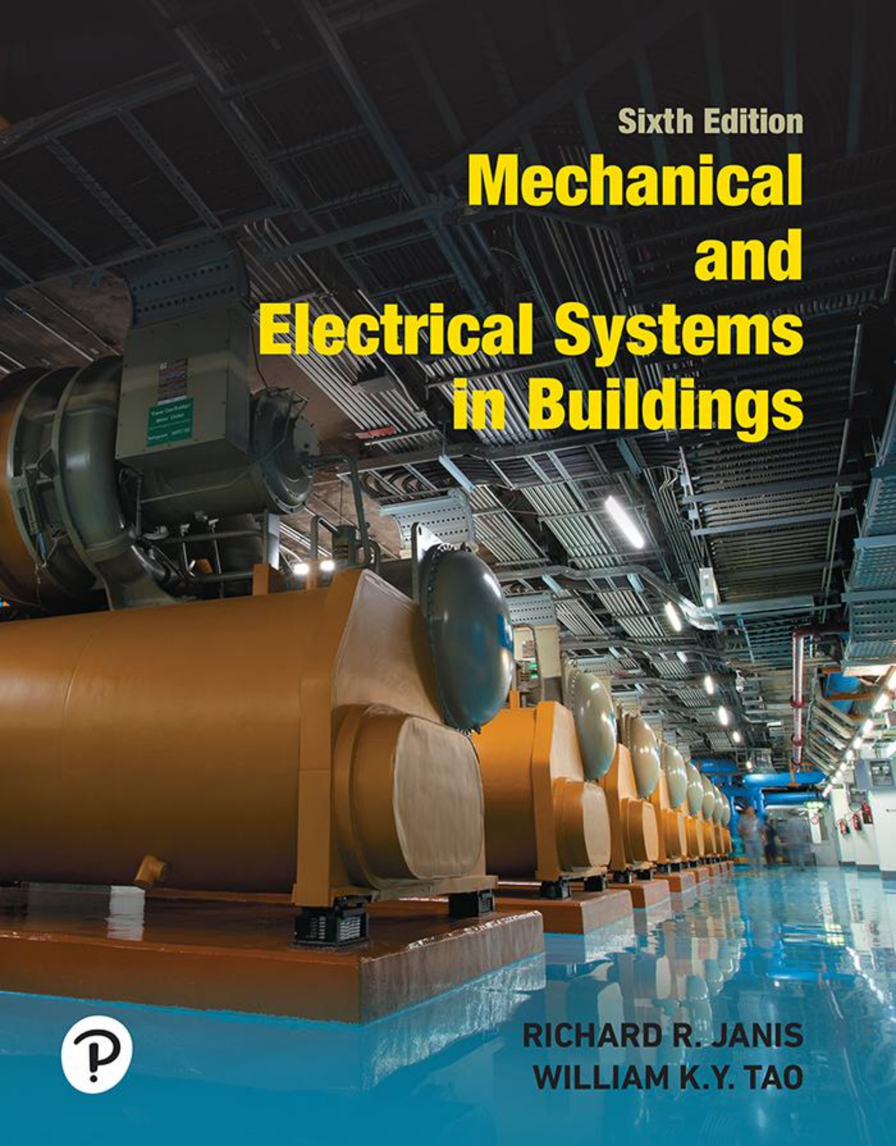

Sixth Edition

Mechanical and Electrical Systems in Buildings

RICHARD R. JANIS
WILLIAM K.Y. TAO

MECHANICAL AND ELECTRICAL SYSTEMS IN BUILDINGS

Sixth Edition

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PREFACE

This book on mechanical and electrical systems covers five major disciplines: HVAC, plumbing and fire protection, electrical power and telecommunications, illumination, and noise and vibration control.

Coauthors Richard R. Janis and William K.Y. Tao have both taught university courses on mechanical and electrical systems for more than 30 years while working as consulting engineers. Their various courses have emphasized the roles of participants in the building process as well as the theories and technologies of system design. In 1989, they finished the first edition of this text in response to the need for a text that was up to date with current practice, emphasizing the *Why?* and the *How?* as well as the *What?*

The topics covered in this book are in a state of continuous advancement, triggering the need for substantial updating every few years. This sixth edition incorporates new developments in all the major disciplines, with updates on electrical, lighting, telecom, plumbing, and HVAC.

NEW TO THIS EDITION:

- Life-cycle cost analyses including societal cost of pollution and economic benefits of indoor environmental quality (IEQ)
- An entire chapter exploring energy impact and economics of various generic architectural and systems options
- VRF heat pump and heat recovery operations using three pipe technology
- Operation of condensing boilers and furnaces and direct fired heating
- Energy and comfort advantages of high-volume, low-speed (HVLS) fans
- EPA WaterSense, the latest development in water conserving fixtures
- Economics of solar photovoltaic systems

The U.S. building industry is embracing sustainable design principles, which were part of the authors' practice and course offerings long before the concept was accepted. *Sustainable design* means that engineers, architects, owners, contractors, and facility managers must interact in a team effort to provide high-quality, productive environments for people while considering the impact of their decisions on the environment. This book is a text and reference for students and professionals interested in an interactive, multidisciplinary approach to the building process.

ORGANIZATION OF THE SIXTH EDITION

In prior editions, Chapters 1 and 2 grew and became unwieldy with the addition of topics. In this sixth edition, even more materials are added to account for advancements in energy and sustainable design. Accordingly, we have reorganized and expanded these new and existing materials into four chapters instead of two. The book is now organized as follows:

Chapter 1, Introduction to Mechanical and Electrical Systems, Sustainable Design, and Evaluating Options, includes new life-cycle cost examples which consider the off-site cost of pollution and the economic effects of indoor environmental quality.

Chapter 2, HVAC Fundamentals, covers the engineering basics required to understand systems.

Chapter 3, HVAC Load Estimating, includes an updated detailed example of heating and cooling load calculations using a building example consistent with current energy codes.

Chapter 4, HVAC Load Management, contains the analysis of various architectural, ventilation, lighting, and appliance options on a hypothetical 30,000 ft^2 building. Analysis includes the effect on load, initial building cost, energy cost, and life-cycle cost for alternative designs involving the following factors:

- Building geometry and form
- Wall orientation
- Altering glass orientation
- Window-to-wall area ratio
- Control solar loads through windows
- Thermal properties of walls and roof
- Infiltration load control potential
- Ventilation criteria and controls
- Task lighting
- Daylighting
- Appliances load control

Chapters 5 through 9 cover HVAC systems and equipment. These chapters are updated to include new technologies to save energy and in some instances reduce the cost of systems. New, enhanced, and expanded topics include condensing boilers and furnaces, direct fired heating, ground source heat pumps, VRF heat pumps and heat recovery, and high-volume, low-speed fans.

Chapter 10 covers plumbing. Updates include condensing water heaters, instantaneous water heaters, and the EPA WaterSense program.

Chapter 11 covers fire protection with general updates for improved clarity.

Chapters 12 through 15 cover electrical systems with general updates including current economics of solar PV systems.

Chapters 16 through 19 cover lighting, including general updates, strategies for high performance, and enhanced coverage of newly dominant LED light sources.

Chapter 20 on noise and vibration takes on a higher level of importance as sound is increasingly recognized for its effect on indoor environmental comfort.

Chapter 21 on architectural accommodation covers system coordination, space planning options, and has many images to convey the visual impact of design.

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Richard R. Janis
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Organizations

Special thanks go to the following organizations for providing valuable design data:

ASHRAE	American Society for Heating Refrigeration and Air Conditioning Engineers
IESNA	Illuminating Engineering Society of North America
ASPE	American Society for Plumbing Engineers
NCAC	National Council of Acoustical Consultants
NEC	National Electrical Code
NFPA	National Fire Protection Association
TIA/EIA	Telecommunication Industry Association
NSPC	National Standard Plumbing Code/ National Association of Plumbing-Heating-Cooling Contractors

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CONTENTS

CHAPTER ONE

INTRODUCTION TO MECHANICAL AND ELECTRICAL SYSTEMS, SUSTAINABLE DESIGN, AND EVALUATING OPTIONS

This book is about mechanical and electrical (M/E) systems in buildings. These systems include:

- Heating, ventilating, and air-conditioning (HVAC)
- Plumbing, consisting of water supply, fixtures, sanitary drainage, sewage treatment and disposal, and storm drainage
- Fire protection, including fire alarm and suppression systems
- Electrical, consisting of power and communications
- Lighting

Over the last 125 years, these systems have been developed and continually improved to make buildings habitable, functional, productive, and safe. In addition, they have allowed flexibility to expand the limits of architectural design. Before modern heating, air-conditioning, and illumination systems, building dimensions were limited due to the need to access windows for light and natural ventilation (see Fig. 1.1). Floors were typically 60 ft or less in depth, or included light wells. Windows were operable and needed to be large and tall enough to allow deep light penetration. Ceiling heights were high to promote stratification of summer heat and to allow the use of operable transom windows over doors for ventilation of interior spaces (see Fig. 1.2).

Air-conditioning and good artificial lighting gave architects the flexibility to design larger floors, and good elevators and life safety systems made high-rise construction possible. These developments occurred when energy to operate buildings was inexpensive by today's standards, and there was very little concern about fossil fuel depletion, dependence on foreign oil, or environmental impact of energy use. As a result, buildings and building systems were designed with little regard for energy efficiency or response to the surrounding environment. With the influence of sustainable design principles, buildings are returning to some of the features which were neglected in recent years, such as daylighting and natural ventilation.

In the early 1970s, the political and economic context of building design changed with the oil embargo, increased energy costs, and the realization that we needed to take care of the environment. Most notably, the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) formulated an energy standard for buildings in cooperation with the Illuminating Engineers Society (IES). Over subsequent years, the standard was updated and made more stringent. The full name of the document is ANSI/ASHRAE/IES Standard 90.1, an it is the basis for energy codes.

Sustainable building design was a simultaneous and complementary development, embracing not only energy but a whole host of environmental, health, and productivity issues related to buildings. Most notably, the U.S. Green Building Council (USGBC) launched the LEED™ rating system in 2000 and has over subsequent versions defined and developed sustainability in the building industry. The rating system promotes an integrated design approach that involves the cooperation of architects, engineering, owners, building users, and contractors to produce buildings, which conserve resources, reduce environmental impact, and produce a healthy productive place to work. The Nidus Center, a research laboratory building shown in Fig. 1.3, was part of the LEED™ pilot program.

This chapter defines sustainability qualities in the context of building and building systems design. It also provides tools for evaluating solutions based on qualitative and quantitative criteria. Life-cycle cost analysis is presented using a discounted cash flow methodology. The reader is encouraged to consider not only "hard costs" but also environmental costs and the economics of productivity in the decision process.

1.1 SUSTAINABLE DESIGN

1.1.1 Overview of Sustainability

Sustainability is a concept that applies not only to buildings but also to industry, agriculture, transportation, and all other aspects of societal activity. "Sustainable" can be defined simply as *having an overall beneficial effect on productivity, health, resources, economics, and the environment.* Sustainable design acknowledges responsibility for future as well as current outcomes. Sustainable design decisions are made for their impact not only at the building level but also at the community and global level. Utility, comfort, energy

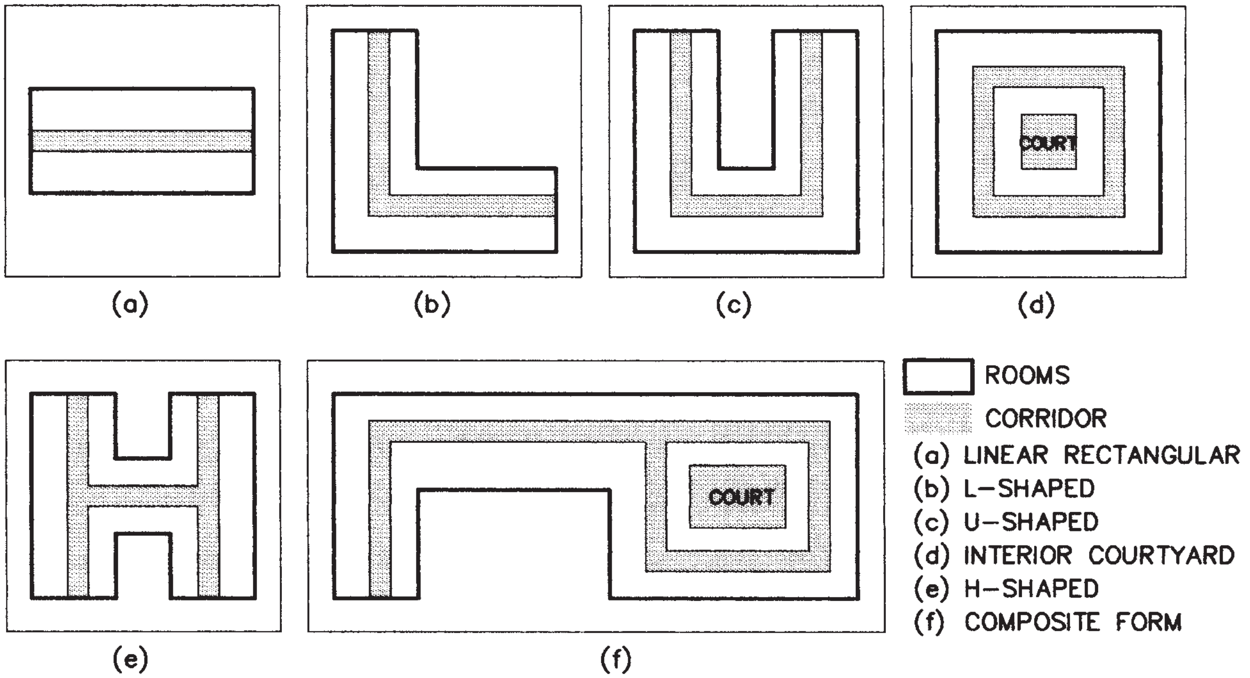


FIGURE 1.1. Common building geometry prior to development of modern M/E systems. With the renewed emphasis on daylight and natural ventilation, these geometries are enjoying a revival.

FIGURE 1.2. Prior to air-conditioning, buildings were equipped with features to take advantage of natural ventilation, such as operable sash, louvered shutters (a), and transom lights (b).

(Courtesy of William Tao & Associates)

FIGURE 1.2. (Continued)

conservation, environmental impact, and appropriate use of technology are basic criteria for mechanical/electrical systems in a sustainable design process.

1.1.2 Design Interactions

Achieving sustainable building solutions requires that many parties work closely together with an understanding of the interactions among building systems and processes. For example, energy usage is affected by architectural form; building materials; lighting; appliances; heating, ventilating, and air-conditioning (HVAC) systems; and even by access to public transportation. There are many participants in the design process who take responsibility for these issues (e.g., architect, lighting designer, owner, consulting engineers, contractors, and suppliers). Too often, each participant makes decisions independent of the others, and opportunities are lost by not understanding the interactions between design factors.

Decisions made by each member of the team will affect systems in which others are also affected. For instance, an architect might design larger windows, which could increase the size of heating and air-conditioning equipment. Or, the lighting designer might design more light fixtures, which would increase the size of air-conditioning equipment.

Interactions also affect health and productivity of building occupants. Daylight and outdoor views, for example, enhance occupants' sense of well-being in buildings, and the effect on performance in the workplace is obvious, though difficult to quantify. Likewise, effective HVAC systems contribute to good indoor air quality to the benefit of occupants' health.

1.1.3 Environmental Impact of Buildings and Building Systems

A building's impact goes beyond the site boundary. Sustainable design must consider how well buildings work to minimize negative environmental impact or even benefit the environment.

Buildings contribute to disruption of storm-water flow, ground erosion, fouling of natural water, light pollution, the growth of landfills from disposal of building materials as construction waste, and, ultimately, demolition. These impacts can be mitigated by good design, and there is potential for well-planned buildings to have zero impact on the environment or even improve the environment.

Buildings account for about 30 percent of overall energy usage in the United States and over 60 percent of electrical usage. This represents not only a depletion of energy resources but also affects the environment by emissions through combustion of fossil fuels, both on the building site and remotely at power-generating stations. Pollution from energy consumption is quantified in Table 1.1.

1.1.4 Water Conservation

Conserving water is a goal of sustainable design. As with most elements of sustainable design, there are economic benefits. Many locations have inexpensive water rates, which alone would not justify significant cost for conservation technology. However, water usage results in sanitary sewer discharge. Sewer charges are generally based on water usage, and are equal or greater in many cases than the water charges. Saving water in buildings will also have a community benefit in reducing the need and cost of constructing, improving, and

FIGURE 1.3. Nidus Center for Scientific Enterprise, which is among the first LEED^{TM} certified buildings, uses principles of sustainable design to conserve resources, reduce environmental impact, and produce a healthy productive place to work. LEED^{TM} (Leadership in Energy & Environmental Design) is a "green" building rating system administered by the U.S. Green Building Council.

(Courtesy of William Tao & Associates)

maintaining water and sewer infrastructure. EPact (Energy Policy Act of 1992) became effective in 1996 to mandate that manufacturers produce conventional fixtures that flow less water. These mandates still apply and have been supplemented by Environmental Protection Agency's (EPA) WaterSense program, which publishes voluntary standards going beyond the mandate. For instance, EPact requires urinals and water closets to use maximum 1.0 and 1.5 gallons per flush, respectively. To be EPA WaterSense Listed, these values must be reduced to 0.5 and 1.28 gallons per flush, respectively. Sensor controls have also become commonplace in building design. Currently, designers are using alternative products on a limited basis, which use even less water, such as waterless urinals, rainwater collection, and composting

Air Pollutants Produced from Energy Conversion

	Air Pollutants Produced, g (lb)		
Energy Converted or Consumed	**CO_2**	**SO_2**	**NO_x**
1 gallon of fuel oil by combustion[a]	10,500 (23.1)	45.0 (0.10)	18.3 (0.04)
1 gallon of gasoline by automobiles[b]	8,500 (18.8)	37.0 (0.08)	15.0 (0.03)
1 pound of coal by combustion[c]	1,090 (2.4)	9.0 (0.02)	4.4 (0.01)
1 therm of natural gas by combustion[d]	6,350 (14.0)	Nil (–)	24.0 (0.05)
1 kWh of electric energy generated by oil[e]	860 (1.9)	3.7 (0.008)	1.5 (0.003)
1 kWh of electric energy generated by gas[e]	635 (1.4)	Nil (–)	2.4 (0.005)
1 kWh of electric energy generated by coal[e]	1,090 (2.4)	9.0 (0.02)	4.4 (0.01)

[a]Calculated by using fuel oil containing 85% carbon and 12% hydrogen, and 7.4 lb/gal.

[b]Calculated by using gasoline mixture of C_8H_{18} and (C_nH_{2n+2}) having 84% carbon and 15% hydrogen, and 6.1 lb/gal.

[c]Calculated by using bituminous coal containing 65% carbon and 3.8% sulfur.

[d]Calculated by using a mixture of methane (CH_4) and ethane (C_2H_6) and 100,000 Btu/therm.

[e]Data from Green Light Program, Environmental Protection Agency.

toilets. These measures will require acceptance by owners and code officials before widespread usage. LEED has also had a conserving impact, granting points to encourage water savings. Water conservation using fixtures certified by EPA Water Sense program is covered in Chapter 10.

1.1.5 Energy Conservation

Sustainable design approaches for energy conservation include:

- Architectural design to limit HVAC loads by methods recommended in Chapter 4
- Effective HVAC delivery systems as described in Chapter 5
- Efficient heating and cooling production and delivery as discussed in Chapters 6, 7, and 8
- Efficient light sources and controls as described in Chapters 17 and 19
- Using renewable energy sources such as solar thermal as described in Chapter 7 and solar photovoltaic as described in Chapter 13

In addition, building owners should be encouraged to use efficient equipment and appliances such as those with Energy Star ratings.

Designers should be cautioned, however, that energy conservation should not be at the expense of comfort or building productivity. Proper ventilation levels, quality lighting, and thermal comfort are essential for building occupants to operate effectively. Buildings and systems that save energy *and* produce a great environment are truly "high performance."

There are many energy technologies vying for use in buildings, and choices among options should be made on value at promoting technology as well as life-cycle economics. Solar collectors installed during the "Energy Crisis" of the 1970s could not be justified economically, but were a valuable technology demonstration to develop systems which might someday be commercially viable (see Fig. 1.4). Some energy technologies are fully mature such as the heat recovery wheel shown in Fig. 1.5. The use of proven technologies that require increased investment should be analyzed by economic methods such as discounted cash flow analysis as described in this chapter.

Energy codes have been enacted based on provisions of ASHRAE Standard 90.1, "Energy Standard for Buildings Except Low-Rise Residential Buildings," which is an industry consensus standard for energy performance of architectural construction, lighting, water heating, and mechanical and electrical equipment. First issued as Standard 90 in 1975, it has evolved to its current version, Std. 90.1-2010, through periodic revisions. LEED's latest version, Version 3, requires that buildings achieve 10 percent or greater energy savings beyond the minimum compliance with the ASHRAE Standard. Points are awarded to encourage higher savings.

1.2 INDOOR ENVIRONMENTAL QUALITY

1.2.1 Components of IEQ

In addition to environmental benefits and resource conservation, sustainable design enhances health, well-being, and productivity of building occupants. These benefits are achieved by several goals of sustainable design:

- Healthful indoor air quality
- Thermal comfort and individual control
- Good lighting
- Connection with the outdoors

FIGURE 1.4. Solar collectors installed after the 1970s oil embargo were an opportunity to explore new technologies, but could not be justified on the basis of economics or life-cycle cost.

(Courtesy of William Tao & Associates)

FIGURE 1.5. Heat recovery wheel allows higher ventilation rates without sacrificing economy of operation, exemplifying sustainable design within economic realities.

(Courtesy of William Tao & Associates)

Combined, these factors contribute to “indoor environmental quality” (IEQ), a term used in the LEED^{TM} rating system, described earlier.

1.2.2 Indoor Air Quality

Indoor air pollution is preventable by good architectural detailing, as shown in Fig. 1.6, effective mechanical systems, and proper maintenance. Indoor air pollution in typical buildings, such as offices, comes from chemicals in finish materials, cleaning products, furniture, and fumes from equipment. In addition, biopollutants such as mold can result if humidity is not properly controlled or if there are moisture problems in building assemblies and systems.

Interior chemical pollution and odors can be diluted to acceptable levels with ventilation by liberal quantities of relatively purer outdoor air. Selecting furnishings, interior finish materials, and cleaning products to be nonpolluting will allow lower ventilation rates and save energy. Using local exhaust over offensive equipment is

FIGURE 1.6. Air intakes for Monsanto Research Center in St. Louis are placed high to avoid street-level air pollution as a measure to improve indoor air quality.

(Courtesy of William Tao & Associates)

also effective at preventing chemicals from entering the larger occupied space.

Building occupants themselves are also sources of pollution. They consume oxygen and emit carbon dioxide and body odors. ASHRAE Standard 62.1-2016, entitled "Ventilation for Acceptable Indoor Air Quality," specifies the amount of outside air needed to cover various levels of occupancy. The amount of outside air required in buildings is based on the nature of building usage, floor area, and density of occupancy.

Condensation in roofs or walls can be a problem. Venting, insulation, and vapor barriers can avoid condensation if properly applied, and HVAC design for proper dehumidification is essential. Interior surfaces of HVAC systems can harbor dust, odors, bacteria, and mold. Filters are porous and microorganisms can breed if they are not changed frequently. Acoustical duct liner is also porous and should be avoided or treated with biocide. Condensate pans in air-conditioning systems are continually moist during hot weather and should drain properly.

Indoor air quality has been correlated with employee productivity. Increasing ventilation rates are reported to render 23–76 percent reductions in the incidence of acute respiratory illnesses. Measured data are also available on the relationship between "sick building syndrome (SBS)" symptoms and worker performance. Workers who reported any SBS symptoms took 7 percent longer to respond in a computerized neurobehavioral test. In another test, workers with symptoms had a 30 percent higher error rate.

One study was performed to determine the effects of ventilation rate on absenteeism. Buildings were classified as moderate ventilation (25 CFM/occ) or high ventilation (50 CFM/occ). Absence rate was 35 percent lower in high-ventilation buildings. Even the moderate ventilation rate cited in the study is higher than rates prescribed by ASHRAE Standard 62.1, indicating potential for improvement in current design practices.

1.2.3 Thermal Comfort

In ASHRAE Standard 55, entitled "Thermal Environmental Comfort Conditions for Human Occupancy," comfort involves factors including temperature, air velocity, and humidity. In general, the standard asserts that these quantities must be maintained within reasonable levels and not allowed to change rapidly.

Basically, the standard identifies conditions of temperature and humidity that 80 percent of research subjects will find acceptable. The obvious corollary is that 20 percent may not find conditions acceptable. This implies that there will be a greater likelihood of satisfying everyone if individual temperature controls are provided. Space heaters and thermostat tampering demonstrate the desire for individual control.

Air temperature has been documented to affect worker performance. Small differences in temperature have been reported to have 2–20 percent performance impact in tasks such as typewriting, learning performance, reading speed, multiplication speed, and word memory.

1.2.4 Individual Control

HVAC systems can be designed that offer opportunities for individuals to control their local thermal environment. This simple notion is generally ignored in typical institutional buildings designed with the goal of providing uniform temperature control.

Allowing greater personal control of indoor environments, and allowing temperatures to fluctuate with outdoor conditions, could improve perceived comfort and reduce energy consumption. Individuals will tolerate a wider range of thermal conditions if they have control over their environment, such as operable windows or the ability to adjust airflow. The effect of individual control on productivity has been documented. Providing ±5°F of individual temperature control has been claimed to increase work performance by 3–7 percent.

Individual control is not practical with many HVAC systems. There are, however, several practical options for giving control to individuals. A few furniture manufacturers can integrate local control features into their workstations, which allows the occupant to adjust the quantity and direction of airflow (see Fig. 1.7). Ironically, table fans used before the advent of air-conditioning are similar in concept.

Other options are to deliver air through floor registers, which allows occupants to adjust the airflow from nearby outlets. Operable windows controlled by occupants are appropriate in some climates and/or some seasons. One analysis revealed that occupants of buildings using central HVAC systems were much more sensitive to temperature variation than occupants of buildings that have operable windows. Having control results in higher perceived comfort. People might even be invigorated by the variability of temperature in naturally ventilated buildings (see Fig. 1.8).

Integrating operable windows with conventional HVAC control systems is a challenge. Typical systems, for instance, might place multiple rooms on the same thermostat. If the occupant with the thermostat opens the window, control will be lost for the other spaces. Other potential problems include possible freezing from cold air through windows left open. Security and infiltration of pollen and dust are other problems which need to be considered. Despite these issues, operable windows are highly desired by building occupants and well worth the effort to work out problems.

1.2.5 Superior Lighting Systems

Lighting affects occupant performance and quality of space as well as energy consumption. Uniform illumination by recessed fluorescent fixtures is the most common lighting solution for work spaces, and often results in glare, shadows, and reflections in computer screens. Indirect (all light up toward ceiling) or semi-indirect (a portion up, a portion down) lighting is an alternative, which produces better visibility of tasks at lower levels of illumination. Indirect lighting is theoretically less efficient than direct lighting due to considerable light being absorbed at the ceiling surfaces. However, indirect light is more uniform, eliminates glare, results in less shadows, and can be designed at lower light levels to produce a better environment at lower energy cost (see Fig. 1.9).

FIGURE 1.7. Personal cooling outlet (left) gives individual personnel control of climate at workstation.

(Courtesy of William Tao & Associates)

FIGURE 1.8. Variations in environment are well tolerated when people have a choice; these shoppers prefer an open-air market to the modern climate-controlled grocery store.

(Courtesy of William Tao & Associates)

Daylight Sustainable lighting strategies generally include daylighting. The challenge in using daylight is to control the glare, avoid thermal discomfort, and minimize HVAC loads. Energy interactions must be considered carefully. While one would expect higher air-conditioning loads due to extra window or skylight area, the extra load may be more than offset by reducing the heat gain from artificial lighting which can be deactivated.

No one would question that an attractive, visually interesting environment contributes to occupant satisfaction and higher levels of productivity. Having an outdoor view or a source of natural light is desirable (see Fig. 1.10). The best publicized study on the effects of daylight and view was performed by the Pacific Gas and Electric Company. The following is quoted from their executive summary:

> Controlling for all other influences, we found that students with the most day lighting in their classrooms progressed 20% faster on math tests and 26% on reading tests in one year than those with the least. Similarly, students in classrooms with the largest window areas were found to progress 15% faster in math and 23% faster in reading than those with the least.

FIGURE 1.9. Semi-indirect lighting in this research laboratory is not only comfortable for occupants but also illuminates building services in the exposed ceiling, resulting in better maintenance and safety.

(Courtesy of William Tao & Associates)

FIGURE 1.10. Light well in this classroom building allows daylight to the interior and gives occupants a sense of outdoor weather and time of day.

(Courtesy of William Tao & Associates)

> And students that had a well-designed skylight in their room, one that diffused the daylight throughout the room and which allowed teachers to control the amount of daylight entering the room, also improved 19–20% faster than those students without a skylight. We also found another window-related effect, in that students in classrooms where windows could be opened were found to progress 7–8% faster than those in rooms with fixed windows. This occurred regardless of whether the classroom also had air conditioning. These effects were all observed with 99% statistical certainty.

1.2.6 Connection with Outdoors

Daylight, views outside, natural ventilation, and temperature variation are ways to give building occupants a sense of connection with the outdoors. Occupants feel better and perform better when they have a sense of time of day and outside weather. These connections need not be exaggerated by using large windows, large skylights, or large ventilation openings. Effective placement is more critical in achieving success as shown in Fig. 1.11.

1.3 COMMISSIONING

1.3.1 Scope of Commissioning

Commissioning is an essential feature of sustainable design. It is a prerequisite for LEED^{TM} certification and highly recommended for any new building. Commissioning can generally be defined as the process of proving that systems will operate as intended and implementing adjustments necessary to achieve that goal. Typically, the commissioning process would include the following steps:

1. Review system criteria, including design temperatures.
2. Review and assure that design (load calculations, equipment selections) is able to achieve criteria.

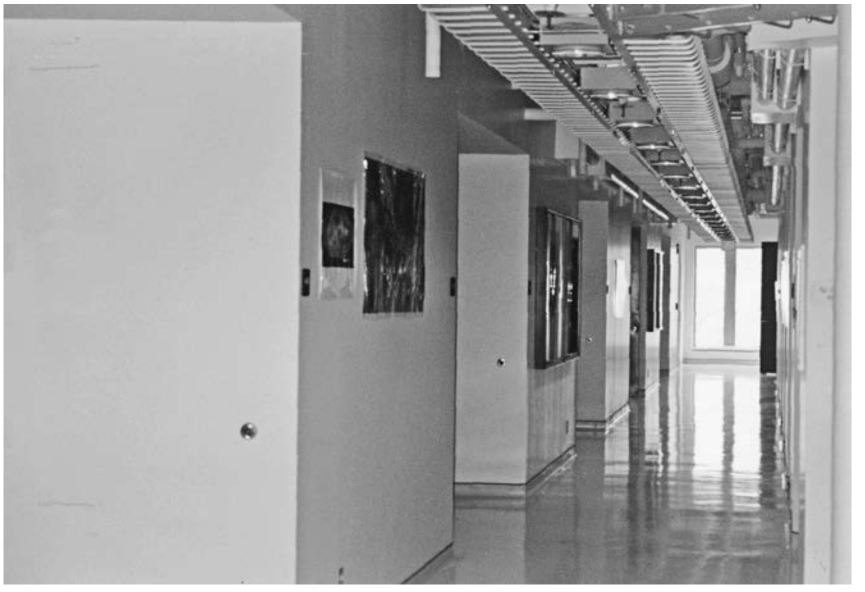


FIGURE 1.11. A simple window at the end of this laboratory corridor provides daylight and view. Lights are rarely turned on during the day in this space. *(Courtesy of William Tao & Associates)*

3. Review plans and specifications for consistency with design.
4. Observe construction to assure that equipment and systems are installed per plans and specifications.
5. Verify that contractor has performed prefunctional checkout of systems and equipment (e.g., proper wiring connections, clean filters).
6. Measure system component performance, review test results.
7. Verify control sequences (e.g., thermostat call for cooling starts compressor).
8. Document that these procedures have been performed along with their outcome.
9. Make sure that appropriate owner's staff are trained in operation of the systems.
10. Verify that operating manuals are turned over to the owner.
11. Follow up during the first year of operation to check seasonal performance and address any owner concerns.

Most of the commissioning scope can be performed by the design and construction team; however, the tasks involving review of design are generally done by a third-party commissioning agent.

1.3.2 Benefits of Commissioning

Making sure that systems operate properly will produce better comfort and save energy. In addition, commissioning reduces the need for warranty work and callbacks to adjust systems during the first year. The commissioning report and associated documentation also provide a baseline of performance for tracking the condition of systems and equipment over the life of the building. Commissioning also aids in organizing maintenance materials (manuals and training) for ongoing use by the building's operations staff.

1.3.3 Range of Applications

The scope of commissioning will depend on how simple or complicated the systems are and on the relative importance of proper system operation. A shortened commissioning process might be quite satisfactory for a small commercial building with simple heating and cooling equipment. If system performance is critical, the commissioning process will be extensive. Examples of buildings requiring emphasis on commissioning include museums, data centers, and correctional facilities.

Museums require that systems operate reliably to produce a precision environment with respect to temperature and humidity. Tight control is needed to prevent damage to valuable artifacts. In most climates, systems have extra components and controls for humidification and dehumidification. Systems must be demonstrated to operate properly before valuable artifacts are moved into the building and placed at risk if systems do not operate properly.

Many enterprises rely on continuous operation of data centers for business-critical and safety-critical functions, such as market transactions, air traffic control, and reservations. Systems are designed with redundancy in the event of failure and must transfer load to backup equipment without interruption of service. Commissioning is essential to test failure modes as well as normal operations.

Correctional facilities may have simple HVAC systems, but they are located in facilities that have limited access for correcting systems problems once the facility is put in service. For this reason, a rigorous commissioning process is necessary to make sure the systems are complete and to minimize callbacks. Other systems such as security and alarm require extensive commissioning due to the critical nature of their performance and their complexity in comparison with similar systems for other buildings.

Buildings with less critical functions can generally suffice with the typical start-up and checkout procedures used by conscientious contractors based on manufacturers' recommendations for particular pieces of equipment. For many simple buildings, ongoing maintenance is outsourced, and there is no need for the owner to receive training or operating and maintenance documentation.

1.3.4 Checklists and Forms

Forms are used in commissioning to assist field personnel through the checkout procedure and to record and sign off on results. In most instances, the equipment manufacturers' start-up procedures and forms will be satisfactory with minor modifications for use in commissioning of individual equipment items. Commissioning at the system level (as opposed to individual equipment checkout) requires procedures and checklists customized for the particular system. Control sequences in the specifications or from the control subcontractor's shop drawing submittals are generally the basis for producing system commissioning procedures and forms. Websites of various commissioning organizations and equipment vendors are good sources of standard forms that can be customized for particular projects.

1.4 EVALUATING DESIGN OPTIONS

1.4.1 Subjective Viewpoints

System quality cannot be assessed without defining criteria. Criteria will vary depending on viewpoint. For instance, a contractor might assess a design solely on the basis of ease of construction, whereas a CFO might consider cost most important, and the director of a physical plant might look more closely at maintenance issues. The purpose of the building must also be considered. A developer-built speculative office building might be designed with nondurable, low-cost materials, meet budget, and be economically feasible; whereas a corporate headquarters office building might command a higher level of quality. Building life expectations are also important. For instance, a building for a 5-year research program need not be equipped with 20-year life systems, whereas a long-term, institutional building might be designed for 50+-year systems.

1.4.2 Qualitative Versus Quantitative Analysis

The goal of sustainable design is that buildings be healthy, pleasant, and productive and that they minimize negative impacts on the local and global environment. Achieving this goal with the best solution requires that many factors be considered in the process. Some factors can be quantified economically and some can only be judged qualitatively based on relative importance.

1.4.3 Decision Matrix Method

The decision matrix is a method for evaluating criteria difficult to quantify. The decision matrix can be used to supplement life-cycle cost analyses and weigh options qualitatively. Decision matrix forces the decision makers to assess what is important to them for defining a successful outcome.

A sample decision matrix analysis is shown in Table 1.2. The analysis includes ranking on factors and on qualitative factors for which precise economic quantification is difficult, especially in the early phases of design, when making decisions is most important. The key feature of the matrix method

Decision Matrix Method

A. How a Corporate Owner Might Think About His Options for HVAC of an Office Building

		VAV/Reheat		VAC/Convectors		VAV/Dual Duct		Multizone		VAV/FTU		Fancoils	
		Score	Weighted	Score	Weighted	Score	Weighted	Score	Weighted	Score	Weighted	Score	Weighted
Criteria	**Weight**												
Comfort	8	5	30	8	42	5	30	5	30	8	48	7	42
Flexibility	6	10	60	7	42	8	48	1	4	8	48	7	42
Initial cost	3	10	30	8	24	6	18	4	12	7	21	6	18
Energy consumption	6	7	42	8	48	7	42	7	42	9	54	9	54
Ease of maintenance	6	7	42	8	48	9	54	10	60	6	36	5	30
Longevity	6	9	54	7	42	9	54	9	54	6	36	5	30
Acoustics	5	8	40	8	40	8	40	8	40	5	25	5	25
Total score			**299**		**308**		**296**		**252**		**284**		**255**
% score (normalized)			97%		100%		96%		82%		92%		85%
Grade			A		A+		B		F		B		C

(Continued)

Continued

B. How a Developer Might Think About His Options for HVAC of an Office Building

Criteria	Weight	VAV/Reheat Score	VAV/Reheat Weighted	VAC/Convectors Score	VAC/Convectors Weighted	VAV/Dual Duct Score	VAV/Dual Duct Weighted	Multizone Score	Multizone Weighted	VAV/FTU Score	VAV/FTU Weighted	Fancoils Score	Fancoils Weighted
Comfort	3	5	15	8	21	5	15	5	15	8	24	7	21
Flexibility	3	9	30	7	21	8	24	1	4	8	24	7	21
Initial cost	10	9	100	8	80	6	60	4	40	7	70	6	60
Energy consumption	2	7	14	8	16	7	14	7	14	9	18	9	18
Ease of maintenance	2	7	14	8	16	9	18	10	20	6	12	5	10
Longevity	2	9	18	7	14	9	18	9	18	6	12	5	10
Acoustics	5	8	40	8	40	8	40	8	40	5	25	5	25
Total score			**213**		**211**		**189**		**151**		**185**		**165**
% score (normalized)			100%		97%		87%		69%		85%		76%
Grade			A+		A–		B		D		B		C

is inclusion of weighting factors which allow quantitative inclusion of real, albeit qualitative, criteria in a comprehensive comparison among alternatives.

This process involves the following steps:

- Define important criteria.
- Score options 1–10 on these criteria.
- Assign weight according to perceived importance of each criterion.
- Multiply weight × score for weighted score.
- Add weighted scores for total score.
- Normalize scores as percent of score of highest ranking option.
- Grade "on a curve."

Note that different constituencies may have various opinions on weighting of criteria, exemplified here by an owner (A) and a developer (B).

1.4.4 Economic Evaluation

The basis for making good business decisions is economics, and two methods are commonly used to evaluate options. They are *simple payback period* and *life-cycle cost analysis*. Payback analysis is a simple tool often used to screen options. It generally considers only the initial cost of implementing the idea and recurring savings in energy. The cost of maintenance and financing are sometimes neglected, and the interactions with other building systems are generally ignored. Life-cycle cost analysis includes maintenance and financing cost, but usually leaves out many important parameters. Virtually never do these analyses include environmental or the impact on occupant productivity. These are considered "soft costs," which are sometimes beyond the realm of defensible quantification. Nonetheless, soft costs can be the most important factors in deciding among options for design.

1. *Payback analysis.* Despite its limitations, payback analysis is often used to evaluate and compare options. Given options with different initial cost and different operating costs, the simple payback period can be calculated to determine which of the options will recoup initial cost most quickly. Simple payback period is calculated by the following equation:

$$\text{Payback Period} = \text{Extra Cost}/\text{Savings}$$

where
Payback Period = the time required for savings between two options to equal the difference in cost
Extra Cost = the difference in initial cost between the two options
Savings = the annual difference in operating cost, generally including utilities and maintenance

Example 1.1 What is the payback period for a hypothetical energy-saving device in a manufacturing plant? It costs \$20,000 to install, lasts 5 years, and saves \$7500 per year in utilities, consisting of 15,000 therms* of gas at \$0.50 per therm. The device will require \$500 per year for maintenance and repairs. These costs are assumed for first year of operation.

* A therm is 100,000 Btu of heating; this unit is used by utilities for billing gas usage.

Solution: Divide $20,000 by net annual savings, which are $7500 utilities, less $500 maintenance, or $7000.

Payback Period = $20,000 / ($7500 − $500) = 2.9 years

Example 1.2 What is the payback period for installing an improved version of the hypothetical energy-saving device? It is constructed of more durable materials, which will last 10 years and costs $30,000 to install. Savings are identical at $7500 per year in utilities, consisting of 15,000 therms of gas at $0.50 per therm and requires $500 per year for maintenance and repairs. These costs are assumed for first year of operation.

Solution: Divide $30,000 by net annual savings, which are $7500 utilities, less $500 maintenance, or $7000.

Payback Period = $30,000/($7500 − $500) = 4.3 years

Results of payback analysis can be deceiving. A low-investment, quick-payback option (Example 1.1) might appear superior to a higher investment with higher savings and a longer payback (Example 1.2). Over the economic life of the option, the higher investment could produce superior results despite a longer payback period.

2. *Life-cycle cost analysis.* Life-cycle cost analysis is performed by listing the cash flows associated with an option over a given period defined as the life cycle. Initial cost can be accounted for as a single outlay at year zero. Cost could also be treated as payments to amortize a loan or as yearly depreciation of the asset for accounting and tax purposes. The single outlay method is used for simplicity to illustrate the process.

Economics of alternative decisions are best demonstrated by life-cycle cost analysis. Life-cycle cost analysis is an appropriate tool to compare options on the basis of economics, but is often performed solely on the basis of those criteria that are easiest to document. Cost of construction, financing, maintenance, and utilities can be estimated and have a reportable impact on balance sheets and income statements. Life-cycle cost analysis can be more effective if owners, engineers, and architects are willing and convinced to include cash flows attributable to effects on productivity in the workplace. Good indoor environmental quality results in real economic benefit from productivity. These benefits should be included in life-cycle cost analysis.

During the first year and subsequent years of the life cycle, there will be expenses for utilities, maintenance, and repairs. These costs will likely escalate over time.

All future cash flows must be brought back to current value using a discount rate. The discount rate represents the cost of money or foregone return on investment. Using the "foregone return" logic, a dollar invested today at, say, 6 percent investment return will be worth more in the future. Its value will be escalated by 6 percent per year. Conversely, a dollar in the future is worth less today, its value being "de-escalated" or discounted by 6 percent per year. Discount rate can be also considered the expected investment return.

The following examples illustrate life-cycle cost for the same energy-saving device described in the previous section on payback.

Example 1.3 What is the life-cycle cost for the energy-saving device described in Example 1.1. Recall that the hypothetical energy-saving device is being considered for a manufacturing plant. It costs $20,000 to install, will last 5 years, and will save $7500 in utilities during the first year of operation, consisting of 15,000 therms of gas at $0.50 per therm. The device will require $500 per year for maintenance and repairs during the first year. Assume that energy cost will escalate at 3% per year and that maintenance/repair cost will also escalate at 3% per year. Assume also that the owner expects a 15% rate of return for investment.

Solution:

Life-Cycle Cost Analysis: $20,000 Energy-Saving Device, 5-Year Life, 3.6-Year Simple Payback

Life cycle of investment (years)	5
Installation cost	20,000
First-year energy saving (utility rates first year)	7500
Annual maintenance/repair cost (first-year value)	500
Energy escalation rate	3%
Repair and maintenance escalation rate	3%
Discount rate (expected investment return)	15%

Year	Cash Flows in Year of Occurrence				
	Install Cost	Energy Saving	Repair and Maintenance	Total Annual Cash Flow	Present Value Total Annual
0	(20,000)	-	-	(20,000)	(20,000)
1	-	7,500	(500)	7,000	6,087
2	-	7,725	(515)	7,210	5,452
3	-	7,957	(530)	7,426	4,883
4	-	8,195	(546)	7,649	4,373
5	-	8,441	(563)	7,879	3,917
Net Present Value of Life-Cycle Cash Flows ($)					**4,712**

Example 1.4 What is the life-cycle cost of the more durable device described in Example 1.2 that costs $30,000 to install. It will last 10 years and will save $7500 in utilities during the first year of operation, consisting again of 15,000 therms of gas at $0.50 per therm. The device will require $500 per year for maintenance and repairs during the first year. Assume that energy cost will escalate at 3% per year and that maintenance/repair cost will also escalate at 3% per year. Assume also that the owner expects a 15% rate of return for investment.

Solution: Note that the $20,000 option in Examples 1.1 and 1.3 has a shorter payback than the $30,000 option in Examples 1.2 and 1.4; however, the life-cycle savings of the $30,000 option are superior.

Installing the hypothetical $30,000 energy-saving device will have a net present value of $8,955, which is considerably higher than the $4,712 for the $20,000 option.

Life-Cycle Cost Analysis: $30,000 Energy-Saving Device, 10-Year Life, 5.5-Year Simple Payback

Life cycle of investment (years)	10
Installation cost	30,000
First-year energy saving (utility rates first year)	7,500
Annual maintenance/repair cost (first year)	500
Energy escalation rate	3%
Repair and maintenance escalation rate	3%
Discount rate (expected investment return)	15%

Year	Cash Flows in Year of Occurrence				
	Install Cost	Energy Saving	Repair and Maintenance	Total Annual Cash Flow	Present Value Total Annual
0	(30,000)	-	-	(30,000)	(30,000)
1	-	7,500	(500)	7,000	6,087
2	-	7,725	(515)	7,210	5,452
3	-	7,957	(530)	7,426	4,883
4	-	8,195	(546)	7,649	4,373
5	-	8,441	(563)	7,879	3,917
6	-	8,695	(580)	8,115	3,508
7	-	8,955	(597)	8,358	3,142
8	-	9,224	(615)	8,609	2,814
9	-	9,501	(633)	8,867	2,521
10	-	9,786	(652)	9,133	2,258
Net Present Value of Life-Cycle Cash Flows ($)					**8,955**

1.4.5 Considering Environmental Emissions in Life-Cycle Analysis

Environmental factors such as "carbon footprint" can be considered in life-cycle cost analysis. Table 1.1 lists pollutants attributable to building energy usage. Various agencies have assigned social costs to the emission of CO_2, ranging from \$15 per ton to upwards of \$50 per ton. These costs are not directly assessed to the building owner who uses the energy, but are sometimes considered in life-cycle cost analysis by institutions which desire to take a more comprehensive view of their energy operational responsibilities.

Example 1.5 shows a life-cycle cost analysis of the \$30,000 investment from Example 1.4 including a value assigned to CO_2 emissions reduction.

Example 1.5 The firm's CEO knows that his board of directors is socially conscious and would appreciate inclusion of the environmental impact of the option. From Table 1.1, we find that 14 lbs. of CO_2 emission will be eliminated for every therm saved. The board subscribes to a governmental agency's opinion that the social cost of CO_2 emission is \$25/ton, resulting in a "carbon saving" of (15,000 therms × 14 lbs)/(2000 lbs./ton) = 105 tons per year; 105 tons × \$25/ton = \$2625.

What is the life-cycle cost of the alternative energy-saving device from Example 1.4 if a value is assigned to reduction of CO_2 emissions?

Carbon cost is assumed to escalate at 3% per year in the analysis presented below:

Solution: Note that the net present value of cash flow is almost three times the value when only energy savings are considered. Considering emissions cost can have a significant effect on the decision process who consider their effect on the environment.

Life-Cycle Cost Analysis: \$30,000 Energy-Saving Device, 10-Year Life, Including Productivity and Environmental Values

Life cycle of investment (years)	10
Installation cost	30,000
First-year energy saving (utility rates first year)	7,500
Annual maintenance and repair	500
Cost assigned to CO_2 emissions reduction (gain)	2,625
Energy escalation rate	3%
Repair and maintenance escalation rate	3%
CO_2 emissions cost escalation rate	3%
Discount rate	15%

	Cash Flows in Year of Occurrence					
Year	Install Cost	Energy Saving	Repair and Maintenance	Value of CO_2 Reduction	Total Annual Cash Flow	Present Value Total Annual
0	(30,000)	-	-	-	(30,000)	(30,000)
1	-	7,500	(500)	2,625	9,625	8,370
2	-	7,875	(515)	2,704	10,064	7,610
3	-	8,269	(530)	2,785	10,523	6,919
4	-	8,682	(546)	2,868	11,004	6,292
5	-	9,116	(563)	2,954	11,508	5,722
6	-	9,572	(580)	3,043	12,036	5,203
7	-	10,051	(597)	3,134	12,588	4,732
8	-	10,553	(615)	3,228	13,167	4,304
9	-	11,081	(633)	3,325	13,773	3,915
10	-	11,635	(652)	3,425	14,408	3,561
Net Present Value of Life-Cycle Cash Flows (\$)						**26,628**

1.4.6 Energy Usage in Perspective with Other Operating Expenses

This section examines the components of building operating cost and the relative importance of energy in comparison with other costs associated with building operations, especially labor costs for personnel working in commercial buildings. An office building is used for example. Office buildings in temperate climates use energy for HVAC, water heating, lighting, office appliances, and vertical transportation. The table here shows how a typical low-rise building might use energy for these functions.

Energy Use	Percent
HVAC	53
Hot water	1
Lighting	28
Appliances	18

Generally, a Midwest office building will experience energy bills of $1.50 to $2.50 per year for gas and electric.

To place the cost of energy in perspective, here is an example of operating costs for a typical office building in a temperate climate:

Facility Expenses, Typical Office Building

Expense Component	Expense (%)	$ per Year
Investment return	59	8.35
Repairs and maintenance	5	0.73
Preventative maintenance	4	0.60
Janitorial	7	1.01
Site maintenance	0	0.06
Gas	**1**	**0.19**
Electric	**13**	**1.84**
Water	1	0.09
Sewer	0	0.05
Environmental	1	0.13
Life safety	1	0.11
Security	5	0.73
Space planning	2	0.29
Total facilities expense	**100**	**14.18**

Energy (highlighted in the table) is about $2.00/ft^2 per year, representing less than 15 percent of the overall cost of owning and operating a facility, exclusive of personnel costs. This value might be higher or lower depending on climate and utility rates, but the general relationship will be fairly consistent with these values.

1.4.7 Energy Cost Compared with Personnel Cost

A typical office building will likely have an occupancy density of 200 gross square feet per workstation, including circulation, toilets, and lobby. Personnel cost can be expressed in terms of $/ft^2 per year for comparison with other operating costs, including energy:

Analysis of Personnel Cost in $/ft^2 per Year

Occupancy density	200 ft^2/employee
Salary (example)	$50,000/yr
Fringes @ 30%	$15,000/yr
Personnel annual cost	$65,000/yr
Personnel annual cost per ft^2	$325 ft^2/yr

1.4.8 Economics of Productivity and Energy

Relative to energy cost at $2.00/ft^2 per year, employee cost is a very large number! Energy conservation is a key issue in sustainable design, but considering the purpose of buildings, saving energy at the expense of occupant well-being and performance is not advisable. Here are a few arithmetic scenarios to make the point.

Suppose that a facility manager changes thermostat set points and reduces lighting levels to effect a 20 percent reduction in utility bills. Savings would be about $0.40/ft^2 per year. Suppose further that these reductions in building service quality reduced employee productivity by 1 percent. The loss in productivity would be 1 percent of $325/ft^2 per year, or $3.25. The energy savings of 20 percent, or $0.40/ft^2 per year, sound impressive and would likely win praise from the CFO, but such a program is not a good idea if there are harmful side effects on productivity.

Conversely, suppose that a facility manager installs better temperature controls and increases lighting levels at the expense of a 20 percent *increase* in utility bills. Extra cost would be about $0.40/ft^2 per year. Suppose further that these improvements in building service quality increased employee productivity by 1 percent. The gain in productivity would be 1 percent of $325/ft^2 per year, or $3.25. Obviously, the extra cost for energy, though considerable, would be a great investment considering the beneficial side effects on productivity.

These examples are not intended to detract from the importance of energy conservation, but rather to point out that energy should be used or conserved with an eye toward *overall* building performance. A high performance building uses energy effectively and enables high performance personnel.

1.4.9 Considering Personnel Productivity in Life-Cycle Analysis

Example 1.6 shows a life-cycle cost analysis of the $30,000 energy investment from Example 1.4, including an assessment for productivity changes on the plant floor.

Example 1.6 At the last minute the plant foreman was told about the installation of the energy-saving equipment described in the preceding example problems. He was pleased with the potential energy savings and enthusiastic about the reduction in carbon footprint. Unfortunately, he was familiar with a similar installation in a competitor's shop and knows that the device will cause a decrease in temperature and considerable breezes on the plant floor. The foreman of the plant estimates that this will cause extra "warm-up" breaks during the winter for the plant's 20 $15/hour fork truck drivers. Assuming a loss of half hour per day for 12 cold weather weeks (60 days), the productivity impact will be $9000 annual loss.

Productivity cost is assumed to escalate at 3% per year in the analysis presented below:

Solution:

Life-Cycle Cost Analysis: $30,000 Energy-Saving Device, 10-Year Life, Including Productivity as well as Environmental Values

Life cycle of investment (years)	10
Installation cost	30,000
First-year energy saving (utility rates first year)	7,500
Annual maintenance and repair	500
Productivity loss	9,000
Cost assigned to CO_2 emissions reduction (gain)	2,625
Energy escalation rate	3%
Repair and maintenance escalation rate	3%
Labor (productivity) escalation rate	3%
CO_2 emissions cost escalation rate	3%
Discount rate	15%

	Cash Flows in Year of Occurrence						
Year	**Install Cost**	**Energy Saving**	**Repair and Maintenance**	**Value of CO_2 Reduction**	**Productivity Gain (Loss)**	**Total Annual Cash Flow**	**Present Value Total Annual**
0	(30,000)	-	-	-	-	(30,000)	(30,000)
1	-	7,500	(500)	2,625	(9,000)	625	543
2	-	7,875	(515)	2,704	(9,270)	794	600
3	-	8,269	(530)	2,785	(9,548)	975	641
4	-	8,682	(546)	2,868	(9,835)	1,170	669
5	-	9,116	(563)	2,954	(10,130)	1,378	685
6	-	9,572	(580)	3,043	(10,433)	1,602	693
7	-	10,051	(597)	3,134	(10,746)	1,842	692
8	-	10,553	(615)	3,228	(11,069)	2,098	686
9	-	11,081	(633)	3,325	(11,401)	2,372	674
10	-	11,635	(652)	3,425	(11,743)	2,665	659
Net Present Value of Life-Cycle Cash Flows ($)							**(23,457)**

This hypothetical example shows that so-called "soft costs" can make a difference in the economic performance of concepts. Inclusion of even small changes in productivity of building occupants can have a very large effect on the decision process. In this case loss of productivity makes the energy saving feature ill-advised.

QUESTIONS

1.1 What are the benefits of buildings with shallow floor depths?

1.2 How much CO_2 will be liberated to the atmosphere in a year's time due directly to a lighting system consuming 300,000 kWh per year?

1.3 If a corporation is concerned with its carbon footprint and accepts a value of $25 per ton to account for societal costs, what is their perceived economic impact of this much CO_2?

1.4 What will the relative impact be on CO_2 for heating and for cooling, assuming the buildings in Question 1.1 are located in the Midwest (hot summers, cold winters)?

1.5 How does "sustainable" design differ from energy-effective design?

1.6 What factors should the architect and engineer consider to produce a high performance environment for building occupants?

1.7 How does saving energy help to protect the environment?

1.8 What is the role of maintainability in sustainable buildings?

1.9 How could building site selection affect the environment?

1.10 What factors should interior designers consider in terms of indoor air quality? Architects? HVAC engineers?

1.11 What design features would you suggest to allow personal climate control in a single-story residence? A high-rise office building? A classroom building?

1.12 Compare the importance of commissioning for a data center versus a classroom building.

1.13 What sustainable design issues should architects consider in deciding window materials and locations?

1.14 Name a few quantitative factors involved in the analysis of a building HVAC system. Name a few qualitative factors.

1.15 Is there ever a time when energy conservation is unwise? If so, give examples.

1.16 Prepare three decision matrices to evaluate operable windows versus fixed windows in an office building. Use the process described in Section 1.4.3. Fill out the matrices as an occupant, a maintenance staffer, and a building owner.

1.17 Will a commercial building developer use a higher or lower discount rate than an institutional building owner? Why?

1.18 An energy conservation option has a first cost of \$50,000. It requires \$4000 per year maintenance and saves \$10,000 per year in utilities. What is the simple payback period for the option?

1.19 The system in Question 1.18 will last 15 years with no salvage value. What is the 15-year life-cycle cost assuming energy cost escalation of 4% annually, maintenance cost escalation of 2% annually, and a 5% discount rate? What if the discount rate is 15%?

1.20 Assume the option in Question 1.18 is installed in a building with 200 occupants, with average personnel cost of \$60,000 per year. If the device interferes with temperature control, resulting in a 2% decrease in productivity, what would the simple payback be?

1.21 What would the payback be if the option in Question 1.18 improved temperature control and resulted in a 2% increase in productivity?

1.22 Calculate the life-cycle costs for the two cases (2% decrease, 2% increase in productivity) using data from Questions 1.19 and 1.20 for a 5% discount rate and a 15% discount rate.

HVAC FUNDAMENTALS

The initial chapters of this book deal with heating, ventilating, and air-conditioning (HVAC) systems. HVAC systems use and convert energy and move fluids to make buildings provide thermal comfort and healthy air quality. Fig. 2.1 illustrates that air and water are conducted through ducts and pipes to accomplish the task. A basic understanding of fluid flow, energy forms, and conversion factors is essential to the study of HVAC systems along with other mechanical and electrical (M/E) systems.

2.1 BASICS OF ENERGY AND POWER

M/E systems use and convert energy and move fluids to make buildings habitable and functional. Energy forms applicable to building systems include thermal energy, electricity, mechanical energy, and chemically stored energy (fuels).

Thermal energy is measured in British thermal units (Btu). A Btu is the amount of heat required to raise 1 lb of water 1°F. Stated another way, if we heat 1 lb of water (about 1 pint) 1°F, the water will have absorbed 1 Btu. If we heat a pound of water 2°F, we will need 2 Btu. Pound for pound, water will absorb much more heat than most other materials for a given temperature rise. Only 0.156 Btu will be necessary to raise 1 lb of concrete 1°F. If we normalize the heat-absorbing capacity of water at 1.0, the heat capacity (C) of concrete will be 0.156. These relationships can be combined into the following equation:

$$q = M \times C \times \Delta T \tag{2.1}$$

where

q = heat absorbed (or released) (Btu)
M = mass (lb)
C = heat capacity (often called "specific heat") (Btu lb °F)
ΔT = temperature increase or decrease, °F

The quantity C, heat capacity or specific heat, is listed for many common materials in Table 2.1.

Example 2.1 A 10'-by-10' concrete floor is 8" thick. If the floor is warmed by the sun to 80°F during the day and cools to 70°F overnight, how much heat is stored and released by the floor on a daily basis?

Solution: The specific heat of concrete is 0.21 Btu per lb °F. The density of concrete is approximately 144 lb/ft^3. Heat storage is calculated as follows:

$$\begin{aligned} Q &= M \times C \times \Delta T \\ &= 144 \times (10 \times 10 \times 8/12) \times 0.21 \times (80 - 70) \\ &= 20{,}200 \text{ Btu} \end{aligned}$$

The words "energy" and "power" are often used interchangeably, but there is an important distinction between the two. Energy is a quantity, such as heat; power is the rate at which the quantity is transferred or used. Table 2.2 shows the forms of energy and power, their units of measure, and conversion factors.

The unit of *energy* for heat is the Btu. The unit of *power* for heat will be Btu per hour, abbreviated Btuh. This unit is used in quantifying the amount of heating gained or lost by a structure (load) and the amount of heating or cooling capacity required by equipment to offset the heat or load.

For all forms of energy the following equation will apply, but units will depend on energy form:

$$\text{Power} = \text{Energy}/\text{time}$$

or

$$\text{Energy} = \text{Power} \times \text{time} \tag{2.2}$$

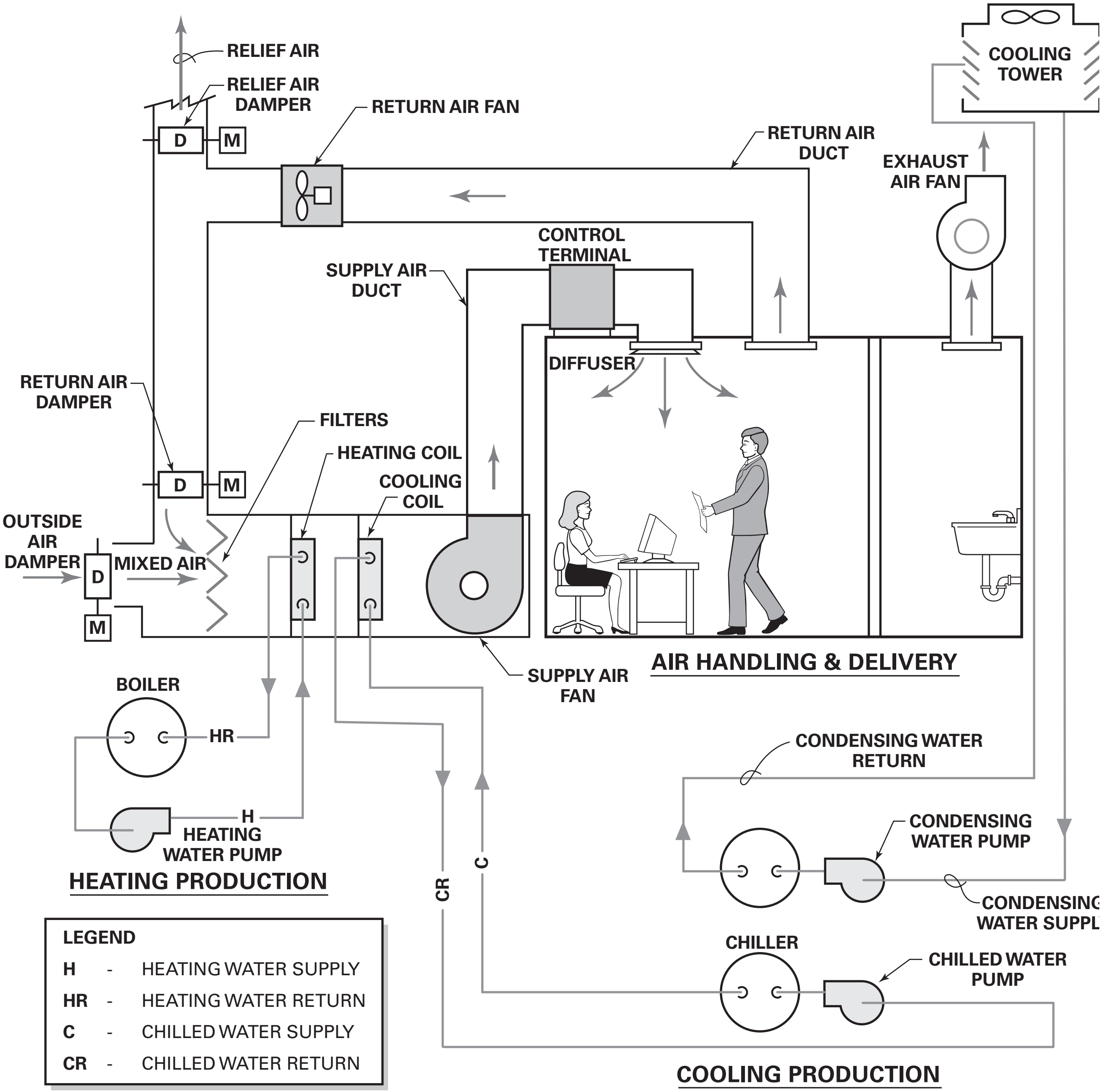


FIGURE 2.1. Components of a large HVAC system (based on hot–chilled water system).

Heat Capacities of Common Materials

Material	Density lb/ft^3	Heat Capacity Btu/°F lb
Water	62.4	1.0
Wood	45	0.57
Foam insulation	2.5	0.34
Air	0.075	0.24
Concrete	144	0.21
Steel	489	0.12

Forms and Units of Energy and Power

Energy Form	Unit of Measure: Energy	Unit of Measure: Power	Conversion to Btu
Heat	British thermal unit (Btu)	British thermal unit per hour (Btuh)	1.00
Electric	Watt-hour (Wh)	Watt (W)	3.41
	Kilowatt-hour (kWh)	Kilowatt (kW)	3,412
Mechanical	Horsepower-hour (hp-hr)	Horsepower (hp)	2,545

Example 2.2 In the previous example, the heat was released from the concrete slab during a night setback period from 10:00 p.m. to 6:00 a.m. What was the average capacity of the slab over this period to assist in heating the building?

Solution: The amount of heat is 15,000 Btu. It was released over an 8-hour period; therefore, the average capacity was

$$\begin{aligned} \text{Power} &= \text{Energy}/\text{time} \\ &= 20{,}200\ \text{Btu}/8\,\text{hr} = 2500\ \text{Btuh} \end{aligned}$$

Electric power is measured in watts (W) or kilowatts (1000 W). These are power units. If power is applied over time, energy is the product:

$$\begin{aligned} &\text{Electric energy (kilowatt-hours, or kWh)} \\ &\quad = \text{Electric power (kilowatts, or kW)} \\ &\qquad \times \ \text{time (hours)} \end{aligned} \quad \textbf{(2.3)}$$

Example 2.3 A 100-W light is on for 10 hours per day. How much energy will the light use in a year's time?

Solution:

$$\begin{aligned} \text{Energy} &= \text{power} \times \text{time} \\ \text{Energy(kWh)} &= 100\ \text{Watts} \times 10\ \text{hr/day} \\ &\qquad \times 365\ \text{days per year} \\ &= 365{,}000\ \text{watt-hours, or } 365\ \text{kWh} \end{aligned}$$

Electrical energy can be converted to mechanical energy in a motor, to light in a lamp, or to heat in a resistance heater. All of the electrical energy used in a heater becomes heat. In a motor, the majority of the electrical energy becomes mechanical power, measured in horsepower. A small portion of the electrical energy is lost as heat. Eventually, even the mechanical energy degrades into heat. In a lamp, a portion of the electrical energy becomes light, and a portion becomes heat. Eventually, virtually all the light is absorbed by room surfaces and becomes heat.

Example 2.4 An electric motor running a large copier draws 1.6 kW. How much heat is produced in the space as a result of the copier's operation?

Solution: From Table 2.2 we find the conversion factor from electric to heat energy or power:

$$\begin{aligned} \text{Heat power (Btuh)} &= \text{Electric power (kW)} \times 3412\ \text{Btuh/kWh} \\ &= 1.6 \times 3412 = 5460\,\text{Btuh} \end{aligned}$$

2.2 FUELS

2.2.1 Energy Content

Fuels are burned to produce thermal energy, which can be used to heat buildings or run engines to produce mechanical energy. The mechanical energy can be used to operate machinery, vehicles, or to produce electricity in a generator. Fuels commonly associated with building systems include natural gas (primarily methane), propane (LP), oil (various grades), and coal (for very large applications). The thermal energy produced by burning various fuels is shown in Table 2.3.

2.2.2 Relative Cost of Fuels

The relative cost of fuels is an important consideration in selection of the energy source for heating buildings. Based on approximate average 2016 U.S. costs for various fuels, Table 2.4 shows the cost per million Btu of net heating energy, considering typical equipment efficiencies. Energy costs can vary greatly between locations, and due to market fluctuations, so current, local analysis should be performed for actual projects. The analysis should consider not only current prices but also projected future trends based on best available opinions.

Heating Values of Various Fuels

Fuel	Unit of Measure[a]	Nominal Heating Value/Unit, Btu (kJ)
Natural gas	cu ft	1,000 (1,055)
LP (propane gas)	gal	93,000 (98,000)
No. 1 oil (diesel)	gal	138,000 (146,000)
No. 5 oil (heavy)	gal	145,000 (153,000)
No. 6 oil (bunker C)	gal	153,000 (161,000)
Soft coal (bituminous)	lb	13,000 (14,000)
		13,700 (14,800)
Hard coal (anthracite)	lb	12,500 (13,500)
		13,200 (14,300)
Electrical resistance[b]	kWh	3,412 (3,600)
Electric heat pump[c]	kWh	10,200 (10,800)

[a]1 gallon = 3.78 liters; 1 cubic foot = 28.32 liters; 1 pound = 0.454 kilogram.

[b]Electrical to heat energy conversion efficiency.

[c]Heat available with coefficient of performance (COP) of 3. COP for heat pump is the ratio of useful heat output divided by energy input.

Example 2.5 A 75% efficient boiler is required to produce 800,000 Btuh to offset a heating load. If the boiler uses natural gas, what will the input rate be in cubic feet per hour?

Solution: From Table 2.3, each cubic foot of gas has a heating value of 1000 Btu. At 75% efficiency, each cubic foot will produce a net heating value of 0.75 $\times$ 1000, or 750 Btu. To produce 800,000 Btuh, we will need 800,000 Btuh/750 Btu/ft^3, or 1067 ft^3 per hour.

Electric Rates Determining electric rates is not a simple issue due to complexity in rate structure. Cost per kWh from most utilities is generally lower in winter, so the cost used for a heating energy comparison should be lower than the average annual cost per kWh. For instance, a Midwest utility might have an average summer cost of $0.12 per kWh, and an average winter cost of $0.08. The lower cost is due to supply and demand factors. More power is demanded in summer due to air conditioning, and the electric utility has more available capacity in winter, so more efficient power plants can be base loaded.

An additional complexity results from rate "steps," which could be considered as a volume discount. The lowest "block" of usage has a higher unit cost per kWh than subsequent blocks. Accordingly, a large residence will pay a lower average cost for electricity than a small residence because the former has more usage in the higher blocks.

The stepped rate structure also promotes electric heating. A homeowner, for instance, might pay $0.09 per kWh for blocks representing the kWh quantity that is typically used for general power usage (lights and appliances). The block above general power usage could be priced at $0.06, which the electric company might statistically assume is used for heating.

A very large building may have a rate structure that includes a demand charge for the peak monthly power requirement (kW) as well as an energy charge (kWh). In many areas demand charges can be half or more of the total electric bill in the summer months. The application of demand charge is shown in Example 2.6.

Example 2.6 The summer rate structure for a large building is $0.04 per kWh and $15.00 per kW of peak monthly demand. If a large building under this rate uses 300,000 kWh in July and the peak July demand is 1,200 kW, what is the July electric bill? What is the average cost per kWH? (Ignore basic charges and taxes for simplicity.)

Solution:

Usage charge 300,000 kWh $\times$ $0.04/kWh = $12,000

Demand charge 1,200 kW $\times$ $15.00/kW = $18,000

Total July bill (exclusive of basic charges and taxes) $30,000 (Avg. $0.10 per kWh)

Range of Cost for Various Fuels

Energy Source for Heating a Building	Billing Unit	Energy (Btu) per Unit	Typical Efficiency (%)	Range of Unit Cost $/Unit		Range of Cost $/ Million Btu	
				Low	High	Low	High
Natural gas	Therm	100,000	80	0.30	0.95	3.75	12.88
LP (propane)	Gallon	93,000	80	2.00	4.75	13.44	63.84
No. 1 oil (diesel)	Gallon	138,000	80	2.50	2.75	13.59	24.91
Electric resistance	kWh	3,412	100	0.04	0.08	10.26	23.45
Heat pump	kWh	3,412	300	0.04	0.08	3.42	7.82

For large buildings, most utility companies measure demand in two periods: peak and off-peak. The time period between 10:00 a.m. and 10:00 p.m. weekdays is commonly considered as peak period. Demand during the off-peak period may be allowed to exceed peak period demand by a factor of 2 times before being used as a basis for the demand charge. Since most heating occurs at night and during the early morning hours, the cost per unit for most of the kWh used for electric heat could be very low since this power doesn't contribute to the demand charge. *Usage* charges (exclusive of demand) between $0.035 and $0.06 per kWh are not uncommon for large buildings in the Midwest. Much higher values could apply in other parts of the United States.

Heat Pumps For heat pumps the coefficient of performance (COP) is defined as the heating energy output divided by the electric energy input. If a heat pump has a COP of 3, then 3 units of heat are produced by 1 unit of electric input. Stated another way, 1 kWh input transports 2 kWh of heat from outdoor air (air source heat pump) or from the earth (ground source heat pump) for a total of 3 kWh of heat. COP varies greatly by system and location, so a COP of 3 is used in Tables 2.3 and 2.4 to give a general comparison of electric heat pumps with other heating energy sources.

Natural Gas Cost of gas is similarly complicated by rates and contracts. In 2016 the average cost of gas for residential customers was about $0.95 per therm according to the U.S. Energy Information Administration. Commercial customers paid, on average, about $0.65 per therm. These values vary by location and size of building and are similar to electric rates. Very large facilities may have much lower gas prices if they directly procure from producers and pay the local utilities only for distribution overhead. Gas costs as low as $0.30 are not uncommon for this practice, but it's only used for very large facilities such as college campuses and industrial complexes.

LP and Oil During the same general time frame referenced for electric and gas, LP (propane) cost varied from about $2.00 per gallon to $4.75 per gallon. Heating oil cost varied from $2.50 per gallon to $2.75 per gallon, with a national average of about $2.10 per gallon. Heavy oils and coal are not commonly used as a primary heating energy for buildings, so they are excluded from Table 2.4.

Example 2.7 A proposed building has an estimated annual heating energy load of 3000 mmBtu per year. The owner wishes to consider using a heat pump concept (air source or ground source) instead of natural gas, which is the most commonly used fuel in his town, despite the fact that the town's gas rates are on the high side. Before spending a lot of time in design, he'd like to get a ballpark estimate of potential savings. His engineer suggests that electric to run the heat pumps during the winter would cost about $0.06 per kWh. How much might he expect to save using the heat pump concept?

Solution: From Table 2.4, natural gas would likely cost about $12 per mmBtu, and cost of heat pumps would lie between $3.42 and $7.82, or about $5.50 per mmBtu. This results in an estimate of 3000 mmBtu × $12.00 per mmBtu = $36,000 for gas and 3000 mmBtu × $5.50 = $16,500 for heat pump. Savings would likely be in the range of $20,000 per year.

2.3 PROPERTIES OF AIR–WATER MIXTURES

The design of environmental control systems relies on an understanding of the properties of air, including temperature and humidity. These properties affect loads on buildings, and HVAC systems are used to alter the properties of air and produce comfort.

2.3.1 Psychrometry

Psychrometry is the study of properties of air–water mixtures. A psychrometric chart is a convenient source for data on the properties of such mixtures. Fig. 2.2 shows how important properties are presented on a psychrometric chart. Fig. 2.3 is a complete chart that can be used in analysis of processes associated with HVAC.

2.3.2 Absolute and Relative Humidity

Two basic properties of air–water mixtures are temperature and humidity. The humidity of the air can be expressed in two ways: absolute and relative. *Absolute humidity*, also known as the *humidity ratio* (W), is the amount of water in the air and is measured in grains or pounds of water per pound of dry air. A grain is equivalent to 1/7000 of a pound. This unit is preferred owing to the very small amount of water present in air. *Relative humidity* (RH) is the ratio of the actual water content to the maximum possible moisture content at a given temperature, expressed as a percent. If the air is currently holding all the moisture possible, the relative humidity is 100 percent, and the air is termed *saturated.*

2.3.3 Effect of Temperature on Humidity

The moisture-holding capacity of the air depends on the air temperature. Warm air can hold more moisture than cold air. For this reason, the same absolute humidity results in different relative humidities at different temperatures. The psychrometric chart illustrates the relationship of temperature, absolute humidity, and RH.

2.3.4 Wet-Bulb Temperature

If a wet sock is placed over the bulb of a conventional thermometer, a lower temperature will be recorded owing to evaporative cooling. The drier the air, the more effective will be the evaporative cooling, and the lower will be the temperature measured. If the air is saturated, then there will be no evaporation and the wet-bulb thermometer will measure the same temperature as a dry-bulb thermometer. The temperature and humidity of the air can be determined by measuring both dry-bulb and wet-bulb temperatures. A combination of wet- and dry-bulb temperature represents a discrete point on the psychrometric chart.

2.3.5 Sensible, Latent, and Total Heat

Air contains thermal energy in two forms: sensible heat and latent heat. Water vapor, or humidity, in the air contains the water's latent heat of vaporization (approximately 1000 Btu/lb of water). Temperature is a measure of sensible heat, while water vapor content is a measure of latent heat. Total heat—the sum of sensible and latent heat—is *enthalpy*, symbolized by the Greek letter eta, or by H. Enthalpy is expressed in units of Btu/lb of dry air. High temperature or high humidity constitutes high energy.

On the psychrometric chart, horizontal movement is associated with sensible heat change (no change in absolute humidity), and vertical movement is associated with latent heat change (no change in temperature). Moving upward or to the right indicates a higher energy level; moving downward or to the left indicates a lower energy level. Lines of

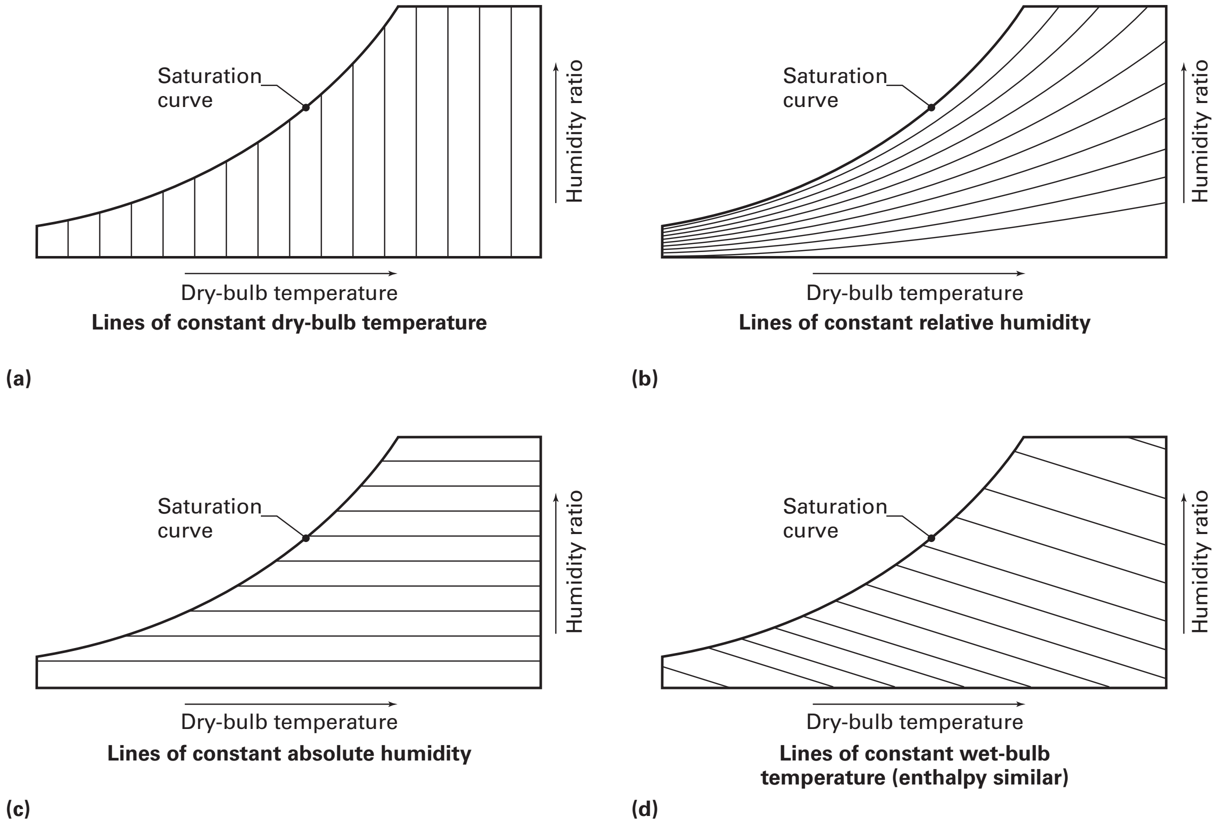


FIGURE 2.2. Lines representing major properties of air–water mixture on the ASHRAE psychrometric chart. (a) Vertical lines: constant dry-bulb (DB) temperature. (b) Curved lines: constant relative humidity (RH). (c) Horizontal lines: constant humidity ratio (W), also commonly referred to as *absolute humidity.* (d) Sloped lines: constant wet-bulb (WB) temperature; lines with same slope: constant enthalpy (H).

constant enthalpy slope upward and to the left at approximately the same slope as lines of constant wet-bulb temperature. This is no coincidence, for wet-bulb temperature is a good measure of total energy.

Often, changes in air conditions result in changes in both humidity and temperature as shown in Fig. 2.4. The net change in energy level, or enthalpy, can be determined by plotting the initial and final conditions on the psychrometric chart.

2.3.6 Sensible Heating and Cooling

Sensible heating (cooling) occurs when the temperature of an air–water mixture is raised (lowered) but the absolute moisture content remains the same. Sensible heating or cooling occurs as air in spaces is warmed or cooled by building loads that do not change the moisture content of the air. Sensible heating or cooling is also performed by systems to compensate for loads. For instance, room air may be cooled by an outside wall during cold winter weather. To compensate, a heater at the base of the wall may warm the air. Sensible heating or cooling is represented by a horizontal movement along the psychrometric chart.

2.3.7 Processes Involving Latent Heat

Heating and cooling represent a change of sensible heat; humidification and dehumidification represent a change of latent heat. The amount of moisture liberated or absorbed by air is measured by its initial and final absolute humidities.

Air can be humidified either by adding dry steam to it or by evaporating moisture into it. If dry steam is added, the air will have a higher energy level, taking on the latent heat of the steam. (There will also be a slight increase in temperature owing to the sensible heat of the steam, but the effect is small and generally ignored in practice.) On the psychrometric chart, this process is represented by a vertical movement.

If water is evaporated into air, the air will cool, but the final energy level of the air does not change. The heat required to vaporize the water cools the air. The sensible heat loss equals the latent heat gain, resulting in constant enthalpy. This process is called *adiabatic saturation.* (No energy is added or removed.) Evaporative humidification is accompanied by evaporative cooling and is represented on the psychrometric chart by an upward movement along a line of constant enthalpy (approximately parallel to a line of constant wet-bulb temperature).

Cooling is a method for dehumidifying air. If moist air is cooled to the saturation curve, further cooling will not only reduce temperature but also remove moisture. The temperature at which moisture begins to condense is termed the *dew point.* Liquid moisture removed from the air by this process is termed *condensate.* The air that results from the process is both cooler and less humid than it was initially.

Air also can be dehumidified by absorption. Some substances are *hygroscopic*, meaning that they absorb moisture. Hygroscopic substances, or desiccants, such as silica gel and lithium bromide are used in certain applications to absorb moisture from the air. As moisture condenses in the desiccant its latent heat is liberated, heating the desiccant and the air. Absorption is represented by a movement on the psychrometric chart approximately opposite in direction to evaporative cooling.

2.3.8 Examples to Understand the Psychrometric Chart

1. Air at 70°F DB and 75% RH is heated to 84°F. What is the RH of the air at this higher temperature?

   Solution: In Fig. 2.3, locate the air at the initial condition (70°F DB and 75% RH) and follow the horizontal line to the right until it meets the 84°F DB line (vertical). The heated air is now at 47% RH.

2. Air at 90°F DB and 70% RH is cooled to 75°F. What is the relative humidity?

   Solution: In Fig. 2.3, from the intersecting point of 90°F DB and 70% RH, draw a line to the left. This line meets the saturation curve at 79°F, which is the dew point temperature of the air. The air is then cooled further, following the saturation curve until it stops at 75°F. Between 79°F and 75°F, the air is saturated, and moisture condenses out of it. The RH of the air is now 100%.

3. Outside air at 95°F DB and 78°F WB is mixed with air returning from a room at 75°F/50% RH. The mix is 20% OA/80% return. What is the condition of the mix?

   Solution: The dry-bulb temperature of the mixed air can be determined as follows:

   $$\begin{aligned}\text{Mixed air temp} &= 20\% \times \text{OA temp} + 80\% \times \text{RA temp} \\ &= 0.2 \times 95 + 0.8 \times 75 = 79°\text{F}\end{aligned}$$

   The humidity ratio of the mixed air can be determined by a similar equation. From the psychrometric chart, the humidity ratio of air at 95°F DB and 78°F WB is 0.0168 lbs/lb, and the humidity ratio of air at 75°F DB and 50% RH is 0.0093 lbs/lb. The mix is 0/0108 lbs/lb.

4. Air at 95°F DB and 78°F WB is cooled to 55°F. What is the change in humidity ratio?

   Solution: In Fig. 2.3, from the intersecting point of 95°F DB and 78°F WB, draw a line to the right to the vertical axis, and read the humidity ratio, which is 0.0168 lbs/lb. Then draw a line from the initial condition to the left. This line meets the saturation curve at 72°F, which is the dew point temperature of the air. The air is then cooled further, following the saturation curve until it stops at 55°F. Between 72°F and 55°F, the air is saturated, and moisture condenses out of it. Humidity ratio at 55°F saturated is determined by drawing a line to the right to the vertical axis, and read the humidity ratio, which is 0.0092 lbs/lb. The change is humidity ratio is 0.0168 minus 0.0092, or 0.0076 lbs/lb.

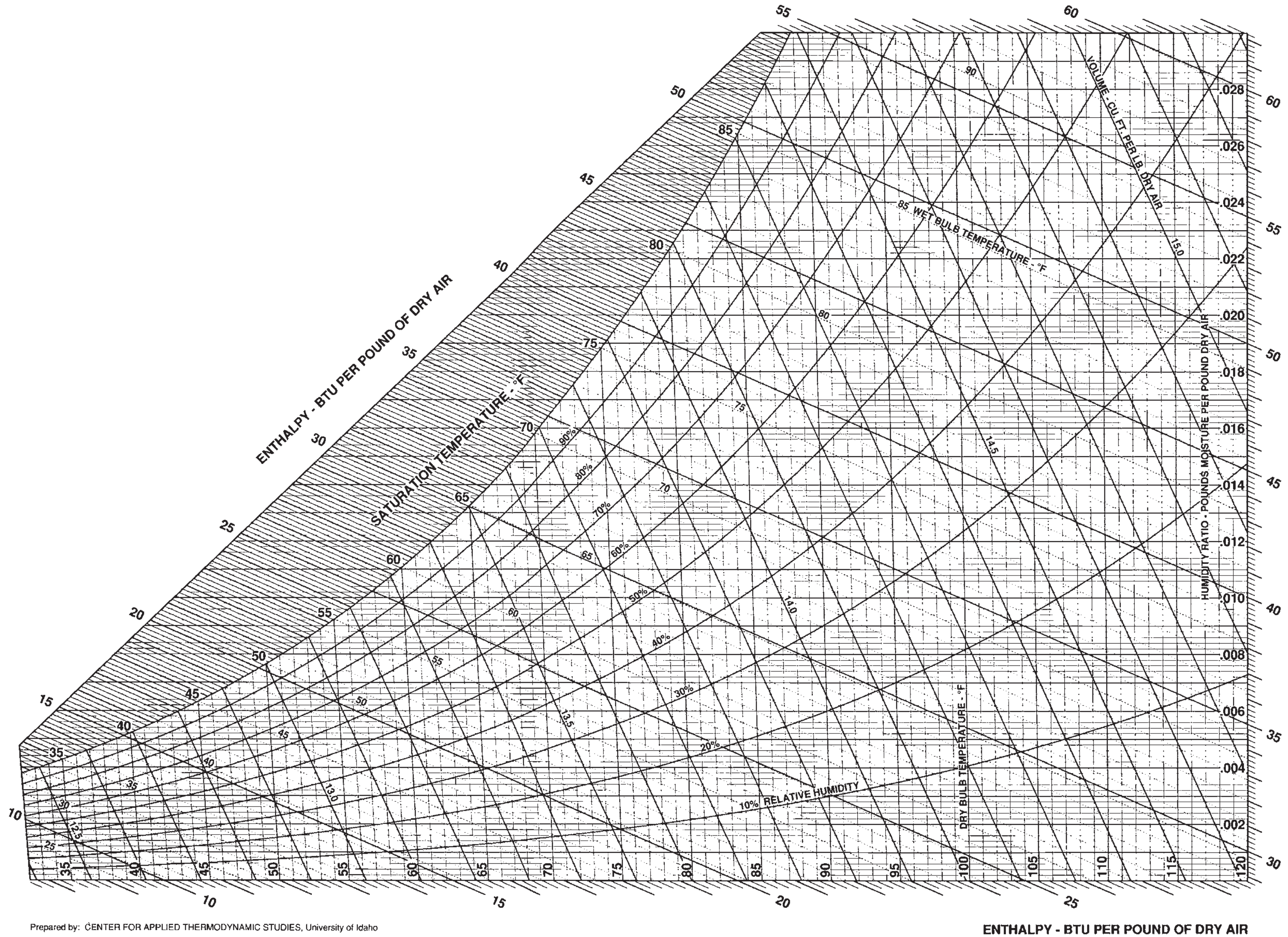
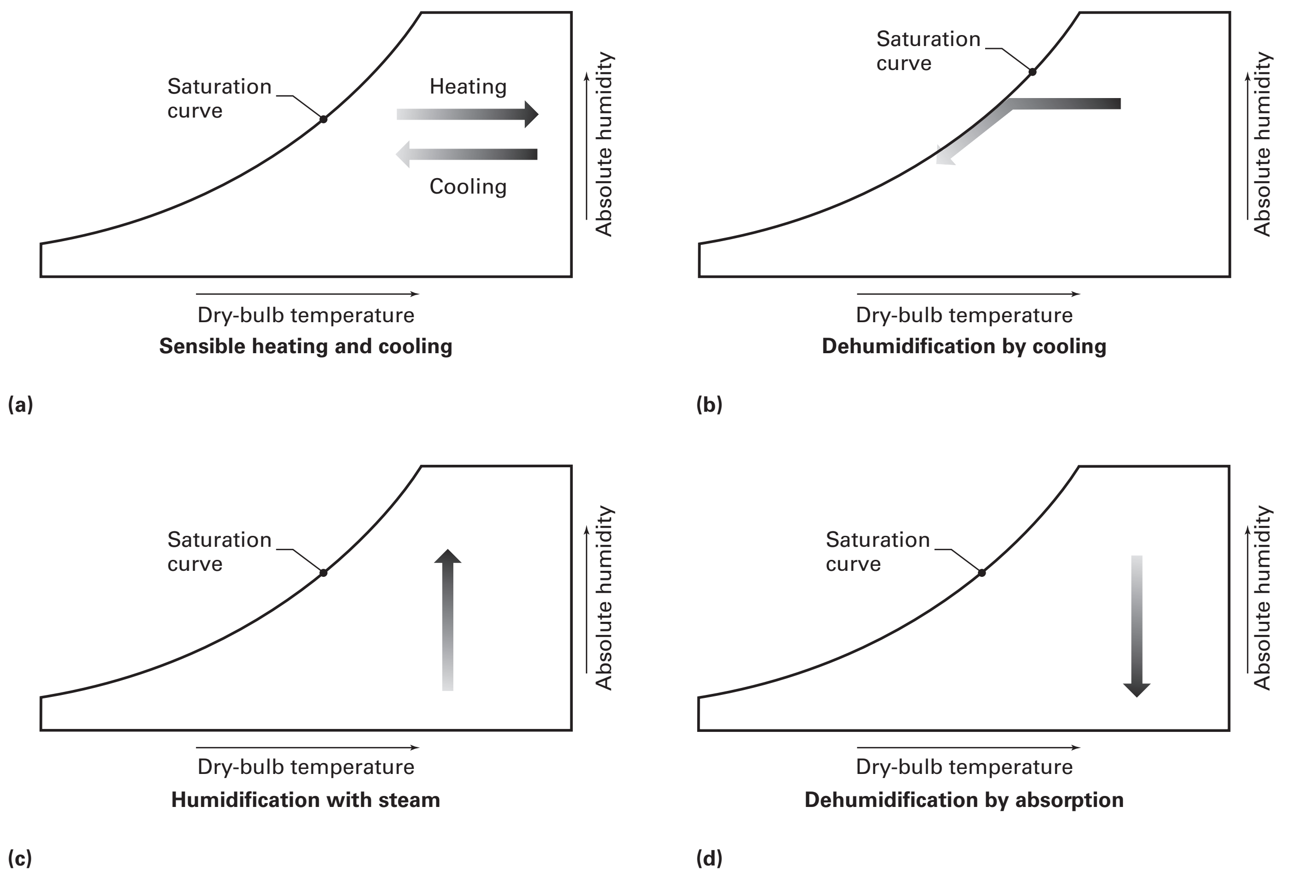


FIGURE 2.3. Psychrometric chart.

(©ASHRAE, www.ashrae.org. (2017) ASHRAE Handbook—(IP version page 1.15, their Fig.1 ASHRAE Psychrometric Chart No. 1.). Used with permission.)



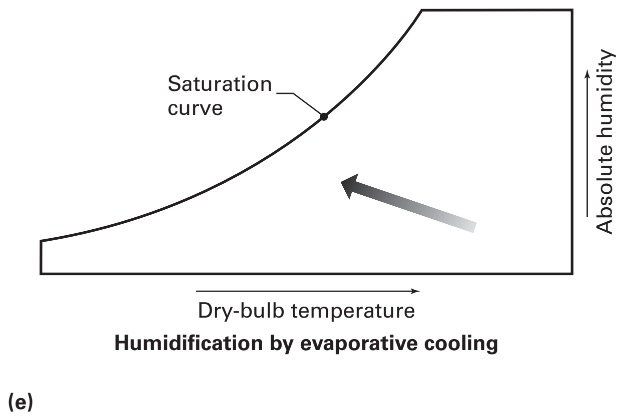


FIGURE 2.4. Basic psychrometric process. (a) Sensible heating and cooling—horizontally. (b) Dehumidification by cooling—follows saturation curve. (c) Humidification with steam—essentially up vertically with slight heating. (d) Dehumidification by absorption—essentially down vertically with slight heating. (e) Humidification by evaporative cooling—approximately follows the wet-bulb line.

2.4 FLUID FLOW AND PRESSURE IN MECHANICAL SYSTEMS

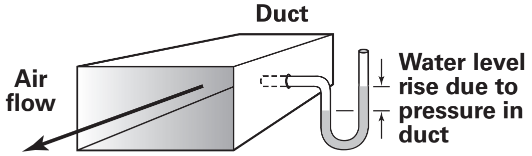
Mechanical systems use the flow of air, water, and steam to transfer energy. Airflow is measured in cubic feet per minute, abbreviated CFM. Air pressures in heating and air-conditioning systems are very low, and measuring in the familiar unit of psi (pound per sq in) would result in numbers too small to be used conveniently. Pressure is measured in "inches of water column" as explained in Fig. 2.5.

Water flow is measured in gallons per minute, abbreviated GPM. Pressures are measured in two units: psig and "ft of head." Head, measured in feet, is equal to the pressure at the bottom of a column of water. For instance, a dam that has a headwall holding water 100 ft in depth has a pressure of 100 ft of head at the bottom. A pressure measurement of 100 ft of head at a particular point in a piping system would be the same as water pressure at 100 ft depth. One psig is equal to 2.31 ft of head.

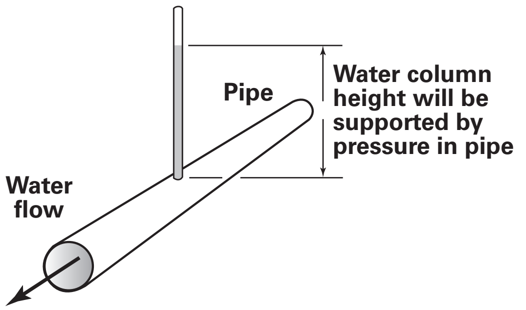
Steam is measured in pounds (lbs), and flow is measured in lbs per hour. Pressure is measured in psig.

2.5 ENERGY TRANSPORT IN HVAC SYSTEMS

Fig. 2.1 shows that HVAC systems use fluids to transport heat and cold to satisfy loads and maintain comfort. Such fluids include air, water, steam, and refrigerant. Equations



Air pressure measured in "inches of water column."



Water pressure measured in "feet of head."

FIGURE 2.5. Measuring pressure of air and water in HVAC systems.

are developed in this section that can be used to determine heat transport based on the flow rate and the initial and final conditions of the fluid. These equations can also be used in equipment design to specify flow rates or conditions, based on requirements for heat transport. The rate of heat flow is measured in Btus per hour, or Btuh. Fluids are used to transport heat in HVAC systems.

2.5.1 Heat Transport in Air

Sensible Heat Transfer in Air The natural property of a fluid that affects heat transfer is called *specific heat*; this is the amount of energy (Btu) required to raise the temperature of 1 lb of a substance 1°F. The specific heat of water is 2.0 and that of air is 0.24. The heat liberated from a quantity of fluid is equal to the specific heat of the fluid multiplied by the number of pounds of the fluid and the temperature change between the initial and final states of the fluid; that is,

$$q = m \times C \times \Delta T \quad \textbf{(2.4)}$$

where q = heat energy (Btu)
m = mass (lb)
C = specific heat (Btu / (lb·°F))
ΔT = temperature difference, °F (final temperature minus initial temperature)

HVAC equipment loads, equipment capacity, and output are expressed as quantities per unit time, or *rates*:

$$Q = M \times C \times \Delta T \quad \textbf{(2.5)}$$

where Q = heat flow (Btu / hr, or Btuh)
M = mass flow (lbm / hr)
C = specific heat (Btu / (lb·°F))
ΔT = temperature difference, °F

Mass flow is quantified in units of cubic feet per minute, or CFM. An equation for sensible heat transfer in air can be derived given that the density of air at standard pressure is 0.075 lb / ft^3 and that the specific heat of air is 0.24 Btu / lb·°F:

$$\begin{aligned} Q_{\text{sensible}} &= 0.076 \text{ lbs/ft}^3 \times .024 \text{ Btu/lb} \cdot {}^\circ\text{F} \\ &\quad \times 60 \text{ min/hr} = 1.1 \times \text{CFM} \times \Delta T \end{aligned} \quad \textbf{(2.6)}$$

where CFM = airflow (ft^3 / min)

Latent Heat Transfer in Air Mass flow is quantified in units of cubic feet per minute, or CFM. An equation for latent heat transfer in air can be derived given that the density of air at standard pressure is 0.075 lb / ft^3 and that the latent heat of vaporization of water is 1076 Btu / lb. The difference in humidity ratio WD can be expressed in lbs of H_2O / lb of dry air or in grains H_2O / lb of dry air:

$$\begin{aligned} Q_{\text{latent}} &= 0.075 \text{ lbs/ft}^3 \times 1076 \text{ Btu/lb} \\ &\quad \times 60 \text{ min/hr} = 4840 \times \text{CFM} \times \Delta W \end{aligned} \quad \textbf{(2.7)}$$

where CFM = airflow (ft^3 / min)
ΔW = difference in humidity ratios (lbs of H_2O / lb of dry air)

Or, if difference in humidity ratio WD is expressed in grains of H_2O per lb of dry air:

$$\begin{aligned} Q_{\text{latent}} &= (0.075 \text{ lbs/ft}^3 \times 1760 \text{ Btu/lb} \times 60/\text{min/hr})/ \\ &\quad 7000 \text{ grains/lb} = 0.68 \times \text{CFM} \times \Delta W \end{aligned} \quad \textbf{(2.7a)}$$

where CFM = airflow (ft^3 / min)
ΔW = difference humidity ratios (grains of H_2O / lb of dry air)

Total Heat Transfer in Air Total heat, termed *enthalpy (H)*, is the sum of sensible and latent heat. The total heat of air at various conditions of temperature and humidity can be taken from a psychrometric chart or tables, and the following equation can be used to determine energy flow:

$$Q = 4.5 \times \text{CFM} \times \Delta H \quad \textbf{(2.8)}$$

where ΔH = change in enthalpy (Btu / lb) Δ

2.5.2 Heat Transport in Water

Mass flow is quantified in units of gallons per minute (GPM). Knowing that 1 gal of water has a mass of 8.35 lb and that there are 60 minutes in 1 hour, the following equation can be derived:

$$\begin{aligned} Q &= 1 \text{ Btu/lb} \cdot {}^\circ\text{F} \times 8.35 \text{ lbs/gal} \\ &\quad \times 60 \text{ min/hr} \times \text{flow (gal/hr)} \\ &\quad \times \text{temp. difference (}^\circ\text{F)} \\ &= 500 \times \text{GPM} \times \Delta T \end{aligned} \quad \textbf{(2.9)}$$

where Q = Heat flow (Btu / hr, or Btuh)
GPM = water flow (gal / min)
ΔT = temperature difference, °F

2.5.3 Heat Transport by Fluid Phase Change

Heat Transfer in Steam Heat is liberated from steam by a change of phase from vapor to liquid. One pound of steam liberates approximately 1000 Btu as it condenses. Conversely, a boiler must produce 1000 Btu to boil 1 lb of water. For steam, heat flow is approximated by the equation

$$Q = 1000\ \text{Btu/lb} \times \text{lb/hr} = 1000 \times \text{SFR} \quad \textbf{(2.10)}$$

where SFR = steam flow rate (lb/hr)

Heat Transfer in Refrigerants Refrigerants absorb heat by changing phase from liquid to gas. The heat absorbed is equal to the latent heat of vaporization, measured in Btu per pound, times the refrigerant flow rate, measured in pounds per hour. There are many types of refrigerants, and each has its own distinct latent heat of vaporization.

2.5.4 Selecting Fluid Flow Rates for HVAC Systems

HVAC systems and subsystems are designed to satisfy heat loads by using heat transport fluids. Fans, pumps, boilers, and distribution elements are sized according to flow requirements, which must be determined by the HVAC designer. The first step is to estimate building heat loads. Methods for estimating loads are presented later in this chapter. Once they are estimated, the HVAC designer must decide on the proper combination of flow and conditions for fluids used to transfer heat and thereby compensate for loads. Initial and final conditions are generally selected on the basis of accepted general practice found to achieve satisfactory results. Equations (2.1) through (2.5) can be used to calculate flow, given the heat transfer requirement and the initial and final conditions of the fluid.

Water Flow For devices using hot water for heating, a supply temperature of 160°F might be chosen and the load equipment selected to allow a 20°F drop in water temperature, resulting in a 140° return temperature. Once this decision is made, the required water flow rate can be calculated using Equation 2.11.

$$\text{GPM} = Q/(500 \times \Delta T) \quad \textbf{(2.11)}$$

Similarly, chilled water can be used for cooling. Chilled-water supply temperatures between 40°F and 50°F are common for building HVAC applications, and systems are designed for water temperature rises ranging from 10°F to 15°F. The same equation applies for determining the required chilled-water flow rate.

Airflow For systems using warm air for heating, supply temperatures between 105°F and 140°F will be appropriate to maintain a space at, say, 75°F. Given a space temperature and a selected supply temperature, the required airflow rate can be calculated using Equation 2.12.

$$\text{CFM} = Q/(1.1 \times \Delta T) \quad \textbf{(2.12)}$$

Systems using chilled air for cooling generally have supply air temperatures between 50°F and 60°F. Equation 2.12 can also be used to determine the airflow rate required to satisfy the sensible portion of cooling loads. Once the airflow rate is determined, the humidity can be determined from a psychrometric chart.

Steam Flow The rate of steam flow required to satisfy a given heating load is determined by using Equation 2.13.

$$\text{SFR} = Q/1000 \quad \textbf{(2.13)}$$

where SFR = steam flow rate (lb/hr)
Q = heat flow (Btuh)
1000 = heat (Btu) liberated by condensation of 1 lb of steam

Refrigerant Flow The rate of refrigerant flow for cooling is determined by dividing the cooling load by the latent heat of vaporization.

The foregoing concepts and equations are used to estimate theoretical fluid flow rates required to meet a given load. The resulting estimates are the basis for sizing the piping and duct systems, along with the pumps and fans required to transport heating and cooling.

2.5.5 Examples for Understanding Heat Transfer by Fluid Flow

1. A room has a sensible cooling load of 55,000 Btuh. How many CFM at 55°F will be required to keep the room at 75°F?

   Solution: Using Equation 2.12, the required airflow will be as follows:

   $$\text{CFM} = Q/(1.1 \times \Delta T) = 55{,}000/(1.1 \times (75 - 55)) = 2500\ \text{CFM}$$

2. The air in the preceding example is cooled and dehumidified in a cooling coil. The coil inlet condition is 79°F, 0.0108 lbs of H_2O/lb of dry air, and the coil outlet condition is 55°F, 0.0092 lbs of H_2O/lb of dry air.

   What is the sensible heat removal at the coil?

   Solution: Using Equation 2.6, the sensible heat removal will be as follows:

   $$Q_{\text{sensible}} = 1.1 \times \text{CFM} \times \Delta T = 1.1 \times 2500 \times (79 - 55) = 66{,}000\ \text{Btuh}$$

   What is the latent heat removal at the coil?

   Solution: Using Equation 2.7, the latent heat removal will be as follows:

   $$Q_{\text{latent}} = 4840 \times \text{CFM} \times \Delta W = 4840 \times 2500 \times (0.0108 - 0.0092) = 19{,}400\ \text{Btuh}$$

   What is the total heat removal at the coil?

   Solution: From the psychrometric chart we determine that the entering air enthalpy (H) is approximately 30.7 Btu/lb and the enthalpy of the leaving air is approximately 23.2 Btu/lb. Using Equation 2.8, the total heat removal will be as follows:

$$Q_{\text{total}} = 4.5 \times \text{CFM} \times \Delta H = 4.5 \times 2500 \times (30.8 - 23.2) = 85{,}500 \text{ Btuh}$$

3. The coil receives 45°F chilled water, and is selected based on a 15°F temperature rise to 60°F. What chilled water flow rate is required to remove the 85,500 Btuh?

   Solution: Using Equation 2.11, the required chilled water flow will be as follows:

$$\text{GPM} = Q/500 \times \Delta T = 85{,}500/500 \times (60 - 45) = 11.4 \text{ GPM}$$

4. A building has a heating load (heat loss) of 6000 mBh. The heating system uses hot water and is designed for a 40°F ΔT at full load conditions. What is the required hot water flow?

   Solution: Using Equation 2.11, the required hot water flow will be as follows:

$$\text{GPM} = Q/500 \times \Delta T = 6{,}000{,}000/500 \times (40) = 300 \text{ GPM}$$

5. If the building in the preceding example were heated by steam, what steam flow would be required?
6. Solution: Using Equation 2.13, the steam flow will be as follows:

$$\text{SFR} = Q/1000 = 6{,}000{,}000/1000 = 6000 \text{ lbs/hr}$$

2.6 ENVIRONMENTAL COMFORT

2.6.1 Comfort for Occupants

The temperature of a space is not the only factor affecting a person's comfort. Even if the temperature is within an acceptable range, the space may seem warm if the humidity is too high, the airflow is too low, or there are warm surfaces radiating heat to occupants. Conversely, a space may seem cool if the humidity is low, the space is drafty, or there are cold surfaces absorbing heat radiated from occupants. Comfort for building occupants is affected by a number of environmental variables, including the following:

- Temperature
- Airflow
- Humidity
- Radiation

Indoor air quality is another aspect of comfort. In air of good quality, sufficient oxygen is present and objectionable impurities such as dust, pollen, odors, and hazardous materials are absent.

Different conditions may be deemed comfortable, depending on the type of activity that goes on in a space. Appropriate conditions for an office would be too warm for a gymnasium and too dry and cool for a natatorium. Expectations must also be considered: Saunas are hot on purpose, and a wide variety of conditions are commonly tolerated without complaint in factories. The physical condition of the occupants, including their age and health, also affects their comfort. Even the seasons affect comfort: Warmer environments are tolerated during the summer and cooler environments in winter, because of clothing and acclimatization.

Economics and concerns about energy conservation are also considered in defining comfort. People might be satisfied with less comfort when they know the purpose is energy conservation or to save money.

2.6.2 Temperature and Humidity

Both temperature and humidity affect our sense of comfort. An ASHRAE graph shows the acceptable range of each for persons wearing typical summer and winter clothing involved in sedentary activities. The lower comfort limit in cold weather is 68°F at about 30 percent RH. More recent versions of the ASHRAE graph have eliminated the lower limit on humidity, recognizing that most commercial buildings do not use humidifiers during the heating season, due primarily to cost and high maintenance. The upper limit in hot weather is 79°F at about 55 percent RH. HVAC systems are generally designed to maintain temperature and RH within a tighter range during the cooling season, and to maintain temperatures, but not humidity in a tighter range during the heating season.

An interior design temperature of about 75°F is considered comfortable by most people in general-use spaces. During the summer, a slightly higher temperature may be appropriate because of light clothing and acclimatization to warm weather; this should be considered in designing air-conditioning systems. Conversely, slightly cooler temperatures are acceptable and can be considered in the design of heating systems. Most air-conditioning systems are designed to maintain a summer temperature of 72–78°F. During winter, heavier clothing and acclimatization to cold weather result in a recommended design temperature of 68–72°F for heating systems. These interior design temperatures will be appropriate for the majority of buildings.

Humidity in excess of 60 percent is considered high in general-use spaces. High humidity not only is uncomfortable but also can result in indoor air-quality problems due to mold growth. Humidity lower than 25–30 percent can result in discomfort due to drying of breathing passages and also cause problems with electronic equipment due to static electricity.

2.6.3 Airflow

Systems must be designed for adequate airflow to prevent complaints of "stuffiness" or drafts. The measure of airflow is velocity. Space air velocities less than 10 feet per minute will be stuffy; those more than 50 feet per minute may seem drafty.

2.6.4 Air Quality

Systems must provide sufficient amounts of clean air to keep oxygen at an acceptable level and to dilute contaminants generated within occupied spaces. Air should be reasonably free of dust, and spaces free of odors or other pollutants that may be hazardous or objectionable. These conditions are generally achieved through the use of filters and by the introduction of outside air into the system at rates specified in ASHRAE Standard 62, “Ventilation for Acceptable Indoor Air Quality.”

Ventilation rates for indoor air quality have been subject to change based on social context. They decreased during the energy crisis of the 1970s and subsequently increased with reports of “sick-building syndrome” shortly thereafter. There is current research available which might increase ventilation rates further based on reported health and productivity benefits. Accordingly, there are LEED Credits associated with using ventilation rates higher than ASHRAE Standard 62.

2.6.5 Radiant Effects

Even if the temperature, humidity, and airflow in a space are acceptable, the space may be uncomfortable owing to radiant effects from cold windows or walls. Systems must therefore compensate for these effects with radiant heat or higher temperatures. Similarly, cooler temperatures or higher air velocities will be needed to offset the effects of warm surfaces. Downdrafts from cold surfaces are also uncomfortable and can be offset by proper placement of heating devices, generally below windows.

2.6.6 Special Considerations

Buildings such as museums, computer centers, and laboratories have special requirements for temperature, humidity, airflow, and air quality. In some instances these requirements are consistent with the comfort of the occupants, but in others they are at odds with comfort.

Interior environmental criteria are often based on specifications for equipment used within an occupied space. Computer rooms, for example, are often drafty and cold in the aisles where air is supplied to rack fronts and hot in the aisles where the racks discharge their heat. This will be an uncomfortable environment for staff in the computer room, and special provisions may necessary for comfort certain areas of the room. Similarly, materials stored in a warehouse may tolerate cold or hot temperature, but the warehouse employees need a refuge of human comfort.

Economics and expectations of comfort also affect design criteria. Energy conservation is a component of sustainable design, and occupants might accept less comfortable conditions to save energy. However, as noted in Chapter 1, comfort should not be sacrificed at the expense of productivity.

QUESTIONS

2.1 If the lighting load for a 10,000-ft^2 building is estimated at 1 W/ft^2, what will be the resulting heat generated by lighting in units of MBtu for 3000 hours of lights on?

2.2 If the lighting load were increased, what would be the effect on other building systems in a Midwestern U.S. climate? Would you increase the capacity of the heating system? The cooling system? What would the energy impact of higher lighting loads be on gas for heating, electric for cooling, and overall electric usage?

2.3 How much heat (Btus) will be stored in a 100-ft^2 concrete wall 1 ft thick if it is warmed from 65°F to 85°F by exposure to sunlight?

2.4 What is the equivalent value of the heat in Question 2.3 compared with gas at $0.65 per therm burned in a boiler at 85% efficiency? What equivalent value compare with electric at $0.06 per kWh?

2.5 Compare the annual cost of heating by propane at $2.00/gallon in a 85% efficient furnace versus electric heat pump with a COP of 3 using electric at $0.06 per kWh. The building is 3000 ft^2, and the engineer assumes an annual heating requirement of 30,000 Btu/ft^2/yr.

2.6 What is the difference between absolute humidity, often called *humidity ratio*, and relative humidity? What are the units used to express each of these quantities?

2.7 If the dry-bulb temperature is 95°F and the wet-bulb temperature is also 78°F, what is the relative humidity? What is the dew point? What is the humidity ratio? What is the enthalpy?

2.8 If the dry-bulb temperature is 55°F and the wet-bulb temperature is also 55°F, what is the relative humidity? What is the dew point? What is the humidity ratio? What is the enthalpy?

2.9 If 20,000 CFM of air at the condition in Question 2.7 is cooled to the condition in Question 2.8, what is the rate of sensible heat removal (Btuh)? What is the rate of latent heat removal (Btuh)? What is the rate of total heat removal (Btuh)?

2.10 If 2000 CFM of air at 5°F is mixed with 8000 CFM of air at 75°F, what is the temperature of the mixed air?

2.11 If the humidity ratio of the 5°F air in Question 2.10 is 0.002 lbs of H_2O / lb of dry air, and the humidity ratio of the 75°F air is 0.0093, what is the humidity ratio of the mixed air? How much moisture in lbs/hour would be needed to raise the mixed air humidity to 0.0093 lbs of H_2O /lb of dry air?

2.12 A space has a heat gain of 40,000 Btuh sensible. How much 55°F air needs to be supplied to the space to maintain the space temperature at 75°F? How much would be needed if the supply air were 50°F?

2.13 If the 55°F air in Question 2.12 is being discharged in a saturated (100% RH) condition from a chilled water coil, and the inlet air to the coil is 100% outside air at 95°F DB and 78°F WB, what is the sensible load on the coil (Btuh)? Latent load? Total load?

2.14 If the chilled water is being supplied at 45°F, and the coil is selected so that the chilled water temperature rise is 10°F, what is the required chilled water flow through the coil in GPM?

2.15 A space has a 60,000 Btuh heat loss in winter. It is heated by a furnace discharging 110°F air. How much air will be needed to keep the space at 72°F?

2.16 If the space in Question 2.15 were heated with air from a hot water coil discharging air at 110°F, what hot water flow would be required through the coil if the hot water supply temperature is 140°F, and the hot water return temperature is 120°F?

2.17 If a steam coil were used in Question 2.16, what would be the required steam flow in lbs/hr?

2.18 Discuss the effects of humidity on interior comfort. What would you recommend for upper and lower limits during summer and winter? How does temperature influence your answer?

2.19 What are the effects of excessively high and excessively low air velocities in occupied spaces? What range of values might be appropriate for design?

2.20 In general, nonnumerical terms, how would you define good air quality?

2.21 How might you compensate for discomfort from a cold window?

2.22 Historically, what factors have caused variations in standards for ventilation of buildings in the United States? What is the authoritative source of these values?

CHAPTER THREE

HVAC LOAD ESTIMATING

3.1 NATURE OF HVAC LOADS

The design of HVAC systems starts with an estimate of the loads the system must satisfy. Heating loads represent how much heat is lost and therefore must be made up by the system. Cooling loads represent how much heat is gained and must be removed. Humidity must also be considered. Internal and external moisture gains and losses may need to be counteracted in maintaining proper humidity levels. Estimates of these loads involve both sensible and latent heat transfer and conversion into and within the building.

3.1.1 Methods for Estimating Loads

Many methods are available for calculating heating and cooling loads for buildings. All, however, should be considered estimates, the precision of which depends on how the method accommodates the nonuniform qualities of building assemblies and contents and the non-steady-state nature of building loads. Heat transfer in building systems is a dynamic process, with ever-changing loads from outside and within the building.

3.1.2 Manual Versus Computer Calculations

Building load calculations are done almost exclusively by using computer programs. Many good programs are available. Most use data, algorithms, and methods developed by ASHRAE. Manual load calculations should be considered only for preliminary design or simple buildings; therefore, the simplified methods presented here tend to be appropriately conservative.

3.1.3 Accuracy and Precision

Oversized HVAC equipment operates at lower efficiency than properly sized equipment. The results are higher operating cost and higher initial cost for installation. Accuracy in load calculations and proper equipment sizing are part of the sustainable design process, and have a significant effect of initial and operating costs.

The architect and the HVAC engineer generally work to the same deadline for completion of design. This necessitates that the engineer make many assumptions regarding the construction prior to the architect's documents being finalized. These assumptions are prone to be conservative. In addition, engineers generally apply a safety factor to account for poor construction. Both of these causes of oversizing could be eliminated with proper scheduling of the design/construction process and better commissioning and inspection to reduce the risk of poor-quality construction, which is a major reason why engineers apply safety factors in sizing equipment.

Accepted methods for calculating heating and cooling loads are documented in the ASHRAE *Handbook of Fundamentals*, which is revised periodically. ASHRAE methods have become increasingly precise and have become the basis for algorithms in computer programs. In the following discussion we use one of the earlier, simpler methods to describe the basic principles of calculating heating and cooling loads. In most cases, the results will be conservative. Greater precision can be achieved by computer analysis using more complex methods; however, reasonable safety factors and allowances for unknown developments still need to be applied in sizing HVAC systems and components.

3.2 LOAD CRITERIA

3.2.1 Critical Conditions for Design

The designer must select an appropriate set of conditions for calculating the load. Relevant conditions include the outside weather, solar effects, the inside temperature and humidity, the status of building operations, and many other factors.

For heating, the critical design condition occurs during cold weather, at a time when there is little or no heating assistance from radiant solar energy or internal heat gains from lights, appliances, or people. The selection of an appropriately cold outside air temperature for design is an important decision.

For cooling load calculations, the critical design condition is the peak coincident occurrence of heat, humidity, solar effects, and internal heat gains from equipment, lights, and people. The position of the sun varies by season and through the day, as does the weather. Building operations also vary. Sometimes, several estimates must be performed for different times to determine the highest combination of individual load components.

3.2.2 Temperature Criteria

The inside temperature and humidity criteria are based on expectations of comfort. Energy codes recommend a maximum winter indoor design temperature of 72°F and a minimum summer indoor design temperature of 75°F.

Outside weather conditions affect heating and air-conditioning loads from introduction of outside air for ventilation, infiltration (leakage) of air and conduction of heat through the building envelope. Historic extremes of temperature and humidity are the basis for design load calculations here. Statistical data compiled for locations throughout the world are used by HVAC designers.

The outside temperature criteria used to calculate loads depend on the nature of the building. If it is essential that the system be capable of always meeting demand, the designer might assume the coldest recorded temperature. Buildings are seldom designed according to that criterion, however.

A few buildings are designed to meet an outside temperature criterion corresponding to a median of extremes. For heating systems, this is the mean of the coldest recorded temperatures. The median value has as many annual extremes above as below it. This is a stringent criterion that may be appropriate when system performance is critical, during the coldest temperatures. For instance, performance is important in a hospital because of the condition of the patients. Also, the occupants are present during the early morning hours, when the lowest outside temperatures generally occur.

For most buildings, criteria need not be so stringent. If the inside temperature of an office building falls a few degrees lower than the intent of the design, no great harm results. In addition, the schedule of office occupancy is such that the lowest outside temperature is not coincident with the presence of many people in the building. For most buildings, outside design temperatures are selected on the basis of a "percent" concept.

For heating load calculations, temperatures are stated in the ASHRAE *Handbook of Fundamentals* corresponding to 99.6 and 99.0 percent annual frequency of occurrence. If, for instance, the 99.6 percent value for a specific location is 6°F, then statistically 0.4 percent of the year will be colder than 6°F (8760 × 0.4% = 35 hours colder than 6°F). A 99.0 percent value of 10°F would imply that statistically 1.0 percent of the year would be below the listed value (8760 × 1% = 88 hours colder than 10°F). The former criterion is more stringent than the latter. If even more stringent criteria are warranted (lower outside design temperature), the public weather records can be searched to find record low or median of the extremes.

Cooling load criteria are similar, but they use different percentage values, including 0.4 percent, 1.0 percent, and 2.0 percent. If a 2.0 percent design criterion is used, we can expect that, on average, 2.0 percent of the year will be warmer than anticipated by the load calculation, and similarly for 1.0 percent and 0.4 percent.

Extremes of hot weather are expressed as the dry-bulb temperature (DB) and the mean coincident wet-bulb temperature (MCWB). The wet-bulb temperature (WB) is also listed in Table 3.1 according to percent occurrence along with mean coincident dry-bulb temperature (MCDB). This information is provided for specifying evaporative equipment for which performance is sensitive to the wet-bulb temperature. Such equipment includes cooling towers, evaporative condensers, and evaporative coolers.

For example, the following values from ASHRAE Handbook would be applicable for St. Louis MO:

Climatic Conditions for HVAC Design St. Louis MO

Heating DB	99.6%	7.5 DB
	99%	12.7 DB
Cooling DB & MCWB	0.4%	96.2 DB 76.7 MCWB
	1%	93.5 DB 76.1 MCWB
	2%	91.1 DB 75.0 MCWB
	0.4%	79.5 WB 90.9 MCDB
Evaporative WB & MCDB	1%	78.2 WB 89.3 MCDB

3.3 CALCULATING HEATING LOADS

The first step in the design of HVAC systems is to calculate loads. Heat losses include conducted loads through building envelope elements (walls, windows, roof, etc.) and outside air loads from leakage (infiltration) and ventilation air. (See Fig. 3.1.) The amount of ventilation air required depends on the type of occupancy, the number of occupants contemplated, and the area of the space.

3.3.1 Conduction

Heat transfer by conduction is proportional to the temperature difference between the warm and cold sides of the building envelope element (wall, roof, window, etc.). Inside temperature will depend on design criteria for either personal comfort or manufacturing process, whichever is applicable, and weather conditions—such as outside air temperature, wind velocity, and humidity—selected for design.

Conduction is proportional to the difference between the outside and inside temperatures. It is also proportional to the area through which the heat is transferred; that is, twice the wall begets twice the heat transfer. Conduction also depends on the insulating quality of the wall, which is measured by resistance to heat transfer: the *R*-value. Envelope elements are made up of several layers, and heat must flow through each layer in sequence. The insulating value of a total assembly is the sum of the *R*-values of each component. The higher the resistance, the lower is the heat transfer. Heat transfer, temperature difference, area, and resistance are related by the equation

$$Q = A \times \mathrm{TD}/R \qquad \textbf{(3.1)}$$

where

Q = heat transfer (Btuh)
A = area of assembly (ft^2)
TD = temperature difference (°F)
R = resistance ($\mathrm{hr \cdot ft^2 \cdot °F/Btu}$)

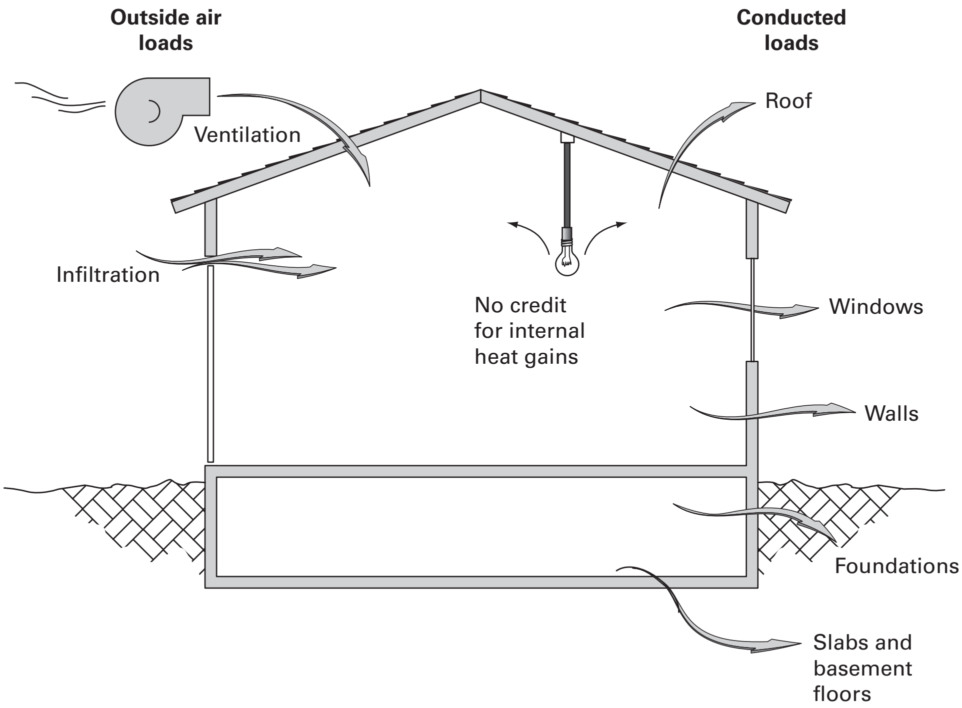


FIGURE 3.1. Components of building heating load.

The tendency of an assembly to conduct heat is called the *U*-factor, or *U*, and is mathematically the reciprocal of *R*. Thus, heat transfer is inversely proportional to *R* and directly proportional to *U*. Substituting *U* for *R* in the preceding equation, we have

$$Q = U \times A \times \text{TD} \qquad \textbf{(3.2)}$$

where $U = U$-factor (Btu/hr·ft^2·°F)

This equation is used in calculating heating load to estimate conduction heat loss through a wall, roof, or window. For calculating air-conditioning load, different equations must be used that consider the effects of the heat of the sun as well as the outside air temperature.

3.3.2 Estimating *U*-Factors for Building Assemblies

Resistance to heat flow and the *U*-factor for a wall or roof can be calculated using thermal properties for the elements that make up the assembly. The sum of the resistances of individual layers of the assembly will be the total resistance for the assembly.

Thermal resistances for representative materials are listed in Table 3.2. These values are used in sample calculation and in the problem set. For some materials, resistance is tabulated by the inch. For example, concrete has a resistance of 0.10 per inch. Thus, 8 in. of concrete has a thermal resistance of 8×0.10, or 0.80. For certain commonly used modules, resistances are tabulated for the module. An 8-in. concrete masonry unit has a resistance of 1.35. Some insulating materials are specified according to their *R*-values, such as an *R*-13 fiberglass batt.

Airspaces contained in building assemblies offer significant resistance to heat flow. The resistance of an airspace depends on the thickness of the space, the temperature and orientation of the space, and the emissivity of surfaces facing the airspace. Emissivity is a measure of a surface's ability to reflect and absorb radiant heat. For winter conditions and high emissivity (generally, dull and non-reflective) surfaces facing the cavity, resistances of air spaces in vertical walls are approximately 1.1 hr·ft^2·°F/Btu for thicknesses from 0.5 to 3.5 inches. A value of approximately R-1 adds significantly to the overall resistance of a wall assembly, and air spaces need to be considered in estimating *U*-factors.

A film of air clings to any surface and has a resistance to heat flow that depends on the thickness of the film. In still air, the film will be thick. If wind is present, its thickness will be less. Films on the exterior have resistances between 0.17 and 0.34 hr·ft^2·°F/Btu. The lower resistance is appropriate for winter due to higher expected wind speed. The higher resistance would be appropriate for summer. Films on the interior have resistances between 0.68 and 1.13 hr·ft^2·°F/Btu. The lower value would be appropriate for walls; the higher, for ceilings. Similar to air spaces, air films add significantly to the overall resistance of a wall assembly and should be considered in *U*-factor estimating.

U-factors for glass are published by manufacturers and include allowances for summer and winter values of air films. Values for selected glass materials are shown in Table 3.3. Additional glass properties pertaining to cooling load

Representative Thermal Resistances for Building Elements Used in U-Factor Calculation Exampless

Item	Resistance (ft^2–°F) per (in.-Btuh)	Resistance (ft^2–°F) per Btuh
Gypsum board, 1/2″	—	0.45
Plywood, 1/2″	—	0.62
Plywood, 3/4″	—	0.93
3-1/2″ fiberglass batt	—	13.00
Fiberglass	4.00	—
Expanded polyisocyanurate	7.20	—
Expanded polyurethane	6.25	—
Expanded polystyrene	4.00	—
Facebrick	0.17	—
6″ concrete block	—	1.20
8″ concrete block	—	1.35
12″ concrete block	—	1.45
Concrete (140 lb./ft^3)	—	0.10
Concrete, light weight	—	0.30
Shingle roofing	—	0.44
Builtup roofing (3/8″)	—	0.33
Framing lumber	1.00	—
Air space	—	1.00
Outside air film	—	0.17
Inside air film, horizontal heat flow	—	0.68
Inside air film, heat flow up	—	0.61
Inside air flow, heat flow down	—	0.92

calculations are shown in Table 3.5 in section 3.4.1. Note in Table 3.3 that maximum *U*-factors are specified by ANSI/ASHRAE/IES Std. 90.1-2013 for various Climate Zones. Values for Climate Zones 1, 4, and 6 are listed in the table.

***U*-Factors of Various Glass Materials**

Type of Glass	*U*-Factor
Single strength clear glass	1.05
1″ insulating glass, 1/4″ lites, clear	0.47
ANSI/ASHRAE/IES Std. 90.1-2013 Zone 1 minimum performance	0.50
ANSI/ASHRAE/IES Std. 90.1-2013 Zone 4 minimum performance	0.35
ANSI/ASHRAE/IES Std. 90.1-2013 Zone 6 minimum performance	0.32
Low solar gain double glazed reflective with lo-E coating	0.29
Low solar gain double glazed reflective with lo-E coating and argon fill	0.24

Zone 1 is the warmest, Zone 2 is mid range, and Zone 6 is coldest. Values given below are for glazing materials. Vendor can also provide performance for window assemblies with specific frame construction. Design of window frame will affect overall *U*-factor according to the conductivity of the frame material and design of thermal breaks between the inside and outside surfaces of the frame. The size of the window will also affect thermal performance according to the area ratio of glass to frame.

Figs. 3.2 through 3.5 illustrate the procedure for calculating the total assembly resistance and *U*-factor. Note in the examples that some assemblies have a nonuniform construction, such as stud walls with insulation between framing. The *U*-factor of the assembly will be the average based on areas of the different constructions.

3.3.3 Infiltration

The HVAC system must have sufficient capacity to heat or cool air infiltration (air leakage) at windows and entries and through loose construction. Infiltration loads will depend on the amount of outside air leakage and the difference in conditions between the outside and inside air. For heating load calculations, the following equation, derived in the previous chapter (Equation 2.6), is used:

$$Q = 1.1 \times \text{CFM} \times \text{TD} \tag{3.3}$$

If the space must be humidified, the humidification load can be determined by comparing the absolute humidity of the outside air and the desired space condition. Absolute humidity (W) is expressed in terms of grains or pounds of water per pound of dry air (1 lb = 7000 grains). The humidification load is determined approximately using either of the following equations:

$$Q = 4840 \times \text{CFM} \times (W_{\text{room}} - W_{\text{oa}}) \tag{3.4a}$$

where Q = Humidification energy (Btuh)
W_{room} = desired room humidity ratio (pounds moisture/pound dry air)
W_{oa} = outside air humidity ratio (pounds moisture/pound dry air)

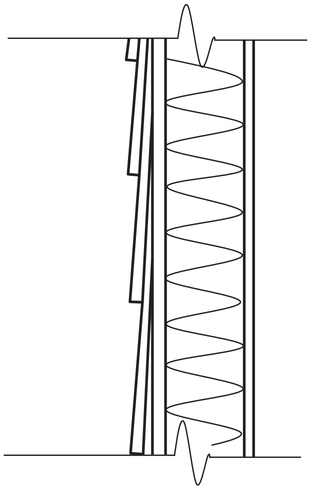
$$Q = 0.68 \times \text{CFM} \times (W_{\text{room}} - W_{\text{oa}}) \tag{3.4b}$$

where Q = humidification energy (Btuh)
W_{room} = desired room humidity ratio (grains moisture/pound dry air)
W_{oa} = outside air humidity ratio (grains moisture/pound dry air)

If steam is being used to humidify the space, the steam load can be calculated using Equation (2.13), or

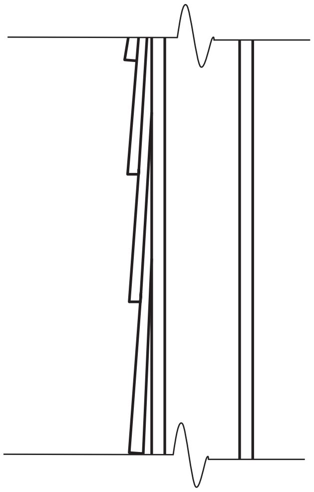
$$\text{SFR} = Q/1000 \tag{2.13}$$

where SFR = steam flow rate (lb/hr)



Section at Insulation (85% of wall area)

Item	Thickness	r per in.	r for item
Outside air film	-	-	0.17
Wood siding	1/2"	1	0.50
3/4" plywood	-	-	0.93
R-13 batt insulation	-	-	13.00
1/2" drywall	-	-	0.45
Inside air film	-	-	0.68
R_{total}			**15.73**
U-factor ($1/R_{total}$)			**0.064**

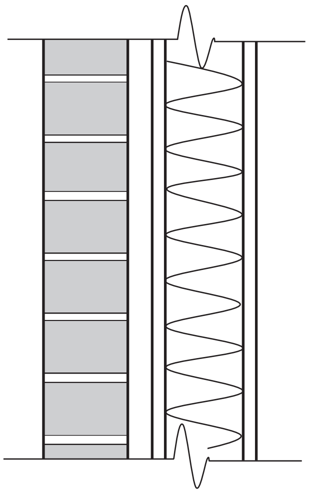


Section at framing (15% of wall area)

Item	Thickness	r per in.	r for item
Outside air film	-	-	0.17
Wood siding	1/2"	1	0.50
3/4" plywood	-	-	0.93
Wood stud	3-1/2"	1	3.50
1/2" drywall	-	-	0.45
Inside air film	-	-	0.68
R_{total}			**6.23**
U-factor ($1/R_{total}$)			**0.161**

$\text{U-factor}_{average} = 0.85 \times 0.064 + 0.15 \times 0.161 = 0.079$

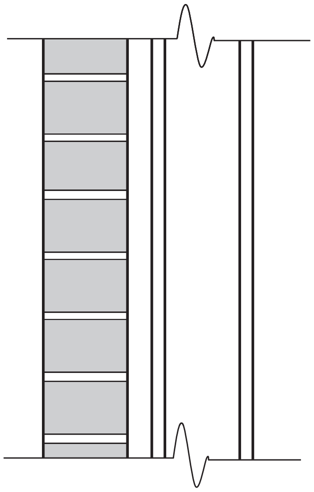
FIGURE 3.2. Sample calculation of *U*-factor for frame wall.



Section at Insulation (85% of wall area)

Item	Thickness	r per in.	r for item
Outside air film	-	-	0.17
4" face brick	3-5/8"	0.17	0.62
Air space	-	-	1.00
3/4" plywood	-	-	0.93
R-13 batt insulation	-	-	13.00
1/2" drywall	-	-	0.45
Inside air film	-	-	0.68
R_{total}			**16.85**
U-factor ($1/R_{total}$)			**0.059**

FIGURE 3.3. Sample calculation of *U*-factor for brick veneer wall.

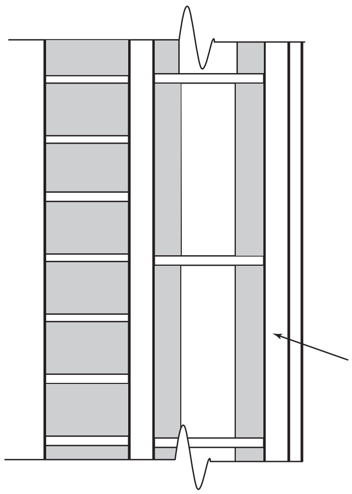


Section at framing (15% of wall area)

Item	Thickness	r per in.	r for item
Outside air film	-	-	0.17
4" face brick	3-5/8"	0.17	0.62
Air space	-	-	1.00
3/4" plywood	3-1/2"	1.0	0.93
3-1/2" wood stud	-	-	3.50
1/2" drywall	-	-	0.45
Inside air film	-	-	0.68
R_{total}			**7.35**
U-factor ($1/R_{total}$)			**0.136**

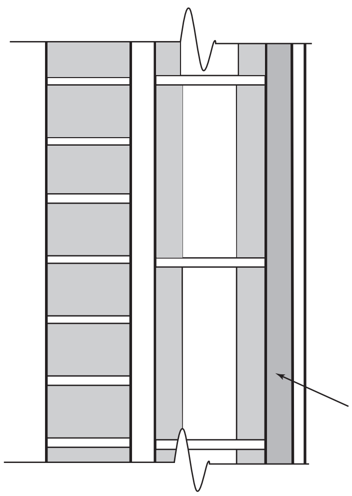
$\text{U-factor}_{average} = 0.85 \times 0.059 + 0.15 \times 0.136 = 0.071$

FIGURE 3.3. *Continued.*



Section at insulation (85% of wall area)

Item	Thickness	r per in.	r for item
Outside air film	-	-	0.17
4" face brick	3-5/8"	0.17	0.62
Air space	-	-	1.00
6" concrete block	-	-	1.20
1-1/2 " polystyrene	1-1/2"	4	6.00
2x2 furring @ 16" o.c.	-	-	-
1/2" drywall	-	-	0.45
Inside air film			0.68
R_{total}			**10.12**
U-factor ($1/R_{total}$)			**0.10**

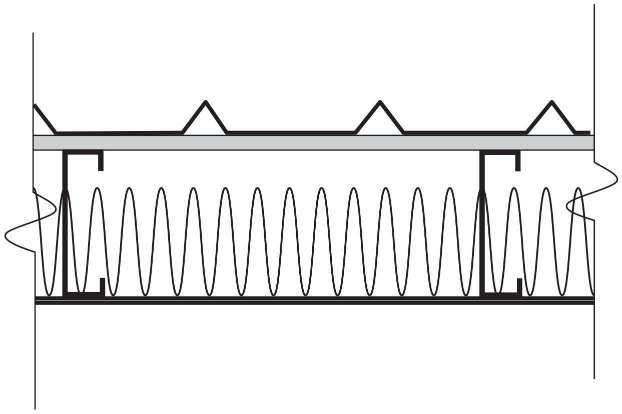


Section at furring (15% of wall area)

Item	Thickness	r per in.	r for item
Outside air film	-	-	0.17
4" face brick	3-5/8"	0.17	0.62
Air space	-	-	1.00
6" concrete block	-	-	1.20
1-1/2" polystyrene	-	-	-
2x2 furring @ 16" o.c.	1-1/2"	1.0	1.50
1/2" drywall	-	-	0.45
Inside air film			0.68
R_{total}			**5.62**
U-factor ($1/R_{total}$)			**0.18**

$\text{U-factor}_{average} = 0.85 \times 0.10 + 0.15 \times 0.18 = 0.11$

FIGURE 3.4. Sample calculation of *U*-factor for masonry cavity wall.



Section at Insulation (85% of roof area)

Item	Thickness	r per in.	r for item
Outside air film	-	-	0.17
Metal roof	-	-	0.00
3/4" plywood	-	-	0.93
Air space	-	-	1.00
6" fiberglass	6"	4	24.00
1/2" drywall	-	-	0.45
Inside air film	-	-	0.61
R_{total}			**27.16**
U-factor ($1/R_{total}$)			**0.037**

Section at Framing (15% of roof area)

Item	Thickness	r per in.	r for item
Outside air film	-	-	0.17
Metal roof	-	-	0.00
3/4" plywood	-	-	0.93
Air space	-	-	0.00
Metal joist	-	-	0.00
1/2" drywall	-	-	0.45
Inside air film	-	-	0.61
R_{total}			**2.16**
U-factor ($1/R_{total}$)			**0.463**

$$\text{U-factor}_{average} = 0.85 \times 0.037 + 0.15 \times 0.463 = 0.101$$

FIGURE 3.5. Sample calculation of *U*-factor for metal roof.

Q = heat flow (Btuh)
1000 = heat (Btu) liberated by condensation of 1 lb of steam

In the early phases of design, most designers prefer to use a simple procedure to estimate infiltration called the *air change method*. Airflow into a space can be estimated in air changes per hour, where the air flow is based on the volume of the *perimeter* space. Air leaks only into spaces with exposure to outside. For instance, a perimeter room 12 ft. deep, 10 ft. wide, and 10 ft. high has a volume of 1200 ft^3. One air change per hour would be 1200 ft^3 per hour, or $1200/60 = 20$ CFM. If the space in question is a deep open office plan, only the perimeter portion of the space (generally considered 10 ft to 15 ft deep) is used in calculating the volume for air change. For instance, a 100 ft. wide, 15 ft. deep, 10 ft. high perimeter area of an office building floor has a volume of 15,000 ft^3. Space interior of the perimeter space is not considered in the estimate. One air change would be 15,000 ft^3/hr, or 250 CFM.

For estimating infiltration, air change rates are assumed on the basis of the engineer's judgment regarding the tightness of the wall, windows, or doors that separate the interior from outside. Winter air change rates estimated for design will generally vary from 0.25 air changes per hour for perimeter rooms of a tight building to 2 air changes per hour for a loosely constructed building. Summer values are generally estimated at half of the winter values due to low wind speeds during hot weather. Exceptional buildings might warrant higher or lower estimates. Doorways to outside will instantly admit very large volumes of

TABLE 3.4 Outdoor Air Requirements for Ventilation, Typical Spaces with Code Occupancy Densities

Minimum Ventilation Rates in Breathing Zone

			Default Values	
Occupancy Category	People Outdoor Air Rate (CFM/person)	Area Outdoor Air Rate (CFM/ft^2)	Occupant Density (#/1000 ft^2)	(CFM/person)
Classrooms (ages 5–8)	10	0.12	25	15
Classrooms (age 9 plus)	10	0.12	35	13
Lecture hall (fixed seats)	7.5	0.06	150	8
Office space	5	0.06	5	17
Reception areas	5	0.06	30	7
Main entry lobbies	5	0.06	10	11
Libraries	5	0.12	10	17
Retail sales	7.5	0.12	15	16
Gym, stadium (play area)	—	0.30	30	—
Spectator areas	7.5	0.06	150	8

Source: Data from ASHRAE Standard 62.1—2013 (www.ashrae.org).

air when they open. The load in this instance is based on the desired "recovery time" rather than the instantaneous load.

3.3.4 Ventilation

Outside air is introduced by the HVAC system to dilute building air contaminants and to make up for exhaust. During cold weather, the air must be heated to the temperature of the space. The ventilation load is calculated by the same equations used for infiltration.

Determining the proper amount of outside air requires analyses of the building's exhaust systems and fresh air requirements for occupancy and a consideration of excess air to pressurize the building slightly and prevent undue infiltration.

Minimum values for outside air are specified by code to maintain acceptable indoor air quality. Model codes generally incorporate recommendations of ASHRAE Standard 62.1. Code requirements for outside air have varied significantly over the years, owing to varying concerns about energy conservation and attention to indoor air quality. Table 3.4 shows the consensus standard for outside air rates used in most current codes. Rates are specified according to usage of space.

3.3.5 Miscellaneous Loads

In addition to conduction, infiltration, and ventilation, heating loads should take into account miscellaneous factors such as losses through walls below grade and slabs on grade. Basement floor and wall losses can be estimated from 2 to 6 and 1 to 3 Btuh per ft^2, respectively, depending on ground water temperature. If performance is critical, as for an occupied church basement assembly area, more precise methods of estimating basement loads are available in ASHRAE Handbook of Fundamentals.

3.3.6 Heating Load Problem 3.1

This sample problem demonstrates methods for calculating heating and humidification loads for a small office building,

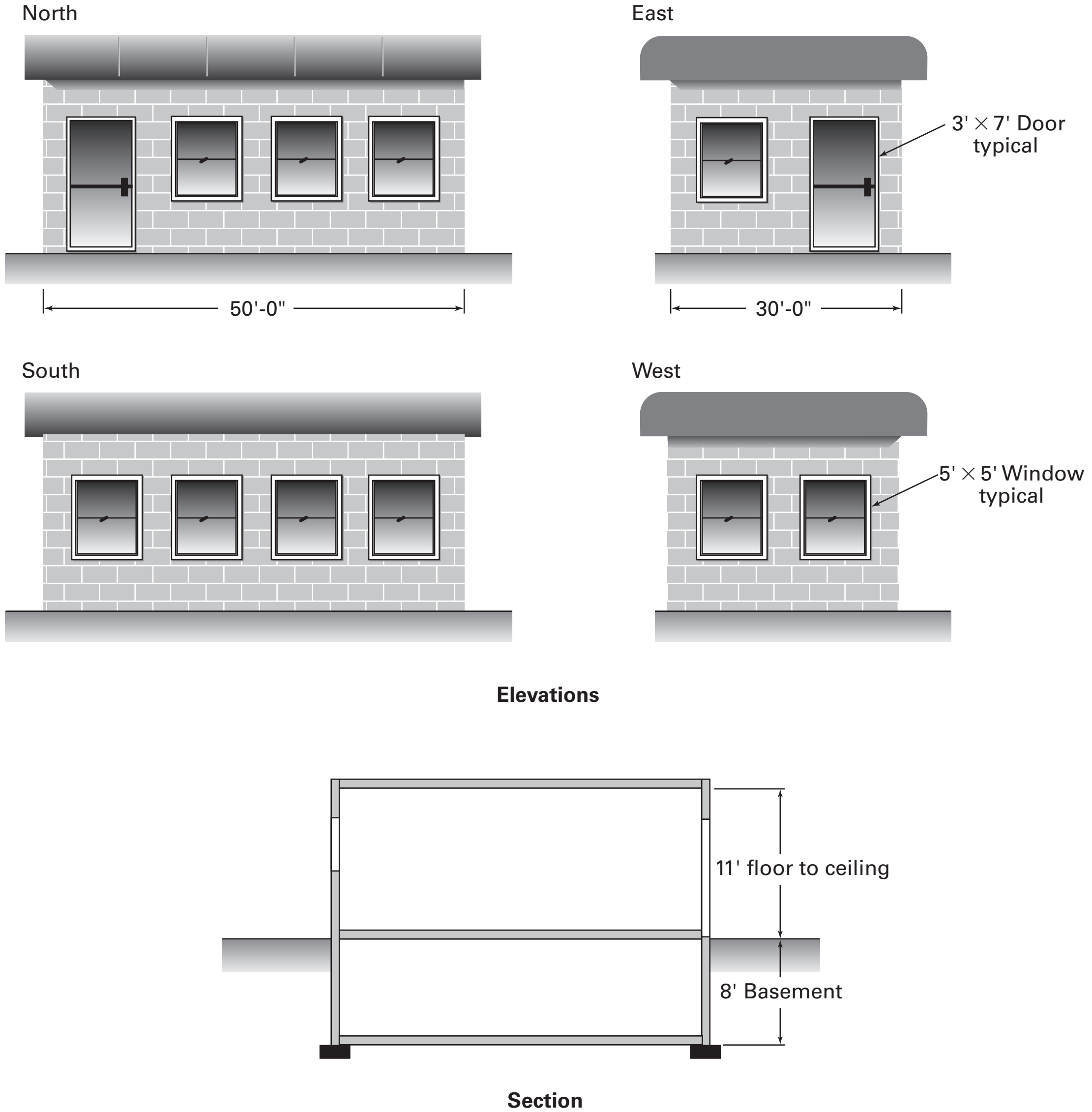


FIGURE 3.6. Elevations and section of the building, Problem 3.1. *U*-factors are as follows: walls, 0.064; roof, 0.32; windows, 0.35; and doors, 0.75.

CALCULATIONS FOR PROBLEM 3.1 ▼

Load Components			Load (Btuh)
Roof		$Q = U \times A \times \text{TD}$	
		$Q = 0.032 \times (30 \times 50) \times (72 - 8)$	3070
Walls[#]		$Q = U \times A \times \text{TD}$	
	North	$Q = 0.064 \times (11 \times 50) \times (72 - 8)$	2250
	South	$Q = 0.064 \times (11 \times 50) \times (72 - 8)$	2250
	East	$Q = 0.064 \times (11 \times 30) \times (72 - 8)$	1350
	West	$Q = 0.064 \times (11 \times 30) \times (72 - 8)$	1350
Doors		$Q = U \times A \times \text{TD}$	
	North	$Q = 0.75 \times (3 \times 7) \times (72 - 8)$	1010
	East	$Q = 0.75 \times (3 \times 7) \times (72 - 8)$	1010
Windows		$Q = U \times A \times \text{TD}$	
	North	$Q = 0.35 \times (3 \times 5 \times 5) \times (72 - 8)$	1680
	South	$Q = 0.35 \times (4 \times 5 \times 5) \times (72 - 8)$	2240
	East	$Q = 0.35 \times (1 \times 5 \times 5) \times (72 - 8)$	560
	West	$Q = 0.35 \times (2 \times 5 \times 5) \times (72 - 8)$	1120
Basement floor		$Q = \text{Btuh}/\text{ft}^2 \times \text{area}$	
		$Q = 3 \times (30 \times 50)$	4500
Basement walls		$Q = \text{Btuh}/\text{ft}^2 \times \text{area}$	
		$Q = 6.0 \times (8 \times (30 + 50 + 30 + 50))$	7680
Infiltration, sensible only		$Q = 1.1 \times \text{CFM} \times \text{TD}$	
		CFM = (air exchanges per hour × volume)/60 minutes per hour	
		$Q = 1.1 \times ((1.5 \times (30' \times 50' \times 10'))/60) \times (72 - 8)$	26,400
Ventilation, sensible only		$Q = 1.1 \times \text{CFM} \times \text{TD}$	
		CFM = 500	
		$Q = 1.1 \times 500 \times (72 - 8)$	35,200
		Total Heat Loss =	**91,700 Btuh**
Humidification (Optional)			
Infiltration air		$Q = 4840 \times \text{CFM} \times (W_{\text{room}} - W_{\text{oa}})$	
		CFM = (air exchanges per hour × volume)/60 minutes per hour	
		$Q = 4840 \times ((1.5 \times (30' \times 50' \times 10'))/60) \times (0.005 - 0)$	9,080
Ventilation air		$Q = 4840 \times \text{CFM} \times (W_{\text{room}} - W_{\text{oa}})$	
		CFM = (air exchanges per hour × volume)/60 minutes per hour	
		$Q = 4840 \times 500 \times (0.005 - 0)$	12,100
		Total Humidification of Outside Air =	**21,200 Btuh (21.2 lbs./hr. steam)**

[#]For simplicity, areas of doors and windows are not deducted from wall area calculations in this example. Adjustment is appropriate if doors and window areas are significant in comparison with the respective wall areas.

defined in Fig. 3.6. The criteria and physical properties of the building are as follows:

- Design conditions—indoor (72°F, 30% RH), outdoor (8°F, near 0% RH)
- Infiltration—1.5 air changes per hour
- Ventilation—500 CFM (based on exhaust; Table 3.4 would require only 140 CFM)

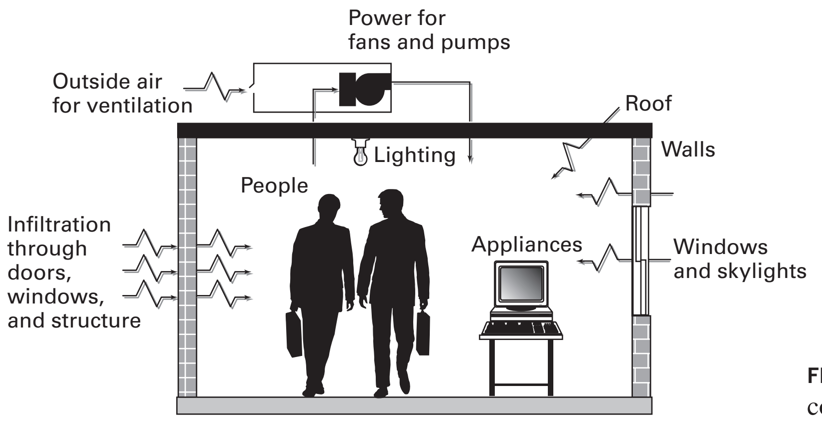


FIGURE 3.7. Components of building cooling loads.

3.4 CALCULATING COOLING LOADS

The first step in sizing air-conditioning equipment is to calculate loads. Heat gains include conduction, solar effects, outside air loads, and internal heat loads, as illustrated in Fig. 3.7.

3.4.1 Conducted and Solar Heat Through Glazing

Solar and conducted effects must be considered in estimating heat gains through windows and skylights. These heat loads are considered in two parts—simple conduction and Conduction and solar heat gains must be considered in estimating cooling loads through windows and skylights. Equation (3.5) applies:

$$Q = U \times A \times \mathrm{TD} + \mathrm{SC} \times A \times \mathrm{SHGF} \qquad \textbf{(3.5)}$$

where SHGF = solar heat gain factor (Btuh/ft^2)

SC = shading coefficient (dimensionless)

The term "$U \times$ A $\times$ TD" represents the conducted heat gain, and the term "SC $\times$ A $\times$ SHGF" represents the solar heat gain.

The *shading coefficient* (SC) is a property of the glazing material and accessories such as blinds or draperies. SC is the ratio of solar heat admitted in comparison with what is admitted by clear single-strength glass, which has a shading coefficient of 1.0. Thus, a glazing with 0.5 SC allows only half as much solar heat into the space as clear single-strength glass does. Shading coefficients of 0.2 or lower can be achieved with heavily reflective films or coatings.

Low shading coefficients can be accompanied by low visible light transmission (VT). High VT is desirable for daylight and view through windows. Sacrificing VT to achieve low SC, or SHGC is discouraged by ANSI/ASHRAE/IES Std. 90.1 by setting a lower limit on the ratio of VT to SHGC of 1.10. Accordingly, some glazing material options are not appropriate for buildings under the jurisdiction of this document. However, lower VT might be advantageous for skylights to prevent excessive glare.

Table 3.5 gives properties for selected glazing materials (nonresidential, nonmetal framing).

Glass manufacturers also publish *solar heat gain coefficient* (SHGC) for their products. SHGC is the ratio of energy transmitted to energy incident on the glazing material. SHGC is equal to 86% of shading coefficient (SC).

Thermal Properties of Glass

	Glass Properties				
Type of Glass	***U*-Factor**	**SC**	**SHGC**	**VT**	**VT/SHGC**
Single strength clear glass	1.05	1.00	0.86	0.90	1.05
1″ insulating glass, 1/4″ lites, clear	0.47	0.80	0.70	0.79	1.13
ANSI/ASHRAE/IES Std. 90.1-2013 Zone 1 (southern US) minimum performance	0.50	0.29	0.25	0.28	1.10
ANSI/ASHRAE/IES Std. 90.1-2013 Zone 4 (middle US) mínimum performance	0.35	0.46	0.40	0.44	1.10
ANSI/ASHRAE/IES Std. 90.1-2013 Zone 6 (northern US) minimum performance	0.32	0.46	0.40	0.44	1.10
Solar control double glazed clear with lo-E coating	0.29	0.44	0.38	0.70	1.84
Low solar gain double glazed reflective with lo-E coating and argón fill	0.24	0.20	0.17	0.14	0.82

SC = shading coefficient
SHGC = solar heat gain coefficient
VT = visible transmittance

Solar Heat Gain Factors (SHGF) for Estimating Peak Room Loads and Peak Block Loads

Orientation	Peak value	Time of peak	4:00 PM value
North	38	Noon	28
South	109	Noon	29
East	216	08:00 AM	26
West	216	04:00 AM	216
Horizontal	267	Noon	153

The *solar heat gain factor* (SHGF) is the amount of solar heat in Btuh/ft^2 that will enter a clear single-pane window at a given latitude, time of year, and time of day, facing a specific orientation.

Solar loads from windows are generally the largest component of overall envelope loads, and, as such, are important to consider in formulating load management strategies as covered in Chapter 4. Plotted values of SHGF are shown in Fig. 4.4. Values relevant to most load estimating problems are shown in Table 3.6 for July 21, 40°N.

For most buildings at 40° north latitude the peak overall cooling load occurs around 4:00 PM when window solar loads are highest due to intense heating from the western sun. However, loads in individual rooms or zones which face other directions will exhibit their peak loads at other times of day. Rooms facing east will have their peak loads occur at 8:00 AM when the eastern solar heat is greatest. Rooms facing south and north will experience peak loads at noon. In order to estimate the peak overall building load, referred to as "block load," values of SHGF are given for 4:00 PM for all orientations. In order to estimate peak room loads, SHGF is given for the peak values experienced for the various orientations.

To use Table 3.6 for estimating solar load for a building block load, apply the 4:00 PM values for all orientations. To estimate a room load (generally one orientation), apply the peak value for the window(s) facing that orientation. For instance, if the room faces south, use 109 Btuh/ft^2 for the south facing window in the room. If a corner room has glass facing two orientations, determine which glass orientation has the higher load at the time of the peak value for that orientation. A good answer will generally result from using the peak value for this orientation and the 4:00 PM value for the other.

One observation should be made regarding south facing glass. Peak cooling loads are generally assumed to occur on July 21, but south facing walls with significant glass areas may exhibit peak load on a warm day in December. The low southern sun on or about the solstice has an SHGF of over 250 Btuh/ft^2 (see Fig. 4.4f).

3.4.2 Conduction Through Walls and Roofs

In the estimation of cooling loads due to walls and roofs, a simple temperature difference between inside and outside air will not account for solar heat. The outside surface of a wall or roof may be much warmer than the surrounding air, owing to solar effects. Accordingly, conduction through walls and roofs is estimated by equations using a total equivalent temperature difference (TETD) that includes solar effects as well as temperature difference. The value of TETD will vary with the latitude, orientation, time of day, absorption property of the surface, and thermal mass of the building assembly. Thermal mass affects the timing of heat entry to the interior.

Light thermal mass would be associated with wood or metal frame construction or curtain wall. Medium thermal mass would typically be the case if masonry veneer were added to the assembly. Heavy thermal mass would be associated with concrete or concrete masonry units and brick veneer. For roofs, light thermal mass would be associated with metal deck. An example of medium thermal mass would be light weight concrete over metal deck. Heavy thermal mass would be structural concrete deck or precast concrete planks.

For walls and roofs, the cooling load is calculated by using the equation

$$Q = U \times A \times TETD \qquad \textbf{(3.6)}$$

where TETD = total equivalent temperature difference (°F)

Generally, wall and roof loads are much less than loads for windows. Due to high window loads and the influence of west sun, most buildings experience peak block load at 4:00 PM as stated earlier. Thus the wall loads at 4:00 PM are of interest for their contribution to the peak block load for the building. For east facing rooms or zones, the peak room load will occur at 8:00 AM due to the influence of eastern sun through east facing glass. Thus the load due to TETD at 8:00 AM is of interest for east facing rooms. Similarly, the noon TETD values for south and north walls are of interest in determining the peak loads for rooms facing these orientations. The 4:00 PM TETD value is appropriate for west facing rooms. TETD values for estimating peak block load and room loads are shown in Table 3.7.

3.4.3 Infiltration and Ventilation

Generally, the amount of air infiltration is much lower during hot weather than during cold weather. This is because winds are milder, and lower temperature differentials cause less of a chimney effect. Accordingly, air change rates should be estimated lower for summer than for winter.

Infiltration loads have two components: sensible and latent. The equations governing such loads are identical to those cited earlier for heating load calculations:

$$Q_{\text{sensible}} = 1.1 \times \text{CFM} \times \text{TD} \qquad \textbf{(3.7)}$$

$$Q_{\text{latent}} = 4840 \times \text{CFM} \times (W_{\text{final}} - W_{\text{initial}}) \qquad \textbf{(3.8)}$$

Total heat, termed *enthalpy (H)*, is the sum of sensible and latent heat. The total heat of air at various conditions of temperature and humidity can be taken from a psychrometric chart or tables, and the following equation can be used to determine energy flow:

$$Q = 4.5 \times \text{CFM} \times \Delta H \qquad \textbf{(3.9)}$$

where ΔH = change in enthalpy (Btu/lb of air)

Outside air introduced for ventilation by the air-conditioning equipment will result in sensible and latent loads calculated according to the same equations. Recommended minimum ventilation rates are

Total Equivalent Temperature Differences (TETD) for Estimating Peak Room Loads and Peak Block Load

Orientation	Thermal mass	Peak	Time of peak	8:00 AM value	Noon value	4:00 PM value
North	Light	21	4:00 PM	14	17	21
	Medium	25	8:00 PM	6	10	19
	Heavy	23	10:00 PM	8	9	15
South	Light	33	2:00 PM	14	28	31
	Medium	38	6:00 PM	5	12	34
	Heavy	33	8:00 PM	9	10	26
East	Light	41	10:00 PM	32	37	28
	Medium	42	2:00 PM	9	38	40
	Heavy	37	6:00 PM	10	28	37
West	Light	48	6:00 PM	17	24	42
	Medium	55	8:00 PM	8	9	25
	Heavy	46	10:00 PM	14	10	19
Roof	Light	96	2:00 PM	24	88	81
	Medium	58	4:00 PM	6	28	58
	Heavy	53	6:00 PM	13	26	48

shown in Table 3.4. When determining ventilation requirements, no credit can be taken for infiltration, which will vary depending on wind and temperature, and how well the building is sealed.

3.4.4 Internal Heat Gains

Heat is generated inside buildings by lights, appliances, and people. People liberate both sensible and latent heat. Latent heat results from exhaled moisture and evaporation of perspiration. Loads will depend on the level of activity. For normal office activities many engineers use 250 Btuh sensible and 250 Btuh latent per occupant. The ASHRAE Handbook of Fundamentals provides data for a wide range of activities and building types.

Heat from lights and appliances can be calculated using the factor for conversion of electrical to thermal energy:

$$Q = 3.41 \times P \qquad \textbf{(3.10)}$$

where P = power input to light fixture or appliance (watts)

Often, a precise figure for the heat from building lighting and appliances is unavailable at the time the air-conditioning system is being designed. In that case, an allowance is assumed in watts per square foot of building or room area. Guidelines for estimating electrical loads for common types of buildings are given in Chapter 14, which covers Electrical Design and Wiring.

3.4.5 Loads in Return Air Plenums

If lighting is recessed in a ceiling that is used as a return air plenum, heat from the back side of the fixtures will not enter the occupied space. Heat to the plenum will still need to be removed by the air-conditioning system but will not affect the amount of supply air delivered to cool the occupied space. Similar adjustments should be made for roof loads that occur above return air plenum ceilings. The actual load to the space and required supply air quantity should include conduction from the plenum through the ceiling. For the overall system load there is an equal offset to the load in the plenum. Space loads and plenum loads can be distinguished by comparing the two cases shown in Fig. 3.14.

3.4.6 Sample Problems 3.2 and 3.3

The following problems illustrate the calculation of cooling loads for a small office building, shown in Fig. 3.9. Problem 3.2 is a load calculation for the building with direct return as shown in Fig. 3.10(b). Problem 3.3 is a load calculation for the building with a return air plenum ceiling and shown in Fig. 3.10(a). Criteria and physical properties of the building are as follows:

- Design conditions: inside (78°F, 50% RH); outside (95°F DB and 78°F WB).
- Construction: wall, masonry (heavy thermal mass); *U*-factor, 0.064 roof— insulation over metal deck (light thermal mass); *U*-factor, 0.032 windows—*U*-factor, 0.35; shading coefficient, 0.46 doors—*U*-factor, 0.75 ceiling (Problem 3.3 only)—*U*-factor, 0.30.
- Outside air: infiltration, ½ air exchange per hour; ventilation, 500 CFM (based on general exhaust; code required minimum per Table 3.4 would be only 140 CFM).
- Lighting: 30 Fluorescent fixtures 60 W each all heat to occupied space in Problem 3.2 and 50% in Problem 3.3.
- Appliances: allowance of 1.5 W per square foot.
- Occupants: (10) adults, general office work.

Completing the problems shows that the overall air-conditioning load will be about the same, regardless of whether there is a return air plenum; however, the load in the occupied space will be considerably lower if the design includes a return air plenum. This will have an important effect on the amount of supply air required for cooling.

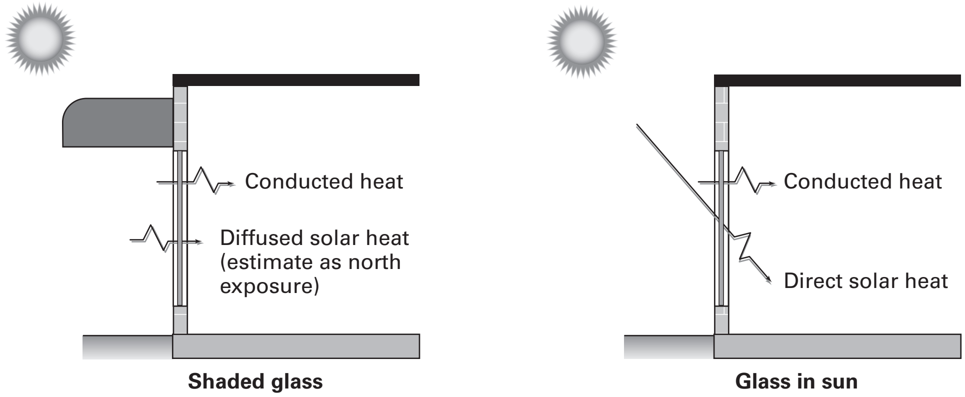


FIGURE 3.8. Heat gain through glazing.

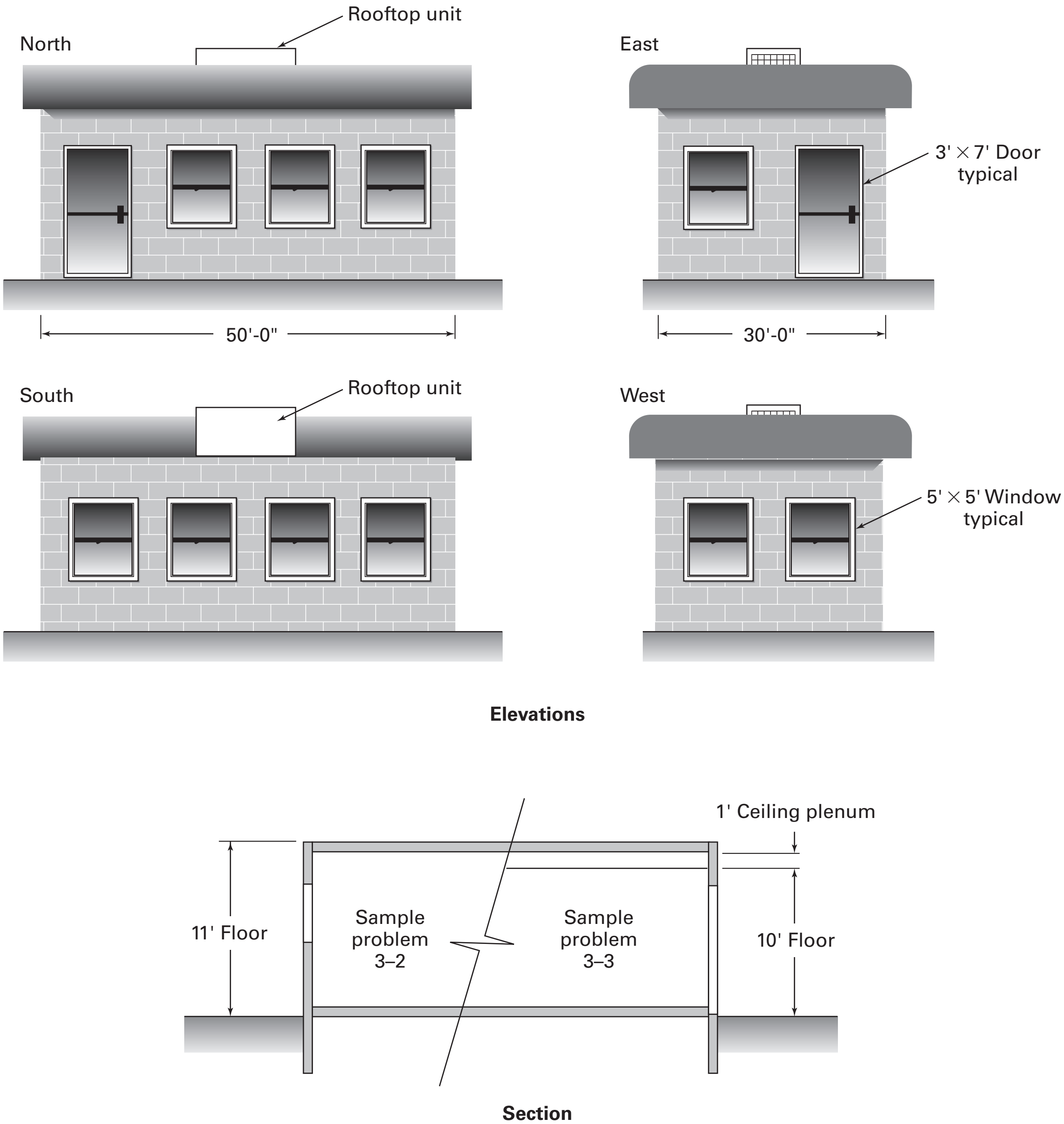
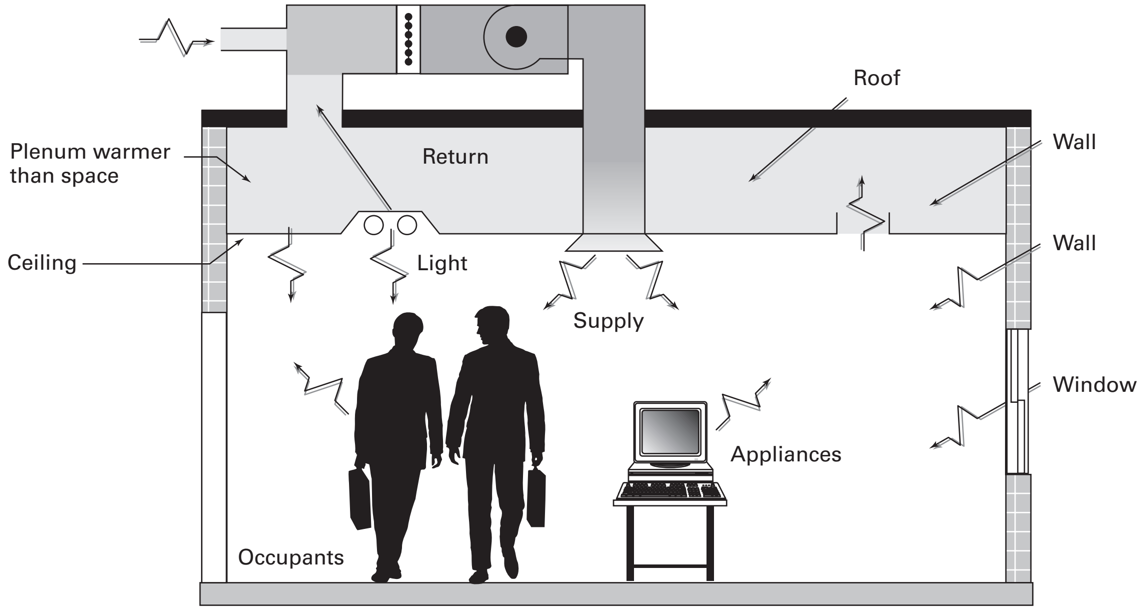
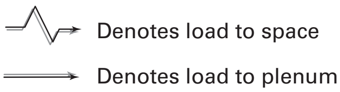
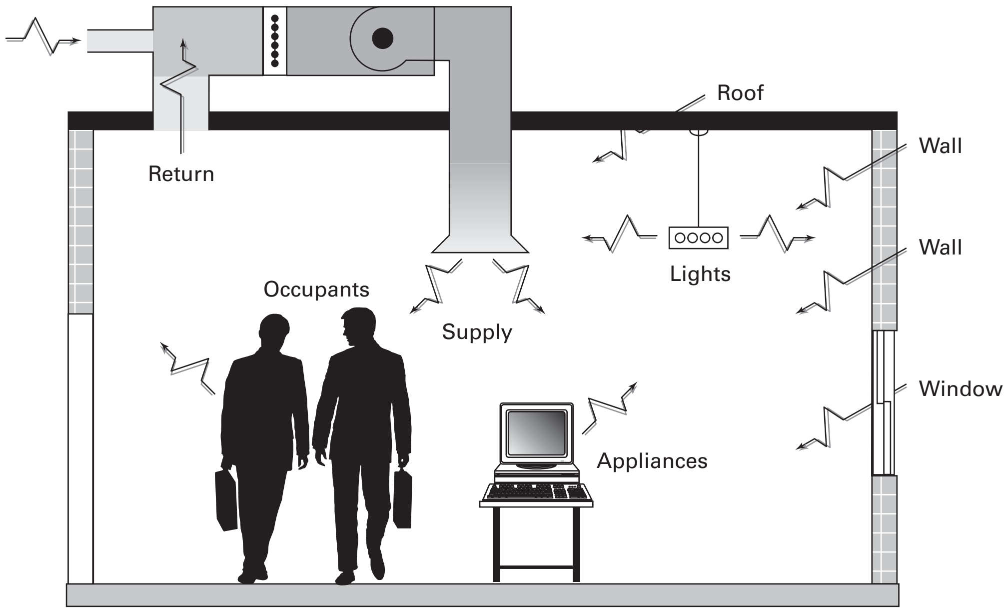


FIGURE 3.9. A small office building.



(a) **Room with return air plenum**





Room with direct return

(b)

FIGURE 3.10. Effect of return air plenum on load to conditioned space. (a) Room with return air plenum. (b) Room with direct return.