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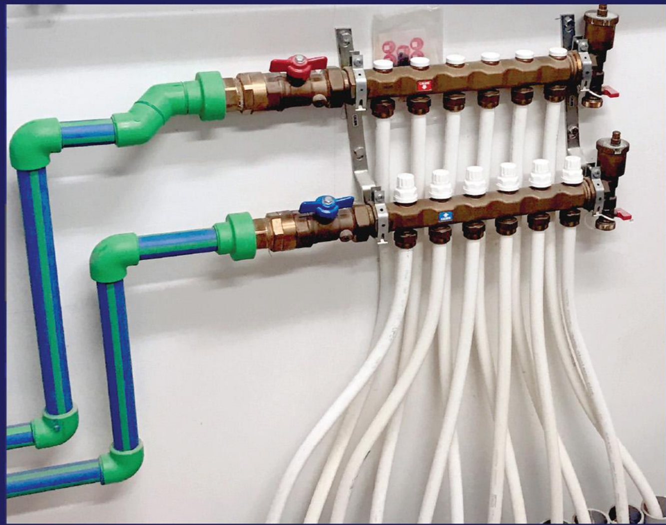
FOURTH EDITION

Modern Hydronic Heating & *Cooling*

for residential and light commercial buildings

John Siegenthaler, P.E.



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Modern Hydronic Heating & *Cooling*

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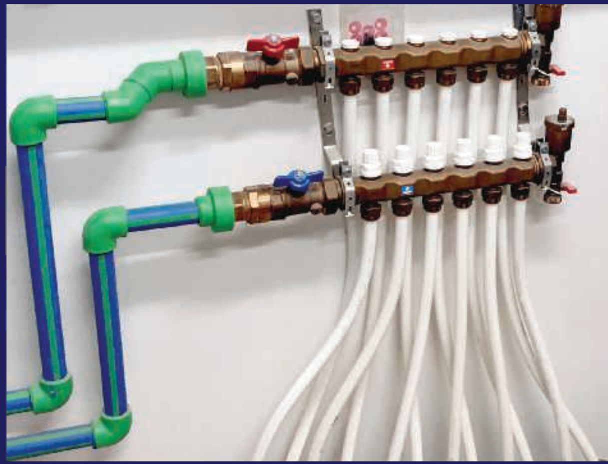

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Modern Hydronic Heating and Cooling: For Residential and Light Commercial Buildings, **4e**
John Siegenthaler, P.E.

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Preface

This book, like its preceding editions, focusses on the design and installation of state-of-the-art hydronic heating systems for residential and light commercial buildings. It was written to provide comprehensive, contemporary, and unbiased information for heating technology students, as well as heating professionals.

I believe that residential and light commercial buildings deserve better comfort systems than they often get. After all, even smaller heating systems affect the comfort and well being of countless people over many years. The information in this book is for those wanting to design and install superior heating systems for such buildings. Systems that are efficient, reliable, and deliver unsurpassed comfort.

For decades, much of the available information on designing hydronic systems has been aimed at engineers, and intended for use in larger buildings. It is often impractical and imprudent to scale such systems down for use in smaller buildings.

Manufacturer's information, while accurately representing products, seldom leads a heating professional through the entire design process.

Most heating systems for smaller buildings are designed by the same firm that performs the installation. Over the last four decades, I've had opportunity to see both ends of the design and installation spectrum. From systems that are "textbook examples" of proper design and consummate craftsmanship, to multithousand dollar assemblies of hardware that will never deliver what is expected of them.

One of the consistent differences between these successes and failures has been a commitment by the designer/installer to go beyond joining pipes and turning wrenches, to learn the fundamental principles at work in every hydronic system. Without this commitment installers can quickly slide into design stagnation, where they approach every project with the same method regardless of its appropriateness, or just rely on hardware suppliers to sketch out a system that they can assemble. They often fail to capitalize on the versatility of modern hydronics technology and the many profitable opportunities it offers.

Since the third edition was released in 2011, there have been many new developments in the hardware and design concepts that now represent cutting-edge hydronics technology. This edition reflects those changes. It emphasizes simplicity wherever possible, and conservative use of distribution energy to move heat through hydronic distribution systems.

Interest in renewable energy, and decarbonization has also grown exponentially since the release of the previous edition. A thorough examination of renewable energy heating systems reveals that *hydronics technology is the "glue" that holds nearly all these systems together.* Although heat sources such as solar thermal collectors, geothermal heat pumps, air-to-water heat pumps, and solid fuel boilers are the "engines" of such systems, hydronics technology is the "drivetrain" that delivers the heat when and where it's needed. Without proper hydronic detailing none of these renewable energy heat sources will perform up to their full potential.

Current global trends aimed at phasing out fossil fuels, and replacing them with renewably-sourced electricity, will profoundly impact the future of hydronics technology. Heat pumps, in particular, will continue to gain market share against fossil fuel boilers as heat sources for residential and light commercial systems.

The vast majority of heat pump capable of supplying warm water for hydronic heating can also supply chilled water for cooling. This is a "game changer" when it comes to the potential for hydronic-based systems to supply the *year-round* comfort requirements of smaller buildings. It eliminate a shortfall that has constrained the potential growth of hydronics technology for decades - the complexity and cost of installing a completely separate cooling system in a building that uses hydronic heating.

This edition recognizes the emerging market for hydronic cooling in smaller buildings. To that end it includes a new chapter that covers the fundamentals of small scale chilled water cooling.

Other additions to this fourth edition include:

- Broader applications for variable speed circulators
- Discussion of new piping materials and joining methods
- Application of air-to-water heat pumps
- Improving water quality in hydronic systems
- Increased use of modern air, dirt, and magnetic particle separator technology
- Additional buffer tank piping options
- Use of panel radiators at lower water temperatures
- Extended analysis and applications for plate heat exchangers

Organization

The initial chapters of this book acquaint readers with the fundamental physical processes involved in hydronic heating. Topics such as basic heat transfer, heating load calculations, and properties of fluids are discussed. A solid understanding of these basics is essential for both design and troubleshooting.

Later chapters use the fundamental principles for overall system design. Readers are referred to relevant sections of earlier chapters during the design process to reinforce the importance of these principles. Example systems at the end of some of the later chapters show complete system piping and control wiring diagrams.

Chapter 1 provides a summary of hydronic heating, and the basic concepts involved. It emphasizes comfort as the ultimate goal of the heating professional. It encourages an attitude of craftsmanship and professionalism as the reader continues through the text.

Chapter 2 covers heating load estimating calculations. Experience indicates that such calculations can be a stumbling block for students, as well as for those in the trade, who want to jump right into design and layout without first determining what the system needs to provide. Without proper load information, any type of heating system can fail to deliver the desired comfort. A complete method for determining design heating loads is presented.

Chapter 3 surveys a wide spectrum of hydronic heat sources. These include conventional gas and oil-fired boilers, as well as contemporary devices such as geothermal heat pumps, air-to-water heat pumps, modulating/condensing boilers, pellet-fueled boilers, cordwood gasification boilers, and solar thermal collectors. Emphasis is placed on the need to match the balance of the system to the operating requirements of these heat sources.

Chapter 4 describes, in simple terms, the physical properties of water including specific heat, density, viscosity, incompressibility and solubility of air. Two very important equations that relate heat and heat flow to water temperature and flow rate are introduced. It also covers water quality and the use of demineralized water.

Chapter 5 is a "show and tell" chapter covering the proper use of tubing, fittings, and valves in hydronic systems. The use of copper tubing as well as materials such as PEX and PEX-AL-PEX is discussed. Common valves are illustrated and their proper use is emphasized. Several specialty fittings and valves are also discussed.

Chapter 6 remains as the key analytical chapter of the text. It defines vital parameters associated with the hydraulic performance of any hydronic system. It also introduces a method for calculating the head loss of a fluid as it flows through a piping system. This method uses the concept of *hydraulic resistance* to build an analogy between fluid flow in piping circuits and the principles of current, voltage, and resistance in electrical circuits. This method can be applied to both simple and complex piping circuits.

Chapter 7 presents qualitative and quantitative information on circulators for hydronic systems. The pump curve is introduced, and uses to properly match a circulator to a piping system. The intersection of the pump curve of a candidate circulator, and system head loss curve of the piping system, reveals the operating flow rate of the circulator/piping system combination. Quantitative and graphical methods are shown for finding this point. The chapter also places strong emphasis on what circulator cavitation is, and how to avoid it. The fourth edition presents the latest information on variable speed pressure-regulated circulators with electronically commutated motors. These circulators represent a major shift in capability and energy efficiency, and are quickly becoming the “new normal” in residential and light commercial hydronic systems, It is essential that heating professional understand how to apply them.

Chapter 8 surveys several types of hydronic heat emitters including finned-tube baseboard convectors, fan-coils, panel radiators, and radiant baseboards. The advantages and disadvantages of each type are discussed. Performance and sizing information is also given. The fourth edition presents new methods for analyzing the performance of panel radiators at lower water temperatures, allowing them to be successfully used with contemporary hydronic heat sources.

Chapter 9 is a major chapter dealing with control components and systems for hydronic heating. Control systems for multiload/multitemperature systems are discussed. Ladder diagrams are used as a framework on which to design such control systems. The continued growth of wireless and Internet-based control technology is reflected in this fourth edition.

Chapter 10 is devoted entirely to hydronic radiant panel heating. This edition includes coverage of radiant floors, walls and ceilings including new materials and thermographic images of operating panels.

Chapter 11 covers several types of distribution piping configurations including series loop, one-pipe diverter tee, two-pipe reverse return, homerun, and primary/secondary systems. The analytical methods first introduced in Chapters 6 and 7 are reference through step by step design procedures for a wide range of distribution systems. The fundamental concepts of hydraulic separation and distribution efficiency are emphasized.

Chapters 12 and **13** deal with the specialized topics of expansion tanks and air removal. Both survey the latest types of hardware, and show how to properly select it. The fourth edition includes new material on dirt and magnetic particle separation technology.

Chapter 14 covers several miscellaneous topics. Auxiliary loads such as indirect water heating, pool/spa heating, and intermittent garage heating are discussed. Extensive material on the design and application of hydronic snow and ice melting (SIM) systems is also presented. Unique concepts for incorporating buffer tanks are presented. The fourth edition provides expanded coverage of flat plate heat exchangers, and heat metering technology.

Chapter 15 is entirely new to the fourth edition. It covers the fundamentals of chilled water cooling and how it can be applied in residential and light commercial buildings. Topics include the thermodynamic properties of air, basic psychometrics, performance of chillers, importance of pipe insulation, discussion of hardware suitable for chilled water cooling, and a range of example systems.

Design Assistance Software

This edition references several software products for expediting calculations and documenting hydronic system designs. These softwares include:

- Hydronics Design Studio
- Heat Load Pro
- HydroSketch

The ***Hydronics Design Studio*** is built around the analytical methods presented in this book. It allows rapid simulation of flow in user-defined piping systems including those with multiple parallel branch circuits. It also includes modules for room heat loss estimating, expansion tank sizing, and detailed simulation and sizing of series-loop baseboard systems. Several examples of how the software can be used to evaluate long numerical calculations are given.

Heat load Pro is a subset of the Hydronics Design Studio. It was created as a simple, low-cost tool for calculating building heating loads, and estimating the operating costs associated with different fuel types and heat sources.

HydroSketch is a cloud-based tool for quickly constructing piping and electrical schematic diagrams. It includes many of the component symbols given in Appendix A and used throughout this book. HydroSketch uses a simple "drag and drop" interface to quickly place component symbols on a drawing canvas and connect them into circuits.

More information on these software products is available at www.hydronicpros.com

Other Features of the Fourth Edition

- Updated photographs and illustrations show more installation details and proper use of the latest hydronic hardware.
- A full description of all variables, along with the required units, is given whenever an equation is introduced. All quantities are expressed in the (IP) units commonly used in much of the North American HVAC trade.
- Many example calculations are organized as a situation statement, solution procedure, and discussion of the results.
- A summary of key terms is given at the end of each chapter. An extended glossary is provided for quick reference to concise descriptions of key terms. Readers are encouraged to test their understanding of each chapter by briefly describing each of these key terms.
- Additional questions and exercises have been developed for several chapters.

Previous editions of this text have been used by the author as the basis of a one semester course on hydronic heating. This course is offered in the second year of a two-year Associate Degree program in Air Conditioning Technology at Mohawk Valley Community College in Utica, NY.

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My appreciation is extended to the many manufacturers who have supplied images and information for this edition. Their cooperation, and willingness to accommodate what have often been tight schedules has been truly outstanding.

Finally, I wish to thank the Cengage Learning production team, and especially Vanessa Myers, Jennifer Alverson, Jennifer Ziegler, and Kimberly Klotz for their diligence and patience as the fourth edition has taken form.

Dedication

This edition is dedicated to Mario Restive, an accomplished engineer, professional associate at Mohawk Valley Community College, and long time friend. Mario worked diligently to transform ideas and algorithms for the previously mentioned software tools into thousand of lines finished code. Without his skill this key companion to the text would not exist. His loyal friendship over the last 40 years is deeply appreciated.

John Siegenthaler, P.E.

Chapter 1

Fundamental Concepts

Objectives

After studying this chapter, you should be able to:

- Describe the advantages of hydronic heating.
- Define heat and describe how it is measured.
- Describe three methods by which heat travels.
- Explain thermal equilibrium within a hydronic heating system.
- Define four basic hydronic subsystems.
- Explain the differences between a radiator and a convector.
- Explain the differences between an open-loop and a closed-loop hydronic system.
- Summarize the basic components of a hydronic heating system and explain how they operate.
- Explain the concept of distribution efficiency and use it to compare different distribution systems.

1.1 What Is a Hydronic Heating System?

Hydronic heating systems use water (or water-based solutions) to move thermal energy from where it is produced to where it is needed. The water within the system is neither the source of the heat nor its destination, only its "conveyor belt." **Thermal energy** is absorbed by the water at a **heat source**, conveyed by the water through the **distribution system**, and finally released into a heated space by a **heat emitter**. Ideally, the same water remains in the system year after year.

Hydronic systems are not limited to heating. They can also be used to convey cooling effect (e.g., the lack of heat) from a source of chilled water to cooling emitters located in one or more locations within a building.

Water has many characteristics that make it ideal for heating and cooling applications. It is readily available, nontoxic, nonflammable, and has one of the highest heat storage abilities of any material known to man. All three states of water (solid, liquid, and vapor) are used for various building heating and cooling applications. The modern hydronic systems discussed in this book make use of the liquid state only.

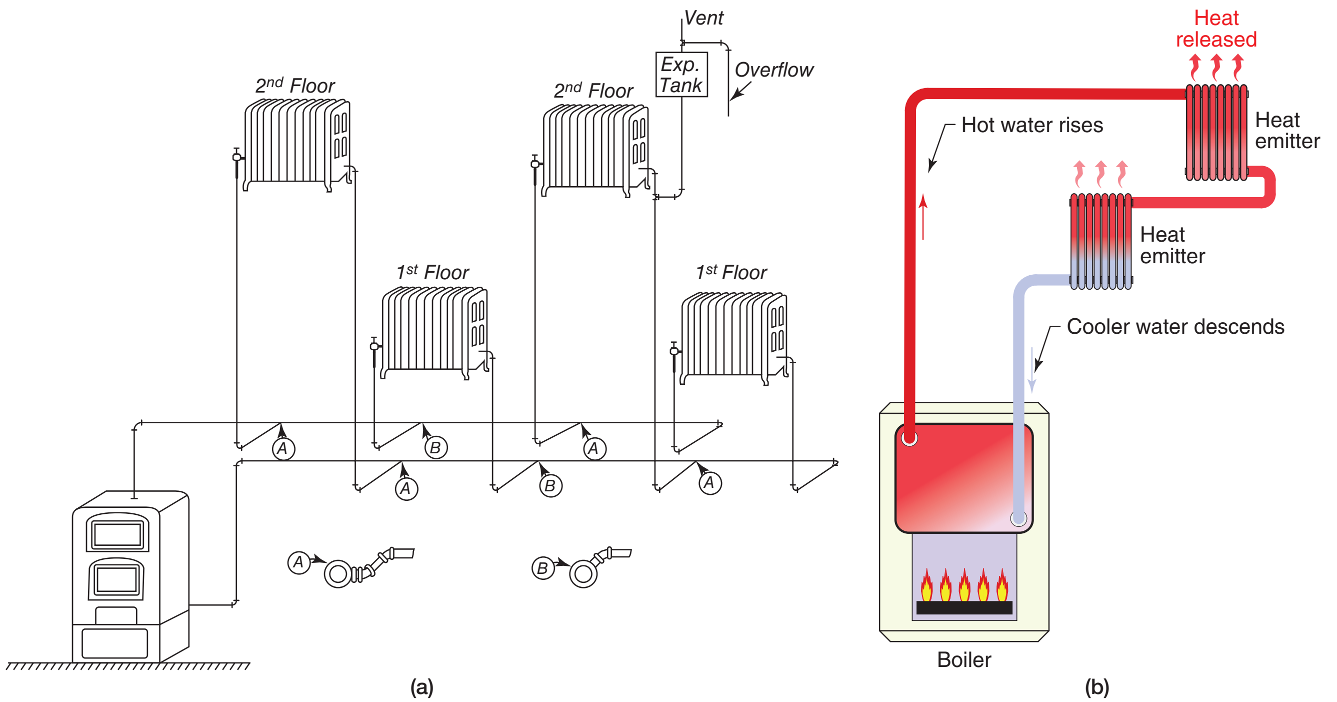


Figure 1-1 (a) An early (1930s vintage) hydronic heating system without a circulator. (b) Flow occurs due to differences in density between hot water and cooler water.

The practical temperature range for water in residential and light commercial buildings is from about 32 to 250 °F.

At the upper end of this range, the water is maintained in a liquid state by system pressurization. The lower end of the range can be extended well below 32 °F by the addition of antifreeze. Such a solution is called **brine**. It would be used in special applications such as hydronic snowmelting or the earth loop of a geothermal heat pump system.

Early hydronic systems, as depicted in Figure 1-1a and 1-1b, relied on the **buoyancy** of hot water to move water between the boiler and the heat emitters. Because of its lower density, hot water would rise upward from a boiler through supply pipes into heat emitters. After releasing heat, the now slightly denser water flows downward back to the boiler, as shown in Figure 1-1b.

These early hydronic heating systems required careful pipe sizing and installation since buoyancy-driven flows are weak, and their designs were significantly limited in comparison to what can be done with modern methods. The emergence of electrically powered **circulators** made it possible to move water at higher flow rates through much more elaborate piping systems.

Modern hydronics technology enables heat to be delivered precisely when and where it is needed. Hundreds of system configurations are possible, each capable of meeting the exact comfort requirements of its owner. Some may be as simple as a boiler serving a single piping circuit through several series-connected heat emitters. Others may use two or more hydronic heat sources operated in stages, releasing their heat through a wide assortment of heat emitters. Those same heat sources could also provide the building's **domestic hot water**. They might even heat the swimming pool or melt snow as it falls on the driveway. If the heat source is a reversible heat pump, it could also supply chilled water for cooling. Well-designed and properly installed hydronic systems provide unparalleled versatility, unsurpassed comfort, and fuel efficiency for the life of the building.

1.2 Benefits of Hydronic Heating

This section discusses several benefits of hydronic heating. Among these are

- Comfort
- Energy savings
- Design flexibility
- Clean operation
- Quiet operation
- Noninvasive installation
- High distribution efficiency

Comfort

Contrary to common belief, heating systems are not created to heat buildings. Instead, they are created for sustaining human thermal comfort within buildings. Think about it: Does a window, concrete block, or insulation batt really "care" whether its temperature is 40 or 70 °F? Of course not. However, the selection and placement of these materials can have a profound impact on human thermal comfort, or lack thereof.

Providing comfort should be the primary objective of any heating system designer or installer. Unfortunately, this objective is too often compromised by other factors, the most common of which is cost. Even small residential heating systems affect the health, productivity, and general well-being of several people for many years. It makes sense to plan and install them accordingly.

The average North American building owner spends little time thinking about the consequences of the heating system they select. Many view such systems as a necessary but uninteresting part of a building. When construction budgets are tightened, it is often the heating system that is compromised to save money for other, more impressive amenities.

Heating professionals should take the time to discuss the full range of benefits of hydronic systems, including superior comfort, as well as price with their clients. Often people who have lived with uncomfortable heating systems do not realize what they have been missing. In retrospect, many would welcome the opportunity to live or work in truly comfortable buildings and would willingly spend more money, if necessary, to do so.

Maintaining comfort is not a matter of supplying heat to the body. Instead, it is a matter of controlling how the body loses heat. *When interior conditions allow heat to leave a person's body at the same rate at which it is generated, that person feels comfortable.* If heat is released faster or slower than the rate at which it is produced, some degree of discomfort is experienced.

A normal adult engaged in light activity generates heat at a rate of approximately 400 **British thermal units** per hour (Btu/h). Figure 1-2 shows the various processes by which the body of a person at rest releases heat to a typical indoor environment.

Note that a large percentage of the body's heat loss comes from **thermal radiation** to surrounding surfaces. Most people will not be comfortable in a room containing several cool surfaces such as large windows or cold floors, even if the room's air temperature is 70 °F. Remaining heat loss occurs through a combination of **convection**, **evaporation**, respiration, and a small amount of **conduction**. The latter occurs through surfaces in direct contact with the body.

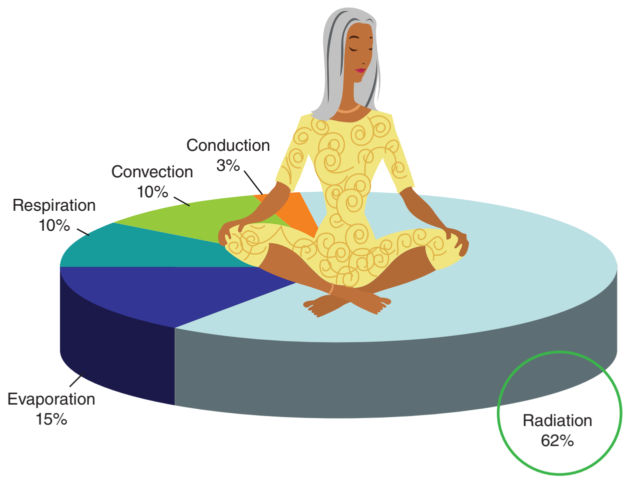


Figure 1-2 Processes the body uses to release heat to a typical indoor environment. *Courtesy of Robert Bean.*

The body can adjust these heat loss processes, within certain ranges, to adapt to different interior environments. For example, if air temperature around the body increases, convection heat loss will be suppressed. The body responds by increasing evaporation heat loss (perspiration) and increasing skin temperature to increase radiative heat loss.

Properly designed hydronic systems influence both the air temperature and surface temperatures of rooms to maintain optimal comfort. Modern controls can maintain room air temperature to within ±1 °F of the desired **setpoint temperature**. Heat emitters such as radiant floors or radiant ceilings raise the average surface temperature of rooms. Since the human body is especially responsive to radiant heat loss, these heat emitters significantly enhance comfort. Comfortable humidity levels are also easier to maintain in hydronically heated buildings.

Several factors such as activity level, age, and general health determine the comfortable environment for a given individual. When several people are living or working in a common environment, any one of them might feel too hot, too cold, or just right. Heating systems that allow various "zones" of a building to be maintained at different temperatures can better adapt to the comfort needs of several individuals. This is called **zoning**. Although both forced-air and hydronic heating systems can be zoned, the latter is usually much simpler and easier to control.

Energy Savings

Ideally, a building's rate of heat loss would not be affected by how that heat is replaced. However, experience has shown that otherwise identical buildings can have significantly different rates of heat loss based on the types of heating systems installed. *Buildings with hydronic heating systems have consistently shown lower heating energy use than equivalent structures with forced-air heating systems.*

A number of factors contribute to this finding. One is that hydronic systems do not adversely affect room air pressure while operating. Small changes in room air pressure occur when the blower of a ducted forced-air heating system is operating. Increased room air pressure is often created by a lack of adequate return air flow from the rooms back to the furnace or air handler. This condition drives heated air out through every small crack, hole, or other opening in the exterior surfaces of the room.

One study that compared several hundred homes, some with ducted forced-air systems and others with hydronic baseboard convectors, found that air leakage rates averaged 26% higher and energy usage averaged 40% greater in homes with forced-air heating.

Another factor affecting building energy use is air temperature **stratification** (e.g., the tendency of warm air to rise toward the ceiling while cool air settles to the floor). In extreme situations, the difference in air temperature from floor to ceiling can exceed 20 °F. Stratification tends to be worsened by high ceilings, poor air distribution, and heating systems that supply air into rooms at high temperatures. Maintaining comfortable air temperatures in the occupied areas of rooms plagued with a high degree of temperature stratification leads to significantly higher air temperatures near the ceiling as shown in Figure 1-3. This, in turn, increases heat loss through the ceiling.

Hydronic systems that transfer the majority of their heat by thermal radiation reduce air temperature stratification and thus reduce heat loss through ceilings. Comfort can often be maintained at lower air temperatures when a space is radiantly heated. This leads to further energy savings. Zoned hydronic systems provide the potential for unoccupied rooms to be kept at lower

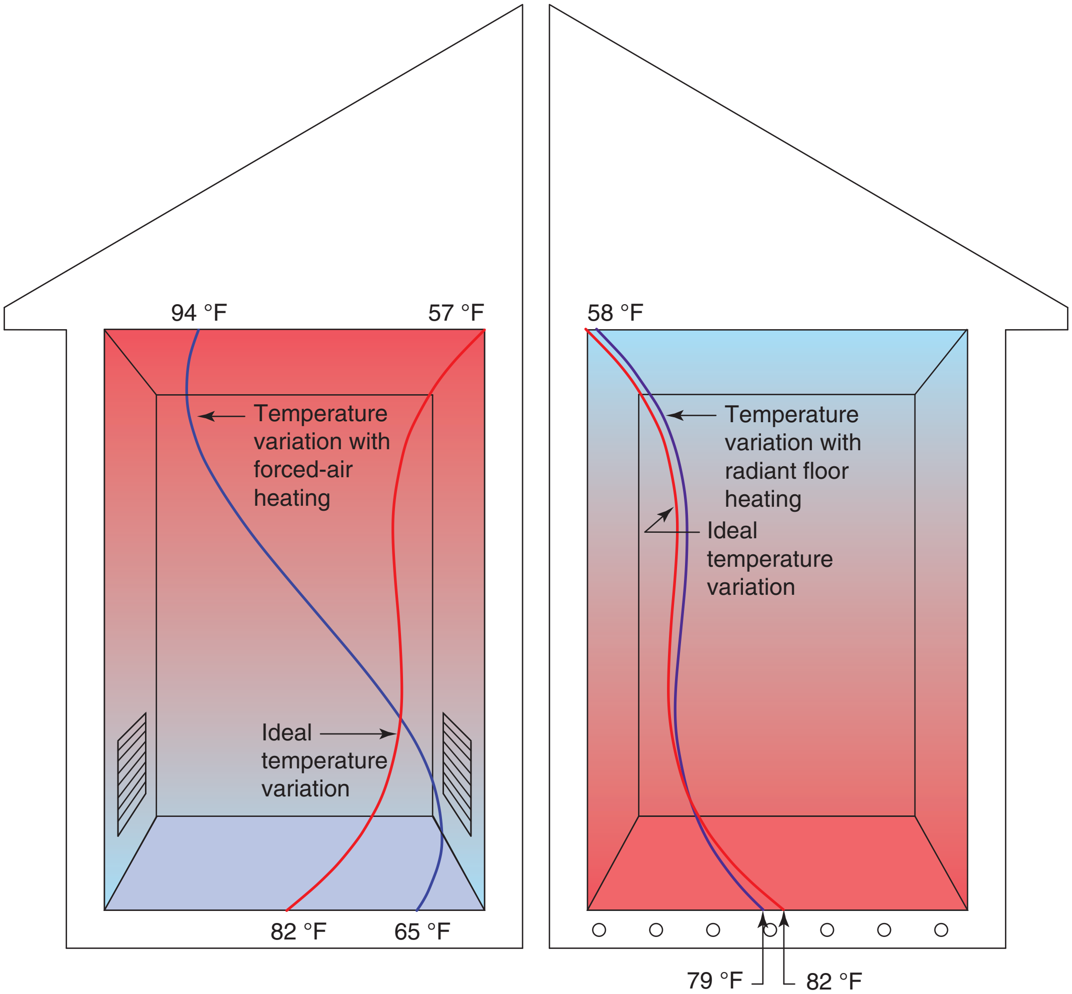


Figure 1-3 | Comparison of air temperatures from floor to ceiling for forced-air heating (left) and radiant floor heating (right).

temperatures, which lowers heat loss and reduces fuel consumption. Some hydronic systems also automatically reduce the water temperature in their distribution piping as the outdoor temperature increases. This reduces heat loss from piping and increases heat source efficiency.

The electrical energy consumption of the circulator(s) used in a well-planned modern hydronic system can be a small fraction (often less than 10%) of the electrical energy required by a blower in a forced-air heating system of equal capacity. This saving is often overlooked by those who only consider the energy use associated with *producing* heat or chilled water for cooling. This difference in energy use by various heating or cooling distribution systems can be quantified using the concept of **distribution efficiency**, which is discussed later in this chapter.

Design Flexibility

Modern hydronics technology offers virtually unlimited potential to accommodate the comfort needs, usage, aesthetic tastes, and budget constraints of almost any building. A single system can be designed to supply **space heating**, domestic hot water, and specialty **loads** such as pool heating or **snowmelting** (see Figure 1-4). Such **"multiload" systems** reduce installation costs because redundant components such as multiple heat sources, exhaust systems, electrical hookups, safety devices, and fuel supply components are eliminated. They also tend to improve the heat source efficiency and thus reduce fuel usage relative to systems where each load is served by its own dedicated heat source.

The space heating needs of some buildings are best served through the use of different heat emitters. For example, it is possible for hydronic radiant floor heating to be used in the basement and first floor of a house while the second floor rooms are heated using panel radiators or fin-tube baseboard. Modern hydronics technology makes it easy to combine different heat emitters into the same system.

The wide variety of hydronic heat sources now available also allows systems to easily adapt to special circumstances or opportunities such as "time-of-use" electrical rates, on-site renewable energy availability, or waste heat recovery.

Clean Operation

A common complaint about forced-air heating is its propensity to move dust and other airborne particles such as pollen and smoke throughout a building. In buildings where air-filtering equipment is either of low quality or is poorly maintained, dust streaks around ceiling diffusers, as seen in Figure 1-4a, are often evident. Eventually duct systems, such as that shown in Figure 1-5, require internal cleaning to remove dust and dirt that have accumulated over several years of operation.

In contrast, few hydronic systems involve forced-air circulation. Those that do create room air circulation rather than whole building air circulation. This reduces the dispersal of airborne particles and microorganisms, which is a major benefit in situations where air cleanliness is imperative, such as for people with allergies or respiratory illness, or in health care facilities and laboratories. When floor heating is used in entry areas,

Figure 1-4 The hydronic system that heats this house also melts snow on the walkway. *Courtesy of G. Todd.*

Figure 1-4a Dust streaks surrounding a ceiling diffuser indicate poor air filtering in a ducted forced-air distribution system. *Courtesy of John Siegenthaler*

Figure 1-5 Dirt and dust accumulation in ducting. *Courtesy of L. Drake.*

Figure 1-5a Ducting enclosed in a framed soffit within living space. *Courtesy of Angela McCormick.*

the floor can dry rapidly to reduce tracking water and dirt further into the building.

Quiet Operation

Most people want their home to be a quiet refuge from the pace and *noise* of modern life. They don't want to hear sounds emanating from their heating and cooling systems. A properly designed and installed hydronic system can operate with virtually no detectable sound levels in the occupied areas of a home. Modern systems that use constant circulation and variable water temperature control minimize expansion noises that can occur when high-temperature water is injected directly into a room temperature heat emitter. These characteristics make hydronic heating ideal in sound-sensitive areas such as home theaters, reading rooms, or recording studios.

Noninvasive Installation

Consider the difficulty encountered when ducts have to be concealed from sight within a typical house. The best that can be done in many situations is to encase the ducting in exposed soffits, as shown in Figure 1-5a. Such situations often lead to compromises in duct sizing and/or placement.

By comparison, hydronic heating systems are easily integrated into the structure of most small buildings without compromising their structure or the aesthetic character of the space. The underlying reason for this is the high **heat capacity of water**. *A given volume of water can absorb over 3,400 times more heat as the same volume of air for the same temperature change.* The volume of water that must be moved through a building to deliver a certain amount of heat is only about 0.03% that of air, assuming that the air and water undergo the same temperature change. This greatly reduces the size of the distribution "conduit."

For example, a 3/4-inch-diameter tube carrying water at 6.0 gpm around a hydronic system operating with a 20 °F temperature drop transports as much heat as a 14-inch by 8-inch duct carrying 130 °F air at 1,000 feet/min. Figure 1-7 depicts these two options side by side.

Notching into floor joists to accommodate the 14-inch by 8-inch duct would destroy their structural integrity. By contrast, smaller tubing is easily routed through the framing, especially if it happens to be one of several flexible tube products now available.

If the distribution system is insulated, which is now a code requirement in many areas, considerably less material is required to insulate the tubing compared to the ducting. When insulated with the same material, the heat loss of the 14-inch by 8-inch duct is almost 10 times greater than that of the 3/4-inch tube.

Hydronic systems using small flexible tubing are much easier to retrofit into existing buildings in comparison to ducting. The tubing can be routed through open or closed framing cavities much like electrical cable, as seen in Figure 1-8.

Figure 1-6 Hydronic systems operate with virtually no noise in occupied space. *Courtesy of John Siegenthaler*

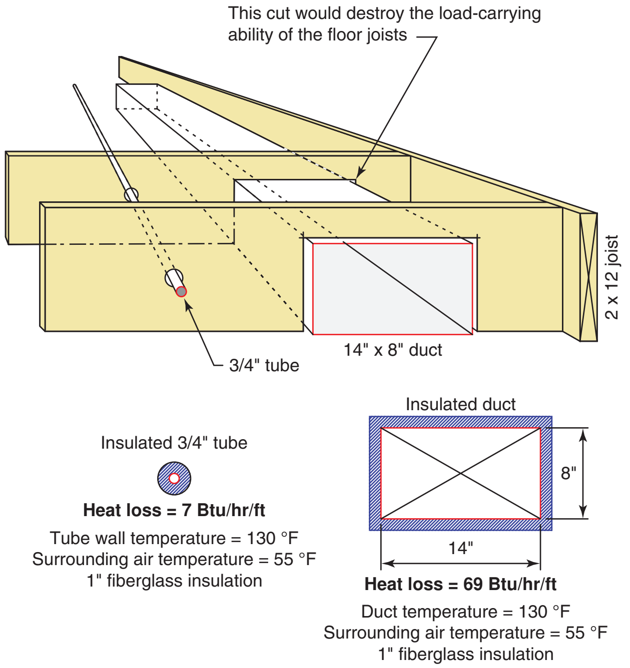


Figure 1-7 A 3/4-inch-diameter tube carrying water can transport heat at the same rate as a 14-inch by 8-inch duct in a forced-air system. If the water in the tube and air in the duct are of the same temperature, there is almost 10 times more heat loss from the duct due to its larger surface area.

One modern strategy is to route 3/8-inch or 1/2-inch flexible tubing from a central **manifold station** to a heat emitter in each room. This "homerun" approach allows the option of maintaining different temperatures in each room. The concept is shown in Figure 1-9.

For buildings where utility space is minimal, small wall hung boilers using sealed combustion systems can often be mounted in a closet. In many cases, these compact boilers supply the building's domestic hot water as well as its heat. The entire system might occupy less than 10 square feet of floor area.

Figure 1-8 Small-diameter flexible hydronic tubing routed through framing cavities. *Courtesy of John Siegenthaler*

Distribution Efficiency

The electrical power required to move heat or cooling effect through a building is an important consideration when trying to reduce a building's overall energy use. It's possible to compare the relative electrical energy use of various heating and cooling distribution systems using the concept of distribution efficiency, which is defined by Equation 1.1.

Equation 1.1:

$$n_d = \frac{q}{p_e},$$

where,

n_d = distribution efficiency (Btu/h/watt)
q = rate of thermal energy delivery by distribution system (Btu/h)
p_e = electrical power required to operate distribution system (watt)

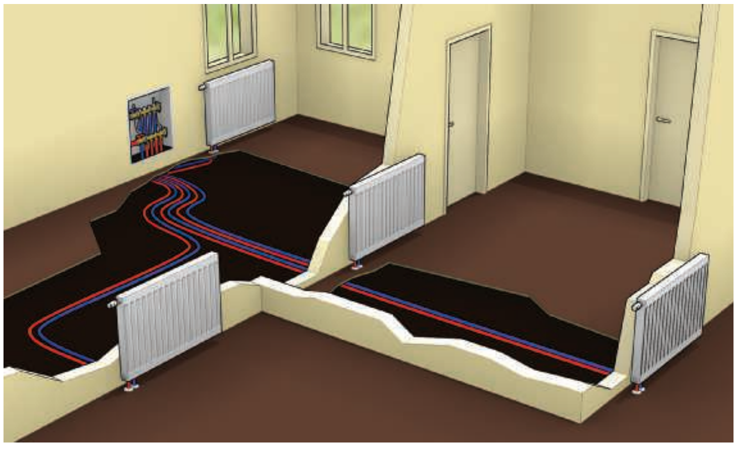


Figure 1-9 A homerun distribution system uses supply and return tubing from each heat emitter to a central manifold station. *Courtesy of Caleffi North America.*

Example 1.1

A hydronic heating distribution system has three circulators, each requiring 45 watts of electrical power input when operating at peak capacity. Under this condition the distribution system delivers 160,000 Btu/h of thermal energy to the building. Determine the distribution efficiency of this system.

Solution:

Since both the rate of thermal energy delivery and the associated electrical power input to the distribution system are known, the distribution efficiency is easily calculated:

$$n_{\mathrm{d}} = \frac{q}{p_{\mathrm{e}}} = \frac{160{,}000\ \mathrm{Btu/h}}{3(45\ \mathrm{watt})} = 1{,}185\ \frac{\mathrm{Btu/h}}{\mathrm{watt}}.$$

Discussion:

Neither the type of heat source that produced the 160,000 Btu/h of heat output, nor the efficiency of that heat source, is relevant when determining distribution efficiency.

The calculated value of 1,185 Btu/h/watt can be interpreted as follows: This hydronic distribution system can transfer 1,185 Btu/h of heat from where it is produced to where it's needed in the building for each watt of electrical power required to operate the distribution system. Still, the value of 1,185 Btu/h/watt is somewhat meaningless without something to compare it to. This will be done in Example 1.2.

Example 1.2

A ducted forced-air heating system uses a blower to deliver 84,000 Btu/h to a building. The blower requires 740 watts of electrical power input when operating. What is the distribution efficiency of this system?

Solution:

Again the calculation is straightforward:

$$n_{\mathrm{d}} = \frac{q}{p_{\mathrm{e}}} = \frac{84{,}000\ \mathrm{Btu/h}}{(740\ \mathrm{watt})} = 113.5\ \frac{\mathrm{Btu/h}}{\mathrm{watt}}.$$

Discussion:

In this case the distribution system only delivers 113.5 Btu/h to where it's needed in the building for each watt of electrical power required to operate the distribution system. A ratio of the distribution efficiencies calculated in these two examples is 1,185/113.5 = 10.4. This implies that the hydronic system delivered about 10.4 times as much heat per watt of electrical power input compared to the forced-air system. It could also be interpreted as the hydronic system delivering a given amount of heat using about 9.6%, of the electrical power required by the forced-air system. In either case the hydronic distribution system described in Example 1.1 holds a clear advantage over the ducted forced-air distribution system described in Example 1.2. That advantage could result in thousands of dollars of electrical energy savings over the life of these heating systems. Bear in mind that this comparison was for two specific systems, and that the results should not be generalized to all hydronic versus ducted forced-air system comparisons.

Using state-of-the-art design techniques and modern hardware, it is possible to construct hydronic systems having distribution efficiencies over 3,000 Btu/h/watt. This distinct benefit has the potential to significantly reduce electrical energy use in a wide range of buildings using all types of hydronic heating and cooling sources.

1.3 Heat and Heat Transfer

Before attempting to design any type of heating system, it is crucial to understand the entity being manipulated: heat.

What we commonly call heat can also be described as energy in thermal form. Other forms of energy such as electrical, chemical, mechanical, and nuclear energy can be converted into heat through various processes and devices. All heating systems consist of devices that convert one form of energy into another.

Heat is our perception of atomic vibrations within a material. Our means of expressing the intensity of these vibrations is called temperature. The more intense the vibrations, the greater the temperature of the material and the greater its heat content. Any material above absolute zero temperature (–460 °F) contains some amount of heat.

There are several units for expressing a quantity of heat. In North America, the most commonly used unit of heat is the British thermal unit (Btu). A Btu is defined as the amount of heat required to raise 1 lb. of water by 1 °F.

Heat always moves from an area of higher temperature to an area of lower temperature. In hydronic systems, this occurs at several locations. For systems using a boiler as their heat source, the heat moves from the hot gases in the combustion chamber to the cooler metal walls of the boiler's heat exchanger. The heat continues to move through the metal walls of the heat exchanger into the cooler water within. After being transported to a heat emitter on a flowing "conveyor belt" of water, the heat passes through the metal walls of the heat emitter into the still cooler air and objects of a room. Finally, heat moves through the exposed surfaces of a room into the outside air. Figure 1-10 depicts this heat transfer.

In every instance, heat moved from an area of higher temperature to an area of lower temperature. *Without this temperature difference, there would be no heat transfer.*

In this book, the *rate* of heat transfer is expressed in British thermal units per hour, abbreviated as Btu/h, or Btuh. It is very important to distinguish between the *quantity* of heat present in an object (measured in Btu) and the *rate* at which heat moves in or out of the object (measured in Btu/h). These terms are often misquoted by people, including those in the heating and cooling profession.

The *rate* of heat transfer from one location to another is governed by several factors. One is the temperature difference between where the heat is and where it is going. *Temperature difference is the driving force that causes heat to move.* Without a temperature difference between two locations, there can be no heat transfer. The greater the temperature difference, the faster heat flows. In most instances, the rate of heat transfer through a material is directly proportional to the temperature difference across the material. Thus, if the temperature difference across a material were doubled, the rate of heat transfer through the material would also double.

Another factor affecting the rate of heat transfer is the type of material through which the heat moves. Some materials, such as copper and aluminum, allow

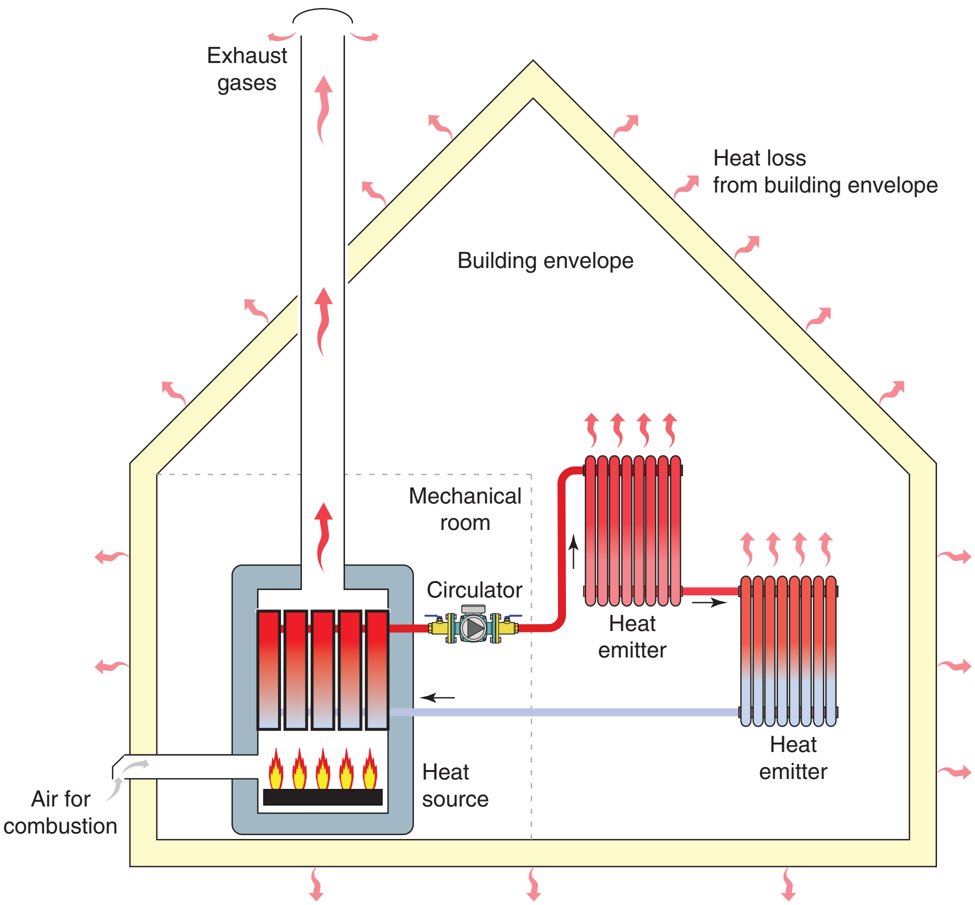


Figure 1-10 | Heat movement within a hydronic heating system and the building it serves.

heat to move through very quickly. Other materials, such as polyurethane foam, greatly inhibit the rate of heat transfer.

Three Modes of Heat Transfer

Thus far, two important principles of heat transfer have been discussed. First, heat moves from an area of higher temperature to an area of lower temperature. Second, the rate of heat transfer depends on temperature difference and the type of material. To gain a more detailed understanding of how the thermal components of a hydronic system work, we need to classify heat transfer into three modes: conduction, convection, and thermal radiation.

Conduction is the type of heat transfer that occurs through solid materials. Recall that heat has already been described in terms of atomic vibrations. Heat transfer by conduction is a dispersal of these vibrations from a source of heat, out across trillions of atoms that are bonded together to form a solid material. The index that denotes how well a material transfers heat is called its **thermal conductivity**. The higher a material's thermal conductivity, the faster heat can pass through it, all other conditions being equal. Heat passing though the walls of a coffee cup, and then to hands wrapped around that cup, as shown in Figure 1-10a is an example of conduction heat transfer. Heat moving from the hot inner surface of a pipe to its cooler outer surface is another example.

Figure 1-10a Heat passes through the wall of the coffee cup by conduction.
Africa Studio/Shutterstock.com

The rate of heat transfer by conduction is directly proportional to both the temperature difference across the material and its thermal conductivity. It is inversely proportional to the thickness of a material. Thus, if one were to double the thickness of a material while maintaining the same temperature difference between its sides, the rate of heat transfer through the material would be cut in half.

In some locations within hydronic systems, designers try to enhance conduction. For example, the higher the thermal conductivity of a flooring material installed over a heated floor slab, the faster heat can pass upward through it and into the room. In contrast, the slower the rate of conduction through the insulation of a hot water storage tank, the better it retains heat. Equations for calculating heat flow by conduction are presented in Chapter 2.

Convection heat transfer occurs when a fluid at some temperature moves along a surface at a different temperature. The term "fluid" can refer to a gas (such as air), or a liquid (such as water).

Consider the example of water at 100 °F flowing along a surface maintained at 150 °F. The cooler water molecules contacting the warmer surface absorb heat from that surface. These molecules are churned about as the water moves along. The heated molecules are constantly swept away from the surface into the bulk of the water stream and replaced by cooler molecules. One can think of the heat as being "scrubbed" off the surface by the flowing water.

The speed of the fluid moving over the surface greatly affects the rate of convective heat transfer. We've all experienced the increased "wind chill" effect as cool air is blown across our skin rather than lying relatively stagnant against it. Although the air temperature may not be extremely cold, the speed at which it moves over our skin greatly increases the rate of convective heat loss. Although it may feel like the air is very cold, we are actually sensing the *rate* of heat loss from our skin rather than the air temperature. To achieve the same cooling sensation in calm air requires a much lower air temperature.

When fluid motion is caused by either a circulator (for water) or a blower or fan (for air), the resulting heat transfer is more specifically called **forced convection**. When buoyancy differences within the fluid cause it to move without any influence from a circulator or blower, the heat transfer is more specifically called **natural convection**. Generally, heat moves much slower by natural convection than by forced convection. The warm air currents rising from the fin-tube element in Figure 1-11 are an example of natural convection. The heat transferred to the air pushed along by the blower is an example of forced convection.

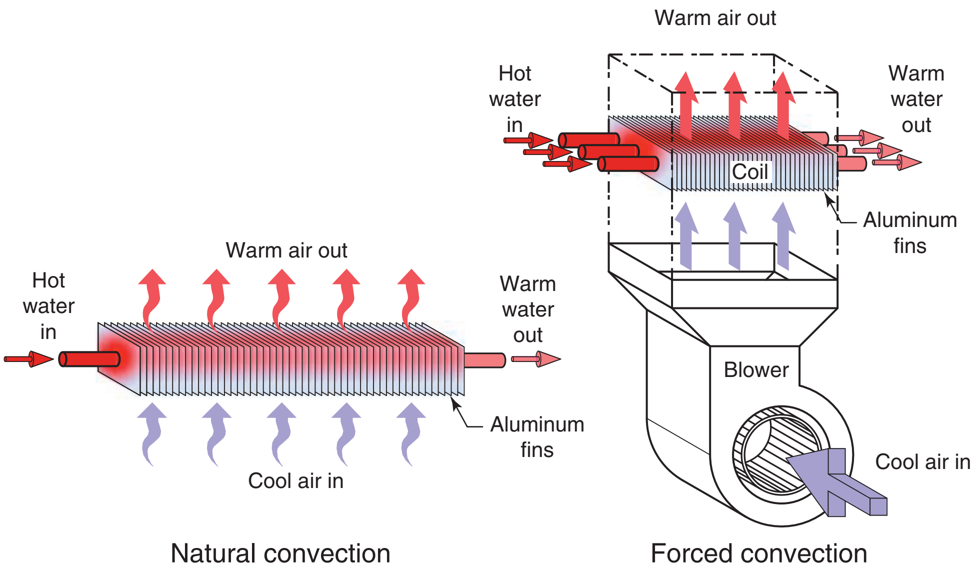


Figure 1-11 | Two types of convective heat transfer.

Some hydronic heat emitters, such as a finned-tube baseboard, are designed to release the majority of their heat output to the surrounding air by natural convection. Such devices are appropriately called **convectors**. Heat emitters that use fans or blowers to force air through a heat exchanger are usually called **fan coils** or **air handlers**.

Thermal radiation is probably the least understood mode of heat transfer. Just like visible light, thermal radiation is **electromagnetic energy**. It travels outward from its source in straight lines, at the speed of light (186,000 miles/s), and cannot bend around corners, although it can be reflected by some surfaces. Unlike conduction or convection, thermal radiation needs no material to transfer heat from one location to another.

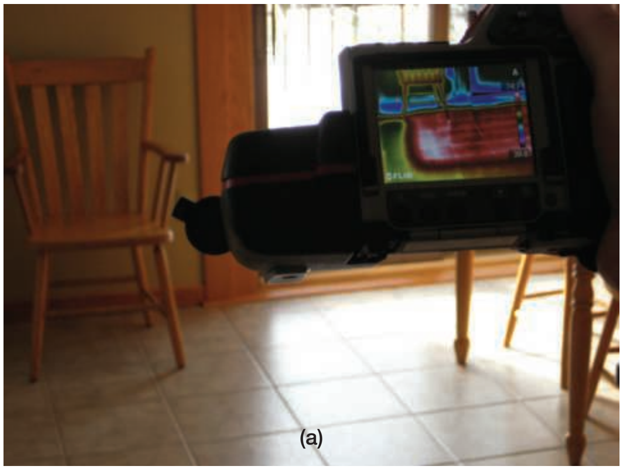
Consider a person sitting a few feet away from a campfire on a cold winter day, as shown in Figure 1-11a. If pointed toward the fire, the person's face probably feels warm, even though the surrounding air is cold. This sensation is the result of thermal radiation emitted by the fire traveling through the air and being absorbed by their exposed skin. The air between the fire and the person's face is not heated as the thermal radiation passes through it. Likewise, thermal radiation emitted from the warm surface of a heat emitter passes through the air in a room without directly heating that air. When the thermal radiation strikes another surface in the room, most of it is absorbed. At *that instant*, the energy carried by the thermal radiation becomes heat.

The main difference between thermal radiation and visible light is the wavelength of the radiation. Anyone who has watched molten metal cool has noticed how the bright orange color eventually fades to duller shades of red, until finally the metal's surface no longer glows. As the surface of the metal cools below about 970 °F, our eyes can no longer detect *visible* light from the surface. Our skin, however, still senses that the surface is very hot. Though unseen, thermal radiation in the nonvisible infrared portion of the electromagnetic spectrum is still being strongly emitted by the metal's surface.

Any surface continually emits thermal radiation to any cooler surface within sight of it. The surface of a heat emitter that is warmer than our skin or clothing surfaces

Figure 1-11a | Thermal radiation passes from the fire to skin surfaces without warming the air between. *Pavel Vaschenkov/ Shutterstock.com*

Figure 1-11b Comfort is reduced due to radiant heat loss to cold glass surfaces even though interior air temperature is acceptable. *Courtesy of Caleffi North America.*



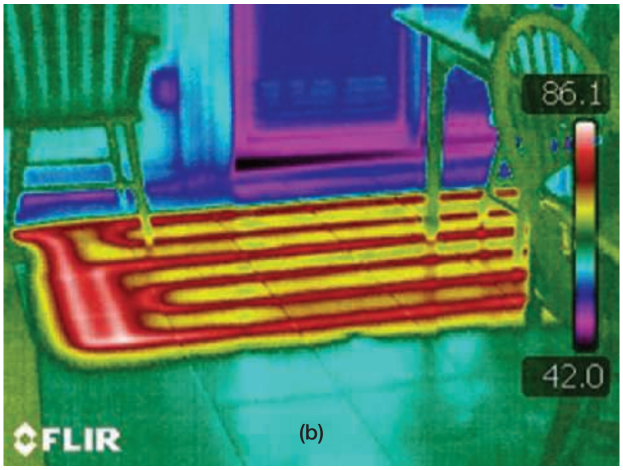


Figure 1-12 (a) An infrared camera aimed at a floor. *Courtesy of John Siegenthaler* (b) Infrared thermograph of same floor reveals heat from tubing installed below floor. *Courtesy of John Siegenthaler*

transfers heat to us by thermal radiation. Likewise, our skin and clothing give off thermal radiation to any surrounding surfaces at lower temperatures.

The term **mean radiant temperature** describes the area-weighted average temperature of all surfaces within a room. As the mean radiant temperature of a room increases, the air temperature required to maintain comfort decreases. As the temperature of the room's surfaces increases, the heat released from the body by thermal radiation decreases, so the amount released by convection must increase to keep the total rate of heat release constant. The converse is also true. This is why a person in a room with an air temperature of 70°F may still feel cool if surrounded by cold surfaces, such as large, undraped windows, on a cold winter day, as shown in Figure 1-11b.

As thermal radiation strikes an opaque surface, part of it is absorbed as heat and part is reflected away from the surface. The percentage of incoming radiation that is absorbed or reflected is determined by the optical characteristics of the surface and the wavelength of the radiation. Most interior building surfaces absorb the majority of thermal radiation that strikes them. The small percentage that is reflected typically strikes another surface where most of it will be absorbed, and so on. Very little, if any, thermal radiation emitted by warm surfaces in a room escapes from the room.

Although the human eye cannot see thermal radiation, there are devices that can detect it and display an image that uses colors to represent different surface temperatures. Such an image is called an **infrared thermograph**. Figure 1-12a shows an infrared-detecting camera pointed at a tiled floor. The visible floor surface gives no evidence that it is being heated. Figure 1-12b is an infrared thermograph of the same floor area produced by the infrared camera. The bright colors show areas of different surface temperatures. From the shape of these color patterns, it is evident that a source of heat is installed under the floor; in this case, it is tubing carrying heated water. The color spectrum on the far right of the image gives the range of surface temperatures in the image. In this case they range from the relatively cool sill of the patio door (about 42.0°F) to the warmest floor areas directly above the embedded tubing (about 86°F). Infrared thermography is a powerful tool in diagnosing the thermal characteristics of buildings as well as mechanical systems. It can also be used to locate heated objects embedded behind surfaces as Figure 1-12b demonstrates.

The rate at which thermal radiation transfers heat between two surfaces depends upon their temperatures, an optical property of each surface called emissivity, and the angle between the surfaces.

Hydronic heat emitters deliver a portion of their heat output to the room by convection and the rest by radiation. The percentages delivered by each mode depend on many factors such as surface orientation, shape, type of finish, air flow rate past the surface, and surface temperature.

When a heat emitter transfers over 50% of its heat output by radiation, it is called a **radiant panel**. Heated floors, walls, and ceilings are all examples of radiant panels. They are discussed in detail in Chapter 10.

Thermal Equilibrium

If a material is not gaining or losing heat and remains in a single physical state (solid, liquid, or gas), its temperature does not change. This is also true if the material happens to be gaining heat from one object while simultaneously releasing heat to another object at the same rate. These principles have many practical applications in hydronic heating.

For example, if you observed the operation of a hydronic heating system for an hour and found no change in the temperature of the water leaving the boiler, although it was firing continuously, what could you conclude? Answer: Since there is no change in the water's temperature, it did not undergo any net gain or loss of heat. The rate that the boiler injects heat into the water stream passing through it is the same as the rate the heat emitters remove heat from the water stream (see Figure 1-13).

Under these conditions, the system is in thermal equilibrium. It is important to understand that *all hydronic heating systems inherently attempt to find a condition of* ***thermal equilibrium*** *and remain in operation at that condition*. The goal of the system designer is to ensure that thermal equilibrium is established at a condition that maintains comfort in the building and does not adversely affect the operation, safety, or longevity of the system's components.

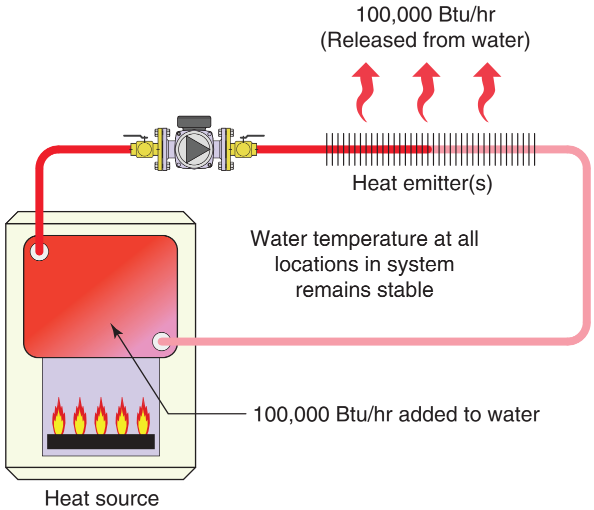


Figure 1-13 | When the rates of heat input and heat release are equal, the system is in thermal equilibrium and fluid temperatures at all locations in the system are stable.

1.4 Four Basic Hydronic Subsystems

In a hydronic heating system, water is heated by a heat source and conveyed by means of a distribution system to heat emitters where it is released to the building. A control system regulates these elements in an attempt to keep the rate of heat delivery very close to the rate of building heat loss. The overall hydronic system thus consists of four interrelated subsystems:

1. Heat source
2. Distribution system
3. Heat emitters
4. Control system

This section shows how these subsystems are connected to form a simple hydronic heating system. The discussion begins with the concept of a simple piping loop. Several essential components are described and then situated in the loop. Finally, a composite schematic shows all components in their preferred locations relative to each other. After reading this section, you will have a basic understanding of what the essential components of a hydronic system are and what they do. The details of proper component selection and sizing are covered in later chapters.

Basic Hydronic Circuit

The simplest hydronic system can be described as a loop or piping circuit. If the circuit is sealed off from the atmosphere at all locations (as is true for most modern hydronic systems), it is called a **closed-loop system**. If the circuit is open to the atmosphere at any point, it is called an **open-loop system**. Figure 1-14 shows the

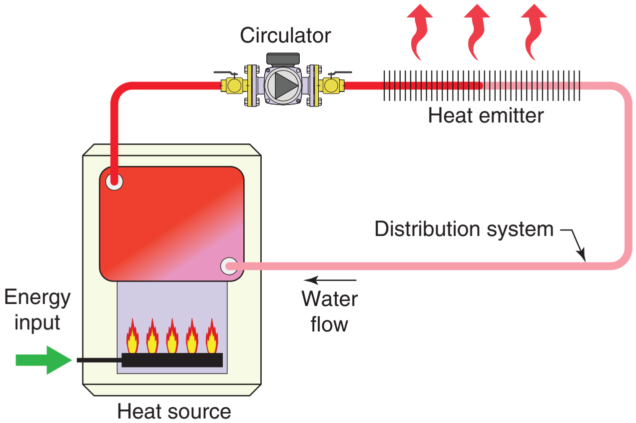


Figure 1-14 | The basic hydronic heating circuit.

simplest form of a piping circuit. The basic components are represented by schematic symbols.

When there is a demand for heat, water flow in the circuit is established by the circulator. The water carries heat from the heat source to the heat emitter where the heat is released into the space. Think of the flowing water as a "conveyor belt" for the heat.

In an ideal application, the rate of heat production by the heat source would exactly match the rate of heat dissipation by the heat emitters. This heat flow would also match the rate of heat loss from the building. Unfortunately, in real systems such ideal conditions seldom exist.

For example, it is fairly common to select a boiler with a heating capacity slightly greater than the building's rate of heat loss on the coldest day of the year. During a milder day, a boiler selected by this method could deliver heat to the building much faster than the building loses heat. Continuous operation of the boiler under such circumstances would quickly overheat the building. Obviously, some method of controlling the system's heat delivery is needed.

Temperature Controls

Figure 1-15 adds two simple control devices, the **room thermostat** and the **temperature-limiting controller**, to the basic system. The room thermostat determines when the building requires heat based on its **setpoint temperature** and the current indoor air temperature.

When the indoor air temperature drops slightly below the thermostat's setpoint, the switch contacts inside the thermostat close. This in turn signals other electrical circuits in the system to turn on the circulator and "enable" the heat source to produce heat.

The temperature-limiting controller ensures that the water temperature within the heat source remains within a predetermined range while the demand for heat is present. It does so by turning the heat-producing components of the heat source on and off as needed. Like the room thermostat, the temperature-limiting controller is a temperature-operated switch. For example, assume that the temperature-limiting controller on a typical gas- or oil-fired boiler is set for 160 °F with a 10 °F **differential**. If the water temperature inside the boiler drops to 150 °F (160 °F setpoint − 10 °F differential), a switch contact in the temperature-limiting controller closes to operate the burner. The burner remains on until the water in the boiler reaches its upper limit temperature of 160 °F or the heat demand is no longer present (e.g., the room thermostat is "satisfied").

Together, the room thermostat and temperature-limiting controller provide the necessary start/stop signals to the heat source and circulator to reasonably match the heat output to the heating requirement of the building. Such components have been used in millions of hydronic heating systems for several decades. These devices, as well as several more sophisticated controllers, are discussed in Chapter 9.

Expansion Tank

As water is heated, it expands. This increase in volume is an extremely powerful but predictable characteristic that must be accommodated in any type of closed-loop hydronic system. Figure 1-16 shows an **expansion tank**

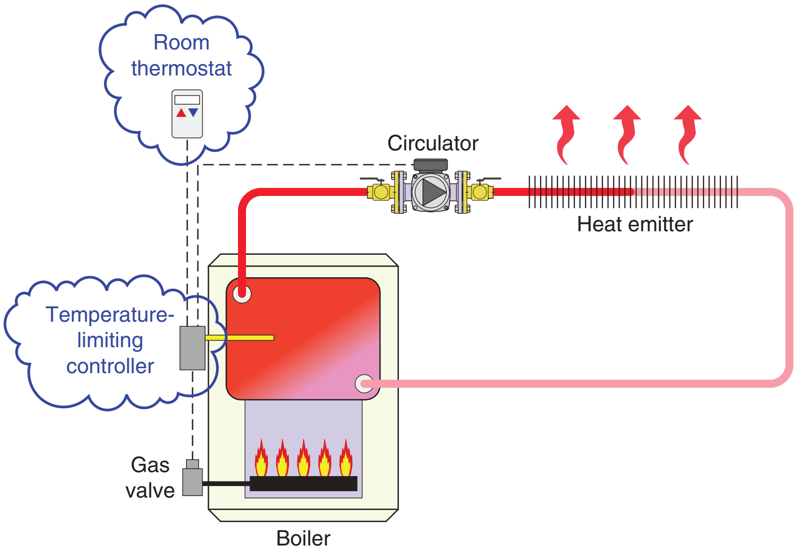


Figure 1-15 | Adding a room thermostat and heat source temperature-limiting controller to the system.

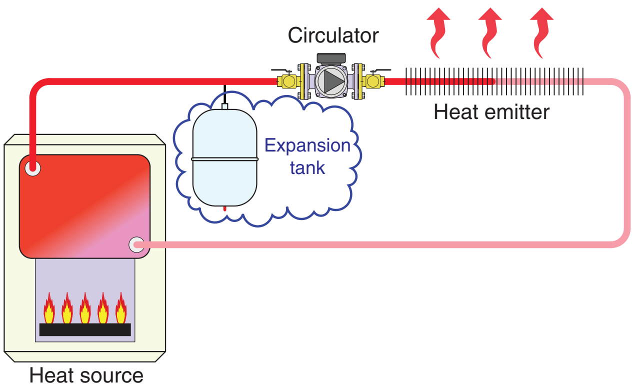


Figure 1-16 | Adding an expansion tank to the basic circuit.

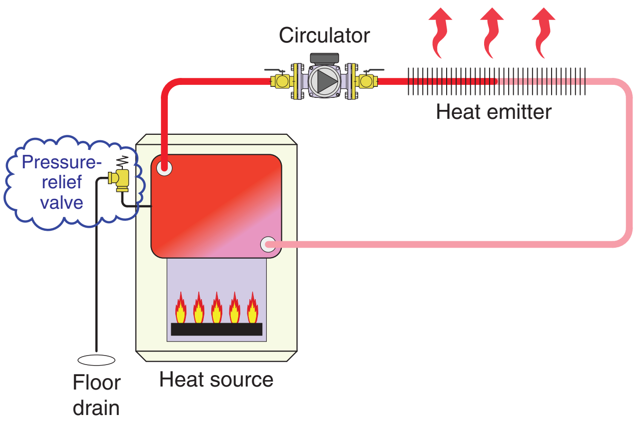


Figure 1-17 | Adding a pressure-relief valve to the heat source.

added to the basic system. The tank contains a captive volume of air. As the heated water expands, it pushes into the tank and slightly compresses the captive air volume. As a result, the system pressure rises slightly. As the water cools, its volume decreases, allowing the compressed air to expand, and the system pressure returns to its original value. This process repeats itself each time the system heats up and cools off.

Most modern hydronic systems use diaphragm-type expansion tanks. Such tanks contain their captive air in a sealed chamber. Older hydronic systems often used expansion tanks without diaphragms. Such tanks had to be considerably larger than modern diaphragm-type tanks. They also had to be mounted higher than the heat source.

The sizing and placement of the system's expansion tank are crucial to proper system operation. Both are discussed in Chapter 12.

Pressure-Relief Valve

Consider the fate of a closed-loop hydronic system in which a defective controller fails to turn off the heat source after its upper temperature limit has been reached. As the water gets hotter and hotter, system pressure steadily increases due to the water's expansion. This pressure could eventually exceed the pressure rating of the weakest component in the system. Most residential system components have pressure ratings of at least 60 pounds per square inch (psi) and may withstand two or more times that pressure before bursting. The consequences of a system component bursting at such high pressures and temperatures could be devastating. For this reason, all closed-loop hydronic systems must be protected by a **pressure-relief valve**. This is a universal requirement of all mechanical codes in North America.

Pressure-relief valves are designed and labeled to open at a specific pressure. Residential and light commercial systems typically have pressure-relief valves rated to open at 30 psi pressure. In systems with boilers, the pressure-relief valve is almost always attached directly to the boiler, or mounted very close to the boiler with nothing that could impede its operation between it and the boiler. A pressure-relief valve has been added to the heat source in Figure 1-17. Pressure-relief valves are discussed in more detail in Chapter 5.

Make-Up Water System

Most closed-loop hydronic systems experience minor water losses over time due to evaporation from valve packings, pump seals, air vents, and other components. These losses are normal and must be replaced to maintain adequate system pressure. The common method for replacing the water is through a **make-up water system** consisting of a **pressure-reducing valve**, **backflow preventer**, pressure gauge, and shutoff valves.

Because the pressure in a municipal water main or private water system is often higher than the pressure-relief valve setting in a hydronic system, such water sources cannot be directly connected to the loop. A pressure-reducing valve, also known as a **feed water valve**, is used to reduce and maintain a constant minimum pressure in the system. This valve allows water into the system whenever the pressure on the outlet side of the valve drops below the valve's pressure setting. This often occurs as air is vented from the system at start-up or after servicing. Most pressure-reducing valves have an adjustable pressure setting. Determining the proper setting is covered in Chapter 12.

The backflow preventer does just what its name implies. It stops any water that has entered the system from returning and possibly contaminating the potable

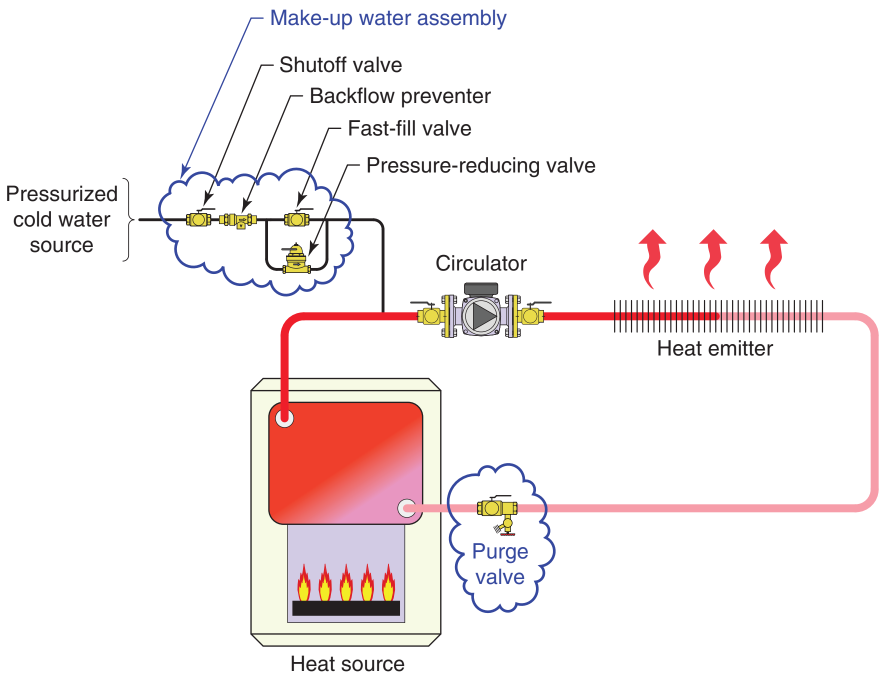


Figure 1-18 | Adding a make-up assembly.

water supply system. Most municipal codes require such a device on any heating system connected to a public water supply. Even in the absence of code requirements, installing a backflow preventer is a wise decision, especially if any antifreeze, corrosion inhibitors, or other chemicals are, or ever might be, used in the system.

The shutoff valves are installed to allow the system to be isolated from its water source and to allow components between the shutoff valves to be isolated if they need to be serviced. An optional "fast-fill" valve is sometimes installed in parallel with the pressure-reducing valve so water can be rapidly added to the system. This is particularly beneficial when filling larger systems. The components that constitute the make-up water system are shown in Figure 1-18.

Also shown in the lower portion of Figure 1-18 is a **purging valve** that allows most of the air in the system to be eliminated as it is filled with water. Purging valves consist of a ball valve, which is in line with the distribution piping, and a side-mounted drain port. To **purge** the system, the inline ball valve is closed, and the drain port is opened. As water first enters the system through the make-up water assembly, air is expelled through the drain port. Eventually, most of the air originally in the system will have been expelled through the drain port of the purging valve.

Flow-Check Valve

Another component commonly used in older hydronic systems is a **flow-check valve**. This valve contains a weighted plug that sits over the orifice within the valve. Some minimum differential pressure is required to lift this plug off the valve's orifice and thus enable flow through the valve. That minimum differential pressure will only be present when the circulator for that circuit is operating.

Another valve, known as a **spring-loaded check valve**, is sometimes used in place of a flow-check valve. This practice is common in modern systems. Both types of valves are discussed in more detail in Chapter 5. Either of these valves can serve one or two purposes depending upon the system it is installed in.

In single-loop systems, the flow-check valve prevents hot water in the boiler from slowly circulating through the distribution system when the circulator is off. Whenever a device containing heated water is part of an unblocked piping path having some vertical displacement, the potential for such flow exists. This flow is driven by the difference in density of hot water in the heat source relative to that of the cooler water in the distribution system. If not prevented, such **thermosiphoning** allows heat to be dissipated by slow but persistent flow through a piping circuit in an uncontrolled manner, often ending up where it is not desired. Figure 1-19 shows the

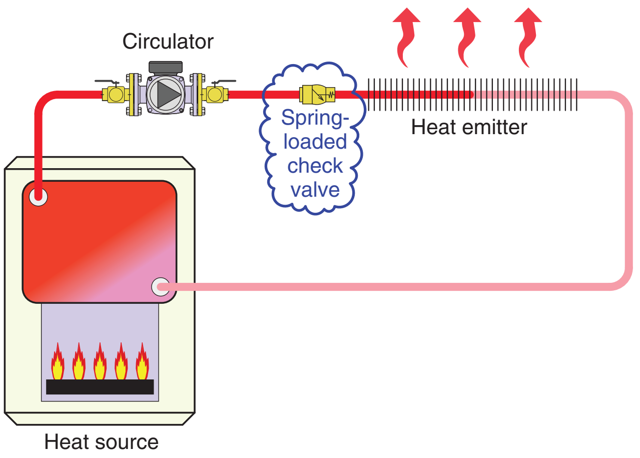


Figure 1-19 | Adding a flow-check valve to the basic circuit.

preferred placement of a flow-check valve (or spring-loaded check valve) in a single-loop system.

In multizone systems using circulators, a flow-check valve is installed in each zone circuit to eliminate thermosiphoning as well as flow reversal through inactive zones.

Air Separator

An **air separator** is designed to separate air from water and eject it from the system. Modern air separators create regions of reduced pressure as water passes through them. The reduced pressure causes molecules of oxygen, nitrogen, and other gases dissolved in the water to form bubbles. Once formed, these bubbles are guided upward into a collection chamber where an automatic air vent expels them from the system.

The process of separating air from water is enhanced when the water is heated. For best results, the air separator should be located where fluid temperatures are highest—in the supply pipe from the heat source. An air separator is shown in Figure 1-20. Air elimination is covered in detail in Chapter 13.

1.5 The Importance of System Design

Figure 1-21 is a composite drawing showing all the components previously discussed in their proper positions relative to each other. By assembling these components, we have built a simple hydronic heating system. It must be emphasized, however, that just because all the components are present does not guarantee that the system will function properly. Combining these components is not simply a matter of choosing a favorite product for each and connecting them as shown. Major subsystems such as the heat emitters and heat source have temperature and flow requirements that must be properly matched if they are to function together as a system. The objective is to achieve a stable, dependable, affordable, and efficient overall system. Failure to respect the operating characteristics of all components, and how they interact, will result in installations that underheat, overheat, waste energy, or otherwise disappoint their owners. The chapters that follow present detailed information on all the basic components just discussed. This information is essential in planning systems that deliver optimal performance.

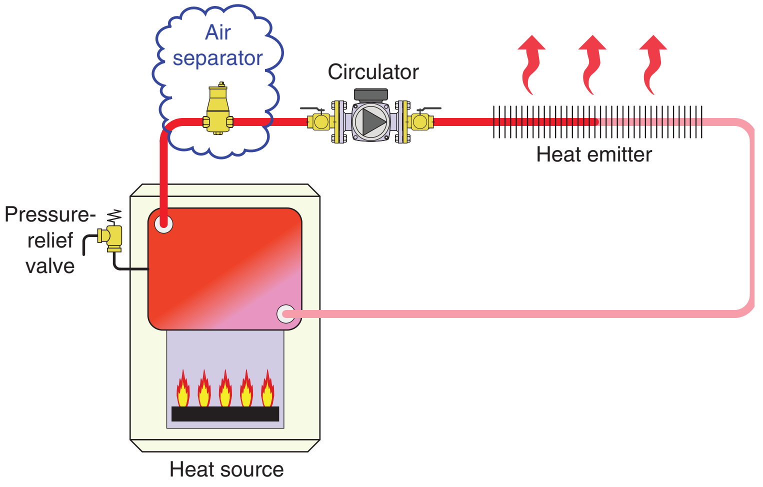


Figure 1-20 | Adding an air separator to the basic circuit.

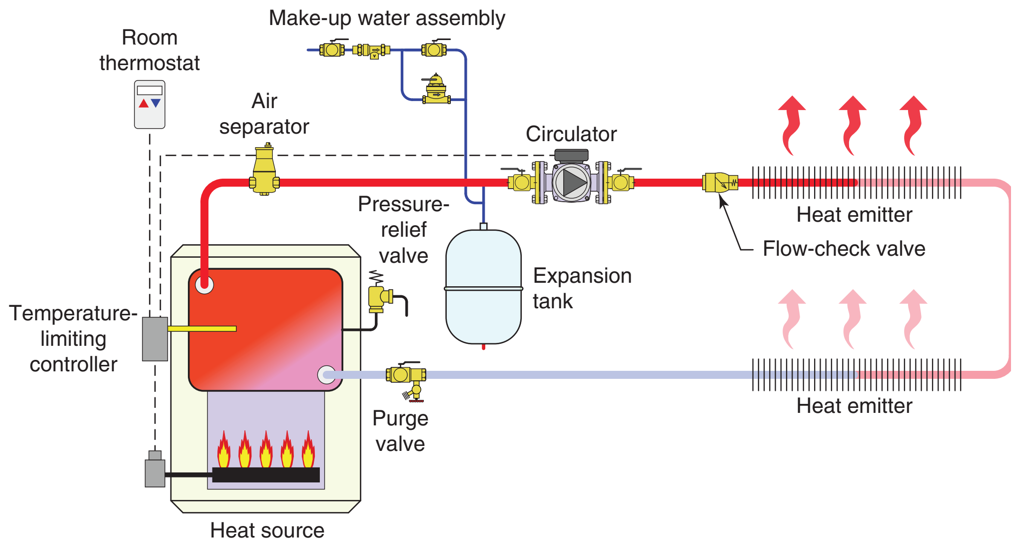


Figure 1-21 | Composite drawing showing all the basic hydronic circuit components.

Summary

The primary objective of any heating system is to provide the best possible comfort for the building occupants. Other important objectives include making efficient use of fuel and electricity, providing quiet and dependable operation, and maximizing the service life of all system components.

Hydronic heating systems offer tremendous potential for attaining all these objectives. The key to turning potential into reality is a solid understanding of the processes and components introduced in this chapter. A lack of understanding of these basics is often the cause of improper system design or difficulty in diagnosing the cause of poor system performance. Before continuing into more detailed aspects of design and installation, take time to test your understanding of these all-important fundamentals.

Key Terms

air handlers
air separator
air vent
backflow preventer
brine
British thermal units
buoyancy
circulators
closed-loop system
conduction
convection
convectors
differential
distribution efficiency
distribution system
domestic hot water
electromagnetic energy
evaporation
expansion tank
fan coils
feed water valve
flow-check valve
forced convection
heat capacity of water
heat emitter
heat source
infrared thermograph
loads
make-up water system
manifold station
mean radiant temperature
multiload systems
natural convection
open-loop system
pressure-reducing valve
pressure-relief valve
purge
purging valve
radiant panel
room thermostat
setpoint temperature
snowmelting
space heating
spring-loaded check valve
stratification
temperature-limiting controller
thermal conductivity
thermal energy
thermal equilibrium
thermal radiation
thermosiphoning
zoning

Questions and Exercises

1. Why does ducting in forced-air heating systems have to be so much larger than the tubing in a hydronic heating system of equal heating capacity?
2. Why is it better to surround a person with warm surfaces as opposed to just warm air?
3. What type of heat transfer creates the wind chill effect we experience during winter?
4. A certain block of material conducts heat at a rate of 100 Btu/h. One side is maintained at 80 °F and the other at 70 °F. Describe what happens to the rate of heat transfer when the:
   a. 80 °F side is raised to 130 °F
   b. thickness of the block is doubled
   c. 80 °F side is raised to 130 °F, and the thickness is cut in half
5. At approximately what temperature does a surface that is cooling from a high temperature stop emitting visible radiation?
6. How does thermal radiation differ from visible light? In what ways are they similar?
7. Describe thermal equilibrium within a hydronic system.
8. What is the function of a feed water valve in a hydronic system?
9. What is the difference between an open-loop system and a closed-loop system?
10. Are there any closed-loop hydronic heating systems on which a pressure-relief valve is not required? Why?
11. List two types of hydronic heat emitters other than finned-tube baseboard convectors.
12. What is the function of a flow-check valve in a single-loop system?
13. What is a "brine"? In what type of hydronic heating application would it be used?
14. Why is it necessary to have a backflow preventer in the make-up water line?
15. What are the customary units for heat and heat transfer rate in the North America?
16. What is a "zoned" hydronic system?
17. What made water flow through early hydronic systems before circulators were used?
18. What causes air bubbles to form within an air separator?
19. How are the expansion tanks used in modern hydronic systems different from those used in older systems?
20. What are some common ways small amounts of water leak out of a closed-loop hydronic system?

Chapter 2

Space Heating and Domestic Water-Heating Loads

Objectives

After studying this chapter, you should be able to:

- Describe what a design heating load is and why it is important to heating system design.
- Explain the difference between room heating loads and building heating loads.
- Determine the thermal envelope of a building.
- Calculate the effective total *R*-value of a building surface.
- Describe how the unit *U*-value for windows and doors is determined and how it is used.
- Estimate infiltration heat loss using the air change method.
- Determine the heat loss of foundations and slab floors.
- Explain what degree days are and how they are used.
- Estimate the annual space heating energy usage of a building.
- Estimate the daily energy used for domestic water heating.

2.1 Introduction

The design of any space heating or domestic water-heating system must start with an estimation of the **thermal load**. The care given to this step will directly affect system cost, efficiency, and most importantly, customer satisfaction.

It is natural for those learning to design hydronic heating systems to want to "dive into" a discussion of the hardware involved. However, this is like selecting the foundation for a building without knowing how much weight that foundation must support. It is pointless to start designing any heating system for a building without knowing the rate at which that building requires heat. To that end, this chapter lays out the basics for determining both space heating and domestic water-heating loads. Using the information in this chapter, designers can accurately estimate these loads and then optimize their system designs to meet them.

The majority of this chapter deals with estimating space heating loads. Simple mathematical methods for **conduction heat loss** and **air infiltration heat loss** are introduced, as are methods of estimating heat losses from slab on grade floors and basements. These methods are then applied to an example house on a room-by-room basis. Software-based approaches to heating load estimation are also discussed. The concept of **degree days** is introduced and used as a way to extrapolate the design heating load into an estimate of annual heating energy use for the building in a specific climate.

Information for estimating daily and peak hourly domestic water-heating loads is also presented. Domestic water-heating load information will be combined with space heating loads and used to design combisystems in later chapters.

2.2 Design Space Heating Load

It is important to understand what a design space heating load is before attempting to calculate it. *The **design heating load** of a building is an estimate* of the rate at *which a building loses heat during the near-minimum outdoor temperature.* This definition contains a number of key words that need further explanation.

First, it must be emphasized that a heating load is a calculated *estimate* of the rate of heat loss of a room or building. Because of the hundreds of construction details and thermal imperfections in an object as complex as a building, even a simple house, it is simply not possible to determine its design heating to the nearest Btu/h. Some factors that add uncertainty to the calculations are:

- Imperfect installation of insulation materials
- Variability in *R*-value of insulation materials
- Shrinkage of building materials, leading to greater air leakage
- Complex heat transfer paths at wall corners and other intersecting surfaces
- Effect of wind direction on building air leakage
- Overall construction quality
- Traffic into and out of the building

Second, the heating load is a *rate* of heat flow from the building to the outside air. It is often misstated, even among heating professionals, as a number of Btus rather than a *rate of flow* in Btus per hour (Btu/h). This is like stating the speed of a car in miles rather than miles per hour. Recall from the discussion of thermal equilibrium in Chapter 1 that when the rate of heat flow into a system matches the rate of heat flow out of a system, the temperature of that system remains constant. In the case of a building, *when the rate of heat input to the building equals the rate the building loses heat, the indoor temperature remains constant.* This is also true for each room within the building.

Finally, the design heating load is estimated assuming the outside temperature is *near* its minimum value. This temperature, called the **97.5 percent design dry bulb temperature**, is not the absolute minimum temperature for the location. It is the temperature that the outside air is at or above during 97.5% of the year. Although outside temperatures do occasionally drop below the 97.5 percent design dry bulb temperature, the duration of these low temperature excursions is short enough that most buildings can "coast" through them using heat stored in their thermal mass. Use of the 97.5 percent design temperature for heating system design helps prevent oversizing of the heat source.

Building Heating Load Versus Room Heating Load

One method of calculating the design heating load of a building yields a single number that represents the design heating load of the entire building. This value is called the **building design heating load**, and it is useful for selecting the heating output of the heat source as well as estimating annual heating energy use. However, this single number is not sufficient for designing the heat distribution system. It does not tell the designer how to properly proportion the heat output of the heat source to each room. Even when the building heating load is properly determined, failure to properly distribute the total heat output can lead to overheating in one room and underheating in another.

One should not assume that individual **room design heating loads** are proportional to room floor area. For example, a small room with a large window area could have a greater heating load than a much larger room with smaller windows.

The proper approach is to perform a design heating load estimate calculation for each room. Then, once these individual room heating loads are determined, add together to obtain the building design heating load.

Why Bother with Heat Load Calculations?

Would an electrician select wiring without knowing the amperage it must carry? Would a structural engineer select steel beams for a bridge without knowing the forces they must carry? Obviously, the answer to these questions is no. Yet, many so-called heating professionals routinely select equipment for heating systems with only a guess as to the building's heating load.

Statements attempting to justify this approach range from "I haven't got time for doing all those calculations," to "I'd rather be well oversized than get called back on the coldest day of the winter," to "I did a house like this one a couple of years ago, it can't be much different." Instead of listing excuses, let's look at some of the consequences of not properly estimating the building heating load.

- If the estimated heat loss is too low, the building will be uncomfortable during cold weather (a condition few homeowners will tolerate). An expensive callback will likely result.
- Grossly *overestimated* heating loads lead to systems that needlessly increase installation cost. Oversized systems may also cost more to operate due to lower heat source efficiency. This reduced efficiency can waste *thousands* of dollars worth of fuel over the life of the system.

Most building owners never know if their heating systems are oversized and simply accept the fuel usage of these systems as normal. Most trust the heating professional they hire to properly size and select equipment in their best interest. Heating professionals who deserve that trust adhere to the following principle:

Before attempting to design a space heating system of any sort, always calculate the design heating load of each room in the building using credible methods and data.

The remainder of this chapter shows you how to do this.

2.3 Conduction Heat Losses

Conduction is the process by which heat moves through a solid material whenever a temperature difference exists across that material. The rate of conduction heat transfer depends on the **thermal conductivity** of the material as well as the temperature difference across it. The relationship between these quantities is given in Equation 2.1.

Equation 2.1:

$$Q = A\left(\frac{k}{\Delta x}\right)(\Delta T)$$

where,

Q = rate of heat transfer through the material (Btu/h)
k = thermal conductivity of the material (Btu/T·h·ft)
Δx = thickness of the material in the direction of heat flow (ft)
ΔT = temperature difference across the material (°F)
A = area across which heat flows (ft^2)

Example 2.1

Determine the rate of conduction heat transfer through the 6-inch-thick wood panel shown in Figure 2-1.

Solution:

As shown in Figure 2-1, wood has a thermal conductivity of 0.1 Btu/°F•h•ft. Substituting the data into Equation 2.1:

$$Q = A\left(\frac{k}{\Delta x}\right)(\Delta T)$$

$$Q = 200\ \text{ft}^2\left(\frac{0.1\ \text{Btu/°F·h·ft}}{0.5\ \text{ft}}\right)(70°\text{F} - 10°\text{F})$$

$$= 2{,}400\ \text{Btu/h}$$

Discussion:

Notice how the units of the inserted data cancel out so that the resulting units are Btu/h. Thermal conductivity data for various materials may be stated in units other than those given below Equation 2.1. If so, be sure to convert that conductivity value to the stated units, or the results of the calculation will be invalid.

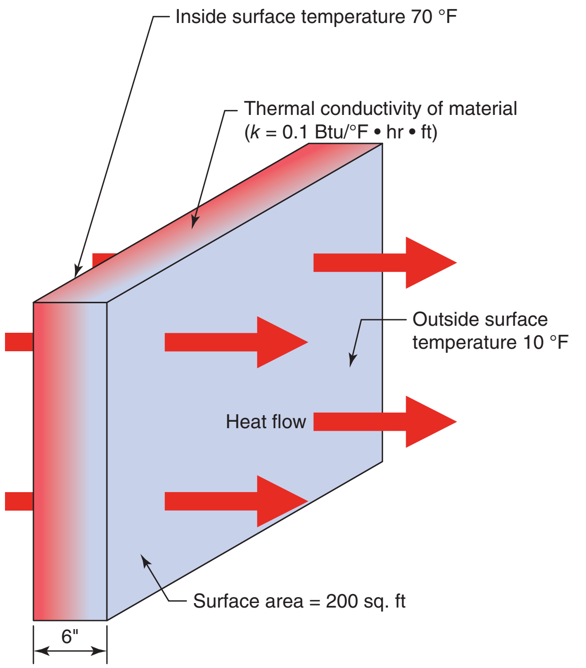


Figure 2-1 Heat flow through a material by conduction.

Thermal Resistance of a Material

Since building designers are usually interested in *reducing* the rate of heat flow from a building, it is more convenient to think of a material's *resistance* to heat flow rather than its thermal conductivity. An object's **thermal resistance** (also referred to as its ***R*-value**) can be defined as its thickness in the direction of heat flow divided by its thermal conductivity.

Equation 2.2:

$$R\text{-value} = \frac{\text{thickness}}{\text{thermal conductivity}} = \frac{\Delta x}{k}$$

The greater the thermal resistance of an object, the slower heat passes through it when a given temperature differential is maintained across it. Equations 2.1 and 2.2 can be combined to yield an equation that is convenient for estimating heating loads:

Equation 2.3:

$$Q = \left(\frac{A}{R}\right)(\Delta T)$$

where,

Q = rate of heat transfer through the material (Btu/h)
ΔT = temperature difference across the material (°F)
R = R-value of the material (°F·ft²·h/Btu)
A = area across which heat flows (ft²)

Some interesting facts can be demonstrated with this equation. First, *the rate of heat transfer through a given object is directly proportional to the temperature difference (ΔT) maintained across it.* If this temperature difference were doubled, the rate of heat transfer would also double.

Second, *the rate of heat transfer through an object is inversely proportional to its R-value.* For example, if its R-value were doubled, the rate of heat transfer through the object would be halved (assuming the temperature difference across the object remained constant).

Finally, the larger the object's surface area, the greater the rate of heat transfer through it. For example, a 4-foot by 4-foot window with an area of 16 ft² would transfer heat twice as fast as a 2-foot by 4-foot window with an area of 8 ft².

Example 2.2

Determine the rate of heat transfer through the 6-inch-thick wood panel shown in Figure 2-2.

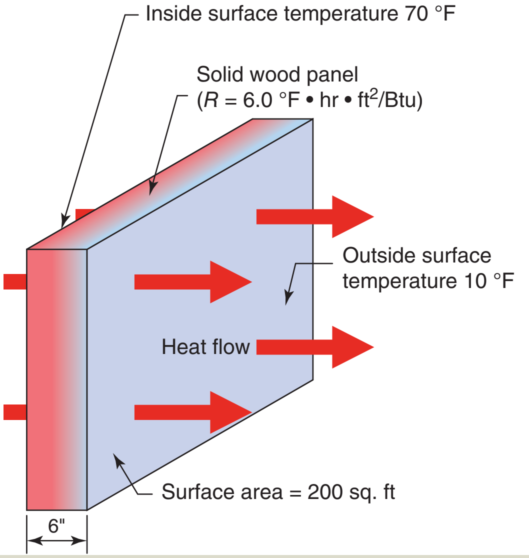


Figure 2-2 | Heat flow through a panel by conduction (using R-value).

Solution:

Substituting the relevant data into Equation 2.3:

$$Q = \left(\frac{A}{R}\right)(\Delta T) = \left(\frac{200}{6}\right)(70 - 10) = 2{,}000 \text{ Btu/h}$$

The R-value of a material is also directly proportional to its thickness. If a 1-inch-thick panel of extruded polystyrene insulation has an R-value of 5.4, a 2-inch-thick panel would have an R-value of 10.8, and a 1/2-inch-thick piece would have an R-value of 2.7. This relationship is very useful when the R-value of one thickness of a material is known, and the R-value of a different thickness needs to be determined.

The R-value of a material is slightly dependent on the material's temperature. As the temperature of the material is lowered, its R-value increases slightly. However, for most building materials, the change in R-value is small over the temperature ranges the material typically experiences and thus may be assumed to remain constant.

Total *R*-Value of an Assembly

Since the walls, floors, and ceilings of buildings are rarely constructed of a single material, it is often necessary to determine the total R-value of an assembly made up of several materials in contact with each other. This is done by adding up the R-values of the individual

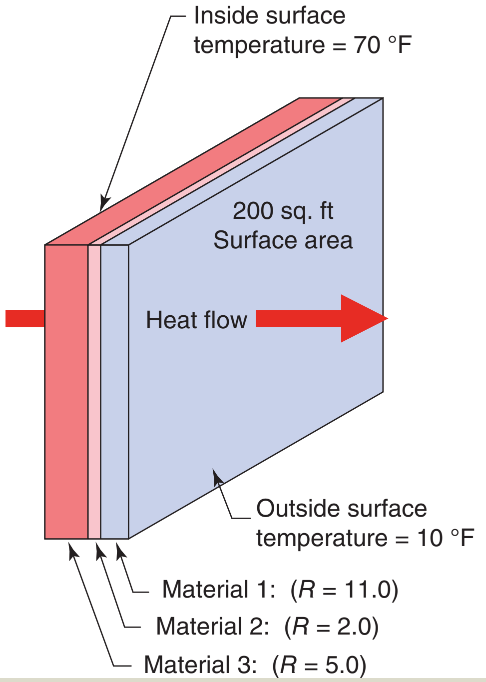


Figure 2-3 | Wall assembly consisting of three materials having different *R*-values.

materials. For example, the total *R*-value of the assembly shown in Figure 2-3 is:

$$R_{\text{total}} = 11.0 + 2.0 + 5.0 = 18.0$$

The heat flow across the panel can now be calculated using the **total *R*-value** substituted into Equation 2.3:

$$Q = \left(\frac{A}{R}\right)(\Delta T) = \left(\frac{200}{18}\right)(70 - 10) = 667 \text{ Btu/h}$$

The *R*-values of many common building materials can be found in Appendix B. To find the *R*-values of other materials, consult manufacturer's literature or the current edition of the *ASHRAE Handbook of Fundamentals.*

Air Film Resistances

In addition to the thermal resistances of the solid materials, there are two thermal resistances, called the **inside air film** resistance and **outside air film** resistance, which affect the rate of heat transfer through a panel separating inside and outside spaces. These thermal resistances represent the insulating effects of thin layers of air that cling to all surfaces. The *R*-values of these air films are dependent on surface orientation, air movement along the surface, and reflective qualities of the surface. Values for these resistances can also be found in Appendix B. The *R*-values of air films include the combined effect of conduction, convection, and radiation heat transfer, but are stated as a conduction-type thermal resistance for simplicity.

Example 2.3 shows how the total *R*-value of the wall assembly shown in Figure 2-3 is affected by the inclusion of the air film resistances. Notice that the temperatures used to determine the ΔT in this example are the indoor and outdoor *air temperatures*, rather than the *surface temperatures* used in the previous examples.

Example 2.3

Determine the rate of heat transfer through the assembly shown in Figure 2-4. Assume still air on the inside of the wall and 15-mph wind outside. See Appendix B for values of the air film resistances.

Solution:

The total *R*-value of the assembly is again found by adding the *R*-values of all materials, including the *R*-values of the inside and outside air films:

$$R_{\text{total}} = 0.17 + 11.0 + 2.0 + 5.0 + 0.68 = 18.85$$

The rate of heat transfer through the assembly can once again be found using Equation 2.3:

$$Q = \left(\frac{A}{R}\right)(\Delta T) = \left(\frac{200}{18.85}\right)(70 - 10) = 637 \text{ Btu/h}$$

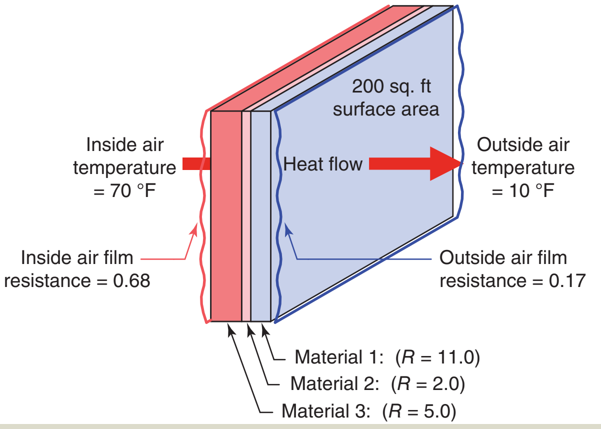


Figure 2-4 | Air film resistances add *R*-value to a wall assembly.

Discussion:

Including the R-value of the inside and outside air films caused a slight decrease in the rate of heat transfer through the panel. For the panel construction shown in Figure 2-4, the decrease in the rate of heat transfer was about 4.5%. The greater the total R-value of the materials making up the panel, the smaller the effect of the air film resistances. However, the R-value of the air films should always be included when determining the total R-value of an assembly.

Effect of Framing Members

The rate of heat flow through assemblies made of several materials stacked together like a sandwich can now be calculated. In such situations, and when edge effects are considered insignificant, the heat flow through the assembly is uniform over each square foot of surface area. However, few buildings are constructed with walls, ceilings, and other surfaces made of simple stacked layers of materials. This is particularly true of wood-framed buildings with wall studs, ceiling joists, window headers, and other framing members. Wooden framing members that span across the insulation cavity of a wall tend to reduce its **effective total *R*-value**. This is because the thermal resistance of wood, about 1.0 per inch of thickness, is usually less than the thermal resistance of insulation materials it displaces. Thus, the rate of conduction through framing materials is usually faster than through the surrounding insulation materials. This effect can be detected with infrared thermography, as shown in Figure 2-5. In this case, the wall studs as well as top and bottom plates are readily visible to the infrared camera because they create "stripes" of higher-surface temperature along the outside wall surface on a cold winter night.

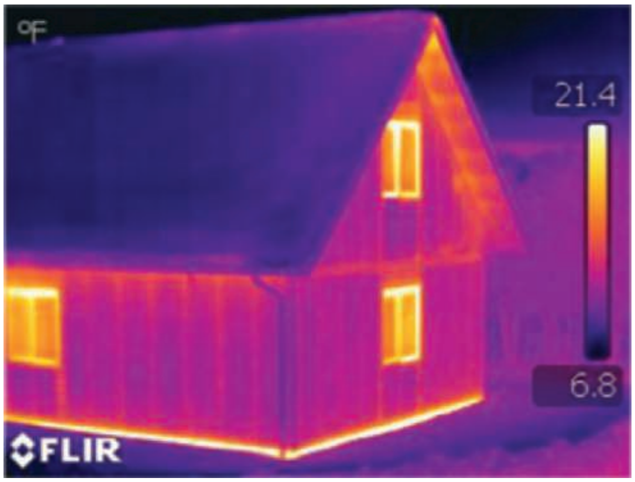


Figure 2-5 Infrared thermograph of framed wall reveals "stripes" of higher outside surface temperatures due to lower R-value of studs relative to cavity insulation.

It is possible to adjust the R-value of an assembly to include the thermal effects of framing. This requires that the assembly's total R-value be calculated both between the framing and at the framing. The resulting total R-values are then weighted according to the percentage of solid framing in the assembly. Equation 2.4 can be used to calculate an "effective total R-value," which can then be applied to the entire panel area.

Equation 2.4:

$$R_{\text{effective}} = \frac{(R_i)(R_f)}{p(R_i - R_f) + R_f}$$

where,

$R_{\text{effective}}$ = effective total R-value of panel (°F·ft²·h/Btu)

P = percentage of panel occupied by framing (decimal %)

R_f = R-value of panel at framing (°F·ft²·h/Btu)

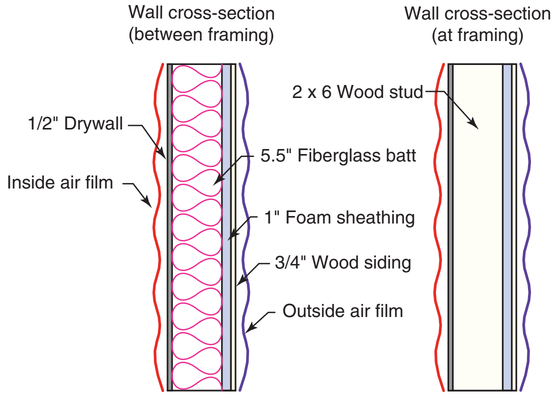
R_i = R-value of panel at insulation cavities (°F·ft²·h/Btu)

Although the percentage of the panel occupied by framing could be calculated for every construction assembly, such calculations can be tedious and often make insignificant changes in the results. Typical wood-framed walls will have between 10 and 15% of the wall area as solid framing across the insulation cavity. The lower end of this range would be appropriate for 24 inches on-center framing, with insulated headers over windows and doors. The upper end of this range is typical of walls with 16 inches on-center framing and solid headers.

To illustrate these calculations, the effective total R-value of the wall assembly shown in Figure 2-6 will be calculated assuming that 15% of the wall is solid framing.

Notice that the effective R-value of this wall (R-22.9) is about 13.6% lower than the R-value through the insulation cavity (R-26.5). If the person estimating the heat loss simply ignored the effect of framing, the estimated heat loss for the wall would be 13.6% low. It follows that a heating system designed according to this underestimated heat loss might not be able to maintain comfort on a design load day.

To obtain the total heat loss of a room, the effective total R-value of each different **exposed surface** within the room must be determined using a procedure similar to that shown in Figure 2-6. Fortunately, many buildings have the same type of construction for most exposed



Material	*R*-value
Inside air film	0.68
1/2" Drywall	0.45
5.5" Fiberglass batt	19.0
1" Foam sheathing	5.4
3/4" Wood siding	0.8
Outside air film	0.17
Total *R*-value (between framing)	26.5

Material	*R*-value
Inside air film	0.68
1/2" Drywall	0.45
2 x 6 Stud	5.5
1" Foam sheathing	5.4
3/4" Wood siding	0.8
Outside air film	0.17
Total *R*-value (at framing)	13.0

$$R_{\text{effective}} = \frac{(R_i)(R_f)}{p\,(R_i - R_f) + R_f} = \frac{(26.5)(13.0)}{0.15\,(26.5 - 13.0) + 13.0} = 22.9$$

Figure 2-6 Calculating the effective total *R*-value of a wall assembly.

walls, ceilings, and so on. In such cases, the effective *R*-value of the assembly need only be calculated once.

It is good practice to make a sketch or CAD drawing of the cross-section of each exposed assembly (i.e., wall, ceiling, floor), similar to that shown in Figure 2-6. The *R*-value of each material can then be determined from Appendix B and the effective total *R*-value calculated. These sheets or CAD drawing files showing the cross-sections of the assemblies and their effective total *R*-values can be saved and referenced on other buildings with the same construction.

2.4 Foundation Heat Loss

Heat flow from a basement or slab-on-grade foundation is determined by complex interactions between the building, the surrounding soil, insulation materials (if present), and the air temperature above grade.

Figure 2-7a shows a **finite element analysis** computer simulation of heat flow from a residential basement during February, in a cold winter climate. The basement has a width of approximately 26 feet. Only half of this width is shown because heat flow is assumed to be symmetrical about a vertical centerline through the basement. The floor slab is heated by embedded tubing and is assumed to be covered by a ¾-inch-thick hardwood flooring. *There is no under-slab insulation.* The basement walls are insulated with 2 inches of expanded polystyrene.

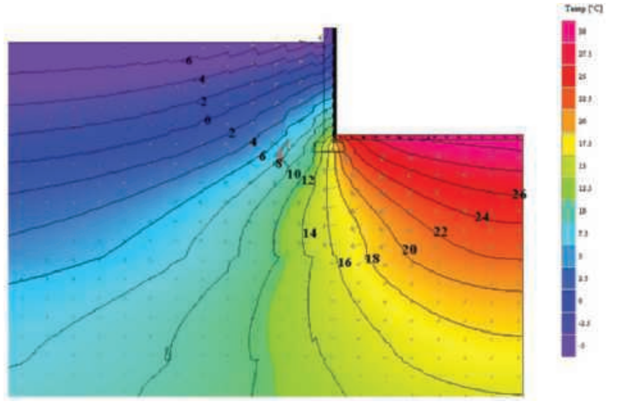


Figure 2-7a Simulated heat flow through a basement wall and uninsulated heated slab floor. *Courtesy of Beaver Plastics LTD.*

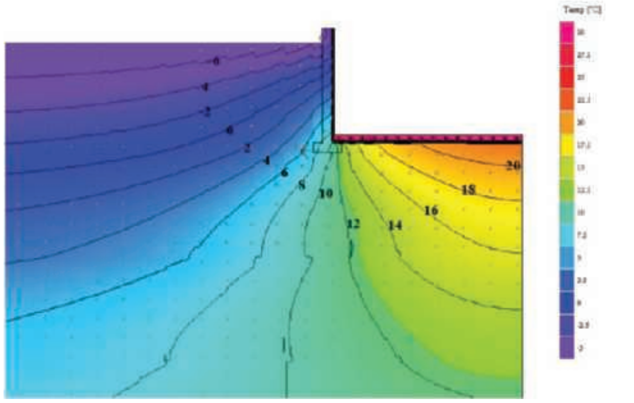


Figure 2-7b Simulated heat flow through a basement wall and heated slab floor with 2 inches of expanded polystyrene insulation below floor slab. *Courtesy of Beaver Plastics LTD.*

The contour lines with temperatures labeled in °C are called **isotherms**. They represent locations having the same temperature. Heat flow is perpendicular to the isotherms in all locations. In this case, heat flows downward from the slab, passes under the footing, then passes outward and upward toward the cold exposed soil surface. In this case, almost half of the purchased energy used to heat the slab floor is being driven downward into the soil.

Figure 2-7b shows the same situation, but with 2 inches of expanded polystyrene insulation installed under the floor slab. Notice that soil temperatures below the slab are significantly lower, because less heat is being driven downward. It follows that heat loss from this basement will be significantly less. For the cold northern climate assumed in this simulation, which represented an average between the typical weather conditions in Toronto, Ontario, and Calgary, Alberta, the use of 2-inch expanded polystyrene insulation under the heated slab reduced heat loss by approximately 50%. The energy savings over a complete heating season were estimated at 8,945 Btu/ft^2 of floor area.

The results of complex computer simulations of foundation heat loss have been used to develop simpler calculation procedures that can be used for load estimating. One such method is useful for estimating the heat loss of partially buried basement walls, the other for estimating the heat loss from floor slabs.

Heat Loss Through Basement Walls

The heat loss of a basement wall is significantly affected by the height of soil against the outside of the wall. Soil adds thermal resistance between the wall and the outside air and thus helps reduce heat loss.

Figure 2-8 illustrates the theoretical heat flow paths through a uniformly insulated basement wall and adjacent soil. Notice that the heat flow paths from the inside basement air to the outside air are longer for the lower portions of the wall. As the path length increases, so does the total thermal resistance of the soil along that path.

One method of approximating the heat loss through basement walls is based on dividing the wall into horizontal strips based on the height of finish grade. The upper strip includes all wall area exposed above grade as well as the wall area to a depth of 2 feet below grade. The implicit assumption is that by mid-winter, shallow soils will be at approximately the same temperature as the outside air. The middle strip includes the wall area from 2 feet to 5 feet below grade. The lower strip includes all wall area deeper than 5 feet below grade. Figure 2-9 illustrates these areas.

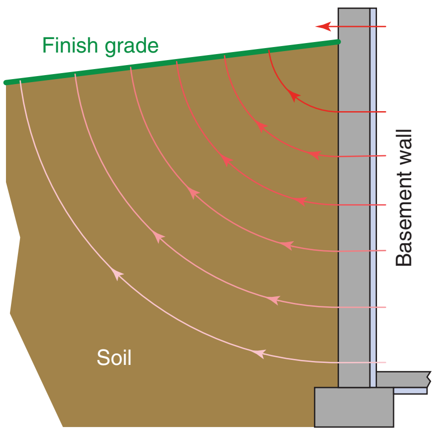


Figure 2-8 | Theoretical heat flow paths through a uniformly insulated basement wall.

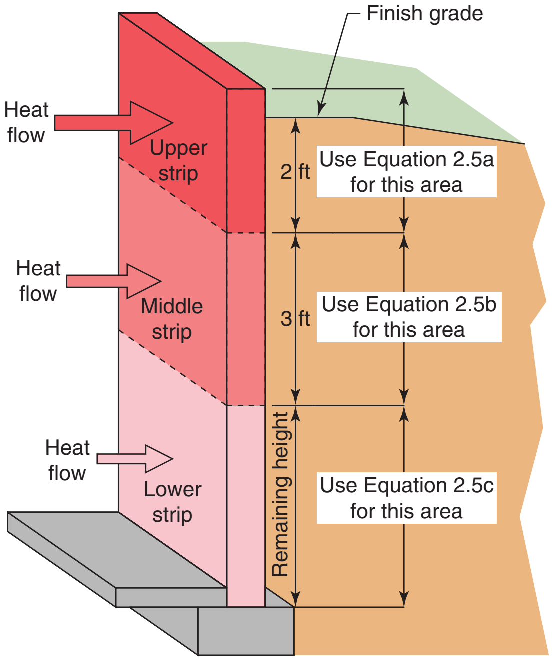


Figure 2-9 | Dividing the basement wall into three horizontal strips to estimate heat loss.

Use Equation 2.5a to calculate the effective *R*-value of basement walls for all above grade areas as well as areas down to 2 feet below grade.

Equation 2.5a:

$$R_{\text{effective}} = R_{\text{wall}} + R_{\text{insulation}}$$

Use Equation 2.5b to calculate the effective *R*-value of basement walls for areas from 2 feet to 5 feet below grade.

Equation 2.5b:

$$R_{\text{effective}} = 7.9 + 1.12(R_{\text{insulation}})$$

Use Equation 2.5c to calculate the effective *R*-value of basement walls for areas deeper than 5 feet below grade.

Equation 2.5c:

$$R_{\text{effective}} = 11.3 + 1.13(R_{\text{insulation}})$$

The parameter ($R_{\text{insulation}}$) in Equations 2.5a through 2.5c represents the *R*-value of any insulation *added* to the foundation wall over the particular strip for which the effective *R*-value is being determined. The foundation wall itself is assumed to be a masonry or concrete wall between 8 and 12 inches thick.

Once the *R*-values of each wall strip have been determined, use Equation 2.3 to calculate the heat loss of each area. In each case, the ΔT in Equation 2.3 is the difference between the basement air temperature and the outside air temperature.

Example 2.4

Estimate the heat loss through a 10-inch-thick concrete basement wall 9 feet deep with 6 inches exposed above grade. The wall is 40 feet long and has 2 inches of extruded polystyrene insulation (*R*-11) added on the outside over its full depth. The basement air temperature is 70 °F. The outside air temperature is 10 °F. Assume the *R*-value of the 10-inch concrete wall is 1.0. A drawing of the wall is shown in Figure 2-10.

Solution:

For the upper wall strip, use Equation 2.5a to determine the effective *R*-value. Then use Equation 2.3 to calculate the heat loss.

$$R = 1.0 + 11 = 12$$
$$A = 40\,\text{ft}\,(2.5\,\text{ft}) = 100\,\text{ft}^2$$
$$\Delta T = (70\,°\text{F} - 10\,°\text{F}) = 60\,°\text{F}$$

$$Q = \left(\frac{A}{R}\right)(\Delta T) = \left(\frac{100}{12}\right)(60) = 500\ \text{Btu/h}.$$

For the middle wall strip, use Equation 2.5b to determine the effective *R*-value. Then use Equation 2.3 to calculate the heat loss.

$$R = 7.9 + 1.12(R_{\text{added}}) = 7.9 + 1.12(11) = 20.2$$
$$A = 40\,\text{ft}(3\,\text{ft}) = 120\,\text{ft}^2$$
$$\Delta T = (70\,°\text{F} - 10\,°\text{F}) = 60\,°\text{F}$$
$$Q = \left(\frac{A}{R}\right)(\Delta T) = \left(\frac{120}{20.2}\right)(60) = 356\ \text{Btu/h}.$$

For the lower wall strip, use Equation 2.5c to calculate the *R*-value. Then use Equation 2.3 to calculate the heat loss.

$$R = 11.3 + 1.13(R_{\text{added}}) = 11.3 + 1.13(11) = 23.7$$
$$A + 40\,\text{ft}(3.5\,\text{ft}) = 140\,\text{ft}^2$$
$$\Delta T = (70\,°\text{F} - 10\,°\text{F}) = 60\,°\text{F}$$
$$Q = \left(\frac{A}{R}\right)(\Delta T) = \left(\frac{140}{23.7}\right)(60) = 354\ \text{Btu/h}.$$

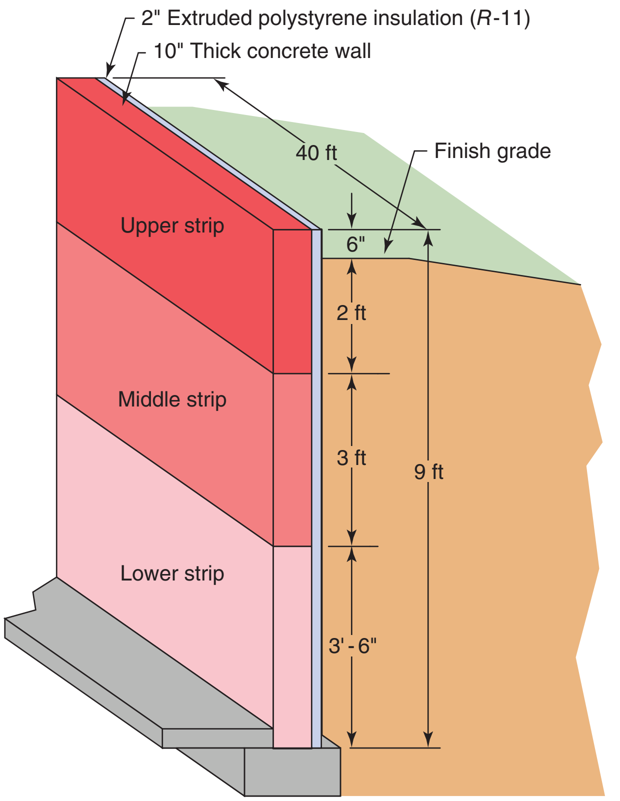


Figure 2-10 | Basement wall used in Example 2.4.

The total heat loss for this wall is the sum of the heat losses from each wall strip:

$$Q_{\text{total}} = 500 + 356 + 354 = 1{,}210\,\text{Btu/h}.$$

Discussion:

Note that the same ΔT term (the difference between the basement air temperature and the outside air temperature) was used in all three calculations. It should also be noted that these results are based on a two-dimensional heat transfer model of the situation. Such a model does not account for the more complex three-dimensional heat transfer at wall corners, which will slightly increase basement heat loss. In general, narrow basements with several corners will have greater rates of heat loss compared to wider basements with minimal corners.

Heat Loss Through Basement Floors

Equation 2.6a can be used to estimate downward heat loss for basement floor slabs that are at least 2 feet below finish grade:

Equation 2.6a:

$$Q_{\text{bslab}} = \left(\frac{A}{R_{\text{bslab}}}\right)\Delta T$$

where,

Q_{bslab} = rate of heat loss through the basement floor slab (Btu/h)

A = floor area (ft^2)

R_{bslab} = effective R-value of slab *(designated as R_i in Equation 2.6b, or as R_u in Equation 2.6c)*

ΔT = a. (basement air temperature minus outside air temperature) (°F) *for unheated slabs*

b. (average slab temperature minus outside air temperature) (°F) *for heated slabs*

The effective R-value of the basement slab (R_{bslab}) is found using either Equation 2.6b or 2.6c. Use Equation 2.6b if the slab is *insulated with a least R-3 (°F·h·ft^2/Btu) underside insulation*. Use Equation 2.6c if the slab is *uninsulated*. Equations 2.6b and 2.6c are based on the width of the shortest side of the floor slab.

Equation 2.6b:

$$R_i = 20.625 + 1.6063(w)$$

Equation 2.6c:

$$R_u = 14.31 + 1.1083(w)$$

where,

R_i = effective R-value for *insulated* slab (°F·h·ft^2/Btu)

R_u = effective R-value for *uninsulated* slab (°F·h·ft^2/Btu)

w = width of *shortest* side of floor slab (ft)

Example 2.5

Determine the rate of heat loss from an unheated basement floor slab measuring 30 feet by 50 feet, when the outside temperature is 10 °F, and the basement air temperature is 65 °F. Assume the basement floor has no underside insulation.

Solution:

Since the slab has no underside insulation, Equation 2.6c will be used to calculate the effective total R-value.

$$R_u = 14.31 + 1.1083(w)$$
$$= 14.31 + 1.1083(30) = 47.56$$

The heat loss from the slab can now be determined using Equation 2.6a.

$$Q_{bslab} = \left(\frac{A}{R_{bslab}}\right)\Delta T = \frac{30 \times 50}{47.56}(65 - 10) = 1{,}735 \text{ Btu/h}$$

Discussion:

The effective R-value of the slab determined using either Equation 2.6b or Equation 2.6c includes the effects of the slab, underside insulation (if present), and several feet of soil in the path of heat flow from the basement air to the outside air. Thus, the effective R-value in relatively high, even for an uninsulated slab.

Heat Loss Through Slab-on-Grade Floors

Many residential and commercial buildings have concrete slab-on-grade floors rather than crawl spaces or full basements. Heat flows from the floor slab downward to the surrounding earth and outward through any exposed edges of the slab. Of these two paths, outward heat flow through the slab edge tends to dominate. This is because the outer edge of the slab is exposed to outside air or to soil at approximately the same temperature as the outside air. Soil under the slab and several feet in from the perimeter tends to stabilize at somewhat higher temperatures once the building is maintained at normal comfort conditions. Hence, downward heat losses are relatively small from interior areas of the floor slab. Areas of the floor close to the exposed edges have higher rates of heat loss. This often justifies the use of higher R-value insulation near the perimeter of the slab compared to under interior areas.

As with basement walls, the heat flow paths from a slab to the soil and outside air are complex and vary with foundation geometry, soil conditions, and the insulation materials used. Figure 2-11 shows isotherms and arrows indicating the direction of heat flow for a heated slab on grade with continuous underside and edge insulation.

The method used to estimate the heat loss from a slab-on-grade floor depends on the amount of insulation (if any) used near the perimeter of the slab, the orientation of that insulation, and the type of soil present at the perimeter of the slab. Dense soils with significant water content will conduct heat better than lighter/drier soils and thus increase rates of heat loss.

Two scenarios will be considered: One in which no underslab or edge insulation is used, and another where a rigid insulating panel having an R-value between 4 and 16 (°F·h·ft^2/Btu) is installed between the edge of the slab and the foundation as well as under the outer four feet of the slab. Both scenarios are shown in Figure 2-12.

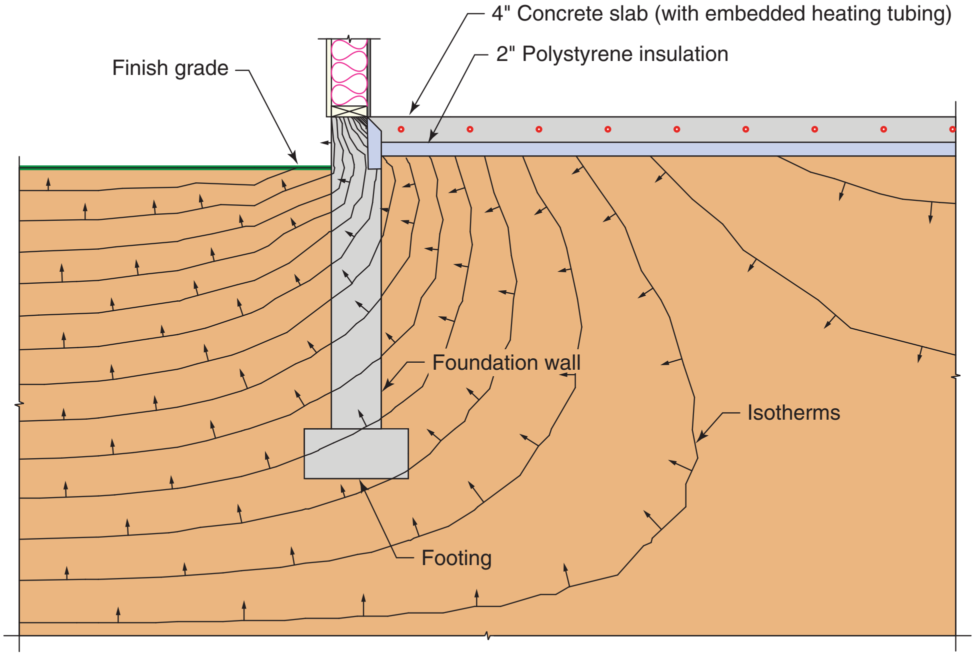


Figure 2-11 | Isotherms and arrows indicating the direction of heat flow fora heated slab-on-grade foundation.

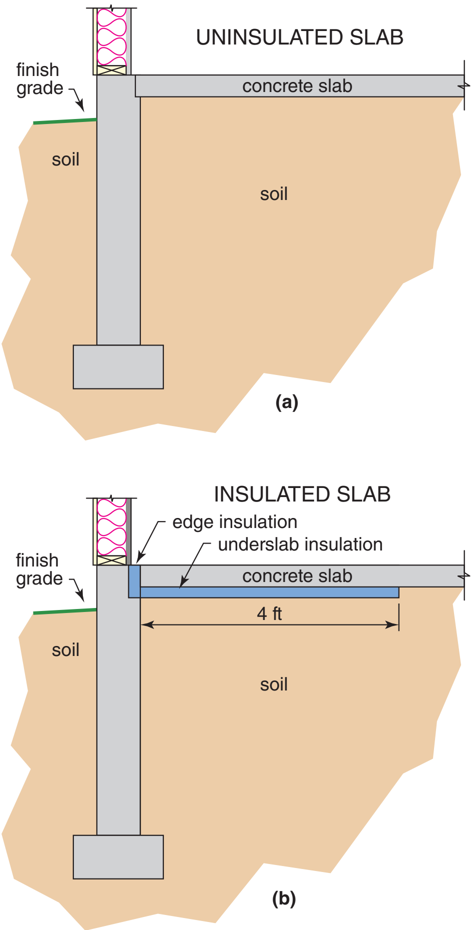


Figure 2-12 | (a) Uninsulated floor slab. (b) Insulated floor slab.

The heat loss from an *uninsulated* slab can be estimated using Equation 2.7a.

Equation 2.7a:

$$Q_{slab} = c_0 L(\Delta T)$$

where,

Q_{slab} = rate of heat loss through the slab edge and interior floor area (Btu/h)

L = exposed edge length of the floor slab (ft)

ΔT = a. difference between inside and outside air temperature (°F) for unheated slabs

b. difference between average slab temperature and outside air temperature (°F) for heated slabs

c_0 = a number based on soil type (see table in Figure 2-13)

Heavy/damp soil	Heavy/dry soil (or) light/damp soil	Light/dry soil
c_0 = 1.358	c_0 = 1.18	c_0 = 0.989

Figure 2-13 | Values of c_0 for Equation 2.7a (uninsulated slabs).

Example 2.6

Determine the rate of heat loss from the shaded portion of an unheated and uninsulated floor slab shown in Figure 2-14. The soil outside the slab is classified as heavy and damp. The inside air temperature is 70 °F and the outside air temperature is 10 °F.

Solution:

Only two sides of the room's slab are exposed to the outside. The total linear footage of exposed slab edge is 20 + 25 = 45 feet. The value of c_0 for heavy/damp soil in Figure 2-13 is 1.358. Substituting the data into Equation 2.7a yields:

$$Q_{slab} = c_0 L(\Delta T) = 1.358(45)(70 - 10) = 3{,}670 \text{ Btu/h}$$

Discussion:

It should also be stressed that unlike many of the equations used for estimating heat loss, Equation 2.7a is based on the *length* of the exposed perimeter of the slab and not the slab's area.

If the slab contained embedded tubing for heating, it would typically be at temperatures higher than room air temperature during the heating season. This would increase the rate of downward and outward heat loss. *It is suggested that the air temperature in Equation 2.7a be replaced by a temperature that is 25 °F higher than room air temperature when estimating the heat loss of a heated floor slab under design load conditions.*

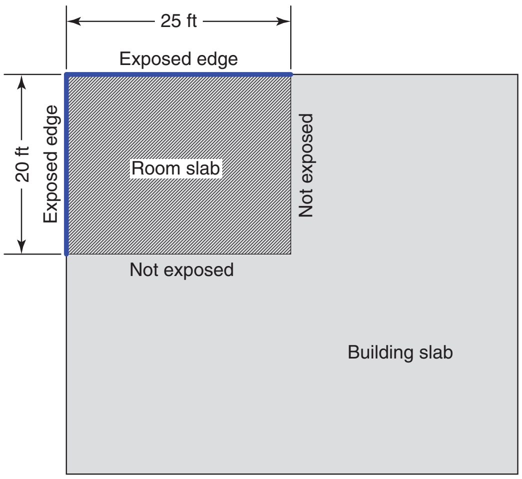


Figure 2-14 Room with exposed slab edge used in Example 2.6.

If the slab-on-grade floor is *insulated* as shown in Figure 2-12b, with insulation having an R-value between 4 and 16 (°F·h·ft²/Btu), the heat loss can be determined using Equation 2.7b.

Equation 2.7b:

$$Q_{slab} = L(\Delta T)\left[b_0 + b_1 R_{edge} + b_2 (R_{edge})^2\right]$$

where,

Q_{slab} = rate of heat loss through the slab edge and interior floor area (Btu/h)

L = exposed edge length of the floor slab (ft)

ΔT = (a. difference between inside and outside air temperature (°F) for unheated slabs)/(b. difference between the average slab temperature and outside air temperature (°F) for heated slabs)

R_{edge} = R-value of the edge and underslab insulation (°F·h·ft²/Btu) ($4 \le R_{edge} < 16$)

b_0, b_1, b_2 = constants based on soil type (see Figure 2-15)

Example 2.7

Determine the rate of heat loss from the shaded portion of the unheated floor slab shown in Figure 2-14, *assuming 2 inches of extruded polystyrene insulation has been placed at the edge of the slab and under the outer 4 feet of the slab*. The soil outside the slab remains classified as heavy and damp. The inside air temperature is 70 °F and the outside air temperature is 10 °F.

Solution:

Using the data in Appendix B, the R-value of 2 inches of extruded polystyrene insulation is $2 \times (5.4) = 10.8$ (°F·h·ft²/Btu). Substituting the values of b_o, b_1, and b_2

Heavy/damp soil	$b_0 = 0.755$	$b_1 = -0.0425$	$b_2 = -0.00126$
Heavy/dry soil (or) Light/damp soil	$b_0 = 0.595$	$b_1 = -0.0360$	$b_2 = -0.00108$
Light/dry soil	$b_0 = 0.411$	$b_1 = -0.0293$	$b_2 = -0.0009$

Figure 2-15 Values of b_0, b_1, and b_2 for Equation 2.7b (insulated slabs).

for heavy/damp soil along with the remaining data into Equation 2.7b yields the following:

$$Q_{slab} = 45(70 - 10)\,[0.755 + (-0.0425)10.8 + (0.00126)\,(10.8)^2]$$
$$= 1{,}196 \text{ Btu/h}$$

Discussion:

Comparing the results of Examples 2.6 and 2.7 shows that there is a 67% decrease in heat loss from the slab when 2 inches of extruded polystyrene insulation is installed. Insulating slab floors in heated buildings is essential to good performance. The estimated heat loss includes loss through the exposed edge of the slab as well as downward heat flow from the shaded interior area. Equation 2.7b should only be used when the *R*-value of the underslab insulation is between 4 and 16 (°F·h·ft²/Btu).

Designers and building owners should consider that there is most likely only one opportunity to properly insulate a concrete slab-on-grade floor. Although "retrofitting" underslab insulation is not impossible, it is extremely disruptive and very expensive. From a practical standpoint, such retrofitting is very unlikely to happen. Thus, the choice of underslab insulation will most likely have to suffice for the life of the building. It is imperative to "get it right" the first time. The author recommends a *minimum* of 2-inch-thick extruded polystyrene insulation ($R - 10.8$ °F•h•ft²/Btu) under *all* slab-on-grade floor areas in heated spaces. If the building program emphasizes high energy efficiency, even more underslab insulation (3 to 4 inches of extruded polystyrene) is likely justified, especially in cold northern climates. This insulation should be placed under all floor areas with exception of footings for load bearing columns or walls. Extruded polystyrene insulation is available with compressive stress ratings of 15 to 100 psi to accommodate slabs ranging from interior residential floors to heavy vehicle maintenance garages and aircraft hangers.

Heat Loss Through Windows, Doors, and Skylights

The windows, doors, and skylights in a room can represent a large percentage of that room's total heat loss. The construction of windows and skylights can range from old, single glass assemblies with metal frames to modern multi-pane assemblies filled with Argon gas to suppress convection, low-E (low emissivity) coatings to suppress radiation heat transfer, and insulated polymer frames to reduce edge conduction losses. Doors can range from solid wood to foam-core insulated panels with steel or fiberglass claddings.

The thermal resistance of a typical window, door, or skylight unit varies depending upon *where* on the cross-section of the unit the thermal resistance is evaluated. For example, a wooden window frame will provide higher thermal resistance than will two panes of glass with an air space in between. Similarly, the thermal resistance at the edge of a multi-pane glazing assembly will be less than that measured at the center of the glazing unit. This is the result of greater conduction through the edge spacers that separate the panes versus the air space at the center of the unit.

To provide a uniform and simplified approach to dealing with these effects, manufacturers, working cooperatively through the National Fenestration Rating Council (NFRC), have developed a standard for determining the **unit *U*-value** of a window, door, or skylight unit.

The *R*-value is the reciprocal of *U*-value. If either the *U*-value or *R*-value is known, the other can be easily determined.

$$R = \frac{1}{U}$$

The lower the unit *U*-value of a window, door, or skylight, the lower its heat loss, all other conditions being equal.

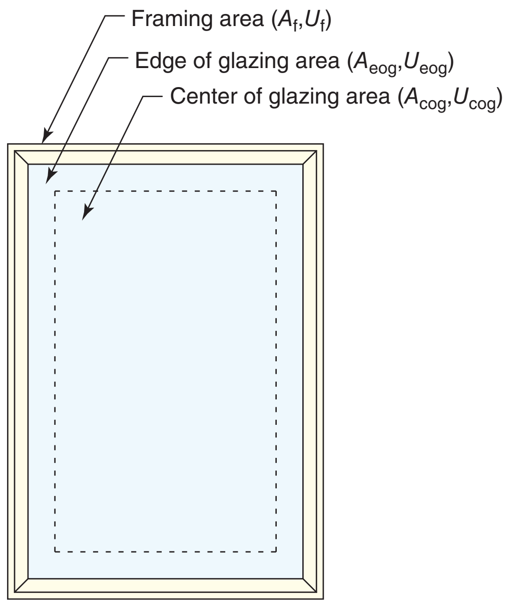
The methods used in the standard NFRC 100 *Procedures for Determining Fenestration Product U-Factor* account for the thermal resistance at the window frame as well as at the edge and center of the glazing unit. These thermal resistances are then "area weighted" based on the shape of the unit. An example of how this would apply to a simple rectangular window unit is shown in Figure 2-16.

The unit *U*-value number can be thought of as the *effective U*-value of the unit. Values for it can be found in the specifications published by window and door manufacturers. Many building codes now require windows and doors used in new construction or remodeling to have unit *U*-values at, or below, a specified value.

Once the unit *U*-value is known, its reciprocal (unit *R*-value) can be used in the same manner at the effective *R*-value of a wall or ceiling when heat loads are calculated.

Thermal Envelope of a Building

Heat is lost through all building surfaces that separate heated space from unheated space. Together, these surfaces (walls, windows, ceilings, doors, foundation, etc.)



$$U_{unit} = \frac{(A_f U_f + A_{eog} U_{eog} + A_{cog} U_{cog})}{(A_f + A_{eog} + A_{cog})}$$

$$R_{unit} = \frac{1}{U_{unit}}$$

Figure 2-16 | Areas and formula for determining unit U-value of a rectangular window unit.

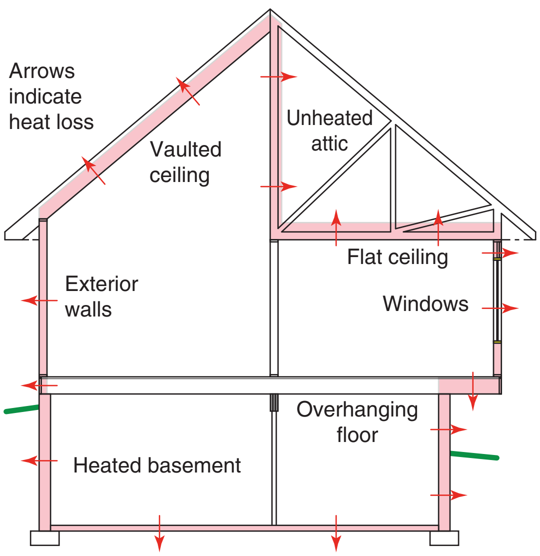


Figure 2-17 | A portion of the thermal envelope of a house (shown shaded).

are called the **thermal envelope** of a building. *It is only necessary to calculate heat flow though the surfaces that constitute this thermal envelope.* Walls, ceilings, floors, or other surfaces that separate one heated space from another should not be included in these calculations. Figure 2-17 illustrates a part of the thermal envelope for a house.

One can also envision each room in a building as a "compartment." The thermal envelope of that compartment would consist of all surfaces that *separate heated space from unheated space*. Heat loss calculations would only be performed for these surfaces.

2.5 Infiltration Heat Loss

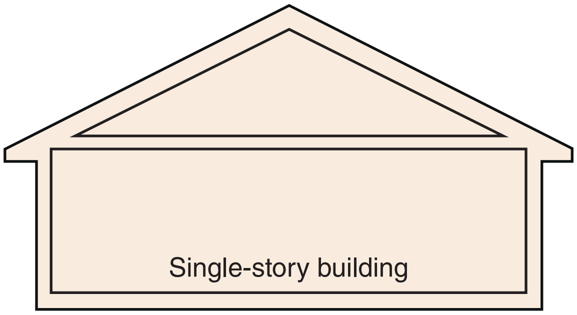
In addition to conduction losses, heat is also carried out of buildings by uncontrolled air leakage. This is called **infiltration heat loss**. The faster air leaks into and out of a building, the faster heat is carried away by that air. The thermal envelope of an average building contains hundreds of imperfections through which air can pass. Some leakage points, such as fireplace flues, exhaust hoods, and visible cracks around windows and doors, are obvious. Others are small and out of sight, but when taken together represent significant amounts of leakage area. These include cracks where walls meet the floor deck, air leakage through electrical outlets, small gaps where pipes pass through floors or ceilings, and many other small imperfections in the thermal envelope. Some of these may be the result of structural or aesthetic detailing of the building.

It would be impossible to assess the location and magnitude of all the air leakage paths in a typical building. Instead, an estimate of air leakage can be made based on the air sealing quality of the building. This approach is known as the **air change method** of estimating infiltration heat loss. It relies on a somewhat subjective classification of air sealing quality as well as experience. The designer chooses a rate at which the interior air volume of the building (or an individual room) is exchanged with outside air. For example: 0.5 air change per hour means that half of the entire volume of heated air in a space is replaced with outside air once each hour.

The infiltration rates shown in Figures 2-18a and 2-18b are suggested in *ACCA Manual J* (8th edition). They are categorized based on single- or two-story constructions. Use Figure 2-18a for single-story construction and Figure 2-18b for two-story construction.

The descriptions for air sealing quality (e.g., tight, semi-loose) are as follows:

- **Tight:** All structural joints and cracks are sealed using *meticulous* work along with air infiltration barriers, caulks, tapings, or packings. Window and doors are rated at less than 0.25 CFM per foot of

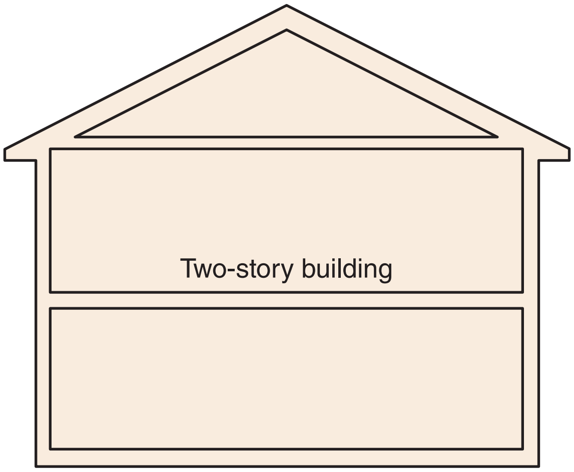


Single-story Floor area (ft^2)	900 or less	901–1,500	1,500–2,000	2,000–3,000	3,001 or more
Tight	0.21	0.16	0.14	0.11	0.10
Semi-tight	0.41	0.31	0.26	0.22	0.19
Average	0.61	0.45	0.38	0.32	0.28
Semi-loose	0.95	0.70	0.59	0.49	0.43
Loose	1.29	0.94	0.80	0.66	0.58

Figure 2-18a | Air change rates for single-story construction.

crack at 25-mph wind speed. All exhaust fans are equipped with backdraft dampers. The building either does not have recessed light fixtures piercing through thermal envelope surfaces, or has recessed lights with negligible air leakage. The house does not have powerful (150 CFM or greater) exhausting range hoods. The house either does not have combustion-type heat sources within conditioned space, or any combustion heat sources present are direct vented. If fireplaces are present, they are provided with their own combustion air and have tight-fitting glass doors.

- **Semi-Tight:** Conditions range between tight and average.
- **Average:** All structural joints and cracks are sealed using *average* work along with air infiltration barriers, caulks, tapings, or packings. Window and doors are rated at 0.25 to 0.50 CFM per foot of crack at 25-mph wind speed. All exhaust fans are equipped with backdraft dampers. The building either does not have recessed light fixtures piercing through the thermal envelope or has recessed lights with negligible air leakage. The house does not have powerful (150 CFM or greater) exhausting range hoods. The house either does not have combustion-type heat sources within conditioned space or any combustion heat sources present are direct vented. If fireplaces are present, they are provided with their own combustion air and have tight fitting glass doors.
- **Semi-Loose:** Conditions range between average and loose.



Two-story Floor area (ft^2)	900 or less	901–1,500	1,500–2,000	2,000–3,000	3,001 or more
Tight	0.27	0.20	0.18	0.15	0.13
Semi-tight	0.53	0.39	0.34	0.28	0.25
Average	0.79	0.58	0.50	0.41	0.37
Semi-loose	1.23	0.90	0.77	0.63	0.56
Loose	1.67	1.22	1.04	0.85	0.75

Figure 2-18b | Air change rates for two-story construction.

- **Loose:** There is little or no effort to seals joints or cracks in the building envelope. Windows and doors are either not rated or are rated for more than 50 CFM per foot of crack at 25-mph wind speed. There are many recessed light fixtures piercing through the thermal envelope that are not airtight. Exhaust fans are not equipped with backdraft dampers. Fireplaces, if present, do not have a source of outside combustion air or tight glass doors. Powerful kitchen exhaust blowers are present.

These air exchange rates are at best estimates and they vary over a wide range. For example, in a single-story building with 1,800 square feet of floor area, the air leakage rate associated with "loose" quality air sealing results in almost six times more heat loss due to air leakage compared to the same size building with "tight" air sealing.

Experience is also helpful in estimating infiltration rates. A visit to the building while it is under construction helps give the heating system designer a feel for the air sealing quality being achieved. An inspection visit to an existing building, though not as revealing, can still help the designer assess air sealing quality. Things to look for or consider include:

- Visible cracks around windows and doors
- Continuous spray foam insulation versus batt insulation

- Poorly installed or maintained weather stripping on doors and windows
- Slight air motion detected near electrical fixtures
- Smoke that disappears upward into ceiling lighting fixtures
- Deterioration of paint on the downwind side of building
- Poor-quality duct-system design (especially a lack of proper return air ducting)
- Interior humidity levels (drier buildings generally indicate more air leakage)
- Presence or lack of flue damper on fossil fuel heating system
- Windy versus sheltered building site
- Storm doors and entry vestibules help reduce air infiltration
- The presence of fireplaces (especially those of older masonry construction)
- Casement and awning windows generally have less air leakage than double hung or sliding windows
- Cold air leaking in low in the building generally indicates warm air leaking out higher in the building

Figure 2-18c | A blower door test being performed during house construction. *Courtesy of © John P. Gordon, AIA.*

Blower Door Testing

The airtightness of an existing or partially constructed building can often be assessed through blower door testing. A **blower door** consists of an adjustable frame covered by an airtight fabric that expands to form an airtight seal around the perimeter of an exterior door opening, as shown in Figure 2-18c. A variable speed fan mounts through the fabric panel.

When the fan in the door panel is operated, a negative air pressure is created within the building. This causes outside air to move into the building (e.g., infiltrate) through any possible leakage paths. Infiltrating air can be detected by a **smoke pencil** held near suspected leakage paths such as the perimeter of doors and windows, electrical junction boxes or pipe penetrations, or seams and cracks in surfaces. The smoke pencil emits an inert smoke that responds to the slightest air movement. In many cases, the detected leakage paths can be repaired using sealants, tapes, or similar measures. This is especially true when the building is under construction. Ideally, a blower door test is performed when the thermal envelope is completed, but the wall, ceiling, and floor finishes have not yet been installed. This allows good access to detect and repair leakage paths. The blower door can then be used to assess the degree to which the leakage paths have been reduced or eliminated.

Some blower doors are equipped with instruments that precisely measure the differential pressure between outside and inside the building. The overall air tightness of the building is determined by increasing the fan speed until this differential pressure reaches 50 Pascals, which is approximately 0.2 inches of water column pressure. This creates a condition that approximates a 20-mph wind striking all surfaces of the building simultaneously. The air changes per hour leakage rate at 50 Pascals differential pressure can then be determined based on the fan speed and house volume. An average house without special air leakage detailing could have upward of 15 air changes per hour at a 50 Pascal differential pressure. An **EnergyStar-certified** house requires an air leakage rate less than or equal to four air changes per house at 50 Pascals. An ultra-tight house built to the German **PassivHaus standard** must have a tested air leakage rate not exceeding 0.6 air changes per hour at 50 Pascals differential pressure.

Estimating Infiltration Heat Loss

Air leakage rates can be converted into rates of heat loss using Equation 2.8:

Equation 2.8:

$$Q_i = 0.018(n)\ (v)(\Delta T)$$

where,

Q_i = estimated rate of heat loss due to air infiltration (Btu/h)

n = number of air changes per hour, estimated based on air sealing quality (1/h)

v = interior volume of the heated space (room or entire building) (ft^3) 0.018 = heat capacity of air ($Btu/ft^3/°F$)

ΔT = inside air temperature minus the outside air temperature (°F)

Example 2.7

Determine the rate of heat loss by air infiltration for a 30-foot by 56-foot single-story building with 8-foot ceiling height and average quality air sealing. The inside temperature is 70 °F. The outside temperature is 10 °F.

Solution:

The floor area of this building is 30 × 56 = 1,680 ft^2. Referring to Figure 2-18a for a 1,680-square-foot *single-story* building with average air sealing quality, the suggested air leakage rate is 0.38 air changes per hour. Substituting this value along with the other data into Equation 2.8 yields:

$$\begin{aligned} Q_i &= 0.018(n)\ (n)\ (\Delta T) \\ &= 0.018(0.38)\ (30 \times 56 \times 8)\ (70 - 10) \\ &= 5{,}516 \text{ Btu/h} \end{aligned}$$

Discussion:

As the conduction heat losses of a building are reduced, air infiltration becomes a larger percentage of total heat loss. When higher insulation levels are specified for a building, greater efforts at reducing air infiltration should also be undertaken. These include the use of high-quality windows and doors, close attention to caulking, sealed infiltration barriers, and so on.

2.6 Example of Complete Heating Load Estimate

Finding the total design heat loss of a room is simply a matter of calculating both the conduction heat loss and the infiltration heat loss and then adding them together. When this is done on a room-by-room basis, the resulting numbers can be used to size the heat emitters as the hydronic distribution system is designed. The sum of the room heat losses—the total building design heat loss—helps determine the required heat output of the heat source.

This section demonstrates a complete heat loss estimate for a small house having the floor plan shown in Figure 2-19. Simple tables show the results of using the equations present earlier in this chapter. Other assumed values for the calculations are as follows:

- Unit *R*-value for all windows: *R*-3.0
- Unit *R*-value for exterior door: *R*-5
- *R*-value of foundation edge insulation: *R*-10
- Rate of air infiltration in vestibule and utility room: 1.0 air change/h
- Rate of air infiltration in all other rooms: 0.5 air change/h
- Outdoor design temperature: –5 °F
- Desired indoor temperature: 70 °F
- Window height: 4 feet
- Exterior door height: 6 feet 8 inches
- Wall height: 8 feet
- Light/dry soil under slab

Determining the Total *R*-Value of Thermal Envelope Surfaces

Walls: The wall cross-section is shown in Figure 2-20. It is made up of 2 × 6 wood framing spaced 24 inches on center with *R*-21 fiberglass batt insulation in the stud cavities. The inside finish is 1/2-inch drywall. The outside of the wall is sheathed with 1/2-inch plywood, and covered with a Tyvek® infiltration barrier and vinyl siding.

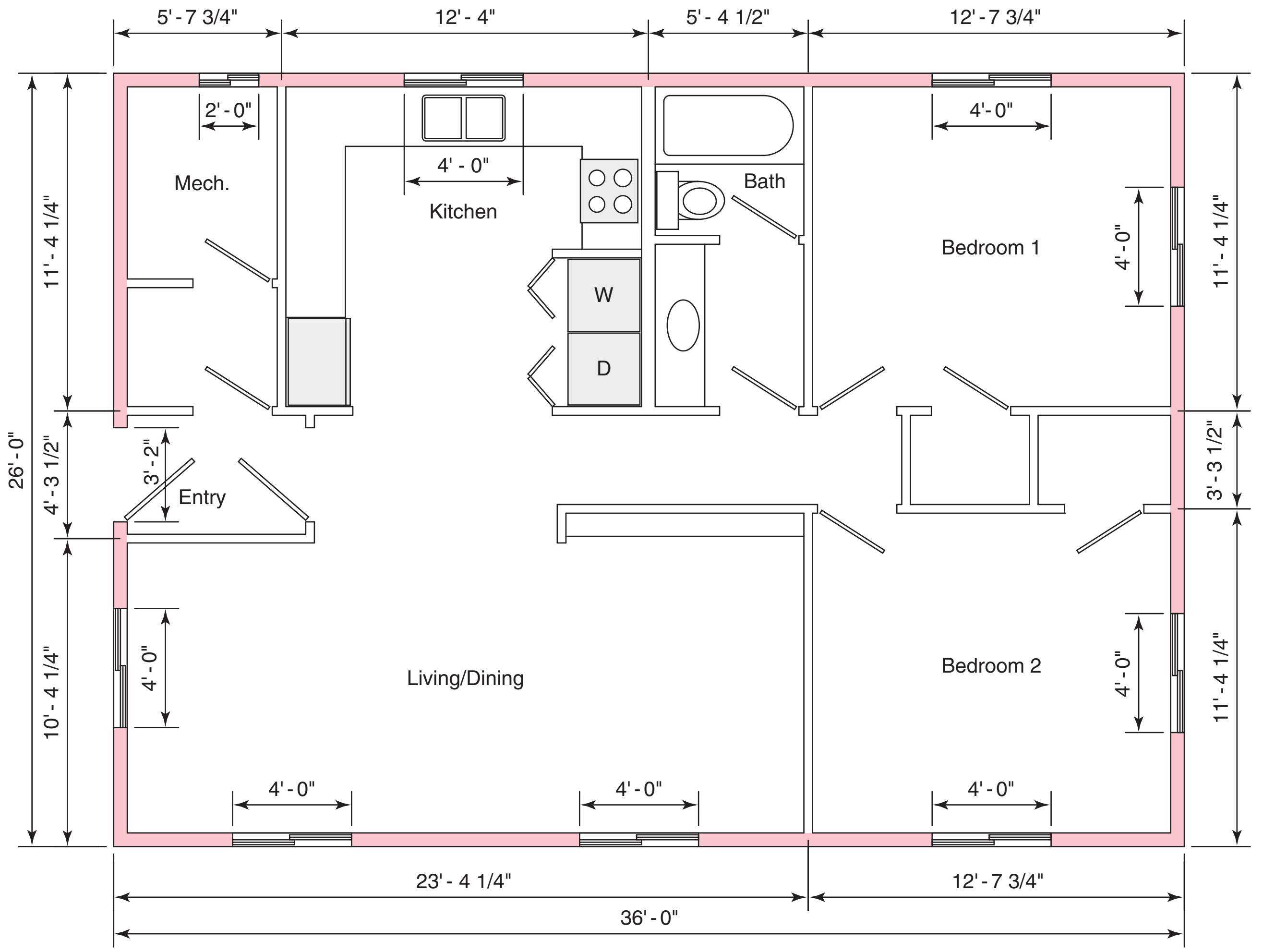


Figure 2-19 | Floor plan of example house for which heating load will be estimated.

The *R*-values of the materials are as follows:

Material	*R*-value between framing	*R*-value at framing
Inside air film	0.68	0.68
1/2-inch drywall	0.45	0.45
Stud cavity	21.0	5.5
1/2-inch plywood sheathing	0.62	0.62
TYVEK® infiltration barrier	~0	~0
Vinyl siding	0.61	0.61
Outside air film	0.17	0.17
Total	23.5	8.03

Assuming 15% of wall is solid framing, the effective total *R*-value of the wall is determined using Equation 2.4:

$$R_{\text{effective}} = \frac{(R_i)(R_f)}{p(R - R_f) + R_f} = \frac{(23.53)(8.03)}{0.15(23.53 - 8.03) + 8.03} = 18.3$$

Ceilings: The ceiling consists of 1/2-inch drywall covered with approximately 12 inches of blown fiberglass insulation. A cross-section is shown in Figure 2-21. The roof trusses displace a minor amount of this insulation.

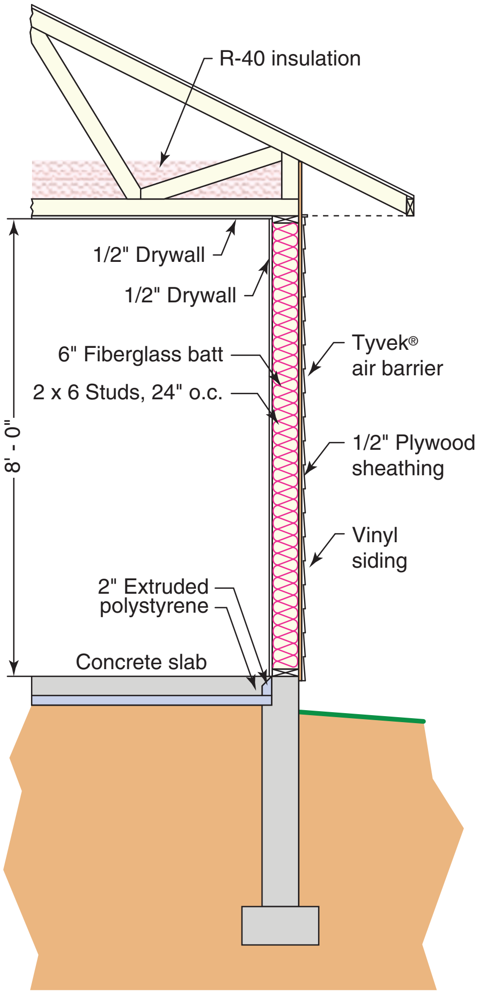


Figure 2-20 Wall cross-section for example house.

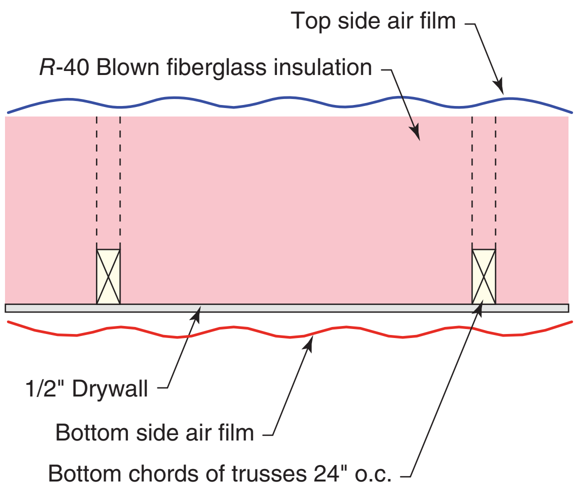


Figure 2-21 Cross-section of ceiling in example house.

The *R*-values of the materials are as follows:

Material	*R*-value between framing	*R*-value at framing
Bottom air film	0.61	0.61
1/2-inch drywall	0.45	0.45
Insulation	40	28.3
Framing	0	3.5
Top air film	0.61	0.61
Total	41.67	33.47

Assuming 10% of ceiling is solid framing, the effective total *R*-value of the ceiling is also determined using Equation 2.4:

$$R_{\text{effective}} = \frac{(R_i)(R_f)}{p(R_i - R_f) + R_f} = \frac{(41.67)(33.47)}{0.10(41.67 - 33.47) + 33.47} = 40.7$$

This effective *R*-value is very close to the *R*-value between framing. The wooden truss chords have minimal effect on the effective total *R*-value.

Room-by-Room Calculations

Using information from the floor plan and the total effective *R*-values of the various surfaces, the design heating load of each room can be calculated. This is done by breaking out each room from the building and determining all necessary information, such as areas of walls, and windows. This information will be entered into tables to keep it organized.

To expedite the process, many of the dimensions are shown on the floor plan of the individual rooms. If such were not the case, the dimensions would have to be estimated using an architectural scale along with the printed floor plan. If the floor plan were available as a CAD file, dimensions could also be determined using the dimensioning tool of the CAD system without having to print the drawing. As a matter of convenience, the room dimensions are taken to the centerline of the common walls.

When working with architectural dimensions, it is best to convert inches and fractions of inches into decimal feet since many dimensions will be added and multiplied together. Example 2.8 illustrates this.

Example 2.8

Convert the dimension 10 ft 4 1/4 in to decimal feet.

Solution:

$$10 \text{ ft } 4\tfrac{1}{4} \text{ in} = 10 \text{ ft} + \left(\frac{4.25}{12}\right)\text{ft} = 10 \text{ ft } + 0.35 \text{ ft} = 10.35 \text{ ft}$$

Discussion:

The conversion from feet and inches into decimal feet has been performed before the calculation of areas and lengths in Figures 2-22 through 2-26.

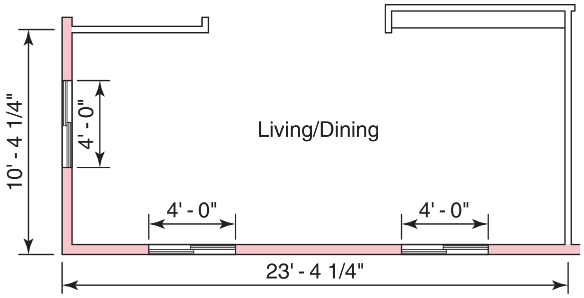
Total Building Heating Load

The total building heating load is obtained by summing the heating loads of each room.

Living/dining room	4,384 Btu/h
Bedroom 1	3,102 Btu/h
Bedroom 2	3,102 Btu/h
Kitchen	2,278 Btu/h
Bathroom	688 Btu/h
Mechanical room/entry	2,533 Btu/h
Total	**16,087 Btu/h**

A number of simplifying assumptions were used in this example.

- The rooms were divided at the centerline of the common partitions. This ensures that all the exterior wall area is assigned to individual rooms and not ignored as being between the rooms.
- Since the two bedrooms are essentially identical in size and construction, it was only necessary to find the heat load of one, then double it when determining the total building load.
- The exterior dimensions of the rooms were used to determine areas, volumes, and so forth. This makes the calculations somewhat conservative because the exterior wall area of a room is slightly greater than the interior wall area.
- The areas and volumes of the closets between the bedrooms were equally divided up between the two bedrooms. Since a separate heat emitter would not be supplied for each closet, this part of the load is simply assigned to the associated bedrooms.



(a)

Gross exterior wall area = (10.35 ft + 23.35 ft)(8 ft) = (33.708 ft)(8 ft) = 269.7 ft^2
Window area = 3 (4 ft × 4 ft) = 48 ft^2
Exterior door area = 0 ft^2
Net exterior wall area = Gross exterior wall area − Window and door area = 221.7 ft^2
Ceiling area = (10.35 ft)(23.35 ft) = 241.7 ft^2
Exposed slab edge length = (10.35 ft + 23.35 ft) = 33.7 linear ft
Room volume = (10.35 ft)(23.35 ft)(8.0 ft) = 1933.4 ft^3

		1	2	3	4	5	6	7	8	9	10
Room name		Exposed walls	Windows	Exposed doors	Exposed ceilings	Exposed floor	Exposed slab edge	Room volume	Air changes	Infiltration loss	Total room load
Living & dining	1	A = 221.7	A = 48	A = 0	A = 241.7	A = 0	L = 33.7	V = 1933	N = 0.5		
	2	R = 18.3	R = 3	R = n/a	R = 40.7	R = n/a	Re = 11				
	3	$\frac{A}{R}(\Delta T)$ = 909	$\frac{A}{R}(\Delta T)$ = 1200	$\frac{A}{R}(\Delta T)$ = 0	$\frac{A}{R}(\Delta T)$ = 445	$\frac{A}{R}(\Delta T)$ = 0	Q_{slab} = 525			Q_i = 1305	4384

(b)

Figure 2-22 (a) Floor plan for living/dining room of example house. (b) Table with values for living/dining room of example house.

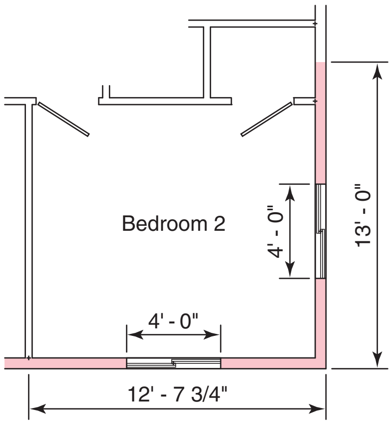
2.7 Computer-Aided Heating Load Calculations

After gathering the data and performing all the calculations needed for a building heating load estimate, it becomes apparent that a significant amount of information goes into finding the results. If the designer then decides to go back and change one or more of the numbers, more time is required to unravel the calculations back to the point of the change and calculate the new result. Even for people used to handling this amount of information, these calculations are tedious and always subject to errors. The time required to perform accurate heating load estimates is arguably the chief reason why some "heating professionals" do not do them.

Like many routine design procedures, heating load calculations are now commonly done using computers, and sometimes even on smartphones. The advantages of using load estimating software are many:

- Rapid (almost instant) calculations to quickly study the effect of changes
- Automated referencing of *R*-values, and weather data, in some programs
- Significantly less chance of error due to number handling
- Ability to print professional reports for customer presentations
- Ability to store project files for possible use in similar future projects
- Ability to import area and *R*-value information directly from CAD-based architectural drawings

There are many software packages currently available for estimating the heating loads for residential and light commercial buildings. They vary in cost from free to



(a)

Gross exterior wall area = (12.65 ft + 13.0 ft)(8 ft) = (25.65 ft)(8 ft) = 205.2 ft^2
Window area = 2 (4 ft × 4 ft) = 32 sq. ft
Exterior door area = 0 ft^2
Net exterior wall area = Gross exterior wall area − Window and door area = 173.2 ft^2
Ceiling area = (12.65 ft)(13.0 ft) = 164.5 sq. ft
Exposed slab edge length = (12.65 ft + 13.0 ft) = 25.65 linear ft
Room volume = (12.65 ft)(13.0 ft)(8.0 ft) = 1315.6 ft^3

		1	2	3	4	5	6	7	8	9	10
Room name		Exposed walls	Windows	Exposed doors	Exposed ceilings	Exposed floor	Exposed slab edge	Room volume	Air changes	Infiltration loss	Total room load
Bedroom 2	1	A = 173.2	A = 32	A = 0	A = 164.5	A = 0	L = 25.7	V = 1316	N = 0.5		
	2	R = 18.3	R = 3	R = n/a	R = 40.7	R = n/a	Re = 11				
	3	$\frac{A}{R}(\Delta T)$ = 710	$\frac{A}{R}(\Delta T)$ = 800	$\frac{A}{R}(\Delta T)$ = 0	$\frac{A}{R}(\Delta T)$ = 303	$\frac{A}{R}(\Delta T)$ = 0	Q_{slab} = 401			Q_i = 888	3102

(b)

Figure 2-23 (a) Floor plan for Bedroom 2 of example house. (b) Table with values for Bedroom 2 of example house.

over \$500. The higher-cost programs usually offer more features such as material reference files, automated areas determination from architectural drawings, and customized output reports. The lower-cost programs offer no frills output, but can nonetheless still yield accurate results.

Figure 2-27a shows the main screen of a program named ***Building Heat Load Estimator***, which is part of the more comprehensive software tool ***Hydronics Design Studio 2.0***, which was co-developed by the author. This software allows the user to build a list of rooms, each of which is defined based on its thermal envelope. The total heat loss of each room as well as the overall building is continually displayed as the user interacts with the software.

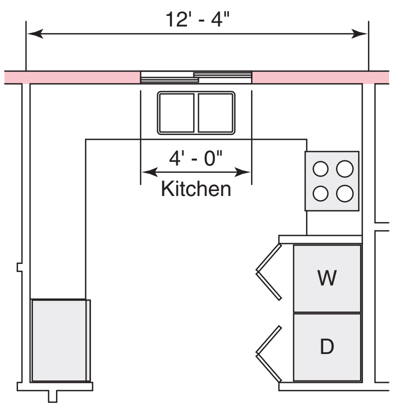
Building Heat Load Estimator allows the user to define the thermal envelope surfaces with each room, using pull-down menus for materials. It also provides a point-and-click selection of common wall, ceiling, floor, window and door assemblies. An example of how a framed wall is specified is shown in Figure 2-27b.

Building Heat Load Estimator also assists the user in determining surface areas and room volumes, as shown in Figure 2-27c.

A free demo version of this software can be downloaded at **www.hydronicpros.com**.

2.8 Estimating Annual Heating Energy Use

Although heating system designers need to know the design heat loss of each room in a building before sizing a heating system, such information is often meaningless to building owners. Their interest usually lies in what it will cost to heat their building. Such estimates can be made once the building's design heating load and the seasonal efficiency of the heat source are established.



(a)

Gross exterior wall area = (12.65 ft + 13.0 ft)(8 ft) = (25.65 ft)(8 ft) = 205.2 ft^2
Window area = 2 (4 ft × 4 ft) = 32 sq. ft
Exterior door area = 0 ft^2
Net exterior wall area = Gross exterior wall area − Window and door area = 173.2 ft^2
Ceiling area = (12.65 ft)(13.0 ft) = 164.5 sq. ft
Exposed slab edge length = (12.65 ft + 13.0 ft) = 25.65 linear ft
Room volume = (12.65 ft)(13.0 ft)(8.0 ft) = 1315.6 ft^3

		1	2	3	4	5	6	7	8	9	10
Room name		Exposed walls	Windows	Exposed doors	Exposed ceilings	Exposed floor	Exposed slab edge	Room volume	Air changes	Infiltration loss	Total room load
	1	A = 82.6	A = 16	A = 0	A = 135.6	A = 0	L = 12.3	V = 1085	N = 0.5		
Kitchen	2	R = 18.3	R = 3	R = n/a	R = 40.7	R = n/a	Re = 11				
	3	$\frac{A}{R}(\Delta T)$ = 339	$\frac{A}{R}(\Delta T)$ = 400	$\frac{A}{R}(\Delta T)$ = 0	$\frac{A}{R}(\Delta T)$ = 250	$\frac{A}{R}(\Delta T)$ = 0	Q_{slab} = 401			Q_i = 888	2278

(b)

Figure 2-24 (a) Floor plan for kitchen of example house. (b) Table with values for kitchen of example house.

Degree Day Method

The seasonal energy used for space heating depends upon the climate in which the building is located. A relatively simple method for factoring local weather conditions into estimates of heating energy use is the concept of heating degree days. *The number of heating degree days that accumulate in a 24-hour period is the difference between 65 °F and the average outdoor air temperature during that period* (see Equation 2.9). The average temperature is determined by averaging the high and low temperature for the 24-hour period.

Equation 2.9:

$$DD_{\text{daily}} = (65 - T_{\text{ave}})$$

where,

DD_{daily} = daily degree days accumulated in a 24-hour period

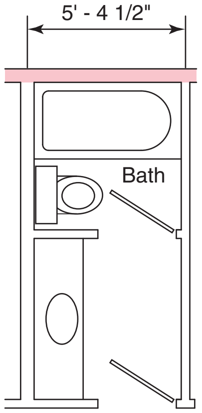
T_{ave} = average of the high and low outdoor temperature for that 24-hour period

The total heating degree days for a month, or an entire year, is found by adding up the *daily* heating degree days over the desired period. Many heating reference books contain tables of monthly and annual heating degree days for many major cities in the United States and Canada. A sample listing of annual degree days is given in Figure 2-28.

Heating degree day data has been used to estimate fuel usage for several decades. Such data is often recorded by fuel suppliers, utility companies, and local weather stations. Degree day statistics are also frequently listed in the weather section of newspapers as well as on Web-based weather reports.

When the heating degree day method was first developed, fuel was inexpensive and buildings were poorly insulated. Building heat loss was substantially greater than for comparably sized buildings using modern construction techniques.

Estimating seasonal energy consumption using heating degree days assumes that buildings need heat input from their heating systems whenever the outside



(a)

Gross exterior wall area = (5.38 ft)(8 ft) = 43.0 ft²
Window area = 0 ft²
Exterior door area = 0 ft²
Net exterior wall area = Gross exterior wall area − Window and door area = 43.0 ft²
Ceiling area = (5.38 ft)(11.0 ft) = 59.2 ft²
Exposed slab edge length = 5.38 linear ft
Room volume = (5.38 ft)(11.0 ft)(8.0 ft) = 473 ft³

		1	2	3	4	5	6	7	8	9	10
Room name		Exposed walls	Windows	Exposed doors	Exposed ceilings	Exposed floor	Exposed slab edge	Room volume	Air changes	Infiltration loss	Total room load
	1	A = 43	A = 0	A = 0	A = 59.2	A = 0	L = 5.38	V = 473	N = 0.5		
Bathroom	2	R = 18.3	R = 3	R = n/a	R = 40.7	R = n/a	Re = 11				
	3	$\frac{A}{R}(\Delta T)$ = 176	$\frac{A}{R}(\Delta T)$ = 0	$\frac{A}{R}(\Delta T)$ = 0	$\frac{A}{R}(\Delta T)$ = 109	$\frac{A}{R}(\Delta T)$ = 0	Q_{slab} = 84			Q_i = 319	688

(b)

Figure 2-25 (a) Floor plan for bathroom of example house. (b) Table with values for bathroom of example house.