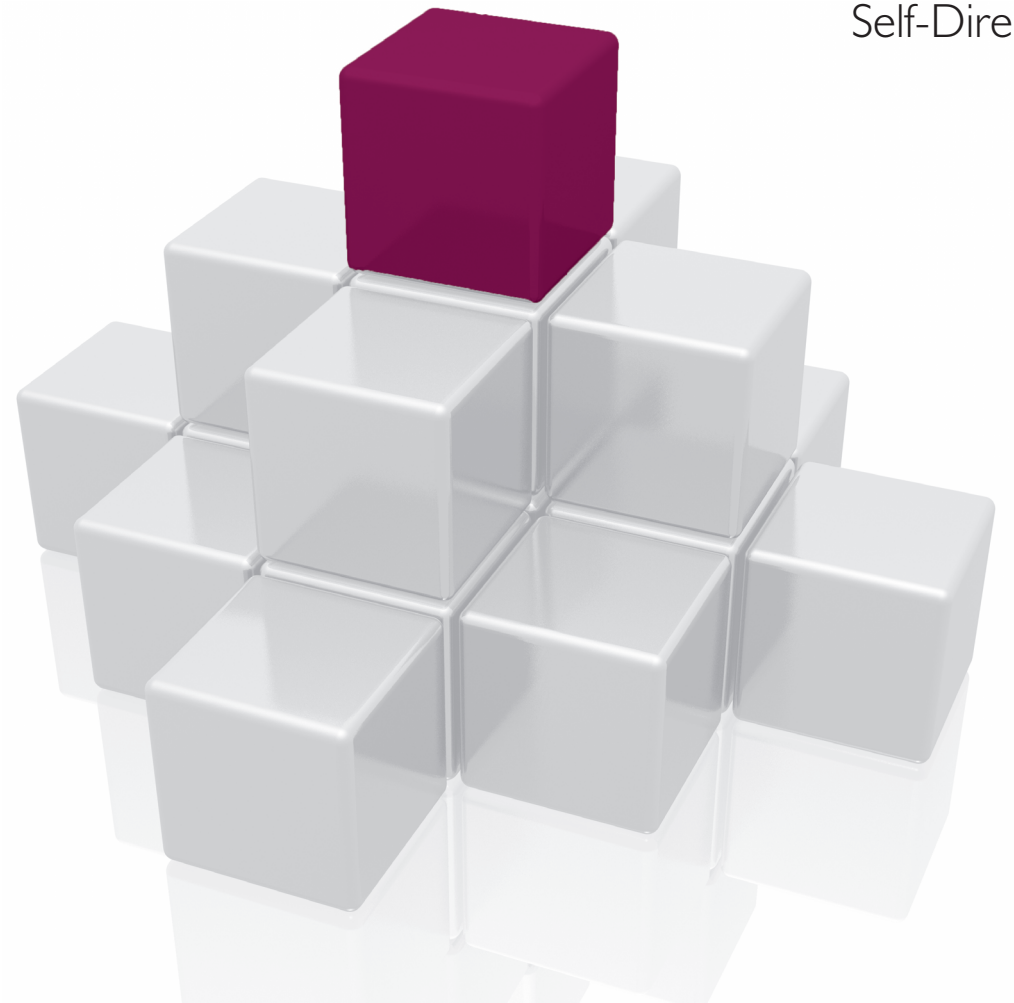
Fundamentals of Water System Design

Second Edition

Mark Hegberg
Richard A. Hegberg

I-P
Inch-Pound

A Course Book for
Self-Directed or Group Learning



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Atlanta

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Mark C. Hegberg: Mark Hegberg has spent major portions of his career involved in direct digital control system application for Tour & Andersson, consulting engineering for A. Epstein and Sons and others, and hydronic training and product development for ITT Bell & Gossett. In recent years Mark has been significantly involved in hydronic system testing and balancing and the issues associated with flow control in the field. Mark has chaired and been involved with several ASHRAE committees and is a recipient of the ASHRAE Exceptional Service Award. He too has been associated with the Illinois Chapter HVAC training class for many years.

Richard (Dick) A. Hegberg, PE: Dick Hegberg spent the majority of his career in the area of temperature control application for Powers Regulator company (now Siemens Building Technology), where he lead their early application engineering efforts and development of their building automation systems. In particular, he devoted much time to valves and dampers. Dick was actively involved in several ASHRAE TCs regarding controls and hydronics, as well as other groups that used both as part of their processes. Dick is an ASHRAE Life Member, Fellow, and recipient of the Distinguished Service Award. Locally, he inaugurated and directed for over thirty years the Illinois Chapter HVAC training class which is taught to this day.

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Preface

Fundamentals of Water System Design, Second Edition is an entry level text for the principles and fundamentals of water, or *hydronic*, HVAC system design and calculations. It uses the ASHRAE Handbook, and its related chapters on hydronics, as well as various research projects, ASHRAE documents, and articles on hydronics. The text represents the fundamentals of design and is an equally sound footing for work in commissioning, operation, and troubleshooting. The manual has been evolving over thirty years under the sponsorship of ASHRAE and is supported by ASHRAE Technical Committee (TC) 6.1, Hydronic and Steam System Design. For the majority of its history, this writing effort was lead by Richard (Dick) A. Hegberg.

Hydronic systems are a key foundation technology of HVAC system design. Hydronics provides an important tool in the designer's toolbox of providing effective and efficient HVAC systems to buildings around the world. One can argue exactly when hydronics started to be used for heating buildings; however, the modern application of hot-water systems may have started in the 1870s in the United States, and the technology has been refined ever since. Of the many benefits that water plays as a heat transfer medium for heating or cooling, fundamentally important is the ability to transport a lot of energy in the small area of a pipe (especially when compared to the equivalent size of an air duct). That benefit allowed for the development of central systems, eliminating the necessity of a local heating production system for each heated zone. The advancement allowed us to exercise individuality in our own building zone comfort control and lead to widespread application of hydronic systems in HVAC. From the early days of gravity circulation hot-water heating systems to the most complex of district energy chilled-water systems that are used today, the fundamentals presented here can be employed to successfully design the modern hydronic system.

Acknowledgments

ASHRAE TC 6.1 and its members have played a key role in collecting, researching, and communicating the details and experiences for the successful application of hydronic systems over the years. TC 6.1 is an ad-hoc collection of individuals interested in hydronic system technology who take the time to volunteer and attend meetings and then communicate what they have learned to the other 53,000 members of ASHRAE.

Key in the educational organizing efforts of TC 6.1 in hydronic systems and ASHRAE, a few individuals stand out when this text and the general concept of the ASHRAE's professional development series (be they Self Directed Learning manuals or the face-to-face version of the Professional Development Seminar) are considered. At the apex of this is William (Bill) J. Coad, former ASHRAE President and longtime member of several ASHRAE committees. When the professional development series was first developed, Bill Coad, J. Barry Graham, and Gerry Williams developed the first two classes on Air System Design and Energy Efficient HVAC. Bill's significant involvement in leading many ASHRAE activities left little time to develop the Water Systems manual, which lead to Dick Hegberg taking up the charge. All of these volunteer member contributors are owed a deep debt of gratitude for their devotion to helping ASHRAE develop the initial active training efforts that have evolved into the many practical training programs that are available today.

ASHRAE has, from its very inception as ASHVE, represented a unique cooperative collaboration between all aspects of the HVAC industry. Designers, owners, operators, academicians, and manufacturers have all met and debated the merits of all things HVAC&R for over 100 years, seeking the answers to the problems associated with the HVAC system challenges of the day and providing the fundamental knowledge required by all professionals in a noncommercial manner. In that regard, while there are many professionals from the manufacturing or "product" side of the business that have contributed to advancing the area of hydronics, we would be remiss to not mention the significant contributions of Gil Carlson of Bell & Gossett. As hydronics shifted from steam heating system application to hot- and chilled-water systems in the 1950s and 1960s and then rolled into the early days of the first energy crisis of the 1970s, Gil's application engineering, research expertise, and ability to communicate was an invaluable resource to the industry and ASHRAE TC 6.1. His significant ASHRAE research and journal articles on hydronic application, as well as the simple design tools he developed, were the educational precursor to what later became the professional development series. The industry is indebted to Gil's contributions and to Bell & Gossett, now a division of Xylem Corporation, for their support to this day of the fundamentals of application engineering education in the area of hydronics.

On a personal note, thanks to all those within ASHRAE who, for no other reason than their commitment to the spirit of mentoring that ASHRAE represents, helped provide me the practical education that comes from beyond col-

lege corridors. I thank you all and in particular Dick Hegberg, Bill Coad, and Roy Ahlgren. I have had the great fortune to come to know many of the ASHRAE members that contributed to this manual over the years, successfully designing and operating systems even before the rise of computers and spreadsheets. Amazingly, we can still employ those fundamentals to the same end today, keeping in mind the guidance of Carlson, “A difference to be a difference must be a big enough difference to make a difference,” and Coad paraphrasing Einstein, “The secret is to keep it simple… but not too simple.”

Mark Hegberg

Water System Design Concepts

Study Objectives

After studying the material in this chapter, you should be able to do the following:

- ❑ know what determines the load;
- ❑ understand the difference between closed and open systems;
- ❑ know the components of a hydronic system;
- ❑ understand heating versus cooling source devices;
- ❑ understand how systems meet part-load conditions;
- ❑ identify temperature and pressure ranges for low-, medium-, and high-temperature water systems;
- ❑ know what sensible, latent, and total heat loads are and how they affect design water flow;
- ❑ identify examples of heating and cooling load devices; and
- ❑ know how load diversity suggests a reduction in the total cooling capacity required.

Instructions

Read Chapter 1, and answer all of the questions at the end.

Introductory Concepts

Water system design depends on the designer's ability to evaluate the space loads, occupancy patterns, and indoor environment requirements. This chapter examines the actual process of water system design and provides information on how to evaluate space loads. It also provides strategies and formulas for mastering the key requirements for water systems.

Water systems that convey heat to or from a conditioned space or process with hot or chilled water are frequently called *hydronic systems*. In general, these systems employ centrifugal pumps to force water flow from a heating or a cooling source to the conditioned space or load by means of various piping, pumping, control, and terminal arrangements (ASHRAE 2012).

Given the design requirements, it is the designer's task to evaluate the space loads resulting from building construction, weather distribution, occupancy patterns, indoor environment requirements, and other internal loads to determine the total load (Figure 1-1) subject to the local building codes. The loads include transmission, solar radiation, infiltration, ventilation air, people, lights, power, appliances, and materials in and out (Sauer and Howell 2013).

The historical weather distribution for the project location is important, and a means to control the systems at part-load conditions to maintain comfort conditions for the occupants must be studied for proper design. The designer must weigh the cost of the source utilities available and also the efficiencies of boilers and chillers or other hydronic sources to determine the most efficient system design (Figure 1-2). The system must be able to operate between part-load and full-load conditions. In many cases, the hydronic system is a support system providing the heating or cooling medium for heat transfer equipment in an air distribution system. This course is intended to acquaint the student with the various hydronic principles and practices available for consideration in a project's design concept.

Figure 1-3 shows the basic components of a hydronic system that the designer must define for an HVAC system, namely, a source of heating or cooling, a distribution system, and the load components. There are different classifications of hydronic systems; the most common types are summarized below. As you familiarize yourself with the various systems, remember that different factors may come into play for each.

Water systems may be closed or open types. The fundamental difference between them is the interface of the water with a compressible gas (such as air) or an elastic surface (such as a diaphragm). A closed water system is

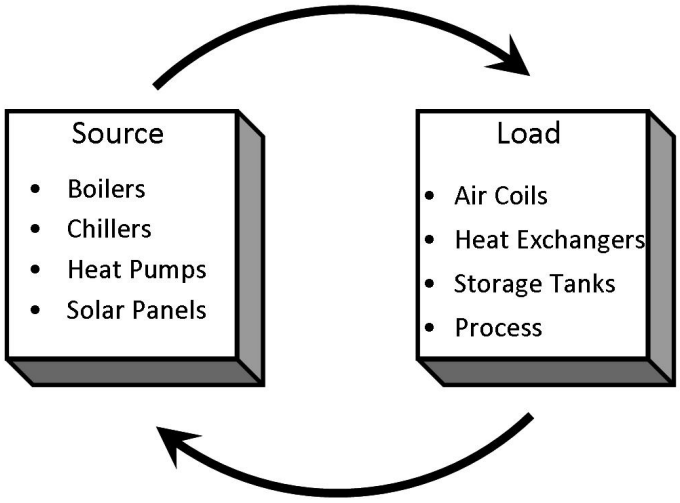


Figure 1-1 Source/load.

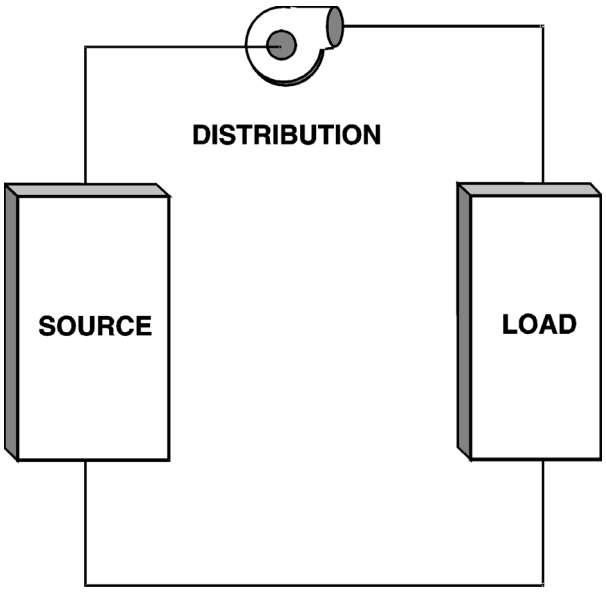


Figure 1-2 Source–distribution–load.

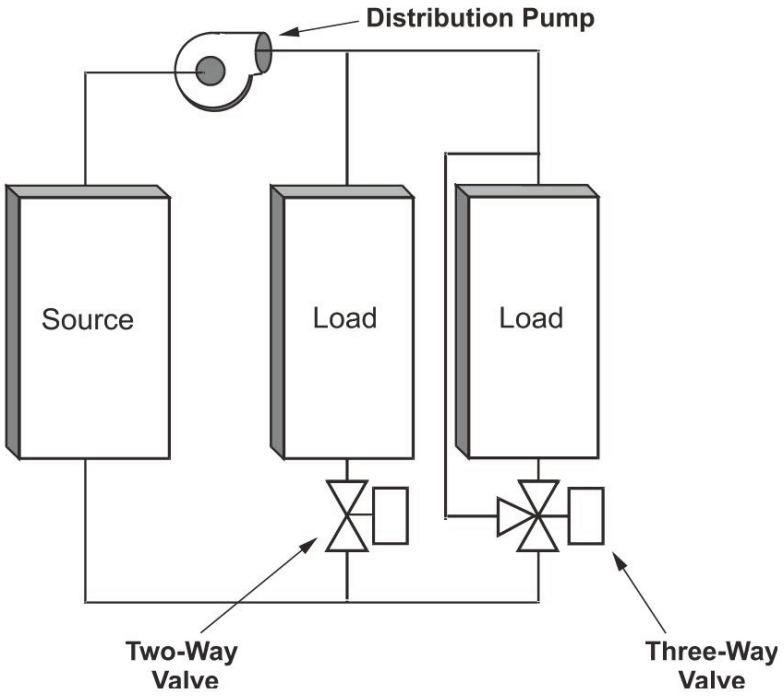


Figure 1-3 Source–distribution–part-load.

defined as one with no more than one point of interface with a compressible gas (air) or surface (Figure 1-4). This definition is fundamental to understanding the hydraulic dynamics of these systems (to be discussed in the section Expansion Chamber).

An open system has more than one such interface. For example, a cooling tower has at least two points of interface: the tower basin and the discharge pipe or nozzles entering the tower.

Comparing Figure 1-4 and Figure 1-5, the differences between the hydraulics of the systems become evident as one analyzes the two systems. However, one major difference is that certain hydraulic characteristics of open systems do not occur in closed systems. For example, in a closed system,

- flow cannot be motivated by static head difference,
- pumps do not provide static lift, and
- the entire piping system is always filled with water.

These factors affect the installation and operating costs of the system over its service life.

Closed water systems are classified by operating temperature:

- **Low-temperature water (LTW) system.** This hydronic heating system operates within the pressure and temperature limits of the ASME *Boiler and Pressure Vessel Code* for low-pressure boilers (Figure 1-6). The maximum allowable working pressure for low-pressure boilers is 160 psig, with a maximum temperature of 250°F. The usual maximum working pressure for LTW boiler systems is 30 psig; however, boilers specifically designed, tested, and stamped for higher pressures are frequently used. Steam-to-water and water-to-water heat exchangers are also used for heating LTW.

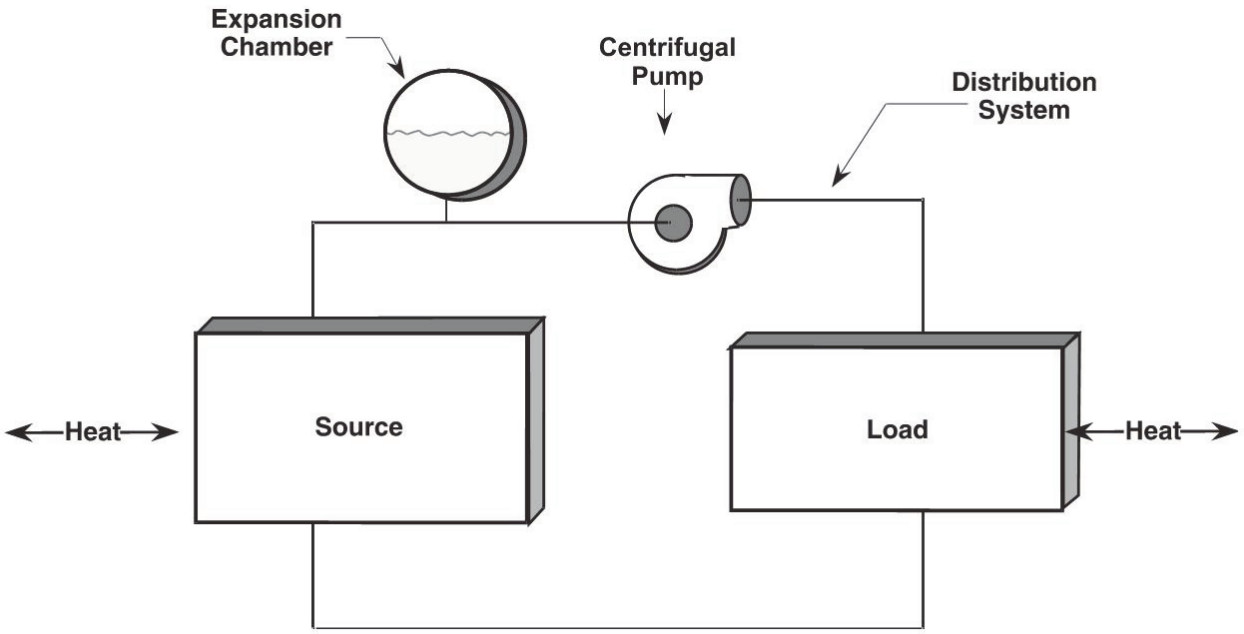


Figure 1-4 Hydronic system fundamentals (closed system).

- **Medium-temperature water (MTW) system.** This hydronic heating system operates at temperatures between 250°F and 350°F, with pressures not exceeding 160 psig (Figure 1-7). The design supply water temperature is approximately 250°F to 325°F, with a pressure rating of 150 psig for boilers and equipment.
- **High-temperature water (HTW) system.** This hydronic heating system operates at temperatures over 350°F, with pressures not exceeding 300 psig. The maximum design supply water temperature is about 400°F, with a pressure rating for boilers and equipment of 300 psig (Figure 1-7). The pressure/temperature rating of each component should be checked for compliance with the system's design versus the manufacturer's rating.

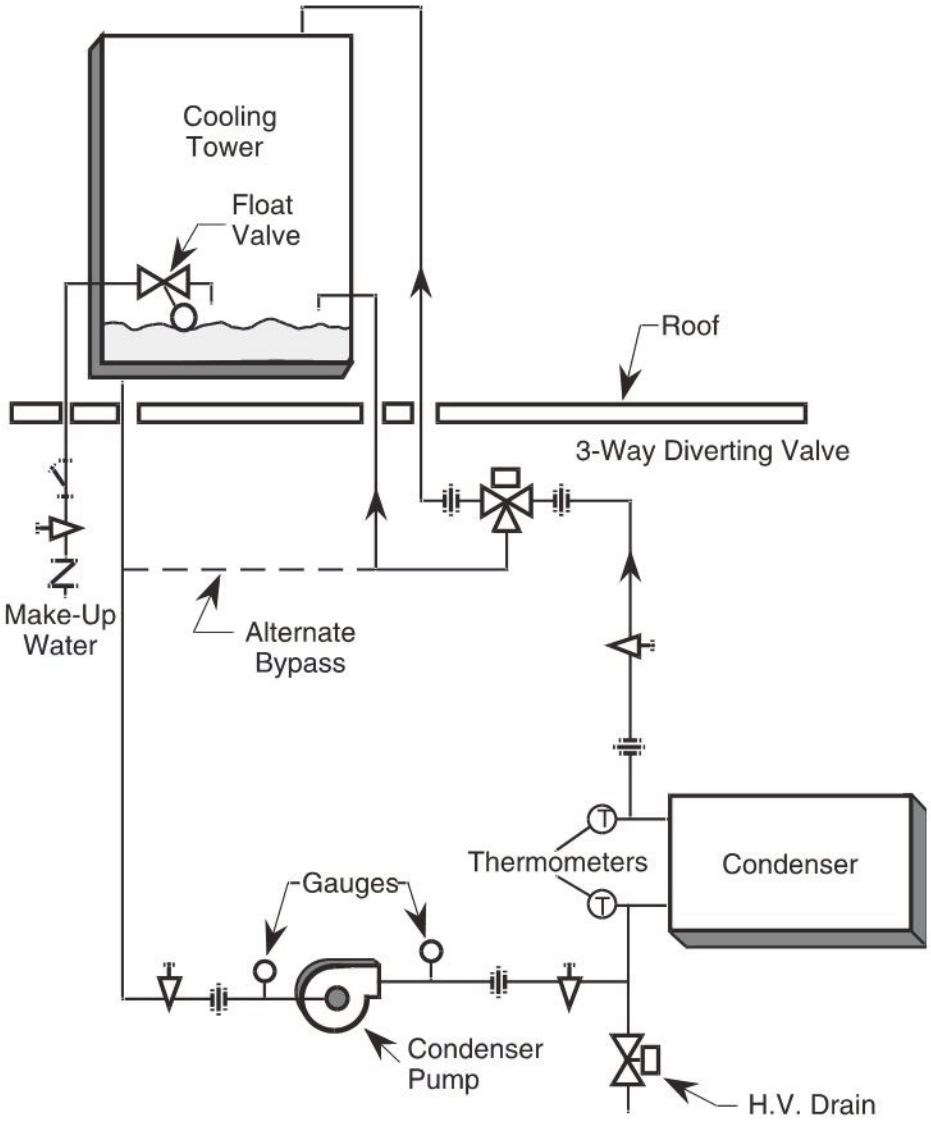


Figure 1-5 Cooling tower (open system).

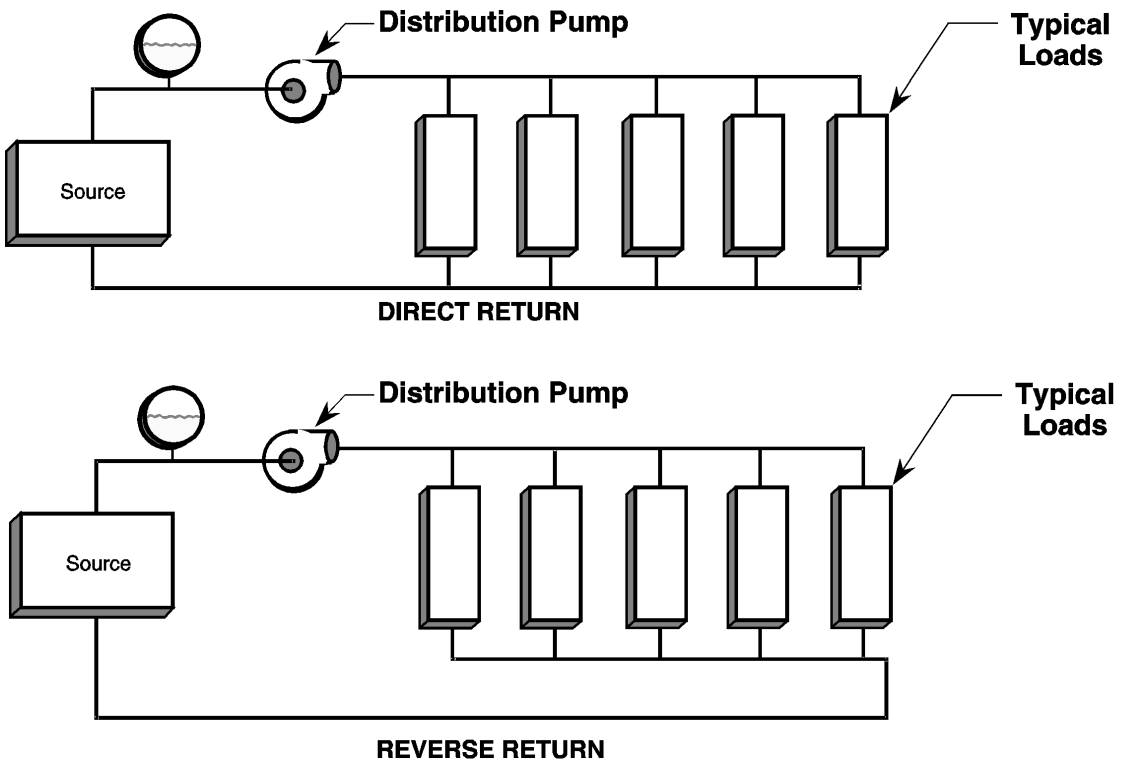


Figure 1-6 Low-temperature water system—direct or reverse return.

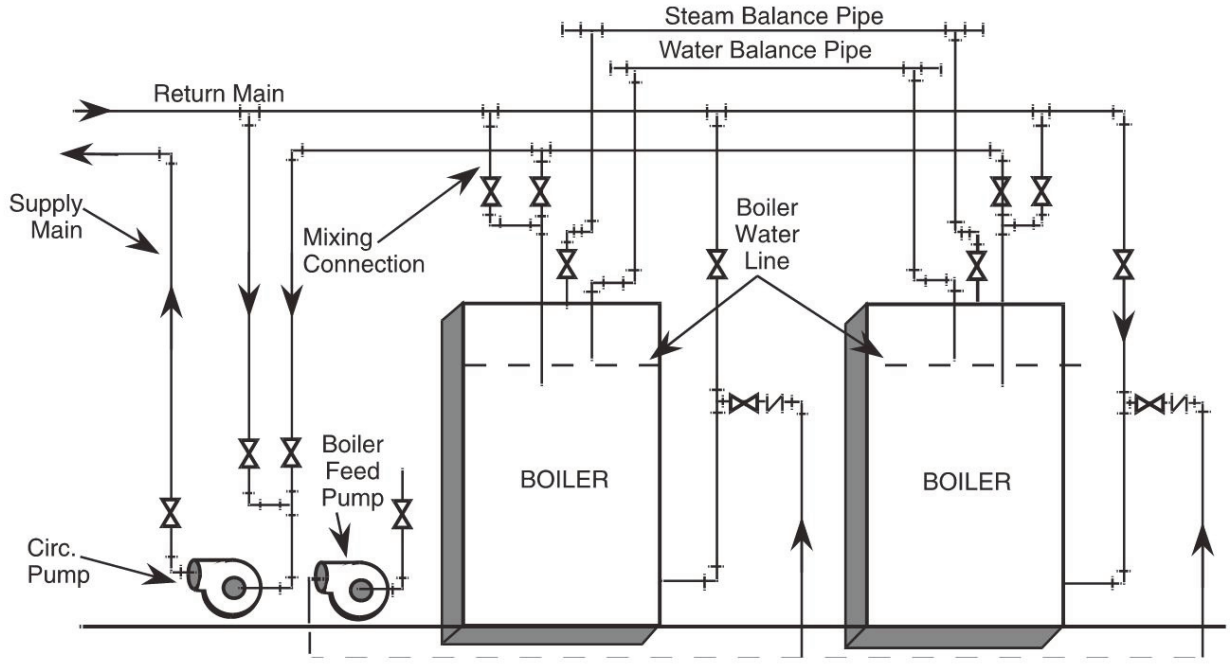


Figure 1-7 Medium- or high-temperature water system.

- **Chilled-water (CHW) system.** A hydronic cooling system normally operates with a design supply water temperature of 40°F to 55°F (usually 44°F or 45°F) with a pressure rating of 120 psig. Figure 1-8 shows a small- to medium-sized system with constant-speed pumping using three-way valves to ensure constant flow in the chiller source and balancing valves on each load for flow measurement and adjustment. Larger systems may employ two-way control valves and different chiller piping and pumping arrangements to reduce pumping power.
- **Condenser water (CW) system.** Open water systems are typically used in refrigeration CW systems as once-through or cooling tower systems. Figure 1-9 shows a water-cooled condenser using city, well, or river water. The return is run higher than the condenser so that the condenser is always full of water. Water flow through the condenser is modulated by a control valve in the supply line. This is usually actuated by a condenser head pressure to maintain a constant condensing temperature with load variations.

Figure 1-10 shows two cooling tower applications to protect against low outdoor temperature conditions. Water flows to the pump from the tower basin, and the level should be above the top of the pump casing for positive prime, and piping pressure drop should be minimized.

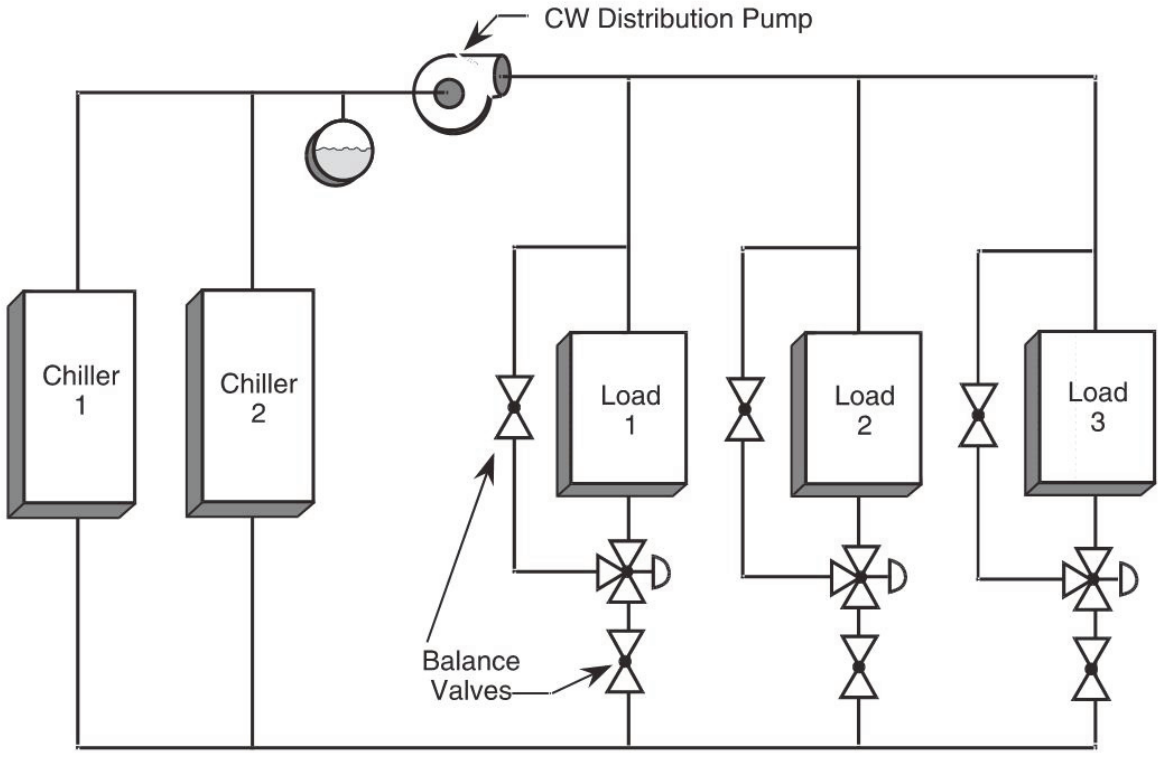


Figure 1-8 Chilled-water system—direct return piping.

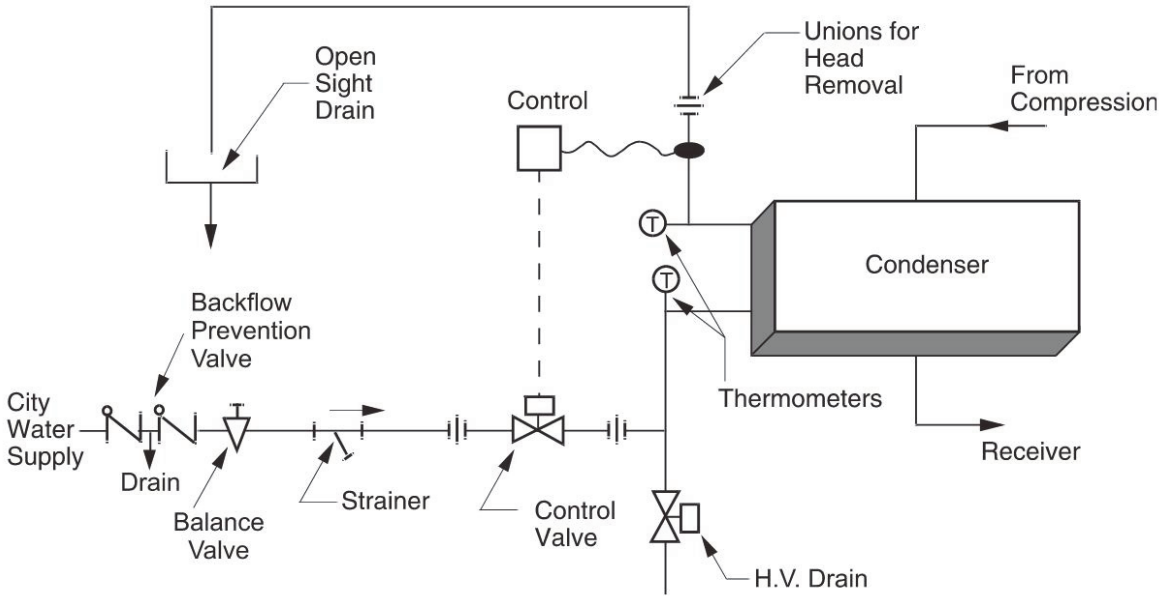


Figure 1-9 Condenser open water system (once through).

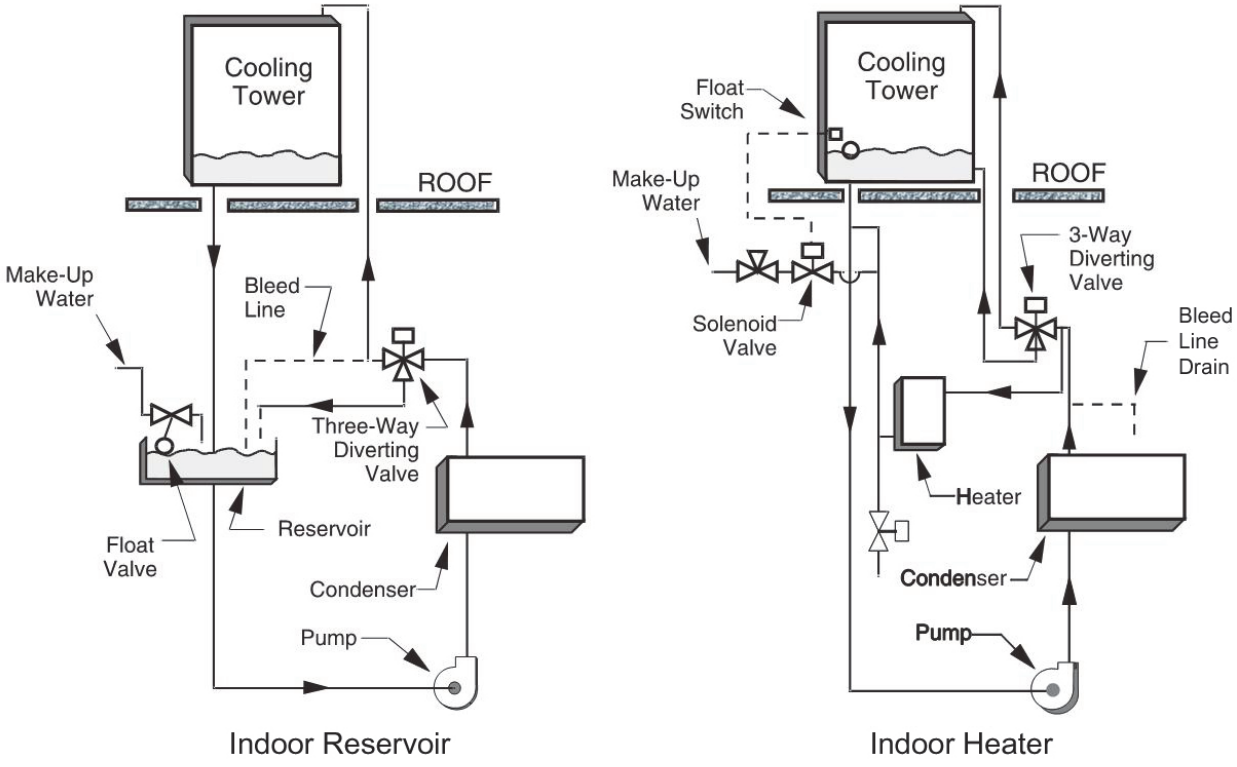


Figure 1-10 Condenser cooling tower system inside reservoir or heated sump.

Antifreeze or brine solutions may be used for applications (process applications) that require temperatures below 40°F or for coil freeze protection. Well water systems can use supply temperatures of 60°F or higher.

In addition, there are other forms of system classification that tend to blur the lines between system operating temperatures, pipe layout styles, and/or a reference to the operated system style.

Four-pipe dual temperature water systems are one such example of this blur. This older example of system design combines the heating and cooling system, circulating hot and/or chilled water through separate supply and return pipes to common terminal coils. Figure 1-11 shows a very simplified piping diagram for this type of system, which would not normally be piped this way. This system operates within the pressure and temperature limits of LTW systems, with usual winter design supply water temperatures of about 100°F to 150°F and summer supply water temperatures of 40°F to 45°F. A reading of ANSI/ASHRAE/IES Standard 90.1 (ASHRAE 2013) can suggest that this type of system (especially as shown) is not allowed due to energy performance issues associated with potential mixing of hot or cold conditioned water. Of course, given enough thought and control logic, this issue may be overcome.

A more common application is to have a two-pipe dual-temperature system where a common piping system serves common loads with both hot and chilled water supplied seasonally to the system. In either case, some of the issues a designer must deal with start to show themselves so that the system operates properly. Think through the components and issues like the designer!

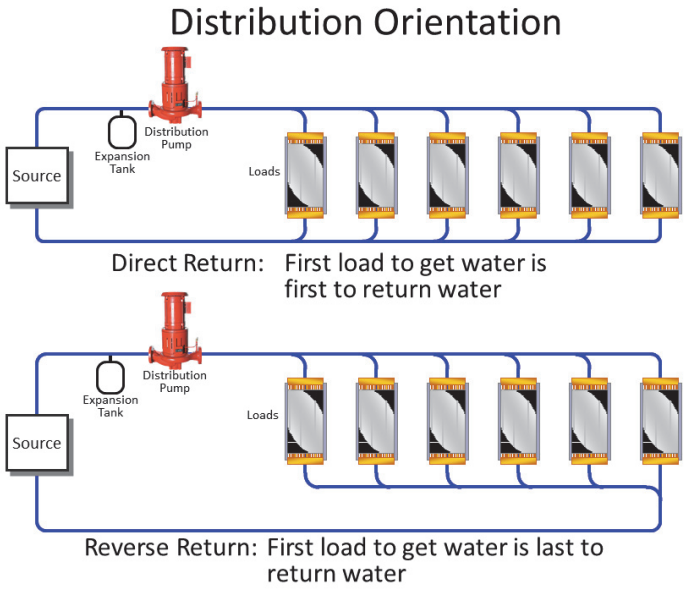


Figure 1-11 Distribution orientation.

The design should consider source protection to prevent temperature shocks to the chiller or boiler on cycle changeover. The designer might consider timing mechanisms, temperature mechanisms, or combinations of both so that the two sources never operate together and that neither source gets water that is too hot or too cold.

Think about heat transfer. In general, a hot-water heating coil with its higher operating-water temperatures and differential temperatures as compared to CHW requires a much smaller heat transfer coil. The water, on the other hand, does not know or care about this, it just acts as it does because there is heat transfer surface area. Therefore, it performs to whatever capabilities are inherently a part of that seasonal operation. The designer then has to carefully consider the design temperature differentials and flow rates associated with the coil selection and whether the sources are capable of handling the developed differentials.

Consider how the coil will be controlled. For example, will flow be proportionally modulated or be positioned for either full or no flow?

Consider whether seasonal changes in operating flow will affect the water flow velocity, possibly impacting air control and pressure management in the system.

There are other issues to be considered, too. This manual teaches a framework of issues to be dealt with in design that can be adapted to deal with any type of system.

Basic System Components

Figure 1-12 shows the fundamental components of a closed hydronic system. Actual systems generally have additional components (such as valves, vents, etc.), but these are not essential to the basic principles underlying the concept of the system. These fundamental components are as follows:

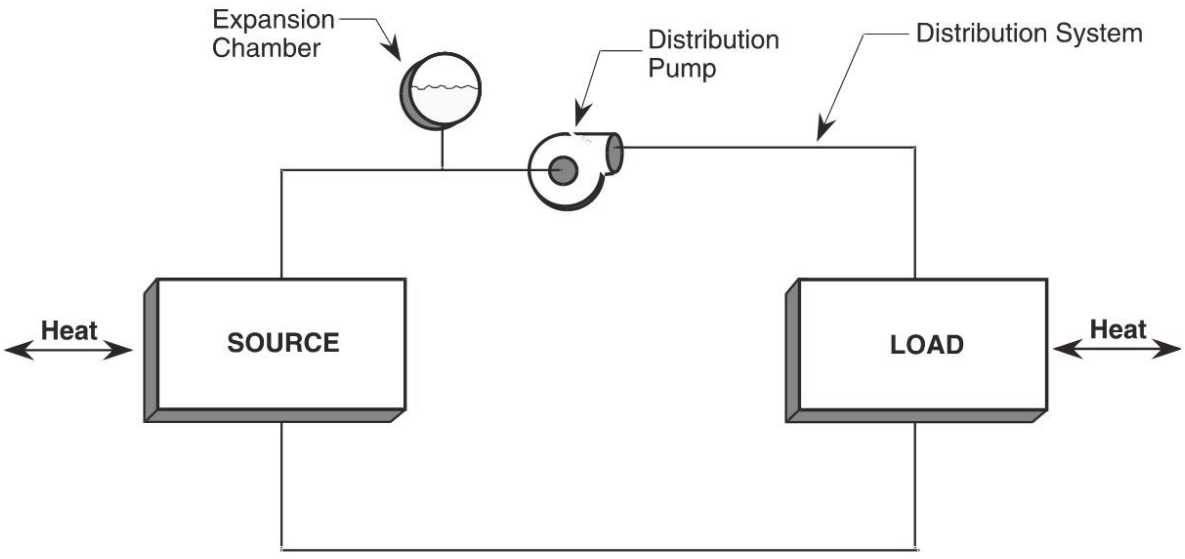


Figure 1-12 Closed hydronic system fundamental components.

- Source system
- Load system
- Pump system
- Distribution system
- Expansion chamber

Source

The source is the point where heat is added in a heating system or removed from a cooling system. Ideally, the amount of energy entering or leaving the source equals the amount entering or leaving through the load system. Under steady-state conditions, the load energy and source energy are equal and opposite. In reality, energy conversion and/or transfer is not perfect and the source has an efficiency of less than 100%.

Each type of source has its own efficiency characteristics as a function of load. You must consider this source efficiency in the system design process. Any device that can be used to heat or cool water under controlled conditions can be used as a source device. Sources typically function in one of two ways:

- By converting chemical, electrical, or solar energy to heat, which is then transferred to water in the system
- By transferring heat from one system to another

The most common source devices for heating and cooling systems are the following:

- **Heating-source devices:** Hot-water generator or boiler, steam-to-water heat exchanger (Figure 1-13), water-to-water heat exchanger (Figure 1-13), solar collector or panels, heat recovery or salvage heat device, exhaust gas heat exchanger, incinerator heat exchanger, heat pump condenser, and air-to-water heat exchanger
- **Cooling-source devices:** Electric compression chiller (Figure 1-14), thermal absorption chiller (Figure 1-14), heat pump evaporator, air-to-water heat exchanger, and water-to-water heat exchanger

A typical large CHW system with multiple chillers, various load controls, and compound pumping is shown in Figure 1-14. This system provides variable flow, constant supply temperature CHW, multiple chillers, two-way valve control, and the advantage of adding CHW storage. One design issue shown is the placement of the common pipe for the chillers. With the common pipe located at the opposite end of the chiller production section, the chillers will unload from right to left. With the common pipe in the alternative location (between the CHW production and the loads), the chillers will load and unload equally in proportion to their capacity (see Chapter 8 for further discussion).

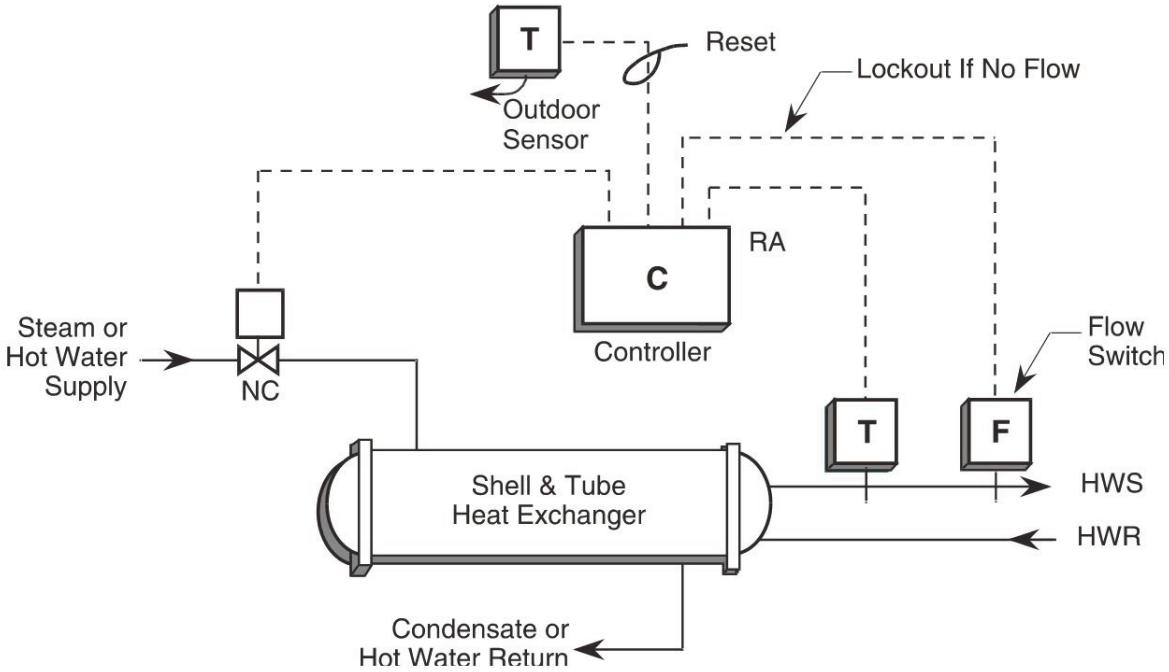


Figure 1-13 Steam-to-water or water-to-water heat exchanger.

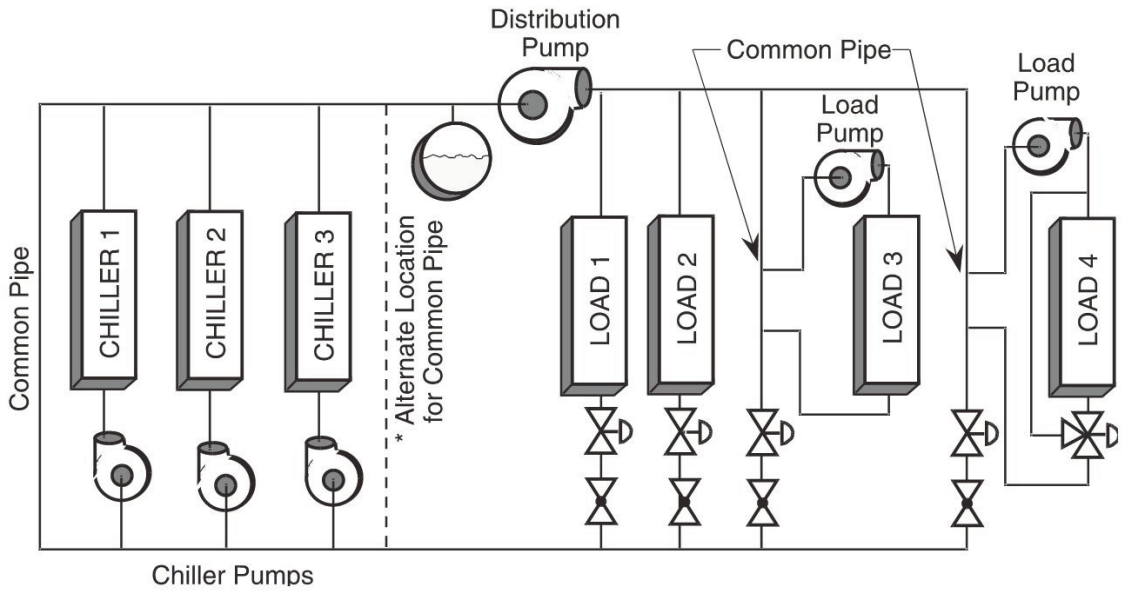


Figure 1-14 Multiple chiller variable-flow chilled-water system.

The two primary considerations in selecting a source device are the design capacity and the part-load capability, which, when combined, define the turndown ratio. The turndown ratio, expressed in percent of design capacity, is

$$\text{Turndown Ratio (\%)} = 100 \times \frac{\text{Minimum Capacity}}{\text{Design Capacity}} \qquad \textbf{(1-1)}$$

The reciprocal of the turndown ratio is sometimes used. For example, a turndown ratio of 25% may also be expressed as a turndown ratio of four.

The turndown ratio has a significant effect on the successful performance of a system, and lack of consideration for this capability of the source system has been responsible for many systems that do not function properly or that excessively consume energy. The turndown ratio has a significant impact on the ultimate system design selection because operating efficiencies tend to decrease as the turndown ratio decreases.

Generally the larger the boiler or chiller on a single unit basis, the more difficult it is to achieve acceptable operation efficiencies and therefore acceptable costs for the low-load portion of its seasonal operation. Seasonal operation at less than 50% load can, in the case of heating, be over 50% of its total duty hours; in northern climates, a similar ratio may occur during cooling seasons. This presents the designer with a dilemma. You must specify a system that meets the maximum requirements, but in doing that the system may operate ineffectively for most of its duty season. One approach to solving this dilemma is by using multiple sources of lower capacity (Figure 1-14).

Another design consideration is the diversity of the cooling load. Carrier (1965) states

> diversity of cooling load results from the probable non-occurrence of part of the cooling load on a design day. Diversity factors are applied to the refrigeration capacity in large air-conditioning systems. These factors vary with location, type, and size of the application, and are based entirely on the judgment of the engineer.

The diversity factor, as a ratio of actual load/design load, can be applied to people and lighting loads in large multistory office, hotel, or apartment buildings. For example, in an office building the diversity factor for the refrigeration capacity due to people may be from 0.75 to 0.90 and due to the lighting from 0.70 to 0.85. In addition, the design engineer must consider the storage load factors for heat gain due to lighting and solar gains by glass.

Specific methods of load control for CHW plants are discussed in Chapter 9.

Load

The load is the point where heat flows out of or into the system from the space or process; it is the independent variable to which the remainder of the system must respond. Outward heat flow characterizes a heating system, and inward heat flow characterizes a cooling system. The quantity of heating or cooling is calculated by one of the means discussed in the following section. Typically, loads are a form of heat exchanger, either air to water or water to water.

Pump System

If we consider the modern era of hydronic system application to be the last 100 or so years, historically the first hydronic systems were heating systems, and they did not apply a pump to the system to circulate water. These systems were gravity fed, relying on the difference in head caused in the system by water being heated in a boiler and cooled in a radiator and pipes. These were exceptionally simple systems, circulating water with differential pressures (less than a few inches of pressure). They tended to be small in size and used a large pipe (2 to 3 in.) to circulate flows less than 1 gpm. These systems were slow to react and operation-sensitive to how well the pipe was sized. There was no real control short of turning the flow on or off. The first recorded use of a pump in a heating system appeared in 1926 in the *Transactions* of ASHVE, forerunner to ASHRAE (Giesecke 1926). Giesecke, a researcher and professor at Texas A&M, delivered a paper on testing pipe fittings, and an engineer from Toronto, H.H. Angus, commented that they had solved flow problems in gravity heating systems by applying a small pump to boost the water flow. A new era of hydronic system design dawned and a term that is still used today, *booster pump*, came into use. Pumps create flow by converting captured centrifugal force into a higher pressure of water. In essence, the pump is a differential pressure machine with the flow of water being created as the pressure moves to a point of lower pressure.

Distribution System

The distribution system in a hydronic system is a series of pipes, transporting water to the location of the loads where it is required.

Expansion Chamber

Recognizing that water is transported in rigid pipes and that water expands and contracts as it is heated and cooled, water must have a place to expand into to prevent damage to the distribution system. The expansion chamber is this place, and there are two types of chambers that are commonly applied. By using this device, the pressure of the water in the system can be controlled and managed so that useful work is done with the system without damage.

Design Trade-Offs

Remember, improved efficiency comes with an initial installation cost penalty. As the designer, you must work out the acceptable trade-off in the initial installation cost that a customer will accept for reduced operational expenses over the life of the system. This will be easier to accomplish when backup system requirements are considered in the decision.

System Temperatures

As the designer, you must design temperatures and temperature ranges by considering the performance and economics of the components. For example, for a cooling system that must maintain 50% rh at 75°F (Figure 1-15), the dew-point temperature is 55°F. This sets the maximum return water temperature near 55°F (60°F maximum); the lowest practical temperature for refrigeration, considering the freezing point and the economics, is about 40°F. This temperature spread then sets constraints for a CHW system.

For a heating system, the maximum hot-water temperature is normally established by the ASME *Boiler and Pressure Vessel Code* (2015) as 250°F, and, with space temperature requirements of little over 75°F, the actual operating supply temperatures and the temperature ranges are set by the design of the load devices. Most economic considerations relating to the distribution and pumping systems favor the use of the maximum possible temperature range Δt.

Heat Transfer in Hydronic Systems

Sensible Heating or Cooling of Air

The quantity of heat entering or leaving the airstream is expressed by

$$q = 60 \times \rho_a \times c_p \times Q_a \times \Delta t \qquad \textbf{(1-2)}$$

where

c_p = specific heat of air, Btu/lb·°F

q = heat transfer rate, Btu/h

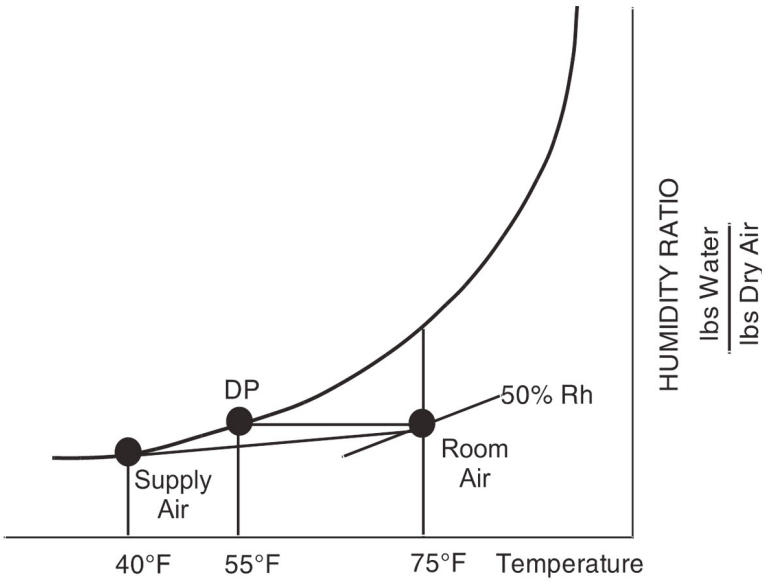


Figure 1-15 Psychrometric chart chilled-water example.

Q_a = airflow rate, cfm

Δt = air temperature change, °F

ρ_a = density of air, lb/ft^3

For standard air with a density of 0.075 lb/ft^3 and a specific heat of 0.24 Btu/lb·°F, this equation becomes

$$q \text{ Btu/h} = \frac{60 \text{ min}}{1 \text{ h}} \left| \frac{0.24 \text{ Btu}}{\text{lb·°F}} \right| \frac{0.075 \text{ lb}}{\text{ft}^3} \left| \frac{Q_a \text{ ft}^3}{\text{min}} \right| \Delta t\text{°F}$$

$$q \text{ Btu/h} = 1.08 \times Q_a(\text{cfm}) \times \Delta t \quad \textbf{(1-3)}$$

Example 1-1

In the system shown in Figure 1-16, the air upstream of the heating coil is 60°F and the air temperature leaving the heating coil is 130°F. Given that c_p = 0.24 Btu/lb·°F, Q_a = 5000 cfm, and the density is 0.075 lb/ft^3, calculate the heat transfer rate of the system.

$q = 1.08\ Q_a\ \Delta t$

$= 1.08$ Btu/h·cfm·°F

$q = (1.08)(5000 \text{ cfm})(130°\text{F} - 60°\text{F})$

$q = 378{,}000$ Btu/h

Solution

The heat exchanger or coil must then transfer this heat to the water. The quantity of sensible heat transferred to the heated or cooled medium in a specific heat exchanger is a function of the surface area, the mean temperature difference between the water and the medium, and the overall heat transfer coefficient, which is a function of the fluid velocities, properties of the medium, geometry of the heat transfer surfaces, and other factors (Figure 1-16). It may be expressed by

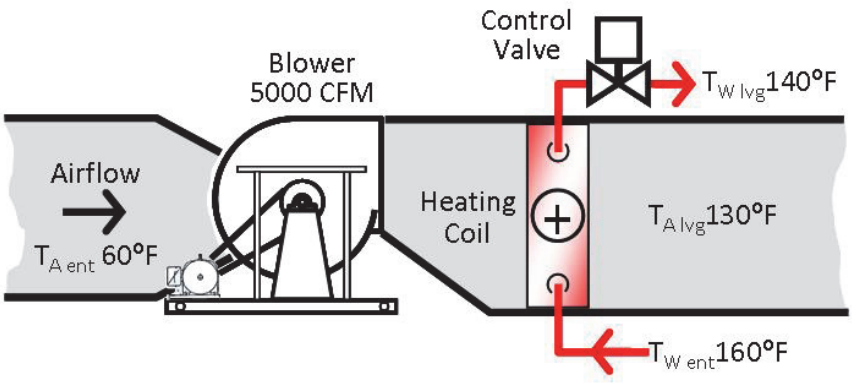


Figure 1-16 Example system with heating coil.

$$q = UA(\text{LMTD}) \quad \textbf{(1-4)}$$

where

A = surface area, ft^2
LMTD = logarithmic mean temperature difference, heated medium to water, °F
Q = heat transfer rate, Btu/h
U = overall coefficient of heat transfer, $Btu/h \cdot ft^2 \cdot °F$

Example 1-2

In Figure 1-17, assume that the coil has a U of 155 $Btu/h \cdot ft^2 \cdot °F$ per row. The upstream temperature is 60°F, and the air temperature leaving the coil is 130°F. Water enters the heating coil at 160°F and leaves at 140°F. The duct size is 48 × 36 in. and the coil has four rows. Find the heat transfer rate (see also Figure 1-18).

Solution

$$q = UA(\text{LMTD})$$

$$\text{LMTD} = \frac{\Delta t_{max} - \Delta t_{min}}{\text{Ln}\left(\frac{\Delta t_{max}}{\Delta t_{min}}\right)} \qquad \Delta t_{max} = 140 - 60 = 80 \qquad \Delta t_{min} = 160 - 30 = 30$$

$$\text{LMTD} = \frac{80 - 30}{\text{Ln}\left(\frac{80}{30}\right)} = 51°\text{F}$$

Next, using LMTD, find q.

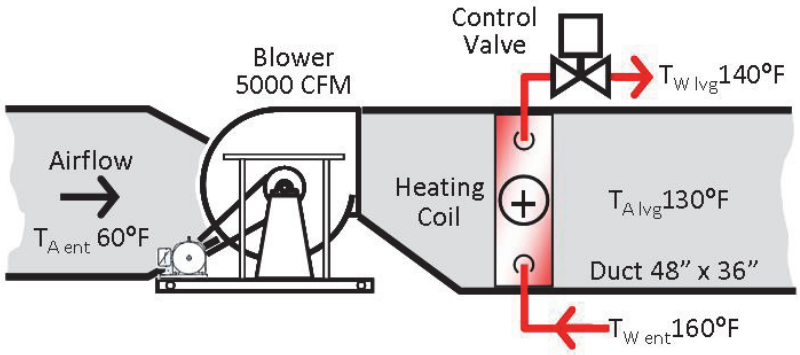


Figure 1-17 Example system with heating coil.

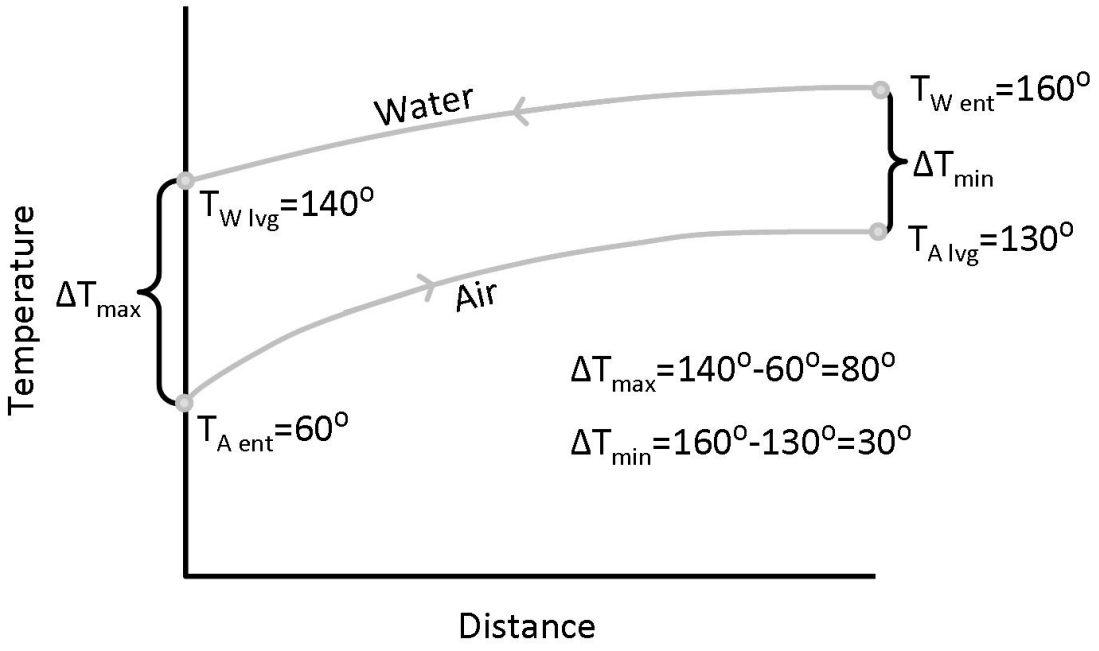


Figure 1-18 Coil LMTD example.

$$q = UA(\text{LMTD})$$

$$= \left(155\ \frac{\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}{\text{Row}}\right)(3\ \text{ft} \times 4\ \text{ft})(51^\circ\text{F})(4\ \text{Rows})$$

$$q = 379{,}440\ \text{Btu/h}$$

Latent Cooling and Dehumidification of Air

The quantity of heat removed from the cooled medium (Figure 1-18) when both sensible cooling and dehumidification are present is expressed by

$$q_t = W\Delta h \qquad \textbf{(1-5)}$$

where

q_t = total heat transfer rate, Btu/h
W = mass flow rate of cooled medium, lb/h
Δh = enthalpy difference of entering and leaving conditions of cooled medium, Btu/lb

Expressed for a cooling coil, this equation becomes

$$q_t = 60Q_a\rho_a\Delta h \qquad \textbf{(1-6)}$$

where

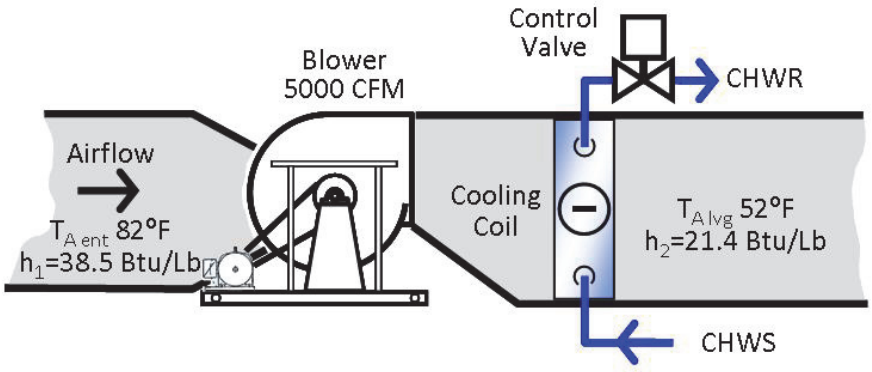


Figure 1-19 Example system with cooling coil.

Q_a = airflow rate, cfm

ρ_a = density of air, lb/ft^3

For standard air, the density is 0.075 lb/ft^3 and the formula reduces to

$$q_t = 4.5 Q_a \Delta h \quad \textbf{(1-7)}$$

Example 1-3

For the system shown in Figure 1-19, determine the heat transfer rate for the sensible cooling/dehumidification process, assuming entering air is 82°F, enthalpy is 38.5 Btu/lb, and discharge air is 52°F and saturated at enthalpy of 21.4 Btu/lb.

Solution

$q_t = 4.5 Q_a \Delta h$

= (4.5)(5000 cfm)(38.5 – 21.4) Btu/h

= 384,750 Btu/h

Heat Transferred to or from Water

The quantity of heat transferred to or from the water is a function of the flow rate, the specific heat, and the temperature drop or rise of the water as it passes through the heat exchanger. The heat transferred to or from the water is expressed by

$$q_t = m c_p \Delta t \quad \textbf{(1-8)}$$

where

c_p = specific heat of water, Btu/lb·°F

m = mass flow rate of water, lb/h

q_t = total heat transfer rate, Btu/h

Δt = water temperature increase or decrease across unit, °F

These equations are also used to express the heat-carrying capacity of the piping or distribution system of any portion of that system. In this regard, the temperature differential, sometimes called the *temperature range*, is established or identified. For any flow rate through the piping, q_w is called the *heat-carrying capacity.*

With water systems, it is common to express the flow rate in gallons per minute (gpm), in which case the equation becomes

$$q_t = 8.02\rho_w c_p Q_w \Delta t \qquad \textbf{(1-9)}$$

where

Q_w = water flow rate, gpm

ρ_w = density of water, lb/ft^3

For standard conditions in which density is 62.4 lb/ft^3 and specific heat is 1 Btu/lb·°F, the equation becomes

$$q_t = 500 Q_w \Delta t \qquad \textbf{(1-10)}$$

Equations 1-9 and 1-10 can be used to express the heat transfer across a single load or source device or any quantity of such devices connected across a piping system. In the design or diagnosis of a system, the load side may be balanced with the source side by these equations.

Example 1-4

For a single system similar to that shown in Figure 1-16, assume the heat transfer rate across the coil is 378,000 Btu/h and the Δt of the water supplying the coil is 20°F (water and air at standard conditions). Find the water flow rate required for the system.

Solution

$$q_t = 500 Q_w \Delta t$$

$$Q_w = \frac{q_t}{500 \times \Delta t}$$

$$Q_w = \frac{378{,}000 \text{ Btu/h}}{(500 \text{ Btu·min/°F·h·gal})(20°\text{F})}$$

$$Q_w = 37.8 \text{ gpm}$$

Example 1-5

Assume q_t (cooling coil) = 384,750 Btu/h and Δt = 12°F.

Solution

$$Q_w = \frac{384{,}750 \text{ Btu/h}}{(500 \text{ Btu·min/°F·h·gal})(12°\text{F})}$$

$$Q_w = 64.1 \text{ gpm}$$

Load Systems

Load systems are the devices (terminal units) that convey heat from the water for heating or to the water for cooling of the space or process. Most load systems are basically water-to-air finned-coil heat exchangers or water-to-water heat exchangers. The specific configuration is usually used to describe the device. Common configurations include the following:

- **Heating load devices:** Preheat coils in central air-handling units, heating coils in central air-handling units, zone or central reheat coils, finned-tube radiation, baseboard radiation, convectors, unit heaters, fan-coil units, water-to-water heat exchangers, radiant heating panels, and snow-melting panels
- **Cooling load devices:** Coils in central units, fan-coil units (Figure 1-20), induction unit coils, radiant cooling panels, and water-to-water heat exchangers

It is at this point that further consideration of the coil selection becomes important. Figure 1-21 shows a representative coil. Typically, as air enters the coil it passes over tubes carrying the water back to the source to be reconditioned. Those tubes are arranged in a geometrical pattern selected to achieve the required heat transfer and air and water pressure drop considerations. Tubes are connected to a supply or return header, which either supplies water to the coil or directs it back to the source. The designer has a number of criteria that can be adjusted in achieving the selection, albeit they are bounded by the heat transfer characteristic and the psychrometric process the HVAC process needs to achieve. While the application of the psychrometric chart has been previ-

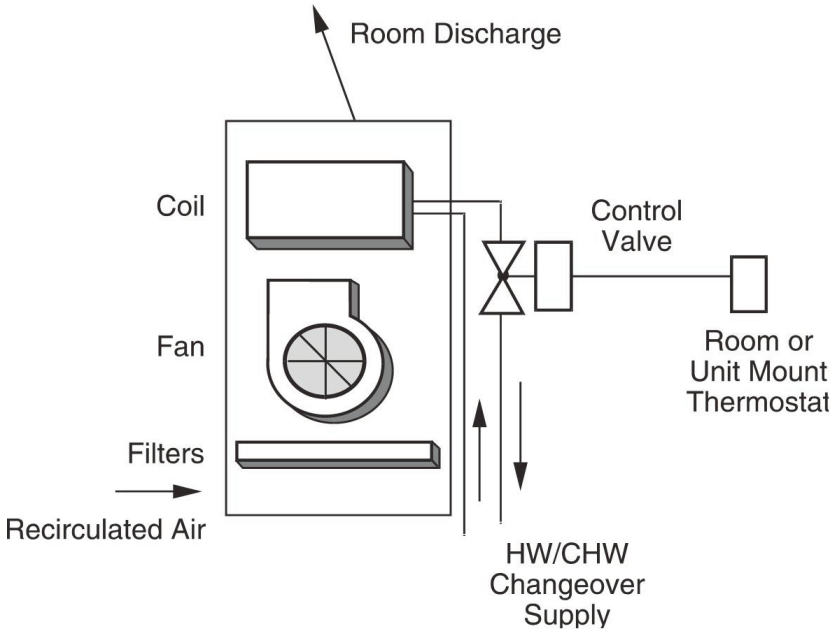


Figure 1-20 Fan-coil unit.

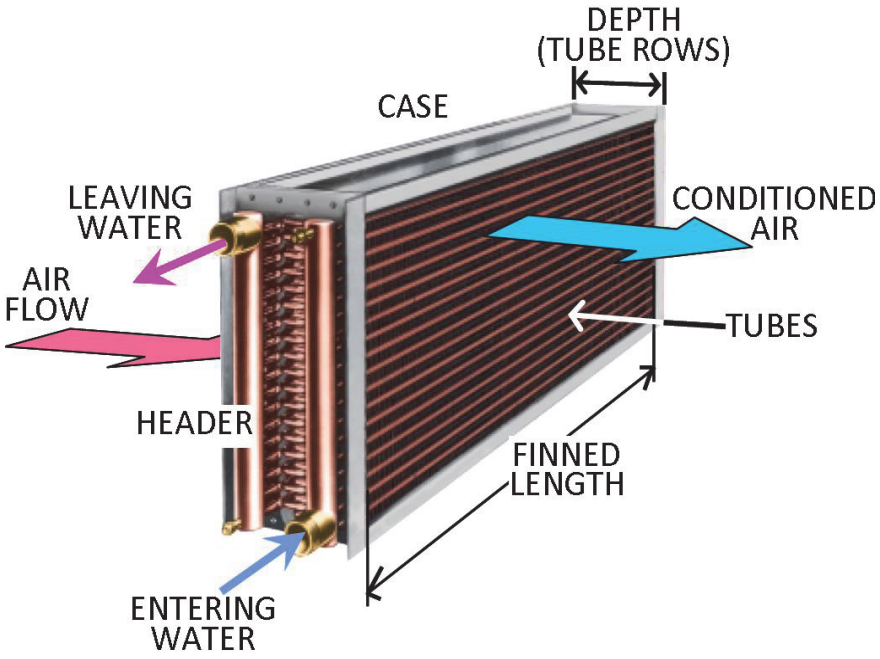


Figure 1-21 Typical coil.

ously noted, it becomes useful in determining what to select for a heat transfer device. Load calculation programs are often applied by designers for calculation of the required heating or cooling, which generates a gross value for the required heating or cooling. While space and outdoor design conditions are specified by the designer, these programs do not select the required heat transfer components, because it is just too early in the deign process to do so. The load calculation program does not have the capability to evaluate how many air-handling units are required and what their respective sizes should be. The designer's link then is to take the load calculation data and apply it to the correct heat transfer device based on the design criteria of applied equipment. This implies selecting physical equipment to perform the air-conditioning function. Often getting the equipment to perform like the theoretical equation can be daunting.

For example, consider the determination of a cooling coil and its characteristics for a CHW coil per the following criteria:

- Office space with a sensible cooling load of 42,000 Btu/h and a total latent load of 18,000 Btu/h
- Space design conditions are 75°F/50% rh
- Outdoor design conditions are 95°F db/75°F wb
- Ventilation air is 30% of the total
- Supply air is delivered at 55°F
- 65% efficient draw-through blower at 5.0 in. total pressure
- 12°F design Δt CHW rise through the coil

In this example, we have to incorporate our basic knowledge of psychrometrics and load calculation but also account for the energy use as the unit operates, represented by the fan energy causing a temperature rise. Similarly, we could also take into account temperature rise for the airflow in the ductwork.

We know the following:

Sensible heating

$$Q = 1.08 \times \text{cfm} \times (\Delta t) \Rightarrow \quad \text{cfm} = \left(\frac{Q}{1.08 \times \Delta t}\right)$$

Fan heat energy balance

$$Q = 1.08 \times \text{cfm} \times (\Delta t)$$

$$P_{\text{WATTS}} = \frac{0.1175 \times \text{cfm} \times \Delta P_{\text{IN}}}{\mu_{\text{F}}}$$

$$P_{\text{BTU}} = P_{\text{WATTS}} \times 3.412$$

Energy balance

$$1.08 \times \text{cfm} \times (\Delta t) = \frac{0.1175 \times 3.412 \times \text{cfm} \times \Delta P_{\text{IN}}}{\mu_{\text{F}}}$$

$$\Delta t = \frac{0.1175 \times 3.412 \times \text{cfm} \times \Delta P_{\text{IN}}}{1.08 \times \text{cfm} \times \mu_{\text{F}}} = \frac{0.371 \times \Delta P_{\text{IN}}}{\mu_{\text{F}}} \qquad \textbf{(I-11)}$$

The process could be sketched psychrometrically per Figure 1-22.

From the psychrometric chart, the desired space conditions are fixed, as is the design temperature for the supply air temperature. In order for the process to work psychrometrically, the supply air condition must be along the sensible heat ratio (SHR) line. Calculating SHR at 0.7, using either ASHRAE Psychrometric Chart No. 1 (1992) compass or other means, a line can be drawn at the SHR slope and through the space condition and the design supply air temperature, showing the entering point at about 55°F/80% rh (51.5°F wb).

Blower energy must be accounted for, which is a sensible rise calculated to be about 3°F. This is also plotted and shows the coil leaving conditions to be 52°F/90% rh (about 50°F wb).

Sensible heating

$$Q = 1.08 \times \text{cfm} \times (\Delta t) \Rightarrow \text{cfm}$$

$$= \left(\frac{Q}{1.08 \times \Delta t}\right) = \left[\frac{42{,}000}{1.08 \times (75 - 55)}\right] \approx 1950 \text{ cfm}$$

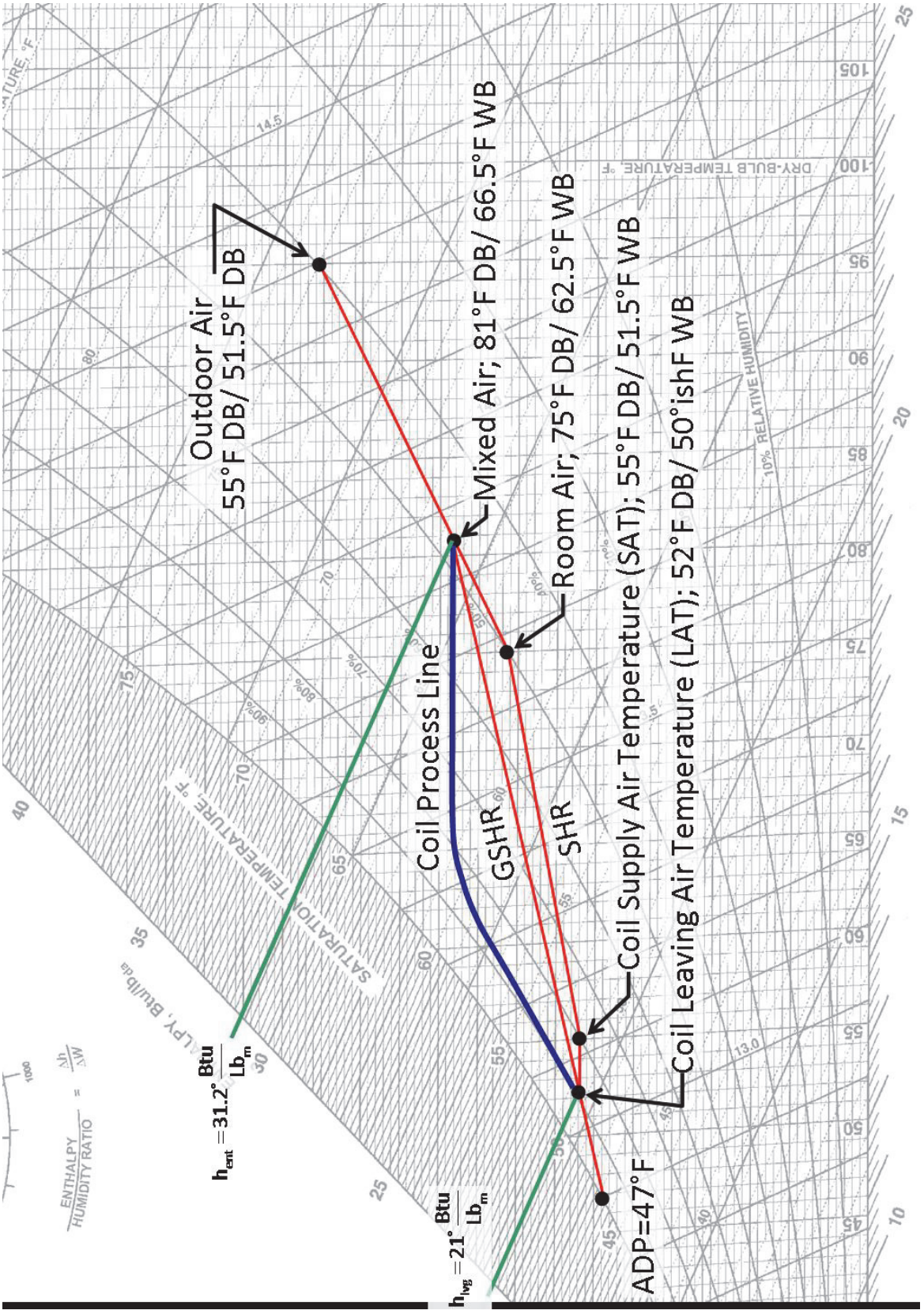


Figure 1-22 Example process diagrammed on ASHRAE Psychrometric Chart.

This allows the linear process line, sometimes called the *grand sensible heat ratio* (GSHR), from the coil entering condition at the mixed-air point for 30% outdoor air at 81°F/48% rh (66.5°F wb) through the leaving-air point. Extend the graphic plot of the line so that it intersects with the saturation line of the psychrometric chart; it must intersect for the process to work psychrometrically. If it does not, the process must be modified to one that does. The intersection point is known as the *apparatus dew point* (ADP) and has a specific function as it relates to coil selection—it helps set the entering CHW temperature for the coil. This is also a relative indicator of how well the process works. If the entering water temperature is too low, again reevaluate the process.

Blower temperature rise

$$\Delta t = \frac{0.371 \times \Delta P_{IN}}{\eta_F} = \frac{0.371 \times 5 \text{ in.}}{0.65} = 2.9°\text{F}$$

Blower sensible heat

$$Q_{FAN} = 1.08 \times 1480 \times 2.9°\text{F} = 4600 \text{ Btu/h}$$

The ADP is an average temperature representing the coil surface and is what allows condensation for dehumidification to occur. The ADP in this example is about 47°F (depending on either how good one's eyesight is or how large the psychrometric chart has been printed!) The coil entering water temperature (EWT) can be estimated using the following equation:

$$\text{Coil EWT} = (2 \times \text{ADP}) - T_{WBLVG} \quad \textbf{(1-12)}$$

$$\text{Coil EWT} = (2 \times 47°\text{F}) - 50°\text{F} = 44°\text{F}$$

It appears plotted on the ASHRAE Psychrometric Chart in Figure 1-22.

Coil selection does not end with the establishment of the selection data; this merely gives the required data to search out the heat transfer device from a manufacturer. In that regard, most manufacturers will provide the selection service or provide complimentary software that will find either a single coil or a series of coils that provide varying levels of performance. An example of a manufacturer's coil selection program is shown in Figure 1-23.

Coil selection is an iteration process, and rarely does a real product provide exactly the desired psychrometric process as specified by the designer. The only datum not established for selection is that of the required flow rate, and to do that the designer must set design criteria for the water-side temperature difference for the coil. The ADP calculation established an entering temperature of 44°F. For the example, we will use a moderate Δt for design of 12°F, implying a leaving water temperature of 56°F, just as, previously, design flow rate is calculated as follows:

$$Q_{COIL} = 4.5 \times \text{cfm} \times \Delta h = 4.5 \times 1950 \times (31.2 - 21) = 89{,}500 \text{ Btu/h}$$

$$q(\text{gpm}) = \frac{Q_{COIL}}{500 \times \Delta t_W} = \frac{89{,}500}{500 \times 12} = 15 \text{ gpm}$$

A variety of options are up to the designer in coil selection. Typical tube sizes are 0.375, 0.5, and 0.625 in. in water coils. The quantity of fins per inch (FPI) of tube has a large effect on overall heat transfer as well as dehumidification. In that regard, the depth of the coil generally referred to as the rows with the quantity of tubes is equally important.

All air that flows through the coil does not come into physical contact with the metal surfaces of the coil. The air that does not is called *bypass air*. Getting a coil to provide the required heat transfer requires playing with all of the dimensions that affect the surface area exposed to the air as it passes through the coil as well as giving the air enough time to come into contact with that metal. For CHW coils, air velocities are normally limited to a maximum of 500 fpm to reduce the potential for condensate carryover in the airstream. Velocity can be lower and often is to achieve the required latent energy removal. Table 1-1 provides guidance on coil depth and velocity to achieve dehumidification. In addition, carefully consider the quantity of FPI of the coil. While packing in the fins increases heat transfer and reduces coil

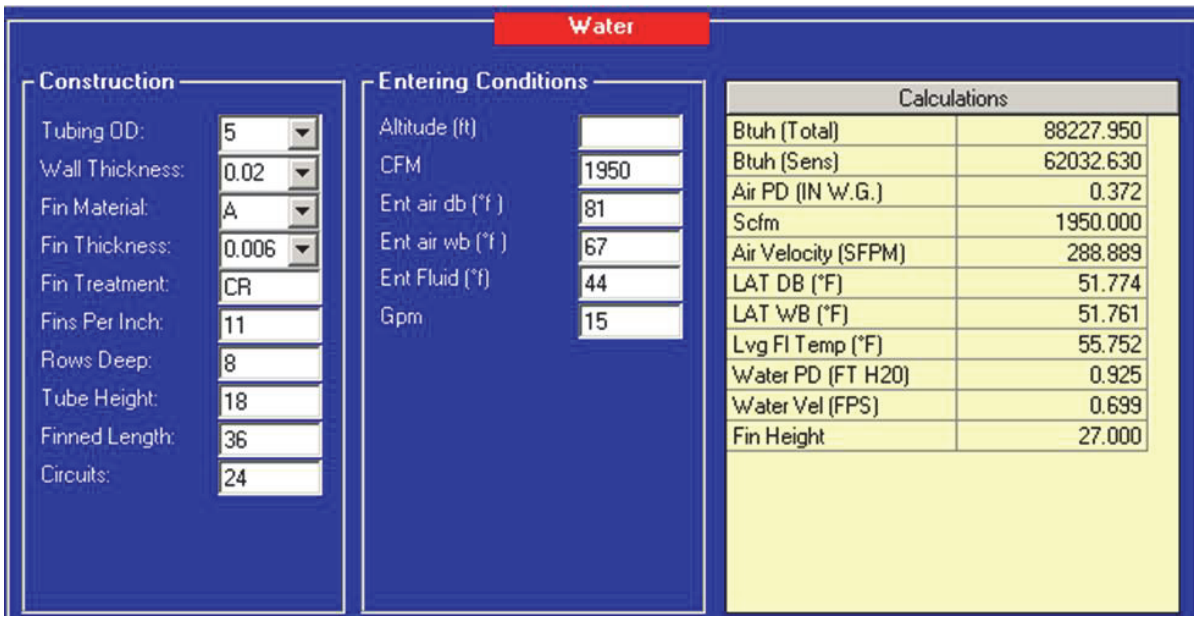


Figure 1-23 Example of manufacturer's coil selection program.

Table 1-1 Coil Depth and Velocity to Achieve Dehumidification

Coil Load, cfm/ton	Face Velocity, fpm	Coil Rows
600	500	3
500	500	4
400	400	4
300	300	6
200	200	8

size, it may degrade latent energy transfer and become a maintenance problem, collecting dirt on the fin surface. (Remember, they get wet!) Generally, try to limit fin spacing to a maximum of 8 FPI in cooling and dehumidification applications. It may not always be possible, but it is a good starting point for evaluation.

Note that in this selection, we are very close to the psychrometric process that we plotted on the chart. We are not exactly at 89,500 Btu/h, but we are very close. Similarly, we are at a little less than the required 52°F discharge temperature and similarly at slightly less than the ideal water-side ΔT. All things considered though, this is not a bad first pass at selection. Air-side pressure drop is a reasonable 0.4 in., and water-side pressure drop is a very respectable 1 ft. Note though that it is 8 tubes deep by 18 tubes high, or 144 copper tubes. Coil cost is all about the quantity of metal and the labor to connect everything together, so this might be a pricey coil selection (only your coil provider knows for sure), and you may only know by selecting a few comparable coils and asking for estimates.

Less easy to get will be a coil characteristic (as portrayed in Figure 1-24), which either will require data from the manufacturer for your calculation or may be given to you by them. This is a helpful graph in determining control strategy

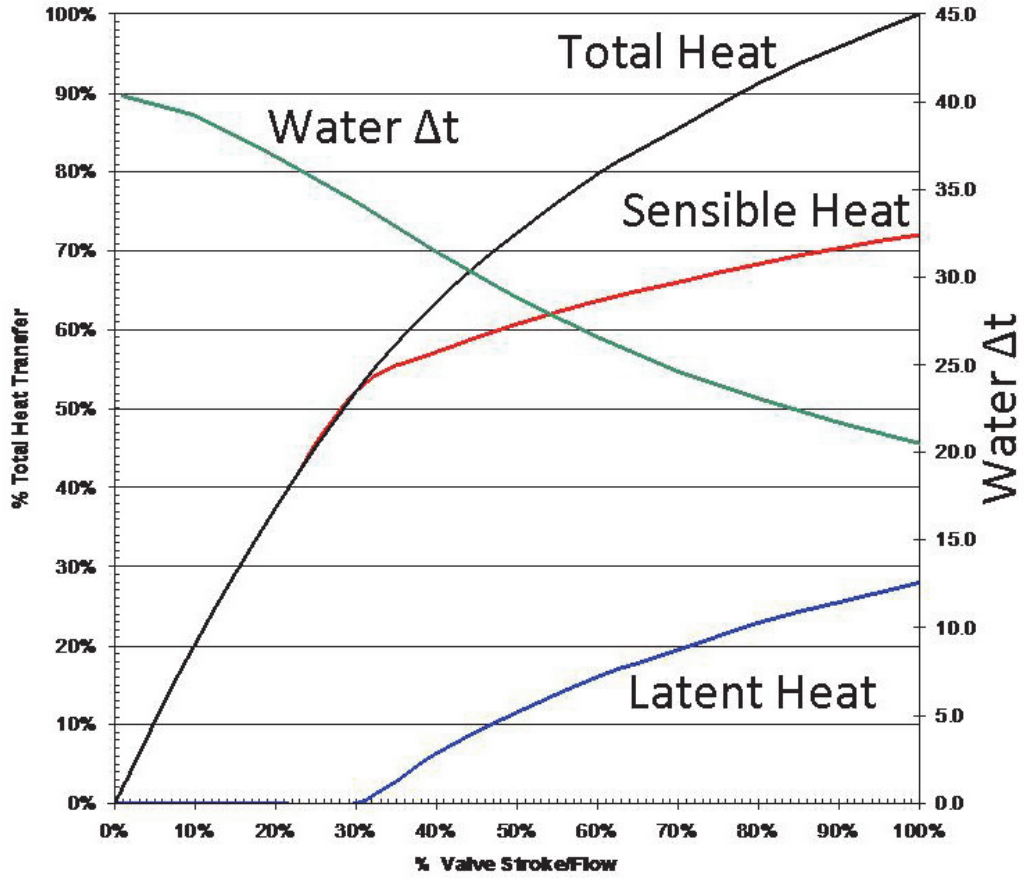


Figure 1-24 Heat transfer versus water flow for a coil.

and part-load performance. It is a normalized graph (percentage heat transfer versus percentage water flow) so that it is not influenced by the magnitude of heat transfer (Btu/h versus flow), and generally it is shown in this form holding the entering air conditions to the coil as fixed.

Generally, the characteristic plots retain this type of shape. It is the sensible heat curve that is important to the control valve selection and performance, with the shape typically being the influence in selection of equal percentage valve characteristics, as is discussed in Chapter 5. Recall that this is generated with a fixed entering condition, and if the same type of evaluation is performed where specific entering conditions at specific loads are examined, one might find a wide range of shapes and required flow rates. Often, it makes the shape more linear, but that is not always the case. Similarly, note how dehumidification starts at about 30% flow to the coil. That take-off point too can vary based on the geometry of the selected coil. Typically, with less depth or higher airflow velocities, that starting point happens to the right, at larger flow rates. In this particular coil selection (which is slightly different from the previous example), a water-side Δt of 20°F was used, and note that as flow decreases to the coil, Δt rises, as would be expected for any throttled condition. This needs to be checked against the chiller to make sure that entering water to the chiller does not have a maximum criteria.

Another observation of this coil characteristic is that it is for throttled flow. Were the coil pumped, the entering water temperature would be controlled by throttling a percentage of the chilled water being circulated to the coil. That makes the coil characteristic more linear in performance with respect to the percentage of chilled water provided to the coil. Why do such a thing? Specifically, when there may be concern about freezing a coil in situations where that can arise or in situations where as a designer you always want to provide the full latent energy (dehumidification) heat transfer effect of the coil. Options such as pumping the coil and changing the heat transfer characteristic are some of the great tools that chilled-water systems provide in air-conditioning system design.

The Next Step

In Chapter 2, you will be introduced to piping system design.

Summary

Chapter 1 covered the following topics:

- What determines the load
- The difference between closed and open systems
- Components of a hydronic system
- Heating versus cooling source devices
- How systems need to meet part-load conditions

- Temperature and pressure ranges for low-, medium-, and high-temperature water systems
- Sensible, latent, and total heat loads and how they affect design water flow
- Examples of heating and cooling load devices
- How load diversity suggests a reduction in total cooling capacity required

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Skill Development Exercises for Chapter 1

Complete these questions by writing your answers on the worksheets at the back of this book.

1-1 Water systems that convey heat to or from a conditioned space or process with hot or chilled water are frequently called _____.

1-2 What is the fundamental difference between closed and open types of water systems?

1-3 A cooling tower has at least two points of interface with air. Where are they?

1-4 What is the maximum working pressure for LTW boiler systems?

1-5 What is a CHW system? How is it different from a CW system?

1-6 What are the fundamental components of a closed hydronic system?

1-7 Explain the most common source devices for heating and cooling systems.

1-8 Explain what *load* means.

1-9 What factors influence the heating and cooling load requirements?

1-10 Define sensible heat transfer.

1-11 Name five heating load devices and describe how each is used in system applications.

2 Piping System Design

Study Objectives

After completing this chapter, you should be able to do the following:

- ❑ understand Bernoulli's principle,
- ❑ know the three steps in design of a fluid distribution system,
- ❑ understand the difference between direct-return and reverse-return piping and if they can be combined,
- ❑ know methods to allow thermal expansion,
- ❑ determine pressure drop in piping,
- ❑ know the difference between laminar and turbulent flow and what index quantifies it,
- ❑ understand piping roughness factors, and
- ❑ know what governs pressure drop in a piping system.

Instructions

Read Chapter 2 and answer all of the questions at the end.

Basic Considerations

The piping system is a key component of the distribution system, and good design practice can significantly affect the performance and energy efficiency of an HVAC system. This chapter discusses the key aspects of piping system design, including pipe sizing and system design philosophy. In the design of any fluid distribution system, you must consider the following steps (Coad 1985):

- Establish the piping design philosophy and objectives.
- Size the pipes.
- Calculate or determine the pressure drop in the system as a whole or in various subparts or branches.

To achieve the best energy efficiency, you may need to repeat these steps several times to optimize the design.