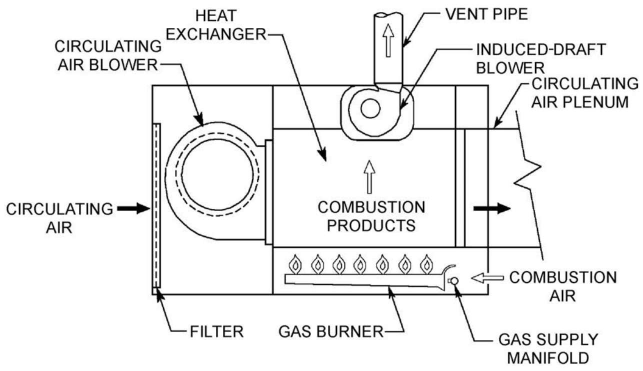


Fig. 19-3 Horizontal Forced-Warm-Air Furnace
(Figure 4, Chapter 33, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

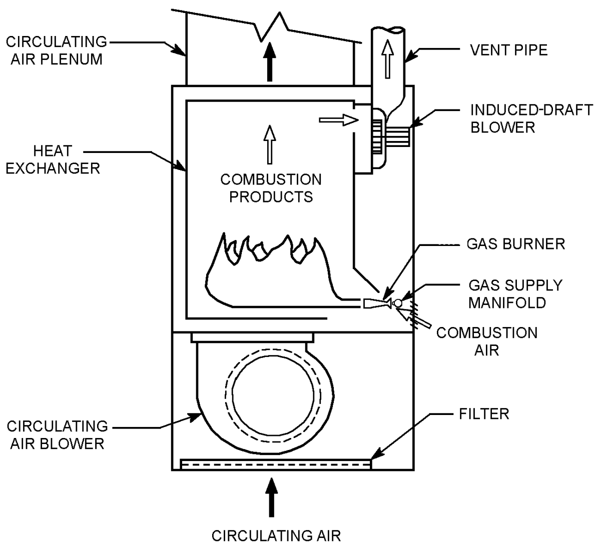


Fig. 19-4 Upflow Forced-Warm-Air Furnace
(Figure 2, Chapter 33, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

exchanger, and discharges out of the opposite end. Such furnaces are used for locations with limited head room (e.g., attics and crawlspaces). These units are often designed to allow for component rearrangement so that airflow may be from left to right or from right to left.

2. The blower in an **upflow furnace** is beneath the heat exchanger and discharges vertically upward (Figure 19-4). Air enters through the bottom or the side of the blower compartment and leaves at the top. Such furnaces may be used in utility rooms on the first floor of homes without basements or in basements with the return air ducted down to the blower compartment entrance.
3. The **basement furnace**, which is a variation of the upflow furnace, requires less head room (Figure 19-5). The blower is located at the bottom beside the heat exchanger. Air enters the top of the cabinet, flows down through the blower, discharges over the heat exchanger, and leaves vertically at the top.

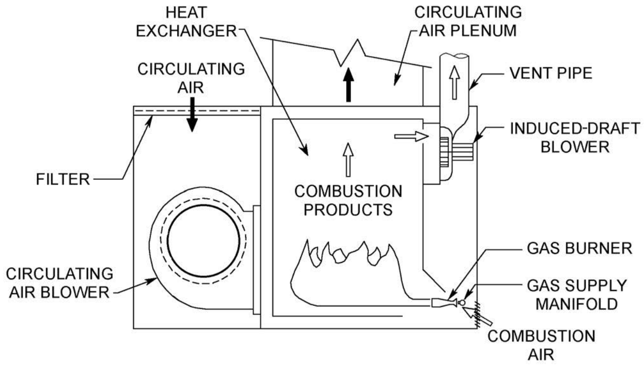


Fig. 19-5 Basement Forced-Warm-Air Furnace
(Figure 5, Chapter 33, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

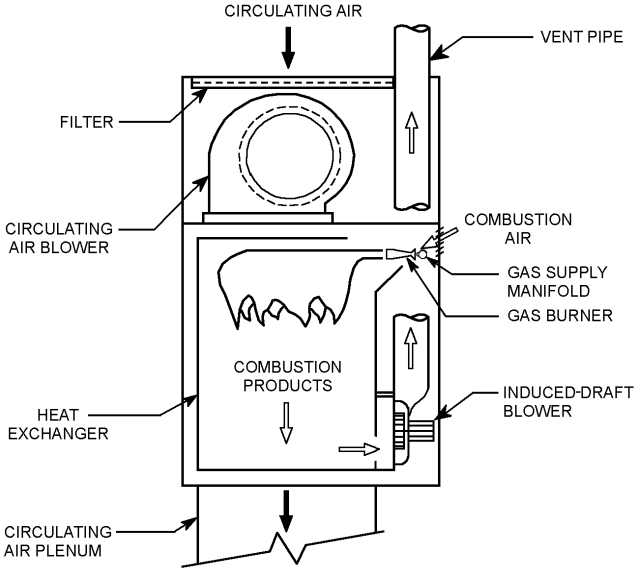


Fig. 19-6 Downflow (Counterflow) Warm-Air Furnace
(Figure 3, Chapter 33, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

4. The blower in the **downflow** or **counterflow furnace** is located above the heat exchanger and discharges downward (Figure 19-6). Air enters at the top and discharges vertically at the bottom. This style of furnace is used in a house without a basement that has a perimeter heating system.

Further discussion about furnaces and space heaters can be found in the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

19.3.2 Capacity Selection

Heating capacity depends on several variables that may operate alone or in combination.

Design heating requirement of building. The heat loss of the structure can be calculated by using the procedures in Chapters 6 or 7.

Additional heating required if furnace is operating on night setback cycle. During the morning recovery period,

additional capacity is required to bring the conditioned space temperature up to the desired level.

Internal loads. Normally, the heat gain from internal loads is neglected when selecting a furnace, but if the internal loads are constant, they should be used to reduce the required capacity of the furnace. This is particularly applicable in nonresidential applications.

Energy required for humidification. Humidification energy depends on the desired level of relative humidity and the rate at which moisture must be supplied to maintain the specified level. Net moisture requirements must take into account internal gains as the result of people, equipment, and appliances, and losses through migration in exterior surfaces, plus air infiltration.

Influence of off-peak storage devices. A storage device, when used in conjunction with a furnace, decreases the required capacity of the furnace. The storage device can supply the additional capacity required during the morning recovery of a night setback cycle or reduce the daily peak loads to assist in load shedding.

Influence of backup systems. A furnace can exist as a backup to a solar system, a heat recovery system, or a structure requiring multiple units for uninterrupted service. Oversizing results in higher initial costs, possible increased operating costs, and decreased comfort control. Undersizing produces unacceptable comfort control near the design conditions.

Capacity to accommodate air conditioning. Even if air conditioning is not initially planned, the cabinet should be large enough to accept a cooling coil that satisfies the design cooling load. The blower and motor should have sufficient capacity to provide increased airflow rates typically required in air-conditioning applications.

19.3.3 Performance Criteria

Some typical efficiencies encountered in performance criteria of a furnace are (1) steady-state efficiency, (2) use efficiency, and (3) annual fuel use efficiency.

These efficiencies are generally used by the furnace industry in the following manner:

Steady-state efficiency is the efficiency of a furnace when it is operated under equilibrium conditions. It is calculated by measuring the energy input, subtracting the losses for exhaust gases, and then dividing by the fuel input (cabinet loss not included), i.e.,

$$\text{SS (\%)} = 100(\text{Fuel input} - \text{Flue loss} \\ - \text{Condensate loss}) / \text{Fuel input}$$

For furnaces tested under the isolated combustion system (ICS) method, cabinet heat loss must also be deducted from the energy input.

$$\text{SS (\%)(ICS)} = 100(\text{Fuel input} - \text{Flue loss} \\ - \text{Condensate loss} - \text{Jacket loss}) / \text{Fuel input}$$

Utilization efficiency is obtained from an empirical equation developed by Kelly et al. (1978) by starting with 100% efficiency and deducting losses for exhausted latent and sensible heat, cyclic effects, infiltration, and pilot-burner effect.

Table 19-5 Typical Values of Efficiency

(Table 1, Chapter 33, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

	AFUE, %	
Type of Gas Furnace	**Indoor**	**ICS[a]**
1. Natural-draft with standing pilot	64.5	63.9[b]
2. Natural-draft with intermittent ignition	69.0	68.5[b]
3. Natural-draft with intermittent ignition and auto vent damper	78.0	68.5[b]
4. Fan-assisted combustion with standing pilot or intermittent ignition	80.0	78.0
5. Same as 4, except with improved heat transfer	82.0	80.0
6. Direct vent, natural-draft with standing pilot, preheat	66.0	64.5[b]
7. Direct vent, fan-assisted combustion, and intermittent ignition	80.0	78.0
8. Fan-assisted combustion (induced-draft)	80.0	78.0
9. Condensing	90.0	88.0
Type of Oil Furnace	**Indoor**	**ICS[a]**
1. Standard—pre-1992	71.0	69.0[b]
2. Standard—post-1992	80.0	78.0
3. Same as 2, with improved heat transfer	81.0	79.0
4. Same as 3, with automatic vent damper	82.0	80.0
5. Condensing	91.0	89.0

[a] Isolated combustion system (estimate).
[b] Pre-1992 design (see text).

Annual fuel utilization efficiency (AFUE) is the same as **utilization efficiency**, except that losses from a standing pilot during nonheating season time are deducted. The equation for AFUE can also be found in Kelly et al. (1978) or ASHRAE Standard 103 (2007).

Federal law requires manufacturers of furnaces to use AFUE to rate efficiency. Typical values of AFUE for various types of furnaces are listed in Table 19-5.

19.4 Commercial Furnaces

19.4.1 Equipment Variations

The basic difference between residential and commercial furnaces is the size and heating capacity of the equipment. The heating capacity of a commercial furnace may range from 150,000 Btu/h (44 kW) to over 2,000,000 Btu/h (600 kW). Generally, furnaces with output capacities less than 320,000 Btu/h (93 kW) are classified as light commercial, and those above 320,000 Btu/h (93 kW) are large commercial equipment. In addition to the difference in capacity, commercial equipment is constructed from material having increased structural strength and more sophisticated control systems.

Light commercial heating equipment comes in many flow arrangements and design variations. Some are identical to residential equipment, while others are unique to commercial applications. Some commercial units function as a part of a ducted system; others operate as unducted space heaters.

Ducted Equipment. Upflow gas-fired commercial furnaces are available up to 300,000 Btu/h (88 kW) and supply enough airflow to handle up to 10 tons (35 kW) of air conditioning. These furnaces may develop high static pressure,

have belt-driven blowers, and frequently consist of two standard upflow furnaces tied together in a side-by-side arrangement.

Horizontal gas-fired duct furnaces are also available for built-in light commercial systems. This furnace is not equipped with its own blower but is designed for uniform airflow across the entire furnace.

Duct furnaces are normally certified for operation either upstream or downstream of an air-conditioner cooling coil. Electric duct furnaces are available in a great range of sizes and are suitable for operation in upflow, downflow, or horizontal positions. These units are also used to supply auxiliary heat with the indoor section of a split-heat pump.

The most common commercial furnace is the combination package unit, which is sometimes called a combination rooftop unit. They are available as air-conditioning units with liquified petroleum gas (LPG) and natural gas furnaces, electric resistance heaters, or heat pumps. Combination oil-heat-electric-cool units are not commonly available. Combination units come in a full range of sizes covering air-conditioning ratings from 5 to 50 tons (18 to 176 kW) with matched furnaces supplying heat-to-cool ratios of approximately 1.5:1.

Combination units of 15 tons (53 kW) and under are available as single-zone units. The entire unit must be in either the heating or cooling mode. All air delivered by the unit is at the same temperature. Large combination units in the 15 to 50 ton (50 to 180 kW) range are available as single-zoned units, as are small units; however, they are also available as multizone units. A multizone unit supplies conditioned air at several different zones of a building in response to individual thermostats controlling those zones. These units are capable of supplying heating to one or more zones at the same time cooling is being supplied to other zones.

Large combination units are normally available only in a curbed configuration; i.e., the units are mounted on a rooftop over a curbed opening in the roof. The supply and return air enter through the bottom of the unit. Smaller units may be available for either curbed or uncurbed mounting. In either case, the unit is always connected to ductwork within the building to distribute the conditioned air.

Unducted Heaters. Three types of commercial heating equipment used as unducted space heaters are unit heaters, infrared heaters, and floor furnaces.

Unit heaters are available from about 25,000 to 320,000 Btu/h (7 to 94 kW). Normally they are mounted from ceiling hangers and blow air across the heat exchanger into the heated space. Natural gas, LPG, and electric heat units are available.

Infrared heaters are mounted from ceiling hangers and transmit heat downward by radiation. Low- and medium-intensity infrared heaters are compact, self-contained, direct-heating devices. They are used in hangars, factories, warehouses, foundries, greenhouses, and gymnasiums and for areas such as loading docks, racetrack stands, under marquees, outdoor restaurants, and around swimming pools. Low-temperature radiant heaters are used in office buildings and other commercial buildings.

These heaters can be used with variable-air-volume systems. Infrared heating units may be electric, gas fired, or oil fired. They consist of an infrared source or generator operating in a temperature range of 350 to 5000°F (180 to 2800°C), with the specific temperature determined by the energy source, size, and application. Reflectors can be used to distribute radiation in specific patterns. Common configurations for gas-fired and electric infrared heaters are shown in Figures 19-7 and 19-8, respectively.

Radiant heaters transfer energy directly to solid objects. As floor and objects are warmed by the infrared energy, they, in turn, reradiate heat to solid objects, as well as transfer heat to the air by convection. Dry-bulb temperatures can be maintained slightly less than the mean radiant temperature. With convective heat, the converse is true. Since human comfort is determined by the average of mean radiant and dry-bulb temperatures, dry-bulb temperature may be lowered when heating with radiation. Heat loss to ventilating air and by transmission is correspondingly lower, as is energy consumption. Because buildings heated by infrared require less heating fuel for a given application, equipment that handles only 80 to 85% of the design heat loss calculated by ASHRAE methodology is typically installed.

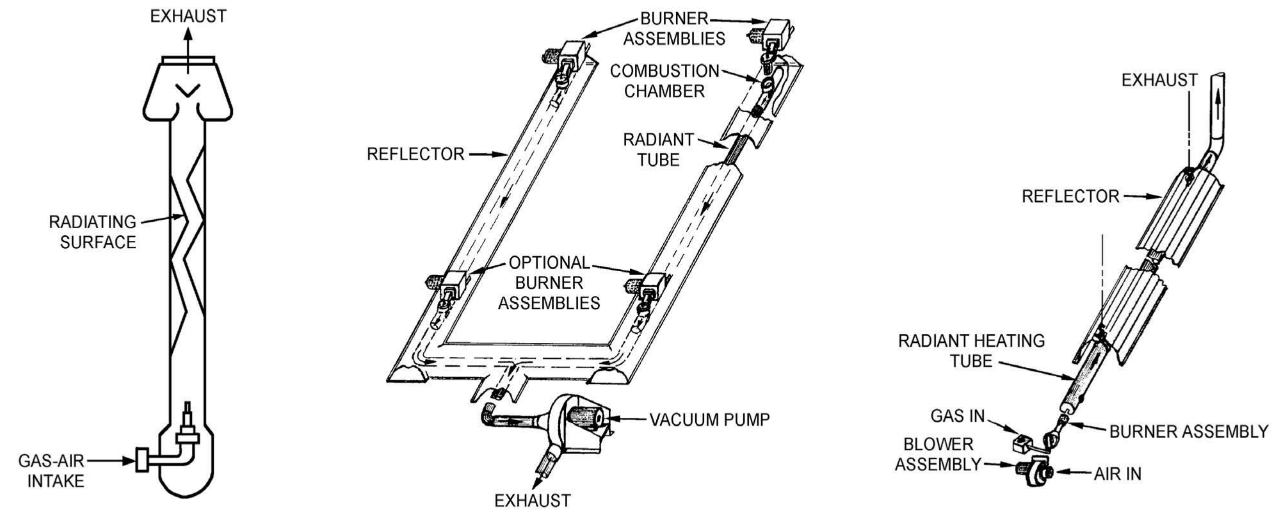
Floor furnaces are used as large area unducted heaters. They are available in capacities ranging from 200,000 to 2,000,000 Btu/h (60 to 590 kW). Floor furnaces direct heated air through nozzles for task heating or use air circulators to heat large industrial spaces.

19.4.2 System Design and Equipment Selection

The procedure for design and selection of a commercial furnace is similar to that for a residential furnace. First, the design capacity of the heating system must be determined, considering structure heat loss, recovery load, internal load, humidification, off-peak storage, waste heat recovery, and backup capacity. Since most commercial buildings use setback periods during weekends, evenings, or other long periods of inactivity, the recovery load is important, as are internal loads and waste heat recovery.

Commercial sizing criteria are essentially the same as for residential furnaces. The furnace should be oversized 30% above the total load if setback is anticipated. Since combination units must be sized accurately for the cooling load, it is possible that the smallest gas-fired capacity available is larger than the 30% oversize value. This is especially true for the warmer climates of the United States.

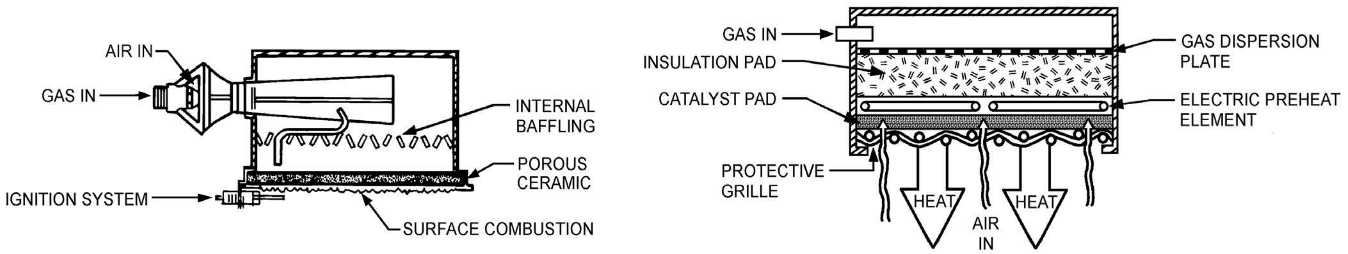
Commercial units have about the same efficiency as residential units. Two-stage gas valves are frequently used with commercial furnaces, but the efficiency of a two-stage system may be lower than that of a single-stage system. At a reduced firing rate, the excess combustion airflow through the burners increases, which decreases the steady-state operating efficiency of the furnace. Multistage furnaces with



A. INDIRECT INFRARED HEATER

B. INDIRECT TUBE-TYPE HEATER (VACUUM OPERATED)

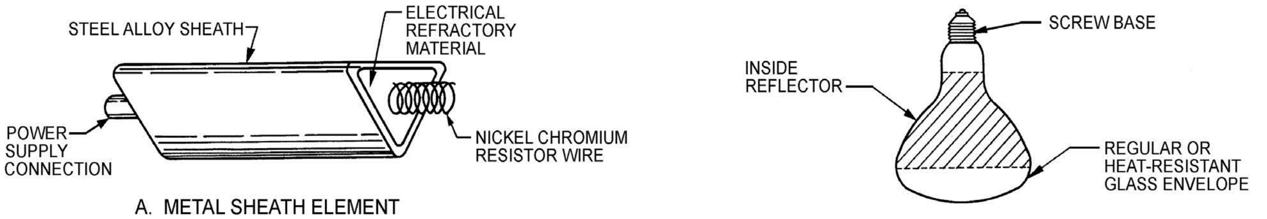
C. INDIRECT TUBE-TYPE HEATER (FORCED DRAFT)



D. POROUS-MATRIX INFRARED BURNER

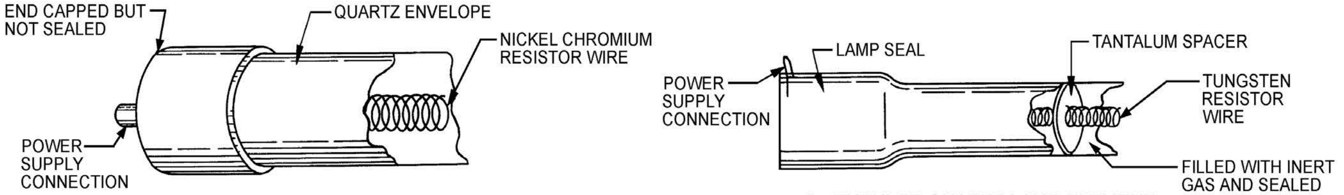
E. CATALYTIC OXIDATION INFRARED BURNER

Fig. 19-7 Gas-Fired Infrared Heaters
(Figure 1, Chapter 16, 2016 ASHRAE Handbook—HVAC Systems and Equipment)



A. METAL SHEATH ELEMENT

B. REFLECTOR LAMP ELEMENT



C. QUARTZ TUBE ELEMENT

D. TUBULAR QUARTZ LAMP ELEMENT

Fig. 19-8 Electric Infrared Heaters
(Figure 2, Chapter 16, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

multistage thermostats and controls provide more uniform distribution of heat within the building.

The useful life of commercial heating and cooling equipment is about 20 years.

19.5 Boilers

A boiler is a pressure vessel designed to transfer heat produced by combustion to a fluid. Heat is normally produced by combustion, but electrical resistance elements or electrodes acting directly on the fluid are also used. In boilers of interest to ASHRAE, the fluid is usually water, in the form of liquid or steam. If the fluid being heated is air, the heat-exchange device is called a furnace. The firebox, or combustion space, of some boilers is also called the furnace.

Excluding special and unusual fluids, materials, and methods, a boiler is a cast-iron, steel, or copper pressure vessel heat exchanger, designed with and for fuel-burning devices and other equipment (1) to burn fossil fuels (or use electric current) and (2) to transfer the released heat to water (in water boilers) or to water and steam (in steam boilers). Boiler heating surface is the area of fluid-backed surface exposed to the products of combustion, or the fire-side surface. Various codes and standards define allowable heat transfer rates in terms of heating surface. Boiler design provides for connections to a piping system, which delivers heated fluid to the point of use and returns the cooled fluid to the boiler.

19.5.1 Boiler Classifications

Boilers may be grouped into classes based on such criteria as working pressure and temperature, fuel used, shape and size, use (such as heating or process), and steam or water.

Working Pressure/Temperature. With few exceptions, all boilers are constructed to meet the ASME *Boiler and Pressure Vessel Code*.

Low-pressure boilers are constructed for maximum working pressures of 15 psi (103 kPa) steam and up to 160 psi (1100 kPa) hot water. Hot water boilers are limited to 250°F (120°C) operating temperature.

Medium- and high-pressure boilers are designed to operate above 15 psi (103 kPa) steam or above 160 psi (1100 kPa) water or 250°F (120°C) water boilers.

Steam boilers are available in standard sizes of up to 100,000 lb steam/h (60,000 to 100,000,000 Btu/h or 17 to 30,000 kW), many of which are used for space heating in both new and existing systems. On larger installations, they may also provide steam for auxiliary uses, such as hot-water heat exchangers, absorption cooling, laundry, and sterilizers. In addition, many steam boilers provide steam at various temperatures and pressures for a wide variety of industrial processes.

Water boilers are available in standard sizes of up to 100,000,000 Btu/h (30 MW) from 50,000 Btu/h (15 kW), many of which are in the low-pressure class and are used for space heating in both new and existing systems. Some water boilers may be equipped with either internal or external heat exchangers to supply domestic (service) hot water.

Every steam or water boiler is rated at the maximum working pressure determined by the ASME *Boiler Code Section* (or other code) under which it is constructed and tested. When installed, it must also be equipped with safety controls and pressure relief devices mandated by such code provisions.

Fuel Used. Boilers may be designed to burn coal, wood, various grades of fuel oil, various types of fuel gas, or to operate as electric boilers. A boiler designed for one specific fuel type may not be convertible to another type of fuel. Some boiler designs can be adapted to burn coal, oil, or gas. Several designs allow firing with oil or gas by burner conversion or by using a dual-fuel burner.

Construction Materials. Most boilers, other than special or unusual models, are made of cast iron or steel. Some small boilers are made of copper or copper-clad steel.

Cast iron boilers are constructed of individually cast sections, assembled into blocks (assemblies) or sections. Push or screw nipples, gaskets, or an external header join the sections pressure-tight and provide passages for the water, steam, and products of combustion. The number of sections assembled determines boiler size and energy rating. Sections may be vertical or horizontal, the vertical design being the most common. The boiler may be dry-base (the firebox is beneath the fluid-backed sections), wet-leg (the firebox top and sides are enclosed by fluid-backed sections), or wet-base (the firebox is surrounded by fluid-backed sections, except for necessary openings).

Steel boilers are fabricated into one assembly of a given size and rating, usually by welding. The heat-exchange surface past the firebox usually is an assembly of vertical, horizontal, or slanted tubes. The tubes may be firetube (flue gas inside, heated fluid outside) or watertube (fluid inside, hot gas outside). The tubes may be in one or more passes. Dry-base, wet-leg, or wet-base design may be used. Most small steel heating boilers are of dry-base, vertical firetube design. Larger boilers usually have horizontal or slanted tubes; both firetube and watertube designs are used. A popular design for medium and large steel boilers is the Scotch, or Scotch Marine, which is characterized by a central fluid-backed cylindrical firebox, surrounded by firetubes in one or more passes, all within the outer shell.

Cast iron boilers range in size from 35,000 to 13,000,000 Btu/h (10 to 3770 kW) gross output. Steel boilers range in size from 50,000 Btu/h (15 kW) to the largest boilers made.

Condensing or Noncondensing Boilers. Because higher boiler efficiencies can be achieved with lower water temperatures, condensing boilers purposely allow the flue gas water vapor in the boiler to condense and drain. Illustrated in Figure 19-9 is a typical relationship of overall boiler efficiency to return water temperature. The dew point of 130°F (55°C) shown varies with the percent of hydrogen in the fuel and the CO_2 (or excess air) in the flue gas.

Low return water temperatures and condensing boilers are particularly important because they are so efficient at part-

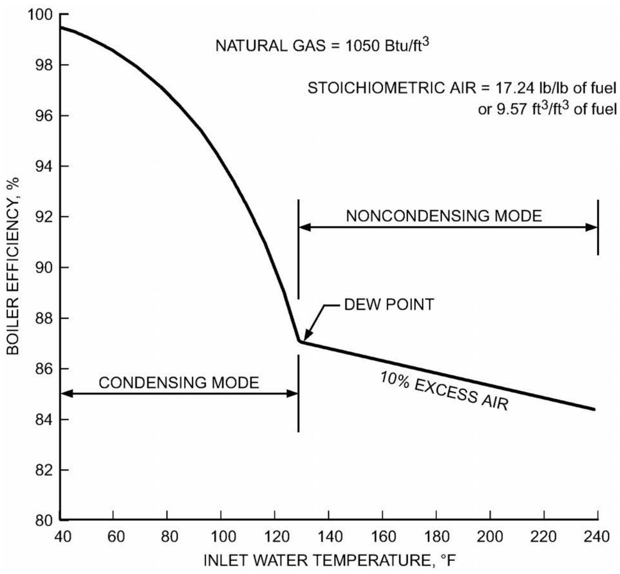


Fig. 19-9 Boiler Efficiency
(Figure 6, Chapter 32, 2016 ASHRAE Handbook—
HVAC Systems and Equipment)

load operation when high water temperatures are not required. For example, a hot water heating system that operates under light load conditions at 80°F (27°C) return water temperature has a potential overall boiler efficiency of 97% when operated with natural gas of the specifications applicable for Figure 19-9.

19.5.2 Selection Parameters

Boiler selection should be based on a competent review of the following parameters:

All Boilers

- ASME Code Section, under which the boiler is constructed and tested
- Net boiler output capacity, Btu/h (kW)
- Total heat transfer surface, ft^2 (m^2)
- Water content, lb (kg)
- Auxiliary power requirements, kWh
- Internal water-flow patterns
- Cleaning provisions for all heat transfer surfaces
- Operational efficiency
- Space requirements and piping arrangement
- Water treatment requirements
- Operating personnel requirements
- Maintenance requirements
- Regulatory emission limitations

Fuel-Fired Boilers

- Combustion space (furnace volume), ft^2 (m^2)
- Internal flow patterns of combustion products
- Combustion air and venting requirements
- Fuel availability

Steam Boilers

- Steam space, ft^3 (m^3)
- Steam quality

Electric Boilers

Electric boilers are in a separate class. Since no combustion occurs, no boiler heating surface and no flue openings are necessary. Heating surface is the surface of the electric elements or electrodes immersed in the boiler water. The design of electric boilers is largely determined by the shape and heat release of the electric heating elements used.

19.5.3 Efficiency: Input and Output Ratings

Efficiency of fuel-burning boilers is defined by combustion efficiency and overall efficiency. Overall efficiency of electric boilers is in the 92 to 96% range.

Combustion efficiency is input minus stack (chimney) loss, divided by input, and ranges from 75 to 86% for most noncondensing, mechanically fired boilers. Condensing boilers operate in the range of 88% to over 95% efficiency.

Overall efficiency is gross output divided by input. Gross output is measured in the steam or water leaving the boiler and depends on individual installation characteristics. Overall efficiency is lower than combustion efficiency by the percentage of heat lost from the outside surface of the boiler (this loss is usually termed **radiation loss**). Overall efficiency can be determined only by testing under fixed conditions. Approximate combustion efficiency can be determined under any operating condition by measuring operating flue-gas temperature and percentage CO_2 or O_2 and by consulting a chart or table for the fuel being used.

Heating boilers are usually rated according to standards developed by (1) The Hydronics Institute [formerly the Institute of Boiler and Radiator Manufacturers (IBR) and The Steel Boiler Institute (SBI)], (2) The American Gas Association (AGA), and (3) The American Boiler Manufacturers Association (ABMA).

19.5.4 Boiler Sizing

Boiler sizing is the selection of boiler output capacity to meet connected load. The boiler gross output is the rate of heat delivered by the boiler to the system under continuous firing at rated input. Net rating (IBR rating) is gross output minus a fixed percentage to allow for an estimated average piping heat loss plus an added load for initially heating the water in a system (sometimes called pickup).

Piping loss is variable. If all piping is run within the space defined as load, loss is zero. If piping runs through unheated spaces, heat loss from the piping may be higher than accounted for by the fixed net rating factor. Pickup is also variable. Pickup factor may be unnecessary when actual connected load is less than design load. On the design's coldest day, extra system output (boiler and radiation) is needed to pick up the load from a shutdown or low night setback. If night setback is not used, or if no extended shutdown occurs, no pickup load exists. Standby system capacity for pickup, if needed, can be in the form of excess capacity in base-load boilers or in a standby boiler.

19.5.5 Control of Boiler Input and Output

Boiler controls regulate the rate of fuel input (on-off, step-firing, or modulating) in response to a control signal representing load change, so that average boiler output equals load within some accepted control tolerance. Boiler controls include safety controls that shut off fuel flow when unsafe conditions develop.

Steam boilers are operated by boiler-mounted, pressure-actuated controls, which vary the input of fuel to the boiler. Common examples of controls are on-off, high-low-off, and modulating. Modulating controls continuously vary the fuel input from 100% down to a selected minimum point. The ratio of maximum to minimum is called the **turndown ratio**. The minimum input is usually between 5% and 25%, i.e., 20:1 to 4:1, and depends on the size and type of fuel-burning apparatus.

19.5.6 Hydronic Systems

Hot water heating systems are frequently called **hydronic systems**. Water systems can be classified by (1) temperature, (2) flow generation, (3) pressurization, (4) piping arrangement, and (5) pumping arrangement. Water systems are either once-through or recirculating systems.

The two types of hot water heating systems classified by flow generation are (1) the **gravity system**, which uses the difference in density between the supply and return water columns of a circuit or system to circulate water, and (2) the **forced system**, in which a pump, usually driven by an electric motor, maintains the flow.

Low-temperature hot water systems (LTHW) are the most widely used heating systems for residential, commercial, and institutional systems where loads consist primarily of space heating and domestic water heating and do not exceed 5,000,000 Btu/h (1.5 MW) total. The maximum allowable working pressure for low-pressure heating boilers is 160 psia (1100 kPa), with a maximum temperature limitation of 250°F (120°C). The usual maximum working pressure for boilers for LTHW systems is 30 psia (200 kPa), although boilers specifically designed, tested, and stamped for higher pressures may frequently be used with working pressures to 160 psia (1100 kPa). Steam-to-water or water-to-water heat exchangers are also often used.

Medium-temperature hot water systems (MTHW) (350°F [175°C] or less) are most commonly used for space heating in large commercial and institutional buildings or in industrial applications with process loads, and where total loads range from 5×10^6 to 20×10^9 Btu/h (1.5 to 6 MW). In a medium-temperature system, the usual design supply temperature is above 250°F (120°C) and below 350°F (175°C) with a usual pressure rating for boilers and equipment of 150 psia (1030 kPa).

High-temperature hot water systems (HTHW) are generally limited to campus-type district heating installations and to applications requiring process heating temperatures of 350°F (175°C) or higher.

The design of water systems is affected by complex relationships between the various system components. The design water temperature, flow rate, piping layout, pump selection, terminal unit selection, and control method are all interrelated. The size and complexity of the system determines the importance of these relationships in affecting the total operating success. Present hot water heating system design practice originated in residential heating applications where a 20°F (11°C) temperature drop (TD) was used to determine flow rate. Besides producing satisfactory operation and economy in small systems, this TD enabled simple calculations because 1 gpm conveys 10,000 Btu/h (1 L/s conveys 41 kW with a 10°C TD).

Elements of a high-temperature water system are illustrated in Figure 19-10. Requirements for such a system include a high limit control, a safety relief valve, and other safety controls and devices on the boiler, and a boiler efficiency with an accepted test rating.

19.6 Terminal Units

Radiators, convectors, baseboard, and finned tube units are commonly used to distribute heat to the space provided by the steam or LTHW systems. They supply heat through a combination of radiation and convection. In general, these units are placed at the points of greatest heat loss of the space such as under windows, along exposed walls, and at door openings.

A **radiator** is generally considered a sectional cast-iron unit of column, large tube, or small tube type (Fig. 19-11).

A **convector** is a heat-distributing unit that operates with gravity-circulated air. The heating element contains two or more tubes with headers at both ends. The heating element is surrounded by an enclosure with an air inlet below and an air outlet above (Fig. 19-11).

Baseboard and **baseboard radiation heat distributing units** are installed at the base of walls in place of the conventional baseboard. They may be made of cast iron, with a substantial portion of the front face directly exposed to the room, or with a finned-tube element in a sheet metal enclosure (Fig. 19-11).

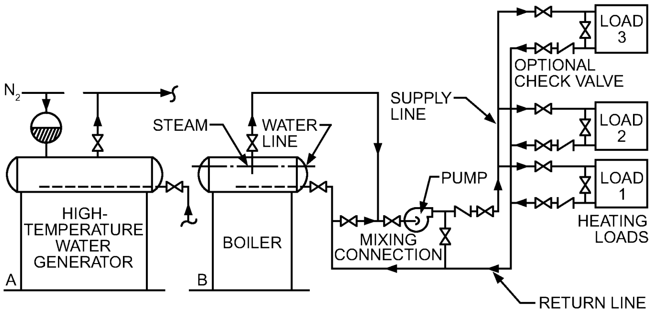


Fig. 19-10 Elements of High-Temperature Water System
(Figure 2, Chapter 15, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

Finned tube or **fin-tube** refers to heat-distributing units fabricated from metallic tubing with metallic fins bonded to the tube. They operate with gravity-circulated room air and may be installed bare, with an expanded metal grille, a cover, or an enclosure having top, front, or inclined outlets.

19.6.1 Radiators

The small-tube type radiators, with a length of only 1 3/4 in. (44 mm) per section, occupy less space than the older column and large-tube units and are particularly suited to installation in recesses. Shown in Figure 19-12 are the types of units now being manufactured.

19.6.2 Convectors

Convectors are made in a wide variety of depths, sizes, and lengths in both cabinet and enclosure style. The heating elements are fabricated in ferrous and nonferrous metals. The air enters the enclosure below the heating element, is heated as it passes through the element, and leaves the enclosure through the outlet grille located above the heating element. Factory-assembled units composed of a heating element and enclosure are widely used. These may be free standing, wall hung, or recessed and may have outlet grilles and arched inlets or inlet grilles (Figure 19-11).

19.6.3 Baseboard Units

Baseboard heat-distributing units are divided into two types: (1) radiant-convector and (2) finned tube.

A **radiant-convector baseboard heating unit** is made of cast iron or steel. The units have air openings at the top and bottom to permit circulation of room air over the wall side of the unit. The wall side of the unit has an extended surface to provide increased heat output. A large portion of the heat is transferred by convection (Fig. 19-11).

A **finned tube baseboard heating unit** has a finned-tube heating element that is concealed by a long, low, sheet-metal enclosure or cover. Most of the heat is transferred to the room by convection. Optimum comfort for room occupants is obtained when units are installed along as much of the exposed wall as possible.

The baseboard unit has the following advantages: (1) it is normally placed along the cold walls and under areas where the greatest heat loss occurs, (2) it is inconspicuous, (3) it offers minimum interference with furniture placement, and (4) it distributes heat near the floor. This last characteristic reduces the floor-to-ceiling temperature gradient to about 2 to 4°F (1 to 2°C) and helps produce uniform temperatures throughout the room. It also makes baseboard heat-distributing units adaptable to homes without basements, where cold floors are prevalent.

19.7 Electric Heating

For many applications, the compactness, simplicity, responsiveness, accuracy of control, safety, and cleanliness of electric heating may outweigh its disadvantages. Electric space heating is often used where minimum initial cost is the dominating factor.

19.7.1 Electric Heating Elements

Electric heating elements usually are composed of metal-alloy wire or ribbons, nonmetallic carbon compounds in rod or other shapes, or printed circuits. Heating elements may have exposed resistor coils mounted on insulators, metallic resistors embedded within refractory insulation encased in a protective metal sheath, or a printed circuit on glass sheets or vitrified panels. Fins or extended surfaces may be used to

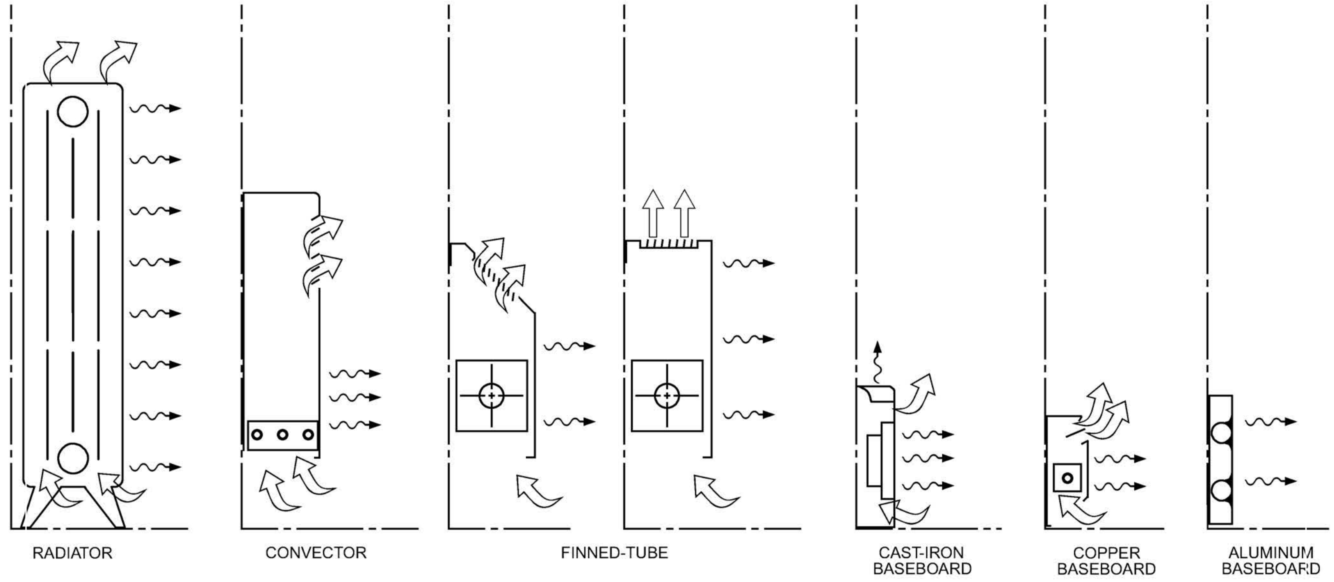


Fig. 19-11 Terminal Units
(Figure 1, Chapter 36, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

increase heat dissipation. Elements are made in many forms. Metal or oxide conductive films on glass and ceramics have been used, usually in panel form. Tubular elements may be immersed in liquids, used bare, formed into coils, or cast into metal.

Cloth fabrics, incorporating flexible resistor wires, are used for low-temperature purposes such as heating pads, sheets, blankets, aviators' clothing, window draperies, and some radiant panel heating installations. Paper or fabric incorporating a surface resistor on the back is applied for ceiling panel heating installations.

19.7.2 Electric Space Heating Systems

Types of electric heating equipment and complete heating systems in use are listed in Table 19-6. The sequence shown is for convenient reference only and does not indicate the relative performance or extent of use. Installations of all types listed are in successful operation, but operating characteristics depend largely on proper application.

19.7.3 Calculating Capacities

The procedure outlined in Chapters 17 and 18 of the 2017 *ASHRAE Handbook—Fundamentals* for calculating heating load of residential and nonresidential buildings may be used for electric systems. The NEMA *Manual for Electric Comfort Conditioning*, published by the National Electrical Manufacturers Association, gives heat loss factors and describes methods for calculating load directly in kilowatts.

The most economical electric heating systems from an operating standpoint are decentralized, with the thermostat provided on each unit or for each room. This permits each room to compensate for heat contributed by auxiliary sources such as sunshine, lighting, and appliances. Such an arrangement also gives a better diversity of the power demand because the electric loads from all units of an installation do not coincide. Manual switches allow the heat to be turned off or allow for reduced temperatures in rooms that

Table 19-6 Types of Electric Space-Heating Equipment

Decentralized Systems
Natural Convection Units
Floor drop-in units
Wall insert and surface-mounted heaters
Baseboard convectors
Hydronic baseboard convectors with immersion elements
Forced Air units
Unit ventilators
Unit heaters
Wall insert heaters
Baseboard heaters
Floor drop-in heaters
Radiant Units (High Intensity)
Radiant wall, insert, or surface-mounted (open ribbon or wire element)
Metal-sheathed element with focusing reflector
Quartz tube element with focusing reflector
Quartz lamp with focusing reflector
Heat lamps
Valance (cove) heaters
Radiant Panel Systems (Low Intensity)
Radiant ceiling with embedded conductors
Prefabricated panels
Radiant floor with embedded conductors
Radiant-convector panel heaters
Centralized Systems
Heated Water Systems
Electric boiler
Electric boiler with hydronic off-peak storage
Heat pumps
Integrated heat recovery systems
Steam Systems
Electric boiler with immersion element or electrode
Heated-Air Systems
Duct heaters
Electric furnaces
Heat pumps
Integrated heat recovery systems
Unit ventilators
Self-contained heating and cooling units
Storage units (ceramic, water)

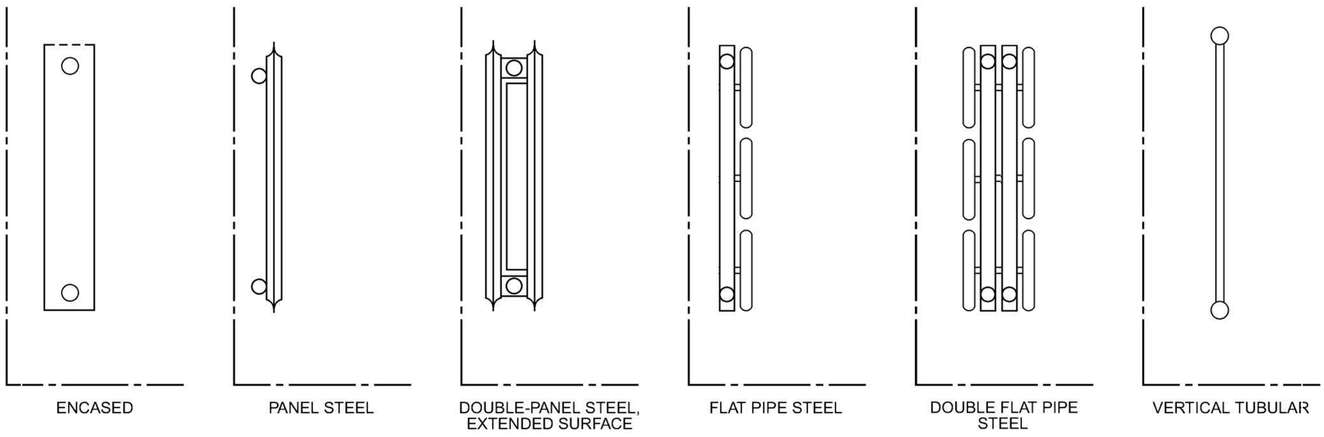


Fig. 19-12 Typical Radiators

(Figure 2, Chapter 36, 2016 ASHRAE Handbook—HVAC Systems and Equipment)

are not in use. When manual switches are used, adequate capacity for warmup should be provided.

Residences and other buildings designed for electric heat should be well insulated. Depending on the climate, storm or multiple-glazed windows and storm doors are generally recommended to reduce heat loss and further increase comfort. Weather stripping doors and windows effectively reduces air infiltration.

19.8 Problems

In all problems, assume air is 79% N_2 and 21% O_2.

19.1 Set up the necessary combustion equations and determine the mass of air required to burn 0.45 kg (1 lb) of pure carbon to equal masses of CO and CO_2.

19.2 The gravimetric analysis of a gaseous mixture is: CO_2 = 32%, O_2 = 54.5%, and N_2 = 11.5%. The mixture is at a pressure of 20.7 kPa (3 psia). Determine (a) the volumetric analysis and (b) the partial pressure of each component.

19.3 A liquid petroleum fuel, C_2H_6OH, is burned in a space heater at atmospheric pressure.

(a) For combustion with 20% excess air, determine the air/fuel ratio by mass, the mass of water formed by combustion per pound of fuel, and the dew point of the combustion products. [Ans: 11.45, 1.34, 133.9°F]
(b) For combustion with 80% theoretical air, determine the dry analysis of the exhaust gases in percentage by volume. [Ans: 5.94% $C0_2$, 11.04% CO, 83.02% N_2]

19.4 Find the air/fuel ratio by mass when benzene (C_6H_6) burns with theoretical air and determine the dew point at atmospheric pressure of the combustion products if an air/fuel ratio of 20:1 by mass is used.

19.5 A diesel engine uses 30 lb_m of fuel per hour (3.8 g/s) when the brake output is 75 hp. If the heating value of the fuel is 19,600 Btu/lb (45,600 kJ/kg), what is the brake thermal efficiency of the engine?

19.6 Methane (CH_4) is burned with air at atmospheric pressure. The Orsat analysis of the flue gas gives: CO_2 = 10.00%, O_2 = 2.41%, CO = 0.52%, and N_2 = 87.07%. Balance the combustion equation and determine the air-fuel ratio, the percent theoretical air, and the percent excess air. [Ans: 10.48 (vol.), 18.89 (mass), 110.1%, 10.1%]

19.7 Fuel oil composed of $C_{16}H_{32}$ is burned with the chemically correct air-fuel ratio. Find:

(a) Moisture formed per kg of fuel; moisture formed per lb of fuel
(b) Partial pressure of the water vapor, kPa; water vapor, psia
(c) Percentage of CO_2 in the stack gases on an Orsat basis
(d) Volume of exhaust gases per unit mass of oil, if the gas is at 260°C (500°F) and 102 kPa (14.8 psia).

19.8 Determine the composition of a hydrocarbon fuel if the Orsat analysis gives: CO_2 = 8.0%, CO = 1.0%, O_2 = 8.7%, and N_2 = 82.3%.

19.9 Determine the air/fuel ratio by mass when a liquid fuel of 16% hydrogen and 84% carbon by mass is burned with 15% excess air. [Ans: 17.49]

19.10 Compute the compositions of the flue gases on a percent by volume on dry basis (same as Orsat) resulting from the combustion of C_8H_{18} with 85% theoretical air.

19.11 A liquid petroleum fuel having a hydrogen to carbon ratio of 0.169 by mass is burned in a heater with an air/fuel ratio of 17 by mass. Determine (a) the volumetric analysis on both wet and dry bases of the exhaust gases and (b) the dew point of the exhaust gas.

19.12 Compare the heating value for semianthracite coal as given in Table 8, Chapter 28, 2009 *ASHRAE Handbook—Fundamentals* with the value predicted using the Dulong Formula. [Ans: 1.24% difference]

19.13 Natural gas with a volumetric composition of 93.32% methane, 4.17% ethane, 0.69% propane, 0.19% butane, 0.05% pentane, 0.98% carbon dioxide, and 0.61% nitrogen burns with 30% excess air. Calculate the volume of dry air at 60°F, 30 in. Hg (15.6°C, 101.5 kPa) used to burn 1000 ft^3 (28.3 mL) of gas at 68°F and 29.92 in. Hg (20°C and 101.4 kPa) and find the dew point of the combustion products.

19.14 The proximate analysis of a coal is: moisture = 4.33%, volatile matter = 40.21%, fixed carbon = 45.07%, and ash = 10.39%. The heating value was determined as 29 000 kJ/kg (12,490 Btu/lb). Find the ASTM rank of the coal.

19.15 A fuel oil shows an API gravity of 36. Calculate the specific gravity at 60°F and the pounds per gallon of fuel. Estimate the ASTM grade. [Ans: 0.845, 7.05, No. 2]

19.16 A representative No. 4 fuel oil has a gravity of 25 degrees API and the following composition: carbon = 87.4%, hydrogen = 10.7%, sulfur = 1.2%, nitrogen = 0.2, moisture = 0, and solids = 0.5%.

(a) Estimate its higher heating value.
(b) Compute the mass of air required to burn, theoretically, 1 gallon of the fuel.

19.17 The following data were taken from a test on an oil-fired furnace:

Fuel rate = 20 gal oil/h
Specific gravity of fuel oil = 0.89% by mass
Hydrogen in fuel = 14.7%
Temperature of fuel for combustion = 80°F
Temperature of entering combustion air = 80°F
Relative humidity of entering air = 45%
Temperature of flue gases leaving furnace = 550°F

(a) Calculate the heat loss in water vapor in products formed by combustion.
(b) Calculate the heat loss in water vapor in the combustion air.

[Ans: 1672.5 Btu/lb (3888 kJ/kg), 29.4 Btu/lb (68.3 kJ/kg)]

19.18 An office building requires 2901 MJ (2.75×10^9 Btu) of heat for the winter season. Compute the seasonal heating costs, if the following fuel is used:

(a) Bituminous coal: 31,380 kJ/kg (13,500 Btu/lb), $70.00 per ton
(b) No. 2 fuel oil: 38,500 kJ/L (38,000 Btu/gal), $2.75 per gallon.

Assume that the conversion efficiency is 75% for the oil and 61% for the coal.

19.19 Saturated air at 41°F dry bulb (5°C) enters a furnace; it leaves the furnace at 110°F dry bulb (43.3°C) and 0.00543 lb_v/lb_a (0.00543 kg/kg) and circulates through a factory. Air leaves the factory at 65°F dry bulb (18.3°C) and 63°F wet bulb (17.2°C).

(a) What is the sensible and latent heat change for the air passing through the factory?
(b) State whether the air gains or loses sensible and latent heat during each process. [Ans: q_s = –10.8 Btu/lb (–25.2 kJ/kg), q_l = +7 Btu/lb (16.3 kJ/kg)]

19.20 A plant is maintained at 70°F dry bulb, 60% rh, and has a low-pressure steam heating system. A makeup air system is being added to the plant and it has been decided that the input air should be 10,127 cfm. Outside design conditions are –1°F dry bulb, 50% rh. The plant is 250 ft by 560 ft and normally has 325 people working per shift.

(a) What are the total steam requirements for the heating coil and the humidifier?
(b) What capacity should the humidifier have in pounds of water per hour?

19.21 A residence with a design heating load of 26 kW (89,000 Btu/h) is to use an oil-fired warm air system with forced circulation. Return air to the furnace is at 22.2°C (72°F). Specify the following:

This problem requires catalog data.

(a) Supply air temperature
(b) Airflow rate
(c) Make and catalog number of suitable furnace

19.22 A residence with a design heating load of 16 kW (55,000 Btu/h) is to use a forced circulation hot-water baseboard radiator system. The baseboard units house copper tubing with aluminum fins and operate with the inlet air temperature at 18.3°C (65°F). Specify the following:

This problem requires catalog data.

(a) Hot water inlet temperature and outlet temperature
(b) Total water flow rate
(c) Total length of radiator panel for house
(d) Location of panels
(e) Make and catalog number of suitable hot water heater

19.23 For the residence of Problem 19-22, electric baseboard units replace the hot water system. Specify:

This problem requires catalog data.

(a) Total rating of electric system, kW
(b) Total length of baseboard units

19.24 A large classroom has a winter design heat loss of 19.9 kW (68,000 Btu/h) with installed forced circulation hot-water baseboard radiators. The baseboard units house copper tubing with aluminum fins and operate with the inlet air temperature at 18.3°C (65°F). Specify the following:

This problem requires catalog data.

(a) Hot water inlet temperature and outlet temperature
(b) Water flow rate
(c) Length of radiator panel

19.25 A large classroom has a winter design load of 26 kW (89,000 Btu/h). A forced circulation warm air system is to be used with return air at 23.3°C (74°F). Specify

(a) Supply air temperature
(b) Airflow rate
[Ans: $t_s \approx$ 135°F, 1330 cfm]

19.26 For a heat loss from the space to be conditioned of Q, write the expression for determining

(a) Amount of air L/s (cfm) that must be supplied if a hot air system is used
(b) Amount of hot water L/s (gpm) that must be supplied if a hydronic system is used
(c) Amount of steam kg/h (lb/h) that must be supplied if a steam heating system is used
(d) Size, in watts, of electric heaters required if electric heat is used

19.27 List the steps taken when designing a forced-circulation hot water heating system.

19.28 Compute the increase in length of 28.3 m (93 ft) of steel steam pipe when the average steam temperature is 113°C (235°F) and the air is 21°C (70°F). The pipe was installed during a period when the temperature was 15.6°C (60°F). [Ans: 1.27 in.]

19.29 The total mass of steel in the boiler and piping of a school's heating system is 9080 kg (20,000 lb). The piping and boiler also contain 6810 kg (15,000 lb) of water. After a weekend shutdown, the temperature of the system is 10°C (50°F). The operating temperature is 93°C (200°F).

(a) Assuming the system should be warmed up in one hour, determine the required furnace size.
[Ans: 764 kW (2,610,000 Btu/h)]
(b) For a furnace size of 146 kW (500,000 Btu/h) output, when should the furnace be started to be up to the operating temperature of 93°C (200°F) by 7:30 AM Monday morning? [Ans: 2:17 AM Sunday]

19.9 Bibliography

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Chapter 20

HEAT EXCHANGE EQUIPMENT

Heat transfer plays a vital role in heating, refrigerating, and air-conditioning as can be seen by the many sections in previous chapters dealing with equipment whose main function is the exchange of heat, thus providing either heating or cooling. The *ASHRAE Handbook* series has a number of chapters devoted to heat transfer and heat transfer applications: in the *2017 Fundamentals*, Chapter 4, Heat Transfer, and Chapter 5, Two-Phase Flow; and 2016 *Systems and Equipment*, Chapter 23, Air-Cooling and Dehumidifying Coils, Chapter 27, Air-Heating Coils, and Chapter 48, Heat Exchangers. This chapter briefly reviews the fundamentals of *applied* heat transfer and illustrates the basic approach to heat exchanger design and analysis.

20.1 Modes of Heat Transfer

Heat transfer or ***heat*** (the "transfer" is redundant) can be defined as the transfer of energy from one region or one body to another due to a ***temperature difference***. Heat transfer is as universal as gravity since differences in temperature exist all over the universe. Unlike gravity, however, heat transfer is governed not by a single relationship but by a combination of various independent laws of physics. Heat transfer is generally divided into three distinct modes: *conduction*, *convection*, and *radiation*. Strictly speaking, only conduction and radiation are both separate and purely heat transfer processes, since convection also involves mass transfer and includes conduction.

20.1.1 Conduction

Conduction is the mode of heat transfer whereby energy is transported between parts of an opaque, stationary medium or between two media in direct physical contact. In gases, conduction is due to the elastic collision of molecules; in liquids and electrically non-conducting solids, it is due primarily to longitudinal oscillations of the lattice structure. In metals, thermal conduction takes place in the same manner as electrical conduction; that is, with the movement of the free electrons.

The theory of heat transfer by conduction was first proposed by Jean B. Fourier in a noted work, published in 1822 in Paris, titled *Theorie analytique de la chaleur*. Fourier's law gives the heat transfer rate past any plane by the following:

$$q = -kA\frac{\partial T}{\partial n} \tag{20-1}$$

where k is the thermal conductivity of the material, A is the area normal to the flow of heat, T is the temperature, and n is the distance in the direction of heat flow. The partial derivative $\partial T/\partial n$ is the temperature gradient in the direction of the heat flow. The minus sign indicates that heat flows of its own accord only in the direction of decreasing temperature (from hot to cold), in accordance with the Second Law of Thermodynamics. The thermal conductivity k is the specific property of matter that indicates a material's ability to transfer heat, expressed as energy transferred per unit time per unit area per unit temperature gradient. Table 20-1 provides a few order of magnitude values of thermal conductivity.

If heat flows in more than one direction, or there are temperature variations in more than one direction, Equation 20-1 cannot be directly integrated but is only a start in the development of the three-dimensional *general conduction equation*. However, for the following two simple, but important, cases to HVAC applications shown in Figure 20-1, direct integration is possible with these results:

Case 1. The slab or plane wall under steady-state conditions:

$$q = kA\frac{(T_1 - T_2)}{L} = \frac{T_1 - T_2}{L/(kA)} \tag{20-2}$$

where L is the wall thickness, T_1 is the temperature at $x = 0$, and T_2 is the temperature at $x = L$. The quantity $L/(kA)$ can be considered the "thermal resistance" to the flow of heat.

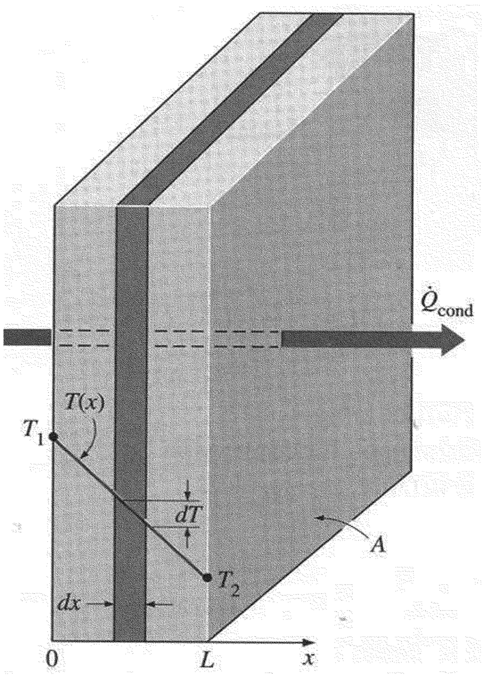
Case 2. The hollow cylinder (tubes, pipes, etc.) under steady-state conditions:

$$q = \frac{2\pi kL(T_1 - T_2)}{\ln(r_2/r_1)} = \frac{T_1 - T_2}{[\ln(r_2/r_1)/(2\pi kL)]} \tag{20-3}$$

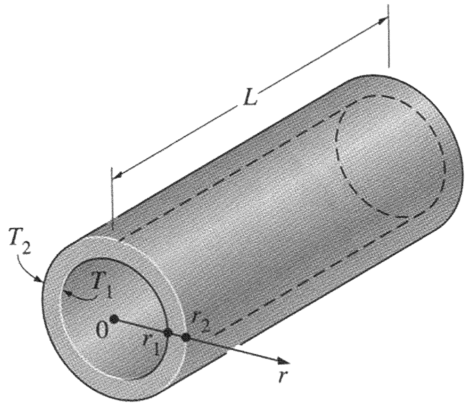
where L is the length of the cylinder, T_1 is the temperature at the inner radius r_1, and T_2 is temperature at the outer radius r_2. Here, the quantity $\ln(r_2/r_1)/(2\pi kL)$ is considered the "thermal resistance."

20.1.2 Convection

Convection is the mode of heat transfer whereby energy is transported by the combined action of conduction, energy storage, and mixing motion. Convection occurs between a solid surface and a moving fluid. When the fluid movement is produced by other than the heat transfer process itself



A. SLAB



B. HOLLOW CYLINDER

Fig. 20-1 Fourier's Law Applied to Two Simple Cases: (a) Slab and (b) Hollow Cylinder

(such as by a fan or pump), the convection is termed *forced* convection. When the only motion is due to the heat transfer and the fact that warmer fluids are less dense and will naturally tend to rise, the convection is termed *free* or *natural* convection. A combination of free and forced convection may occur and is termed *mixed* convection.

The quite complex phenomenon of convection was analyzed by Sir Isaac Newton in 1701 resulting in the often called "Newton's Law of Cooling,"

$$q = hA(T_s - T_f) = \frac{(T_s - T_f)}{(1/hA)} \qquad (20\text{-}4)$$

Table 20-1 Order of Magnitude of Thermal Conductivity at Room Temperature

Material	k, W/(m·K)	K, Btu/h·ft·°F
Copper	400	220
Aluminum	200	110
Mild Steel	64	40
Stainless Steel	15	10
Concrete	1.4	0.8
Glass	1.1	0.6
Water	0.6	0.4
Wood	0.10	0.06
Polyvinyl chloride (PVC)	0.10	0.06
Fiberglass (medium density)	0.04	0.02
Air	0.04	0.02

where h is the convective heat transfer coefficient (also called the "film coefficient"), A is the surface area in contact with the fluid, T_s (T_w sometimes used) is the surface or wall temperature, and T_f (T_∞ often used) is the fluid temperature outside the boundary layer. For convection, the quantity $1/hA$ is taken as the thermal resistance. Unfortunately, Equation (20-4) is actually no more nor less than the defining equation for the convection coefficient, h, and should be written:

$$h = q/[A(T_s - T_f)]$$

The equations for the governing laws for convection actually consist of five partial differential equations, namely:

- Conservation of Mass Equation
- Conservation of Energy Equation
- Momentum Equations (three, one per direction).

The h value could be obtained from solving these equations for q, which is then substituted in Eq. (20-4). Due to the mathematical complexity of the problem, most available values of the convection coefficient have been determined experimentally with empirical correlations provided for future reference and use in similar situations.

In almost every case, the fluid properties found in the correlations for predicting h depend significantly on temperature. In the case of density (and kinematic viscosity), there is also a pressure effect for gases. The temperature dependence means that there may be a significant variation in the quantities through the region of fluid near the surface (through the *boundary layer*). The accuracy of the predictive correlations depends upon the temperature(s) used for evaluating these thermodynamic and thermophysical properties. For convection occurring on an exterior surface, the *film temperature*, which is the average of the surface temperature and the undisturbed fluid temperature, is normally required. For internal flow, the *bulk mean fluid temperature* (also called the *average bulk temperature*), which is generally the average of the mean fluid inlet temperature and the mean fluid outlet temperature, is used as the temperature at which to evaluate the fluid properties. The bulk temperature is also called the "mixing cup" temperature as it represents an energy weighted average temperature. For viscous fluids, the

Table 20-2 Approximate Ranges of Convective Heat Transfer Coefficients

Flow and Fluid	h_c, W/(m^2·K)	h, Btu/h·ft^2·°F
Free convection, air	10	2
Free convection, water	50	10
Forced convection, air	100	20
Forced convection, water	500–10,000	100
Condensing vapor	5000–50,000	1000–10,000
Boiling liquid	1000–100,000	200–20,000

correlation may also include the viscosity evaluated at the surface temperature.

Many engineering applications involve convection transport in noncircular tubes. At least to a first approximation, many of the circular tube results may be applied by using an effective diameter, also termed the hydraulic diameter, as the characteristic length and is defined as

$$D_e = D_h = 4A_c/P$$

where A_c and P are the flow cross-sectional area and the wetted perimeter, respectively. It is this diameter that is used in the calculation of Re_D and the Nu_D.

Table 20-2 provides the typical range of convective heat transfer coefficients for several common processes.

20.1.3 Radiation

Radiation is the mode of heat transfer wherein energy is emitted by one surface (converted from internal energy), transmitted as electromagnetic waves, and then absorbed by a receiving surface. All bodies emit radiant heat continually, with the intensity depending upon the temperature and the nature of the surface. Radiant heat is emitted by a body in the form of finite patches, or quanta, of energy. Their motion in space is similar to the propagation of light and is approximated as traveling in a straight line (slight curvature is neglected). There are many types of electromagnetic radiation, including radio waves, x-rays, gamma rays, as well as light and thermal radiation. The thermal radiation region is from about 0.1 to 100 microns, whereas the visible light portion of the spectrum is very narrow, extending from about 0.40 to 0.65 microns.

Conduction and convection heat transfer rates are driven primarily by temperature gradients and somewhat by temperature due to temperature-dependent properties; however, radiative heat transfer rates are driven by the fourth power of the absolute temperature and increase rapidly with temperature. Unlike conduction and convection, no medium is required to transmit electromagnetic energy and thermal radiation is assumed to pass undiminished through a vacuum and transparent gases. Although the rate of emission of energy is independent of the surroundings, the net heat transfer rate by radiation depends on the temperatures and spatial relationships of all surfaces involved.

The starting point for analyzing radiation heat transfer is with the answers to the three questions:

1. How much radiation is emitted (sent out) by any body?
2. Where does it go?
3. What happens when it gets there?

Depending upon the application, there are actually three answers to Question1:

1a. How much radiation is emitted at a particular wavelength?
1b. How much radiation is emitted over all wavelengths?
1c. How much radiation is emitted between any two wavelengths?

Radiation Emitted. The rate at which thermal radiation is emitted by an ideal surface (perfect emitter) is dependent on its absolute temperature and the wavelength. Such a surface is also a perfect absorber (i.e., it absorbs all incident radiant energy) and is called a ***blackbody***. Planck in 1901 showed that the spectral distribution of energy radiated by a blackbody at an absolute temperature T is given by:

$$E_{b\lambda} = \frac{C_1\lambda^{-5}}{e^{C_2/\lambda T} - 1} \tag{20-5}$$

where

λ = wavelength, μm
T = temperature, K
C_1 = 3.743×10^8 W·μ/m^2 (1.187×10^8 Btu·μm^4/h·ft^2)
C_2 = 1.4387×10^4 μm$^{/K}$ (2.5896×10^4 μm·°R)

The symbol $E_{b\lambda}$ is used to denote the emitted flux per wavelength (monochromatic emissive power) and is defined as the energy emitted per unit surface area at wavelength lambda per unit wavelength interval around lambda. Equation (20-5), called Planck's distribution law, or just ***Planck's law***, is the basic equation for thermal radiation.

The total radiation emitted over all wavelengths for a blackbody may be obtained by the direct integration of Planck's law:

$$E_b = \int_0^{\infty} E_{b\lambda}\, d\lambda = \sigma T^4 \tag{20-6}$$

The constant σ has the value of 5.67×10^{-8} W/m^2·K^4 (0.1714×10^{-8} Btu/h·ft^2·°R^4). This expression was deduced by Stefan in 1879 from experimental data. Boltzmann, in 1884, using classical thermodynamics, derived the expression and placed it on firm theoretical ground. The equation is called the ***Stefan-Boltzmann law***.

The radiation emitted between any two wavelengths can also be obtain by integrating Planck's law with the two wavelengths as upper and lower limits of the integral. However, the integration is not easy but fortunately has been accomplished and recorded as ***blackbody radiation functions*** in form(s) readily used, such as provided in Table 20-3.

The blackbody radiation function, $f_{0\text{-}1}$, represents the fraction of radiation emitted from a blackbody at temperature T in the wavelength range from λ = 0 to λ. The values of *f* are

Table 20-3 Blackbody Radiation Function, $f_{0-\lambda}$

λT, μm·K	f_λ	λT, μm·K	f_λ
200	0.000000	6200	0.754140
400	0.000000	6400	0.769234
600	0.000000	6600	0.783199
800	0.000016	6800	0.796129
1000	0.000321	7000	0.808109
1200	0.002134	7200	0.819217
1400	0.007790	7400	0.829527
1600	0.019718	7600	0.839102
1800	0.039341	7800	0.848005
2000	0.066728	8000	0.856288
2200	0.100888	8500	0.874608
2400	0.140256	9000	0.890029
2600	0.183120	9500	0.903085
2800	0.227897	10,000	0.914199
3000	0.273232	10,500	0.923710
3200	0.318102	11,000	0.931890
3400	0.361735	11,500	0.939959
3600	0.403607	12,000	0.945098
3800	0.443382	13,000	0.955139
4000	0.480877	14,000	0.962898
4200	0.516014	15,000	0.969981
4400	0.548796	16,000	0.973814
4600	0.579280	18,000	0.980860
4800	0.607559	20,000	0.985602
5000	0.633747	25,000	0.992215
5200	0.658970	30,000	0.995340
5400	0.680360	40,000	0.997967
5600	0.701046	50,000	0.998953
5800	0.720158	75,000	0.999713
6000	0.737818	100,000	0.999905

listed in Table 20-3 as a function of λT, where λ is in μm and T is in K. The fraction of radiant energy emitted by a blackbody at temperature T over a finite wavelength band from λ_1 to λ is determined from

$$f_{\lambda_1 - \lambda_2}(T) = f_{\lambda_2}(T) - f_{\lambda_1}(T) \qquad (20\text{-}7)$$

Radiation Between Any Two Surfaces. Since thermal radiation is taken to travel in straight lines, the determination of "how much goes where" becomes a matter of geometry. Hence the factor used to quantitatively describe the fraction of the radiation leaving a surface and going to another surface is called the **geometric factor, angle factor, configuration factor,** or **view factor**. ***View factor*** will be used here.

The view factor F_{ij} by definition is the ***fraction*** of the ***total radiation*** leaving surface i that ***directly*** falls upon surface j. Figure 20-2 shows the geometry for determining the view factor between two surfaces, 1 and 2. The resulting equation for the view factor is

$$\begin{aligned} F_{12} &= F_{A_1 \to A_2} \\ &= \frac{1}{A_1}\int_{A_2}\int_{A_1} \frac{\cos\theta_1 \cos\theta_2}{\pi r^2} dA_1 dA_2 \end{aligned} \qquad (20\text{-}8)$$

where dA_1 and dA_2 are elemental areas of the two surfaces, r is the distance between dA_1 and dA_2, and θ_1 and θ_2 are the angles between the respective normals to dA_1 and dA_2 and the connecting line r. The solution of this equation in closed form is difficult, if not impossible, for all geometries. Numerical, graphical, and mechanical techniques have all provided alternative methods, and numerical values of the view factor for many geometries encountered in engineering may be found in the literature. It must be emphasized that the expression for the view factor is based on the assumption that the directional distribution of radiation leaving a surface is diffuse and uniformly distributed.

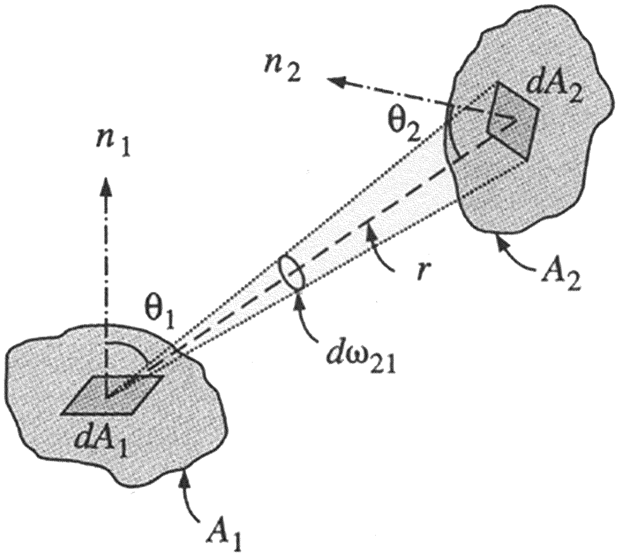


Fig. 20-2 View Factor Nomenclature

Two special properties play a very important role in obtaining numerical values for the complete set of view factors between the surfaces exchanging radiation. If there are n surfaces forming an enclosure, then

$$\sum_{j=1}^{n} F_{ij} = 1 \qquad (20\text{-}9)$$

The other is the reciprocal relationship: $F_{ij}A_i = F_{ji}A_j$. In both cases, it is important to note that F_{jj} is not necessarily 0 since a concave surface may irradiate ("see") itself.

Numerical values of view factors for three common geometries are provided in Table 20-4 and Figure 20-3.

Radiation Falling on Surface. When radiant energy falls on a surface, portions may be absorbed in, reflected from, or transmitted through the material as shown in Figure 20-4.

Therefore, based on conservation of energy,

$$\alpha + \rho + \tau = 1 \qquad (20\text{-}10)$$

where

α = fraction of incident radiation absorbed (absorptivity or absorptance)

ρ = fraction of incident radiation reflected (reflectivity or reflectance)

Table 20-4 View Factors for Three-Dimensional Geometries

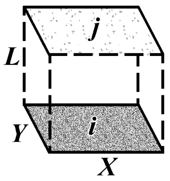
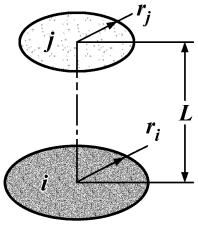
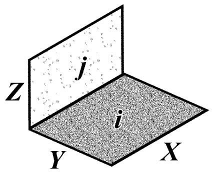
Geometry	Relation
Aligned parallel rectangles  	$\bar{X} = X/L, \qquad \bar{Y} = Y/L$ $F_{ij} = \frac{2}{\pi\bar{X}\bar{Y}}\{\ln\left[\frac{(1+\bar{X}^2)(1+\bar{Y}^2)}{1+\bar{X}^2+\bar{Y}^2}\right]^{1/2}$ $+\bar{X}(1+\bar{Y}^2)^{1/2}\tan\frac{\bar{X}}{(1+\bar{Y}^2)^{1/2}}$ $+\bar{Y}(1+\bar{X}^2)^{1/2}\tan^{-1}\frac{\bar{Y}}{(1+\bar{X}^2)^{1/2}}$ $-\bar{X}\tan^{-1}\bar{X} - \bar{Y}\tan^{-1}\bar{Y}\}$
Coaxial parallel disks  	$R_i = R_i/L, \qquad R_j = r_j/L$ $S = 1 + \frac{1+R_j^2}{R_i^2}$ $F_{ij} = \frac{1}{2}\left\{S - [S^2 - 4(r_j/r_i)^2]^{1/2}\right\}$
Perpendicular rectangles with a common edge  	$H = Z/X, \qquad W = (Y/X)$ $F_{ij} = \frac{1}{\pi W}\Big(W\tan^{-1}\frac{1}{W} + H\tan^{-1}\frac{1}{H}$ $-(H^2+W^2)^{1/2}\tan^{-1}\frac{1}{(H^2+W^2)^{1/2}}$ $+\frac{1}{4}\ln\{\frac{(1+W^2)(1+H^2)}{1+W^2+H^2}$ $\times\left[\frac{W^2(1+W^2+H^2)}{(1+W^2)(W^2+H^2)}\right]^{W^2}$ $\times\left[\frac{H^2(1+H^2+W^2)}{(1+H^2)(H^2+W^2)}\right]^{H^2}\}\Big)$

Table 20-5 Emittance Values of Common Materials

Material and Surface Condition	Total Hemispherical Emittance	Solar Absorptance
Aluminum		
Foil	0.05	0.15
Alloy, as received	0.04	0.37
Weather alloy	0.20	0.54
Asphalt (Roofing/Pavement)	0.88	
Brick	0.90	0.63
Concrete, rough	0.91	0.60
Copper		
Electroplated	0.03	0.47
Oxidized plate	0.76	
Frost, rime	0.99	
Glass (smooth)	0.91	
Gravel	0.30	
Ice (smooth)	0.97	
Iron		
Wrought, polished	0.29	
Wrought, dull	0.91	
Marble		
Polished	0.89	
Smooth	0.56	
Paints		
Black		
Flat	0.97	0.98
Gloss	0.90	
White		
Acrylic resin	0.90	0.26
Gloss	0.85	
Skin	0.95	
Soil	0.94	
Snow (fresh)	0.82	0.13
Stainless Steel		
Polished	0.60	0.37
Dull	0.2	
Vegetation	0.94	
Water	0.90	0.98
Wood (smooth)	0.84	

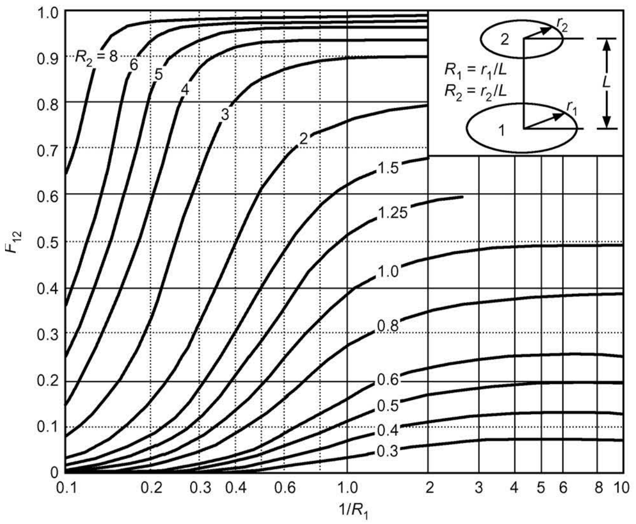
τ = fraction of incident radiation transmitted (transmissivity or transmittance)

For the many materials encountered in HVAC practice (other than fenestrations) that are opaque in the infrared region, $\tau = 0$, and thus $\rho = 1 - \alpha$.

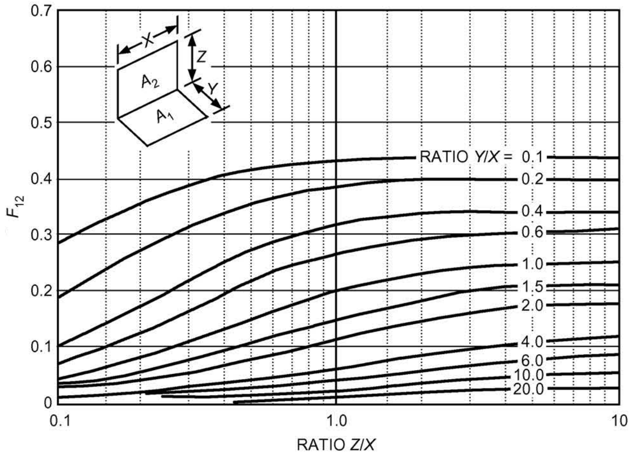
Actual Surfaces (Nonblack Bodies). Materials and surfaces of engineering interest show marked divergences from the Stephan-Boltzmann and Planck laws. Actual surfaces emit and absorb less readily and are called nonblack. The emittance (or emissivity) ε of the actual surface is defined as the ratio of the radiation emitted by the surface to the radiation emitted by a blackbody at the same temperature. Emissivity is a function of the material, the condition of its surface, and its temperature. In general, the emissivity of a surface may vary with wavelengths. To overcome this complexity, gray surface behavior (ε = constant over all wave- actually do approximate this condition, at least in some regions of the spectrum. However, one must be especially careful at high temperature. The emissive power of a nonblack surface, at temperature T, is given by

$$E = \varepsilon E_b = \varepsilon\sigma T^4 \qquad (20\text{-}11)$$

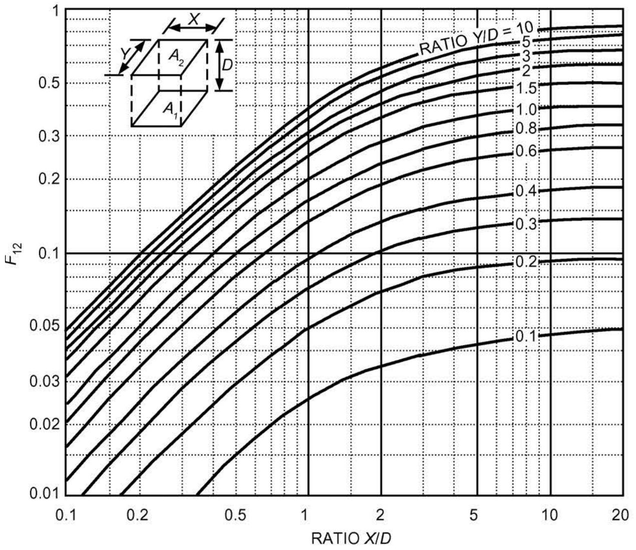
where ε is the total hemispherical emittance (or emissivity) and is a strong function of the condition and temperature of the actual surface. Table 20-5 provides approximate emittance values of some common materials and surface finishes at room and solar temperatures. In general, both ε and α of a surface depend on the temperature and the wavelength of the radiation. ***Kirchhoff's law*** of radiation states that the emittance and the absorptance of a surface at a given temperature and wavelength are equal. In many practical applications, the surface temperature and the temperature of the source of incident radiation (major exception – solar radiation) are of the same order of magnitude, and the average absorptance of a surface is taken to be equal to its average emittance ($\alpha = \varepsilon$).



COAXIAL DISKS



PERPENDICULAR RECTANGLES WITH COMMON EDGE



ALIGNED PARALLEL RECTANGLES

Fig. 20-4 View Factor Graphs for Common Geometries

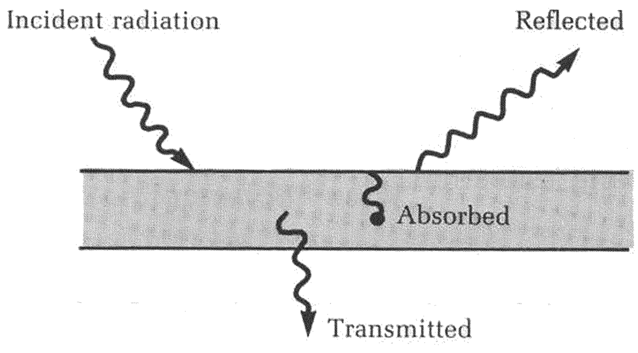


Fig. 20-3 Radiation Incident on Surface

20.1.4 Net Radiant Energy Loss from a Surface

When radiation heat transfer is involved in the energy balance on a surface, the net heat gain (or loss) by radiation from the surface is the quantity of interest. The general problem of determining the radiation exchange in an enclosure consisting of *n* surfaces, which may see one another by no others, requires the solution of n linear algebraic equations to account for the possibly infinite number of reflections of radiation from the participating surfaces. The current method of determining this quantity is called the ***radiosity method.*** It begins with the definitions of the two terms, radiosity and irradiation, and is applicable subject to the following conditions:

- Each surface is opaque, gray, isothermal, and uniformly irradiated.
- The emission and the reflections from each surface are diffuse.

20.1.5 Radiosity

The *radiosity J* is the total radiation leaving a surface, per unit area, per unit time. It includes both the emitted and the reflected amounts and can be expressed for surface *i* as

$$J_i = E_i + \rho G_i = \varepsilon_i \sigma T_i^4 + (1 - \varepsilon_i) G_i \qquad (20\text{-}12)$$

where G_i is the radiation falling upon surface *i*.

20.1.6 Irradiation

The *irradiation G* is the total radiation falling on a surface, per unit area, per unit time. The radiation incident on the *i*th surface is

$$A_i G_i = J_1 A_1 F_{1i} + J_2 A_2 F_{2i} + J_3 A_3 F_{3i} + \cdots$$

or

$$G_i = \sum_{j=1}^{n} J_j F_{ij}$$

(using the reciprocal rule).

The radiosity of surface *i* is $J_i = \varepsilon_i E_{bi} + (1 - G_i)$; substituting for G_i gives

$$J_i = \varepsilon_i E_{bi} + (1 - \varepsilon_i)\sum^{n} J_j F_{ik}; \qquad i = 1, 2, \ldots, n \tag{20-13}$$

a system of *n* linear equations in the *n* unknowns J_i. Upon solving the simultaneous equations and obtaining the values for the *J*s, the net radiant heat loss from each surface is obtained from

$$q_i = [J_i - G_i]A_i$$

$$q_i = A_i\left(J_i - \sum_{j=1}^{n} F_{ij}J_j\right) = \frac{E_{bi} - J_i}{(1 - \varepsilon_i)/E_i A_i} \tag{20-14}$$

For the special case when only two surfaces are involved, the net loss can be written as

$$\begin{aligned} q_i &= \frac{E_{b1} - E_{b2}}{\dfrac{1 - \varepsilon_1}{\varepsilon_1 A_1} + \dfrac{1}{F_{1\text{-}2}A_1} + \dfrac{1 - \varepsilon_2}{\varepsilon_2 A_2}} \\ &= \frac{\sigma(T_1^4 - T_2^4)}{\dfrac{1 - \varepsilon_1}{\varepsilon_1 A_1} + \dfrac{1}{F_{1\text{-}2}A_1} + \dfrac{1 - \varepsilon_2}{\varepsilon_2 A_2}} \end{aligned} \tag{20-15}$$

20.1.7 Radiation and Combined Heat Transfer Coefficients

A special case that occurs frequently involves radiation exchange between a small surface at T_s and a much larger, isothermal surface that completely surrounds the smaller one. The *surroundings* could, for example, be the walls of a room whose temperature T_{sur} differs from that of an enclosed surface $(T_{sur} \neq T_s)$. For such a condition, the *net* rate of radiation heat transfer *from* the surface, per unit area of the surface, is

$$q = \varepsilon A \sigma (T_s^4 - T_{\text{sur}}^4) \tag{20-16}$$

The T^4 dependence of radiant heat transfer complicates engineering calculations. When T_1 and T_2 are not too different, it is convenient to linearize Eq. (20-16) by factoring the term $(\sigma T_1^4 - \sigma T_2^4)$ to obtain

$$\begin{aligned} q_{12} &= \varepsilon_1 A_1 (T_1^2 + T_2^2)(T_1 + T_2)(T_1 - T_2) \\ &\cong \varepsilon_1 A_1 \sigma (4T_m^3)(T_1 - T_2) \end{aligned} \tag{20-17}$$

for $T_1 \cong T_2$, where T_m is the mean of T_1 and T_2. This result can be written more concisely as

$$q \cong A_1 h_r (T_1 - T_2)$$

where $h_r = 4\varepsilon_1 \sigma T_m^3$ is called the **radiation heat transfer coefficient**, in Btu/h·ft²·°F (W/m²·K).

Heat transfer from surfaces is usually a combination of convection and radiation. It is assumed that these modes are additive, and therefore a combined surface coefficient can be used to estimate the heat flow to/from a surface:

$$h_o = h_c + h_r$$

where

h_o = overall surface coefficient, Btu/h·ft²·°F (W/m²·K)
h_c = convection coefficient, Btu/h·ft²·°F (W/m²·K)
h_r = radiation coefficient, Btu/h·ft²·°F (W/m²·K)

Assuming the radiant environment is equal to the temperature of the ambient air, the heat loss/gain at the surface can be calculated as

$$q = h_o A (T_{\text{surf}} - T_{\text{amb}}) \tag{20-18}$$

Radiation is usually significant relative to conduction or natural convection, but negligible relative to forced convection. Thus, radiation in forced-convection applications is usually disregarded, especially when the surfaces involved have low emissivities and low to moderate temperatures.

20.2 Heat Exchangers

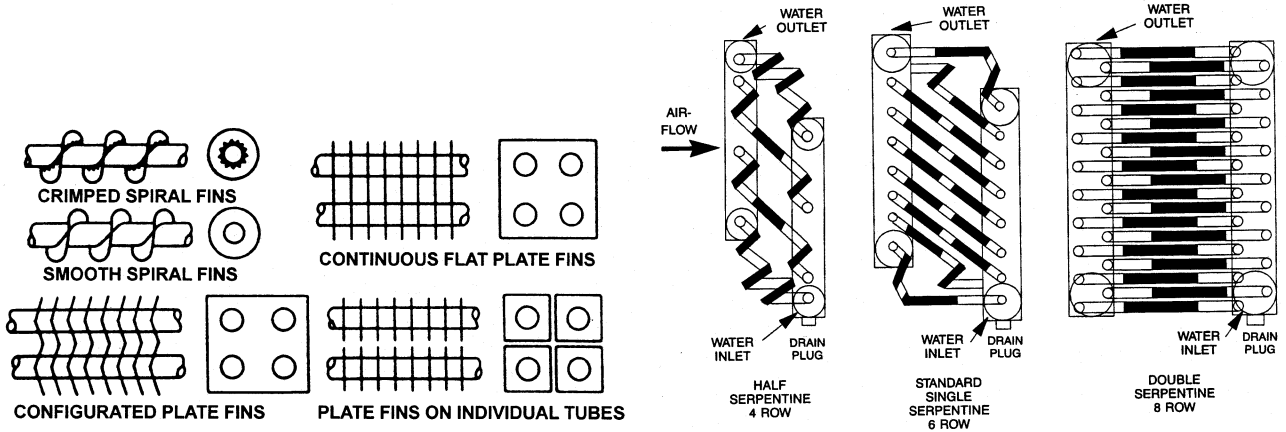
A heat exchanger is a device that permits the transfer of heat from a warm fluid to a cooler fluid through an intermediate surface without mixing of the two fluids. The correct sizing and selection of heat exchangers is probably the most important single factor in designing an efficient and economical building HVAC&R system. Whether the heat exchanger is selected as an off-the-shelf item or designed especially for the application, the following factors are normally considered:

- Thermal performance
- Cost
- Pressure drop
- Space requirements
- Serviceability.

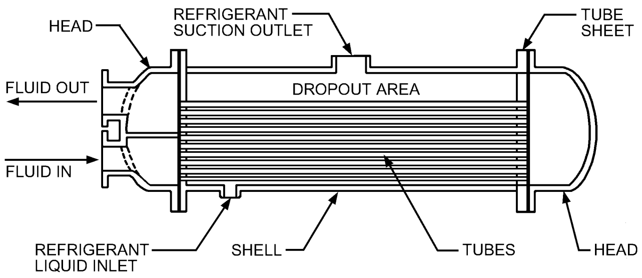
The main types of heat exchangers found in HVAC&R systems are the: finned-tube (coil), shell-and-tube, and plate. Sketches of each are given in Figure 20-5.

20.2.1 Plate Fin (Extended-Surface) Coils

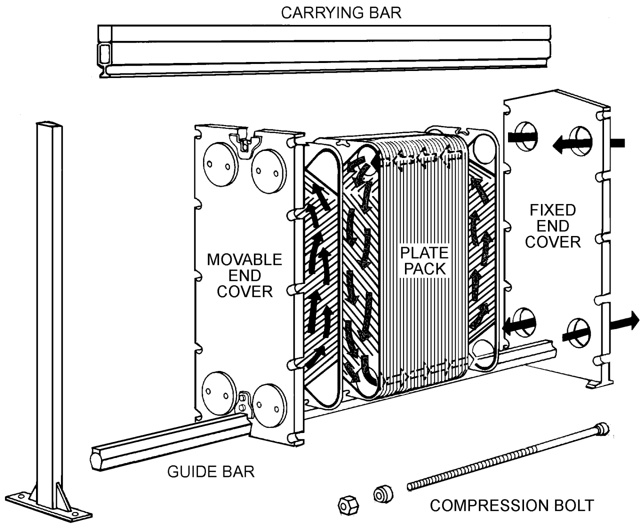
Most coils in HVAC systems consist of tubes with fins attached to their outer surface. Air flows over the outside of the tubes and refrigerant, steam, or water flows inside the tubes. The purpose of the fins is to increase the surface area on the air side where the convection coefficient is usually much lower than on the refrigerant, steam, or water side.



A. VARIOUS HEATING AND COOLING COILS (FINNED-TUBE HEAT EXCHANGERS)



B. SHELL-AND-TUBE HEAT EXCHANGER



C. PLATE HEAT EXCHANGER

Fig. 20-5 Sketches of the Three Types of Heat Exchangers: (A) Finned-Tube, (B) Shell-and-Tube, and (C) Plate (Part A: Figure 1, Chapter 23; Part B: Figure 2, Chapter 42; and Part C: Figure 15, Chapter 48 in 2016 ASHRAE Handbook—HVAC Systems and Equipment)

The *face area* of the coil is the cross sectional area of the air stream at the entrance of the coil and is obtained from the *length* (sometimes called the *width*) of the coil multiplied by the *height* of the coil. The *face velocity* of the air is the volume flow rate of the air divided by the face area. The *surface area* of the coil is the heat transfer surface area in contact with the air. The *number of rows* of tubes and the *depth* of the coil are measured in the direction of the airflow.

In comparison to bare tube coils of the same capacity, finned coils are much more compact, less weight, and usually less expensive. The secondary surface area of a finned coil may be 10 to 40 times that of the bare tubes. The primary surface is that of the tubes or pipes. The secondary surface (fins) consists of thin metal plates or a spiral ribbon uniformly spaced or wound along the length of the primary surface and in intimate contact with it. The bond between fin and tube is a significant parameter in the thermal performance of the coil. The bonding is usually accomplished by expanding the tubes (often copper) into the tube holes in the plate fins (often sheets of aluminum). The tube holes are often punched with a formed fin collar which both provides contact area as well as a means of spacing the fins uniformly along the length of the tubes. Figure 20.5(a) illustrates several finned-tube heat exchangers (coils).

More information of cooling coils can be found in Section 17.2 of this book while more on heating coils appears in Section 17.3.

20.2.2 Shell-and-Tube Heat Exchangers

The most common type of heat exchanger in industrial applications is probably the shell-and-tube heat exchanger. This type also finds extensive use in HVAC&R applications involving water cooled units and water and brine chillers. Figure 20.5(b) provides a sketch of the shell-and-tube heat exchanger.

Shell-and-tube heat exchangers can handle from a single tube to a large number of tubes packed in a shell with their axes parallel to that of the shell. Heat transfer takes place as one fluid flows inside the tubes while the other fluid moves outside the tubes through the shell. Baffles are usually placed in the shell to force the shell-side fluid to maintain uniform and good contact with the outside of the tubes. Shell-and-tube heat exchangers are generally classified according to the number of tube and shell passes involved.

20.2.3 Plate Heat Exchangers

The plate-and-frame (or just plate) type of heat exchanger continues to find additional use in HVAC applications, including the water side economizer. Plate heat exchangers consist of metal plate pairs arranged to provide separate flow paths (channels) for the two fluids. Heat transfer occurs across the plate walls. The hot and cold fluids flow in alternate passages, and thus each cold fluid stream is surrounded by two hot fluid streams, resulting in very effective heat transfer. Figure 20.5(c) provides a pictorial view of the plate exchanger and illustrates the flow paths.

The exchangers have multiple channels in series that are mounted on a frame and clamped or welded together. The rectangular plates have an opening or port at each corner. When assembled, the plates are sealed such that the ports provide manifolds to distribute the fluids through the separate flow paths. The clamped type of plate heat exchanger can be easily enlarged to meet higher heat transfer rates by simply mounting more plates. Plate exchangers are particularly well suited for liquid-to-liquid heat exchange applications, but also find use as condensers and evaporators.

20.3 Basic Heat Exchanger Design Equation

By applying the fundamentals of heat transfer to heat exchangers whose purpose it is to transfer heat from one fluid to another, the overall heat flow from one fluid across a barrier to a second fluid is often expressed as:

$$q = UA\,\Delta T \tag{20-19}$$

where

q = rate of heat transfer, Btu/h or kW

A = surface area of material separating the two fluids, ft^2 or m^2

U = overall coefficient of heat transfer, $\text{Btu/h·ft}^2\text{·°F}$ or $\text{W/m}^2\text{·K}$

ΔT = *mean* temperature difference between the hot and cold fluids, °F or K

Rearranging this equation yields the basic design equation for a heat exchanger as

$$A = \frac{q}{U\,\Delta T_m} \tag{20-20}$$

where A is the total heat transfer area required in the exchanger. Thus, it will now be necessary to first calculate (estimate) q, U, and ΔT, as discussed in the following sections.

20.4 Estimation of Heat Load

The usual first step in designing a heat exchanger is to use the First Law of Thermodynamics to make an energy balance for (a) estimating the heat load (duty) of the heat exchanger, and probably (b) the required flow rate of one of the fluids. Since the heat exchanger is normally assumed to be overall adiabatic (only energy exchange takes place within the heat exchanger, from the hot fluid to the cold fluid), the heat load is calculated in the general case from

$$q = m_h(h_{h,\,\text{in}} - h_{h,\,\text{out}}) = m_c(h_{c,\,\text{out}} - h_{c,\,\text{in}}) \tag{20-21}$$

where m_h and m_c are the mass flow rates of the hot and cold fluids and $h_{h,in}$, $h_{h,out}$, $h_{c,in}$, and $h_{c,out}$ are the respective enthalpies. When there is no change in phase, the enthalpy change can be replaced as follows:

$$\begin{aligned} h_{\text{out}} - h_{\text{in}} &= c_p(T_{\text{out}} - T_{\text{in}}) \\ \text{or}\quad h_{\text{in}} - h_{\text{out}} &= c_p(T_{\text{in}} - T_{\text{out}}) \end{aligned} \tag{20-22}$$

where c_p is the specific heat of the particular fluid.

Upon specifying the function of the particular heat exchanger(e.g., cool a known amount of hot fluid from one temperature to another), Eq. (20-20) can be used to calculate the required heat transfer rate from the hot fluid (also the rate to the cold fluid) as well as to determine either the amount of the other fluid (if both inlet and outlet temperatures are known) or determine its outlet temperature (if its flow rate and inlet temperature are known).

20.5 Mean Temperature Difference

The temperature of the fluids flowing through a heat exchanger generally varies from location to location as heat is transferred from the hotter to the colder fluid. There is no single temperature difference serving as the driving force for the heat transfer and thus the rate of heat transfer with this varying temperature difference must be obtained by integrating

$$dq = U\,dA\,\Delta T$$

over the heat transfer area A along the length of the heat exchanger. The result, for either concurrent (parallel) or countercurrent (counterflow) flow conditions yields

$$q = UA(\Delta T_a - \Delta T_b)/\ln(\Delta T_a/\Delta T_b)$$

where a and b refer to the two ends of the heat exchanger.

The concept of an appropriate mean temperature difference, a single temperature difference which results in the same heat flow value, is useful and widely used in engineering practice:

$$q = UA\,\Delta T_{\text{mean}}$$

For parallel or counterflow conditions, this mean temperature difference is therefore

$$\Delta T_{\text{mean}} = (\Delta T_a - \Delta T_b)/\ln(\Delta T_a/\Delta T_b) \qquad (20\text{-}23)$$

and is called the log mean temperature difference or LMTD.

For stream conditions other than the ideal counterflow (such as the common cross flow), a correction factor F is applied to the LMTD obtained as if the flow had been pure counterflow. Examples of these corrections factors are provided in Figure 20-6 (Bowman et al. 1940).

The resulting equation for the heat transfer rate becomes

$$q = UAF\,\Delta T_{m,\,cf} \qquad (20\text{-}24)$$

For cases where at least one fluid temperature remains constant (e.g., evaporation or condensation), the correction factor is unity regardless of the flow pattern.

20.6 Estimation of the Overall Heat Transfer Coefficient *U*

The governing equations for the design of heat exchangers are as follows:

$$Q = U_o A_o F\,\Delta T_{m,\,cf} = U_i A_i F\,\Delta T_{m,\,cf} \qquad (20\text{-}24a)$$

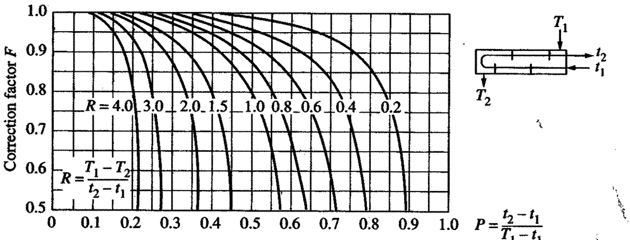
where

Q = amount of heat transfer for the coil to do
U = overall heat transfer coefficient
F = temperature difference correction
ΔT = log mean temperature difference
A_i, A_o = internal or outside areas

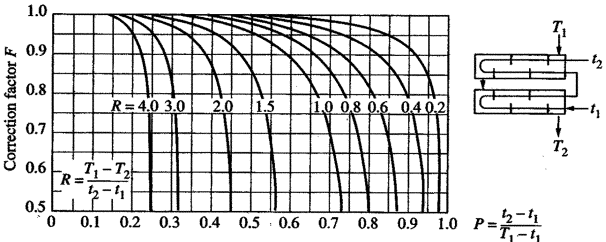
$$U = 1/\Sigma R \qquad (20\text{-}24b)$$

where the Rs are resistances to the flow defined by

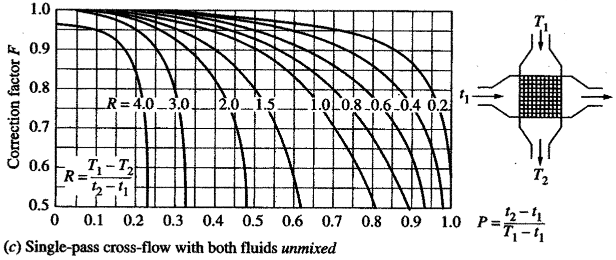
A/H_iA_i = internal convective resistance
AR_{fi}/A_i = internal fouling in the pipes
$A\ln(d_o/d_i)/(2\pi kl)$ = conductive resistance through wall of piping
AR_{fo}/A_o = outside fouling resistance



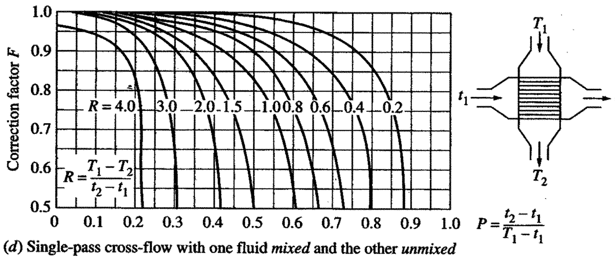
(a) One-shell pass and 2, 4, 6, etc. (any multiple of 2), tube passes



(b) Two-shell passes and 4, 8, 12, etc. (any multiple of 4), tube passes



(c) Single-pass cross-flow with both fluids *unmixed*



(d) Single-pass cross-flow with one fluid *mixed* and the other *unmixed*

Fig. 20-6 LMTD Correction Factors for Several Flow Configurations
(Bowman et al. 1940)

AR_c/A_o = contact resistance between tube and fin
$A/[h_o(A_{\text{unfin}} + \phi A_{fin})]$ = outside convective resistance

$$\Delta T_{m,\,cf} = (\Delta T_a - \Delta T_b)/\ln(\Delta T_a - \Delta T_b) \qquad (20\text{-}24c)$$

These equations become

$$Q = \frac{F\Delta T_{m,\,cf}}{\dfrac{1}{h_iA_i} + \dfrac{R_{fi}}{A_i} + \dfrac{\ln(d_o/d_i)}{2\pi kL} + \dfrac{R_{fo}}{A_o} + \dfrac{R_c}{A_o} + \dfrac{1}{h_o(A_u + \phi A_f)}} \qquad (20\text{-}25)$$

A complicated heat transfer phenomenon is considerably simplified by the assumption of boundary layers or

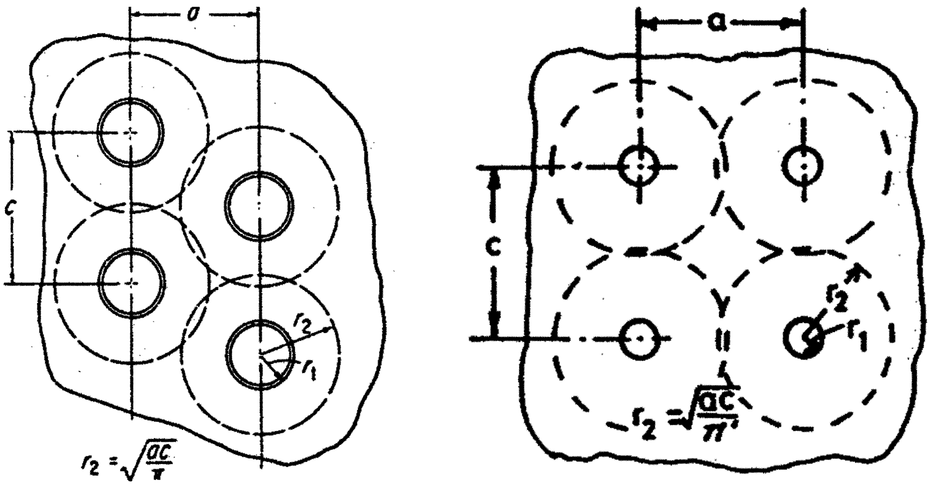


Fig. 20-7 Approximate Method for Obtaining Efficiency of Common Flat Plate Fin

films between the barrier wall and the fluids that offer resistance to heat flow. This mechanism is represented by the following equation, which assumes a constant overall heat transfer coefficient in the entire heat exchanger. This is not unreasonable when the temperature change in each fluid is small and therefore there is little change of physical properties between the inlet and the outlet.

$$U = \frac{1}{\frac{1}{h_o} + \frac{1}{R_{fo}} + \frac{d_w}{k_w}\left(\frac{A_o}{A_{\text{mean}}}\right) + \frac{1}{R_{fi}}\left(\frac{A_o}{A_i}\right) + \frac{1}{h_i}\left(\frac{A_o}{A_i}\right)} \qquad \text{(20-25a)}$$

where

U = overall heat transfer coefficient, Btu/h·ft^2·°F (W/m^2·K)

h_o = film coefficient of fluid outside tube, Btu/h·ft^2·°F (W/m^2·K)

h_i = film coefficient of fluid inside tube, Btu/h·ft^2·°F (W/m^2·K)

R_{fo} = fouling coefficient outside of tube, Btu/h·ft^2·°F (W/m^2·K)

R_{fi} = fouling coefficient inside of tube, Btu/h·ft^2·°F (W/m^2·K)

d_w = thickness of tube wall, ft (m)

k_w = thermal conductivity of tube, Btu·ft/h·ft^2·°F (W/m·K)

A_o/A_i = ratio of outside to inside tube surface

A_{mean} = average tube area per unit length, ft^2/ft (m^2/m)

A_o = outside tube area per unit length, ft^2/ft (m^2/m)

The accuracy of this relationship is limited by the reliability of the correlations for calculating the individual film coefficients, and by the arbitrary selection of fouling coefficients.

20.7 Extended Surfaces, Fin Efficiency, and Fin-Tube Contact Resistance

When the tube is finned on the air side to enhance heat transfer, the total heat transfer surface on the finned side becomes

$$A_s = A_{\text{total}} = A_{\text{fin}} + A_{\text{unfinned}}$$

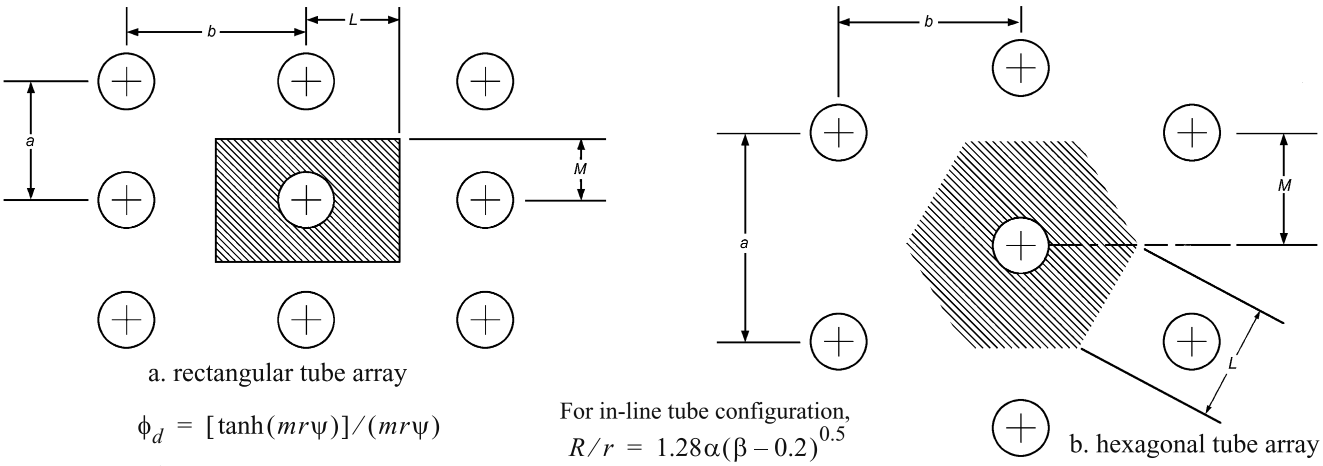
where A_{fin} is the surface area of the fins and $A_{unfinned}$ is the area of the unfinned portion of the tube surface. However, when determining the heat transfer rate using the overall coefficient U, the appropriate total surface area to use is the effective area which means de-rating the area of the fins due to the internal thermal resistance within the fins. Thus, the appropriate surface area is

$$A = A_{\text{unfinned}} + \phi A_{\text{fin}} \qquad \text{(20-26)}$$

The rectangular-plate fin of uniform thickness is commonly used in finned coils for heating or cooling air. It is not possible to obtain an exact mathematical solution for the efficiency of such a fin. It can be shown that an adequate approximation is to assume that the fin area served by each tube is equivalent in performance to a flat circular-plate fin of equal area. Figure 20-7 shows the method for determining the equivalent outer radius for this method. The corresponding efficiency for the flat plate fin can then be obtained from Figure 20-8.

As detailed in Chapter 4 of the 2017 *ASHRAE Handbook—Fundamentals*, the approximate fin efficiencies can also be calculated as provided in Figure 20-9.

The most common means of bonding the fins to the tubes on common heating and cooling coils is by mechanical expansion. Results of Sheffield et al. (1985) have



a. rectangular tube array

b. hexagonal tube array

$$\phi_d = [\tanh(mr\psi)]/(mr\psi)$$

where

$m = [2h_{od}/(k_f y_f)]^{0.5}$

$\psi = (R/r - 1)[1 + 0.35\ln(R/r)]$

For in-line tube configuration,

$$R/r = 1.28\alpha(\beta - 0.2)^{0.5}$$

For triangular tube configuration,

$$R/r = 1.27\alpha(\beta - 0.3)^{0.5}$$

$\beta = L/M$ and $\alpha = M/r$

Fig. 20-8 Approximate Equations for Plate Fin Efficiency

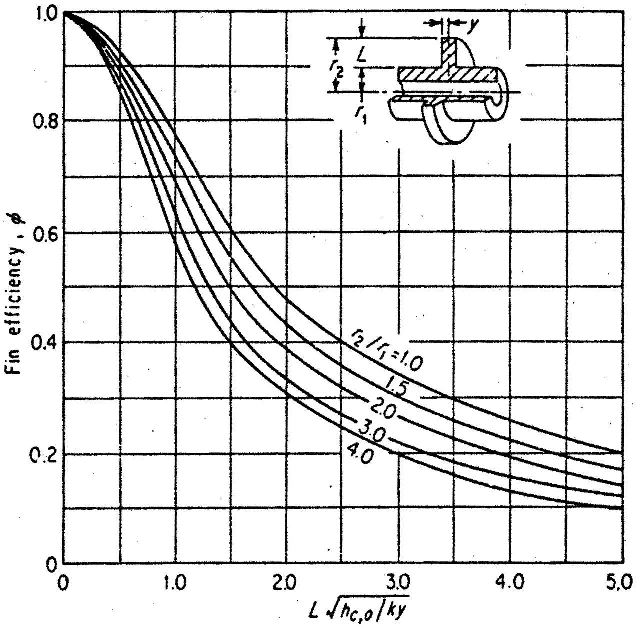


Fig. 20-9 Efficiency of Circular-Plate Fin of Uniform Thickness
(Source: Gardner 1945)

shown that, for properly expanded tubes, there is a relatively narrow range of values of thermal resistance due to the fact that the tubes and fins are only in contact over a relative small area due to surface asperities, non-roundness, and other factors. As reported by Sheffield et al., a reasonable value for the conductance between mechanically expanded copper tubes and aluminum fins is 3750 Btu/h·ft^2·°F (21 293 W/m^2·K). The thermal contact resistance (TCR) is the inverse, or 0.000267 h·ft^2·°F/Btu (0.000047 m^2·K/W).

Table 20-6 Fouling Factors for Water, h·ft^2·°F/Btu (m^2·K/W)

	Water Velocity, ft/s (m/s)	
Type of Water	**3 (1) and less IP (SI)**	**Over 3 (1) IP (SI)**
Cooling tower and spray pond		
Treated makeup	0.001 (0.00018)	0.001 (0.00018)
Untreated	0.003 (0.0005)	0.003 (0.0005)
River water (average)	0.002 (0.00035)	0.001 (0.00018)
Hard	0.003 (0.0005)	0.002 (0.00035)
Distilled or closed cycle condensate	0.001 (0.00018)	0.001 (0.00018)

20.8 Fouling Factors

After a period of operation, the heat transfer surfaces of a heat exchanger may become coated with various deposits from the fluids or may become corroded as a result of interaction between the fluids and the surface material. This coating represents an additional resistance to the flow of heat and results in decreased heat transfer performance. The effect is accounted for by a fouling factor, or fouling resistance, R_f, which is then to be added to the other resistances in the thermal path between the two fluids.

The most common type of fouling is the precipitation of solids from the fluid onto the heat transfer surface. Other types of fouling include chemical fouling (e.g., corrosion) and biological fouling from algae growth.

Table 20-7 Fouling Factors for Various Fluids

	Fouling Factor	
Type of Fluid	**h·ft²·F/Btu**	**m²·K/W**
Gases and vapors		
Steam (non-oil bearing)	0.0005	0.00009
Refrigerant vapors (oil bearing)	0.002	0.00035
Refrigerant vapors (pure)	0	0
Compressed air	0.002	0.00035
Industrial organic heat transfer media	0.001	0.00018
Liquids		
Refrigerant liquids	0.001	0.00018
Industrial organic heat transfer media	0.001	0.00018

The fouling factor depends upon the tube material, the nature of the fluid, and the fluid velocity. Fluid velocities less than about 3 ft/s (0.9 m/s) tend toward excess fouling. A few example values of fouling factors are provided in Table 20-6 and Table 20-7. Considerable uncertainty exists in these values, and they should be used cautiously. More comprehensive tables are available from TEMA (Tubular Exchanger Manufacturers Association) (tema.org).

There is little published data on the rate of fouling for heat exchangers in typical air conditioning and refrigeration service. For many years, the basic reference has been the TEMA Standard. The air conditioning industry has for decades commonly used an assumed fouling level of 0.0005 h·ft²·°F/Btu (0.00009 m²·K/W) in both condensers and coolers. Occasionally, where a condenser was to use river water, engineers would specify as much as 0.0020 h·ft²·°F/Btu (000035 m²·K/W) fouling. Based on more recent studies by the Air-Conditioning, Heating, and Refrigeration Institute (AHRI, formerly ARI), it appears reasonable to specify a fouling factor of 0.00025 h·ft²·°F/Btu (0.000044 m²·K/W) for

- closed-loop liquid chillers
- condensers served by well-maintained cooling towers

20.9 Convective Heat Transfer Coefficients h_i and h_o

The determination of accurate values for the inside and outside convective heat transfer coefficients is critical to the accurate evaluation of heat exchanger performance. Unfortunately, even today, the available correlations for predicting these coefficients often leave much to be desired, particularly if a phase change is occurring.

The correlations presented in the following subsections are included herein primarily as (a) examples, and (b) to provide sample working relations for use with both the example heat exchanger design problems and the homework problems at the end of the chapter. The reader is referred to the current technical literature in heat transfer (e.g., *International Journal of Heat and Mass Transfer*, *Journal of Heat Transfer, International Journal of HVAC & Refrigeration Research*) for improved correlations.

20.9.1 Single-Phase Internal Flow in Tubes

The simplified relation of McAdams (1940) is widely used for turbulent single-phase flow in tubes and pipes”

$$\mathrm{Nu} = 0.023(\mathrm{Re})^{0.8}(\mathrm{Pr})^{1/3} \tag{20-27}$$

The relation is relatively simple, but gives maximum errors of ±25% in the range of $0.67 < \mathrm{Pr} < 100$. A more accurate correlation, which is also applicable for rough ducts, has been developed by Petukhov and coworkers at the Moscow Institute for High Temperature:

$$\mathrm{Nu} = (\mathrm{Re}\,\mathrm{Pr}/X)(f/8)(\mu_b/\mu_w)^n \tag{20-28}$$

where

$$X = 1.07 + 12.7(\mathrm{Pr}^{2/3} - 1)(f/8)^{1/2}$$

and

$$n = \begin{cases} 0.11 & \text{heating } (T_w > T_b) \\ 0.25 & \text{cooling } (T_w < T_b) \\ 0 & \text{gases} \end{cases}$$

This correlation is applicable for fully developed turbulent flow in the range

$$10^4 < \mathrm{Re} < 5 \times 10^6$$
$$2 < \mathrm{Pr} < 140 \quad \text{with 5 to 6\% error}$$
$$0.5 < \mathrm{Pr} < 2000 \quad \text{with 10\% error}$$
$$0.08 < \mu_w/\mu_b < 40$$

All properties, except μ_w, are evaluated at the bulk temperature. For smooth tubes and pipes, the friction factor is evaluated by

$$f = (1.82 \log \mathrm{Re} - 1.64)^{-2} \tag{20-29}$$

20.9.2 Forced Convection Boiling in Tubes

Correlations for forced convection have been developed for boiling refrigerants in horizontal tubes. All are restricted to test conditions for particular refrigerants, and one should be careful in applying them to conditions outside the test range.

Bo Pierre introduced the load factor, $K_f = J\,\Delta x\,h_{fg}/L$, which effectively combines the Boiling and Martinelli numbers. In the load factor expression, J is joules equivalent of heat (778 ft·lb$_f$/Btu [1 J/J]) and Δx is the change in quality that occurred during the evaporation process. Bo Pierre correlated R-12 and R-22 for a wide range of operating conditions with separate correlations for complete and incomplete evaporation. These correlations for the

Nusselt number (Nu) with two-phase (*tp*) flow are as follows:

$$\mathrm{Nu}_{tp} = 0.009(\mathrm{Re}^2 K_f)^{0.5}$$

$$\text{for } 10^9 < \mathrm{Re}^2 K_f < 0.7 \times 10^{12} \qquad (20\text{-}30)$$

and exit vapor quality < 90% (incomplete evaporation)

$$\mathrm{Nu}_{tp} = 0.0082(\mathrm{Re}^2 K_f)^{0.5}$$

$$\text{for } 10^9 < \mathrm{Re}^2 K_f < 0.7 \times 10^{12} \qquad (20\text{-}31)$$

and up to 11°F (6.1°C) superheat (complete evaporation)

20.9.3 Forced Convection Condensation in Tubes

Condensers used for refrigeration and air-conditioning systems often involve vapor condensation inside horizontal tubes. Unfortunately, conditions within the tube are complicated and depend strongly on the velocity of the vapor flowing through the tube.

If this velocity is small, the condensate flow is from the upper portion of the tube to the bottom, from whence it flows in a longitudinal direction with the vapor. For low vapor velocities such that

$$\mathrm{Re}_{v,i} = \left(\frac{\rho_v m_{m,v} D}{\mu_v}\right)_i < 35{,}000$$

where i refers to the tube inlet. An expression of the form

$$\bar{h}_D = 0.555\left[\frac{g\rho_l(\rho_l - \rho_v)k_l^3 h'_{fg}}{\mu_l(T_{\mathrm{sat}} - T_s)D}\right]^{1/4} \qquad (20\text{-}32)$$

is recommended where, for this case, the modified latent heat is

$$h'_{fg} \equiv h_{fg} + \frac{3}{8}c_{p,l}(T_{\mathrm{sat}} - T_s) \qquad (20\text{-}33)$$

At higher vapor velocities, the two-phase flow regime becomes annular, and the following correlation is preferred.

$$hD/k_l = 0.026\,\mathrm{Pr}_l^{1/3}[\mathrm{Re}_l + \mathrm{Re}_v(p_l/p_s)^{1/2}]^{0.8} \qquad (20\text{-}34)$$

where

$$\mathrm{Re}_l = (4M_l/\pi D\mu_l) \qquad \mathrm{Re}_v = (4M_v/\pi D\mu_v)$$

Here, M_l and M_v are, respectively, the mass flow rates of liquid and vapor. This expression is valid for Re > 20,000.

20.9.4 Condensation on Horizontal Tubes

One of the earliest investigations into laminar film condensation on horizontal tubes was carried out by Nusselt (1916). By applying a force and energy balance to the condensate film, Nusselt arrived at the following equation for condensation from a single horizontal tube:

$$h = 0.729[g\rho_l(\rho_l - \rho_v)k_l^3 h_{fg}/D_o\mu_l(T_{\mathrm{sat}} - T_w)]^{1/4} \qquad (20\text{-}35)$$

For a vertical tier of N horizontal tubes, the average convection coefficient (over the N tubes) may be expressed as

$$h = 0.729\left[\frac{g\rho_l(\rho_l - \rho_v)k_l^3 h_{fg}}{N\mu_l(T_{\mathrm{sat}} - T_s)D}\right]$$

That is, $h_N = hN^{-1/4}$, where h is the heat transfer for the first (upper) tube. Such an arrangement is often used in condenser design. The reduction in h with increasing N is due to an increase in the film thickness for each successive tube.

20.9.5 Boiling from Horizontal Tubes

The correlations for heat transfer under fully developed nucleate boiling conditions have been divided into two main groups: those based upon direct curve fitting of experimental data banks, called strictly empirical, and those based upon a physical model, but ultimately curve fitted by experimental results, called semi-empirical. The procedure is based on a straightforward reasoning, according to which, nucleate boiling heat transfer correlations, even those of the second group, can be reduced to a product of powers of the transport properties. These properties can be written in terms of reduced primary thermodynamic properties, such as pressure and temperature p_r and T_r, as in the Law of Corresponding States. Thus, in principle, all the heat transfer correlations could be reduced to a product of powers of p_r and T_r, presenting a single and common form depending on numerical coefficient and exponents that can be obtained by fitting experimental data.

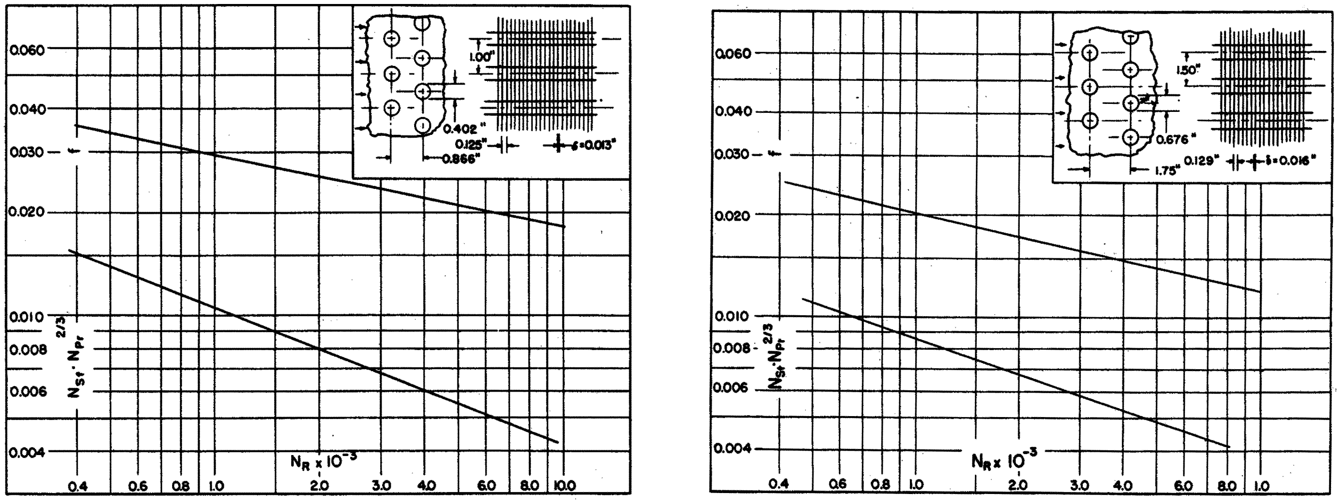
This correlation uses molecular weight, reduced pressure and surface roughness as the correlation parameters and can be written as

$$h = 95 \cdot q^{0.67} M^{-0.5}\left(\frac{p}{p_c}\right)^{0.12 - 0.21\log_{10}R_p} \times \left(-\log_{10}\frac{p}{p_c}\right)^{-0.55} \qquad (20\text{-}36)$$

where M is the molecular weight and R_p is the roughness of the surface.

It is generally assumed that commercial-finish copper tubes have a surface roughness of 0.4 μm.

Rohsenow's Correlation. The first and still most widely used correlation for heat transfer in nucleate pool boiling was proposed by Rohsenow (1973) using experimental data on pool boiling from many different fluids as a guide, Rohsenow obtained



(A) SURFACE 8.0-3/8*T*

(B) SURFACE 7.75-5/8*t*

Fig. 20-10 Dimensional Data for Two Finned-Tube Surfaces

Table 20-8 Values of C_{sf} for Various Fluid-Surface Combinations

[Incropera (2007)]

Fluid-Surface Combination	C_{sf}	*n*
Water-copper		
Scored	0.0068	1.0
Polished	0.0130	1.0
Water-stainless steel		
Chemically etched	0.0130	1.0
Mechanically polished	0.0130	1.0
Ground and polished	0.0060	1.0
Water-brass	0.0060	1.0
Water-nickel	0.006	1.0
Water-platinum	0.0130	1.0
n-Pentane-copper		
Polished	0.0154	1.7
Lapped	0.0049	1.7
Benzene-chromium	0.0101	1.7
Ethyl alcohol-chromium	0.0027	1.7

$$h = \mu_l h_{fg}\left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2}\left(\frac{c_{p,l}}{C_{sf}\, h_{fg} \mathrm{Pr}_l^n}\right)^3 \Delta T_x^2 \tag{20-37}$$

where

μ_l = viscosity of the liquid, lb/ft^2·s (kg/m·s)
h_{fg} = enthalpy of vaporization, Btu/lb (J/kg)
g = gravitational acceleration, ft^2/s (m^2/s)
ρ_l = density of liquid, lb/ft^3 (kg/m^3)
ρ_v = density of vapor, lb/ft^3 (kg/m^3)
σ = surface tension of liquid-vapor interface, lb$_f$/ft (N/m)
c_{pl} = specific heat of the liquid, Btu/lb·°F (J/kg·°C)
ΔT_x = $\Delta T_x = T_s - T_{sat}$
T_s = surface temperature, °F (°C)
T_{sat} = saturation temperature of the fluid, °F (°C)
C_{sf} = experimental constant that depends on surface-fluid combination
Pr_l = Prandtl number of the liquid
n = experimental constant that depends on the fluid = 1.0 for water, 1.7 for other fluids

The Rohsenow method correlates data for all types of nucleate-boiling processes, including pool boiling of saturated or subcooled liquids. Unfortunately, Rohsenow's Correlation can be used **only if** the C_{sf} value is known and the common values are given in Table 20-8. Fortunately, values for several common refrigerants have recently been reported, as shown in Table 20-9.

20.9.6 Airflow Across Finned Tubes

The outside heat transfer coefficient is a very crucial parameter which has to be estimated accurately in the design and performance simulation of finned-tube heat exchangers. It is often the controlling factor in the estimation of the overall heat transfer coefficient for the exchanger when air is the external heat transfer fluid. Most correlations have been obtained experimentally and are (a) valid only for specific surfaces and (b) proprietary for most configured fin surfaces.

Table 20-12 provides dimensional data for two plate fin-and-tube arrangements consisting of aluminum fins bonded to copper tubes (Schedule 18). Figure 20-10 gives a schematic of each surface and presents the corresponding heat transfer correlation for the external surface. The friction factor for the outside surface is also shown. The principal dimensionless groups governing these correlations are the Stanton, Prandtl, and Reynolds numbers:

$$\mathrm{St} = h/Gc_p \qquad \mathrm{Pr} = c_p\,\mu/k \qquad \mathrm{Re} = GD_h/\mu$$

where G is the mass velocity defined as

$$G = m/A_{\min}$$

Table 20-9 C_{sf} s for Refrigerants

Refrigerant	Value of C_{sf}
R-12	0.008339
R-22	0.007947
R-123	0.006706
R-134a	0.006232
R-407C	0.007269
R-410A	0.008294

where m = total mass flow rate of the fluid and A_{min} = minimum free-flow cross-sectional area in the coil. The hydraulic diameter ($D_h = 4r_h$) is specified on each figure.

20.10 Calculation of Heat Exchanger Surface Area and Overall Size

Design methods for shell-and-tube and finned-tube heat exchangers are outlined in Tables 20-10 and 20-11, respectively.

20.11 Fluids and Their Thermophysical Properties

Proper evaluation of the necessary thermodynamic and thermophysical properties of the working fluids in heat exchangers is most important. Properties need to be evaluated at the correct temperature (and pressure, if gas or vapor) which may be the average bulk fluid temperature (average between in and out), the saturation temperature, the "film" temperature (average between fluid and wall), and/or the surface temperature. Unfortunately, there is no single "correct" temperature to use but varies primarily with the correlation selected for predicting the convective heat transfer coefficient.

Unfortunately, very limited property data is included here due to space limitations (see Table 2.2 for R-134a table). However, Chapters 29, 30, and 33 of the 2017 *ASHRAE Handbook—Fundamentals* are an excellent source of such data. In addition, most heat transfer textbooks include some tables, and there are rather extensive reference handbooks either on heat transfer or thermophysical properties. And, of course, the Web is another valuable source of such data.

20.12 Example Finned-Tube Heat Exchanger Design

Task: An aluminum tube with k = 1290 Btu·in/h·ft^2·°F, ID = 1.8 in., and OD = 2 in. has circular aluminum fins $\delta = 0.04$ in. thick with an outer diameter of D_{fin} = 3.9 in. There are N' = 76 fins per foot of tube length. Steam condenses inside the tube at t_i = 392°F with a large heat transfer coefficient on the inner tube surface. Air at t_∞ = 77°F is heated by the steam. The heat transfer coefficient outside the tube is 7 Btu/h·ft^2·°F. Find the rate of heat transfer per foot of tube length.

Solution: From Figure 20-8 efficiency curve, the efficiency of these circular fins is

$$\left.\begin{aligned} L &= (D_{fin} - \text{OD})/2 = (3.9 - 2)/2 = 0.95 \text{ in.} \\ f_2/f_1 &= \frac{3.9/2}{2/2} = 1.95 \text{ in.} \\ L\sqrt{\frac{h}{ky}} &= 0.95 \text{ in.}\sqrt{\frac{7 \text{ Btu/h·ft}^2\text{·°F}}{(1290 \text{ Btu·in/h·ft}^2\text{·°F})(0.02 \text{ in.})}} \end{aligned}\right\}\phi = 0.89$$

The fin area for L = 1 ft is

$$A_s = N'L \times 2\pi(D_{\text{fin}}^2 - \text{OD}^2)/4 = 1338 \text{ in}^2 = 9.29 \text{ ft}^2$$

The unfinned area for L = 1 ft is

$$\begin{aligned} A_p &= \pi \times \text{OD} \times L(1 - N'\delta) \\ &= \pi(2/12) \text{ ft} \times 1 \text{ ft}(1 - 76 \times 0.04/12) \\ &= 0.39 \text{ ft}^2 \end{aligned}$$

and the total area $A = A_s + A_p = 9.68$ ft^2. Surface efficiency is

$$\phi_s = \frac{\phi A_f + A_s}{A} = 0.894$$

and resistance of the finned surface is

$$R_s = \frac{1}{\phi_s hA} = 0.0165 \text{ h·°F/Btu}$$

Tube wall resistance is

$$\begin{aligned} R_{\text{wall}} &= \frac{\ln(\text{OD/ID})}{2\pi Lk_{\text{tube}}} = \frac{\ln(2/1.8)}{2\pi(1 \text{ ft})(1290/12) \text{ Btu·in/h·ft·°F}} \\ &= 1.56 \times 10^{-4} \text{ h·°F/Btu} \end{aligned}$$

The rate of heat transfer is then

$$q = \frac{t_i - t_\infty}{R_s + R_{\text{wall}}} = 18{,}912 \text{ Btu/h}$$

20.13 Problems

20.1 A hot-water coil is to be sized (designed) to heat 5000 cfm of air from 70°F to 130°F with the coil face velocity based on duct space selected as 500 fpm. The inlet and outlet water temperatures are 190°F and 170°F, respectively. Determine: coil height, width, number of rows, depth.

20.2 A supplementary cooling coil is being added to the building's HVAC system to provide additional cooling for a computer room. The cooling coil will do sensible cooling only, taking 1500 scfm of preconditioned air from 70°F db and 60°F wb to 55°F db. A 50/50 antifreeze (ethylene glycol) solution at 50°F will be supplied to the coil. Prepare a preliminary design for the coil using Surface 8.0-3/8T with copper tubes having a final wall thickness of 0.036 in.

Table 20-10 Design Procedure for Shell-and-Tube Heat Exchanger

1. FIND Q OF THE CONDENSER
2. CHOOSE TUBE SIZE
3. CALCULATE MEAN TEMPERATURE
4. CALCULATE REYNOLDS NUMBER
5. USING Q=M*CP*DELT CALCULATE MASS FLOW RATE FOR WATER
6. USING M=RHO*AREA*VELOCITY*N TUBES CALCULATE NUMBER OF TUBES
7. CALCULATE LOG MEAN TEMPERATURE DIFFERENCE USING COUNTERFLOW EXAMPLE
8. USING Q=U*AREA*TEMP LOG MEAN CALCULATE VALUE OF U*L
9. CALCULATE CONVECTIVE COEFFICIENT (H) FOR INSIDE TUBES
10. CALCULATE CONVECTIVE COEFFICIENT (H) FOR OUTSIDE TUBES
11. ASSUME A SURFACE TEMPERATURE FOR OUTER SURFACE OF TUBE, FIND PROPERTIES AT THE MEAN TEMPERATURE
12. LOOK UP FOULING FACTORS FOR WATER AND REFRIGERANT
13. ASSUME COPPER TUBING FOR CONDUCTIVITY (K)
14. CALCULATE NUSSELT NUMBER AND OVERALL RESISTANCE (U)
15. VERIFY ASSUMED TEMPERATURE
16. REPEAT PROCESS 10-15 IF NECESSARY
17. WHEN FINAL H AND U ARE FOUND THEN L MAY BE FOUND FROM EQUATION IN LINE 8

Table 20-11 Design Procedure for Finned-Tube Heat Exchanger

1. DO CALCULATIONS WITH BOTH SURFACES
2. ASSUME TUBE OF GAGE 16 FOR BOTH GIVING DIFFERENCE OF .13 INCHES BETWEEN Do AND Di
3. USING M=RHO*AREA*VELOCITY*N TUBES CALCULATE NUMBER OF TUBES
4. USING HEIGHT=N(TUBES)*SPACING CALCULATE HEIGHT
5. USING M=RHO*AREA*VELOCITY OF FACE CALCULATE AREA OF FACE
6. USING A(FACE)=HEIGHT*WIDTH
7. CALCULATE REYNOLDS NUMBER INSIDE TUBES
8. USING GIVEN VALUE FOR DENSITY OF AIR USE TM=70 F
9. CALCULATE NUSSELT NUMBER, FIND H
10. CALCULATE REYNOLDS NUMBER FOR AIR USING HYDRAULIC DIAMETER
11. USING FIGURE FOR SURFACE FIND StPr2/3, FIND H FROM WITHIN St NUMBER
12. FIND CONTACT RESISTANCE ASSUMING Al FINS AND Cu TUBES
13. ASSUME FINS ARE ANNULAR (CIRCULAR) AND HAVE A RADIUS EQUAL TO THE SPACING BETWEEN THE TUBES
14. CALCULATE ALPHA AND BETA FOR ANNULAR FIN SHAPE
15. CALCULATE K_{an} FOR ANNULAR FINS, FIND ETA
16. CALCULATE U VALUE

20.3 Design a steam coil to heat 8500 scfm of outdoor air from 0°F to 45°F with a face velocity of 600 fpm. Low-pressure saturated steam at 5 psig is used. Surface 7.75-5/8T is to examined first.

20.4 Design both the heating coil and the shell-and-tube water heat exchanger for the heating system shown in the sketch provided.

20.5 Design the evaporator/condenser for a cascade low-temperature refrigeration system using R-410 in the high-temperature loop and R-22 in the low-temperature loop. The shell-and-tube heat exchanger will use standard size copper tubes with a steel pipe as a shell. R-22 at the rate of 0.130 kg/s is to be condensed from saturated vapor to saturated liquid at a pressure of 0.91 MPa as it flows through the tubes. R-410A surrounds the tubes and evaporates under pool boiling conditions at a pressure of 1.1 MPa. The exterior of the heat exchanger shell is to be well insulated. Space limits the length of the exchanger to 2 m.

20.6 A shell-and-tube heat exchanger is to cool 1 L/s of water from 15°C to 5°C using R-22 evaporating at 50 kPa on the outside of the tubes. Tubes are to be of copper with a 1.41 cm ID and 1.59 cm OD. Maximum water velocity in the tubes is to be 2 m/s. Design the heat exchanger including specification

of its duty (thermal rating) in kW, the design U-factor, the number of tubes per pass, and the length of the exchanger.

Boiling performance may be obtained from the ASHRAE data provided in the figure provided.

Table 20-12 Dimensional Data for Two Finned-Tube Surfaces

(a) Surface 8.0-3/8*T*	(b) Surface 7.75-5/8*T*
Tube outside diameter = 0.402 in.	Tube outside diameter = 0.676 in.
Fin pitch = 8 per in.	Fin pitch = 7.75 per in.
Fin thickness = 0.013 in.	Fin thickness = 0.016 in.
Hydraulic diameter = 0.001192 ft	Hydraulic diameter = 0.0114 ft
Free-flow area/face area = 0.534	Free-flow area/face area = 0.481
Fin area/total external area = 0.913	Fin area/total external area = 0.913
Total external area/inside tube area = 12.5	Total external area/inside tube area = 20.5
Total external area/outside tube area = 10.3	Total external area/outside tube area = 17.6
Total outside area/face area = 12.9	Total outside area/face area = 24.6
Heat transfer area/total volume = 179 ft^2/ft^3	Heat transfer area/total volume = 169 ft^2/ft^3

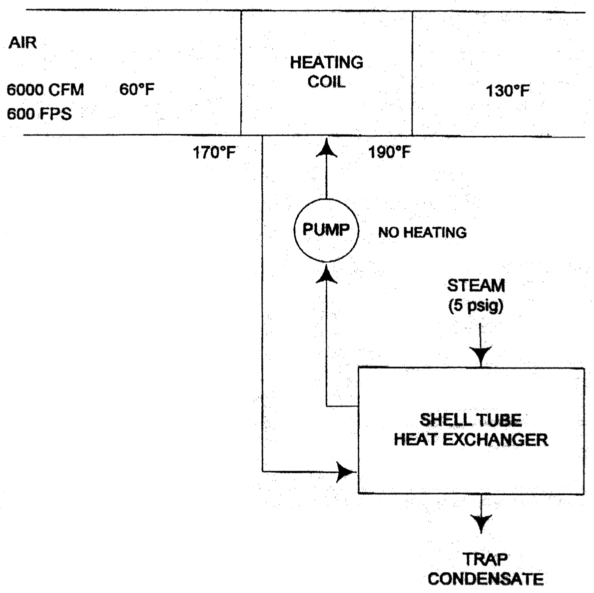


Diagram for Problem 20.4

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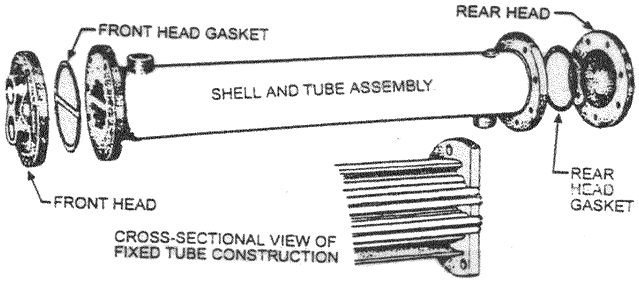

Diagram for Problem 20.5

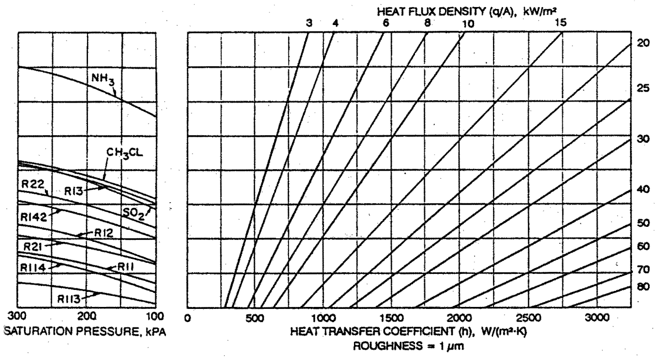

Diagram for Problem 20.6

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Appendix A

SI FOR HVAC&R

This guide conforms to ANSI SI 10-2002, *Standard for Use of the International System of Units (SI): The Modern Metric System*. See ANSI SI 10 for more information and a complete list of conversion factors with more significant digits.

SI PRACTICE

A.1 General

1.1 The International System of Units (SI) consists of seven base units, listed in Table A-1, and numerous derived units, which are combinations of base units (Table A-2).

Table A-1 SI Base Units

Quantity	Name	Symbol
length	metre	m
mass	kilogram	kg
time	second	s
electric current	ampere	A
thermodynamic temperature	kelvin	K
amount of substance	mole	mol
luminous intensity	candela	cd

A.2 Units

2.1 In SI each physical quantity has only one unit. The base and derived units may be modified by prefixes as indicated in Section 4. All derived units are defined by simple formulas using the base units. The basic simplicity of the system can only be kept by adhering to the approved units.

2.2 Angle. The unit of plane angle is the radian. The degree and its decimal fractions may be used, but the minute and second should not be used.

2.3 Area. The unit of area is the square metre. Large areas are expressed in hectares (ha) or square kilometres (km^2). The hectare is restricted to land or sea areas and equals 10 000 m^2.

2.4 Energy. The unit of energy, work, and quantity of heat is the joule (J). The kilowatthour (kWh) is presently permitted as an alternative in electrical applications but should not be introduced in new applications.

1 kilowatthour (kWh) = 3.6 megajoules (MJ)

The unit of power and heat flow rate is the watt (W).

1 watt (W) = 1 joule per second (J/s)

2.5 Force. The unit of force is the newton (N). The newton is also used in derived units that include force.

Examples: pressure or stress = N/m^2 = Pa (pascal)

work = N·m = J (joule)

power = N·m/s = W (watt)

Table A-2 Some SI Derived Units

Quantity	Expression in Other SI Units	Name	Symbol
acceleration			
angular	rad/s^2		
linear	m/s^2		
angle			
plane	dimensionless	radian	rad
solid	dimensionless	steradian	sr
area	m^2		
Celsius temperature	K	degree Celsius	°C
conductivity, thermal	W/(m·K)		
density			
heat flux	W/m^2		
mass	kg/m^3		
energy, enthalpy			
work, heat	N·m	joule	J
specific	J/kg		
entropy			
heat capacity	J/K		
specific	J/(kg·K)		
flow, mass	kg/s		
flow, volume	m^3/s		
force	kg·m/s^2	newton	N
frequency			
periodic	1/s	hertz	Hz
rotating	rev/s		
inductance	Wb/A	henry	H
magnetic flux	V·s	weber	
moment of a force	N·m		
potential, electric	W/A	volt	V
power, radiant flux	J/s	watt	W
pressure, stress	N/m^2	pascal	Pa
resistance, electric	V/a	ohm	Ω
velocity			
angular	rad/s		
linear	m/s		
viscosity			
dynamic (absolute)(m)	Pa·s		
kinematic (n)	m^2/s		
volume	m^3		
volume, specific	m^3/kg		

2.6 Length. The unit of length is the metre. The millimetre is used on architectural or construction drawings and mechanical or shop drawings. The symbol *mm* does not need to be placed after each dimension; a note, "All dimensions in mm" is sufficient.

The centimetre is used only for cloth, clothing sizes, and anatomical measurements.

The metre is used for topographical and plot plans. It is always written with a decimal and three figures following the decimal, i.e., 38.560.

2.7 Mass. The unit of mass is the kilogram (kg). The unit of mass is the only unit whose name, for historical reasons, con-

tains a prefix. Names of multiples of the unit mass are formed by attaching prefixes to the word gram. The megagram, Mg (1000 kg, metric ton or tonne, t), is the appropriate unit for describing large masses. Do not use the term *weight* when *mass* is intended.

2.8 Pressure. The unit of stress or pressure, force per unit area, is the newton per square metre. This unit is called the *pascal* (Pa). SI has no equivalent symbol for psig or psia. If a misinterpretation is likely, spell out Pa (absolute) or Pa (gage).

2.9 Volume. The unit of volume is the cubic metre. Smaller units are the litre, L (m^3/1000); millilitre, mL; and microlitre, μL. No prefix other than m or μ is used with litre.

2.10 Temperature. The unit of thermodynamic (absolute) temperature is the Kelvin. Celsius temperature is measured in degrees Celsius. Temperature intervals may be measured in kelvins or degrees Celsius and are the same in either scale. Thermodynamic temperature is related to Celsius temperature as follows:

$$t_c = T - T_0$$

where

t_c = Celsius temperature, °C
T = thermodynamic temperature, kelvins (K)
T_0 = 273.15 K by definition

2.11 Time. The unit of time is the second, which should be used in technical calculations. However, where time relates to life customs or calendar cycles, the minute, hour, day, and other calendar units may be necessary.

Exception: Revolutions per minute may be used, but revolutions per second is preferred.

A.3 Symbols

3.1 The correct use of symbols is important because an incorrect symbol may change the meaning of a quantity. Some SI symbols are listed in Table A-3.

3.2 SI has no abbreviations—only symbols. Therefore, no periods follow a symbol except at the end of a sentence.

Examples: SI, *not* S.I.; s, *not* sec; A, *not* amp

3.3 Symbols appear in lowercase unless the unit name has been taken from a proper name. In this case the first letter of the symbol is capitalized.

Examples: m, metre; W, watt; Pa, pascal

Exception: L, litre

3.4 Symbols and prefixes are printed in upright (roman) type regardless of the type style in surrounding text.

Example: . . . *a distance of* 56 km *between* . . .

3.5 Unit symbols are the same whether singular or plural.

Examples: 1 kg, 14 kg; 1 mm, 25 mm

Table A-3 SI Symbols

Symbol	Name	Quantity	Formula
A	ampere	electric current	base unit
a	atto	prefix	10^{-18}
Bq	becquerel	activity (of a radio nuclide)	1/s
C	coulomb	quantity of electricity	A·s
°C	degree Celsius	temperature	°C = K
c	centi	prefix	10^{-2}
cd	candela	luminous intensity	base unit
d	deci	prefix	10^{-1}
da	deka	prefix	10^{1}
E	exa	prefix	10^{18}
F	farad	electric capacitance	C/V
f	femto	prefix	10^{-15}
G	giga	prefix	10^{9}
Gy	gray	absorbed dose	J/kg
g	gram	mass	kg/1000
H	henry	inductance	Wb/A
Hz	hertz	frequency	1/s
h	hecto	prefix	10^{2}
ha	hectare	area	10 000 m^2
J	joule	energy, work, heat	N·m
K	kelvin	temperature	base unit
k	kilo	prefix	10^{3}
kg	kilogram	mass	base unit
L	litre	volume	m^3/1000
lm	lumen	luminous flux	cd·sr
lx	lux	illuminance	lm/m^2
M	mega	prefix	10^{6}
m	metre	length	base unit
m	milli	prefix	10^{-3}
mol	mole	amount of substance	base unit
μ	micro	prefix	10^{-6}
N	newton	force	kg·m/s^2
n	nano	prefix	10^{-9}
Ω	ohm	electric resistance	V/A
P	peta	prefix	10^{15}
Pa	pascal	pressure, stress	N/m^2
p	pico	prefix	10^{-12}
rad	radian	plane angle	dimensionless
S	siemens	electric conductance	A/V
Sv	sievert	dose equivalent	J/kg
s	second	time	base unit
sr	steradian	solid angle	dimensionless
T	tera	prefix	10^{12}
T	tesla	magnetic flux density	Wb/m^2
t	tonne, metric ton	mass	1000 kg; Mg
V	volt	electric potential	W/A
W	watt	power, radiant flux	J/s
Wb	weber	magnetic flux	V·s

3.6 Leave a space between the value and the symbol.

Examples: 55 mm, *not* 55mm; 100 W, *not* 100W

Exception: No space is left between the numerical value and symbol for degree Celsius and degree of plane angle.

Note: Symbol for degree Celsius is °C; for coulomb, C.

Examples: 20°C, not 20 °C or 20° C; 45°, not 45 °

3.7 Do not mix symbols and names in the same expression.

Examples: m/s or metres per second,

not metres/second; *not* metres/s

J/kg or joules per kilogram,

not joules/kilogram; *not* joules/kg

3.8 Symbol for product—use the raised dot (·)

Examples: N·m; mPa·s; W/(m^2·K)

3.9 Symbol for quotient—use one of the following forms:

Examples: m/s or $\frac{\mathrm{m}}{\mathrm{s}}$ or use negative exponent

Note: Use only one solidus (/) per expression.

3.10 Place modifying terms such as electrical, alternating current, etc., parenthetically after the symbol with a space in between.

Examples: MW (e); *not* MWe; *not* MW(e)

V (ac); *not* Vac; *not* V(ac)

kPa (gage); *not* kPa(gage); *not* KPa gage

A.4 Prefixes

4.1 Most prefixes indicate orders of magnitude in steps of 1000. Prefixes provide a convenient way to express large and small numbers and to eliminate nonsignificant digits and leading zeros in decimal fractions. Some prefixes are listed in Table A-4.

Examples: 126 000 watts is the same as 126 kilowatts

0.045 metre is the same as 45 millimetres

65 000 metres is the same as 65 kilometres

4.2 To realize the full benefit of the prefixes when expressing a quantity by numerical value, choose a prefix so that the number lies between 0.1 and 1000. For simplicity, give preference to prefixes representing 1000 raised to an integral power (i.e., µm, mm, km).

Exceptions:

1. In expressing area and volume, the prefixes hecto, deka, deci, and centi are sometimes used; for example, cubic decimetre (L), square hectometre (hectare), cubic centimetre.
2. Tables of values of the same quantity.
3. Comparison of values.
4. For certain quantities in particular applications. For example, the millimetre is used for linear dimensions in engineering drawings even when the values lie far outside the range of 0.1 mm to 1000 mm; the centimetre is usually used for body measurements and clothing sizes.

4.3 Compound units. A compound unit is a derived unit expressed with two or more units. The prefix is attached to a unit in the numerator.

Examples: V/m *not* mV/mm

mN·m *not* N·mm (torque)

MJ/kg not kJ/g

Table A-4 SI Prefixes

Prefix	Pronunciation	Symbol	Represents
exa	ex′ a (a as in about)	E	10^{18}
peta	pet′ a (e as in pet, a as in about)	P	10^{15}
tera	as in *terra* firma	T	10^{12}
giga	jig′ (i as in jig, a as in about)	G	10^{9}
mega	as in *mega*phone	M	10^{6}
kilo	kill′ oh	k	10^{3} = 1000
hecto	heck′ toe	h*	10^{2} = 100
deka	deck′ a (a as in about)	da*	10^{1} = 10
deci	as in *deci*mal	d*	10^{-1} = 0.1
centi	as in *centi*pede	c*	10^{-2} = 0.01
milli	as in *mili*tary	m	10^{-3} = 0.001
micro	as in *micro*phone	µ	10^{-6}
nano	nan′ oh (an as in ant)	n	10^{-9}
pico	peek′ oh	p	10^{-12}

*See paragraph 4.2 regarding use of this prefix.

4.4 Compound prefixes formed by a combination of two or more prefixes are not used. Use only one prefix.

Examples: 2 nm *not* 2 mµm

6 m^3 *not* 6 kL

6 MPa *not* 6 kkPa

4.5 Exponential Powers. An exponent attached to a symbol containing a prefix indicates that the multiple (of the unit with its prefix) is raised to the power of 10 expressed by the exponent.

Examples: $1\ \mathrm{mm}^3 = (10^{-3}\ \mathrm{m})^3 = 10^{-9}\ \mathrm{m}^3$

$1\ \mathrm{ns}^{-1} = (10^{-9}\ \mathrm{s})^{-1} = 10^{9}\ \mathrm{s}^{-1}$

$1\ \mathrm{mm}^2/\mathrm{s} = (10^{-3}\ \mathrm{m})^2/\mathrm{s} = 10^{-6}\ \mathrm{m}^2/\mathrm{s}$

A.5 Numbers

5.1 Large Numbers. International practice separates the digits of large numbers into groups of three, counting from the decimal to the left and to the right, and inserts a space to separate the groups. In numbers of four digits, the space is not necessary except for uniformity in tables.

Examples: 2.345 678; 73 846; 635 041; 600.000;

0.113 501; 7 258

5.2 Small Numbers. When writing numbers less than one, always put a zero before the decimal marker.

Example: 0.046

5.3 Decimal Marker. The recommended decimal marker is a dot on the line (period). (In some countries, a comma is used as the decimal marker.)

5.4 Billion. Because billion means a thousand million in the United States and a million million in most other countries, avoid using the term in technical writing.

5.5 Roman Numerals. Do not use M to indicate thousands (MBtu for a thousand Btu), nor MM to indicate millions, nor C to indicate hundreds because they conflict with SI prefixes.

Table A-5 SI Units for HVAC&R Catalogs

Quantity	Unit
Boilers	
Heat output	kW
Heat input	kW
Heat release	kW/m^2
Steam generation rate	kg/s
Fuel firing rate:	
solid	kg/s
gaseous	L/s
liquid	kg/s, L/s
Volume flow rate (combust. products)	m^3/s, L/s
Power input (to drives)	kW
Operating pressure	kPa
Hydraulic resistance	kPa
Draft conditions	Pa
Coil, Cooling and Heating	
Heat exchange rate	kW
Primary medium:	
mass flow rate	kg/s
hydraulic resistance	kPa
Air volume flow rate	m^3/s, L/s
Airflow static pressure loss	Pa
Face area	m^2
Fin spacing, center to center	mm
Controls and Instruments	
Flow rate:	
mass	kg/s
volume	m^3/s, L/s, mL/s
Operating pressure	kPa
Hydraulic resistance	kPa
Rotational frequency	rev/s (rpm)*
Cooling Towers	
Heat extraction rate	kW
Volume flow rate:	
air	m^3/s, L/s
water	m^3/s, L/s
Power input (to drive)	kW
Diffusers and Grilles	
Air volume flow rate	m^3/s, L/s
Airflow pressure loss	Pa
Velocity	m/s
Fans	
Air volume flow rate	m^3/s, L/s
Power input (to drive)	kW
Fan static pressure	Pa
Fan total pressure	Pa
Rotational frequency	rev/s (rpm)*
Outlet velocity	m/s
Air Filters	
Air volume flow rate	m^3/s, L/s
Static pressure loss	Pa
Face area	m^2
Fuels	
Heating value:	
solid	MJ/kg
gaseous	MJ/m^3
liquid	MJ/kg
Heat Exchangers	
Heat output	kW
Mass flow rate	kg/s
Hydraulic resistance	kPa
Operating pressure	kPa
Flow velocity	m/s
Heat exchange surface	m^2
Fouling factor	m^2/W
Induction Terminals	
Heating or cooling output	kW
Primary air volume flow rate	m^3/s, L/s
Primary air static pressure loss	Pa
Secondary water mass flow rate	kg/s
Secondary water hydraulic resistance	kPa
Pumps	
Mass flow rate	kg/s
Volume flow rate	L/s
Power input (to drive)	kW
Developed pressure	kPa
Operating pressure	kPa
Rotational frequency	rev/s (rpm)*
Space Heating Apparatus	
Heat output	kW
Airflow volume flow rate	m^3/s, L/s
Power input (to drive)	kW
Primary medium mass flow rate	kg/s
Hydraulic resistance	kPa
Operating pressure	kPa
Airflow static pressure loss	Pa
Vessels	
Operating pressure	kPa
Volumetric capacity	m^3, L
Air Washers	
Volume flow rate:	
air	m^3/s, L/s
water	m^3/s, L/s
Mass flow rate, water	kg/s
Power input (to drive)	kW
Airflow static pressure loss	Pa
Hydraulic resistance	kPa
Water Chillers	
Cooling capacity	kW
Mass flow rate, water	kg/s
Power input (to drive)	kW
Refrigerant pressure	kPa
Hydraulic resistance	kPa

*Acceptable

A.6 Words

6.1 The units in the international system of units are called SI units—not Metric Units and not SI Metric Units.

(Inch-Pound units are called I-P units—not conventional units, not U.S. customary units, not English units, and not Imperial units.)

6.2 Treat all spelled out names as nouns. Therefore, do not capitalize the first letter of a unit except at the beginning of a sentence or in capitalized material such as a title.

Examples: watt; pascal; ampere; volt; newton; kelvin

Exception: Always capitalize the first letter of Celsius.

6.3 Do not begin a sentence with a unit symbol—either rearrange the words or write the unit name in full.

6.4 Use plurals for spelled out words when required by the rules of grammar.

Examples: metre — metres; henry — henries;
kilogram — kilograms; kelvin — kelvins

Irregular: hertz — hertz; lux — lux;
siemens — siemens

Table A-6 Typical Densities (kg/m^3 at 20°C)

Gases (101.325 kPa)		Liquids		Solids	
butane	2.412	mercury	13 550	lead	11 300
propane	1.829	sulphuric acid	1 830	copper	8 900
oxygen	1.330	refrigerant 12	1 329	steel	7 830
air, dry	1.204	glycerine	1 264	cast iron	7 200
carbon dioxide	1.970	battery electr.	1 260	aluminum	2 700
air, 50% rh	1.191	refrigerant 22	1 213	glass	2 500
acetylene	1.173	water	998	concrete	2 300
nitrogen	1.164	mineral oil	900	brick	1 920
natural gas	0.719	kerosene	820	hardwood	750
helium	0.166	ethyl alcohol	791	softwood	540
hydrogen	0.083	gasoline	730	fiberglass board	80
		propane	580	polystyrene	20

6.5 Do not put a space or hyphen between the prefix and unit name.

Examples: kilometre not kilo metre or kilo-metre;

milliwatt not milli watt or milli-watt

6.6 When a prefix ends with a vowel and the unit name begins with a vowel, retain and pronounce both vowels.

Example: kiloampere

Exceptions: hectare; kilohm; megohm

6.7 When compound units are formed by multiplication, leave a space between units that are multiplied.

Examples: newton metre, not newton-metre;
volt ampere, not volt-ampere

6.8 Use the modifier squared or cubed after the unit name.

Example: metre per second squared

Exception: For area or volume, place the modifier before the units. *Example*: square millimetre; cubic metre

6.9 When compound units are formed by division, use the word *per*, not a solidus (/).

Examples: metre per second, *not* metre/second;
watt per square metre, *not* watt/square metre

TEMPERATURE CONVERSION

(exact)

$$\begin{aligned} t_C &= (t_F - 32)/1.8 & t_F &= 1.8\,t_C + 32 \\ t_C &= T - 273.15 & t_F &= T_R - 459.67 \\ T &= T_R/1.8 & T_R &= 1.8T \\ T &= t_C + 273.15 & T_R &= t_F + 459.67 \end{aligned}$$

where

t_C = Celsius temperature, °C
T = thermodynamic (absolute) temperature, kelvins (K)
t_F = Fahrenheit temperature, °F
T_R = thermodynamic (absolute) temperature, degrees Rankine (°R)
and °C = K = 1.8°F°F = °R = °C/1.8

PHYSICAL PROPERTIES

Atmospheric Pressure

Standard pressure = 101.325 kPa, exact value by definition (approximately 29.921 in. Hg at 32°F; 760 mm Hg at 0°C; 14.696 psi at 32°F).

Gravity

Standard acceleration = 9.806 65 m/s^2, exact value by definition (approximately 32.1740 ft/s^2).

Standard Air

Dry air at 101.325 kPa and 20°C (density ≈ 1.204 kg/m^3)
Specific heat (constant pressure), $c_p = 1.006$ kJ/(kg·K)

Heating of Air

Sensible heat $q_s = 1.2\,Q\Delta t$

Latent heat $q_l = 3.0\,Q\Delta w$

Total heat $q_t = 1.2\,Q\Delta h$

where

Δt = temperature difference, K or °C
Δw = moisture content difference, g/kg (dry air)
Δh = enthalpy difference, kJ/kg (dry air)
Q = volume flow rate, m^3/s (standard air)
q_s, q_l, q_t = heat flow, kW

Water

Heat of vaporization

at 101.325 kPa and 100°C = 2257 kJ/kg

Heat of fusion at 0°C = 334 kJ/kg

CONVERSION FACTORS

When making conversions, remember that a converted value is no more precise than the original value. Round off the final value to the same number of significant figures as those in the original value.

CAUTION: The conversion values are rounded to three or four significant figures, which is sufficiently accurate for most applications. See ANSI SI 10 for additional conversions with more significant figures.

Multiply	By	To Obtain
acre	0.4047	ha
atmosphere, standard	*101.325	kPa
bar	*100	kPa
barrel (42 US gal, petroleum)	159	L
Btu, (International Table)	1.055	kJ
Btu/ft^2	11.36	kJ/m^2
Btu·ft/h·ft^2·°F	1.731	W/(m·K)
Btu·in/h·ft^2·°F (thermal conductivity, k)	0.1442	W/(m·K)
Btu/h	0.2931	W
Btu/h·ft	0.9615	W/m
Btu/h·ft^2	3.155	W/m^2
Btu/h·ft^2·°F (heat transfer coefficient, U)	5.678	W/(m^2·K)
Btu/lb	*2.326	kJ/kg
Btu/lb·°F (specific heat, c_p)	4.184	kJ/(kg·K)
bushel	0.03524	m^3
calorie, (thermochemical)	*4.184	J
calorie, nutrition (kilocalorie)	*4.184	kJ
candle, candlepower	*1.0	cd
centipoise, dynamic vicosity, μ	*1.00	mPa·s
centistokes, kinematic viscosity, ν	*1.00	mm^2/s
clo	0.155	m^2·K/W
dyne/cm^2	*0.100	Pa
EDR hot water (150 Btu/h)	44.0	W
EDR steam (240 Btu/h)	70.3	W
fuel cost comparison at 100% eff.		
cents per gallon (no. 2 fuel oil)	0.0677	\$/GJ
cents per gallon (no. 6 fuel oil)	0.0632	\$/GJ
cents per gallon (propane)	0.113	\$/GJ
cent per kWh	2.78	\$/GJ
cents per therm	0.0948	\$/GJ
ft	*0.3048	m
ft	*304.8	mm
ft/min, fpm	*0.00508	m/s
ft/s, fps	*0.3048	m/s
ft of water	2.99	kPa
ft of water per 100 ft of pipe	0.0981	kPa/m
ft^2	0.09290	m^2
ft^2·h·°F/Btu (thermal resistance, R)	0.176	m^2·K/W
ft^2/s, kinematic viscosity, ν	92 900	mm^2/s
ft^3	28.32	L
ft^3	0.02832	m^3
ft^3/h, cfh	7.866	mL/s
ft^3/min, cfm	0.4719	L/s
ft^3/s, cfs	28.32	L/s
footcandle	10.76	lx
ft·lb$_f$ (torque or moment)	1.36	N·m
ft·lb$_f$ (work)	1.36	J
ft·lb$_f$/lb (specific energy)	2.99	J/kg
ft·lb$_f$/min (power)	0.0226	W
gallon, US (*231 in^3)	3.785	L
gph	1.05	mL/s
gpm	0.0631	L/s
gpm/ft^2	0.6791	L/(s·m^2)
gpm/ton refrigeration	0.0179	mL/J
grain (1/7000 lb)	0.0648	g
gr/gal	17.1	g/m^3
horsepower (boiler)(33,470 Btu/h)	9.81	kW
horsepower (550 ft·lb$_f$/s)	0.746	kW
inch	*25.4	mm
inch of mercury (60°F)	3.377	kPa
inch of water (60°F)	248.8	Pa
To Obtain	**By**	**Divide**

Multiply	By	To Obtain
in/100 ft (thermal expansion)	0.833	mm/m
in·lb$_f$ (torque or moment)	113	mN·m
in^2	645	mm^2
in^3 (volume)	16.4	mL
in^3/min (SCIM)	0.273	mL/s
in^3 (section modulus)	16 400	mm^3
in^4 (section moment)	416 200	mm^4
km/h	0.278	m/s
kWh	*3.60	MJ
kW/1000 cfm	2.12	kJ/m^3
kilopond (kg force)	9.81	N
kip (1000 lb$_f$)	4.45	kN
kip/in^2 (ksi)	6.895	MPa
litre	*0.001	m^3
MBtuh (1000 Btu/h)	0.2931	kW
met	58.15	W/m^2
micron (μm) of mercury (60°F)	133	mPa
mil (0.001 in.)	*25.4	mm
mile	1.61	km
mile, nautical	1.85	km
mph	1.61	km/h
mph	0.447	m/s
millibar	*0.100	kPa
mm of mercury (60°F)	0.133	kPa
mm of water (60°F)	9.80	Pa
ounce (mass, avoirdupois)	28.35	g
ounce (force of thrust)	0.278	N
ounce (liquid, US)	29.6	mL
ounce (avoirdupois) per gallon	7.49	kg/m^3
perm (permeance)	57.45	ng/(s·m^2·Pa)
perm inch (permeability)	1.46	ng/(s·m·Pa)
pint (liquid, US)	473	mL
pound		
lb (mass)	0.4536	kg
lb (mass)	453.6	g
lb$_f$ (force or thrust)	4.45	N
lb/ft (uniform load)	1.49	kg/m
lb$_m$/(ft·h) (dynamic viscosity, μ)	0.413	mPa·s
lb$_m$/(ft·s) (dynamic viscosity, μ)	1490	mPa·s
lb$_f$·s/ft^2 (dynamic viscosity, μ)	47 880	mPa·s
lb/min	0.00756	kg/s
lb/h	0.126	g/s
lb/h (steam at 212°F)(970 Btu/h)	0.284	kW
lb$_f$/ft^2	47.9	Pa
lb/ft^2	4.88	kg/m^2
lb/ft^3 (density, ρ)	16.0	kg/m^3
lb/gallon	120	kg/m^3
ppm (by mass)	*1.00	mg/kg
psi	6.895	kPa
quad (10^{15} Btu)	1.06	EJ
quart (liquid, US)	0.946	L
revolutions per minute (rpm)	*1/60	Hz
square (100 ft^2)	9.29	m^2
tablespoon (approx.)	15	mL
teaspoon (approx.)	5	mL
therm (100,000 Btu)	105.5	MJ
ton, short (2000 lb)	0.907	Mg; t (tonne)
ton, refrigeration (12,000 Btu/h)	3.517	kW
torr (1 mm Hg at 0°C)	133	Pa
watt per square foot	10.8	W/m^2
yd	*0.9144	m
yd^2	0.836	m^2
yd^3	0.7646	m^3
To Obtain	**By**	**Divide**

Note: In this list the kelvin (K) expresses temperature intervals. The degree Celsius symbol (°C) is often used for this purpose as well.

*Conversion factor is exact.

Appendix B

SYSTEMS DESIGN PROBLEMS

B.1 Combination Water Chillers

(Centrifugal and Absorption Machines in Series)

Given:

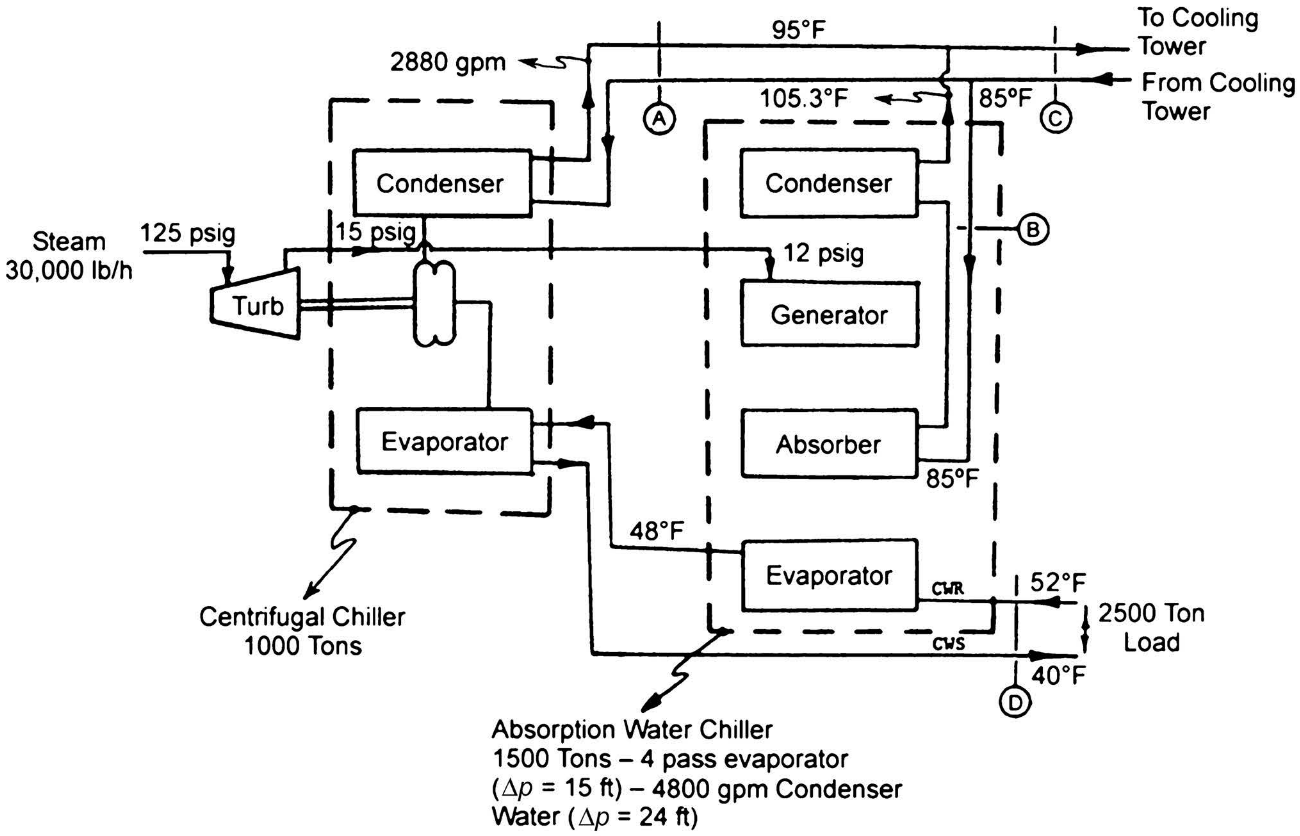
1. The 1000 ton turbine-driven centrifugal compressor in the figure below is supplied with steam at 30,000 lb/h at 125 psig. The turbine exhaust pressure is 15 psig. The temperature rise through the condenser is 10 °F.
2. A 1500 ton lithium bromide water chiller uses exhaust steam at 12 psig from the steam turbine. It has a 4-pass evaporator with a pressure drop of 15 ft. The leaving temperature for the 4800 gpm condenser water is 105.3 °F.
3. Water velocity for the condenser and chilled-water piping is limited to 10 ft/s.
4. Chilled-water supply temperature is 40°; chilled-water return temperature is 52°F.
5. Cooling tower design data: 95 °F dry bulb, 76°F wet bulb, 9°F approach.

Required:

1. Calculate the overall steam rate in pounds per hour per ton for the refrigeration plant.
2. Calculate the chilled-water flow rate in gpm.
3. What is the temperature of water off the tower? What is the temperature of the water entering the cooling tower?
4. In the evaporator of the centrifugal compressor, what is the pressure drop and water velocity in the tubes?
5. In the condenser of the centrifugal compressor, what is the pressure drop and water velocity in the tubes?
6. Using Schedule 40 pipe, what size pipe would you use for the condenser water piping to each machine (a,b), the cooling tower (c), and the chilled-water piping (d)?

B.2 Absorption Chiller Selection

A small college is to be built in the Santa Fe, New Mexico, area (elevation 7000 ft). You have the assignment to design



the mechanical systems for this project. You decide to recommend a central plant for both heating (steam) and cooling (chilled water). Since the available fuel is relatively inexpensive, you decide to use absorption refrigeration to keep the electrical demand as low as possible and to make use of the steam boilers that would otherwise be idle in the summertime.

Your preliminary analysis indicates that the first four buildings to be built will have the following characteristics:

Building	Area, ft^2	Estimated ft^2/ton	Estimated Total Tons
A	75,000	400	190
B	50,000	325	150
C	65,000	350	185
D	25,000	300	85

In addition, you decide to install sufficient additional capacity to handle 100,000 ft^2 of building (you do not know exactly what type of building it will be, so average and estimate). Also estimate a 5% loss in capacity in the piping and distribution system.

Steam is available at 30 psi and can be reduced to any pressure you desire. You decide to have 42°F to 58°F or a 16°F temperature difference (TD0 as your chilled-water design temperatures. Condensing water is available from the cooling tower at 80°F. The maximum allowable pressure drop through the chiller is 40 ft for the condenser-absorber and 20 ft for the evaporator. You want at least a 0.001 fouling factor on the condenser and a 0.0005 factor on the evaporator.

A. Select and specify an absorption chiller to handle the determined capacity. Indicate water and steam flow rates and unit pump motor horsepower requirements.

B. What is the rate of refrigerant flow at maximum load?

C. What is the theoretical pump horsepower required for the chilled water and condensing water due to the pressure drop through the unit?

D. What is the total hourly purchased energy requirement for this chiller (electrical + thermal)? How does this compare with an equivalent motor-driven centrifugal machine? Which one is more economical to operate (assume electricity costs five times as much as steam)?

E. What is the cooling tower requirement for the absorption machine compared with a centrifugal machine? How might this affect your answer to question D above?

B.3 Owning and Operating Costs

Management has decided to move a subsidiary of your company to Indianapolis, Indiana. This will necessitate a new office building. The chief engineer was asked to estimate the owning and operating cost for the refrigeration and summer air-conditioning services in the building. The chief just called you in this morning and now it is your estimate. Here is all the information available.

Building: Five-story, 160 ft by 300 ft gross with 90% of floor area air conditioned

Refrigeration: reciprocating compressors, R-134a, 95°F condensing temperature, 40°F evaporator temperature

Power rate: \$0.09 per kWh. Includes both demand and energy charge.

Water rate: \$1.00 per 1000 gal

Water rate for condenser: 3.0 gpm per ton for full load operating hour

Operating hours for auxiliaries: 5 1/2 days per week

Power requirements:

Fan, air: 0.4 bhp per 1000 cfm
Fan, cooling tower: 0.05 bhp per ton
Pump, chilled water: 1.10 bhp per ton
Pump, cooling tower: 0.09 bhp per ton

Annual operating labor and maintenance: 4 1/2% of first cost (average for life of equipment).

Interest: 5%

Taxes: \$2.50 per \$100 of assessed valuation.

Assessed valuation: 25% of first cost.

B.4 Animal Rooms

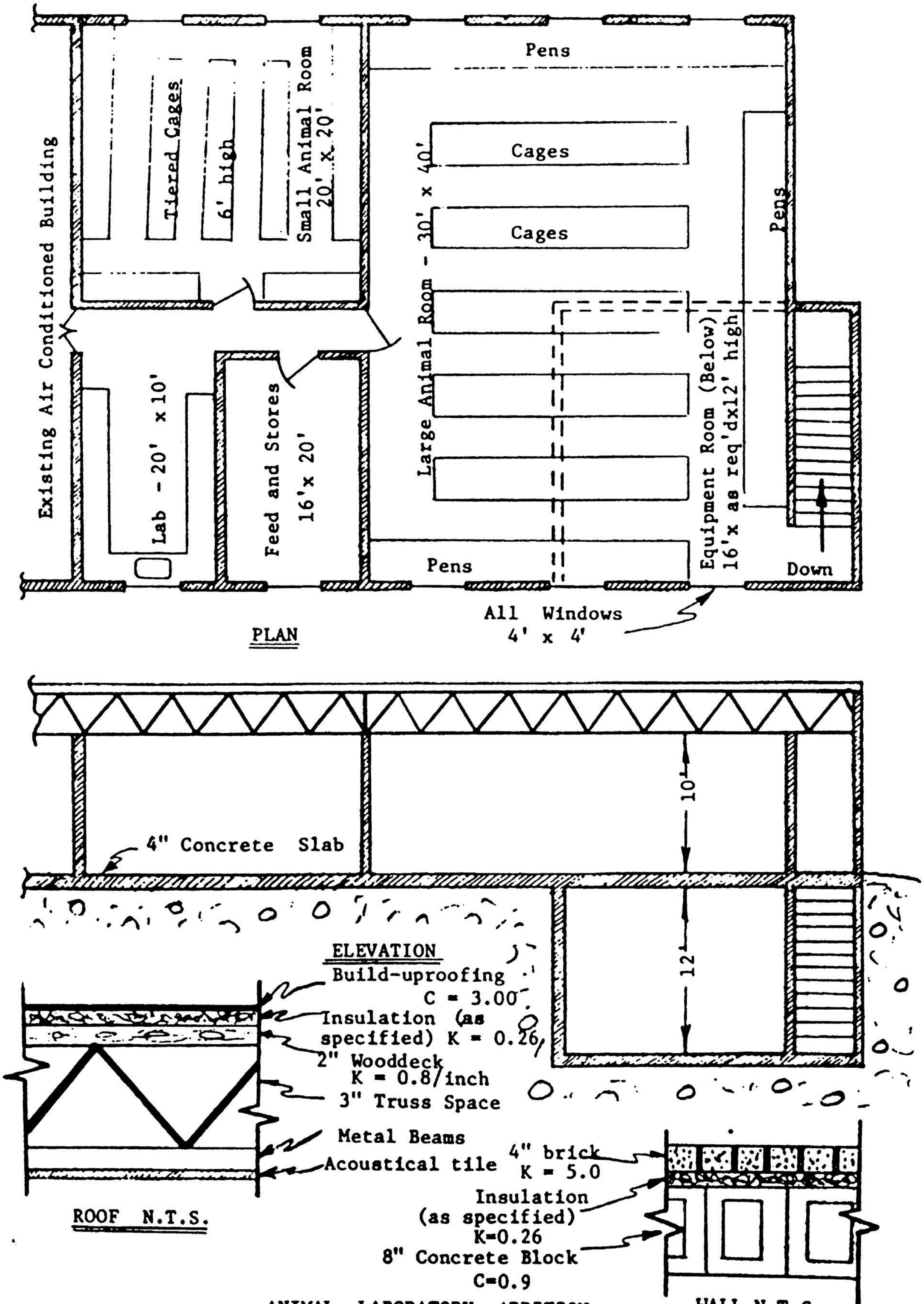
Facilities are to house laboratory animals at control temperature, humidity, air motion, odor, and bacterial count. Design conditions vary widely depending on whether the animals are subjected to test environments or simply quartered. A general range of design temperatures and relateve humidity is tabulated in Table 1.

Recommended tolerances are +2°F dry bulb and ±5% rh at the point of control. Low temperature gradients are desirable. The maximum spread vertically and horizontally within the cage zones should be limited to within ±1.5°F of the control point. Air motion limits in the cage zone are 35 to 50 fpm for general applications and 25 to 35 fpm in quarters for mice. See the diagram on the following page.

The approximate amount of heat released by laboratory animals at rest and during normal activity is shown in Table 2.

Conformance to temperature and humidity requirements requires control on a room basis and preferably on a module basis, because cage loading, occupancy distribution, and animal heat release are variable. A module constitutes two rows of cages with a working aisle separation. Temperature and velocity gradient control requires low supply air to room air temperature differential, overhead high induction diffusion, uniform horizontal and vertical air distribution, and low return outlets.

Odor control within animal rooms requires 100% outdoor air for odor removal from recirculated air. Unidirectional single pass airflow through the room is an aid for lowering odor levels. Air rates range from 10 to 20 air changes per hour depending on the animal occupancy and density.



ANIMAL LABORATORY ADDITION

Table B-1 Animal Room Temperature

Animal	Temp., °F	Relative Humidity, %
Mice	73 to 77	45 to 50
Rats	73 to 77	45 to 50
Guinea Pigs	72 to 74	45 to 50
Rabbits	70 to 72	45 to 50
Cats	75 to 77	45 to 50
Dogs	70 to 72	45 to 50
Monkeys	76 to 79	75

Table B-2 Heat Generated by Laboratory Animals

		Heat Generation, Btu/h per Animal			
		Response (Basal)	Normally Active (est.)		
Animal	Mass, g	Total Heat	Sensible	Latent	Total
Mouse	21	0.6	3.3	1.7	5.0
Hamster	118	1.65	10	3	13
Pigeon	275	4.63	15	3	18
Rat	300	4.46	22	11	33
Guinea Pig	410	5.80	32	15	47
Chicken	2,100	18.8	64	12	76
Rabbit	2,600	19.3	61	18	79
Cat	3,000	25.1	75	25	100
Monkey	4,200	34.5	92	46	138
Dog	16,000	87.5	250	120	370
Goat	36,000	137.0	410	130	540
Sheep	45,000	192.0	560	190	750
Pig	250,000	718.0	2100	700	2800

Table B-3 Average Odor-Free Requirements

Animal	Mass, g	Gross Space ft^3/animal	Odor-Free Air cfm/animal
Mice	21	1.0	0.10
Rats	200	3.5	0.75
Guinea pigs	410	6.0	1.5
Chickens	2,100	8.0	2.0
Rabbits	2,600	10.0	2.0
Cats	3,000	35.0	8.0
Monkeys	3,200	100.0	20.0
Dogs	14,000	150.0	50.0

Odor-free air rates for various animal occupancies are listed in Table 3. Control of odor dissemination to adjoining spaces requires that the animal room be maintained under negative pressure or that air locks be employed. Bacterial control require high-efficiency filtration or germicidal treatment.

Conditions in animal rooms must be continuously maintained, which requires year-round availability of refrigeration and, in some cases, dual air-conditioning facilities and emergency power for motor drives and control instrument energy.

An air-conditioning system is to be designed for an animal room for housing laboratory animals. The area is a separate wing of a research facility and is to have its own completely independent system to provide year-round temperature and humidity control. The large animal room will have a maximum of 80 dogs or their equivalent and the small animal room will contain mice, rats, rabbits and a few monkeys, with an equivalent heat release of 200 rabbits.

Insulation and window construction shall be selected as follows:

Design Temperature	Insulation	Windows (all sealed)
+30 °F and above	None	Single pane
+10 °F to +30°F	1 in.	Single pane
−20 °F to +10°F	2 in.	Two pane
below −20 °F	3 in.	Two pane

U for single pane = 1.13 Btu/h·ft^2·°F
U for double pane = 0.45 Btu/h·ft^2·°F

Lights:

In animal rooms: 1 W/ft^2
In feed room: 1 W/ft^2 (fluorescent)
In laboratory: 4 W/ft^2

For laboratory: 3 bunsen burners, one 1100 W sterilizer

For feed room: 1 hp electric food grinder

Perform the following:

(a) Calculate the heating loads for each room.
(b) Calculate the cooling loads for each room.
(c) 1. Calculate the air quantities (cfm) for each room.
  2. Make a schematic diagram of the apparatus required for air conditioning. (Heat air with hot water from hot water boiler).
(d) Calculate the areas of the cooling coil required and the face area (frontal area).
(e) 1. Make a single line duct layout.
  2. Size the ducts.
  3. Select the grilles, diffusers, and other components.
(f) 1. Select filter pressure loss.
  2. Calculate pressure losses in other parts of the air distributing system, coils, and outdoor louvers.
  3. Calculate the total pressure loss.
  4. Select fan, fans or units.
(g) 1. Select a refrigeration water chilling unit or units.
  2. Select a hot water boiler.
  3. Select pump or pumps for chilled water circulation.
(h) 1. Draw a plan of the building on 22 in. × 17 in. paper (1/2 in. = 1 ft 0 in.).
  2. Draw the mechanical system designed on the plan. (Make sure mechanical system outline is heavier than building outline.)
  3. Draw a plan and elevation of equipment room showing main items of equipment and ducts (1/2 in. = 1 ft, 0 in.).

B.5 Greenhouse

An environmental control system is to be developed for a greenhouse located in the Lafayette, Indiana, area. This greenhouse is a rigid frame structure, 40 ft wide and 100 ft long, with a roof slope of 6/12 and a sidewall height of 6 ft. The covering material is a single layer of polyethylene plastic.

Determine the following, listing your sources of data and assumptions used:

1. What is the maximum heat requirement for the house in Btu/h if the inside temperature is not to fall below 60°F?
2. On a clear winter day, January 21, with an outside noon temperature of 30°F and 65% rh, how much heat (Btu/h) or ventilation (cfm) is required to maintain an inside temperature somewhere between 60 and 70°F at 12 noon solar time? (Consider sensible heat only.)
3. Reconsider Part 2 assuming that half of the solar radiation load is used to evaporate water from the plants, soil, and floor and manifests itself as a latent heat load. A further restriction of a maximum 50% inside rh is added to prevent condensation on the inside of the plastic covering. Under these conditions, what is the amount of heat (Btu/h) and/or ventilation (cfm) required to maintain an inside temperature of 75°F?
4. On a clear summer day, July 21, with an outside noon temperature of 85° and 50% rh, what is the minimum ventilation rate (cfm) required to prevent the inside temperature from exceeding 90°F? (Consider sensible heat only.)
5. A pad evaporative cooler is to be considered for cooling the greenhouse on a clear day, August 21, with a noon temperature of 90°F and 40% rh. The entering air, after passing through the pad, has 90% rh. Using the ventilation rate from Part 4, what is the exhaust air temperature? How many square feet of pad are required if the velocity through the pad is restricted to 150 fpm?

B.6 Drying Room

You are making a feasibility study for a 90,000 ft^3 drying room built inside a factory. An air conditioner is located outside the drying room. Air enters the conditioner from the plant at 82°F dry bulb, 50% rh, and is distributed to the drying room at 82°F dry bulb, 10% rh. The air leaves the drying room at 112°F dry bulb, 90% rh and is exhausted from the factory through an exhaust system. Product passes through the drying room on a belt conveyor entering at 3600 lb/h at 128°F dry bulb, 40% moisture content.

Assume 10 air changes per hour through the drying room. Assume no heat or moisture flow through the drying room structure.

(a) What is the refrigeration load on the conditioning apparatus in tons?

(b) What is the moisture content of the product leaving the drying room?

You report the results to the product engineer who wants to know if you can dry the product to 20% moisture. The engineer says the 112°F leaving air is maximum; otherwise; the product will be too hot and will have to be cooled after it leaves the drying room to prevent checking.

You study the conveyor and feel it will be possible to modify the air distribution arrangement in the drying room. This will give more intimate contact between air and product and perhaps the same air volume flow rate can be used.

(c) What will be the humidity ratio of the air leaving the drying room for a 20% moisture content of the product? (You retain the 10 air changes per hour and the 112°F dry-bulb leaving air.)

B.7 Air Washer

A chiller supplies water to a washer. The water leaves the chiller at 44°F and is returned at 54°F. Design conditions in the building are 78°F dry bulb and 64.5°F wet bulb. A mixture of outdoor and recirculated air enters the washer at 88°F dry bulb and 72°F wet bulb. The total air circulated is 33,000 cfm. The washer has a performance factor of 0.85. The washer is a two-bank single-stage design.

1. What is the refrigeration load on the chiller?
2. What water quantity should it be handling in gpm?
3. What room SHR will these operating conditions exactly satisfty?
4. How many nozzles would you expect to find in the washer assuming 2 gpm per nozzle?
5. What should be the cross-sectional area of the washer assuming a velocity of 500 fpm?
6. Suppose the design engineer made a mistake in estimating the head on the chilled water circulating pump. The operating head was lower than the selection point for the pump. From the characteristic pump curve you estimate, the pump is now delivering 30% more water than in Part 2. Assuming the same performance factor, load, and air flow, what is your answer to Part 3? Discuss the effect of the conditions in the building.

B.8 Two-Story Building

As both architect and chief mechanical engineer of your own consulting engineering firm, you are responsible for the design of the building and for the sizing and selection of the major components of the HVAC system for the building. A report on your design, analysis, and recommendations for the HVAC system must be prepared for the sponsor and his staff. Sizing and selection are to cover only the heating, cooling, and humidifying equipment with cost estimates. Estimates of annual HVAC energy use and cost are to be included.

Simple tables of design conditions, building data, results, and recommended equipment should be included in the body of report. A cover letter is mandatory. Completely labeled sketches and diagrams should be included as appropriate. Detailed calculations and/or computer printouts are to be included as appended material.

The 24,000 ft^2, two-floor office building is to be located in Louisville, Kentucky. There are to be six, separately thermostated zones, three on each floor. The east and north ends are the be combined into one zone, the west and south into a second, and a center portion into the third. The zones are each approximately the same size. The east and north zones are to

contain a minimum of 70% glass in the exterior walls while the exterior walls of the west and south zones are to contain between 20 and 40% glass.

Since the owner will be picking up the utility bills, he or she is somewhat energy conscious. However, the owner is also very concerned about the first cost of the building and does not plan extreme departures from common building practices and materials. A rooftop installation is planned to conserve interior space.

As the engineer and architect hired by the owner-to-be of the building, you are to design the building, specifying the layout, wall and ceiling descriptions, types of glass, doors, etc. Neglect details such as closets, room dividers, halls, etc. Items to be determined include:

Design heating load for each space.
Design cooling load for each space.
Projected energy requirements for heating and for cooling.
Sizing of major components (heating unit, cooling unit, fan, humidifier).
Sizing and layout of ducts.
Specific recommendations for equipment and fuels.
Estimated initial cost of HVAC equipment and annual operating cost.

B.9 Motel

As both architect and chief mechanical engineer of your own consulting engineering firm, you are responsible for the design of the building and for the sizing and selection of the major components of the HVAC system for the building. A report on your design, analysis, and recommendations for the HVAC system must be prepared for the sponsor and his staff.

Sizing and selection are to cover only the heating, cooling, and humidifying equipment with cost estimates. Estimates of annual HVAC energy use and cost are to be included.

Simple tables of design conditions, building data, results and recommended equipment should be included in the body of the report. A cover letter is *mandatory*. Completely labeled sketches and diagrams should be included as appropriate. Detailed calculations are to be included as appended material.

The 24-unit (plus office) single building motel is to be located in St. Louis, Missouri, in the Lambert Airport area. Each unit is to be 12 ft by 24 ft with the office twice the size of a regular room. Two-thirds of the units are to be nonsmoking. Each unit will be conditioned with its own packaged terminal air conditioner (PTAC) or packaged terminal heat pump (PTHP).

Since the owner will be picking up the utility bills, he or she is somewhat energy conscious. However, the owner is also very concerned about the first cost of the building and does not plan extreme departures from common building practices and materials.

As the engineer and architect hired by the owner-to-be of the building, you are to design the building, specifying the layout, wall and ceiling descriptions, types of glass, doors, etc. Items to be determined include:

Design heating load for each space.
Design cooling load for each space.
Projected energy requirements for heating and for cooling.
Sizing of major components (heating unit, cooling unit, fan, humidifier).
Specific recommendations for equipment and fuels.
Estimated initial cost of HVAC equipment and annual operating cost.

B.10 Building Renovation

As chief mechanical engineer of a consulting engineering firm, you are responsible for the design, sizing, and selection of the major components of the HVAC system for the building. A report on your design, analysis, and recommendations for the HVAC system must be prepared for the sponsor and his staff.

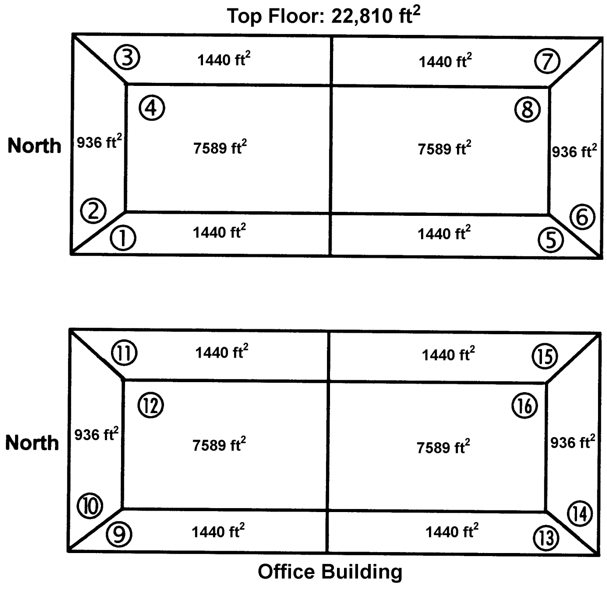
Sizing and selection are to cover the heating, cooling, and humidifying equipment with cost estimates. Estimates of annual HVAC energy use, *both before and after* the modifications, are to be included.

Simple tables of design conditions, building data, results, and recommended equipment should be included in the body of report. A cover letter is *mandatory*.

Completely labeled sketches and diagrams should be included as appropriate.

Detailed calculations and/or computer printouts are to be included as appended material.

The project concerns the complete renovation of an office building located in St. Louis, Missouri, and shown in the following sketch.



Sketch of Project 10 Building

The building is divided into 16 separately thermostated zones. Physical description and building operation (base case) data are

Building roof area: 22,810 ft^2
Building floor area: 45,620 ft^2
Ceiling height: 8.5 ft
Building exterior wall area: 9,460 ft^2 (net)
Building glass area: 7,536 ft^2
Building thermal mass: M (medium)
Uniform internal load density: 2.9 W/ft^2
Occupancy: 408 people (uniformly distributed)
Original U-factors:

- Roof—0.23 $\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$
- Walls—0.18 $\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$
- Glass—0.8 $\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$
- Shading coefficient for glass: 0.5

Originally built in the late 1960s with an all-electric reheat HVAC system, the renovated building will replace the inefficient reheat system with all-electric packaged terminal air conditioners (PTACs) in each space. In the exterior zones, the PTACs will include provisions for ventilation air. For the interior zones, the PTACs will operate without outdoor air provisions and a separate rooftop makeup air system will be used for ventilation requirements. When renovated, all walls and ceilings will include an additional 2 inches of glass fiber, organic bonded rigid insulation, The windows will be upgraded to double pane, 1/4 in. air gap, aluminum frame with thermal break (nonoperable). The internal load density is estimated to have increased over the years from the original 2.9 W/ft^2 to 6.0 W/ft^2.

Include the following items:

- Design heating load for each space.
- Design cooling load for each space.
- Projected annual energy requirements for heating and for cooling.
- Sizing and selection of major components.
- Sizing and layout of ducts for interior zones.
- Estimated initial cost of HVAC equipment and annual operating cost.
- Potential problem areas with this type of equipment.

B.11 Building with Neutral Deck Multizone

As the chief mechanical engineer of your own consulting engineering firm, you are responsible for preliminary design of the HVAC system for the building shown in the figure at right. Selection of all major components is to be included. A report on your analysis and recommendations for the HVAC system must be prepared for the sponsor and his staff.

Sizing and selection will cover all heat exchangers (coils), fans, refrigeration units, boilers and/or other heating equipment, humidifiers, cooling towers, heat reclaim and/or air-to-air energy recovery equipment, and pumps. Piping, ducting, and related fittings need not be sized nor selected at this time.

Simple tables of design conditions, building data, results and recommended equipment should be included in the body of the report. A cover letter is mandatory. Completely labeled sketches and diagrams should be included as appropriate. Detailed calculations and appropriate manufacturers' catalog data are to be included as appended material.

Building Location: Atlanta, GA

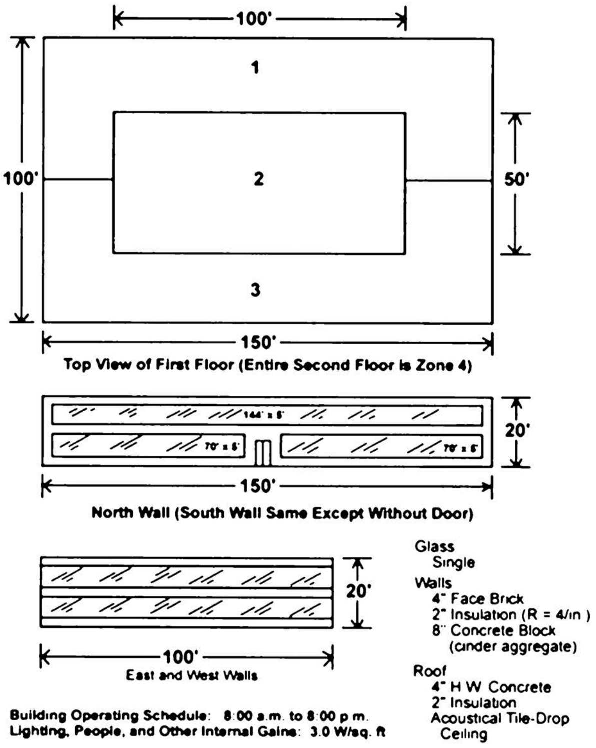
The ventilation requirements are to be in accordance with ASHRAE Standard 62. Anticipate occupany rate is 10 persons per 1000 ft^2 of floor space. Design pressure drop for the ducting system is 3.25 in. of water. The design loads are given in the following table.

Design Loads (Btu)

Zone	Heating (sensible)	Cooling (sensible)	Cooling (latent)
1	−95,000	164,000	47,000
2	+33,000	157,000	14,000
3	−98,000	199,000	40,000
4	−276,000	567,000	72,000

The winter latent load is negligible.

Type of HVAC System:	Multizone with neutral deck
Primary Systems:	R-22 condensing unit and DX coil, multiple gas-fired boilers for steam coil
Auxiliary Equipment:	Spray washer, heat pipe air-to-air energy recovery system, air-side economizer



Sketch for Project 11

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Principles of Heating, Ventilating, and Air Conditioning is a textbook based on the 2017 *ASHRAE Handbook—Fundamentals*. It contains the most current ASHRAE procedures and definitive, yet easy to understand, treatment of building HVAC systems, from basic principles through design and operation.

It is suitable both as a textbook and as a reference book for undergraduate engineering courses in the field of air conditioning, heating, and ventilation; for similar courses at technical and vocational schools; for continuing education and refresher short courses for engineers; and for adult education courses for professionals other than engineers, especially when combined with *ASHRAE Handbook—Fundamentals*.

The material is divided into three major sections: general concepts, Chapters 1–10; air-conditioning systems, Chapters 11–16; and HVAC&R equipment, Chapters 17–20. There are several significant changes in this revised edition. Chapter 4 has new values for climatic design information. Chapter 7 has been extensively revised with new design data. In addition, the chapters on system design and equipment have been significantly revised to reflect recent changes and concepts in modern heating and air-conditioning system practices.

This book includes access to a website containing the Radiant Time Series (RTS) Method Load Calculation Spreadsheets, which are intended as an educational tool both for the student and for the experienced engineer wishing to explore the RTS method. These spreadsheets allow the user to perform RTS cooling load calculations for lights, people, equipment, walls/roofs, and fenestration components using design day weather profiles for any month. Cooling and heating loads can be calculated for individual rooms or block load zones. Twelve-month cooling calculations can be done to determine the month and time of peak cooling load for each room or block load zone. In addition, room/zone worksheets can be copied and modified within the spreadsheet to analyze as many rooms or zones as desired; the number of rooms/zones is limited only by the available computer memory.

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