

## CHAPTER 12

# HYDRONIC HEATING AND COOLING SYSTEM DESIGN

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**W**ATER systems that convey heat to or from a conditioned space or process with hot or chilled water are frequently called *hydronic systems*. Water flows through piping that connects a boiler, water heater, or chiller to suitable terminal heat transfer units located at the space or process.

Water systems can be classified by (1) operating temperature, (2) flow generation, (3) pressurization, (4) piping arrangement, and (5) pumping arrangement.

Classified by flow generation, hydronic heating systems may be (1) **gravity systems**, which use the difference in density between the supply and return water columns of a circuit or system to circulate water; or (2) **forced systems**, in which a pump, usually driven by an electric motor, maintains flow. Gravity systems are seldom used today and are therefore not discussed in this chapter. See the *ASHVE Heating Ventilating Air Conditioning Guide* issued before 1957 for information on gravity systems.

Water systems can be either once-through or recirculating systems. This chapter describes forced recirculating systems.

Successful water system design depends on awareness of the many complex interrelationships between various elements. In a practical sense, no component can be selected without considering its effect on the other elements. For example, design water temperature and flow rates are interrelated, as are the system layout and pump selection. The type and control of heat exchangers used affect the flow rate and pump selection, and the pump selection and distribution affect the controllability. The designer must thus work back and forth between tentative points and their effects until a satisfactory integrated design has been reached. Because of these relationships, rules of thumb usually do not lead to a satisfactory design.

### Principles

Effective and economical water system design is affected by complex relationships between the various system components. The design water temperature, flow rate, piping layout, pump selection, terminal unit selection, and control method are all interrelated. System size and complexity determine the importance of these relationships to the total system operating success. In the United States, present hydronic heating system design practice originated in residential heating applications, where a temperature drop  $\Delta t$  of 20°F was used to determine flow rate. Besides producing satisfactory operation and economy in small systems, this  $\Delta t$  enabled simple calculations because 1 gpm conveys 10,000 Btu/h. However, almost universal use of hydronic systems for both heating and cooling of large buildings and building complexes has rendered this simplified approach obsolete.

The preparation of this chapter is assigned to TC 6.1, Hydronic and Steam Equipment and Systems.

### TEMPERATURE CLASSIFICATIONS

Water systems can be classified by operating temperature as follows.

**Low-temperature water (LTW) systems** operate within the pressure and temperature limits of the ASME *Boiler and Pressure Vessel Code* for low-pressure boilers. The maximum allowable working pressure for low-pressure boilers is 160 psig, with a maximum temperature of 250°F. The usual maximum working pressure for boilers for LTW systems is 30 psi, although boilers specifically designed, tested, and stamped for higher pressures are frequently used. Steam-to-water or water-to-water heat exchangers are also used for heating low-temperature water. Low-temperature water systems are used in buildings ranging from small, single dwellings to very large and complex structures.

**Medium-temperature water (MTW) systems** operate between 250 and 350°F, with pressures not exceeding 160 psi. The usual design supply temperature is approximately 250 to 325°F, with a usual pressure rating of 150 psi for boilers and equipment.

**High-temperature water (HTW) systems** operate at temperatures over 350°F and usual pressures of about 300 psi. The maximum design supply water temperature is usually about 400°F, with a pressure rating for boilers and equipment of about 300 psi. The pressure-temperature rating of each component must be checked against the system's design characteristics.

**Chilled-water (CW) systems** for cooling normally operate with a design supply water temperature of 40 to 55°F (usually 44 or 45°F), and at a pressure of up to 120 psi. Antifreeze or brine solutions may be used for applications (usually 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.

**Dual-temperature water (DTW) systems** combine heating and cooling, and circulate hot and/or chilled water through common piping and terminal heat transfer apparatus. These systems operate within the pressure and temperature limits of LTW systems, with usual winter design supply water temperatures of about 100 to 150°F and summer supply water temperatures of 40 to 45°F.

Terminal heat transfer units include convectors, cast-iron radiators, baseboard and commercial finned-tube units, fan-coil units, unit heaters, unit ventilators, central station air-handling units, radiant panels, and snow-melting panels. A large storage tank may be included in the system to store energy to use when heat input devices such as the boiler or a solar energy collector are not supplying energy.

This chapter covers the principles and procedures for designing and selecting piping and components for low-temperature water, chilled water, and dual-temperature water systems. See [Chapter 14](#) for information on medium- and high-temperature water systems.

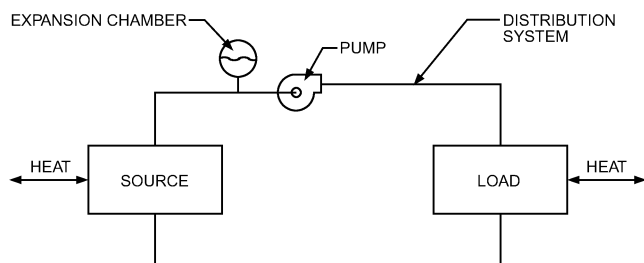


Fig. 1 Fundamental Components of Hydronic System

## CLOSED WATER SYSTEMS

Because most hot- and chilled-water systems are closed, this chapter addresses only closed systems. The fundamental difference between a closed and an open water system is the interface of the water with a compressible gas (such as air) or an elastic surface (such as a diaphragm). A **closed water system** is defined as one with no more than one point of interface with a compressible gas or surface, and that will not create system flow by changes in elevation. This definition is fundamental to understanding the hydraulic dynamics of these systems. Earlier literature referred to a system with an open or vented expansion tank as an “open” system, but this is actually a closed system; the atmospheric interface of the tank simply establishes the system pressure.

An **open system**, on the other hand, has more than one such interface. For example, a cooling tower system has at least two points of interface: the tower basin and the discharge pipe or nozzles entering the tower. One major difference in hydraulics between open and closed systems is that some hydraulic characteristics of open systems cannot occur in closed systems. For example, in contrast to the hydraulics of an open system, in a closed system (1) flow cannot be motivated by static head differences, (2) pumps do not provide static lift, and (3) the entire piping system is always filled with water.

Figure 1 shows the fundamental components of a closed hydronic system. Actual systems generally have additional components such as valves, vents, regulators, etc., but these are not essential to the basic principles underlying the system.

These fundamental components are

- Loads
- Source
- Expansion chamber
- Pump
- Distribution system

Theoretically, a hydronic system could operate with only these five components.

The components are subdivided into two groups: thermal and hydraulic. Thermal components consist of the load, source, and expansion chamber. Hydraulic components consist of the distribution system, pump, and expansion chamber. The expansion chamber is the only component that serves both a thermal and a hydraulic function.

## METHOD OF DESIGN

This section outlines general steps a designer may follow to complete system design. The methodology is not a rigid framework, but rather a flexible outline that should be adapted by the designer to suit current needs. The general order as shown is approximately chronological, but it is important to note that succeeding steps often affect preceding steps, so a fundamental reading of this entire chapter is required to fully understand the design process.

1. **Determine system and zone loads.** Loads are covered in Chapters 27 to 32 of the 2005 *ASHRAE Handbook—Fundamentals*. Several load calculation procedures have been developed, with

varying degrees of calculation accuracy. The load determines the flow of the hydronic system, which ultimately affects the system's heat transfer ability and energy performance. Designers should apply the latest computerized calculation methods for optimal system design. Load calculation should also detail the facility's loading profile facility to enhance the hydronic system control strategy.

2. **Select comfort heat transfer devices.** This often means a coil- or water-to-air heat exchanger (terminal). Coil selection and operation has the single largest influence on hydronic system design. Coils implement the design criteria of flow, temperature drop, and control ability. Coil head loss and location affects pipe design and sizing, control devices, and pump selection. For details on coils, see [Chapters 22](#) and [26](#).
3. **Select system distribution style(s).** Based on the load and its location, different piping styles may be appropriate for a given design. Styles may be combined in a successful hydronic system design to optimize building performance. Schematically lay out the system to establish a preliminary design.
4. **Size branch piping system.** Based on the selection of the coil, its controlling devices, style of installation, and location, branch piping is sized to provide required flow, and head loss is calculated.
5. **Calculate distribution piping head loss.** Although the criteria for pipe selection in branch and distribution system piping may be similar, understanding the relationship and effect of distribution system head loss is important in establishing that all terminals get the required flow for the required heat transfer.
6. **Lay out piping system and size pipes.** After preliminary calculations of target friction loss for the pipes, sketch the system. After the piping system is laid out and the calculations of actual design head loss are complete, note the losses on the drawings for the commissioning process.
7. **Select pump specialties.** Any devices required for operation or measurement are identified, so their head loss can be determined and accounted for in pump selection.
8. **Select air management methodology.** All hydronic systems entrain air in the circulated fluid. Managing the collection of that air as it leaves the working fluid is essential to management of system pressure and the safe operation of system components.
9. **Select pump (hydraulic components).** Unless a system is very small (e.g., a residential hot-water heating system), the pump is selected to fit the system. A significant portion of energy use in a hydronic system is transporting the fluid through the distribution system. Proper pump selection limits this energy use, whereas improper selection leads to energy inefficiency and poor distribution and heat transfer.
10. **Determine installation details, iterate design.** Tuning the design to increase performance and cost effectiveness is an important last step. Documenting installation details is also important, because this communication is necessary for well-built designs and properly operated systems.

## THERMAL COMPONENTS

### Loads

The load is the device that causes heat to flow out of or into the system to or from the space or process; it is the independent variable to which the remainder of the system must respond. Outward heat flow characterizes a heating system, and inward heat flow characterizes a cooling system. The quantity of heating or cooling is calculated by one of the following means.

**Sensible Heating or Cooling.** The rate of heat entering or leaving an airstream is expressed as follows:

$$q = 60Q_a \rho_a c_p \Delta t \quad (1)$$

where

$q$  = heat transfer rate to or from air, Btu/h  
 $Q_a$  = airflow rate, cfm  
 $\rho_a$  = density of air, lb/ft<sup>3</sup>  
 $c_p$  = specific heat of air, Btu/lb · °F  
 $\Delta t$  = temperature increase or decrease of air, °F

For standard air with a density of 0.075 lb/ft<sup>3</sup> and a specific heat of 0.24 Btu/lb · °F, Equation (1) becomes

$$q = 1.1 Q_a \Delta t \quad (2)$$

The heat exchanger or coil must then transfer this heat from or to the water. The rate of sensible heat transfer to or from the heated or cooled medium in a specific heat exchanger is a function of the heat transfer surface area; the mean temperature difference between the water and the medium; and the overall heat transfer coefficient, which itself is a function of the fluid velocities, properties of the medium, geometry of the heat transfer surfaces, and other factors. The rate of heat transfer may be expressed by

$$q = UA(\text{LMTD}) \quad (3)$$

where

$q$  = heat transfer rate through heat exchanger, Btu/h  
 $U$  = overall coefficient of heat transfer, Btu/h · ft<sup>2</sup> · °F  
 $A$  = heat transfer surface area, ft<sup>2</sup>  
 LMTD = logarithmic mean temperature difference, heated or cooled medium to water, °F

**Cooling and Dehumidification.** The rate of heat removal from the cooled medium when both sensible cooling and dehumidification are present is expressed by

$$q_t = w \Delta h \quad (4)$$

where

$q_t$  = total heat transfer rate from cooled medium, Btu/h  
 $w$  = mass flow rate of cooled medium, lb/h  
 $\Delta h$  = enthalpy difference between entering and leaving conditions of cooled medium, Btu/lb

Expressed for an air-cooling coil, this equation becomes

$$q_t = 60Q_a \rho_a \Delta h \quad (5)$$

which, for standard air with a density of 0.075 lb/ft<sup>3</sup>, reduces to

$$q_t = 4.5 Q_a \Delta h \quad (6)$$

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

$$q_w = \dot{m} c_p \Delta t \quad (7)$$

where

$q_w$  = heat transfer rate to or from water, Btu/h  
 $\dot{m}$  = mass flow rate of water, lb/h  
 $c_p$  = specific heat of water, Btu/lb · °F  
 $\Delta t$  = water temperature increase or decrease across unit, °F

With water systems, it is common to express the flow rate as volumetric flow, in which case Equation (7) becomes

$$q_w = 8.02 \rho_w c_p Q_w \Delta t \quad (8)$$

where

$Q_w$  = water flow rate, gpm  
 $\rho_w$  = density of water, lb/ft<sup>3</sup>

For standard conditions in which the density is 62.4 lb/ft<sup>3</sup> and the specific heat is 1 Btu/lb · °F, Equation (8) becomes

$$q_w = 500 Q_w \Delta t \quad (9)$$

Equation (8) or (9) can be used to express the heat transfer across a single load or source device, or any quantity of such devices connected across a piping system. In the design or diagnosis of a system, the load side may be balanced with the source side using these equations.

**Heat-Carrying Capacity of Piping.** Equations (8) and (9) are also used to express the heat-carrying capacity of the piping or distribution system or any portion thereof. The existing temperature differential  $\Delta t$ , sometimes called the temperature range, is identified; for any flow rate  $Q_w$  through the piping,  $q_w$  is called the **heat-carrying capacity**.

### Terminal Heating and Cooling Units

Many types of terminal units are used in central water systems, and may be classified in several different ways:

- **Natural convection units** include cabinet convectors, baseboard, and finned-tube radiation. Older systems may have cast-iron radiators, which are sometimes sought out for architectural restoration.
- **Forced-convection units** include unit heaters, unit ventilators, fan-coil units, air-handling units, heating and cooling coils in central station units, and most process heat exchangers. Fan-coil units, unit ventilators, and central station units can be used for heating, ventilating, and cooling.
- **Radiation units** include panel systems, unit radiant panels, in-floor or wall piping systems, and some older styles of radiators. All transfer some convective heat. These units are generally used in low-temperature water systems, with lower design temperatures. Similarly, chilled panels are also used for sensible cooling and in conjunction with central station air-handling units isolating outdoor air conditioning.

Terminal units must be selected for sufficient capacity to match the calculated heating and cooling loads. Manufacturers' ratings should be used with reference to actual operating conditions. Ratings are either computer selected, or cataloged by water temperature, temperature drop or rise, entering air temperatures, water velocity, and airflow. Ratings are usually given for standard test conditions with correction factors or curves, and rating tables are given covering a range of operating conditions. Because the choice of terminal units for any particular building or type of system is so wide, the designer must carefully consider the advantages and disadvantages of the various alternatives so the end result is maximum comfort and economy.

**Most load devices** (in which heat is conveyed to or from the water for heating or cooling the space or process) are a water-to-air finned-coil heat exchanger or a water-to-water exchanger. The specific configuration is usually used to describe the load device. The most common configurations include the following:

#### Heating load devices

Preheat coils in central units  
 Heating coils in central units  
 Zone or central reheat coils  
 Finned-tube radiators  
 Baseboard radiators

- Convectors
- Unit heaters
- Fan-coil units
- Water-to-water heat exchangers
- Radiant heating panels
- Snow-melting panels

#### Cooling load devices

- Coils in central units
- Fan-coil units
- Induction unit coils
- Radiant cooling panels
- Water-to-water heat exchangers

### Source

The source is the point where heat is added to (heating) or removed from (cooling) the system. Ideally, the amount of energy entering or leaving the source equals the amount entering or leaving through the load. Under steady-state conditions, the load energy and source energy are equal and opposite. Also, when properly measured or calculated, temperature differentials and flow rates across the source and loads are all equal. Equations (8) and (9) express the source capacities as well as the load capacities.

Any device that can be used to heat or cool water under controlled conditions can be used as a source device. The most common source devices for heating and cooling systems are the following:

#### Heating source devices

- Hot-water generator or boiler
- Steam-to-water heat exchanger
- Water-to-water heat exchanger
- Solar heating panels
- Heat recovery or salvage heat device  
(e.g., water jacket of an internal combustion engine)
- Exhaust gas heat exchanger
- Incinerator heat exchanger
- Heat pump condenser
- Air-to-water heat exchanger

#### Cooling source devices

- Electric compression chiller
- Thermal absorption chiller
- Heat pump evaporator
- Air-to-water heat exchanger
- Water-to-water heat exchanger

The two primary considerations in selecting a source device are the design capacity and the part-load capability, sometimes called the **turndown ratio**. The turndown ratio, expressed in percent of design capacity, is

$$\text{Turndown ratio} = 100 \frac{\text{Minimum capacity}}{\text{Design capacity}} \quad (10)$$

The reciprocal of the turndown ratio is sometimes used (for example, a turndown ratio of 25% may also be expressed as a turn-down ratio of 4).

The turndown ratio has a significant effect on system performance; lack of consideration of the source system's part-load capability has been responsible for many systems that either do not function properly or do so at the expense of excess energy consumption. The turndown ratio has a significant effect on the ultimate equipment and/or system design selection.

Note that the turndown ratio for a source is different from that specified for a control valve. Turndown for a valve is comparable to valve rangeability. Whereas **rangeability** is the relationship of the maximum controllable flow to the minimum controllable flow

based on testing, **turndown** is the relationship of the valve's normal maximum flow to minimum controllable flow.

**System Temperatures.** Design temperatures and temperature ranges are selected by consideration of the performance requirements and the economics of the components. For a cooling system that must maintain 50% rh at 75°F, the dew-point temperature is 55°F, which sets the maximum return water temperature at something near 55°F (60°F maximum); on the other hand, the lowest practical temperature for refrigeration, considering the freezing point and economics, is about 40°F. This temperature spread then sets constraints for a chilled-water system. Pedersen et al. (1998) describe a classic method for calculating the required temperature of chilled water from the psychrometric chart.

The entering water temperature follows the relationship

$$t_w = 2t_{ad} - t_{wb} \quad (11)$$

where

- $t_w$  = Coil entering water temperature
- $t_{ad}$  = Apparatus dew point
- $t_{wb}$  = Coil leaving air wet-bulb temperature

The designer should note that there are also constraints imposed on the temperature and differential temperature selection by the chiller selection (e.g., refrigerant choice), and on the coil's flow tolerance with respect to heat transfer. Consult with the chiller manufacturer so that performance requirements of the chiller are taken into account.

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

### Expansion Chamber

The expansion chamber (also called an **expansion** or **compression tank**) serves both a thermal and a hydraulic function. In its thermal function, the tank provides a space into which the non-compressible liquid can expand or from which it can contract as the liquid undergoes volumetric changes with changes in temperature. To allow for this expansion or contraction, the expansion tank provides an interface point between the system fluid and a compressible gas. By definition, a closed system can have only one such interface; thus, a system designed to function as a closed system can have only one expansion chamber.

Expansion tanks are of three basic configurations: (1) a closed tank, which contains a captured volume of compressed air and water, with an air/water interface (sometimes called a **plain steel tank**); (2) an open tank (i.e., a tank open to the atmosphere); and (3) a diaphragm tank, in which a flexible membrane is inserted between the air and the water (a modified version is the **bladder tank**).

Properly installed, a closed or diaphragm tank serves the purpose of system pressurization control with a minimum of exposure to air in the system. Open tanks, commonly used in older systems, tend to introduce air into the system, which can enhance piping corrosion. Open tanks are generally not recommended for application in current designs. Older-style steel compression tanks tend to be larger than diaphragm expansion tanks. In some cases, there may be economic considerations that make one tank preferable over another. These economics usually are relatively straightforward (e.g., initial cost), but there can be significant size differences, which affect placement and required building space and structural support, and these effects should also be considered.

Sizing the tank is the primary thermal consideration in incorporating a tank into a system. However, before sizing the tank, air



control or elimination must be considered. The amount of air that will be absorbed and can be held in solution with the water is expressed by Henry's equation (Pompei 1981):

$$x = p/H \quad (12)$$

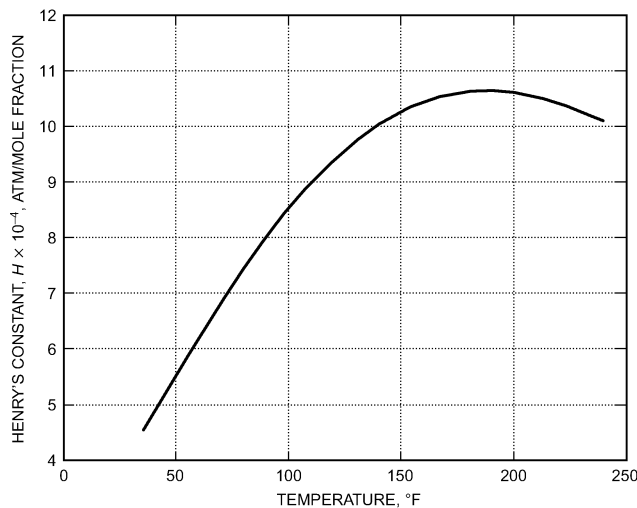
where

$x$  = solubility of air in water (% by volume)

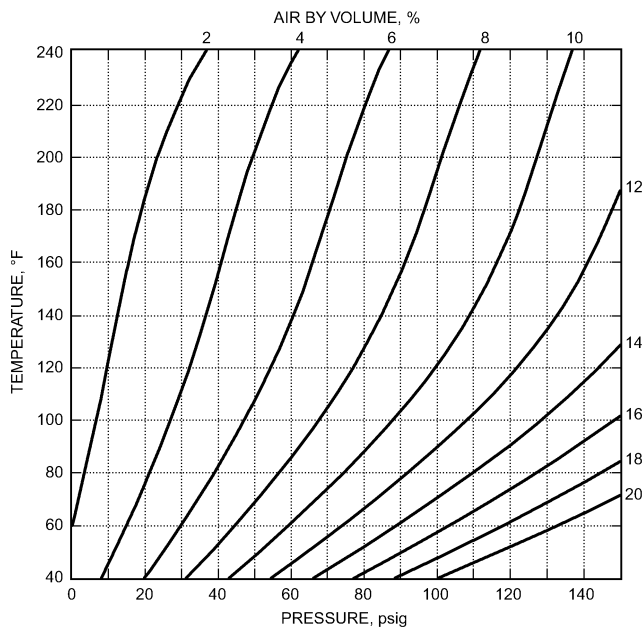
$p$  = absolute pressure

$H$  = Henry's constant

Henry's constant, however, is constant only for a given temperature (Figure 2). Combining the data of Figure 2 (Himmelblau 1960) with Equation (12) results in the solubility diagram of Figure 3.



**Fig. 2 Henry's Constant Versus Temperature for Air and Water**  
(Coad 1980a)



**Fig. 3 Solubility Versus Temperature and Pressure for Air/Water Solutions**  
(Coad 1980a)

With that diagram, the solubility can be determined if the temperature and pressure are known.

If the water is not saturated with air, it will absorb air at the air/water interface until the point of saturation has been reached. Once absorbed, the air will move throughout the body of water either by mass migration or by molecular diffusion until the water is uniformly saturated. If the air/water solution changes to a state that reduces solubility, excess air will be released as a gas. For example, if the air/water interface is at high pressure, the water will absorb air to its limit of solubility at that point; if at another point in the system the pressure is reduced, some of the dissolved air will be released.

In the design of systems with open or plain steel expansion tanks, it is common practice to use the tank as the major air control or release point in the system.

Equations for sizing the three common configurations of expansion tanks follow (Coad 1980b):

**For closed tanks with air/water interface**

$$V_t = V_s \frac{[(v_2/v_1) - 1] - 3\alpha \Delta t}{(P_a/P_1) - (P_a/P_2)} \quad (13)$$

**For open tanks with air/water interface**

$$V_t = 2V_s \left[ \left( \frac{v_2}{v_1} - 1 \right) - 3\alpha \Delta t \right] \quad (14)$$

**For diaphragm tanks**

$$V_t = V_s \frac{[(v_2/v_1) - 1] - 3\alpha \Delta t}{1 - (P_1/P_2)} \quad (15)$$

where

$V_t$  = volume of expansion tank, gal

$V_s$  = volume of water in system, gal

$t_1$  = lower temperature, °F

$t_2$  = higher temperature, °F

$P_a$  = atmospheric pressure, psia

$P_1$  = pressure at lower temperature, psia

$P_2$  = pressure at higher temperature, psia

$v_1$  = specific volume of water at lower temperature, ft<sup>3</sup>/lb

$v_2$  = specific volume of water at higher temperature, ft<sup>3</sup>/lb

$\alpha$  = linear coefficient of thermal expansion, in/in · °F

=  $6.5 \times 10^{-6}$  in/in · °F for steel

=  $9.5 \times 10^{-6}$  in/in · °F for copper

$\Delta t$  =  $(t_2 - t_1)$ , °F

As an example, the lower temperature for a heating system is usually normal ambient temperature at fill conditions (e.g., 50°F) and the higher temperature is the operating supply water temperature for the system. For a chilled-water system, the lower temperature is the design chilled-water supply temperature, and the higher temperature is ambient temperature (e.g., 95°F). For a dual-temperature hot/chilled system, the lower temperature is the chilled-water design supply temperature, and the higher temperature is the heating water design supply temperature.

For specific volume and saturation pressure of water at various temperatures, see Table 3 in Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals*.

At the tank connection point, the pressure in closed-tank systems increases as the water temperature increases. Pressures at the expansion tank are generally set by the following parameters:

- The lower pressure is usually selected to hold a positive pressure at the highest point in the system (usually about 10 psig).
- The higher pressure is normally set by the maximum pressure allowable at the location of the safety relief valve(s) without opening them.

Other considerations are to ensure that (1) the pressure at no point in the system will ever drop below the saturation pressure at the operating system temperature and (2) all pumps have sufficient net positive suction head (NPSH) available to prevent cavitation.

**Example 1.** Size an expansion tank for a heating water system that will operate at a design temperature range of 180 to 220°F. The minimum pressure at the tank is 10 psig (24.7 psia) and the maximum pressure is 25 psig (39.7 psia). (Atmospheric pressure is 14.7 psia.) The volume of water is 3000 gal. The piping is steel.

1. Calculate the required size for a closed tank with an air/water interface.

**Solution:** For lower temperature  $t_1$ , use 40°F.

From Table 3 in Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals*,

$$v_1(\text{at } 40^\circ\text{F}) = 0.01602 \text{ ft}^3/\text{lb}$$

$$v_2(\text{at } 220^\circ\text{F}) = 0.01677 \text{ ft}^3/\text{lb}$$

Using Equation (13),

$$V_t = 3000 \times \frac{[(0.01677/0.01602) - 1] - 3(6.5 \times 10^{-6})(220 - 40)}{(14.7/24.7) - (14.7/39.7)}$$

$$V_t = 578 \text{ gal}$$

2. If a diaphragm tank were used in lieu of the plain steel tank, what tank size would be required?

**Solution:** Using Equation (15),

$$V_t = 3000 \times \frac{[(0.01677/0.01602) - 1] - 3(6.5 \times 10^{-6})(220 - 40)}{1 - (24.7/39.7)}$$

$$V_t = 344 \text{ gal}$$

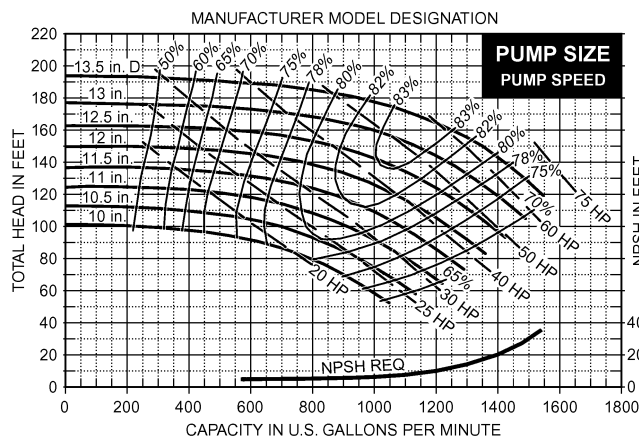
## HYDRAULIC COMPONENTS

### Pump or Pumping System

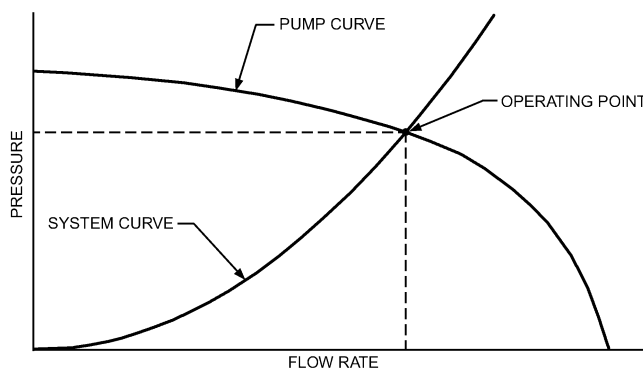
Centrifugal pumps are the most common type in hydronic systems (see [Chapter 43](#)). Circulating pumps used in water systems can vary in size from small in-line circulators delivering 5 gpm at 6 or 7 ft head to base-mounted or vertical pumps handling hundreds or thousands of gallons per minute, with pressures limited only by the system characteristics. Pump operating characteristics must be carefully matched to system operating requirements.

**Pump Curves and Water Temperature for Constant-Speed Systems.** Performance characteristics of centrifugal pumps are described by pump curves, which plot flow versus head or pressure, as well as by efficiency and power information, as shown in [Figure 4](#). Large pumps tend to have a series of curves, designated with a numerical size in inches (10 to 13.5 in. in diameter), to represent performance of the pump impeller and outline the envelope of pump operation. Intersecting elliptical lines designate the pump's efficiency. The **net positive suction head (NPSH)** required line represents the required entering operating pressure for the pump to operate satisfactorily. Diagonal lines represent the required power of the pump motor. The point at which a pump operates is the point at which the pump curve intersects the system curve ([Figure 5](#)).

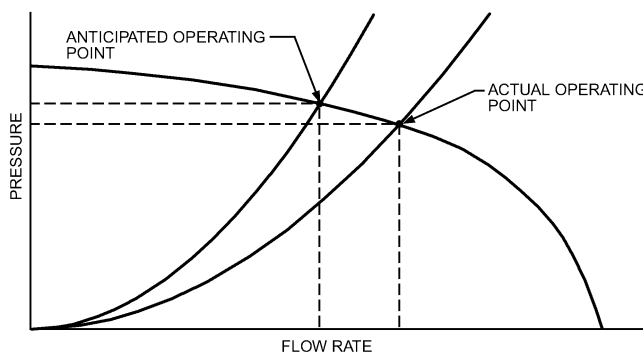
In [Figure 4](#), note that each performance curve has a defined end point. Small circulating pumps, which may exhibit a pump curve as shown in [Figure 6](#), may actually extend to the abscissa showing a run-out flow, and may also not show multiple impellers, efficiency, or NPSH. Large pumps do not exhibit the run-out flow characteristic. In a large pump, the area to the right of the curve is an area of unsatisfactory performance, and may represent pump operation in a state of cavitation. It is important that the system curve always intersect the pump curve in operation, and the design must ensure that system operation stays on the pump curve.



**Fig. 4 Example of Manufacturer's Published Pump Curve**



**Fig. 5 Pump Curve and System Curve**

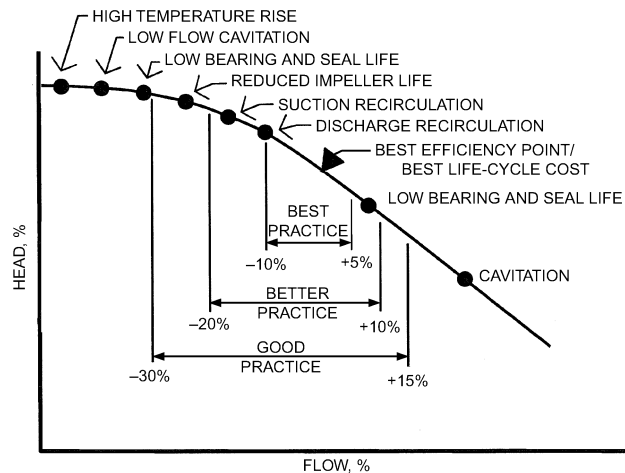


**Fig. 6 Shift of System Curve Caused by Circuit Unbalance**

[Chapter 43](#) also discusses system and pump curves.

A complete piping system follows the same water flow/pressure drop relationships as any component of the system [see Equation (17)]. Thus, the pressure required for any proposed flow rate through the system may be determined and a system curve constructed. A pump may be selected by using the calculated system pressure at the design flow rate as the base point value.

[Figure 6](#) illustrates how a shift of the system curve to the right affects system flow rate. This shift can be caused by incorrectly calculating the system pressure drop by using arbitrary safety factors or overstated pressure drop charts. Variable system flow caused by control valve operation, a larger than required control valve, or improperly balanced systems (subcircuits having substantially lower pressure drops than the longest circuit) can also cause a shift to the right.



**Fig. 7 General Pump Operating Condition Effects**  
(Hydraulic Institute)

As described in [Chapter 43](#), pumps for closed-loop piping systems should have a flat pressure characteristic and should operate slightly to the left of the peak efficiency point on their curves. This allows the system curve to shift to the right without causing undesirable pump operation, overloading, or reduction in available pressure across circuits with large pressure drops.

Many dual-temperature systems are designed so that the chillers are bypassed during winter. The chiller pressure drop, which may be quite high, is thus eliminated from the system pressure drop, and the pump shift to the right may be quite large. For such systems, system curve analysis should be used to check winter operating points.

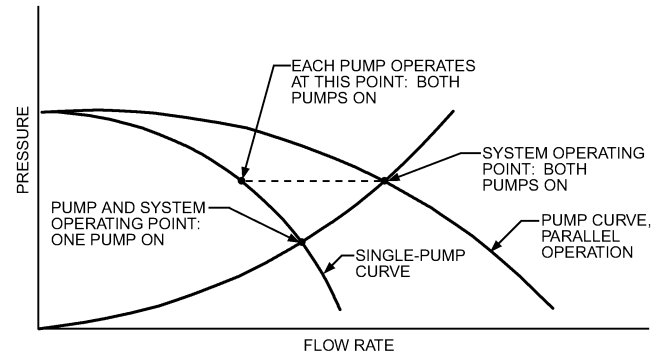
Operating points may be highly variable, depending on (1) load conditions, (2) the types of control valves used, and (3) the piping circuitry and heat transfer elements. In general, the best selection in smaller systems is

- For design flow rates calculated using pressure drop charts that illustrate actual closed-loop hydronic system piping pressure drops
- To the left of the maximum efficiency point of the pump curve to allow shifts to the right caused by system circuit unbalance, direct-return circuitry applications, and modulating three-way valve applications
- A pump with a flat curve to compensate for unbalanced circuitry and to provide a minimum pressure differential increase across two-way control valves

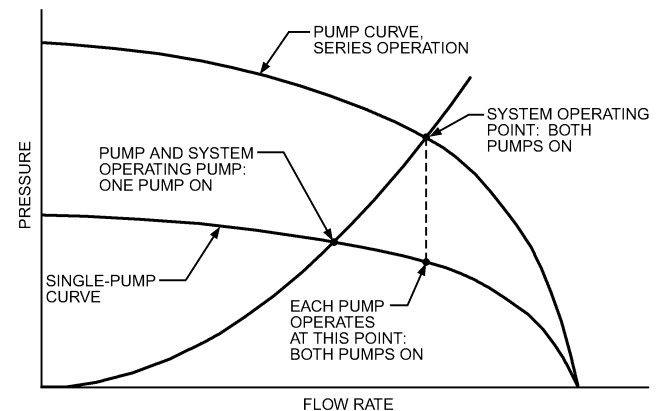
As system sizes and corresponding pump sizes increase in size, more care is needed in analysis of the pump selection. The Hydraulic Institute (HI 2000) offers a detailed discussion of pump operation and selection to optimize life cycle costs. The HI guide covers all types of pumping systems, including those that are much more sophisticated than a basic HVAC closed-loop circulating system, and use much more power. There are direct parallels, though. HI's discussion of pump reliability sensitivity includes a chart similar to that shown in [Figure 7](#).

HVAC pumps tend to be low-energy devices compared to industrial or process pumps, which might be responsible for a wide variety of different fluids and operating conditions. Select a pump as close as possible to the best efficiency point, to optimize life-cycle costs and maximize operating life with a minimum of maintenance. HI's recommendations are also appropriate for HVAC pump selection.

**Parallel Pumping.** When pumps are applied in parallel, each pump operates at the same head, and provides its share of the system flow at that pressure ([Figure 8](#)). Generally, pumps of equal size are used, and the parallel-pump curve is established by doubling the flow of the single-pump curve (with identical pumps).



**Fig. 8 Operating Conditions for Parallel-Pump Installation**



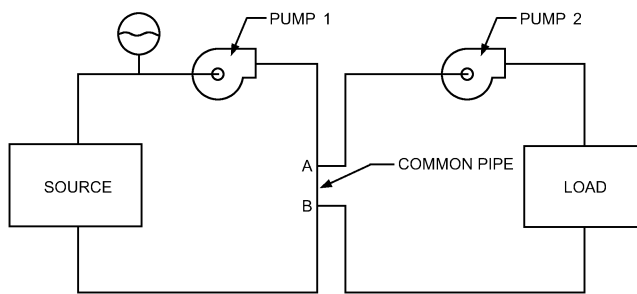
**Fig. 9 Operating Conditions for Series-Pump Installation**

Plotting a system curve across the parallel-pump curve shows the operating points for both single- and parallel-pump operation ([Figure 8](#)). Note that single-pump operation does not yield 50% flow. The system curve crosses the single-pump curve considerably to the right of its operating point when both pumps are running. This leads to two important concerns: (1) the pumps must be powered to prevent overloading during single-pump operation, and (2) a single pump can provide standby service of up to 80% of design flow; the actual amount depends on the specific pump curve and system curve. As pumps become larger, or more than two pumps are placed in parallel operation, it is still very important to ensure in the design that the operating system intersects the operating pump curve, and that there are safeties in place to ensure that, should a pump be turned off, the remaining pumps and system curve still intersect one another.

**Series Pumping.** When pumps are operated in series, each pump operates at the same flow rate and provides its share of the total pressure at that flow. A system curve plotted across the series-pump curve shows the operating points for both single- and series-pump operation ([Figure 9](#)). Note that the single pump can provide up to 80% flow for standby and at a lower power requirement.

Series-pump installations are often used in heating and cooling systems so that both pumps operate during the cooling season to provide maximum flow and head, whereas only a single pump operates during the heating season. Note that both parallel- and series-pump applications require that the actual pump operating points be used to accurately determine the pumping point. Adding artificial safety factor head, using improper pressure drop charts, or incorrectly calculating pressure drops may lead to an unwise selection.

**Multiple-Pump Systems.** Care must be taken in designing systems with multiple pumps to ensure that, if pumps ever operate in either parallel or series, such operation is fully understood and considered by the designer. Pumps performing unexpectedly in



**Fig. 10 Compound Pumping (Primary-Secondary Pumping)**

series or parallel have caused performance problems in hydronic systems, such as the following:

**Parallel.** With pumps of unequal pressures, one pump may create a pressure across the other pump in excess of its cutoff pressure, causing flow through the second pump to diminish significantly or to cease. This can cause flow problems or pump damage.

**Series.** With pumps of different flow capacities, the pump of greater capacity may overflow the pump of lesser capacity, which could cause damaging cavitation in the smaller pump and could actually cause a pressure drop rather than a pressure rise across that pump. In other circumstances, unexpected series operation can cause excessively high or low pressures that can damage system components.

**Standby Pump Provision.** If total flow standby capacity is required, a properly valved standby pump of equal capacity is installed to operate when the normal pump is inoperable. A single standby may be provided for several similarly sized pumps. Parallel- or series-pump installation can provide up to 80% standby, which is often sufficient.

**Compound Pumping.** In larger systems, compound pumping, also known as **primary-secondary pumping**, is often used to provide system advantages that would not be available with a single pumping system. Compound pumping is illustrated in [Figure 10](#).

In [Figure 10](#), pump 1 can be referred to as the **source** or **primary pump** and pump 2 as the **load** or **secondary pump**. The short section of pipe between A and B is called the **common pipe** (also called the **decoupling line** or **neutral bridge**) because it is common to both the source and load circuits. In the design of compound systems, the common pipe should be kept as short and as large in diameter as practical to minimize pressure loss between those two points. Care must be taken, however, to ensure adequate length in the common pipe to prevent recirculation from entry or exit turbulence. There should never be a valve or check valve in the common pipe. If these conditions are met and the pressure loss in the common pipe can be assumed to be zero, then neither pump will affect the other. Then, except for the system static pressure at any given point, the circuits can be designed and analyzed and will function dynamically independently of one another.

In [Figure 10](#), if pump 1 has the same flow capacity in its circuit as pump 2 has in its circuit, all of the flow entering point A from pump 1 will leave in the branch supplying pump 2, and no water will flow in the common pipe. Under this condition, the water entering the load will be at the same temperature as that leaving the source.

If the flow capacity of pump 1 exceeds that of pump 2, some water will flow downward in the common pipe. Under this condition, Tee A is a diverting tee, and Tee B becomes a mixing tee. Again, the temperature of the fluid entering the load is the same as that leaving the source. However, because of the mixing taking place at point B, the temperature of water returning to the source is between the source supply temperature and the load return temperature.

On the other hand, if the flow capacity of pump 1 is less than that of pump 2, then point A becomes a mixing point because some water must recirculate upward in the common pipe from point B. The temperature of the water entering the load is between the supply water temperature from the source and the return water temperature from the load.

For example, if pump 1 circulates 25 gpm of water leaving the source at 200°F, and pump 2 circulates 50 gpm of water leaving the load at 100°F, then the water temperature entering the load is

$$t_{load} = 200 - (25/50)(200 - 100) = 150^\circ\text{F} \quad (16)$$

Mixing is a primary reason against application of compound pumping systems, particularly in chilled-water systems with constant-speed pumping applied to the primary pump circuit of the source, and variable-speed pumping applied to the secondary system connected to the loads. The issues associated with this are generally source- and control-related. At one time, chiller manufacturers restricted water flow variation through a chiller, despite the fact that designers and system operators wanted this ability, to increase energy operating economy and reduce operating costs by reducing flow. Mixing was an inevitable by-product, reducing the chiller's operating efficiency. This situation was exacerbated when variable-speed drives were applied to pumps, while chiller pumps stayed constant-speed. Eventually, changes to chiller operating controls allowed flow rate through the chiller evaporator to be varied. Some system designs have shifted away from compound pumping to variable-speed, variable-flow primary pumping to enhance system efficiency. Depending on locale, system size, and selection criteria for the chiller, compound pumping may still be beneficial to system design and operating stability. Chiller selections often are limited to maximum velocities of 7 fps, and minimum velocities of 3 fps. Equal-percentage valve characteristics, though, should operate most often at valve strokes less than 80%, which is a flow rate of less than 40%, and less than the 42% that might represent a low-flow limit for the chiller in a 3 fps operation criterion. Variable-speed pumping in both primary and secondary circuits can be applied to reduce mixing that might occur as chillers are sequenced on and off to compensate for the loads.

The following are some advantages of compound circuits:

- They allow different water temperatures and temperature ranges in different elements of the system. Compound pumping can be used to vary coil capacity by controlling coil temperature, which leads to better latent energy control.
- They decouple the circuits hydraulically, thereby making the control, operation, and analysis of large systems much less complex. Hydraulic decoupling also prevents unwanted series or parallel operation. Large water circuits often experience pressure imbalances, but compound pumping allows a design to be hydraulically organized into separate smaller subsystems, making troubleshooting and operation easier.

### Variable-Speed Pumping Application

Centrifugal pumps may also be operated with variable-frequency drives, which adjust the speed of the electric motor, changing the pump curve. In this application, a pump controller and typically one frequency drive per pump are applied to control the required system variable.

The most typical application is to control differential pressure across one or more branches of the piping network. In a common application, the differential pressure is sensed across a control valve or, alternatively, the valve, coil, and branch ([Figure 11](#)). The controller is given a set point equal to the pressure loss of the sensed components at design flow. For example, a 5 psi differential pressure was used to size the control valve; the sensor is connected to sense differential pressure across the valve, so a 5 psi set point is given to



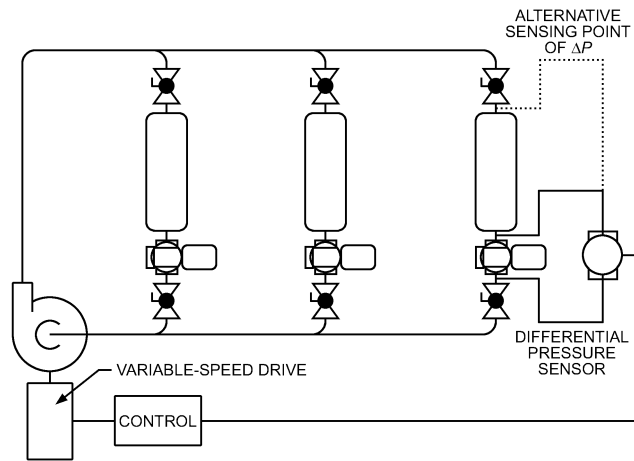


Fig. 11 Example of Variable-Speed Pump System Schematic

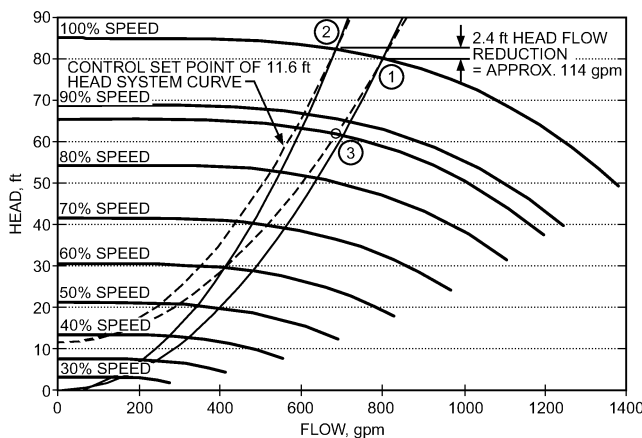


Fig. 12 Example of Variable-Speed Pump and System Curves

the controller. As the control valve closes in reaction to a control signal (Figure 12) from 1 to 2, the differential pressure across the branch rises as the system curve shifts counterclockwise up the pump curve. The pump controller, sensing an increase in differential, decreases the speed of the pump to about point 3, roughly 88% speed. Note that each system curve is shown with two representations. The system curve is the simple relationship of the flow ratio squared to the head ratio. Under control of a pump controller with a single sensor across the control valve, as shown in Figure 11, the pump decreases in speed until the theoretical zero-flow point, at which the pump goes just fast enough to maintain a differential pressure under no flow. In this case, the speed is about 88%, and the power is about 68%. In operation, expect there to be a difference in performance. Figure 12 shows the change as a step function, opposite of the way in which the pump controller functions. Based on the control method, the controller adjustment settings (gain, integral time, and derivative time), system hydraulics, valve time constant, sensor sensitivity, etc., it is far more likely that a small incremental rise in differential pressure will have a corresponding small decrease in pump speed, as the valve repositions itself in reaction to its control signal. This appears as a small sawtoothed series of system curve and pump curve intersection points, and is not of real importance. What is important is that many factors influence operation, and theoretical variable-speed pumping is different from reality.

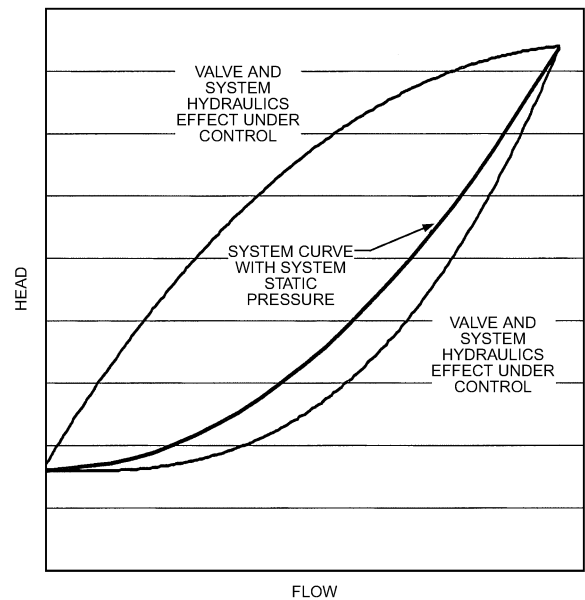


Fig. 13 System Curve with System Static Pressure (Control Area)

Decreasing pump speed is analogous to using a smaller impeller size in the volute of the pump, and the result is that motor power is reduced by the cube of the speed reduction, closely following the affinity laws. (Variable-speed pump curves are shown in Chapter 43.) Design flow conditions should be minimal hours of operation per year; as such, the energy savings potential for a variable-speed pump is great. In the general control valve application, a reasonably selected equal percentage valve with a 50% valve authority should reduce flow by about 30% when the valve is positioned from 100% open (design flow, minimal hours per year) to 90% open. The 10% change in stroke should reduce pump operating power about 70%. Hydronic systems should operate most of the time at flow rates well below design. The coil characteristic suggests that, in sensible applications, a small percentage of flow yields an exceptionally high degree of coil design heat transfer, adequate for most of the year's operation.

Depending on system design, direct digital control may also allow more advanced control strategies. There are various reset control strategies (e.g., cascade control) to optimize pump speed and flow performance. Many of these monitor valve position and drive the pump to a level that keeps one valve open while maintaining set point for comfort conditions. Physical operational requirements of the components must be taken into account. There are some concerns over operating pumps and their motor drives at speeds less than 30% of design, particularly about maintaining proper lubrication of the pump mechanical seals and motor bearings. From a practical perspective, 30% speed is a scant 3% of design power, so it may be unnecessary to reduce speed any further. Consult manufacturers for information on device limits, to maintain the system in good operating condition.

Exceptional energy reduction potential and advances in variable-frequency drive technology that have reduced drive costs have made the application of variable-speed drives common on closed hydronic distribution systems. Successful application of a variable-speed drive to a pump is not a given, however, and depends on the designer's skill and understanding of system operation.

For instance, the designer should understand the system curve with system static pressure, as shown in Figure 13. This control phenomenon of variable-speed, variable-flow pumping systems is often called the **control area**; Hegberg (2003) describes the analysis used to create these graphs.

The plot represents the system curves of different operation points, created by valve positions, and their intersection points with a pump curve. These discrete points occur under specific imposed points of operation on the control valves of the hydronic system, and represent potential worst-case operation points in the system. Typically, this imposed sequence is that control valves are closed in specific order relative to their location to the controlled pump, and the total dynamic head of the pump is calculated at each position. When there is one sensor of control, the boundaries as shown are created. The upper system boundary represents system head as valves are closed sequentially, starting with those nearest the pump and ending with those closer to the sensor. The lower boundary curve represents system head and flow when valves are closed sequentially from the location of the sensor toward the controlled pump.

The importance of these two boundaries is that, for any given system flow, there are an infinite number of system heads that may be required, based on which control valves are open and to what position. This is opposite of the system curve concept, which is one flow, one head loss in a piping system. The graph is a representation of a controlled operation; the piping system follows the principles of the Darcy equation, but the intervening feedback control adjustments to the pump speed produce an installed characteristic in the system different from what might otherwise be expected. The implication of any boundary above the generic system curve relationship to design flow is that, as flow decreases in the system, it does not follow the affinity law relationship in reducing operating power. Conversely, the boundary below the system curve operates with a greater reduction in power than the generic system curve.

Although operating below the system curve may seem to be advantageous, saving energy and pumping costs, the designer is cautioned to remember that these represent a system flow less than that required for comfort control. This operating series of conditions represents control valve interaction. One valve flow affects flow to all other circuits between it and the pump. This interaction can be quite large. Depending on distribution system head loss and system balancing techniques, one valve may reduce system flow by two to three times the individual valve's rated flow. This can negatively affect the control and sequencing of the source, and possibly also the control of the affected terminal units. Controlling this interaction is done by engineering the hydraulic losses and distribution pipe friction head losses, using pressure-independent valves (both balancing and control), using sensors at critical points of the hydronic system, and combinations of these strategies.

These issues are part of what has caused debate over system balancing, pump method of control, etc. Simply put, system design dictates the requirements of adjustment and control. Balance of a system is more than having a balancing valve; it involves pipe selection and location combined with valves and coils and, in some cases, pumps, as well as measurement and adjustment devices. Similarly, the effects of one method of pump control over another (e.g., differential pressure control, valve position of pump speed reset) must also be considered. Traditional system design and feedback control techniques use a series of single-loop controllers. These techniques also work on variable-speed pumping systems, but the interaction of all of the individual controlled systems requires analysis by the designer and calculated design choices based on a thorough understanding of the loads, both theoretical and practical, and should also take into account that operators may run systems in a manner not intended by the designer.

Address these issues during design, because they are difficult to understand in the field, and the required instrumentation to monitor the effects is rarely installed. Despite these challenges, application of variable-speed, variable-flow pumping can be very satisfactory. However, variable-speed pumping designs require that the designer allow adequate time in calculation and design iteration to mitigate potential operation issues, and ensure the design criteria of a comfort providing system with operating energy and cost efficiency.

## Pump Connection

Pump suction piping should be at least as large as the nozzle serving the pump, and there should be minimal fittings or devices in the suction to obstruct flow. Typically, pump manufacturers prefer five to eight pipe diameters of unobstructed (i.e., no fittings) straight pipe entering the pump. Fittings such as tees and elbows, especially when there is a change in planar direction, cause water to swirl in the pipe; this can be detrimental to pump performance, and may also lead to pump damage. Manufacturers may recommend special fittings in applications where piping space is unavailable to overcome geometry factors. These fittings need to be carefully reviewed in application.

Piping to the pump should be independently supported, adding no load to the pump flanges, which, unless specifically designed for the purpose, are incapable of supporting the system piping. Supporting the pipe weight on the flange can cause serious damage (e.g., breaking the flange), or may induce stresses that misalign the pump. Similar results can also occur when the pipe and pump are improperly aligned to each other, or pipe expansion and contraction are unaccounted for. Flexible couplings are one way to overcome some of these issues, when both the pump and the pipe are supported independently of each other, and the flexible coupling is not arbitrarily connected to the pump and pipe.

When pumps are piped in parallel, these requirements are extended. Pump entering and discharge pressures should be equal in operation. In addition to maintaining the recommended straight inlet pipe to the pump, manifold pipe serving the inlets should also have a minimum of two manifold pipe diameters between pump suction center lines. Pump discharge manifolds should be constructed to keep discharge velocities less than 10 to 15 fps, and lower when a check valve is applied, because it is necessary to prevent hydraulic shock (water hammer). Soft-seating discharge check valves are required on the pumps to prevent reverse flow from one pump to another. Various specialty valves and fittings are available for serving one or more of these functions.

## Distribution System

The distribution system is the piping connecting the various other components of the system. The primary considerations in designing this system are (1) sizing the piping to handle the heating or cooling capacity required and (2) arranging the piping to ensure flow in the quantities required at design conditions and at all other loads.

The flow requirement of the pipe is determined by Equation (8) or (9). After  $\Delta t$  is established based on the thermal requirements, either of these equations (as applicable) can be used to determine the flow rate. First-cost economics and energy consumption make it advisable to design for the greatest practical  $\Delta t$  because the flow rate is inversely proportional to  $\Delta t$ ; that is, if  $\Delta t$  doubles, the flow rate is reduced by half.

The three related variables in sizing the pipe are flow rate, pipe size, and pressure drop. The primary consideration in selecting a design pressure drop is the relationship between the economics of first cost and energy costs.

Once the distribution system is designed, the pressure loss at design flow is calculated by the methods discussed in Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*. The relationship between flow rate and pressure loss can be expressed by

$$Q = C_v \sqrt{\Delta p} \quad (17)$$

where

$Q$  = system flow rate, gpm

$\Delta p$  = pressure drop in system, psi

$C_v$  = system constant (sometimes called valve coefficient, discussed in [Chapter 42](#))

Equation (17) may be modified as follows:

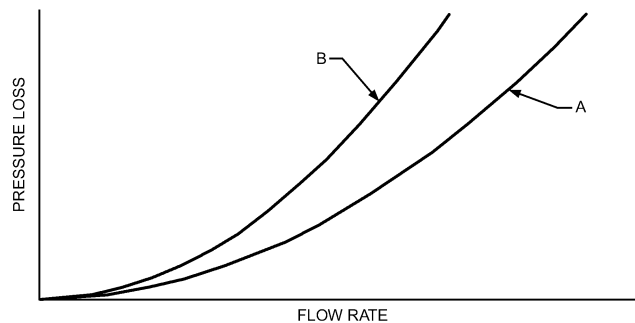


Fig. 14 Typical System Curves for Closed System

$$Q = C_s \sqrt{Dh} \quad (18)$$

where

$\Delta h$  = system head loss, ft of fluid [ $\Delta h = \Delta p / \rho$ ]  
 $C_s$  = system constant [ $C_s = 0.67 C_v$  for water with density  
 $\rho = 62.4 \text{ lb/ft}^3$ ]

Equations (17) and (18) are the system constant form of the Darcy-Weisbach equation. If the flow rate and head loss are known for a system, Equation (18) may be used to calculate the system constant  $C_v$ . From this calculation, the pressure loss can be determined at any other flow rate. Equation (18) can be graphed as a system curve (Figure 14).

The system curve changes if anything occurs to change the flow/pressure drop characteristics. Examples include a strainer that starts to block or a control valve closing, either of which increases the head loss at any given flow rate, thus changing the system curve in a direction from curve A to curve B in Figure 14.

This type of evaluation can help determine the effects of control and component selection in system operation. The designer cannot assume that the control system will compensate for poor selections and unnoticed installation mistakes. Anecdotal data suggest that numerous adjustments are required when selection and operation techniques are poor. There is also a need for system balance when the pump is correctly sized: no excess head is added to the pump; the balance device adds head only to the circuits for coils 1 and 2 to compensate for the difference in friction loss as water goes from one terminal takeoff to another. System flow is reduced to design, and no extra energy input is required for the pump.

## Expansion Chamber

As a hydraulic device, the expansion tank serves as the reference pressure point in the system, analogous to a ground in an electrical system (Lockhart and Carlson 1953). Where the tank connects to the piping, the pressure equals the pressure of the air in the tank plus or minus any fluid pressure caused by the elevation difference between the tank liquid surface and the pipe (Figure 15).

A closed system should have only one expansion chamber. Having more than one chamber or excessive amounts of undissolved air in a piping system can cause the closed system to behave in unintended (but understandable) ways, causing extensive damage from shock waves or water hammer.

With a single chamber on a system, assuming isothermal conditions for the air, the air pressure can change only as a result of displacement by the water. The only thing that can cause the water to move into or out of the tank (assuming no water is being added to or removed from the system) is expansion or shrinkage of the water in the system. Thus, in sizing the tank, thermal expansion is related to the pressure extremes of the air in the tank [Equations (13), (14), and (15)].

The point of connection of the tank should be based on the pressure requirements of the system, remembering that the pressure at

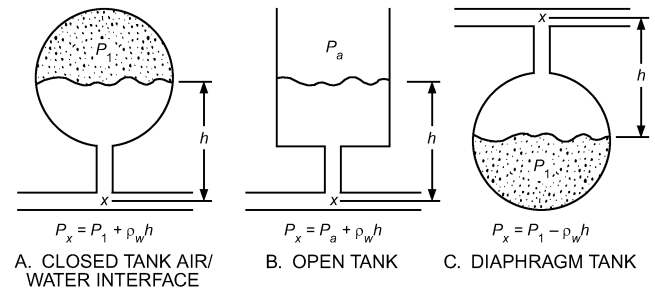


Fig. 15 Tank Pressure Related to System Pressure

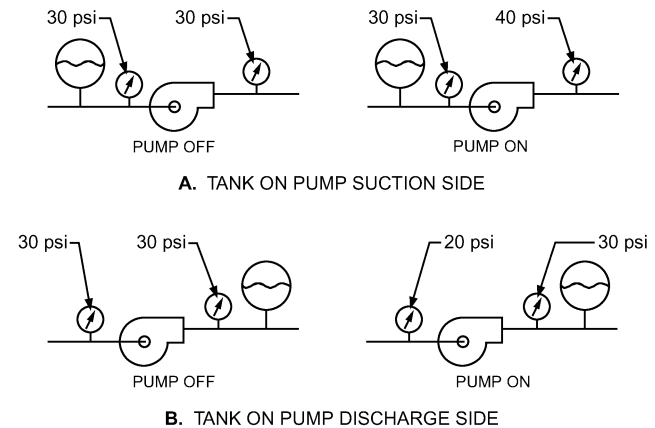


Fig. 16 Effect of Expansion Tank Location with Respect to Pump Pressure

the tank connection will not change as the pump is turned on or off. For example, consider a system containing an expansion tank at 30 psig and a pump with a pump head of 23.1 ft (10 psig). Figure 16 shows alternative locations for connecting the expansion tank; in either case, with the pump off, the pressure will be 30 psig on both the pump suction and discharge. With the tank on the pump suction side, when the pump is turned on, the pressure increases on the discharge side by an amount equal to the pump pressure (Figure 16A). With the tank on the discharge side of the pump, the pressure decreases on the suction side by the same amount (Figure 16B).

Other tank connection considerations include the following:

- A tank open to the atmosphere must be located above the highest point in the system, or be equipped with pressure-sustaining valves (as used in thermal storage applications).
- A tank with an air/water interface is generally used with an air control system that continually vents air into the tank. For this reason, it should be connected at a point where air can best be released.
- Within reason, the lower the pressure in a tank, the smaller the tank is [see Equations (13) and (15)]. Thus, in a vertical system, the higher the tank is placed, the smaller it can be.

## PIPING CIRCUITS

Hydronic systems are designed with many different configurations of piping circuits. In addition to simple preference by the design engineer, the method of arranging circuiting can be dictated by such factors as the shape or configuration of the building, the economics of installation, energy economics, the nature of the load, part-load capabilities or requirements, and others.

Each piping system is a network; the more extensive the network, the more complex it is to understand, analyze, or control. Thus, a major design objective is to maximize simplicity.

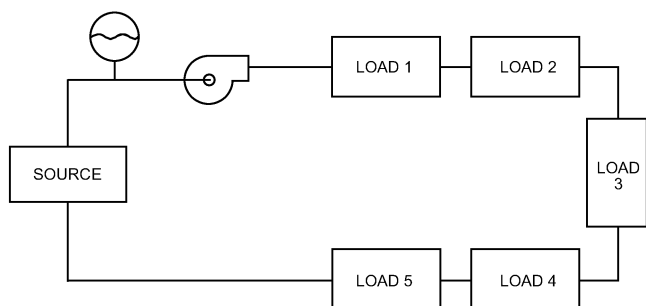


Fig. 17 Flow Diagram of Simple Series Circuit

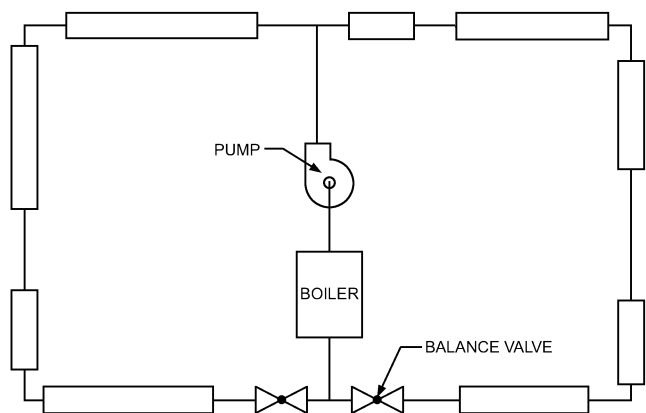


Fig. 18 Series Loop System

Load distribution circuits are of four general types:

- Full series
- Diverting series
- Parallel direct return
- Parallel reverse return

A simple **series** circuit is shown in Figure 17. Series loads generally have the advantages of lower piping costs and higher temperature drops, resulting in smaller pipe size and lower energy consumption. A disadvantage is that the different circuits cannot be controlled separately. Simple series circuits are generally limited to residential and small commercial standing radiation systems. Figure 18 shows a typical layout of such a system with two zones for residential or small commercial heating.

The simplest **diverting series** circuit diverts some flow from the main piping circuit through a special diverting tee to a load device (usually standing radiation) that has a low pressure drop. This system is generally limited to heating systems in residential or small commercial applications.

Figure 19 illustrates a typical one-pipe diverting tee circuit. For each terminal unit, a supply and a return tee are installed on the main. One of the two tees is a special diverting tee that creates pressure drop in the main flow to divert part of the flow to the unit. One (return) diverting tee is usually sufficient for upfeed (units above the main) systems. Two special fittings (supply and return tees) are usually required to overcome thermal pressure in downfeed units. Special tees are proprietary; consult the manufacturer's literature for flow rates and pressure drop data on these devices. Unit selection can be only approximate without these data.

One-pipe diverting series circuits allow manual or automatic control of flow to individual heating units. On/off rather than flow modulation control is preferred because of the relatively low pressure drop allowable through the control valve in the diverted flow circuit.

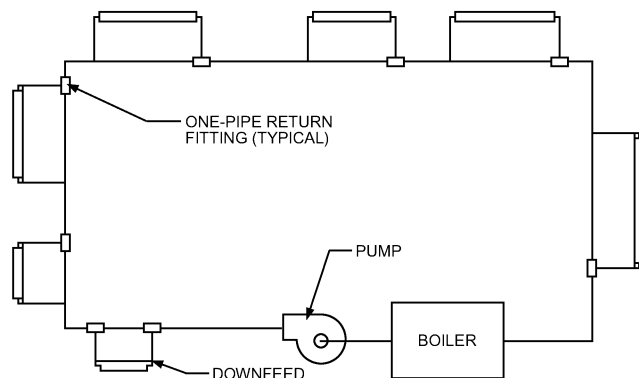


Fig. 19 One-Pipe Diverting Tee System

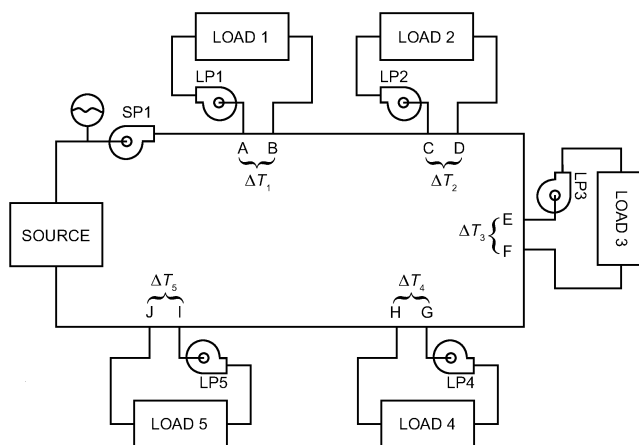


Fig. 20 Series Circuit with Load Pumps

This system is likely to cost more than the series loop because extra branch pipe and fittings, including special tees, are required. Each unit usually requires a manual air vent because of the low water velocity through the unit. The length and load imposed on a one-pipe circuit are usually small because of these limitations.

Because only a fraction of the main flow is diverted in a one-pipe circuit, the flow rate and pressure drop are less variable as water flow to the load is controlled than in some other circuits. When two or more one-pipe circuits are connected to the same two-pipe mains, the circuit flow may need to be mechanically balanced. After balancing, sufficient flow must be maintained in each one-pipe circuit to ensure adequate flow diversion to the loads.

When coupled with compound pumping systems, series circuits can be applied to multiple control zones on larger commercial or institutional systems (Figure 20). Note that in the series circuit with compound pumping, the load pumps need not be equal in capacity to the system pump. If, for example, load pump LP1 circulates less flow ( $Q_{LP1}$ ) than system pump SP1 ( $Q_{SP1}$ ), the temperature difference across Load 1 would be greater than the circuit temperature difference between A and B (i.e., water would flow in the common pipe from A to B). If, on the other hand, the load pump LP2 is equal in flow capacity to the system pump SP1, the temperature differentials across Load 2 and across the system from C to D would be equal and no water would flow in the common pipe. If  $Q_{LP3}$  exceeds  $Q_{SP1}$ , mixing occurs at point E and, in a heating system, the temperature entering pump LP3 would be lower than that available from the system leaving load connection D.

Thus, a series circuit using compound or load pumps offers many design options. Each of the loads shown in Figure 20 could also be a complete piping circuit or network.



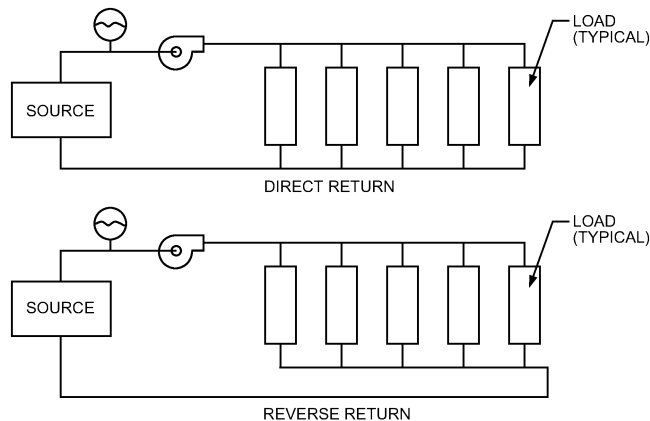


Fig. 21 Direct- and Reverse-Return Two-Pipe Systems

**Parallel piping** networks are the most commonly used in hydronic systems because they allow the same temperature water to be available to all loads. The two types of parallel networks are direct-return and reverse-return (Figure 21).

In the **direct-return** system, the length of supply and return piping through the subcircuits is unequal, which may cause unbalanced flow rates and require careful balancing to provide each subcircuit with design flow. Ideally, the **reverse-return** system provides nearly equal total lengths for all terminal circuits.

Direct-return piping has been successfully applied where the designer has guarded against major flow unbalance by

- Providing for pressure drops in the subcircuits or terminals that are significant percentages of the total, usually establishing pressure drops for close subcircuits at higher values than those for the far subcircuits
- Minimizing distribution piping pressure drop (in the limit, if the distribution piping loss is zero and the loads are of equal flow resistance, the system is inherently balanced)
- Including balancing devices and some means of measuring flow at each terminal or branch circuit
- Using control valves with a high head loss at the terminals

Carlson (1968) described the effects of distribution piping friction loss on total system flow to terminals in constant speed pumping systems through a **branch-to-riser pressure drop ratio (BRPDR)**. This concept helps minimize distribution losses. In constant-speed pumping systems, a BRPDR of 4:1 yielded a flow of 95% of design in all terminals at part-load conditions, 90% at 2:1, and 80% at 1:1. Application of this ratio helps alleviate system balancing problems without adjusting a controlling device. When used with variable-speed, variable-flow pumping systems, the control area can be minimized and control interaction can be alleviated. It also helps minimize overall friction losses, which reduces required pump horsepower and energy use.

### CAPACITY CONTROL OF LOAD SYSTEM

The two alternatives for controlling the capacity of hydronic systems are on/off control and variable-capacity or modulating control. The on/off option is generally limited to smaller systems (e.g., residential or small commercial) and individual components of larger systems. In smaller systems where the entire building is a single zone, control is accomplished by cycling the source device (the boiler or chiller) on and off. Usually a space thermostat allows the chiller or boiler to run, then a water temperature thermostat (aquastat) controls the capacity of the chiller(s) or boiler(s) as a function of supply or return water temperature. The pump can be either cycled with the load device (usually the case in a residential heating

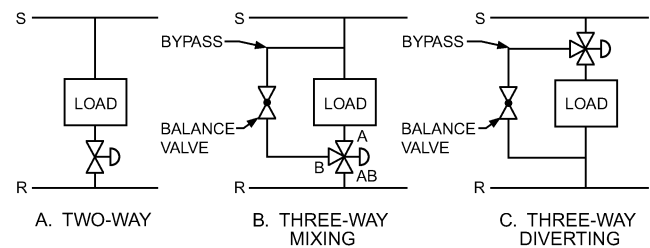


Fig. 22 Load Control Valves

system) or left running (usually done in commercial hot- or chilled-water systems).

In these single-zone applications, the piping design requires no special consideration for control. Where multiple zones of control are required, the various load devices are controlled first; then the source system capacity is controlled to follow the capacity requirement of the loads.

Control valves are commonly used to control loads. These valves control the capacity of each load by varying the amount of water flow through the load device when load pumps are not used. Control valves for hydronic systems are straight-through (two-way) valves and three-way valves (Figure 22). The effect of either valve is to vary the amount of water flowing through the load device.

With a two-way valve (Figure 22A), as the valve strokes from full-open to full-closed, the quantity of water flowing through the load gradually decreases from design flow to no flow. With a three-way mixing valve (Figure 22B) in one position, the valve is open from Port A to AB, with Port B closed off. In that position, all the flow is through the load. As the valve moves from the A-AB position to the B-AB position, some of the water bypasses the load by flowing through the bypass line, thus decreasing flow through the load. At the end of the stroke, Port A is closed, and all of the fluid flows from B to AB with no flow through the load. Thus, the three-way mixing valve has the same effect on the load as the two-way valve—as the load reduces, the quantity of water flowing through the load decreases.

The effect on load control with the three-way diverting valve (Figure 22C) is the same as with the mixing valve in a closed system: the flow is either directed through the load or through the bypass in proportion to the load. Because of the dynamics of valve operation, diverting valves are more complex in design and are thus more expensive than mixing valves; because they accomplish the same function as the simpler mixing valve, they are seldom used in closed hydronic systems.

In terms of load control, a two-way valve and a three-way valve perform identical functions: varying flow through the load as the load changes. The fundamental difference is that as the source or distribution system sees the load, the two-way valve provides a variable-flow load response and the three-way valve provides a constant-flow load response.

According to Equation (9), load  $q$  is proportional to the product of  $Q$  and  $\Delta t$ . Ideally, as the load changes,  $Q$  changes, while  $\Delta t$  remains fixed. However, as the system sees it, as the load changes with the two-way valve,  $Q$  varies and  $\Delta t$  is fixed, whereas with a three-way valve,  $\Delta t$  varies and  $Q$  is fixed. This principle is illustrated in Figure 23. Understanding this concept is fundamental to design or analysis of hydronic systems.

The flow characteristics of two- and three-way valve ports are described in Chapter 15 of the 2005 *ASHRAE Handbook—Fundamentals* and in Chapter 46 of this volume, and must be understood. The equal percentage characteristic is recommended for proportional control of load flow for two- and three-way valves; the bypass flow port of three-way valves should have the linear characteristic to maintain a uniform flow during part-load operation.

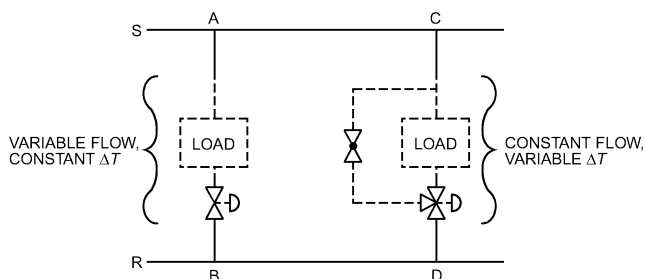


Fig. 23 System Flow with Two-Way and Three-Way Valves

### Sizing Control Valves

For stable control, the pressure drop in the control valve at the full-open position should be no less than one-third of the pump head, or controlled branch differential pressure in a variable speed pumping system. This is a simple rule of thumb recommendation, and caution should be used in application. Systems designed around closed-loop, constant-speed pumping systems with low-differential-temperature coils are considered forgiving in operation; as system complexity increases, however, the advanced strategies used require diligence and examination to evaluate valve performance and selection.

General pressure drops are commonly applied to valve sizing. Values chosen roughly correspond to the pressure loss on a coil, although there is no literature to explain why these values are used.

Using the concept of valve authority, the relationship is

$$\beta = \frac{\Delta P_{\text{valve}}}{\Delta P_{\text{system}}} \times 100 \quad (19)$$

where  $\beta$  is authority.

Common practice has been to assume that, if the coil and the valve have the same pressure drop, they have equal authority of 50%, and that stable control is established. This belief is a rule of thumb, and depends on the type of pumped system and intervening control actions. Using flow coefficient analysis, however, results in a slightly modified definition for authority, comparing the flow coefficient of the valve ( $C_v$ ) to the coefficient of the remaining system components ( $C_s$ ). The authority of 50%, then, is representative of the two components having equal flow coefficients, which implies that the valve pressure drop is much greater than the coil pressure drop.

Selection of the proper valve and its corresponding pressure drop requires analysis of the method of control to be used based on the desired response of the controlled system. Depending on the applied control theory and required level of performance, some systems may be run with an on/off control mode, in which case a two-position valve is used. The pressure drop applied on the valve may be minimal, as long as flow through the corresponding circuit is limited in some way to design flow and heat transfer flow tolerance.

More complex systems with fast time constants or that attempt to match seasonal production to load use modulating throttling valves. In these applications, heat transfer characteristic of the coil and the flow control characteristic of the installed control valve (under the effects of authority) and the desired controller gain are analyzed (Figures 24 and 25). The graphic solution of the two characteristics (coil and valve authority) has traditionally yielded a linear function so that the simple proportional controller could have a constant gain of one over the operation of the modulating control valve (Figures 26 to 28). This should not imply that this is the only method for sizing a valve, or that linearity is the only allowable control characteristic. Controllers represent devices implementing a mathematical argument. Knowledge of control theory or programming of different

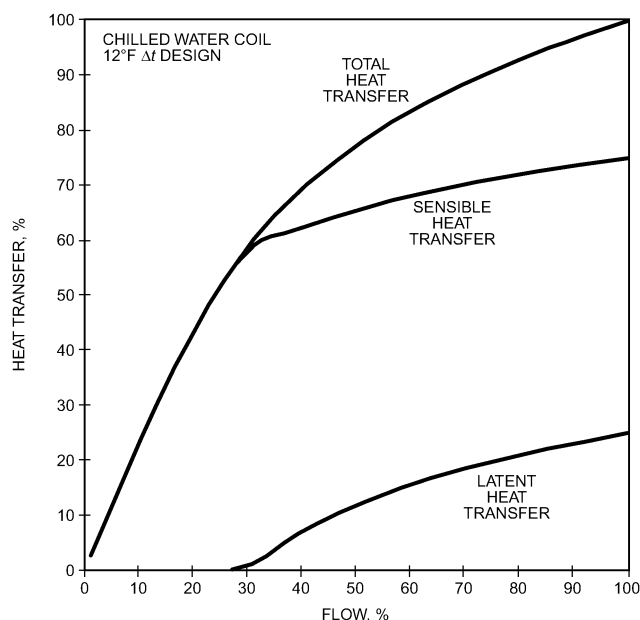


Fig. 24 Chilled-Water Coil Heat Transfer Characteristic

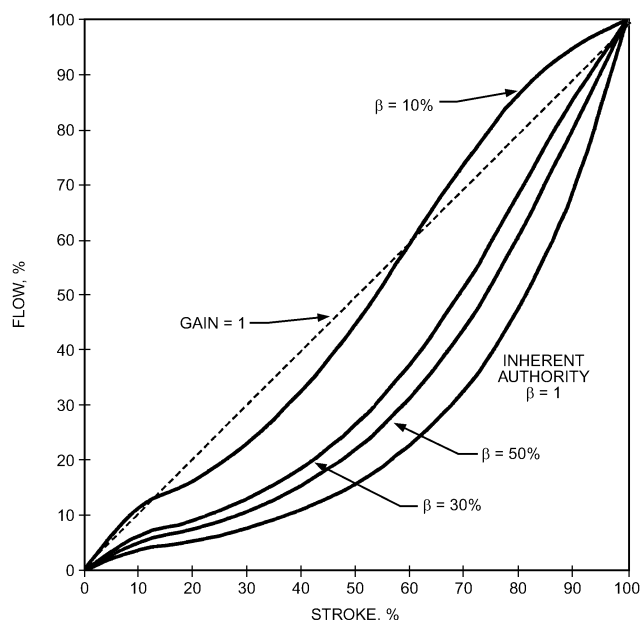


Fig. 25 Equal-Percentage Valve Characteristic with Authority

mathematical relationships is acceptable as long as all of the effects of such action are determined by the designer. However, linear control algorithms such as proportional with integral (PI) and proportional with integral and derivative (PID) control are the most common in HVAC systems, and should be considered in the application of the devices applied to the hydronic system.

Experience in of applied control systems shows the following general guidance.

**Proportional control** is adequate for slowly changing, single-variable systems such as space temperature control. Many of these applications may have slow response time, with time constants for temperature change on the order of 10 to 50 min or more, so a properly applied proportional controller with a properly applied valve and coil characteristic can perform adequately, and may approach

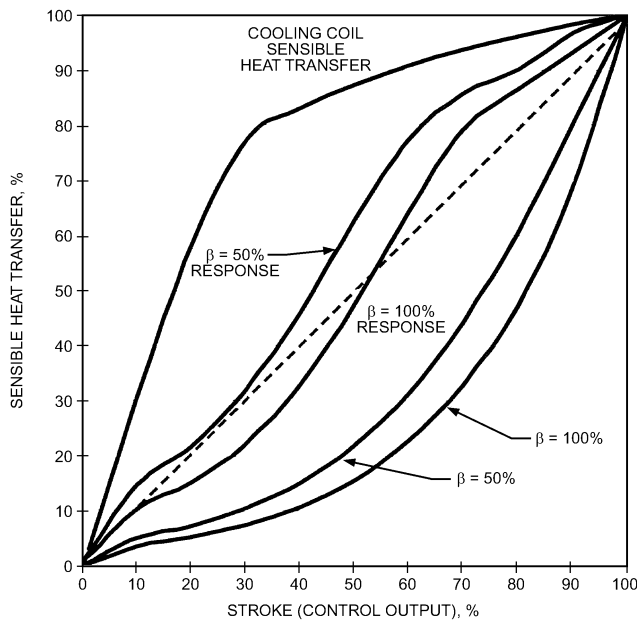


Fig. 26 Control Valve and Coil Response, Inherent and 50% Authority

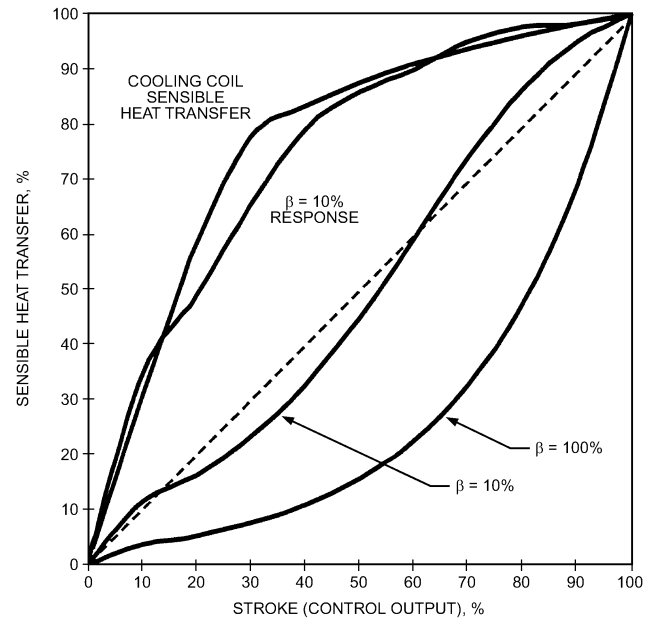


Fig. 28 Coil Valve and Coil Response, 10% Authority

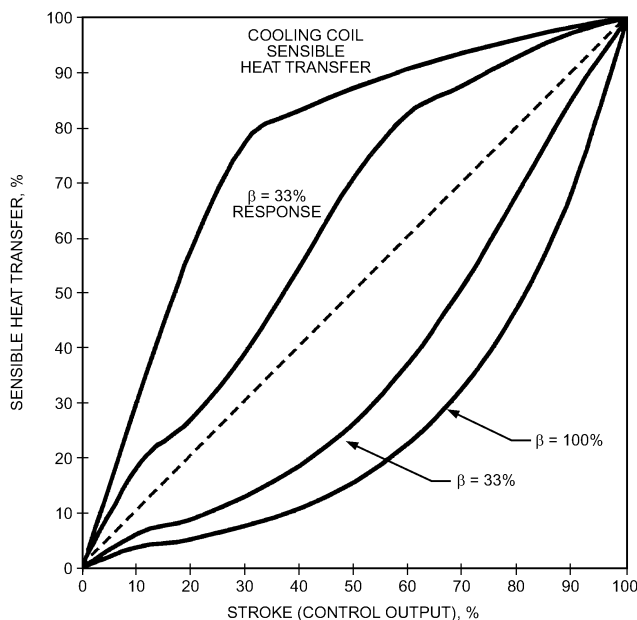


Fig. 27 Control Valve and Coil Response, 33% Authority

simple on/off or valve-open/valve-closed implementation with minimal noticeable temperature hunting in the space. There is a reasonable history of the application of PI control in this application, also. In systems with extra effects, such as variable-speed pumping systems, PI control may be necessary to deal with the control errors introduced when variables outside the direct control loop modify the valve. Note that applying variable-speed drives to pumps and blowers significantly affects the ability of a controlled device to achieve its design differential pressure, because there is an extra variable in the system, with corresponding downstream effects. The effects are the result of the changes that occur in piping system pressure distribution because of changes in flow and corresponding pressures, as affected by the control set point and measurement

responses in the pump controller. These effects can be significant or negligible, depending on how the piping system friction loss has been designed and distributed. Understanding variable-speed pump system control area and distribution of predicted dynamic flow in the building through energy analysis requires significant calculation, but may be necessary in certain applications.

Although **proportional-integral controllers** can be applied to simple systems, they are more often required for primary space and fluid conditioning systems such as discharge temperature control of an air-handling unit, or pressure/flow control of a pumping system. In practice, devices that exhibit fast response time and have low time constants are candidates for PI controller application, and possibly also for additional derivative control action. Generally, these control modes are applied through application of direct digital controls (DDC). Thorough understanding of the applied controller's implementation of the PID algorithm is required to understand the complete effects of each control action. In some cases, the cyclic nature of the generic DDC device or a specific manufacturer's feature may limit unwanted responses, or add stabilization to the output signal very different from the generic mathematical PID controller algorithm.

For example, in [Figure 23](#), the pressure drop at full-open position for the two-way valve should be great enough from A to B that, when the flow coefficient analysis is performed and combined with the coil characteristic ([Figure 24](#)) a straight line is developed ([Figures 25 and 26](#)). Typically, authorities in the range of 30 to 50% (for piping systems with 2:1 branch-to-riser pressure drop ratios) are adequate for HVAC coil applications, with higher  $\Delta t$  coils requiring less valve authority. Authorities less than 30% should be avoided because they lead to a flow characteristic shift that makes the equal-percentage valve appear to be more linear in nature ([Figures 27 and 28](#)). For the three-way valve shown in [Figure 23](#), the full-open pressure drop should be from C to D. The pressure drop in the bypass balancing valve in the three-way valve circuit should be set to equal that in the coil (load), [Chapter 46](#) discusses valve authority in three way modulating applications.

Control valves should be sized on the basis of the required valve coefficient  $C_v$  for the required pressure drop and flow. For more information, see the section on Control Valve Sizing under Automatic Valves in [Chapter 46](#). Briefly, in variable-speed pumping

systems using differential pressure control of the branch, the pressure drop of concern is the controlled differential, and is generally placed to measure the valve and coil pressure drops. The valve pressure drop for control, then, is that required to provide complementary characteristic. In the previous coils, a 12°F  $\Delta t$  was used, and valve authority of 50 to 100% provides good results, implying a pressure drop on the valve equal to that of the coil. Control valves that follow are thus dependent on the controlled hydraulic performance. In these valves, attention must also be paid to valve construction (especially the maximum pressure allowed on the valve body, and maximum allowed pressure drop across the valve).

If a system is to be designed with multiple zones of control such that load response is to be by constant flow through the load and variable  $\Delta t$ , control cannot be achieved by valve control alone; a load pump is required.

Several control arrangements of load pump and control valve configurations are shown in Figure 29. Note that, in all three configurations, the common pipe has no restriction or check valve. In all configurations, there is no difference in control as seen by the load. However, the basic differences in control are

- With the two-way valve configuration (Figure 29A), the distribution system sees variable flow and constant  $\Delta t$ , whereas with both three-way configurations, the distribution system sees constant flow and variable  $\Delta t$ .
- Configuration B differs from C in that the pressure required through the three-way valve in Figure 29B is provided by the load pump, whereas in Figure 29C it is provided by the distribution pump(s).

### Alternatives to Control Valves

Use of variable-frequency drives has greatly increased during the past few years because advances in technology and decreases in cost have made these drives an attractive alternative to using control valves for heat transfer control (Figure 30).

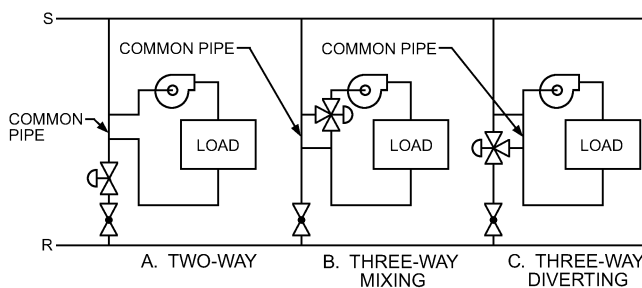


Fig. 29 Load Pumps with Valve Control

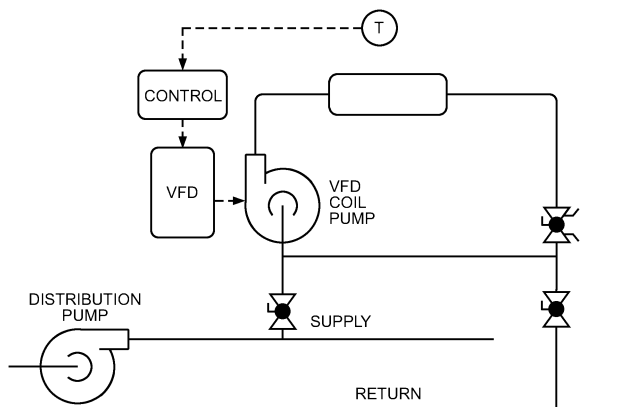


Fig. 30 Schematic of Variable-Speed Pump Coil Control

Green (1994) tested the control stability of variable-speed circulating pumps compared to control valves. Stability of the controlled discharge temperature of the coil was on the order of  $\pm 0.5^\circ\text{F}$ . Large coils with pump and separate drive may also be economical compared to similarly sized valves and actuators. The stability of on/off pump control was also found to be reasonable. The attraction of a pump over a control valve is the reduction of head and thus energy used by the distribution pumping system. For the valve to work properly, pump head must be throttled to control flow. Any throttling process is inherently energy-inefficient. Direct load control by a pump provides only the energy required to overcome the friction loss of the load heat transfer device and piping. Properly sequenced, the distribution pumping system can reduce flow, which causes higher coil entering water temperatures, which causes the circulating pump to operate at a higher flow. This is helpful in allowing for higher percentages of heat transfer capability of the sensible and latent loads through the coil. Applications of this control method must also take into account the potential of gravity flow in the piping scheme.

### LOW-TEMPERATURE HEATING SYSTEMS

These systems are used for heating spaces or processes directly, as with standing radiation and process heat exchangers, or indirectly, through air-handling unit coils for preheating, for reheating, or in hot-water unit heaters. They are generally designed with supply water temperatures from 180 to 240°F and temperature drops from 20 to 100°F.

In the United States, hot-water heating systems were historically designed for a 200°F supply water temperature and a 20°F temperature drop. This practice evolved from earlier gravity system designs and provides convenient design relationships for heat transfer coefficients related to coil tubing and finned-tube radiation and for calculations (1 gpm conveys 10,000 Btu/h at 20°F  $\Delta t$ ). Because many terminal devices still require these flow rates, it is important to recognize this relationship in selecting devices and designing systems.

However, the greater the temperature range (and related lower flow rate) that can be applied, the less costly the system is to install and operate. A lower flow rate requires smaller and less expensive piping, less secondary building space, and smaller pumps. Also, smaller pumps require less energy, so operating costs are lower.

### Nonresidential Heating Systems

Possible approaches to enhancing the economics of large heating systems include (1) higher supply temperatures, (2) primary-secondary pumping, and (3) terminal equipment designed for smaller flow rates. The three techniques may be used either singly or in combination.

Using higher supply water temperatures achieves higher temperature drops and smaller flow rates. Terminal units with a reduced heating surface can be used. These smaller terminals are not necessarily less expensive, however, because their required operating temperatures and pressures may increase manufacturing costs and the problems of pressurization, corrosion, expansion, and control. System components may not increase in cost uniformly with temperature, but rather in steps conforming to the three major temperature classifications. Within each classification, the most economical design uses the highest temperature in that classification.

Primary/secondary or compound pumping reduces the size and cost of the distribution system and also may use larger flows and lower temperatures in the terminal or secondary circuits. A primary pump circulates water in the primary distribution system while one or more secondary pumps circulate the terminal circuits. The connection between primary and secondary circuits provides complete hydraulic isolation of both circuits and allows a controlled interchange of water between the two. Thus, a high supply



water temperature can be used in the primary circuit at a low flow rate and high temperature drop, while a lower temperature and conventional temperature drop can be used in the secondary circuit(s).

For example, a system could be designed with primary-secondary pumping in which the supply temperature from the boiler was 240°F, the supply temperature in the secondary was 200°F, and the return temperature was 180°F. This design results in a conventional 20°F Δ*t* in the secondary zones, but allows the primary circuit to be sized on the basis of a 60°F drop. This primary-secondary pumping arrangement is most advantageous with terminal units such as convectors and finned radiation, which are generally unsuited for small flow rate design.

Many types of terminal heat transfer units are being designed to use smaller flow rates with temperature drops up to 100°F in low-temperature systems and up to 150°F in medium-temperature systems. Fan apparatus, the heat transfer surface used for air heating in fan systems, and water-to-water heat exchangers are most adaptable to such design.

A fourth technique is to put certain loads in series using a combination of control valves and compound pumping (Figure 31). In the system illustrated, the capacity of the boiler or heat exchanger is 2 × 10<sup>6</sup> Btu/h, and each of the four loads is 0.5 × 10<sup>6</sup> Btu/h. Under design conditions, the system is designed for an 80°F water temperature drop, and the loads each provide 20°F of the total Δ*t*. The loads in these systems, as well as the smaller or simpler systems in residential or commercial applications, can be connected in a direct-return or a reverse-return piping system. The different features of each load are as follows:

- The domestic hot-water heat exchanger has a two-way valve and is thus arranged for variable flow (the main distribution circuit provides constant flow for the boiler circuit).
- The finned-tube radiation circuit is a 20°F Δ*t* circuit with the design entering water temperature reduced to and controlled at 200°F.
- The reheat coil circuit takes a 100°F temperature drop for a very low flow rate.
- The preheat coil circuit provides constant flow through the coil to keep it from freezing.

When loads such as water-to-air heating coils in LTW systems are valve-controlled (flow varies), they have a heating characteristic of flow versus capacity as shown in Figure 32 for 20°F and 60°F temperature drops. For a 20°F Δ*t* coil, 50% flow provides approximately 90% capacity; valve control will tend to be unstable. For this reason, proportional temperature control is required, and equal percentage characteristic two-way valves should be selected such that

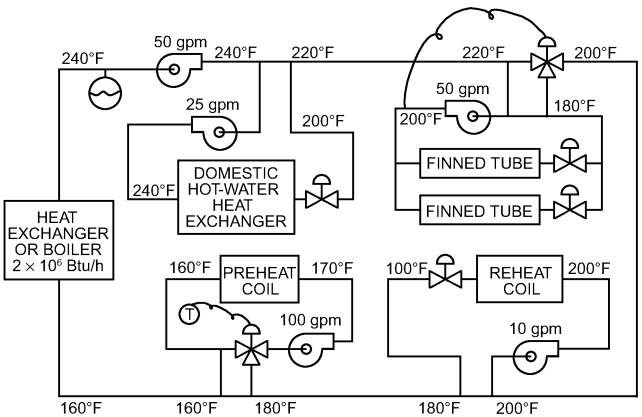


Fig. 31 Example of Series-Connected Loading

10% flow is achieved with 50% valve lift. This combination of the valve characteristic and the heat transfer characteristic of the coil makes control linear with respect to the control signal. This type of control can be obtained only with equal percentage two-way valves and can be further enhanced if piped with a secondary pump arrangement as shown in Figure 29A. See Chapter 46 of the 2007 ASHRAE Handbook—HVAC Applications for further information on automatic controls.

CHILLED-WATER SYSTEMS

Designers have less latitude in selecting supply water temperatures for cooling applications because there is only a narrow range of water temperatures low enough to provide adequate dehumidification and high enough to avoid chiller freeze-up. Circulated water quantities can be reduced by selecting proper air quantities and heat transfer surface at the terminals. Terminals suited for a 16 to 20°F rise rather than a 10 to 12°F rise reduce circulated water quantity and pump power by one-third and increase chiller efficiency.

A proposed system should be evaluated for the desired balance between installation cost and operating cost. Table 1 shows the effect of coil circuiting and chilled-water temperature on water flow and temperature rise. This yields a coil characteristic not unlike the heating coil shown in Figure 33. The characteristic, though, should be shown as total, sensible, and latent heat transfer. The coil rows, fin

Table 1 Chilled-Water Coil Performance

Coil Circuiting	Chilled-Water Inlet Temp., °F	Coil Pressure Drop, psi	Chilled-Water Flow, gpm/ton	Chilled-Water Temp. Rise, °F
Full <sup>a</sup>	45	1.0	2.2	10.9
Half <sup>b</sup>	45	5.5	1.7	14.9
Full <sup>a</sup>	40	0.5	1.4	17.1
Half <sup>b</sup>	40	2.5	1.1	21.8

Note: Table is based on cooling air from 81°F db, 67°F wb to 58°F db, 56°F wb.  
<sup>a</sup> Full circuiting (also called single circuit). Water at the inlet temperature flows simultaneously through all tubes in a plane transverse to airflow; it then flows simultaneously through all tubes, in unison, in successive planes (i.e., rows) of the coil.  
<sup>b</sup> Half circuiting. Tube connections are arranged so there are half as many circuits as there are tubes in each plane (row), thereby using higher water velocities through the tubes. This circuiting is used with small water quantities.

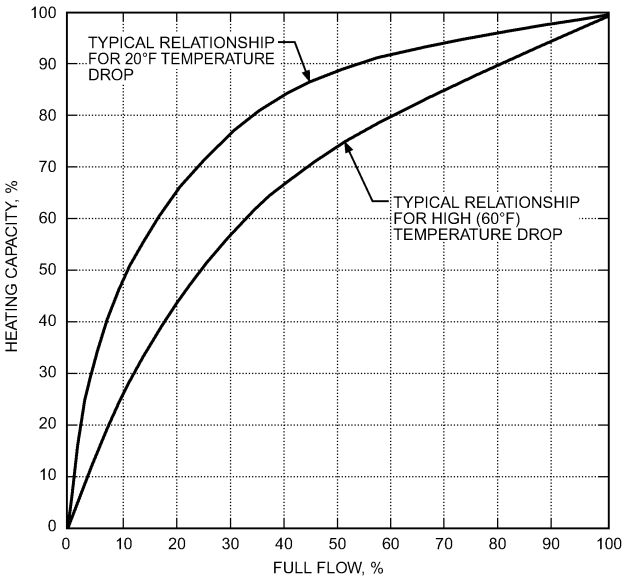


Fig. 32 Heat Emission Versus Flow Characteristic of Typical Hot Water Heating Coil

spacing, air-side performance, and cost are identical for all selections. Morabito (1960) showed how such changes in coil circuiting affect the overall system. Considering the investment cost of piping and insulation versus the operating cost of refrigeration and pumping motors, higher temperature rises, (i.e., 16 to 24°F temperature rise at about 1.0 to 1.5 gpm per ton of cooling) can be applied on chilled-water systems with long distribution piping runs; larger flow rates should be used only where reasonable in close-coupled systems.

For the most economical design, the minimum flow rate to each terminal heat exchanger is calculated. For example, if one terminal can be designed for an 18°F rise, another for 14°F, and others for 12°F, the highest rise to each terminal should be used, rather than designing the system for an overall temperature rise based on the smallest capability.

The control system selected also influences the design water flow. The expense of operating chilled-water systems, combined with the complexity of interrelated system variables, demands direct digital control. Evaluation suggests that, in many cases, a minimum of PI control is required, if not PID. To control flow, the designer should apply two-way modulating control valves, matching the required characteristic to the coil's operational characteristic. The control system programmer may need some method of linearizing through programming the controller output to the combined characteristic of the control valve and the selected coil.

For systems with multiple terminal units, use care in applying diversity factors, which can negatively affect not only chiller selection, but also pump selection. Designers should apply variable-speed pumping techniques to the system pumps. System balancing is best accomplished with automatic flow-limiting valves, with the pumps capable of either providing the coil block flow load, or limiting the stroke of the control valves in conditions where single-loop controllers would tend to open all valves at once (e.g., at start-up). This is necessary to prevent a no-flow condition from occurring in coils farther from the pump.

A primary consideration with chilled-water system design is the control of the source systems at reduced loads. The constraints on the temperature parameters are (1) a water freezing temperature of 32°F, (2) economics of the refrigeration system in generating chilled water, and (3) the dew-point temperature of the air at nominal indoor comfort conditions (55°F dew point at 75°F and 50% rh).

These parameters have led to the common practice of designing for a supply chilled-water temperature of 42 to 45°F and a return water temperature between 55 and 66°F. However, Pederson et al.

(1998) suggest an alternative classic design methodology for selecting entering water temperature to the chilled-water coil, based on the coil's dew-point temperature.

Final selection of the design criteria should be judged against the required flow tolerance of the coil to the required heat transfer characteristics of the coil. Carlson (1981) notes relationships for entering and differential temperatures of the coil as it compares to providing 97% of design heat transfer, and compares the required flow as a percentage of design for the purpose of specifying hydronic system balancing, and selection of the distribution system friction loss.

With increasing entering water temperatures and selection differential temperatures, the coil characteristic becomes very sensitive to changes in flow at the design condition. Carlson (1981) showed that flow tolerance to the coil could be as high as  $\pm 20\%$  flow for low  $\Delta t$  designs, but the tolerance became much closer ( $\pm 5\%$  flow or less) for coils with higher differential-temperature designs (Figure 34). The concern was providing adequate flow for adequate coil heat transfer at a design condition, without grossly overflowing the coil. Carlson (1981) suggested a maximum allowable flow tolerance of  $\pm 10\%$  design flow at design flow conditions to enhance operating system energy efficiency. Many devices are available to help achieve this. However, because coil heat transfer sensitivity to flow increases as water-side differential temperature increases, designers must weigh coil design criteria, pipe sizing decisions, and balancing interactions. From a practical perspective, many balancing devices can achieve  $\pm 10\%$  flow adjustment or better, and in some cases meet or exceed  $\pm 5\%$  flow adjustment. However, as Figure 34 indicates, changes in water-side design  $\Delta t$  and entering air conditions to the coil offer the potential for losing some high-flow heat transfer if design flows cannot be accomplished. This requires that flow be balanced as closely to design as possible so the required heat transfer for design condition can be met. If the designer has concerns

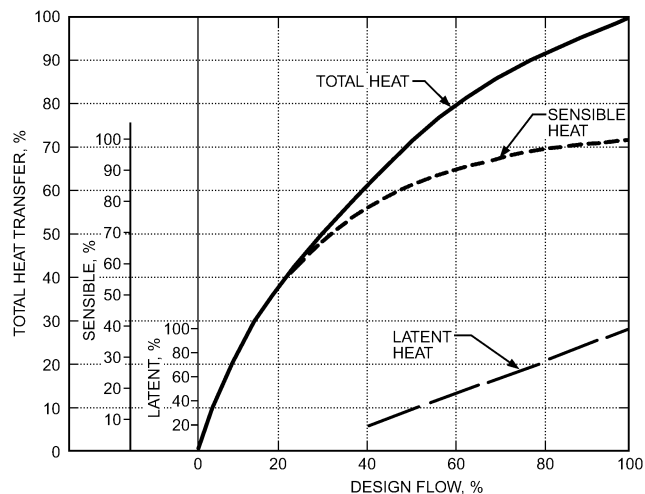


Fig. 33 Generic Chilled-Water Coil Heat Transfer Characteristic

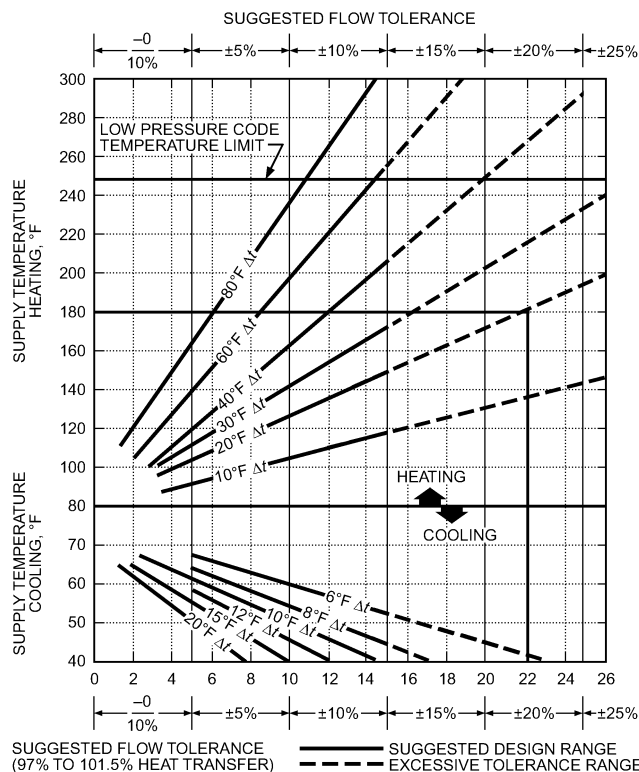


Fig. 34 Recommendations for Coil Flow Tolerance to Maintain 97% Design Heat Transfer (Carlson 1981)

with temperature and humidity control in the space, the maximum potential operating energy efficiency of the system (characterized by higher coil design differential temperatures and lower flows) must be balanced with the required comfort conditions of the space and the ability to control them based on the operating interactions of all system components (pipe, valve, fitting friction loss, coil operation, and control system sensitivity to adjustment). Care must be taken by the designer to achieve the desired operation. Combined with energy-sensible concepts such as variable-speed, variable-flow pumping and the potential flow interaction of the control valve, the designer must carefully analyze the system loads and consider all system effects. Figure 34 shows one approach for of this type of analysis. However, it shows that heat transfer, and potentially comfort, is affected if flow to a coil varies too much. System designers should consider similar types of analysis based on their selected design parameters.

Figure 35 shows a typical configuration of small chilled-water systems, using two parallel chillers and loads with three-way valves. Note that flow should be essentially constant, although the valve characteristic and authority could cause more or less than design flow at part-load (valve stroke) conditions. A simple energy balance [Equation (9)] dictates that, with a constant flow rate, at one-half of design load, the water temperature differential drops to one-half of design. At this load, if one of the chillers is turned off, the return water circulating through the *off* chiller mixes with the supply water. This mixing raises the temperature of the supply chilled water and can cause a loss of control if the designer does not consider this operating mode. A better approach is to use a modulated system instead, to enhance operating efficiency and meet the requirements of building energy codes as presented in ASHRAE *Standard* 90.1.

A typical configuration of a large chilled-water system with multiple chillers and loads and compound piping is shown in Figure 36. This system provides variable flow, essentially constant-supply-temperature chilled water, multiple chillers, more stable two-way

control valves, and the advantage of adding chilled-water storage with little additional complexity. As mentioned, mixing in the transition of chillers can be of concern, but using variable-speed drives can help eliminate the issue while keeping the benefit of hydraulic organization.

One design issue illustrated in Figure 36 is the placement of the common pipe for the chillers. With the common pipe as shown, the chillers unload from left to right. With the common pipe in the alternative location shown, the chillers unload equally in proportion to their capacity (i.e., equal percentage).

The **one-pipe chilled-water system**, also called the **integrated decentralized chilled-water system**, is another system that has seen considerable use in campus-type chilled-water systems with multiple chillers and multiple buildings (Coad 1976). A single pumped main circulates water in a closed loop through all the connected buildings. Each load and/or chiller is connected to the loop, with the chillers usually downstream from a load connection. The loop capacity is limited only by the fact that the flow capacity for any single load or chiller connection cannot exceed the flow rate of the loop. Because the loads are in series, the cooling coils must be sized for higher entering water temperatures than are normally used.

## DUAL-TEMPERATURE SYSTEMS

Dual-temperature systems are used when the same load devices and distribution systems are used for both heating and cooling (e.g., fan-coil units and central station air-handling unit coils). In dual-temperature system design, the cooling cycle design usually dictates the requirements of the load heat exchangers and distribution systems. Dual-temperature systems are basically of three different configurations, each requiring different design techniques:

1. Two-pipe systems
2. Four-pipe common load systems
3. Four-pipe independent load systems

### Two-Pipe Systems

In a dual-temperature two-pipe system, the load devices and the distribution system circulate chilled water when cooling is required and hot water when heating is required (Figure 37). Design considerations for these systems include the following:

- Loads must all require cooling or heating coincidentally; that is, if cooling is required for some loads and heating for other loads at a given time, this type of system should not be used.
- When designing the system, the flow and temperature requirements for both cooling and heating media must be calculated first. The load and distribution system should be designed for the more stringent, and the water temperatures and temperature differential should be calculated for the other mode.
- Changeover should be designed such that the chiller evaporator is not exposed to damaging high water temperatures and the boiler is not subjected to damaging low water temperatures. To accom-

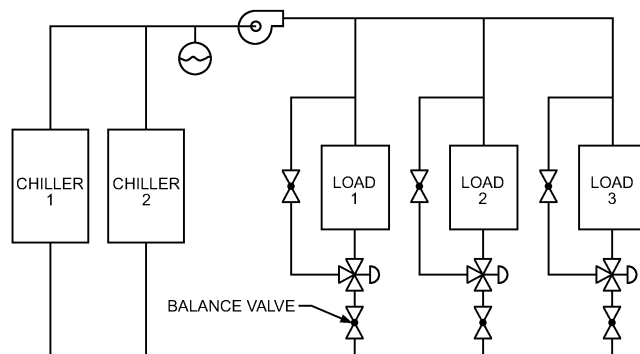


Fig. 35 Constant-Flow Chilled-Water System

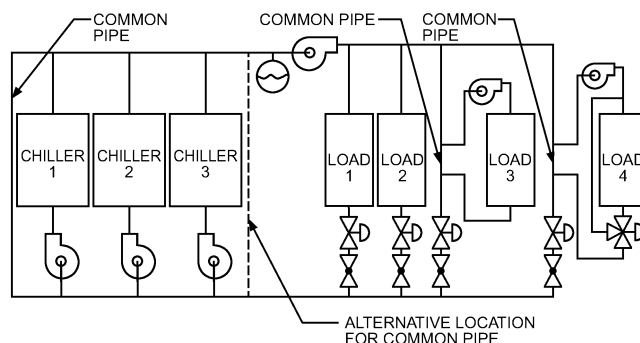


Fig. 36 Variable-Flow Chilled-Water System

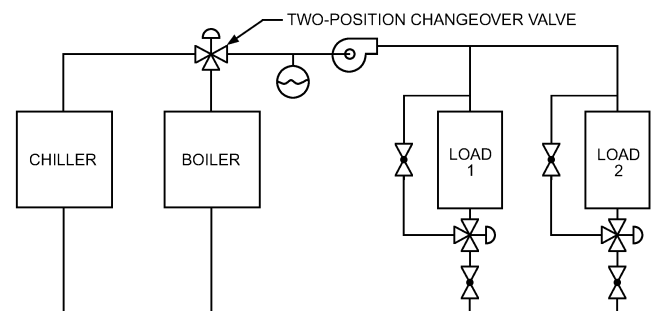


Fig. 37 Simplified Diagram of Two-Pipe System

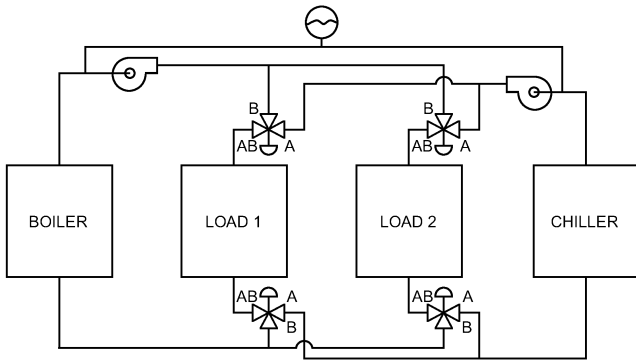


Fig. 38 Four-Pipe Common Load System

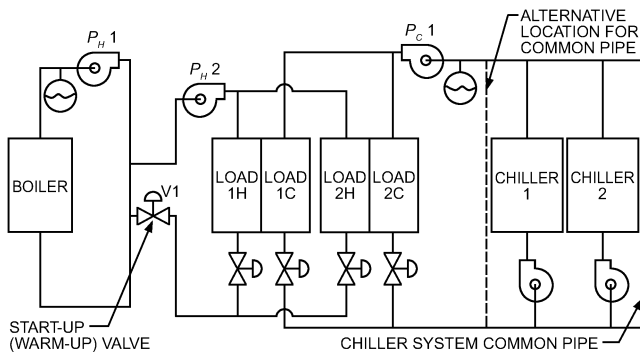


Fig. 39 Four-Pipe Independent Load System

modate these limiting requirements, changeover of a system from one mode to the other requires considerable time. If rapid load swings are anticipated, a two-pipe system should not be selected, although it is the least costly of the three options.

#### Four-Pipe Common Load Systems

In the four-pipe common load system, load devices are used for both heating and cooling as in the two-pipe system. The four-pipe common load system differs from the two-pipe system in that both heating and cooling are available to each load device, and changeover from one mode to the other takes place at each individual load device, or grouping of load devices, rather than at the source. Thus, some load systems can be in cooling mode while others are in heating mode. Figure 38 is a flow diagram of a four-pipe common load system, with multiple loads and a single boiler and chiller.

Many of these systems were installed and did not perform successfully because of problems in implementing the design concepts, and leakage through the control valves. These systems are not allowed for new application in energy standards such as ASHRAE 90.1, and are mentioned here as historical reference.

#### Four-Pipe Independent Load Systems

The four-pipe independent load system is preferred for hydronic applications in which some loads are in heating mode while others are in cooling mode. Control is simpler and more reliable than for the common load systems and, in many applications, the four-pipe independent load system is less costly to install. Also, flow through the individual loads can be modulated, providing both the control capability for variable capacity and the opportunity for variable flow in either or both circuits.

A simplified example of a four-pipe independent load system with two loads, one boiler, and two chillers is shown in Figure 39. Note that both hydronic circuits are essentially independent, so that each can be designed with disregard for the other system.

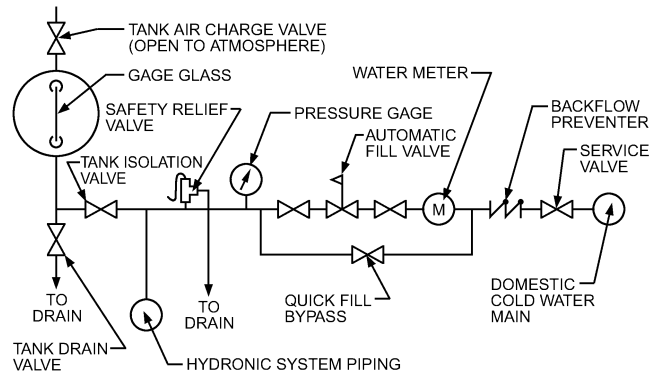


Fig. 40 Typical Makeup Water and Expansion Tank Piping Configuration for Plain Steel Expansion Tank

Although both circuits in the figure are shown as variable-flow distribution systems, they could be constant-flow (three-way valves) or one variable and one constant. Generally, the control modulates the two load valves in sequence with a dead band at the control midpoint.

This type of system offers additional flexibility when some selective loads are arranged for heating only or cooling only, such as unit heaters or preheat coils. Then, central station systems can be designed for humidity control with reheat through configuration at the coil locations and with proper control sequences.

### OTHER DESIGN CONSIDERATIONS

#### Makeup and Fill Water Systems

Generally, a hydronic system is filled with water through a valved connection to a domestic water source, with a service valve, a backflow preventer, and a pressure gage. (The domestic water source pressure must exceed the system fill pressure.)

Because the expansion chamber is the reference pressure point in the system, the water makeup point is usually located at or near the expansion chamber.

Many designers prefer to install automatic makeup valves, which consist of a pressure-regulating valve in the makeup line. However, the quantity of water being made up must be monitored to avoid scaling and oxygen corrosion in the system.

#### Safety Relief Valves

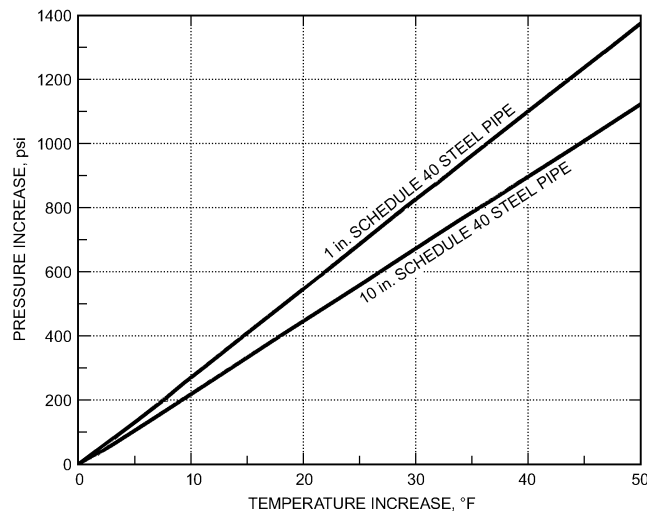
Safety relief valves should be installed at any point at which pressures can be expected to exceed the safe limits of the system components. Causes of excessive pressures include

- Overpressurization from fill system
- Pressure increases caused by thermal expansion
- Surges caused by momentum changes (shock or water hammer)

Overpressurization from the fill system could occur because of an accident in filling the system or the failure of an automatic fill regulator. To prevent this, a safety relief valve is usually installed at the fill location. Figure 40 shows a typical piping configuration for a system with a plain steel or air/water interface expansion tank. Note that no valves are installed between the hydronic system piping and the safety relief valve. This is a mandatory design requirement if the valve in this location is also to serve as a protection against pressure increases due to thermal expansion.

An expansion chamber is installed in a hydronic system, to allow for the volumetric changes that accompany water temperature changes. However, if any part of the system is configured such that it can be isolated from the expansion tank and its temperature can increase while it is isolated, then overpressure relief should be provided.





**Fig. 41 Pressure Increase Resulting from Thermal Expansion as Function of Temperature Increase**

The relationship between pressure change caused by temperature change and the temperature change in a piping system is expressed by the following equation:

$$\Delta p = \frac{(\beta - 3\alpha)\Delta t}{(5/4)(D/E \Delta r) + \gamma} \quad (20)$$

where

- $\Delta p$  = pressure increase, psi
- $\beta$  = volumetric coefficient of thermal expansion of water,  $1/^\circ\text{F}$
- $\alpha$  = linear coefficient of thermal expansion for piping material,  $1/^\circ\text{F}$
- $\Delta t$  = water temperature increase,  $^\circ\text{F}$
- $D$  = pipe diameter, in.
- $E$  = modulus of elasticity of piping material, psi
- $\gamma$  = volumetric compressibility of water,  $\text{in}^3/\text{lb}$
- $\Delta r$  = thickness of pipe wall, in.

Figure 41 shows a solution to Equation (23) demonstrating the pressure increase caused by any given temperature increase for 1 in. and 10 in. steel piping. If the temperature in a chilled water system with piping spanning sizes between 1 and 10 in. were to increase by  $15^\circ\text{F}$ , the pressure would increase between 340 and 420 psi, depending on the average pipe size in the system.

Safety relief should be provided to protect boilers, heat exchangers, cooling coils, chillers, and the entire system when the expansion tank is isolated for air charging or other service. As a minimum, the ASME *Boiler and Pressure Vessel Code* requires that a dedicated safety relief valve be installed on each boiler and that isolating or service valves be provided on the supply and return connections to each boiler.

Potential forces caused by shock waves or water hammer should also be considered in design. Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals* discusses the causes of shock forces and the methodology for calculating the magnitude of these forces.

## Air Elimination

If air and other gases are not eliminated from the flow circuit, they may slow or stop the flow through the terminal heat transfer elements and cause corrosion, noise, reduced pumping capacity, and loss of hydraulic stability (see the section on Principles at the beginning of the chapter). A closed tank without a diaphragm can be installed at the point of the lowest solubility of air in water. With a diaphragm tank, air in the system can be removed by an air separator and air elimination valve installed at the point of lowest solubility.

This type of system is called an air elimination system. Manual vents should be installed at high points to remove all air trapped during initial operation. Shutoff valves should be installed on any automatic air removal device to allow servicing without draining the system.

Air elimination devices are most effective at low velocities. Thus, the pipe leading up to the air elimination device often is smaller than the device piping, and this size difference should be accounted for in the design. Alternatively, with variable-speed pumping systems, the air elimination device could be commissioned at a flow rate less than full design flow, allowing for a lower entering velocity to the device. After commissioning, when air has been purged from the system, the device can be operated at the higher flow. This is less effective for air removal, but air is unlikely to reenter the closed-loop pumping system.

Standard steel vessels used as system compression tanks are air management systems. An air separator installed at the point of lowest solubility collects air recovered from the system and transfers it to the compression tank. The tank must have a monolithic tank fitting, allowing water and air to enter and leave the tank, and preventing gravity flow between the system and the tank. Automatic air removal devices should never be used in these systems, because collected system air provides the gas cushion for water system expansion and contraction.

## Drain and Shutoff

All low points should have drains. Separate shutoff and draining of individual equipment and circuits should be possible so that the entire system does not have to be drained to service a particular item. Whenever a device or section of the system is isolated and water in that section or device could increase in temperature following isolation, overpressure safety relief protection must be provided.

## Balance Fittings

Balance fittings or valves and a means of measuring flow quantity should be applied as needed to allow balancing of individual terminals and subcircuits. Balance, however, cannot be achieved through fittings or devices alone. In a balanced system, at design flow conditions, all terminals receive as a minimum enough flow to create the required design heat transfer. Carlson (1968, 1981) suggested that 97% heat transfer was a reasonable value for required heat transfer accuracy, implying a flow tolerance to the coil based on heat transfer. On a  $200^\circ\text{F}$  entering water temperature (EWT) hot-water coil with a  $20^\circ\text{F}$   $\Delta t$ , flow could be  $\pm 25\%$  to the coil, and the required heat transfer would be achieved. However, as  $\Delta t$  and coil EWT increase, this flow tolerance decreases. Carlson suggested that, for system efficiency, circuits should be balanced to  $\pm 10\%$  flow, which is commonly accepted. However, Carlson also noted that, for some systems, flow tolerance is tighter ( $\pm 5\%$  or, in some cases,  $+10\%$  and  $0\%$ ), particularly in cases where the log mean temperature difference (LMTD) of the coil is limited (e.g., in chilled-water systems) and flow tolerance becomes an issue in proper system operation. One method Carlson suggested to overcome these issues in constant-speed pumping systems was to use a branch-to-riser pressure drop ratio as a criterion for selecting distribution system friction loss. A 4:1 ratio allowed 95% of design flow to always reach the terminal, a 2:1 ratio yielded 90% flow, and 1:1 yielded 80% flow. In variable-speed pumping systems, this concept is helpful to analyze, but yields different results. Regardless, system distribution piping losses should be analyzed with respect to their effects on system flow, balance, and control valve operation, which leads to control stability. Application of flow measurement fittings requires stabilized pipe flow, and is affected by the location and planar geometry of the device to fittings both up- and downstream. In general, allow reasonable straight pipe lengths (anything from a few pipe diameters to 20 diameters) before flow-measurement devices.

## Pitch

Piping need not pitch but can run level, providing that flow velocities exceeding 1.5 fps are maintained or a diaphragm tank is used.

## Strainers

Strainers should be used where necessary to protect system elements, and sparingly to enhance energy efficiency. Strainers in the pump suction must be checked carefully to avoid pump cavitation. Designers should consider strainer designs that can trap particles during the commissioning (flush and clean) phase, and allow more particle trap size modification after commissioning, to reduce system pressure loss. Large separating chambers can serve as main air venting points and dirt strainers ahead of pumps. Automatic control valves or other devices operating with small clearances require protection from pipe scale, gravel, and welding slag, which may readily pass through the pump and its protective separator. Individual fine mesh strainers may therefore be required ahead of each control valve. An alternative is to use two manual three-way valves entering a coil with connected bypasses. During commissioning, all coils are isolated from the system, with the three-way valve bypasses connected, allowing flushing water to serve main distribution piping to the coil, but not carrying particulates through the coil. After piping is flushed and cleaned, the valves are positioned to allow flow to and from the coil. This method removes most system particulates, thus protecting the coil, and, if individual coil strainers are used, greatly reduces labor in flushing each individual strainer.

## Thermometers

Thermometers or thermometer wells should be installed to assist the system operator in routine operation and troubleshooting. Permanent thermometers, with the correct scale range and separate sockets, should be used at all points where temperature readings are regularly needed. Thermometer wells should be installed where readings will be needed only during start-up and infrequent troubleshooting. If a central monitoring system is provided, a calibration well should be installed adjacent to each sensing point in insulated piping systems.

## Flexible Connectors and Pipe Expansion Compensation

Flexible connectors are sometimes installed at pumps and machinery to reduce pipe stress. See Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* for vibration isolation information. Expansion, flexibility, and hanger and support information is in [Chapter 45](#) of this volume. Piping systems should be supported independently of flexible connections.

## Gage Cocks

Gage cocks or quick-disconnect test ports should be installed at points requiring pressure or temperature readings. Gages permanently installed in the system will deteriorate because of vibration and pulsation and may become unreliable. It is good practice to install gage cocks and provide the operator with several high-quality gages for diagnostic purposes. Avoid overuse of gage cocks and test ports, because they represent points of potential system leakage. In general, one port entering a coil and leaving the coil is adequate, and can be combined with other functions to produce data required to verify system operation.

## Insulation

Insulation should be applied to minimize pipe thermal loss and to prevent condensation during chilled-water operation (see Chapter 24 of the 2005 *ASHRAE Handbook—Fundamentals*). On chilled-water systems, special rigid metal sleeves or shields should be installed at all hanger and support points, and all valves should be provided with extended bonnets to allow for the full insulation thickness without interference with the valve operators.

## Condensate Drains

Condensate drains from dehumidifying coils should be trapped and piped to an open-sight plumbing drain. Traps should be deep enough to overcome the air pressure differential between drain inlet and room, which ordinarily will not exceed 2 in. of water. Pipe should be noncorrosive and insulated to prevent moisture condensation. Depending on the quantity and temperature of condensate, plumbing drain lines may require insulation to prevent sweating.

## Common Pipe

In compound (primary-secondary) pumping systems, the common pipe is used to dynamically decouple the two pumping circuits. Ideally, there is no pressure drop in this section of piping; however, in actual systems, it is recommended that this section of piping be a minimum of 10 diameters in length to reduce the likelihood of unwanted mixing resulting from velocity (kinetic) energy or turbulence.

## OTHER DESIGN PROCEDURES

### Preliminary Equipment Layout

**Flows in Mains and Laterals.** Regardless of the method used to determine the flow through each item of terminal equipment, the desired result should be listed on the preliminary plans or in a schedule of flow rates for the piping system.

In an equipment schedule or on the plans, starting from the most remote terminal and working toward the pump, progressively list the cumulative flow and head loss in each of the mains and branch circuits in the distribution system. It is helpful in system commissioning to have a piping system schematic noting the friction loss calculations of each flow path. This should be included with the design drawings of the project.

The designer is responsible for selecting control valves and coordinating them with heat transfer devices. A schedule of devices and detailed connection schematic should be given.

**Preliminary Pipe Sizing.** For each portion of the piping circuit, select a tentative pipe size from the unified flow chart (Figure 1 in Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*), using a value of pipe friction loss ranging from 0.75 to 4 ft per 100 ft of straight pipe. Velocity through the pipe should also be examined, and should not exceed 10 fps in general application. Air management techniques suggest that velocity should also be kept above 2 fps, so entrained air is carried through the system, although piping layout considerations may make this less of an issue. The copper piping trade association suggests establishing maximum velocities of 5 fps for hot-water piping and 8 fps for cold-water piping (CDA 2006). Others suggest that no greater than 4 fps be used on pipes less than 1.5 in. in diameter. Other suggestions are in Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*.

Residential piping size is often based on pump preselection using pipe sizing tables, which are available from the Hydronics Institute or from manufacturers. Allow adequate space for water entrance to pump volute, to reduce pump suction losses in manifolds.

**Preliminary Pressure Drop.** Using the preliminary pipe sizing, determine the pressure drop through each portion of piping. The total pressure drop in the longest circuits determines the maximum pressure drop through the piping, including the terminals and control valves, that must be available in the form of pump pressure.

**Preliminary Pump Selection.** The preliminary selection should be based on the pump's ability to fulfill the determined capacity requirements. It should be selected as close as possible to best efficiency on the pump curve and should not overload the motor. Because pressure drop in a flow system varies as the square of the flow rate, the flow variation between the nearest size of stock pump and an exact point selection will be relatively minor. Note that although efficiency is important, it is secondary to the pipe sizing criteria. Proper pipe sizing reduces overall head losses, allowing a

lower-power pump to be used for the same flow rate, and should be considered first.

### Final Pipe Sizing and Pressure Drop Determination

**Final Piping Layout.** Examine the overall piping layout to determine whether pipe sizes in some areas need to be readjusted. Several principal circuits should have approximately equal pressure drops so that excessive pressures are not needed to serve a small portion of the building.

Consider both the initial cost of the pump and piping system and the pump's operating cost when determining final system friction loss. Generally, lower heads and larger piping are more economical when longer amortization periods are considered, especially in larger systems. However, in small systems such as in residences, it may be most economical to select the pump first and design the piping system to meet the available pressure. In all cases, adjust the piping system design and pump selection until the optimum design is found.

**Final Pressure Drop.** When the final piping layout has been established, determine the friction loss for each section of the piping system from the pressure drop charts (Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*) for the mass flow rate in each portion of the piping system.

After calculating friction loss at design flow for all sections of the piping system and all fittings, terminal units, and control valves, sum them for several of the longest piping circuits to determine the pressure against which the pump must operate at design flow.

**Final Pump Selection.** After completing the final pressure drop calculations, select the pump by plotting a system curve and pump curve and selecting the pump or pump assembly that operates closest to the calculated design point.

### Freeze Prevention

All circulating water systems require precautions to prevent freezing, particularly in makeup air applications in temperate climates where (1) coils are exposed to outside air at freezing temperatures, (2) undrained chilled-water coils are in the winter airstream, or (3) piping passes through unheated spaces. Freezing will not occur as long as flow is maintained and the water is at least warm. Unfortunately, during extremely cold weather or in the event of a power failure, water flow and temperature cannot be guaranteed. Additionally, continuous pumping can be energy-intensive and cause system wear. The following are precautions to avoid flow stoppage or damage from freezing:

- Select all load devices (such as preheat coils) subjected to outside air temperatures for constant-flow, variable  $\Delta t$  control.
- Position the coil valves of all cooling coils with valve control that are dormant in winter to the full-open position at those times.
- If intermittent pump operation is used as an economy measure, use an automatic override to operate both chilled-water and heating-water pumps in below-freezing weather.
- Select pump starters that automatically restart after power failure (i.e., maintain-contact control).
- Select nonoverloading pumps.
- Instruct operating personnel never to shut down pumps in sub-freezing weather.
- Do not use aquastats, which can stop a pump, in boiler circuits.
- Avoid sluggish circulation, which may cause air binding or dirt deposit. Properly balance and clean systems. Provide proper air control or means to eliminate air.
- Install low-temperature-detection thermostats that have phase change capillaries wound in a serpentine pattern across the leaving face of the upstream coil.

In fan equipment handling outside air, take precautions to avoid stratification of air entering the coil. The best methods for proper mixing of indoor and outdoor air are the following:

- Select dampers for pressure drops adequate to provide stable control of mixing, preferably with dampers installed several equivalent diameters upstream of the air-handling unit.
- Design intake and approach duct systems to promote natural mixing.
- Select coils with circuiting to allow parallel flow of air and water.

Freeze-up may still occur with any of these precautions. If an antifreeze solution is not used, water should circulate at all times. Valve-controlled elements should have low-limit thermostats, and sensing elements should be located to ensure accurate air temperature readings. Primary/secondary pumping of coils with three-way valve injection (as in [Figures 29B](#) and [29C](#)) is advantageous. Use outdoor reset of water temperature wherever possible.

### ANTIFREEZE SOLUTIONS

In systems in danger of freeze-up, water solutions of ethylene glycol and propylene glycol are commonly used. Freeze protection may be needed (1) in snow-melting applications (see Chapter 50 of the 2007 *ASHRAE Handbook—HVAC Applications*); (2) in systems subjected to 100% outside air, where the methods outlined above may not provide absolute antifreeze protection; (3) in isolated parts or zones of a heating system where intermittent operation or long runs of exposed piping increase the danger of freezing; and (4) in process cooling applications requiring temperatures below 40°F. Although using ethylene glycol or propylene glycol is comparatively expensive and tends to create corrosion problems unless suitable inhibitors are used, it may be the only practical solution in many cases.

Solutions of triethylene glycol, as well as certain other heat transfer fluids, may also be used. However, ethylene glycol and propylene glycol are the most common substances used in hydronic systems because they are less costly and provide the most effective heat transfer.

### Effect on Heat Transfer and Flow

Tables 6 to 13 and Figures 9 to 16 in Chapter 21 of the 2005 *ASHRAE Handbook—Fundamentals* show density, specific heat, thermal conductivity, and viscosity of various aqueous solutions of ethylene glycol and propylene glycol. Tables 4 and 5 of that chapter indicate the freezing points for the two solutions.

System heat transfer rate is affected by relative density and specific heat according to the following equation:

$$q_w = 500Q(\rho/\rho_w)c_p\Delta t \quad (21)$$

where

$q_w$  = total heat transfer rate, Btu/h  
 $Q$  = flow rate, gpm  
 $\rho$  = fluid density, lb/ft<sup>3</sup>  
 $\rho_w$  = density of water at 60°F, lb/ft<sup>3</sup>  
 $c_p$  = specific heat of fluid, Btu/lb·°F  
 $\Delta t$  = temperature increase or decrease, °F

### Effect on Heat Source or Chiller

Generally, ethylene glycol solutions should not be used directly in a boiler because of the danger of chemical corrosion caused by glycol breakdown on direct heating surfaces. However, properly inhibited glycol solutions can be used in low-temperature water systems directly in the heating boiler if proper operation can be ensured. Automobile antifreeze solutions are not recommended because the silicate inhibitor can cause fouling, pump seal wear, fluid gelation, and reduced heat transfer. The area or zone requiring antifreeze protection can be isolated with a separate heat exchanger or converter. Glycol solutions are used directly in water chillers in many cases.

Glycol solutions affect the output of a heat exchanger by changing the film coefficient of the surface contacting the solution. This change in film coefficient is caused primarily by viscosity changes. [Figure 42](#) illustrates typical changes in output for two types of heat

exchangers, a steam-to-liquid converter and a refrigerant-to-liquid chiller. The curves are plotted for one set of operating conditions only and reflect the change in ethylene glycol concentration as the only variable. Propylene glycol has a similar effect on heat exchanger output.

Because many other variables (e.g., liquid velocity, steam or refrigerant loading, temperature difference, and unit construction) affect the overall coefficient of a heat exchanger, designers should consult manufacturers' ratings when selecting such equipment. The curves indicate only the magnitude of these output changes.

### Effect on Terminal Units

Because the effect of glycol on the capacity of terminal units may vary widely with temperature, the manufacturer's rating data should be consulted when selecting heating or cooling units in glycol systems.

### Effect on Pump Performance

Centrifugal pump characteristics are affected to some degree by glycol solutions because of viscosity changes. Figure 43 shows these effects on pump capacity, head, and efficiency. Figures 12 and 16 in Chapter 21 of the 2005 *ASHRAE Handbook—Fundamentals* plot the viscosity of aqueous ethylene glycol and propylene glycol. Centrifugal pump performance is normally cataloged for water at 60 to 80°F. Hence, absolute viscosity effects below 1.1 centipoise can

safely be ignored as far as pump performance is concerned. In intermittently operated systems, such as snow-melting applications, viscosity effects at start-up may decrease flow enough to slow pickup.

### Effect on Piping Pressure Loss

Friction loss in piping also varies with viscosity changes. Figure 44 gives correction factors for various ethylene glycol and propylene glycol solutions. These factors are applied to the calculated pressure loss for water [Equation (23)]. No correction is needed for ethylene glycol and propylene glycol solutions above 160°F.

### Installation and Maintenance

Because glycol solutions are comparatively expensive, the smallest possible concentrations to produce the desired antifreeze properties should be used. The system's total water content should be calculated carefully to determine the required amount of glycol (Craig et al. 1993). The solution can be mixed outside the system in drums or barrels and then pumped in. Air vents should be watched during filling to prevent loss of solution. The system and cold-water supply should not be permanently connected, so automatic fill valves are usually not used.

Ethylene glycol and propylene glycol normally include an inhibitor to help prevent corrosion. Solutions should be checked each year using a suitable refractometer to determine glycol concentration. Certain precautions regarding the use of inhibited ethylene glycol solutions should be taken to extend their service life and to preserve equipment:

- Before injecting the glycol solution, thoroughly clean and flush the system.
- Use waters that are soft and low in chloride and sulfate ions to prepare the solution whenever possible.

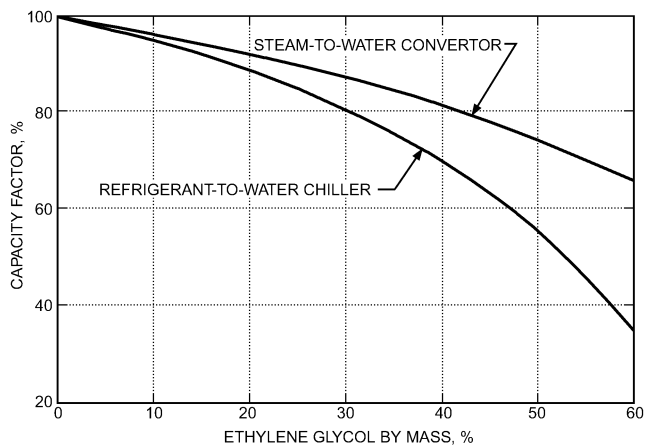


Fig. 42 Example of Effect of Aqueous Ethylene Glycol Solutions on Heat Exchanger Output

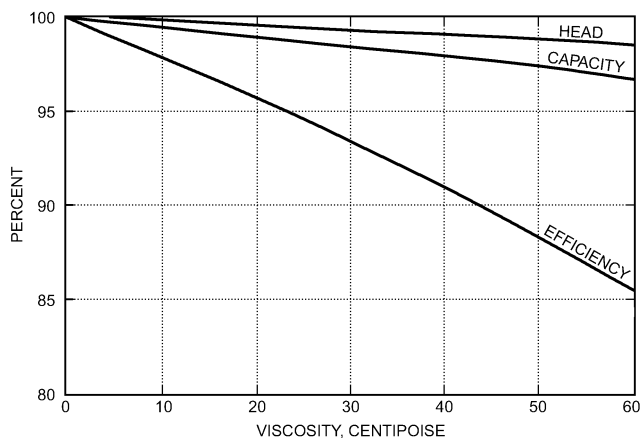


Fig. 43 Effect of Viscosity on Pump Characteristics

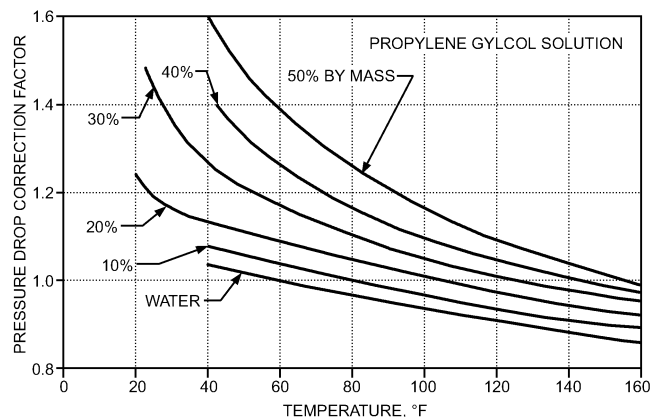
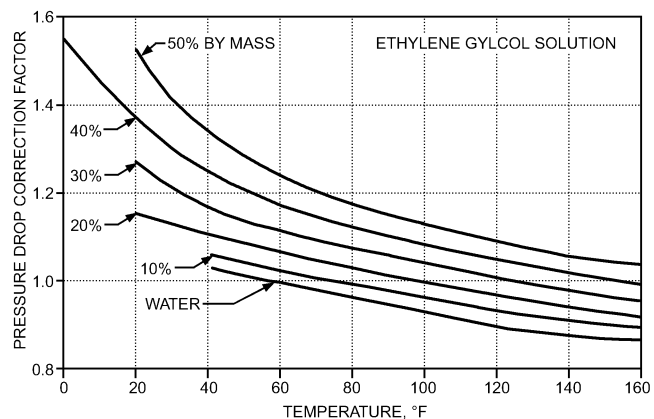


Fig. 44 Pressure Drop Correction for Glycol Solutions



- Limit the maximum operating temperature to 250°F in a closed hydronic system. In a heat exchanger, limit glycol film temperatures to 300 to 350°F (steam pressures 120 psi or less) to prevent deterioration of the solution.
- Check the concentration of inhibitor periodically, following procedures recommended by the glycol manufacturer.

## REFERENCES

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