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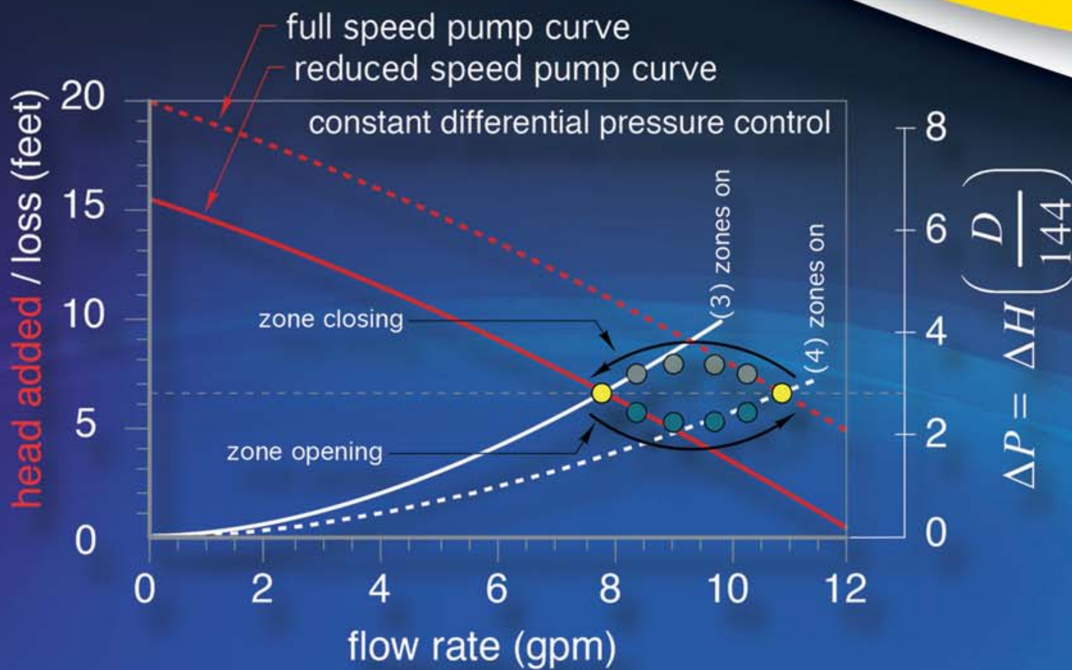
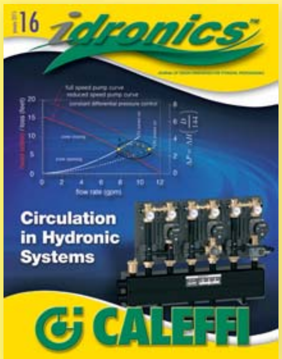
JOURNAL OF DESIGN INNOVATION FOR HYDRONIC PROFESSIONALS



Circulation in Hydronic Systems

CALEFFI



A Technical Journal
from
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Dear Hydronic and Plumbing Professional,

Last year, we surveyed our 25 U.S. and Canadian sales representatives about topics they would like to see addressed in future issues of *idronics*. Their responses included frequent use of words such as circulators, flow and cavitation, and terms like pump selection, pump sizing, fluid velocity, pumping away and pressure drop.

OK, we got the message!

This issue of *idronics* focuses on circulation in hydronic systems. It discusses circulators and how they interact with other system components — the type of components we manufacture, as well as components from other sources. Although Caleffi does not manufacture circulators, an understanding of how they operate, as well as how they are sized and selected, is crucial to good hydronic system design. It's something that's discussed in all of our training efforts.

From time to time we receive photos showing installations of Caleffi products. We sincerely appreciate these submittals. They help us show and explain the best way to apply and install specific products. This issue includes some of those photos. Our thanks are extended to all those who provided them. We encourage you to continue sharing your project photos with us.

We hope you enjoy this 16th issue of *idronics* and encourage you to send us any feedback by e-mailing us at idronics@caleffi.com.

For prior issues, please visit us at www.caleffi.com, and click on the idronics icon. There you can download the PDF files. You can also register to receive hard copies of future issues.

Mark Olson

Mark Olson

General Manager & CEO

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Flow in Hydronic Systems

1. INTRODUCTION

A hydronic circuit is a "conveyor belt" for heat. That heat is loaded on at the heat source, carried through the building by a stream of water, and delivered into the spaces to be heated by one or more heat emitters. During this process, the water is neither creating nor permanently absorbing the heat. It simply serves as temporary storage and transport for the heat. A given quantity of water can repeat this process indefinitely. It never loses its ability to absorb or release heat.

Delivering heat to each conditioned space at the proper rate requires proper flow rates in each part of the system. If the flow rate is too low, the likely result is insufficient heat delivery and loss of comfort. If the flow rate is too high, the system may produce annoying noise or operate under conditions that will ultimately damage some components. It will also require excessive electrical power input to its circulator(s).

This issue of *idronics* focuses on determining the proper flow rates in different portions of a hydronic system, and then selecting piping and circulators that can ensure those flow rates occur when the system is built and commissioned.

The discussion begins with a brief history of how flow was created—without the use of circulators—in older hydronic systems, and compares those early methods with modern technology. It moves on to discuss criteria for selecting pipe sizes in hydronic circuits based on the required rate of heat conveyance. Analytical methods for characterizing the flow characteristics of both simple and complex piping circuits are then described.

After this, the discussion turns to circulator characteristics and how to select a circulator for a specific hydronic circuit so that it operates with minimal power consumption and optimal efficiency. This issue concludes with a discussion of the distribution efficiency of a hydronic system and how to achieve systems that convey unsurpassed comfort using the least amount of electrical energy.

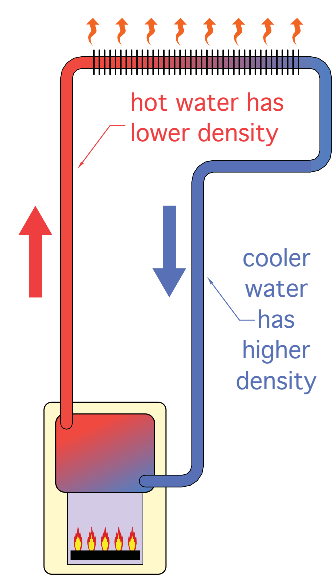
2. A BRIEF HISTORY OF FLOW IN HYDRONIC SYSTEMS

Water-based hydronic heating systems in North America date back over a century. Most accounts point to water-based systems evolving from steam-based systems. The latter were the first means of equipping buildings with "central" heating. Before that, heating was done with fireplaces and wood-fired or coal-fired stoves located throughout the building.

EARLY WATER-BASED SYSTEMS:

When water-based hydronic systems were first developed, and for several decades to follow, there were no electrically powered circulators available. The only "propulsion" effect available to move water through systems was the differential pressure created by the simultaneous presence of hot and cool water in the same system. Hot water in the boiler is slightly less dense, and therefore lighter, than cooler water in the return side piping. This creates a slight weight imbalance in the vertical piping that causes hot water to rise while cool water descends. Thus, a slow circulation occurs within the system, as depicted in Figure 2-1.

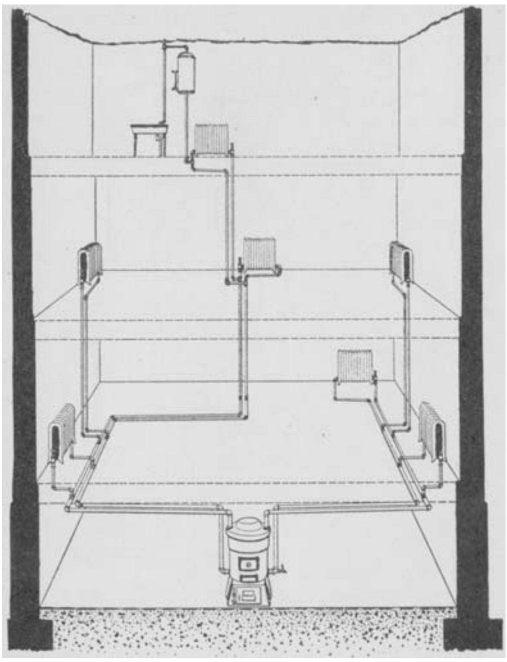
Figure 2-1



These early hot water systems were often referred to as "gravity" hot water systems.

An example of the piping used in these early hydronic systems is shown in Figure 2-2.

Figure 2-2



Notice that the piping is either vertical or slightly sloped. This encourages buoyancy-driven flow and helps prevent air pockets from forming within the piping. Such pockets could block flow.

These early hydronic systems were usually open to the atmosphere at the overflow pipe connected to the top of the attic-mounted expansion tank. As such, the pressure within the boiler and piping were limited. In this respect, they were safer than pressurized steam systems of that vintage. However, due to evaporation, the water level within the system would slowly drop. Water had to be periodically added to maintain proper operation.

Although they took great advantage of natural principles to create circulation, these early systems were very limited in how they could be applied relative to modern hydronic systems.

THE ARRIVAL OF CIRCULATORS:

Many of these limitations were eliminated when electrically powered circulators became available in North America starting in the early 1930s. One example of an early generation (1940) "booster" circulator is shown in Figure 2-3.

Figure 2-3

Courtesy of Bell & Gossett

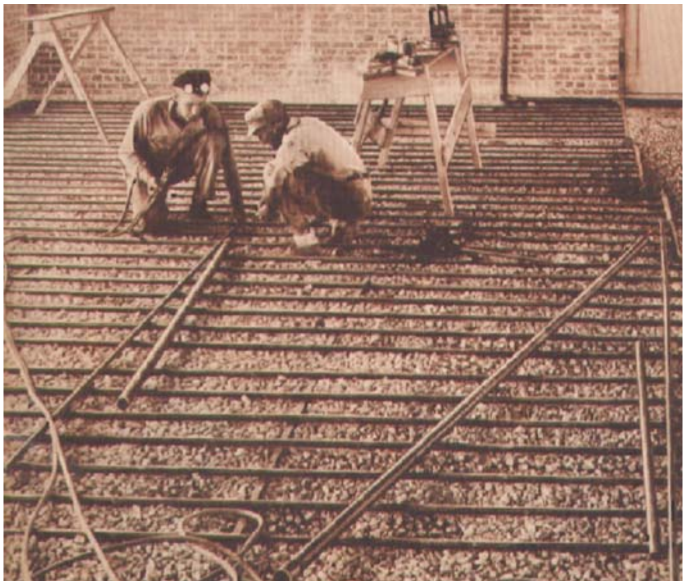
Circulators could create much greater pressure differentials compared to those created solely by the buoyancy differences between hot and cool water. This allowed for smaller diameter piping that could go in virtually any direction. The boiler no longer had to be located at the base of the system. Piping layouts could be very different from those previously required. Response times were also significantly shorter than with natural circulation systems.

In 1958, Taco introduced the first "wet rotor" circulator to the North American hydronics market. Wet rotor circulators slowly gained acceptance as an alternative to traditional 3-piece circulators. By 1980, they had become the predominant type of circulator used in new residential and light commercial hydronic systems.

The availability of circulators also allowed for distribution systems and heat emitters very different from those required by "gravity" flow systems. One example is the hydronic radiant floor panel system shown in Figure 2-4, during its installation in the 1940s.

From the 1960s through the early 1980s, residential and light commercial hydronic heating systems were designed around metal piping (steel, black iron and copper). Distribution systems typically followed established "templates," such as series circuits, 1-pipe Monoflo® systems, 2-pipe direct return or 2-pipe reverse return. Most homes and smaller commercial buildings had one to three zones in which flow was controlled by

Figure 2-4



either zone circulators or zone valves. Pipe sizing and circulator selection was often done using rules of thumb and commonly accepted practice.

Then, during the early 1980s, cross-linked polyethylene tubing (e.g., PEX) made its way to North America after several years of successful use in Europe. PEX and other polymer-based tubing revolutionized the installation of hydronic radiant panel heating, providing fast installation and a long, reliable life.

The availability of PEX tubing was the spark that rekindled interest in hydronic radiant panel heating in North America during the 1980s. This led to expanded expectations in the hydronic heating market. Systems that could simultaneously supply low-temperature loads, such as radiant panel heating, and high-temperature loads, such as fin-tube baseboard, were being requested for newer and larger homes. Additional loads, such as domestic water heating, pool heating and snowmelting, were also being incorporated into these modern “multi-load/multi-temperature” systems. This led to more complicated system designs, some of which worked as expected, while others failed to deliver proper performance.

The complexity of these new systems often compromised the previously used “rule of thumb” design practices. Success often came down to the ability to carefully analyze the hydraulic characteristics of the more complex piping systems, and then choose appropriate piping and circulators that could deliver the required performance. Installers that lacked knowledge of how to do this were often left to “guestimate” piping and circulator selections. Most of these guestimates were very conservative,

Figure 2-5

which often resulted in grossly oversized hardware and unnecessarily high installed cost.

Circulator technology steadily improved during the latter half of the 20th century. The classic 3-piece “booster” circulator that was common in residential hydronic systems during the 1950s and ’60s, largely gave way to wet-rotor circulators. Figure 2-5 shows an example of a wet-rotor circulator from the late 1970s. Some of these circulators are still in operation with zero maintenance over almost 4 decades. This testifies to the robustness of their design and construction. Few other appliances can claim comparable reliability and long-term service.

Still, the increasing cost of electricity and a desire for continuous improvement have motivated circulator manufacturers to develop high-efficiency “smart” circulators such as the one shown in Figure 2-6.

Figure 2-6

These circulator use brushless DC motors (also known as ECM electronically commutated motors). These circulators can operate at a fixed speed or be programmed to vary their speed in proportion to changes in differential pressure or temperature. Use of these circulators greatly reduces the electrical power required to maintain flow in a hydronic circuit in comparison to that required by standard wet-rotor circulators.

Today, the availability of modern piping materials and high-efficiency circulators makes it possible to craft hydronic systems that deliver heating or cooling precisely when and where it is needed. They can do this while operating in virtual silence, and using minimal amounts of both heating fuel and electrical energy. Still, achieving these desired characteristics requires design efforts that go beyond "guestimating." Competent designers need to understand the fundamental relationships between flow, rate of heat transfer and temperatures. These fundamentals are then applied through proper sizing and selection of components such as piping and circulators. The final step is installation that follows "best practices." The remaining sections of this issue of *idronics* show and describe how this is done.

3. THE RELATIONSHIP BETWEEN FLOW AND HEAT TRANSFER

Hydronic system designers often need to know the rate of heat transfer to or from a fluid flowing through a device such as a heat source or heat emitter. This can be done using the sensible heat rate formula given as Formula 3-1:

Formula 3-1

$$q = (8.01Dc)f(\Delta T)$$

Where:

q = rate of heat transfer into or out of the water stream (Btu/hr)
8.01 = a constant based on the units used
D = density of the fluid (lb/ft^3)
c = specific heat of the fluid (Btu/lb/ºF)
f = flow rate of fluid through the device (gpm)
ΔT = temperature change of the fluid through the device (ºF)

When using Formula 3-1, the density and specific heat of the fluid should be based on its *average* temperature during the process by which the fluid is gaining or losing heat.

For cold water only, Formula 3-1 simplifies to Formula 3-2:

Formula 3-2

$$q = 500f(\Delta T)$$

Where:

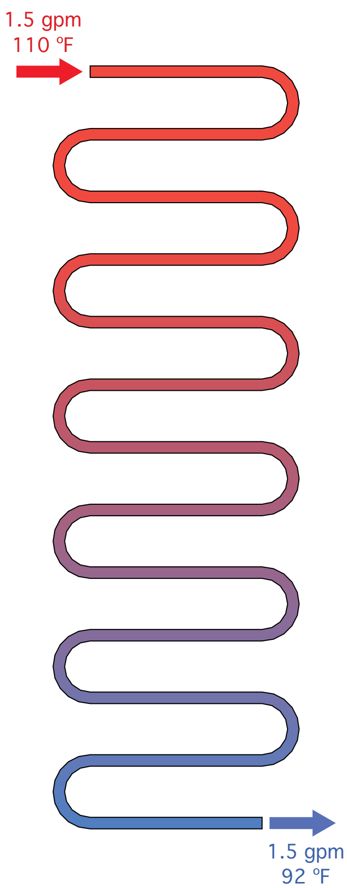
q = rate of heat transfer into or out of the water stream (Btu/hr)
f = flow rate of water through the device (gpm)
500 = constant rounded off from 8.33 x 60
ΔT = temperature change of the water through the device (ºF)

Formula 3-2 is technically only valid for cold water because the factor 500 is based on the density of water at approximately 60ºF. However, because the factor 500 is easy to remember, this formula is often used for quick mental calculations for the rate of sensible heat transfer involving water. While fine for initial estimates, it is better to use Formula 3-1 for final calculations, because it accounts for variations in both the density and specific heat of the fluid. Formula 3-1 can also be used for fluids other than water.

Example: Water flows into a radiant panel circuit at 110°F and leaves at 92°F. The flow rate is 1.5 gpm, as shown in Figure 3-1. Calculate the rate of heat transfer from the water to the heat emitter using:

a. Formula 3-1
b. Formula 3-2

Figure 3-1



Solution: Using Formula 3-2, the rate of heat transfer from the circuit is:

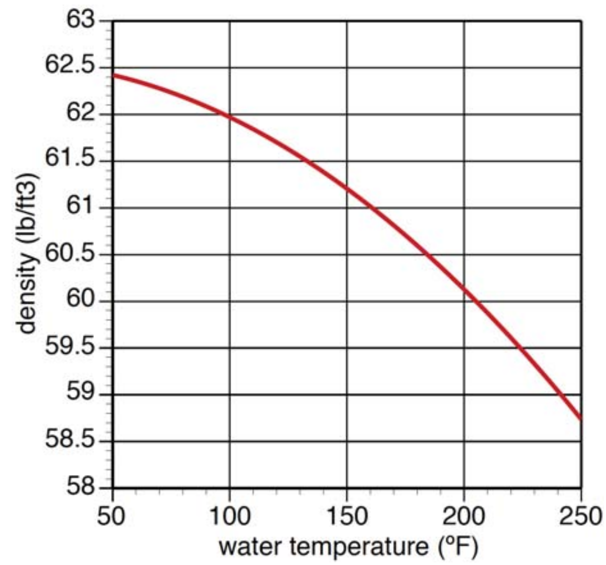
$$q = 500 f(\Delta T) = 500 \times 1.5 \times (110 - 92) = 13{,}500 Btu / hr$$

To use Formula 3-1, the density of water at its average temperature of 101°F must first be estimated. It can be found using the graph in Figure 3-2:

At a temperature of 101 °F, the density of water is 61.96 lb/ft^3

The specific heat of water can be assumed to remain 1.0 Btu/lb/°F.

Figure 3-2



Putting these numbers into Formula 3-1 yields:

$$q = (8.01Dc) f(\Delta T) = (8.01 \times 61.96 \times 1.00) \times 1.5 \times (110 - 92) = 13{,}400 Btu / hr$$

The difference in the two calculated rates of heat transfer is small, only about 0.7%. However, this difference will increase as the water temperature in the circuit increases.

Formula 3-2 is generally accepted in the hydronics industry for estimates of heat transfer to or from a stream of water. However, Formula 3-1 yields more accurate results when the variation in density and specific heat of the fluid can be factored into the calculation. Formula 3-1 should be used whenever calculations are being done for the heat conveyance rate of an antifreeze solution.

Formulas 3-1 and 3-2 can also be rearranged to determine the temperature drop or flow rate required for a specific rate of heat transfer.

Example: Estimate the temperature drop required to deliver heat at a rate of 50,000 Btu/hr using a distribution system operating with water at a flow rate of 4 gpm.

Solution: Just rearrange Formula 3-2 and put in the numbers.

$$\Delta T = \frac{q}{500 f} = \frac{50{,}000}{500 \times 4} = 25^{\circ} F$$

Example: What is the flow rate required to deliver 6,000 Btu/hr of a heat emitter that operates with water and a temperature drop of 30°F?

Solution: Again, Formula 3-2 can be rearranged to determine flow and the known values factored in.

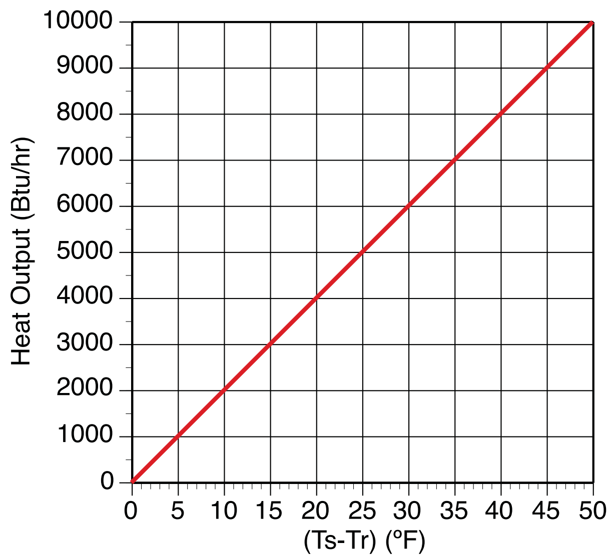
$$f = \frac{q}{500(\Delta T)} = \frac{6{,}000}{500(30)} = 0.6gpm$$

Formulas 3-1 and 3-2 are arguably the most important formulas in hydronic system design. They are constantly used to relate the rate of heat transfer to fluid flow rate and the temperature change of the fluid.

HOW FLUID TEMPERATURE AFFECTS HEAT TRANSFER RATES:

The heat output of a hydronic heat emitter increases in approximate proportion to the difference between temperature of the water supplied to it and room air temperature. A graph of heat output versus this difference is shown in Figure 3-3.

Figure 3-3

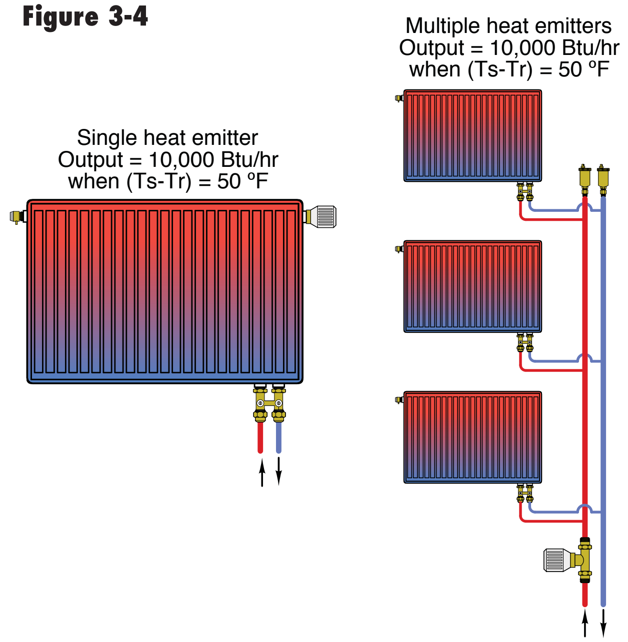


This graph might represent the heat output of a single heat emitter, such as a panel radiator, or it could represent the total heat output of a group of the same heat emitters that are all supplied by a parallel piping distribution system, as shown in Figure 3-4.

If water is supplied to the heat emitter(s) at room temperature (whatever that temperature happens to be), the difference (Ts-Tr) is zero, and so is the heat output from the heat emitter(s). As the temperature of the water supplied to the heat emitter(s) climbs above room air temperature, its heat output also increases.

This graph in Figure 3-3 shows that when the supply water temperature is 50°F above the room air temperature, the heat emitter(s) will release 10,000 Btu/hr into the building. If the supply water temperature is only 25°F above the room's air temperature, the heat emitter releases 5,000 Btu/hr into the building.

Figure 3-4



Formula 3-2 can be used to estimate the *rate* of heat transfer into or out of a device that has a stream of water flowing through it at a *measured* flow rate, and with a *measured* temperature change between the inlet and outlet of that water stream. For example: If a hydronic circuit is operating at a measured flow rate of 8 gpm, and if the temperature drop from its water inlet to its water outlet is measured to be 20°F, then that water stream is delivering heat to the heat emitter at a rate of:

$$q = 500f(\Delta T) = 500 \times 8 \times (20) = 80{,}000\frac{Btu}{hr}$$

You could also apply this formula to calculate the rate of heat transfer delivered by a stream of water flowing through a heat emitter at 16 gpm and undergoing a temperature drop of 10°F.

$$q = 500f(\Delta T) = 500 \times 16 \times (10) = 80{,}000\frac{Btu}{hr}$$

Mathematically, there are an infinite number of combinations of water flow rate and temperature change that, when multiplied together, will give the same result of 80,000 Btu/hr. However, don't interpret this to imply that all of these combinations are *achievable operating conditions*. For example, the mathematics of Formula 3-2 would imply that a circuit operating at a flow rate of 0.2 gpm and a temperature drop of 800ºF would also deliver 80,000 Btu/hr.

$$q = 500f(\Delta T) = 500 \times 0.2 \times 800 = 80{,}000\,Btu/hr$$

Even though the math is valid, there is no practical way to build a hydronic heating system that could operate with an 800ºF temperature drop. This demonstrates that there is a distinct difference between what Formula 3-2 calculates as the rate heat transfer from *measured* flow rate and temperature drop values, and how this formula should be used to *predict* the heat transfer rate from assumed flow rate and temperature drop numbers.

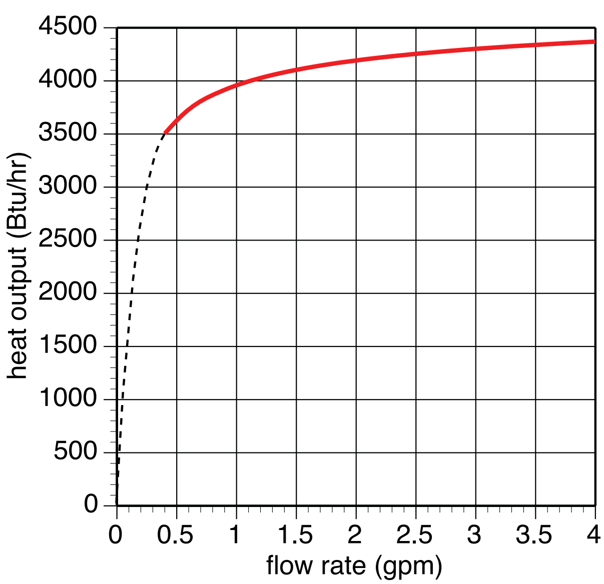
Here's another example: Suppose you designed a hydronic distribution system that would operate at a flow rate of 8 gpm and a 20ºF water temperature drop under design load conditions. Then you thought about how that system should operate under half load conditions (e.g., a heat output of 40,000 Btu/hr). You might assume that by slowing the flow rate from 8 gpm to 4 gpm and maintaining a 20ºF temperature drop, the system would release half its design load output. Mathematically, this works:

$$q = 500f(\Delta T) = 500 \times 4.0 \times (20) = 40{,}000\frac{Btu}{hr}$$

But just because the math is valid doesn't mean that the circuit will actually behave accordingly.

Figure 3-5 shows the heat output versus flow rate relationship for a 12-foot long piece of residential fin-tube baseboard being supplied with 160ºF water. The baseboard this graph is based on, is rated to release 600 Btu/hr per foot of element length when the element contains 200ºF water moving at a flow rate of 1 gpm.

Figure 3-5



The heat output curve in Figure 3-5 is based on a detailed engineering model of fin-tube baseboard that accounts for continually decreasing water temperature along the element. This model is also based on information in the I=B=R rating standard for fin-tube baseboard—specifically, that heat output from fin-tube baseboard varies with the 0.04 power of flow rate. Thus, if the heat output from a baseboard at a flow rate of 1 gpm is 250 Btu/hr/ft, then the heat output from that baseboard at a flow rate of 4 gpm (and the same water temperature) would be:

$$250 \times (4)^{0.04} = 250 \times 1.057 = 264\frac{Btu/hr}{ft}$$

Assume water enters a baseboard that conforms to the relationship shown in Figure 3-5. The water flow rate is 4 gpm, and it enters the baseboard at a temperature of 160ºF. Room air flows into the bottom of the baseboard at 68ºF. Under these conditions, the baseboard's heat output is about 4,350 Btu/hr.

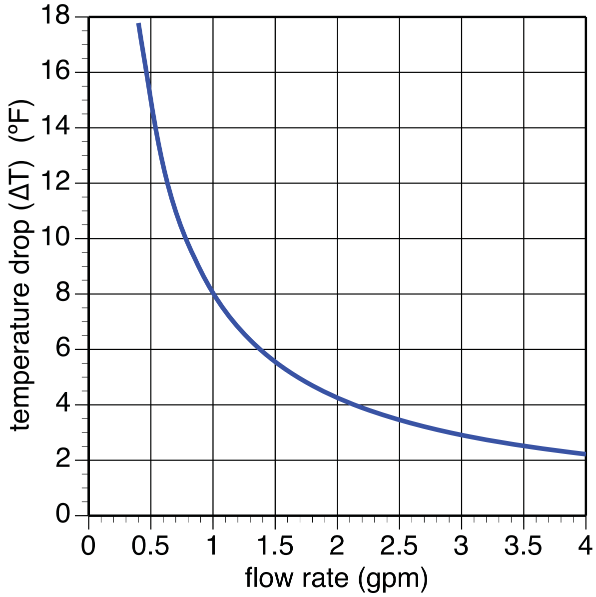
Next, consider what happens if the same water and air inlet temperatures are maintained, but the flow rate through the 12-foot baseboard is reduced by 50% (e.g., from 4 to 2 gpm). The baseboard's heat output drops from 4,350 to about 4,200 Btu/hr. A drop of only about 3.5%. If the flow is cut in half again, down to 1 gpm, the heat output drops to about 3,900 Btu/hr, a drop of about 10.3% from the heat output at 4 gpm flow rate.

The red curve in Figure 3-5 doesn't go below flow rates of 0.4 gpm, because at flow rates in the range of 0.3 gpm, the water passing through the 3/4" copper fin-tube transitions from turbulent to laminar flow. This will cause a significant drop in heat output. Even though the exact drop in heat output at very low flow rates is not directly predictable, it obviously drops to 0 heat output at 0 flow rate.

Figure 3-5 shows that the heat output from the baseboard as a function of flow rate is very "non-linear." Heat output decreases relatively slowly at higher flow rates, but drops off very quickly at low flow rates.

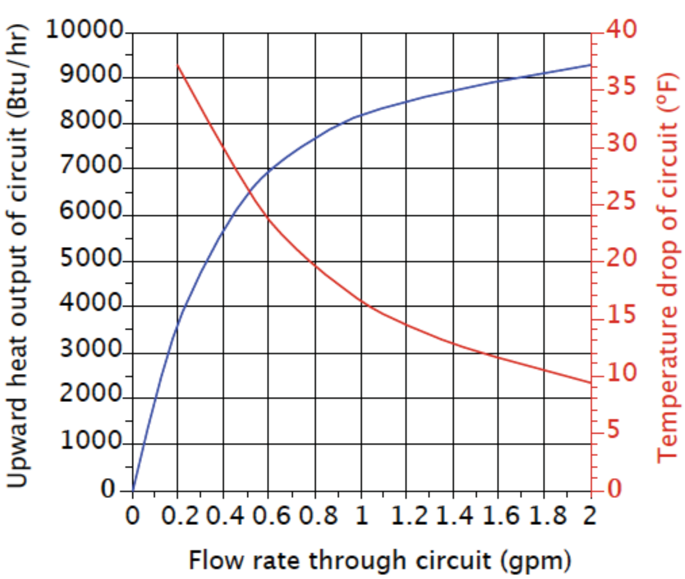
Figure 3-6 shows what happens with the temperature drop (or ΔT) across the same 12-foot baseboard as the flow rate through it varies across the same range, while the supply water temperature remains steady at 160ºF.

Figure 3-6



At 4 gpm, the temperature drop across this baseboard is only about 2.2ºF. At 2 gpm, it's about 4.2ºF, and at 1 gpm, the ΔT is about 8ºF. If the flow gets down to 0.4 gpm, the ΔT is just under 18ºF. The temperature drop along the baseboard naturally changes as the flow rate through the baseboard changes. The notion that any heat emitter "wants to," or even *can*, remain at a fixed temperature drop as the flow rate changes is not supported by these results.

Figure 3-7



This behavior is not limited to fin-tube baseboard. It also applies to other heat emitters such as fan-coils and radiant panel circuits. Figure 3-7 shows what happens with upward heat output and temperature drop for a 300-foot long circuit of1/2" PEX tubing, buried in a bare 4-inch thick concrete slab at tube spacing of 12 inches. The circuit is supplied with water at 110ºF. The room temperature above the floor is 70ºF.

Reducing the flow rate through this circuit from 2 to 1 gpm only reduces heat output from about 9,300 Btu/hr to about 8,200 Btu/hr, a drop of about 12%. The corresponding temperature drop along the circuit changes from about 9.5ºF at 2 gpm to about 17ºF at 1 gpm.

Figures 3-6 and 3-7 show that the relationship between temperature drop (ΔT) and the flow rate through a heat emitter (or a complete distribution system consisting of multiple parallel-connected heat emitters) is not proportional.

SELECTING A DESIGN ΔT:

Before a flow rate for a hydronic heating circuit can be determined, the designer must select a "design temperature drop" (also known as a "design ΔT") for the circuit. These terms both refer to the temperature drop across the distribution system at design load conditions.

The selection of design ΔT usually depends on the type of heat emitters used and how they are piped. It should also account for how the heat output of the heat emitters will change based on the flow rate through them. Finally, this selection should also consider the implications of circulator power requirements, which will depend on flow rate, and thus implicitly depend on design ΔT.

A "traditional" design ΔT for hydronic circuits in North America is 20ºF. However, there is nothing special about this number. The design ΔT of a hydronic circuit can be higher or lower depending on circumstances and design objectives.

For example, in a room where radiant floor heating is expected to deliver "barefoot friendly" floors, a design ΔT of 10º to 15ºF is suggested for the tubing circuits. This reduces the variation in floor surface temperature from where the circuit enters the room and is carrying relatively warm water to where it leaves the room carrying lower temperature water.

Figure 3-8a

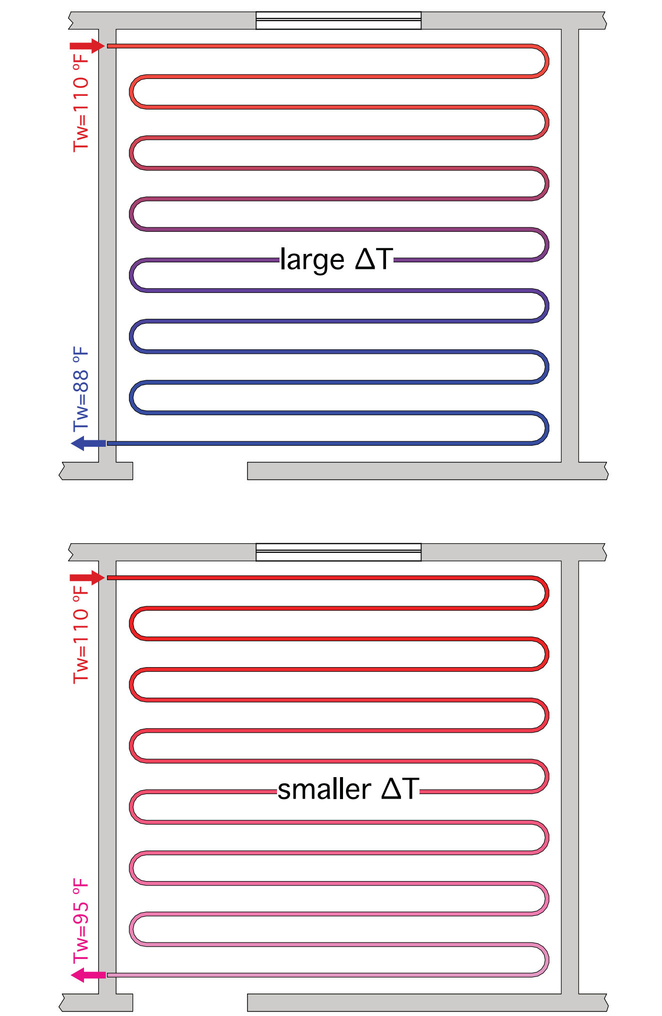


Figure 3-8b

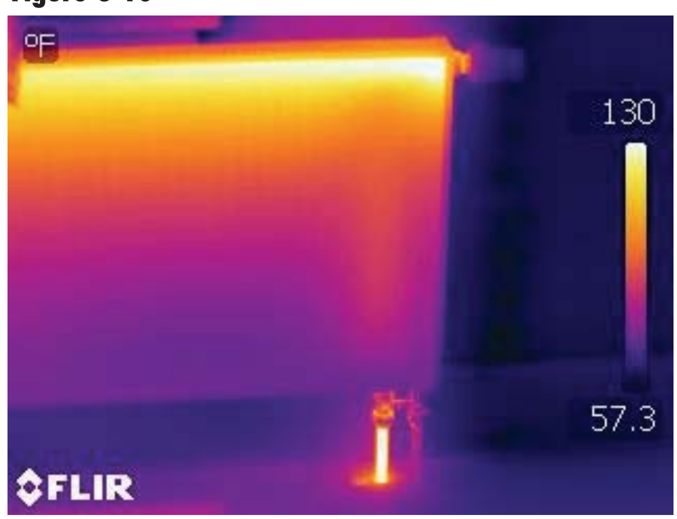
However, if the floor-heating circuit is in an industrial building, where occupants are most likely wearing work boots, the variation in floor surface temperature is usually not as much of a concern. This allows the design ΔT of the floor-heating circuits to be increased to 20º or even 25ºF. These high-circuit temperature drops allow for lower flow rates, which will likely reduce circulator power requirements. It may also allow use of small-diameter tubing.

Figure 3-9

Design ΔTs of 25° to 35ºF can often be used with panel radiators or fan coil units, provided that these heat emitters can still release heat into their assigned spaces at rates that meet design heating load. Again, the benefit of higher design ΔT is lower flow rates, and the possibility of smaller piping, smaller circulators and lower circulator energy consumption.

Figure 3-10 shows an infrared thermograph of a panel radiator operating at a ΔT estimated at over 50ºF.

Figure 3-10

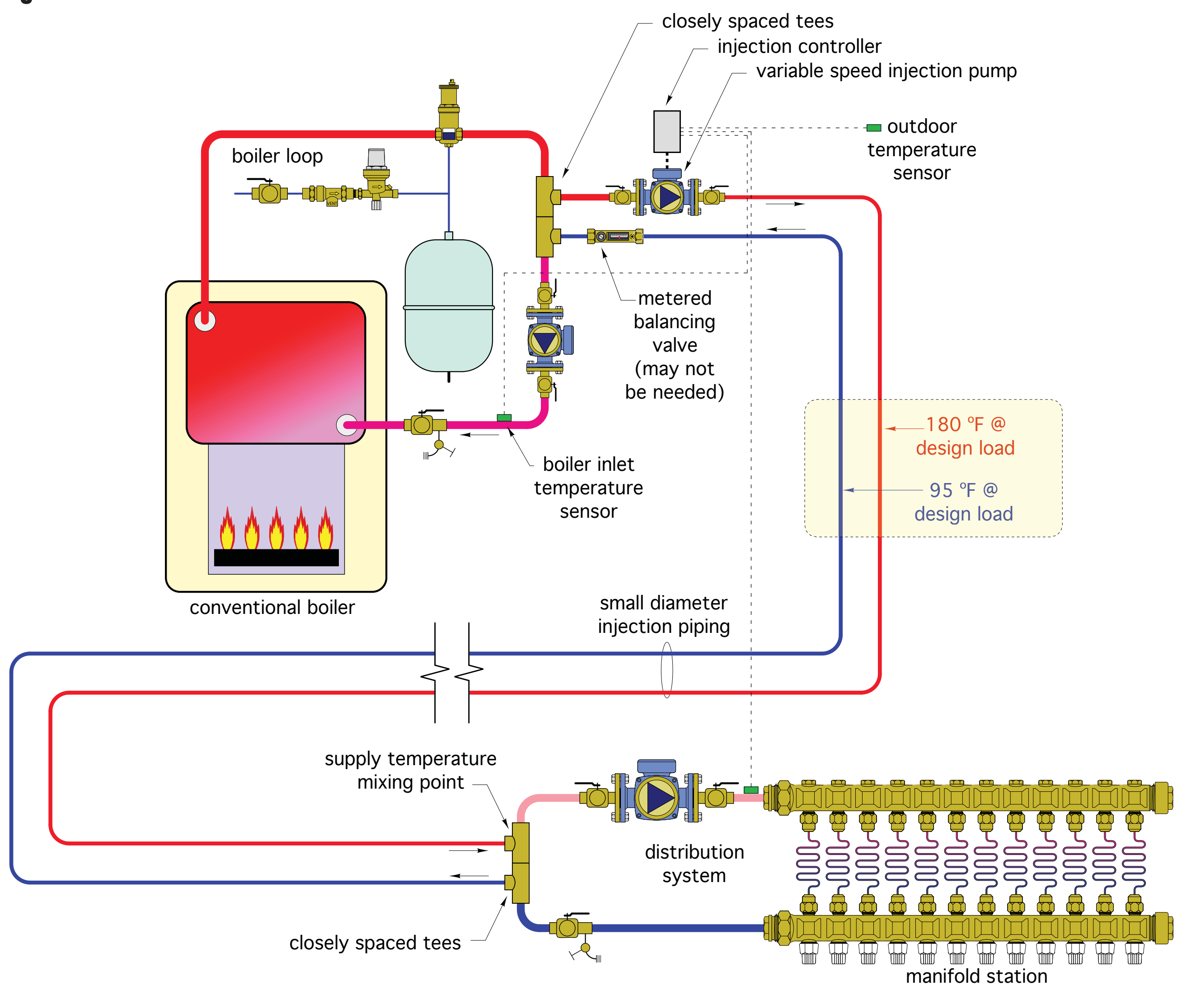


Notice the color gradient from the top of the radiator to the bottom. Compare it to the corresponding temperature scale at the right side of the image to see the wide ΔT across the panel. Also notice that the flow distribution across the vertical water passages on the radiator's surface is relatively uniform.

There are even situations in which certain hydronic circuits can operate at design ΔTs of 80° to perhaps 100°F. Figure 3-11 shows an example of a "minitube" system, which supplies low-temperature floor-heating circuits from a conventional boiler.

The piping between the closely spaced tees in the boiler loop and those in the manifold circuit can operate with a design ΔT of 80° to even 100°F. The water supplied by the boiler to the "hot" injection pipe (shown in red) may be 180ºF under design load conditions. The water returning from the radiant floor circuits under these conditions might only be 95ºF. The difference between these temperatures: 180 - 95 = 85ºF is the design ΔT of the injection piping running between the boiler room and the manifold station. This allow each gallon per minute of water flow through the injection piping to carry approximately 42,500 Btu/hr from the boiler room to the manifold station. Under such conditions, a 3/4" tube could convey about 280,000 Btu/hr. This approach greatly reduces the size and installed cost of the tubing between the mechanical room and a manifold station. The details shown in Figure 3-11 can be repeated to supply multiple manifolds in the same building.

Figure 3-11



DESIGN FLOW RATE:

The design flow rate of a hydronic circuit is determined based on the required rate of heat transport and the selected design ΔT. Once these values are known or selected, the design flow rate is determine using either Formula 3-1 or 3-2.

For example, assume a circuit supplying a panel radiator needs to supply 6,000 Btu/hr to that radiator at design load conditions. The designer decides to operate the panel radiator using a design ΔT of 30ºF. The necessary flow rate will be:

$$f = \frac{q}{500(\Delta T)} = \frac{6{,}000}{500(30)} = 0.6 gpm$$

This relatively low flow rate is easily handled by a 1/2" size tube.

The designer now needs to determine the size of that panel radiator based on the average water temperature within it under design load conditions. This is done by reviewing heat output rating tables or curves for various radiators, which give specific outputs for the radiators based on their dimensions and the average water temperature at which they operate. Assuming a design ΔT of 30ºF, the supply water temperature to the panel at design load conditions needs to be 15ºF higher than this average water temperature.

FLOW VELOCITY:

The speed of the fluid passing through a pipe varies within the cross section of the pipe. For flow in a straight pipe, the fluid moves fastest at the centerline and slowest near the pipe's internal surface. The term flow velocity, when used to describe flow through a pipe, refers to the *average* speed of the fluid. If all fluid within the pipe moved at this average speed, then the volume of fluid moving past a point in the pipe over a given time would be exactly the same as the amount of fluid moved by the varying internal flow velocities.

The common units for flow velocity in North America are feet per second, abbreviated as either ft/sec or FPS. Within formulas, the average flow velocity is represented by the symbol v.

Formula 3-3 can be used to calculate the average flow velocity associated with a given flow rate in a round pipe.

Formula 3-3

$$v = \left(\frac{0.408}{d^2}\right) f$$

Where:

v = average flow velocity in the pipe (ft/sec)
f = flow rate through the pipe (gpm)
d = exact inside diameter of the pipe (inches)

The formulas in Figure 3-12 can be used to find the average flow velocity (v) in (ft/sec) for several types and sizes of tubing based on the flow rate (f) entered in gpm (gallons per minute)

Figure 3-12

Tubing size/type	Flow velocity v in ft/sec f in gpm
3/8" copper	v = 2.02 f
1/2" copper	v = 1.26 f
3/4" copper	v = 0.62 f
1" copper	v = 0.367 f
1.25" copper	v = 0.245 f
1.5" copper	v = 0.175 f
2" copper	v = 0.101 f
2.5" copper	v = 0.0655 f
3" copper	v = 0.0459 f
3/8" PEX	v = 3.15 f
1/2" PEX	v = 1.73 f
5/8" PEX	v = 1.20 f
3/4" PEX	v = 0.880 f
1" PEX	v = 0.533 f
1.25" PEX	v = 0.357 f
1.5" PEX	v = 0.256 f
2" PEX	v = 0.149 f
3/8" PEX-AL-PEX	v = 3.41 f
1/2" PEX-AL-PEX	v = 1.63 f
5/8" PEX-AL-PEX	v = 1.00 f
3/4" PEX-AL-PEX	v = 0.628 f
1" PEX-AL-PEX	v = 0.383 f

SELECTING A PIPE SIZE:

There is no universal method for selecting pipe sizes in hydronic systems. Some designers make selections based solely on flow velocity, while others make selections based on pressure drop.

An ASHRAE suggested guideline is to select smaller pipes (2" nominal diameter or smaller) based on not exceeding an average flow velocity of 4 feet per second.

Pipe sized larger than 2” can be selected based on not exceeding a “specific head loss” of 4 feet of head loss per 100 feet of pipe.

The criteria of not exceeding an average flow velocity of 4 feet per second is based on keeping any flow noise emitted from the piping at levels that are not objectionable in occupied space. This average flow velocity is also low enough that erosion of copper tubing is not an issue. Another benefit of this flow velocity is that it allows expedient separation of air and dirt within high-efficiency air and dirt separators. Finally, an average flow velocity of 4 feet per second produces reasonably low head losses in smaller tubing, as shown in Figure 3-13.

Figure 3-13

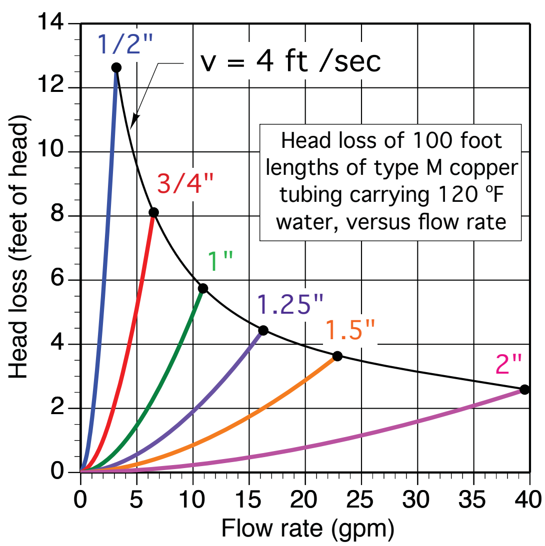


Figure 3-13 plots the head loss of 100 feet of the specified tubing carrying water at 120ºF versus the flow rate in that tubing. The curves only extend to a condition that represents a flow velocity of approximately 4 feet per second. It’s evident that increasing the tube size (e.g., its diameter) greatly reduces head loss, assuming the flow velocity remains constant. For example, at a flow velocity of 4 feet per second, the head loss associated with a 3.2 gpm flow of 120ºF water through a 1/2” copper tube is about 12.7 feet per 100 feet of pipe. Assuming the same 4 feet per second flow velocity, a 2” copper tube carrying the 120ºF water at a flow rate of 39.6 gpm only produces a head loss of about 2.6 feet per 100 feet of pipe. This trend continues as the pipe size increases above 2”.

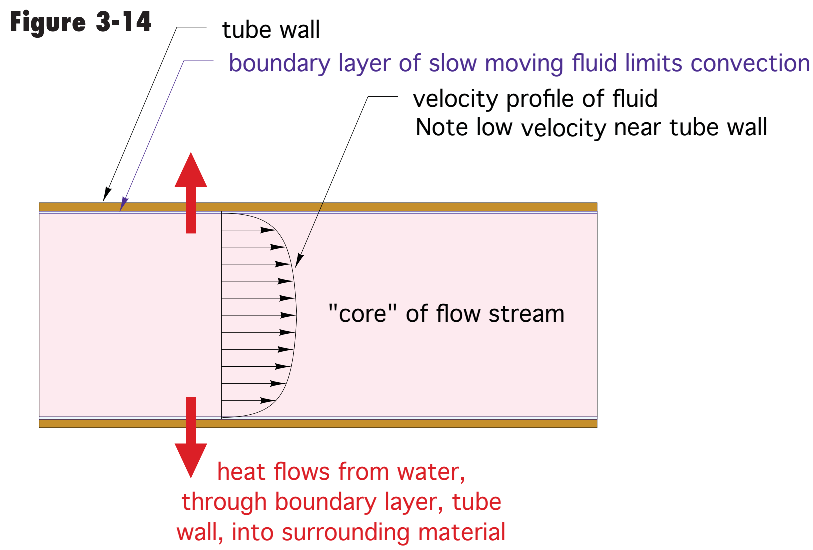
The rapid drop in head loss with increasing pipe size suggests that larger pipes sized to a flow velocity criteria such as 4 feet per second would result in very low head loss. Although this is good from the standpoint of the electrical power required for circulators, it also results in piping that is much larger, and more expensive, than necessary. This is the rationale behind using a different criteria for sizing tubing and piping larger than 2” nominal size.

The generally accepted sizing criteria for piping larger than 2” is called specific head loss. This method sets a maximum value for head loss per 100 feet of pipe. One reference suggests that this value can be anywhere between 0.5 and 3 feet of head loss per 100 feet of pipe. ASHRAE suggests a value of 4 feet of head loss per 100 feet of pipe. This is a relatively wide range, and the values at the extremes of this range will result in very different pipe sizes. The lower end of the range would be appropriate when the design priority is to minimize circulator power requirements. The upper end is appropriate when the design priority is to minimize pipe size and installation cost.

MINIMUM FLOW VELOCITIES

There are conditions under which very low flow velocity through pipes creates undesirable results. One of these conditions relates to air entrainment. Average flow velocities in piping should be no lower than 2 feet per second to enable the flowing water to entrain air bubbles and eventually bring them from remotes areas of the system back to a central air separator. This criteria is especially relevant to vertical piping with downward flow. Any air bubbles in such a pipe will try to rise due to their buoyancy in the surrounding water. An average flow velocity of 2 feet per second is considered adequate to

Figure 3-14



entrain air bubbles and carry them downward, eventually delivering them to a central air separator.

Another consideration regarding low flow velocity in pipes is laminar versus turbulent flow. Turbulent flow is always desirable when water is flowing through a component that either absorbs heat from the water or delivers heat to the water. Turbulent flow reduces the thickness of a slow-moving layer of fluid called the "boundary layer," which exists between the surface of a component that flow is passing through and the bulk of the fluid stream, as shown in Figure 3-14.

Think of the boundary layer as a thin layer of "fluid insulation" that inhibits heat transfer between the solid surface and the bulk of the fluid stream. Turbulence causes vigorous mixing of the fluid molecules. This reduces the thickness of the boundary layer, and thus improves heat transfer across it.

Maintaining turbulent flow is especially important for efficient heat transfer in small tubes, such as those used in water-to-air heat exchanger coils, or some radiant panel circuits.

The accepted criteria for ensuring that turbulent flow exists is to calculate a dimensionless quantity called the Reynolds' number. If the Reynolds' number is 4000 or higher, the flow will be turbulent. If the Reynolds' number is 2300 or lower, the flow will be laminar. If the Reynolds' number is between 2300 and 4000, the flow could be either turbulent or laminar. It may even transition back and forth between turbulent and laminar. This unpredictable flow condition should be avoided, and thus the minimum acceptable condition that ensures turbulent flow is Re# ≥ 4000.

The Reynolds' number is calculated using Formula 3-4.

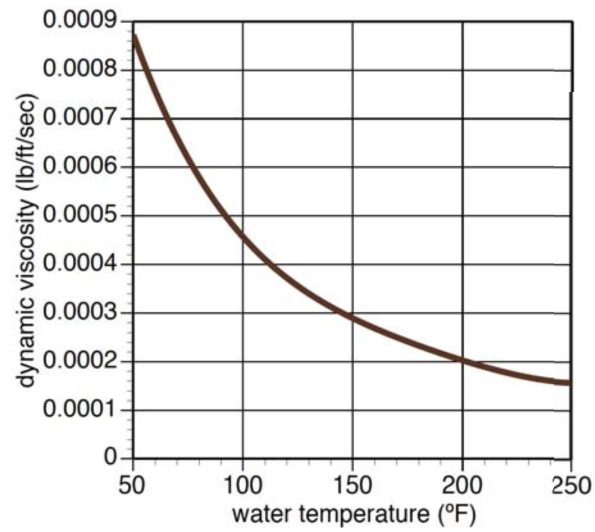
Formula 3-4

$$\text{Re\#} = \frac{vdD}{\mu}$$

Where:
v = average flow velocity of the fluid (ft/sec)
d = internal diameter of pipe (ft)
D = fluid's density (lb/ft^3)
μ = fluid's dynamic viscosity (lb/ft/sec)

This calculation requires the density of the fluid and its dynamic viscosity. Both of these fluid properties vary with the fluid's temperature. Figure 3-2 can be used to find the density of water over a wide temperature range. Figure 3-15 can be used to find the dynamic viscosity of water over this same temperature range.

Figure 3-15



It is possible to calculate the minimum flow rate that corresponds to a Reynolds' number of 4000 based on the fluid being circulated and the internal diameter of the tubing through which it passes. The relationship is given as Formula 3-5.

Formula 3-5

$$f_{min} = \frac{(117{,}503)ud}{D}$$

Where:
f_{min} = minimum flow rate that yields a Reynolds' number of 4000
u = dynamic viscosity of fluid (lb/ft/sec)
d = internal diameter of tube (inches)
D = density of fluid (lb/ft^3)

For example, what is the minimum flow rate for turbulent flow through a 1/2" type M copper tube that is conveying 120ºF water?

Solution: The exact internal diameter of a 1/2" Type M copper tube is 0.569 inches.
The density of water at 120ºF (from Figure 3-2) is 61.6 lb/ft^3.
The dynamic viscosity of water at 120ºF (from Figure 3-15) is 0.00037 lb/ft/sec.

Putting these numbers in Formula 3-5 yields:

$$f_{min} = \frac{(117{,}503)ud}{D} = \frac{(117503)(0.00037)(0.569)}{61.6} = 0.402\,gpm$$

Formula 3-5 applied to a 3/4" type M copper tube, also conveying water at 120ºF, yields a minimum flow rate of

0.57 gpm to ensure a Reynolds' number of 4000, and thus ensure turbulent flow.

The cooler a fluid, the higher its density, and the higher its dynamic viscosity. As its temperature decreases, the dynamic viscosity of any fluid that would be used in a hydronic system increases faster than its density. This implies that the minimum flow rate required for turbulence increases as the temperature of the fluid decreases.

Consider water at 50ºF used in a hydronic cooling system. Its density is 62.4 lb/ft^3, and its dynamic viscosity is 0.00087 lb/ft/sec. In a ¾" type M copper tube with an internal diameter of 0.811 inches, the minimum flow rate required for turbulence can be found using Formula 3-5:

$$f_{min} = \frac{(117{,}503)ud}{D} = \frac{(117503)(0.00087)(0.811)}{62.4} = 1.33gpm$$

This flow rate is more than twice the minimum flow rate required for turbulence when water at 120ºF passes through a ¾" type M copper tube. It is due to a large increase in dynamic viscosity compared to a relatively slight increase in density.

Designers working with chilled-water terminal units, or earth loops for geothermal heat pumps, need to be especially careful about ensuring turbulent flow for good heat transfer.

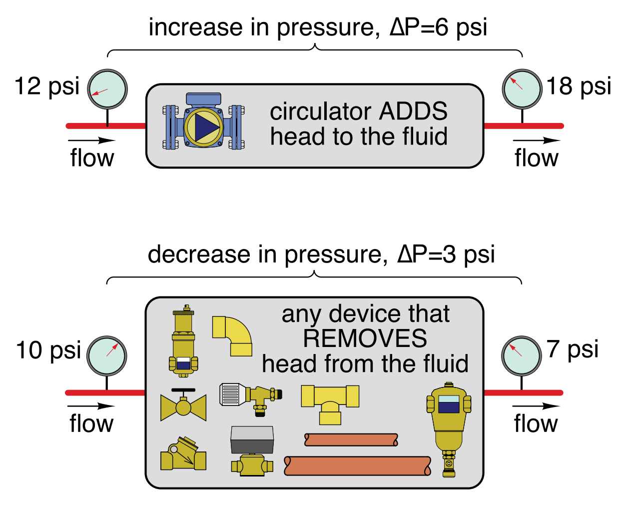
4. HEAD LOSS IN HYDRONIC CIRCUITS

Fluids in a hydronic system contain both thermal and mechanical energy. Thermal energy is sensed by a change in temperature of the fluid. For example, the hot water leaving a boiler contains more thermal energy than cooler water entering the boiler. The increase in temperature of the fluid is "evidence" that thermal energy was added to it as it passed through the boiler.

The *mechanical* energy contained in a fluid is called *head*. The units for head energy are (ft•lb/lb). The unit of ft•lb (pronounced "foot pound") is a unit of *energy*. As such, it can be converted to any other unit of energy, such as a Btu. However, engineers long ago chose to cancel the units of pounds (lb) in the numerator and denominator of this ratio, and express head in the sole remaining unit of feet. To make a distinction between feet as a unit of distance and feet as a unit of fluid energy, the latter can be stated as feet *of head*.

When head energy is added to or removed from a liquid in a closed-loop piping system, there will always be an associated change in the pressure of that fluid. Just as a change in temperature is "evidence" of a gain or loss of thermal energy, a change in pressure is evidence of a gain or loss in head energy. When head is lost, pressure decreases. When head is added, pressure increases. This concept is illustrated in Figure 4-1.

Figure 4-1



Using pressure gauges to detect changes in the head of a liquid is like using thermometers to detect changes in the thermal energy content of that liquid.

The *only device that adds head energy to hydronic systems is an operating circulator. Every other device through which flow passes causes a loss of head energy*. This happens because of friction forces between the fluid molecules, as well as friction between the fluid molecules and the components through which they are moving.

Formula 4-1 can be used to calculate the change in pressure associated with head energy being added or removed from a fluid.

$$\Delta P = H\left(\frac{D}{144}\right)$$

Where:

ΔP = pressure change corresponding to the head added or lost (psi)
H = head added or lost from the liquid (feet of head)
D = density of the fluid at its corresponding temperature (lb/ft^3)

The classic method for calculating the head energy loss associated with a fluid moving through piping is the Darcy-Weisbach formula, which is given as Formula 4-1.

Formula 4-1

$$H_L = f\frac{L}{D}\frac{v^2}{2g}$$

Where:
H_L = head loss (feet of head)
f = Moody friction factor (see Appendix B)
L = length of pipe (feet)
D = inside diameter of pipe (feet)
v = average flow velocity (ft/sec)
g = acceleration of gravity (32.2 ft/sec^2)

Establishing the Moody friction factor is the most tedious aspect in using this formula. The friction factor depends on the internal roughness of the pipe (which is estimated for various types of pipe). It also depends on the Reynolds' number of the flow, which itself depends on the velocity of the flow. Although it is possible to use mathematical iteration to establish the friction factor, such calculations can be tedious.

It is possible to simplify Formula 4-1 for the special case of *turbulent flow in smooth pipes* such as drawn *copper, PEX, PEX-AL-PEX, PE-RT, or PP-R.* The simplified relationship is given as Formula 4-2.

Formula 4-2

$$H_{\mathrm{L}} = (acL)(f)^{1.75}$$

Where:

H_L = head loss of the circuit (feet of head)
α = fluid properties factor (see Figure 4-2)
c = pipe size coefficient (see Figure 4-3)
L = total equivalent length of the circuit (feet)
f = flow rate through the circuit (gpm)
1.75 = an *exponent* applied to flow rate (f)

Formula 4-2 is valid for turbulent flow with Reynold's numbers in the range of 4000 to about 200,000. This covers most of the operating conditions found in residential and light commercial hydronic systems. At Reynold's numbers above 200,000 the equation gradually begins to underestimate head loss. At a Reynold's number of 300,000, it underestimates head loss by about 6%.

Formula 4-2 should *not* be used for systems that are predominantly constructed of iron or steel piping. Such piping has internal roughness that is greater than the smooth tubing products previously mentioned. If head loss through this rougher piping is required, it is best to directly apply the Darcy-Weisbach formula.

To calculate the head loss of a circuit using Formula 4-2, the designer must gather information from other graphs and tables.

The value of the fluid properties factor, (a), can be found in Figure 4-2.

Figure 4-2

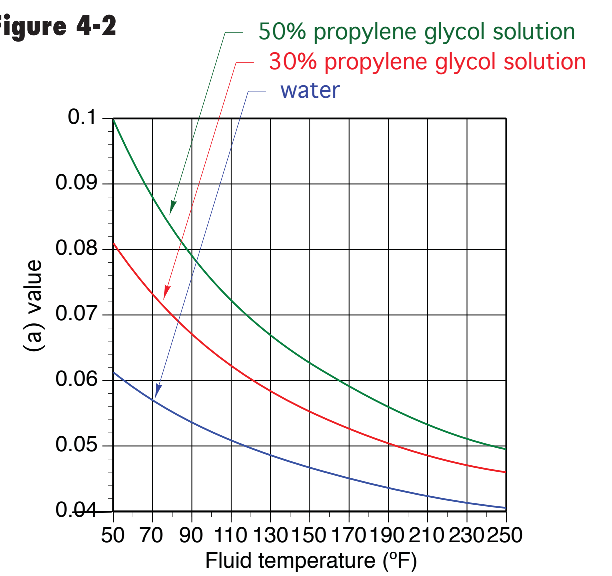


Figure 4-3

Tube (size & type)	C value
3/8" type M copper	1.0164
1/2" type M copper	0.33352
3/4" type M copper	0.061957
1" type M copper	0.01776
1.25" type M copper	0.0068082
1.5" type M copper	0.0030667
2" type M copper	0.0008331
2.5" type M copper	0.0002977
3" type M copper	0.0001278
3/8" PEX	2.9336
1/2" PEX	0.71213
5/8" PEX	0.2947
3/4" PEX	0.14203
1" PEX	0.04318
1.25" PEX	0.01668
1.5" PEX	0.007554
2" PEX	0.002104
3/8" PEX-AL-PEX	3.35418
1/2" PEX-AL-PEX	0.6162
5/8" PEX-AL-PEX	0.19506
3/4" PEX-AL-PEX	0.06379
1" PEX-AL-PEX	0.019718

For fluids not shown in Figure 4-2, the value of (a) can be calculated using Formula 4-3:

$$a=\left(\frac{D}{u}\right)^{-0.25}$$

Where:
a= fluid properties factor
D = density of the fluid (lb/ft^3)
m = dynamic viscosity of the fluid (lb/ft/sec)
-0.25 = an exponent

The value of the pipe size coefficient (c) in Formula 4-2 is a constant for a given tubing type and size. It can be found for several sizes of copper, PEX, and PEX-AL-PEX tubing from the table in Figure 4-3.

EQUIVALENT LENGTH:

To use Formula 4-2, the designer must determine the total equivalent length of the piping circuit. The total equivalent length is the sum of the equivalent lengths of all fittings, valves and other devices in the circuit, plus the total length of all tubing in the circuit.

The equivalent length of a component is the amount of tubing of the same size that would produce the same head loss as the actual component at the same flow rate. By replacing all components in the circuit with their equivalent length of piping, the circuit can be treated as if it were a single piece of pipe having a length equal to the sum of the actual pipe length, plus the total equivalent lengths of all fittings, valves or other devices *in the flow path*.

Figure 4-4

Copper tube sizes									
Fitting or valve[1]	3/8"	1/2"	3/4"	1"	1.25"	1.5"	2"	2 1/2"	3"
90-degree elbow	0.5	1.0	2.0	2.5	3.0	4.0	5.5	7.0	9
45-degree elbow	0.35	0.5	0.75	1.0	1.2	1.5	2.0	2.5	3.5
Tee (straight run)	0.2	0.3	0.4	0.45	0.6	0.8	1.0	0.5	1.0
Tee (side port)	2.5	2.0	3.0	4.5	5.5	7.0	9.0	12.0	15
B&G Monoflo® tee[2]	n/a	n/a	70	23.5	25	23	23	n/a	n/a
Reducer coupling	0.2	0.4	0.5	0.6	0.8	1.0	1.3	1.0	1.5
Gate valve	0.35	0.2	0.25	0.3	0.4	0.5	0.7	1.0	1.5
Globe valve	8.5	15.0	20	25	36	46	56	104	130
Angle valve	1.8	3.1	4.7	5.3	7.8	9.4	12.5	23	29
Ball valve[3]	1.8	1.9	2.2	4.3	7.0	6.6	14	0.5	1.0
Swing-check valve	0.95	2.0	3.0	4.5	5.5	6.5	9.0	11	13.0
Flow-check valve[4]	n/a	n/a	83	54	74	57	177	85	98
Butterfly valve	n/a	1.1	2.0	2.7	2.0	2.7	4.5	10	15.5

1. Data for soldered fittings and valves. For threaded fittings double the listed value.
2. Derived from C_v values based on no flow through side port of tee.
3. Based on a standard-port ball valve. Full-port valves would have lower equivalent lengths.
4. Based on B&G brand “flow control” valves.

Figure 4-5

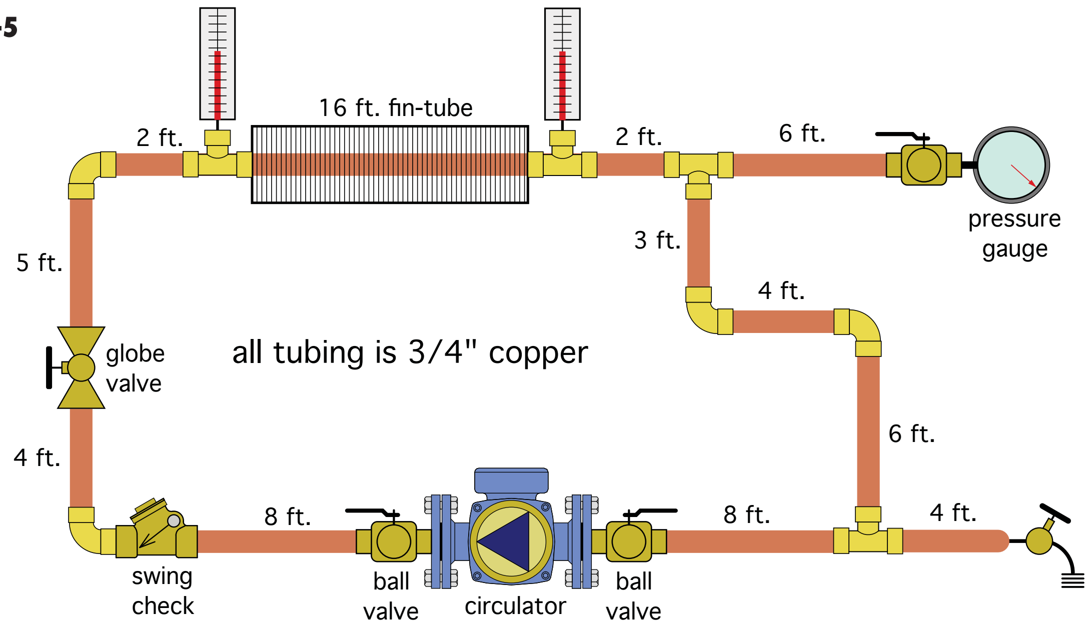


Figure 4-4 lists the equivalent lengths of most common fittings and valves.

Example: Determine the total equivalent length of the piping circuit shown in Figure 4-5.

Only components that are *in the flow path* are evaluated when determining equivalent length. Thus, the pressure gauge seen in the upper right corner of the circuit, and its associated 6 feet of piping and ball valve are not counted. Neither is the 4 feet of piping and drain valve in the lower right corner of the circuit.

The circulator is also not counted because it adds head energy to the circuit, rather than dissipating head energy from it.

Figure 4-6 shows the tally of equivalent lengths for all piping and components in the flow path of the circuit shown in Figure 4-5.

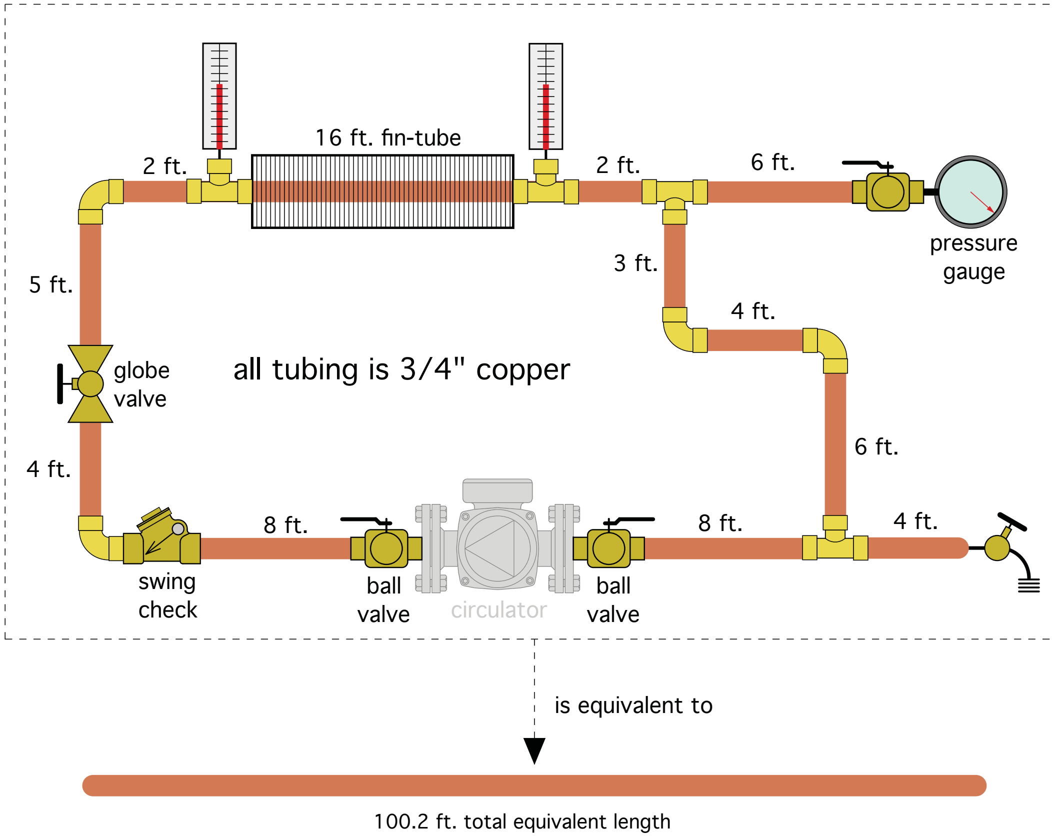
Thus, the total equivalent length of the circuit shown in Figure 4-5 is 100.2 feet of 3/4" copper tubing, as represented in Figure 4-7.

The head loss of this circuit can now be calculated using Formula 4-2, where the value of (L) is 100.2 ft.

Figure 4-6

COMPONENTS	EQUIVALENT LENGTH
3/4" straight tube	58 ft
3/4" x 90° elbows	4 x 2 ft each = 8 ft
3/4" straight run tees	2 x 0.4 ft each = 0.8 ft
3/4" side port tees	2 x 3 ft each = 6 ft
3/4" ball valves	2 x 2.2 ft each = 4.4 ft
3/4" globe valves	1 x 20 ft each = 20 ft
3/4" swing check	1 x 3 ft each = 3 ft
TOTAL EQUIVALENT LENGTH =	**100.2 ft**

Figure 4-7



Example: Determine the head loss of the circuit shown in Figure 4-7, assuming it has 140ºF water flowing through it at 5 gpm.

Solution: To use Formula 4-2, the values of (a) and (c) must first be determined.

The value of (a) for water at 140ºF is found in Figure 4-2: a = 0.0475.

The value of (c) for 3/4" copper is found in Figure 4-3: c = 0.061957

The total equivalent length was just determined to be 100.2 feet.

Putting these numbers into Formula 4-2 yields the head loss of the circuit under these operating conditions.

$$H_L = (acL)f^{1.75} = (0.0475 \times 0.061957 \times 100.2) \times (5)^{1.75} = 4.93\,feet$$

Keep in mind that this calculated circuit head loss is only valid for the stated operating conditions. Any change in the temperature or type of fluid used in the circuit will affect the value of (a), and thus change the head loss calculated using Formula 4-2. This is also true for any change in pipe size or components that affect the equivalent length of the circuit. If the circuit operates at a flow rate other than 5 gpm, the head loss will also change. Lower flow rates will decrease head loss, while higher flow rates will increase head loss.

HEAD LOSS CURVE

The flow rate through any hydronic circuit depends on the head loss characteristics of that circuit, as well as the head being added by the selected circulator. One aspect of finding that flow rate involves constructing a head loss curve for the piping circuit. The head loss curve is just a "picture" of Formula 4-2 applied to a particular piping circuit carrying a specific fluid at a specific average fluid temperature.

For a given circuit operating with a specific fluid and a specific average fluid temperature, the values of (a) (c) and (L) in Formula 4-2 are all fixed values. Therefore, Formula 4-2 can be simplified to the following:

Formula 4-4

$$H_{\mathrm{L}} = (\text{number})(f)^{1.75}$$

Under these conditions, the head loss of the specific circuit depends only on flow rate. Formula 4-4 can be graphed by selecting several flow rates, calculating the head loss at each of them and then plotting the resulting points. Once the points are plotted, a smooth curve can be drawn through them.
Example: Use the piping circuit and operating conditions from the previous example to construct a system head loss curve.

Solution: For this circuit and these operating conditions, Formula 4-2 simplifies to the following:

$$H_L = (acL)f^{1.75} = (0.0475 \times 0.061957 \times 100.2) \times (f)^{1.75} = 0.295 \times (f)^{1.75}$$

The next step is to select several random flow rates and use them in this formula to determine the corresponding head losses. This is best done using a table like the one shown in Figure 4-8.

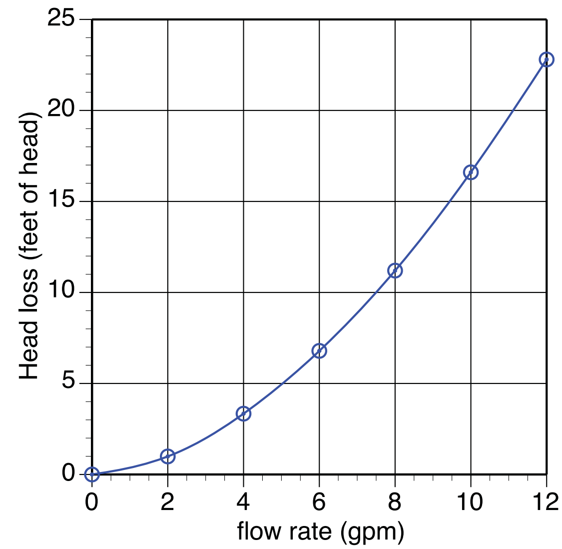
The next step is to plot these points, and draw a smooth curve through them, as shown in Figure 4-9.

This graph is called a head loss curve for the circuit. All piping circuits have a unique head loss curve. It could even be thought of as the analytical "fingerprint" of that circuit. Determining the head loss curve of a piping circuit is an essential step in properly selecting a circulator for that circuit.

Figure 4-8

flow rate (gpm)	head loss (feet)
0	0
2	0.99
4	3.34
6	6.79
8	11.2
10	16.6
12	22.8

Figure 4-9



Notice that the head loss curve in Figure 4-9 indicates zero head loss at zero flow rate. This will always be the case for closed-loop/fluid-filled piping circuits. Any changes to the piping components, tubing, fluid or average fluid temperature will change the value of (acl), and thus alter the circuit's head loss curve. If the value of (acl) increases, the head loss curve gets steeper. If the value of (acl) decreases, the head loss curve becomes shallower.

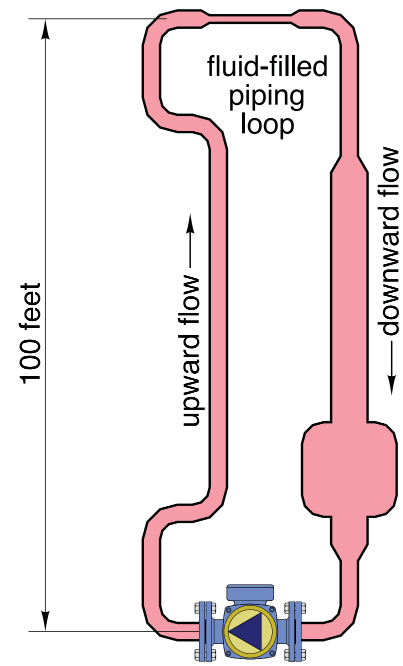
FLUID-FILLED PIPING CIRCUITS:

Most hydronic heating systems consist of *closed* piping systems. After all piping work is complete, these circuits are completely filled with fluid and purged of air. During normal operation, very little, if any, fluid enters or leaves the system.

Consider the fluid-filled piping loop shown in Figure 4-10.

Assume this loop is filled with fluid that has the same temperature at all locations. A static pressure is present at the outlet of the circulator due to the weight of the fluid column on the left side of the circuit. It might seem that the circulator would have to overcome this pressure in order to "push" fluid up the left side of the circuit. However, this is *NOT* true. The reason is that the static pressure exerted by the fluid at the inlet of the circulator, due to the fluid column on the right side of the circuit, is the same as the static pressure at the circulator's outlet.

Figure 4-10



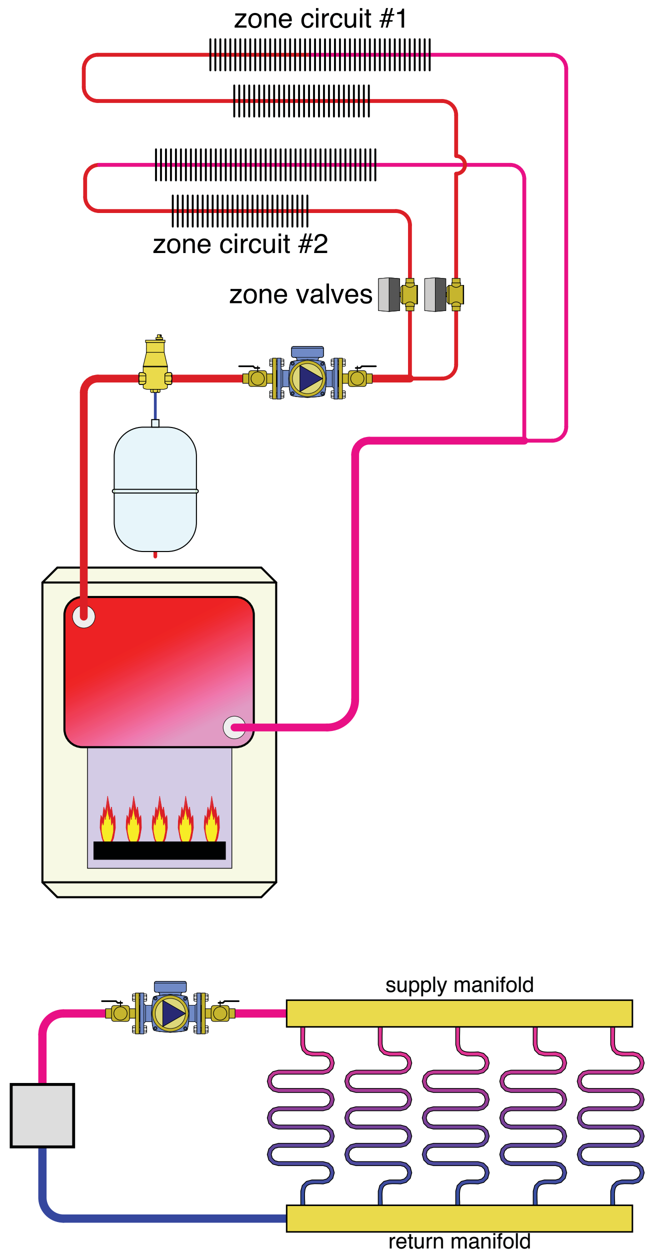
This is true regardless of the size, shape or height of the circuit, as purposefully illustrated in Figure 4-10.

When the circulator operates, the weight of fluid it moves up the left side of the circuit in a given amount of time is the same as the weight of fluid moving downward on the right side of the circuit during that same time. This must be true because the fluid has nowhere else to go within the circuit. If, for example, we assumed that 50 pounds of fluid went up the left side of the circuit in one minute, and only 49 pounds of fluid came down the right side over that minute, the question becomes: Where did the difference (1 pound) of fluid go? In a closed circuit (with no leaks), the only possible answer is nowhere. Thus, it is not possible to have different flow rates occurring simultaneously in a series piping circuit.

This balance between the weight of fluid moving up and the weight of fluid moving down also means that the circulator in a closed fluid-filled circuit is only responsible for replacing the head energy lost due to friction through the piping components. *The circulator is not responsible for "lifting" the fluid in the upward-flowing portion of the circuit.*

This explains why a small circulator can establish and maintain flow in a filled piping loop, even if the top of the loop is several stories above the circulator and contains hundreds, or even thousands, of gallons of fluid.

Figure 4-11



SERIES VERSUS PARALLEL CIRCUITS:

The simplest type of piping arrangement for a hydronic circuit is when all components are connected to form a closed loop. This is called a "series" circuit, and its head loss curve can be created using the methods discussed earlier in this section.

Although series circuits have historically been used in many small hydronic heating systems, especially traditional North American systems using fin-tube baseboard, they are not commonly used in modern systems. The principle reason is that they have to operate as a single zone.

Most modern hydronic heating systems have multiple zones. One of the most common uses a single circulator to create flow through a distribution system consisting of multiple branches. These branches begin at a supply header or supply manifold, and end at a return header or return manifold. Figure 4-11 shows two examples of hydronic circuits with parallel piping branches.

The percentage of the total system flow that passes through a given branch depends on the "hydraulic resistance" of that branch, in comparison to the hydraulic resistance of the other branches. Branches with relatively high hydraulic resistance will have a lower percentage of the total system flow passing through them, and vice versa.

Mathematically, the hydraulic resistance of a branch is the product of the fluid properties factor (a) times the pipe size coefficient (c) times the equivalent length of the branch. This can be expressed as Formula 4-5.

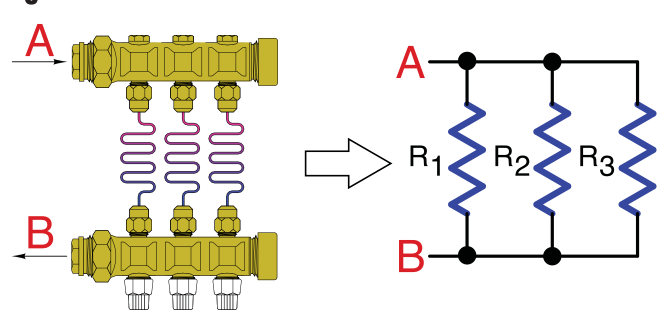
Formula 4-5

$$R_i = (acL)_i$$

Where:
Ri = hydraulic resistance of branch "i"
a = fluid properties factor
c = pipe size coefficient for branch "i"
L = total equivalent length of branch "i" (feet)

The quantities needed to calculate Ri are found the same way as described earlier in this section. The fluid properties factor (a) comes from Figure 4-2. The pipe size coefficient is found in Figure 4-3. The equivalent length (L) is found by totaling up the length of tubing plus the equivalent length of all fittings, valves or other components in the branch. The fluid properties factor only depends on the type of fluid in the system and its average temperature when the system is operating. It will be the same value for all the branches in the system.

Figure 4-12



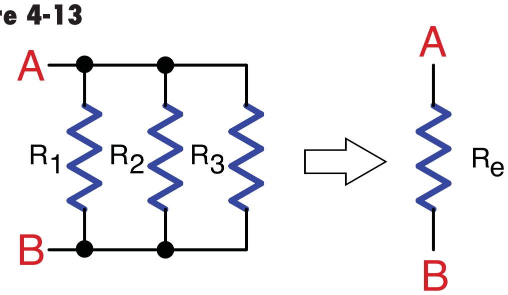
The hydraulic resistance of each branch should be calculated using Formula 4-5. These resistances can be expresses as R1 for branch 1, R2 for branch 2, R3 for branch 3, and so forth.

It is possible to view the hydraulic resistances of the branch circuits much like electrical resistors connected in parallel, as shown in Figure 4-12.

In this case, the blue resistor symbols labelled R1, R2, and R3 represent the hydraulic resistances of each tubing circuit connected between the manifolds. The black lines connecting the three hydraulic resistances together are assumed to have resistances that are so low that they are insignificant in comparison to the hydraulic resistances of the branches. This is a very close approximation of the very low hydraulic resistance of a manifold compared to that of the branch circuits connected to it.

As is possible with electrical resistances, hydraulic resistances connected in parallel can be combined into a single "equivalent resistance." This concept is shown in Figure 4-13.

Figure 4-13

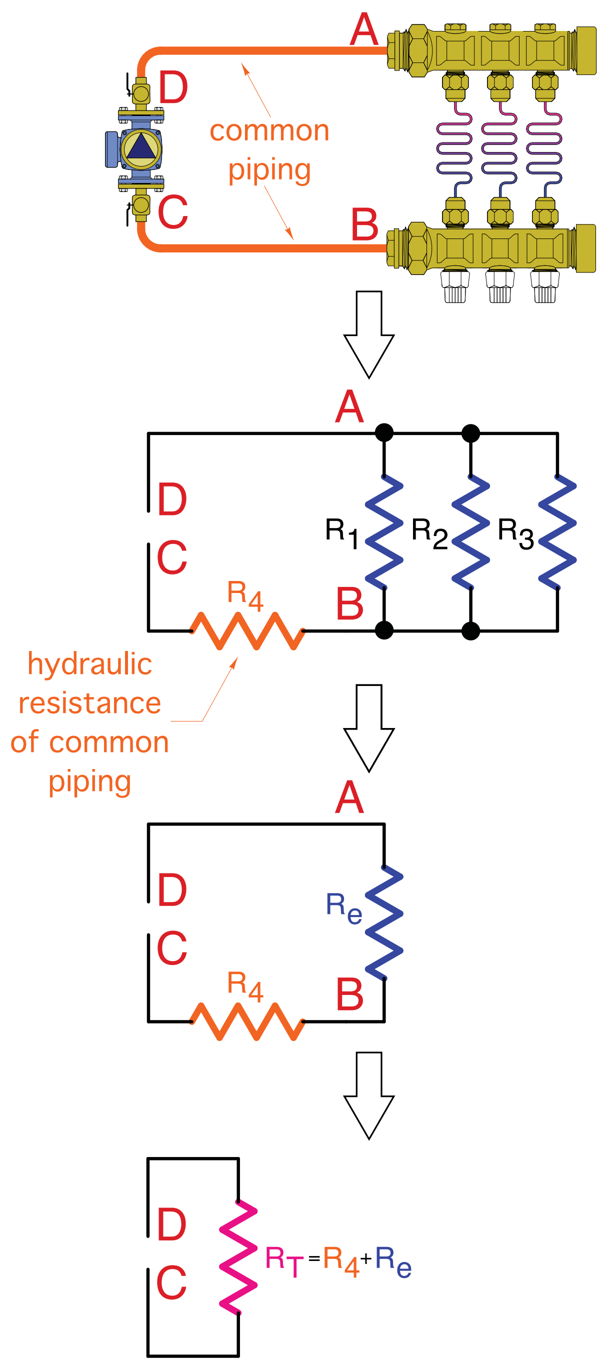


The equivalent resistance of a group of parallel hydraulic resistances represents the same hydraulic characteristics as the original parallel resistances. A given circulator connected between points A and B on the manifold station represented by the three parallel resistors produces a given flow rate and head. It would produce exactly the same flow rate and head if it were connected between points A and B, in a circuit represented by the equivalent hydraulic resistance (Re).

The concept of reducing groups of parallel hydraulic resistances into a single equivalent resistance is a very powerful technique for analyzing complex hydronic circuits. It can be used for any number of parallel hydraulic resistances connected as shown in Figure 4-12 and 4-13.

The mathematics associated with reducing parallel hydraulic resistances into a single equivalent resistance is a bit more complicated than would be the case with parallel- connected electrical resistances. Formula 4-6 can be used to find the equivalent hydraulic resistance of any number of parallel-connected hydraulic resistances.

Figure 4-14



Formula 4-6

$$R_{\text{equivalent}_{\text{parallel}}} = \left[\left(\frac{1}{R_1}\right)^{0.5714} + \left(\frac{1}{R_2}\right)^{0.5714} + \ldots + \left(\frac{1}{R_n}\right)^{0.5714}\right]^{-1.75}$$

Where:

R_1, R_2, … and R_n are the values of the branch hydraulic resistances. The minimum number of parallel resistances is 2, but this nomenclature implies that there can be any number (n) of parallel hydraulic resistances used in this formula.

Once the equivalent resistance of a group of parallel hydraulic resistances has been determined, it can be combined with the hydraulic resistance of the "common piping" in the system, as shown in Figure 4-14.

The "common piping" is the piping that connects the manifold to a circulator. Along with straight piping segments, the common piping may contain fittings, valves, a heat source or other components. The hydraulic resistance of all these components is combined into a single hydraulic resistance using Formula 4-4 and the methods associated with it. This resistance is labelled as R_4 in Figure 4-14.

Since resistance R_4 is in series with the equivalent resistance R_e, they can be added together, as would be the case with electrical resistances, to determine the total hydraulic resistance of the circuit (R_T).

The value of R_T can then be used in Formula 4-7 to create the head loss curve of the circuit.

Formula 4-7

$$H_L = (R_T)(f)^{1.75}$$

Where:

H_L = head loss of circuit (feet of head)

R_T = total hydraulic resistance of circuit

f = flow rate through circuit (gpm)

The curve shown in Figure 4-9 is an example of such a curve, where the value of R_T is 0.295.

This head loss curve can be used along with the pump curve of the circulator to determine the flow rate that will occur in the circuit when the circulator is operating. Methods for doing this will be given in the next section.

Once the flow rate through the common piping in the circuit is determined, the flow rates through the individual branch circuits can be calculated using Formula 4-8.

Formula 4-8

$$f_i = f_{total}\left(\frac{R_e}{R_i}\right)^{0.5714}$$

Where:
f_i = flow rate through parallel path i (gpm)
f_{total} = total flow rate through common piping (gpm)
R_e = equivalent hydraulic resistance of all the parallel hydraulic resistors
R_i = hydraulic resistance of parallel piping path i

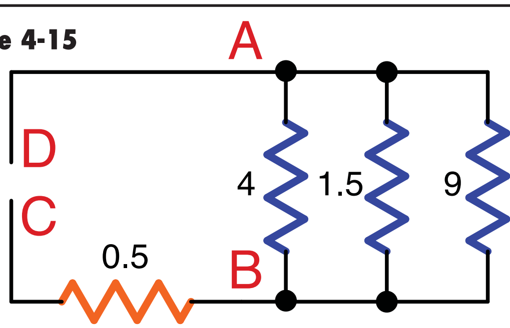
Here's an example: Assume that a 3-circuit manifold station is being analyzed. Formula 4-5 has been used to calculate the individual hydraulic resistances of each branch. They are as follows:

R_1 = 4
R_2 = 1.5
R_3 = 9

The hydraulic resistance of the common piping has also been determined to be 0.5 using Formula 4-5.

Figure 4-15 shows how these hydraulic resistances are connected.

Figure 4-15



To find the overall hydraulic resistance of the circuit, start by combining the three parallel hydraulic resistances into a single equivalent hydraulic resistance using Formula 4-6. This requires use of a calculator that can work with exponents. Any scientific calculator, including those in smart phones, can be used for this calculation.

$$R_{equivalent_{parallel}} = \left[\left(\frac{1}{R_1}\right)^{0.5714} + \left(\frac{1}{R_2}\right)^{0.5714} + \left(\frac{1}{R_n}\right)^{0.5714}\right]^{-1.75}$$

$$R_{equivalent_{parallel}} = \left[\left(\frac{1}{4}\right)^{0.5714} + \left(\frac{1}{1.5}\right)^{0.5714} + \left(\frac{1}{9}\right)^{0.5714}\right]^{-1.75}$$

$$R_{equivalent_{parallel}} = \left[0.4529 + 0.7932 + 0.2849\right]^{-1.75}$$

$$R_{equivalent_{parallel}} = \left[1.531\right]^{-1.75}$$

$$R_{equivalent_{parallel}} = 0.4745$$

Notice that the equivalent resistance is smaller than the smallest of the individual branch resistances. *This will always be true*, and is therefore one way to check that the calculations are done correctly.

This equivalent resistance can now be added to the hydraulic resistance of the common piping to get the total hydraulic resistance of the circuit.

$$R_T = 0.5 + 0.4745 = 0.9745$$

The total hydraulic resistance can now be used in Formula 4-7 to construct the head loss curve for the system.

$$H_L = (R_T)(f)^{1.75} = (0.9745)(f)^{1.75}$$

To graph this formula, just pick a few random values for flow rate (f), and calculate the corresponding head loss (H_L). Figure 4-16 shows some values for flow rate and their corresponding calculated head losses.

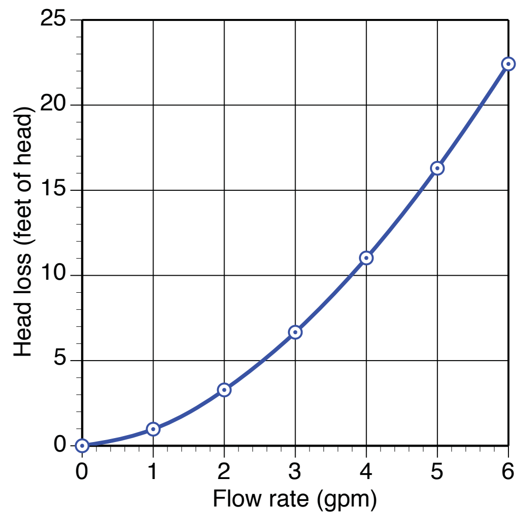
The next step is to plot these points and draw a smooth curve through them to establish the head loss curve for the system. This is shown in Figure 4-17.

Assume that the methods to be discussed in section 5 have determined that a specific circulator will create a flow rate in the common pipe of the circuit of 5.5 gpm. Formula 4-8 can now be used to determine how this total flow rate divides up among the three branches. When doing this, it's

Figure 4-16

Flow rate (f) (gpm)	Head loss (H_L) (feet of head)
0	0
1	0.9745
2	3.28
3	6.66
4	11.02
5	16.29
6	22.42

Figure 4-17



very important to use the equivalent resistance of just the parallel resistors, (R_e), in Formula 4-8, *and not the total hydraulic resistance of the circuit (R_T).*

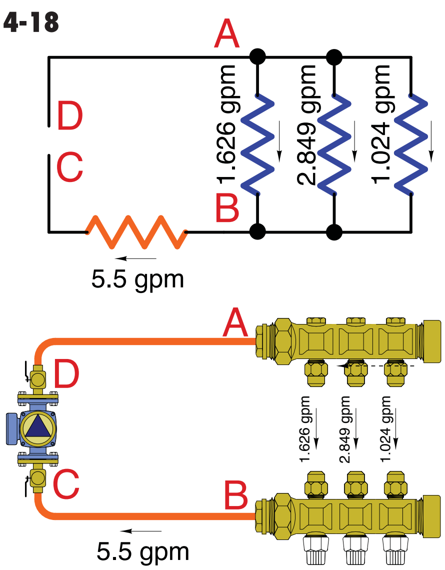
$$f_1 = 5.5\left(\frac{0.4745}{4}\right)^{0.5714} = 1.626\,gpm$$

$$f_2 = 5.5\left(\frac{0.4745}{1.5}\right)^{0.5714} = 2.849\,gpm$$

$$f_3 = 5.5\left(\frac{0.4745}{9}\right)^{0.5714} = 1.024\,gpm$$

Figure 4-18 shows these calculated branch flow rates on the hydraulic resistance diagram and the original piping schematic.

Figure 4-18



As a check, add up the branch flow rates. Their total should equal, or be very close to, the flow rate in the common piping. In this case, the branch flow rates add up to 5.499 gpm, which is very close to 5.5 gpm. The slight difference is due to rounding off the numbers for the branch flow rates.

Notice that the branch with the greatest hydraulic resistance (R_3 = 9) has the lowest of the three branch flow rates passing through it. The branch with the lowest hydraulic resistance (R_2 = 1.5) has the highest of the three branch flow rates. This is intuitive, but it also serves as a "reality check" that the calculations are correct.

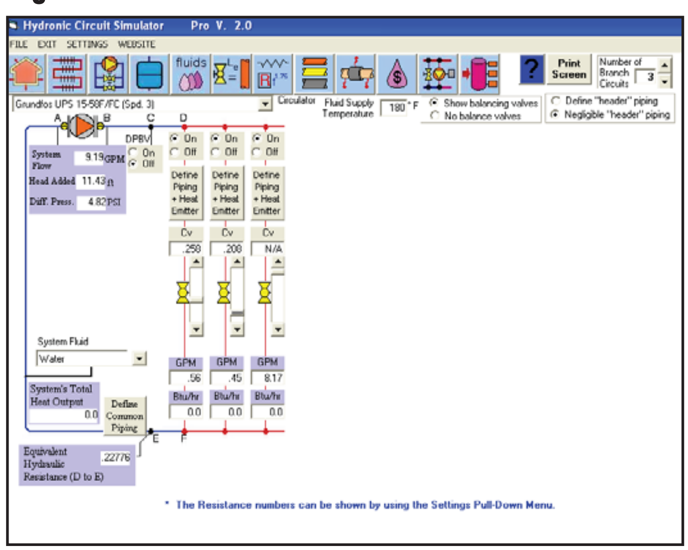
SOFTWARE-BASED CIRCUIT ANALYSIS:

Although the calculations just discussed are precise, they require a considerable amount of effort. The greater the amount of mathematics required, the greater the chance of error. Complex calculations also require more time to complete, especially if they have to be done several times to investigate different design options.

The methods discussed in this section have been incorporated into software. One example, the *Hydronics Design Studio*, is shown in Figure 4-19.

This software uses the same underlying theory and mathematics that have been discussed. However, it allows rapid changes for design options such as the configuration of the branches and common piping, the fluid used in the system and its average operating temperature, user-specified fittings and valves, and a wide range of potential circulators. This software, or other software based on the methods presented in this section, allow for rapid "iterative" analysis of both simple and complex piping systems.

Figure 4-19



5. HYDRONIC CIRCULATORS

There are hundreds of circulators available in North America for use in hydronic systems. They range from the "classic" 3-piece circulator shown in Figure 5-1a, to the state-of-the-art microprocessor-controlled circulator shown in Figure 5-1b.

Figure 5-1a

Figure 5-1b

This broad range of products makes it possible to match the performance of the circulator to the flow and head requirements of almost any practical hydronic system. This is not just a matter of choosing a circulator that is assumed to operate at some predetermined flow rate. All circulators are capable of producing a wide range of flow rates based on how they interact with the piping system into which they are installed. A given circulator might produce a flow rate of 4 gpm in one hydronic circuit and 10 gpm in another.

To understand the proper method of selecting a circulator, it's first important to understand what a circulator does. All circulators can be thought of as "energy converters." Specifically, they convert electrical energy into mechanical energy. In the context of hydronic systems, that mechanical energy is called "head." Section 4 has already described head as the number of foot pounds (ft•lb) of mechanical energy contained in each pound (lb) of fluid.

Mathematically, head can be stated as follows:

$$\text{head} = \frac{ft \cdot \cancel{lb}}{\cancel{lb}} = \text{ft. of head}$$

The units of (lb) cancel out in the top and bottom of the fraction, and thus the resulting units are stated as "feet of head." Still, it's convenient to think of head as the number of foot pounds of mechanical energy contained in each pound of fluid within the system.

The methods given in section 4 allow a designer to determine the head *loss* associated with flow through a given component or a complete hydronic system. *<u>Every component through which flow passes dissipates some head energy from the fluid.</u>* This is the result of friction between fluid molecules, as well as between these molecules and the surfaces or components they are moving through. Friction ultimately converts mechanical energy into heat.

In contrast, head is *added* to fluids as they flow through an operating circulator. In a hydronic system, an operating circulator is the only device that adds head energy to the fluid.

The amount of head energy a given circulator adds to a fluid depends on the flow rate passing through it. The greater the flow rate, the lower the amount of head energy added to each pound of fluid. This is a characteristic of all hydronic circulators and can be represented graphically as a "pump curve." An example of a pump curve for a small hydronic circulator is shown in Figure 5-2.

Figure 5-2

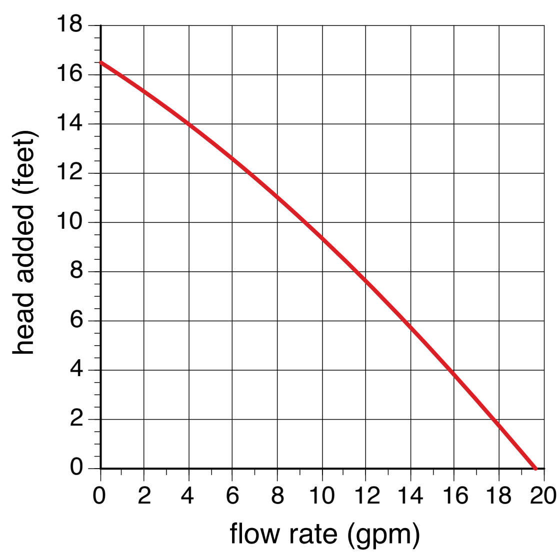
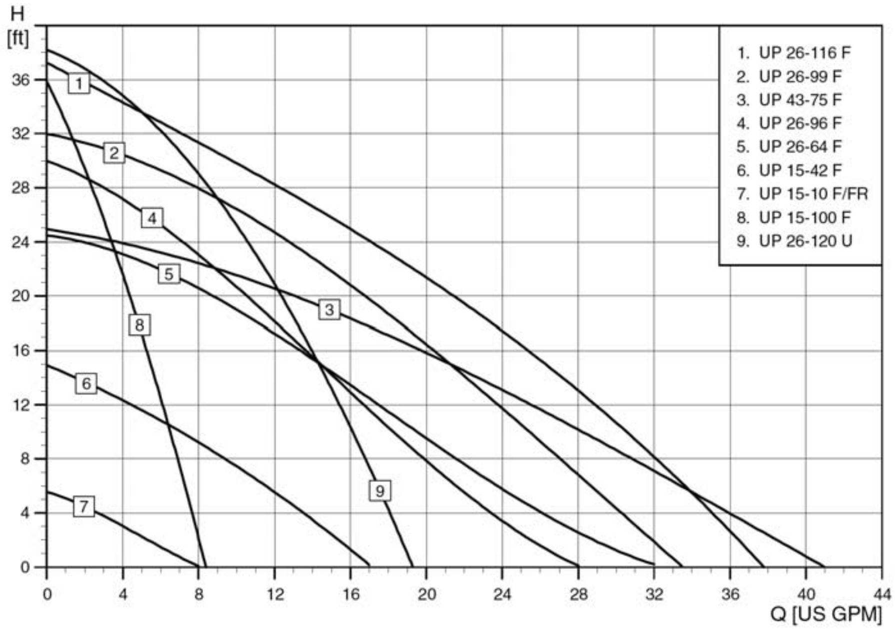


Figure 5-3



Courtesy of Grundfos

The point where the head loss curve of the circuit crosses the pump curve of the circulator is called the "operating point." This is where hydraulic equilibrium occurs.

The flow rate in the circuit at hydraulic equilibrium is found by drawing a vertical line from the operating point down to the horizontal axis. The head input by the circulator (or head loss by the piping system) can be found by extending a horizontal line from the operating point to the vertical axis.

A performance comparison of several "candidate" circulators within a given piping circuit can be made by plotting their individual pump curves on the same set of axes as that circuit's head loss curve. The intersection of each circulator's pump curve with the circuit's head loss curve indicates the operating point for that particular circulator. By projecting vertical lines from these operating points down to the horizontal axis, the designer can determine the flow rate each circulator would produce within the circuit, as shown in Figure 5-5.

The pump curve of a circulator is developed from test data using water in the temperature range of 60° to 80°F. For fluids with higher viscosities, such as glycol-based antifreeze solutions, there is a very small decrease in head and flow rate capacity of the circulator. However, for the fluids and temperature ranges commonly used in hydronic heating systems, this variation is so small that it can be ignored. Thus, for residential and light commercial hydronic systems, *pump curves may be considered to be independent of the fluid being circulated.*

Pump curves are extremely important in matching the performance of a circulator to the flow requirements of a piping circuit or a complete piping system. All circulator manufacturers publish these curves for the circulator models they offer. In many cases, the pump curves for multiple circulators are plotted on the same graph so that performance comparisons can be made. Figure 5-3 shows an example of a family of pump curves.

HYDRAULIC EQUILIBRIUM:

It is possible to predict the flow rate that will develop when in a specific circulator is installed in a specific hydronic circuit. *That flow rate will be such that the head energy added by the circulator is exactly the same as the head energy dissipated by the piping circuit.* This condition is called "hydraulic equilibrium." The flow rate at hydraulic equilibrium is found by plotting the head loss curve of the circuit on the same graph as the pump curve for the circulator. An example of this is shown in Figure 5-4.

Figure 5-4

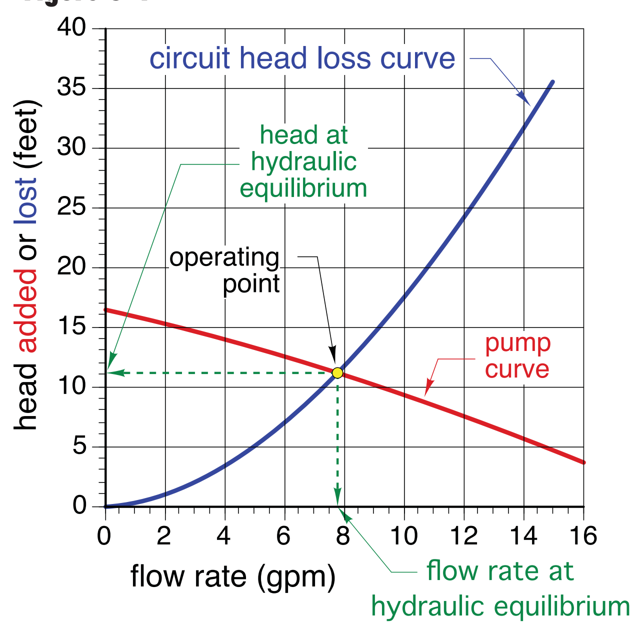
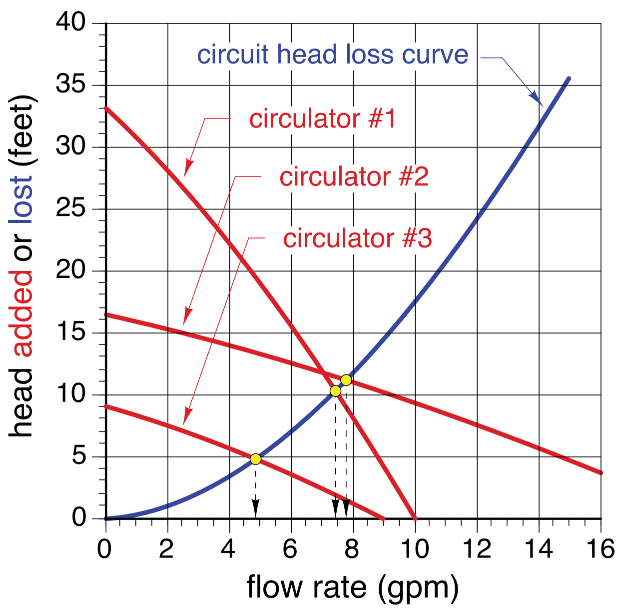


Figure 5-5



Notice that even though the curves for circulators 1 and 2 are quite different, they intersect the circuit's head loss curve at almost the same point. Therefore, these two circulators would yield very similar flow rates of about 7.5 gpm and 7.8 gpm in this circuit. The flow rate produced by circulator 3, about 5 gpm, is considerably lower.

For any combination of piping circuit and circulator, hydraulic equilibrium will be established within a few seconds of turning on the circulator. Once established, the system will remain at the flow rate corresponding to hydraulic equilibrium *unless* something occurs that affects either the head loss curve, or the pump curve.

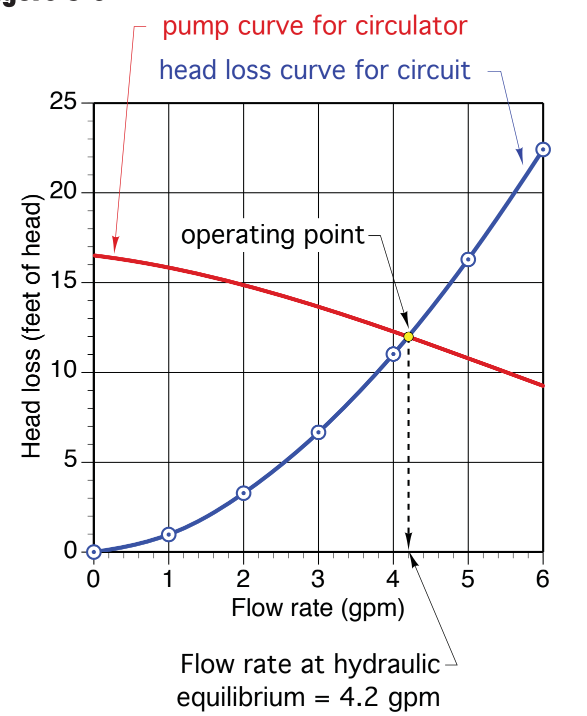
Examples of what could change the circuit head loss curve include:
• The type of fluid in the circuit changes
• The temperature of the fluid changes
• Changes are made to the pipe type, pipe size or to the components in the piping circuit
• Valves setting are changed

Examples of what could change the pump curve include:
• Change to a different circulator
• Change the circulator to different speed setting
• Altering the circulator's impeller

Example: Determine the flow rate for the circuit analyzed in section 4, assuming it operated with a circulator having the pump curve shown in Figure 5-6.

Solution: The head loss curve for this piping circuit, as determined in section 4, is shown in Figure 5-6. So is the pump curve for a specific circulator. The point where these curves cross is the operating point, where the head added by the circulator exactly balances the head dissipated by the circuit. In this case, that occurs when the flow through the circulator, and hence through the common piping, is 4.2 gpm.

Figure 5-6



USING A PUMP CURVE TO ESTIMATE FLOW RATE:

A pump curve can be used in combination with a reading of differential pressure to estimate the flow rate through an operating circulator.

Recall that Formula 4-1 gave the relationship between head added by a circulator and the associated increases in pressure across that circulator. If Formula 4-1 is rearranged as in Formula 5-1, the pressure increase across the circulator can be converted to the head energy added to the fluid by the circulator.

Formula 5-1

$$H_{added} = \Delta P_{gain}\left(\frac{144}{D}\right)$$

Where:

ΔP_{gain} = pressure increase measured across the circulator (psi)
H_{added} = head added to the fluid by the circulator (feet of head)
D = density of the fluid at its corresponding temperature (lb/ft^3)

Once the head added by the circulator is known, it is easy to find this head value on the vertical axis of the pump curve, draw a horizontal line from that value to the pump curve, and then a vertical line from this intersection down to the horizontal axis to read the flow rate through the circulator.

For example, a pressure gauge on the inlet of a circulator reads 15 psi. Another pressure gauge on the outlet of the circulator reads 20.5 psi. The pump curve for the circulator is given by Figure 5-2. The fluid passing through the circulator is water at 120ºF. Estimate the flow rate through the circulator under these conditions.

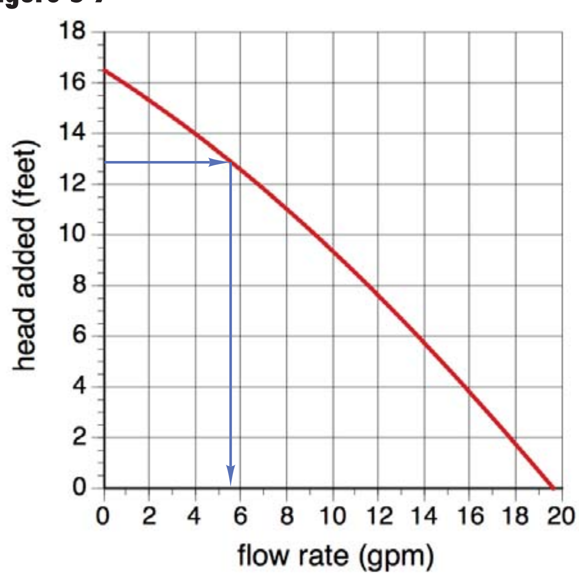
Solution: The density of water at 120ºF is 61.6 lb/ft^3, as determined using Figure 3-2.

The differential pressure across the circulator is 20.5 - 15 = 5.5 psi

It is now possible to convert the differential pressure gain across the circulator to the corresponding head using Formula 5-1.

$$H_{added} = \Delta P_{gain}\left(\frac{144}{D}\right) = 5.5\left(\frac{144}{61.6}\right) = 12.9\,ft$$

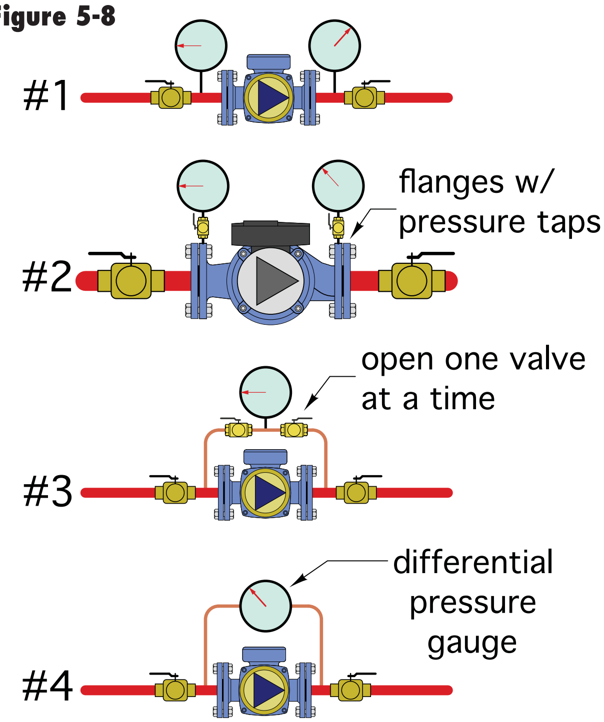
Figure 5-7



Starting at a head of 12.9 feet on the vertical axis of Figure 5-2, draw a horizontal line to intersect the pump curve. Then draw a vertical line down to the horizontal axis to determine the flow rate. The result is an estimated flow rate of 5.5 gpm, as shown in Figure 5-7.

There are several possible ways to get the differential pressure measurement across a circulator, as shown in Figure 5-8.

Figure 5-8



Configuration #1 shows two pressure gauges mounted into the piping just upstream and downstream of the circulator. The differential pressure is the pressure on the discharge side of the circulator minus the pressure on the intake side. This method requires accurate pressure gauges in both locations, since any error in either gauge will affect the differential pressure value.

Configuration #2 is common on medium and large circulators. The intake and discharge flanges of these circulators are often drilled and tapped to receive a 1/8" or 1/4" MPT pressure gauge. Some designers also specify small isolation ball valves between each pressure gauge and the flange tapping. These allow either gauge to be replaced if necessary without stopping and isolating the circulator.

Configuration #3 uses two small isolating ball valves on either side of a single pressure gauge. Only one of these ball valves is opened at a time to read the pressure on either the intake or discharge side of the circulator. This method has the advantage that any consistent error in the pressure gauge will be cancelled out when the intake pressure is subtracted from the discharge pressure.

Configuration #4 uses a direct-reading differential pressure gauge with two tappings. Each of the two connections on this gauge connect to the piping just upstream and downstream of the circulator flanges. The isolation valves for the circulator are outside of these tees to eliminate the slight pressure drop they create from affecting the differential pressure reading.

CIRCULATOR EFFICIENCY:

It has already been stated that circulators can be thought of as electrical-to-mechanical energy converters. The mechanical energy they impart to the water is called head.

Like most energy converters, a circulator is not 100% efficient in converting electrical energy into head energy. Some reasons for this are attributable to the hydraulic characteristics of the circulator—for example, the friction of the water as it passes through the circulator's volute and impeller. There are also recirculation losses where some of the fluid coming off the periphery of the impeller internally migrates back to the inlet of the impeller.

The net effect of all the inefficiencies in the mechanical and hydraulic portions of a circulator can be represented by a number called pump efficiency. That number can be calculated using Formula 5-2.

Formula 5-2

$$n_{pump} = \frac{f(\Delta P)}{1716(hp)}$$

Where:

n_{pump} = pump efficiency (decimal %)
f = flow rate through circulator (gpm)
ΔP = differential pressure across the operating circulator (psi)
hp = mechanical horsepower supplied to the impeller shaft (horsepower)

This formula shows that the mechanical/hydraulic efficiency of the pump depends on both flow rate and differential pressure. Although Equation 5-2 is easy to derive from basic physics, it is not easy to use because measurements of input horsepower to the pump's shaft are difficult to obtain outside of a laboratory.

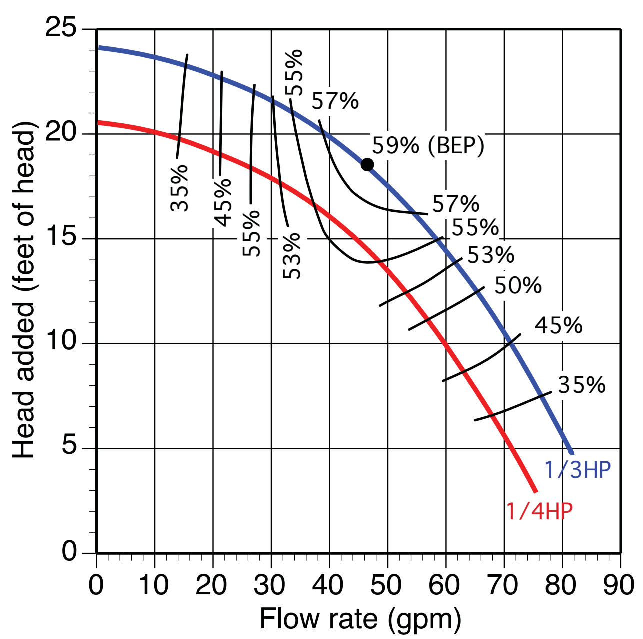
To provide efficiency information to designers, most manufacturers plot the pump efficiency of their medium- and larger-size pumps as a group of contour lines overlaid on the pump curve(s), as shown in Figure 5-9.

The red and blue curves are pump curves, in this case for a pump with two different motor options. The black contour lines represent the pump efficiency. They indicate the percentage of shaft input power that is converted to head energy and imparted to the fluid. They do not include any inefficiency associated with the electric motor driving the impeller shaft.

The pump efficiency is highest (59%) at a point that is near the middle of the pump curve. This is typical of all centrifugal pumps. This point is aptly named the "best efficiency point," or BEP. The ideal situation is for the head loss curve of the hydronic circuit to pass through the pump curve at the BEP. This is where the pump is most efficient in converting shaft input power to head energy.

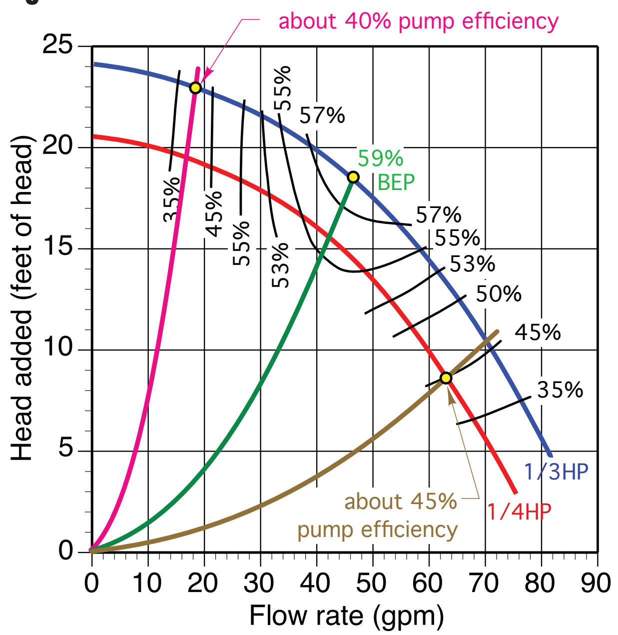
The black contour lines each represent a specific pump efficiency. If the head loss curve of a hydronic circuit crosses the pump curve at the same place the efficiency contour does, that contour indicates the pump efficiency at which the pump will be operating. If the head loss curve crosses the pump curve between efficiency contours, the pump efficiency can be estimated by interpolation.

Figure 5-9



For example, the pink head loss curve shown in Figure 5-10 passes through the blue pump curve midway between the contour lines labelled 35% and 45%. Therefore the pump efficiency of the pump operating at this condition is about 40%. This implies that 40% of the mechanical power supplied to the impeller shaft is being converted to head energy and imparted to the fluid. The remaining 60% of the shaft input power is being converted to heat through the various friction forces occurring within the fluid or the mechanical bearing and seals of the pump.

Figure 5-10



The green head loss curve in Figure 5-10 intersects the blue pump curve at the BEP, and thus the pump, with the 1/3 HP motor, operating in a circuit with this head loss curve, would be operating at its highest possible pump efficiency of 59%.

The brown pump curve intersects the lower red pump curve at the contour line marked 45%, and thus indicates a pump efficiency of 45% if operating within a hydronic circuit having this head loss curve.

WIRE-TO-WATER EFFICIENCY:

The pump efficiencies represented by the contour lines in Figure 5-9 and 5-10 do not include the efficiency of the electric motor (or other "prime mover") that turns the impeller shaft. This is common practice because some medium and large pumps can be fitted with several motor options, each of which may have a different electrical-to-mechanical efficiency.

Most small to mid-size circulators are sold with a specific motor. This includes all wet-rotor circulators, as well as some 3-piece and 2-piece circulators.

When the motor is an integral part of the circulator, it is convenient to express the overall efficiency of converting electrical input power to head. This overall efficiency includes both the mechanical/hydraulic characteristics of the pump and the electrical-to-mechanical conversion efficiency of the motor. It is called "wire-to-water" efficiency.

The wire-to-water (w/w) efficiency of a circulator can be calculated using Formula 5-3.

Formula 5-3

$$n_{w/w} = \frac{0.4344\, f(\Delta P)}{w}$$

Where:

$n_{w/w}$ = wire-to-water efficiency of the circulator (decimal percent)
f = flow rate through the circulator (gpm)
ΔP = pressure differential measured across the circulator (psi)
w = input wattage required by the motor (watts)
0.4344 = units conversion factor

Figure 5-11 shows the pump curve of a small wet-rotor circulator, along with another curve that indicates the circulator's calculated wire-to-water efficiency. Peak wire-

Figure 5-11

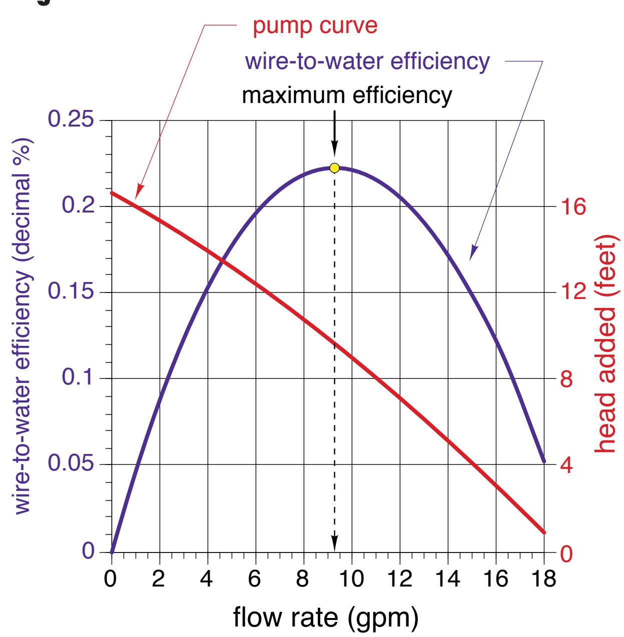
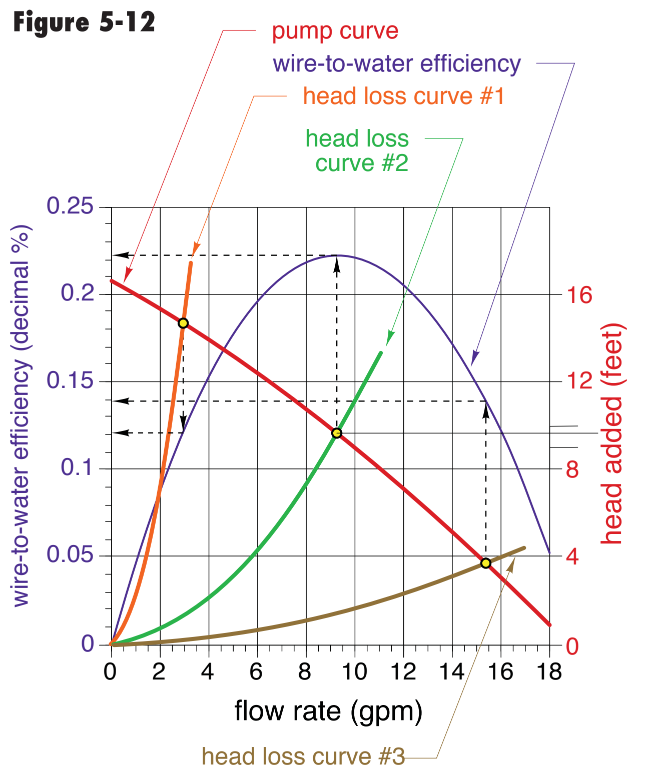


Figure 5-12



to-water efficiency occurs at a flow rate that is close to the center of the pump curve. For this particular circulator, it occurs at a flow rate of about 9.3 gpm. Ideally, the intersection of the pump curve and the circuit's head loss curve will be at or near this point.

The wire-to-water efficiency at which this wet-rotor circulator operates depends on where the head loss curve of the circuit crosses the pump curve. Figure 5-12 shows three head loss curves that represent three different hydronic circuits.

Head loss curve #1 intersects the pump curve at a flow rate of about 3 gpm. A vertical line is drawn from this operating point to the wire-to-water efficiency curve. A horizontal line drawn from that intersection to the left vertical axis indicates that the wire-to-water efficiency of the circulator operating in this circuit will only be about 0.12 (e.g., 12%). This means that the circulator is only converting about 12% of the electrical energy it takes in to head energy. In comparison, head loss curve #2 intersects near the middle of the pump curve. A vertical line from this operating point to the wire-to-water efficiency curve, and then to the left vertical axis, indicates a wire-to-water efficiency just over 22%. Although still relatively low in comparison to other efficiencies , such as the thermal efficiency of a modern boiler, it is still 83% higher than the wire-to-water efficiency associated with head loss curve #1. Head loss curve #3 is a very shallow curve. The wire-to-water efficiency associated with the circulator operating in a system with this head loss curve is only about 14%.

The preferable situation is to match circulator pump curves with circuit head loss curves so that their intersection is near the middle of the circulator's pump curve. A guideline is to select circulators so that this intersection falls within the middle third of the pump curve. As Figure 5-12 shows, operating a circulator near either end of its pump curve, while possible, forces the circulator to operate with very low wire-to-water efficiency.

ESTIMATING CIRCULATOR INPUT POWER:

It's possible to estimate the electrical power required by a circulator operating at a specified condition. This can be done using Formula 5-4.

Formula 5-4

$$w = \frac{0.4344\, f(\Delta P)}{n_{\text{wire-to-water}}}$$

Where:

$n_{w/w}$ = wire-to-water efficiency of the circulator (decimal percent)
f = flow rate through the circulator (gpm)
ΔP = pressure differential measured across the circulator (psi)
w = input wattage required by the motor (watts)
0.4344 = units conversion factor

Formula 5-4 is a simple algebraic manipulation of Formula 5-3. It requires values for flow rate and differential pressure. It also requires a value for the wire-to-water efficiency of the circulator operating under these conditions. The latter is hard to measure and most likely would have to be inferred based on a known efficiency curve and the measured flow rate and differential pressure.

For example, head loss curve #2 in Figure 5-12 crosses the circulator's pump curve at a flow rate of about 9.3 gpm. The wire-to-water efficiency at this point is about 22%, and the head added is about 9.7 feet, as shown on the right side vertical axis. To estimate the electrical power input to the circulator under this condition, the head added needs to be converted to differential pressure. This can be done using Formula 4-1, which requires specification of the fluid being circulated.

Assuming water at 120ºF as the circulating fluid, the differential pressure across the circulator at this operating point is:

$$\Delta P = H_{added}\left(\frac{D}{144}\right) = 9.7\left(\frac{61.6}{144}\right) = 4.15\,psi$$

The electrical input power can now be calculated using Formula 5-4.

$$w = \frac{0.4344\, f(\Delta P)}{\eta_{\text{wire-to-water}}} = \frac{0.4344\ (9.3)(4.15)}{0.22} = 76.2\,watt$$

CAVITATION:

All liquids will boil if the pressure exerted on them is less than their vapor pressure. The vapor pressure of liquids varies with the liquid's temperature. The higher the temperature, the higher the pressure that must be exerted on the liquid to prevent it from boiling.

Boiling involves the formation of vapor pockets in the liquid. These pockets look like bubbles, but should not be confused with air bubbles. They will form even in water that has been completely deaerated, instantly appearing whenever the liquid's pressure drops below the vapor pressure corresponding to its current temperature.

The density inside a vapor pocket is about 1,500 times lower than the surrounding liquid water. This is comparable to the change in density that would occur if of a single kernel of popcorn expanded to the size of a baseball.

If the pressure on the liquid then increases above the vapor pressure, the vapor pockets instantly collapse in a process known as implosion. When this occurs, small amounts of liquid are accelerated into "micro-jets" that can attain velocities on the order of twice the speed of sound. When liquid at this speed impacts a metal surface, it can erode that surface, eventually causing serious damage to the impeller and volute of a circulator. A circulator operating under such conditions is said to be "cavitating." This is a condition that must be avoided through proper design.

NET POSITIVE SUCTION HEAD AVAILABLE (NPSHA):

The best way of preventing cavitation is to know what operating conditions allow it to occur, and then design the system to avoid these conditions.

A standardized method has been developed for predicting the conditions that cause cavitation in circulators. This method requires the calculation of a quantity called the Net Positive Suction Head Available, or NPSHA. The NPSHA is a precise description of the state of the fluid within a specific piping system as it *enters* the circulator. It includes the effects of temperature, pressure, velocity and the fluid's vapor pressure in a single number, which can be calculated using Formula 5-5.

Formula 5-5

$$\text{NPSHA} = \frac{v^2}{64.4} + \left((p_\text{i} + 14.7 - p_\text{v}\right)\left(\frac{144}{D}\right)$$

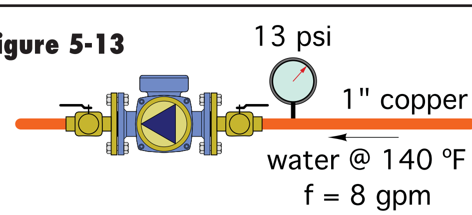
NPSHA = net positive suction head available at the circulator inlet (ft of head)
v = velocity of the liquid in the pipe entering the circulator (ft/sec)
p_i = gauge pressure of the moving fluid measured at the circulator entrance (psig)
p_V = vapor pressure of the liquid as it enters the circulator (psi absolute)

The NPSHA is the difference between the total head of the fluid at the inlet of the circulator and the head at which the fluid will boil.

NPSHA is a characteristic of the piping system and the properties of the fluid being circulated. It is NOT dependent on the circulator being used.

Example: Determine the NPSHA for the piping system shown in Figure 5-13. Assume the water temperature is 140ºF.

Figure 5-13



Solution: A number of quantities need to be determined before using Equation 5.5 to calculate the NPSHA.

The velocity of the water in the pipe entering the circulator is found using the formula for 1" copper tubing from Figure 3-12:

$$v = 0.367f = 0.367(8) = 2.94 \text{ ft/sec}$$

The vapor pressure of water is a function of its temperature, and can be read from Figure 5-14. Note that this graph gives the vapor pressure as an absolute pressure.

At a temperature of 140ºF, the vapor pressure of water is 2.9 psia.

Figure 5-14

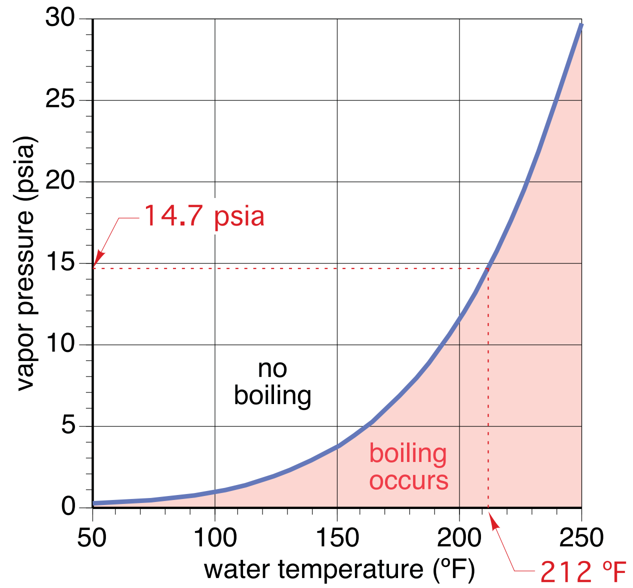
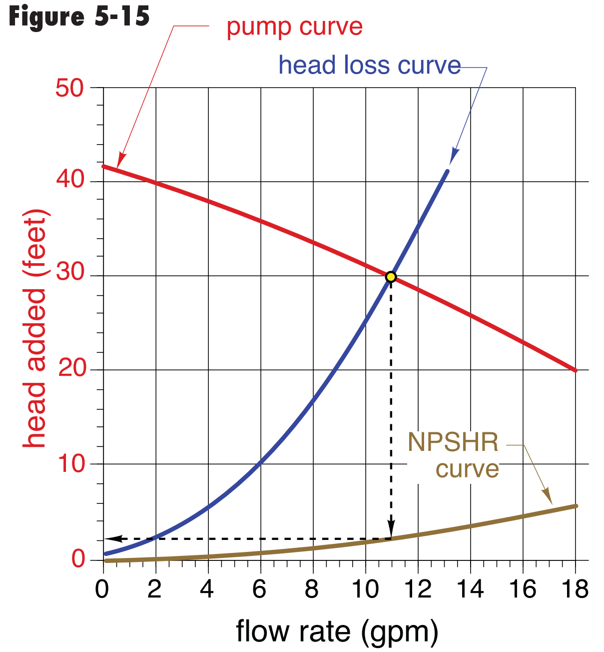


Figure 5-15



The density of water at 140°F is found from Figure 3-2: D = 61.35 lb/ft^3.

Substituting these values into Formula 5.5 yields:

$$\text{NPSHA} = \frac{v^2}{64.4} + \left((p_i + 14.7 - p_v\right)\left(\frac{144}{D}\right) = \frac{2.94^2}{64.4} + \left((13 + 14.7 - 2.9\right)\left(\frac{144}{61.3}\right) = 58.4\,ft$$

By itself, this number is not very useful. However, it will soon be compared to another number that will determine if cavitation will occur.

NET POSITIVE SUCTION HEAD REQUIRED (NPSHR):

Every circulator has a minimum value of net positive suction head that is required in order for that circulator to operate without cavitation. That value is called the *Net Positive Suction Head Required*, or NPSHR. Its value depends on the design of the circulator, as well as the flow rate through it.

It's important to distinguish between Net Positive Suction Head *Available* (NPSHA), which is totally determined by the piping system and the fluid, and the Net Positive Suction Head *Required*, which is strictly determined by the circulator characteristics.

Circulator manufacturers determine values of NPSHR by lowering the NPSHA to an operating circulator until cavitation occurs. This is done at several flow rates so that a curve can be plotted. The NPSHR curve is sometimes plotted on the same graph as the pump curve, as shown in Figure 5-15.

Avoiding cavitation is simply a matter of ensuring the piping circuit's NPSHA exceeds the NPSHR of the circulator under all possible operating conditions.

The higher the NPSHA is in comparison to the NPSHR, the wider the safety margin against cavitation. Common design practice is to ensure that the circuit's NPSHA is at least 2 feet of head higher than the NPSHR of a circulator.

In Figure 5-15, the operating point where the circuit's head loss curve crosses the pump curve corresponds to a flow rate of approximately 11 gpm. Drawing a line from this point down to the circulator's NPSHR curve shows that a NPSHR of about 2 feet is required at that flow rate to prevent cavitation. The system designer should therefore calculate the NPSHA for the circuit and ensure that it is at least 4 feet (e.g., 2 foot of head safety margin above the NPSHR of the circulator).

GUIDELINES FOR AVOIDING CAVITATION:

There are several qualitative guidelines for avoiding cavitation. Observing these guidelines typically widens the safety margin against the onset of cavitation. Most strive to make the NPSHA of the circuit as high as possible. The degree to which each guideline affects the NPSHA of the circuit can be evaluated using Formula 5.5.

- Keep the static pressure on the system as high as practical.
- Keep the fluid temperature as low as practical.

• Always place the expansion tank near the inlet side of the circulator.
• Keep the circulator low in the system to maximize static pressure at its inlet.
• Do not place any components with high flow resistance (especially flow-regulating valves) near the inlet of the circulator.
• If the system has a static water level, such as a partially filled tank, keep the inlet of the circulator as far below this level as possible.
• Be especially careful in the placement of high-head circulators or close-coupled series circulator combinations because they create greater pressure differentials.
• Provide a straight length of pipe at least 12 pipe diameters long upstream of the circulator's inlet.
• Install a good air separator in the system.

PLACEMENT OF THE CIRCULATOR IN THE CIRCUIT:

The location of the circulator(s) relative to other components in a hydronic system can make the difference between quiet, reliable operation and constant problems. One guiding rule summarizes the situation: *Always install the circulator so that its inlet is close to the connection point of the system's expansion tank.*

To understand why, consider the interaction between the circulator and expansion tank. In a closed-loop piping system, the amount of fluid, including that in the expansion tank, is fixed. It does not change regardless of whether the circulator is on or off. The upper portion of the expansion tank contains a captive volume of air at some pressure. The only way to change the pressure of this air is to either push more fluid into the expansion tank to compress the air, or to remove fluid from the expansion tank to expand the air. This fluid would have to come from, or go to, some other location within the system. However, since the system's fluid is incompressible, and the amount of fluid in the system is fixed, this cannot happen regardless of whether the circulator is on or off. Thus, the expansion tank fixes the pressure of the system's fluid at its point of attachment to the piping. This is called the *point of no pressure change* for that system.

Consider a horizontal piping circuit filled with fluid and pressurized to a static pressure of 10 psi. This static pressure is indicated by the solid red line shown above the piping in Figure 5-16.

When the circulator is turned on, it immediately creates a pressure difference between its inlet and discharge ports. However, the pressure where the expansion tank connects to the circuit (e.g., the point of no pressure change) remains at 10 psi. The combination of the pressure increase across the circulator, the pressure drop due to head loss in the piping and the point of no pressure change results in the dynamic pressure distribution shown by the dashed green lines in Figure 5-16.

Notice that the pressure increases in nearly all parts of the circuit when the circulator is operating. This is desirable because it helps eject air from vents. It also reduces the potential for cavitation. The short segment of piping between the expansion tank and the inlet port of the circulator experiences a slight drop in pressure due to head loss in the piping. The numbers used for pressure in Figure 5-16 are illustrative only. The actual numbers will depend on flow rates, fluid properties and pipe sizes.

To see how problems could develop, imagine the same piping circuit, but with the expansion tank connected near the discharge of the circulator and with a static pressure of only 5 psi. When the circulator operates, the same 9 psi pressure differential is established between its inlet and outlet ports. The pressure at the point where the expansion tank connects (e.g., the point of no pressure

Figure 5-16

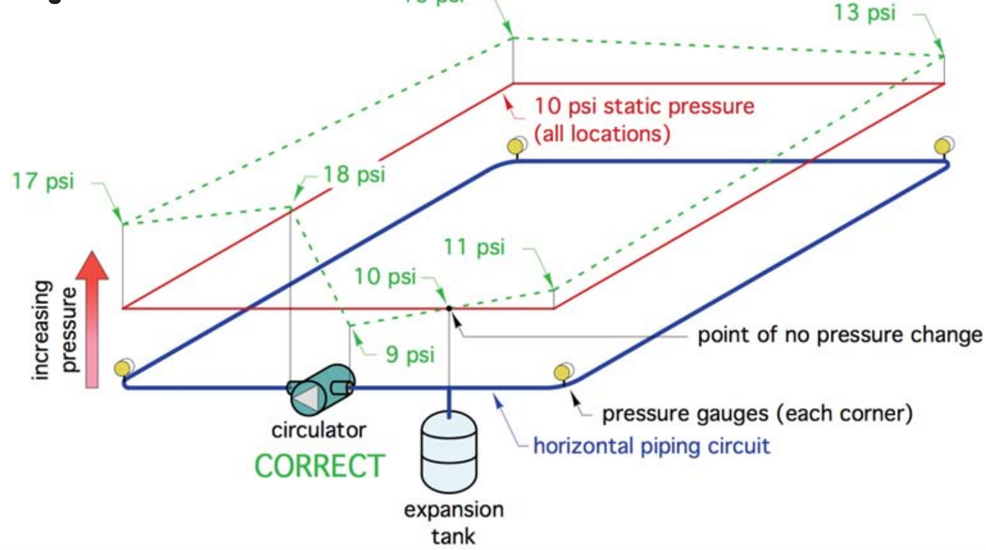


Figure 5-17

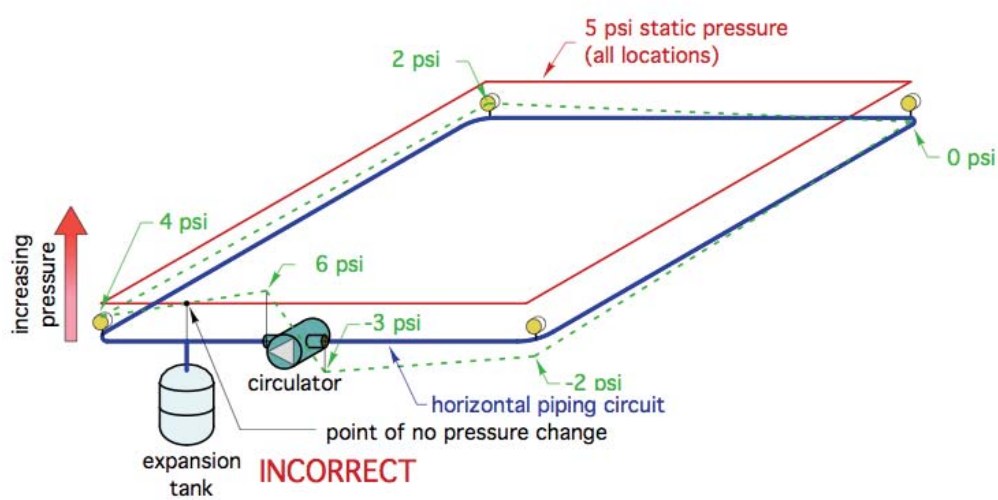


Figure 5-18

change) remains at 5 psi. The resulting dynamic pressure profile is shown in Figure 5-17.

Notice that the pressure in the piping between the upper right-hand corner of the circuit and the inlet port of the circulator is now negative (e.g., below atmospheric pressure). If air vents or an air separator were located in this portion of the circuit, the negative pressure would suck air into the system every time the circulator operates. This situation must be avoided by proper placement of the circulator relative to the expansion tank. Always "pump away" from the location where the expansion tank connects to the system.

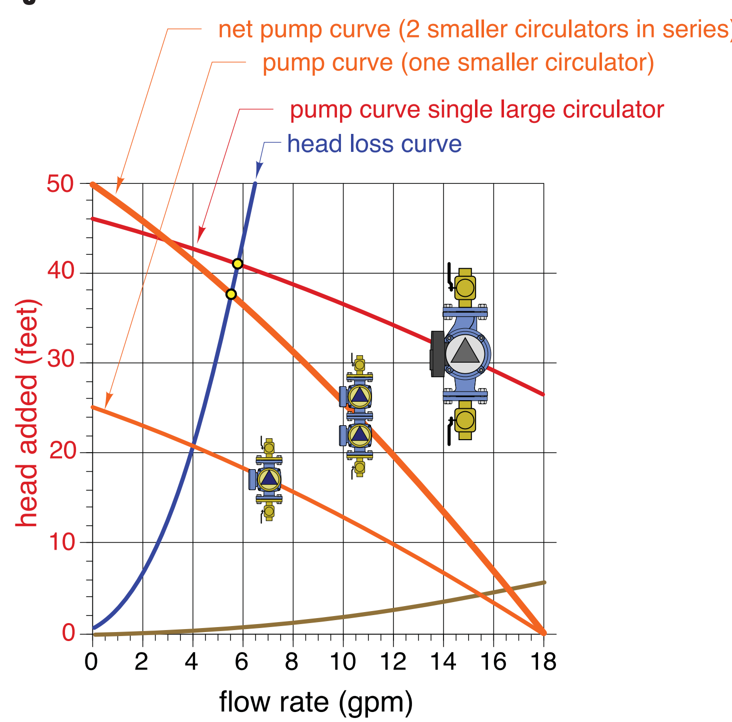
CIRCULATORS IN SERIES:

Some hydronic circuits are characterized by high head loss relative to the flow rate passing through them. Two examples of such circuits are the initial filling of a drainback-protected solar thermal system and a long earth-loop circuit in a geothermal heat pump system.

One way to meet these requirements is to specify a single large circulator capable of meeting the flow and head requirement. However, in many cases this approach causes the large circulator to operate near the upper end of its pump curve, and thus at low wire-to-water efficiency. The result may be a high electrical power requirement for the circulator relative to another alternative. That alternative is to operate two circulators in series, as shown in Figure 5-18.

When two identical circulators are connected in series, their "net" pump curve is found by doubling the head of one circulator at every flow rate. Thus, if one circulator has a head of 25 feet at zero flow, two of these circulators in series would have a head of 50 feet at zero flow. Figure 5-19 shows this method of creating a net pump curve for two identical series-connected circulators.

Figure 5-19



Notice that the operating point for the two series-connected circulators is quite close to that of the single large circulator. Assuming a target flow rate of about 5 gpm, either of these circulator options could provide the necessary flow and head. However, the total electrical power required for the two series-connected circulators might be significantly lower than that to the single large circulator. Thus, energy savings would be possible using the series-connected circulator approach.

When connecting two identical circulators in series, be sure their flow arrows point in the same direction.

CIRCULATORS IN PARALLEL:

It's also possible to connect two circulators in parallel, as shown in Figure 5-20. The "net" pump curve of two identical circulators in parallel is found by doubling the flow rate at each head, as shown in Figure 5-21.

Figure 5-20

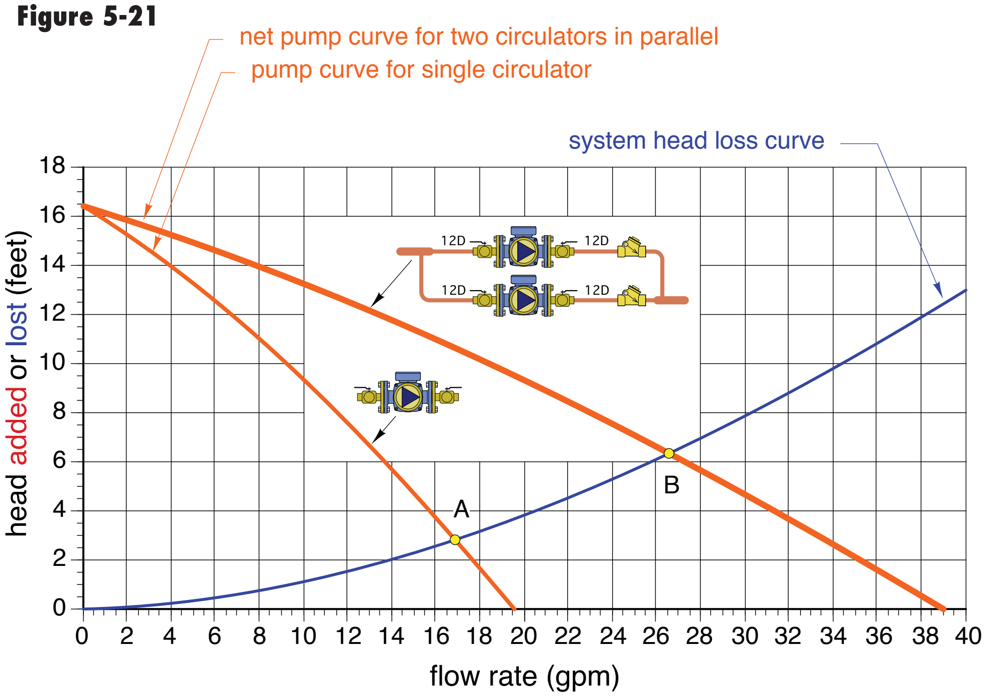
As is true with circulators in series, two identical circulators operating in parallel may reduce the circulator electrical power requirement compared to a single larger circulator that would yield the same flow rate in the circuit.

The "shallow" head loss curve shown in Figure 5-21 intersects the pump curve for the single circulator at a flow rate of about 17 gpm. The net pump curve for two of these circulators in parallel intersects the head loss curve at about 26.5 gpm. Although there is a substantial increase in flow rate, two circulators in parallel will never double the flow rate of a single circulator when applied in any practical system.

When two circulators are connected in parallel, it's import to have a check valve downstream of each circulator. Some circulators are supplied with internal spring-loaded check valves that serve this purpose. If the circulator does not have an internal check valve, include a swing check valve mounted at least 12 pipe diameters downstream of each circulator. This length of straight pipe reduces turbulence entering the check valve and prevents the internal flapper in the check valve from vibrating. These check valves prevent recirculation flow if one circulator is operating while the other is not.

Parallel-piped circulators allow the possibility of control scenarios where one circulator is operating under low load conditions, and the other turns on when needed for high flow rates at high loads. It also allows for redundancy where only one circulator operates at a time, while the other remains in reserve in case the operating circulator fails. This is a common design approach in mission-critical applications where the failure of a single circulator would interrupt heating or cooling to an entire building or other large area. When this approach is used, a controller typically switches operation between the two circulators based on accumulated running hours. The goal is to provide approximately equal running times for each circulator.

Figure 5-21



6. APPLYING CIRCULATORS IN HYDRONIC CIRCUITS

This section discusses some unique circulator applications that take advantage of the concepts discussed in previous sections.

DIFFERENTIAL PRESSURE CONTROL:

Many multiple-zone hydronic heating and cooling systems use some type of valve to allow or prevent flow through each individual zone circuit. Some systems use electrically operated zone valves, and others use electrically operated manifold valve actuators connected to each internal valve within a manifold station. Still others use non-electric thermostatic radiator valves. All three types of systems are shown in Figure 6-1.

All hydronic systems that use valve-type zoning should have a means of controlling differential pressure across the branch circuits. To understand why, consider what happens as the zone valves close in the 3-zone distribution system with a *fixed-speed* circulator shown in Figure 6-2. Each time a zone valve closes, the head loss curve

Figure 6-1

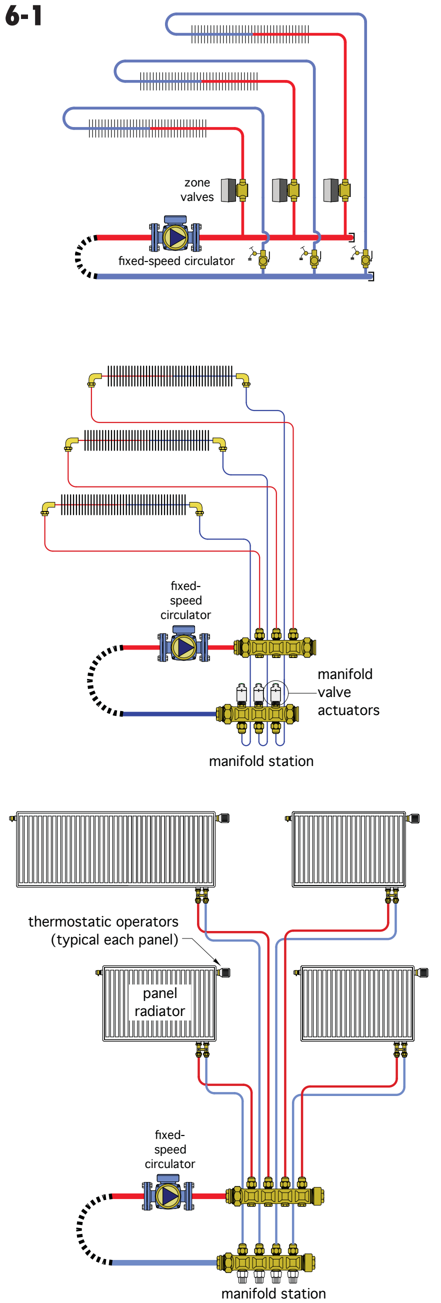


Figure 6-2a

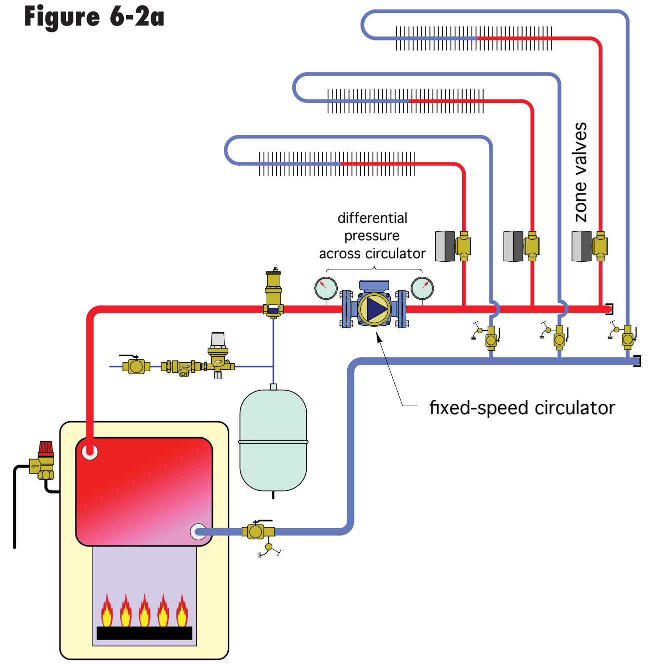


Figure 6-2b

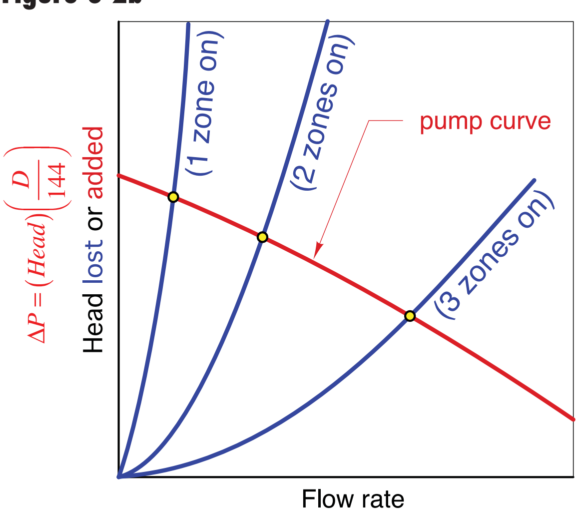


Figure 6-3

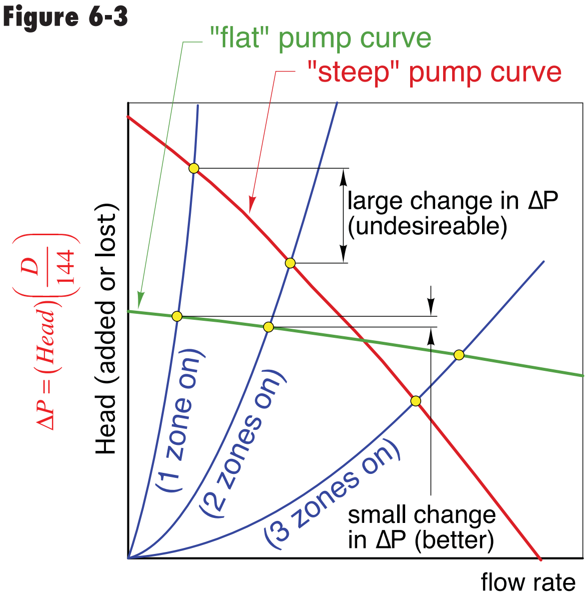
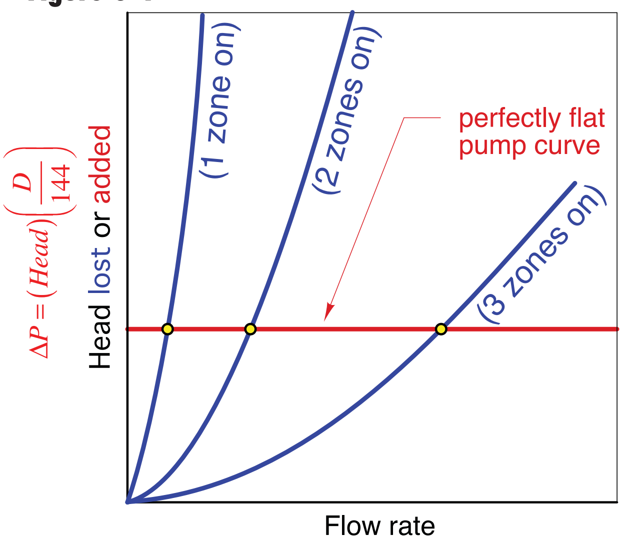


Figure 6-4



for the distribution system gets steeper. This happens because the distribution system, as a whole, becomes more resistant to flow as the number of available flow paths decrease. Think of this as similar to what happens with the "resistance" to traffic flow on a 3-lane highway when one of the lanes is closed.

Each time a zone valve (or manifold valve actuator) closes, and the system head loss curve steepens, the operating point immediately moves farther up the pump curve. This causes the differential pressure across each operating branch circuit to increase. An increase in differential pressure causes an increase in flow rate through each operating branch circuit. While small increases in flow rate are seldom a concern, significant increases can lead to problems such as velocity noise and internal erosion of piping or other components. In systems with large high-head circulators, the increase in differential pressure can even be great enough to cause "bleed-through" flow through zone valves that are supposed to be closed. All of these situations must be avoided.

The traditional approach to minimizing changes in differential pressure in systems with valve-type zoning is to use a circulator with a relatively "flat" pump curve. Such a curve is compared to a "steep" pump curve in Figure 6-3. Notice the relatively small change in differential pressure when the circulator with the "flat" pump curve is used, compared to the change that would occur if the circulator with the "steep" pump curve were used.

Although use of a fixed-speed circulator with relatively flat curve is preferable, there is still a change in differential pressure as zone valves open and close. This change is undesirable because it causes an increase in flow rate through zone circuits that remain on whenever other zones close, and vice versa.

The "ideal" circulator for a system using valve-type zoning would have a perfectly flat pump curve, as illustrated in Figure 6-4.

A perfectly flat pump curve would eliminate any change in differential pressure across the headers (or manifold) regardless of which zones are open or closed. Notice that the three operating points shown in Figure 6-4 move horizontally, but not vertically. When a zone valve closes, flow through the system is reduced, but the differential pressure across any zones that are on doesn't change.

Unfortunately, it is impossible to create a centrifugal-type circulator with a perfectly flat pump curve. However, other techniques that "mimic" the effect of a perfectly flat pump curve are possible, and will now be discussed.

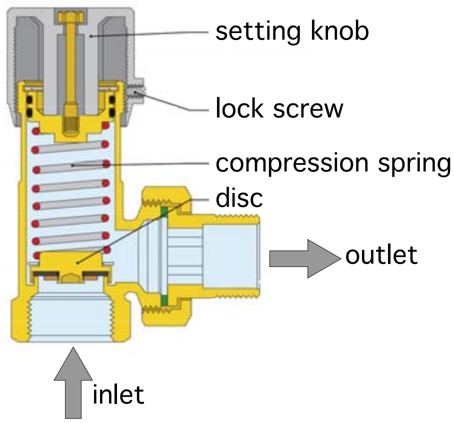
DIFFERENTIAL PRESSURE BYPASS VALVES:

Changes in differential pressure can be further reduced through use of a differential pressure bypass valve (DPBV). Figure 6-5a shows an example of such a valve. Figure 6-5b shows the valve in cross section.

The setting knob on a DPBV adjusts the force exerted by the compression spring on the disc. This force determines the differential pressure between the valve's inlet and outlet at which the disc begins to lift off its seat. If conditions in the system attempt to further increase the differential pressure across the valve, the disc moves farther off the

Figure 6-5a

Figure 6-5b

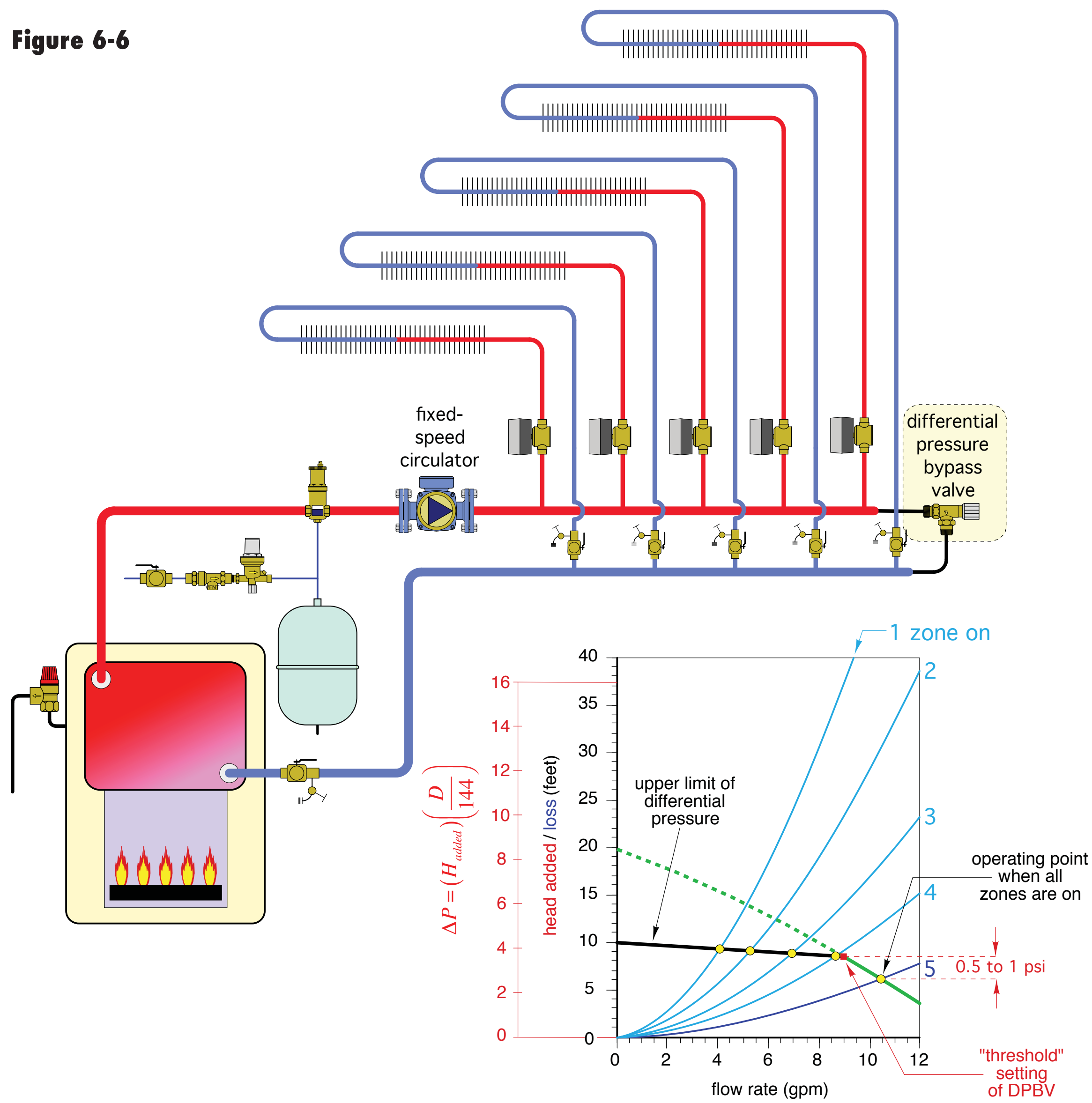


seat to compensate. This allows increased flow through the valve with minimal variation in the differential pressure between the inlet and outlet ports.

An example of a five-zone system with a DPBV installed across the headers is shown in Figure 6-6

When properly set, a DPBV is fully closed when all zone circuits are operating. With this setting, the DPBV has no effect on the system under design load conditions. When the load decreases, and zone valves close, the differential pressure across the headers increases. A properly set DPBV

Figure 6-6



will begin opening when the differential pressure reaches 0.5 to 1 psi above the differential pressure present when all zone circuits are operating. This is shown on the graph in Figure 6-6.

The point where the DPBV begins to open is called the "threshold" setting. At that point, the action of the DPBV limits the upward increase in differential pressure to an almost horizontal line that begins at the threshold setting and slopes very slightly upward to the left. The slight upward slope is caused by increased flow through the DPBV, and thus increased pressure drop across it as the system's head loss curve steepens.

As zones close, the operating point shifts to the left along this slightly sloping line rather than tracking up the pump curve. The result is minimal vertical movement in the operating points, and thus minimal change in the differential pressure between the supply and return headers. Although this is not quite as good as having a hypothetical circulator with a perfectly flat pump curve, it is much better than having no means of differential pressure control.

SIZING DIFFERENTIAL PRESSURE BYPASS VALVES:
To properly size a DPBV, it's necessary to estimate the flow through it assuming all zone circuits are closed. It's possible to estimate the flow under these conditions by plotting the head loss curve for the DPBV along with the pump curve for the circulator, and finding where they intersect, as shown in Figure 6-7.

A vertical line drawn downward from this intersection indicates the flow rate through the DPBV under this condition. A properly sized DPBV can now be selected based on the manufacturer's maximum recommend flow rate for a given valve size.

VARIABLE SPEED CIRCULATOR CONTROL:
Another way of regulating differential pressure in hydronic systems is through use of variable-speed circulators. This method has been used in larger hydronic systems for several years. It is accomplished through use of electronic pressure transducers that communicate to a variable frequency drive (VFD) that electrically adjusts the speed of an AC circulator motor.

The more recent availability of smaller circulators with electronically commutated motors now makes similar control techniques available in smaller hydronic heating and cooling systems. These "pressure-regulated" ECM circulators are ideal for systems using valve-based zoning. They eliminate the need for a differential pressure bypass valve. They also eliminate the need of pressure transducers and variable frequency drives.

CONSTANT DIFFERENTIAL PRESSURE CONTROL:
A variable-speed pressure-regulated circulator can be set to provide "cruise control" for differential pressure. The microprocessor-based speed controller within the circulator is set to maintain the differential pressure required by the distribution system at design load conditions when all branch circuits are fully open. Using electronic sensing techniques, the circulator continuously calculates the differential pressure at which it is operating. When a valve in a branch circuit closes or modulates for less flow, the circulator senses an "attempt" for the differential pressure across the circulator to increase. It quickly reacts by lowering its speed to "cancel out" the attempted change in differential pressure. As other valves in other branch circuits close or modulate for less flow, the circulator continues to decrease its speed as necessary to maintain its set differential pressure. This is called "constant differential pressure control" mode.

The movement of the operating points in the first few seconds after a zone valve closes is represented by the yellow dots in Figure 6-8.

Initially, the operating point moves upward and to the left as the head loss of the distribution system increases. The electronics within the circulator quickly sense this departure from a set

Figure 6-7

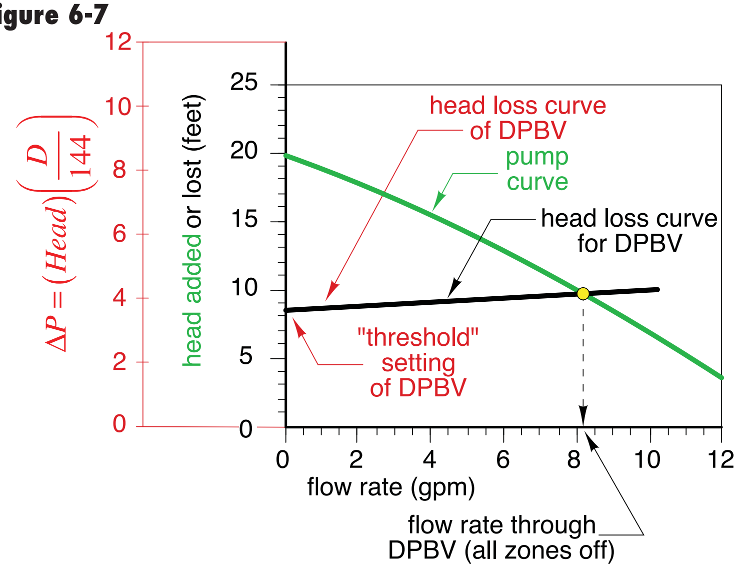
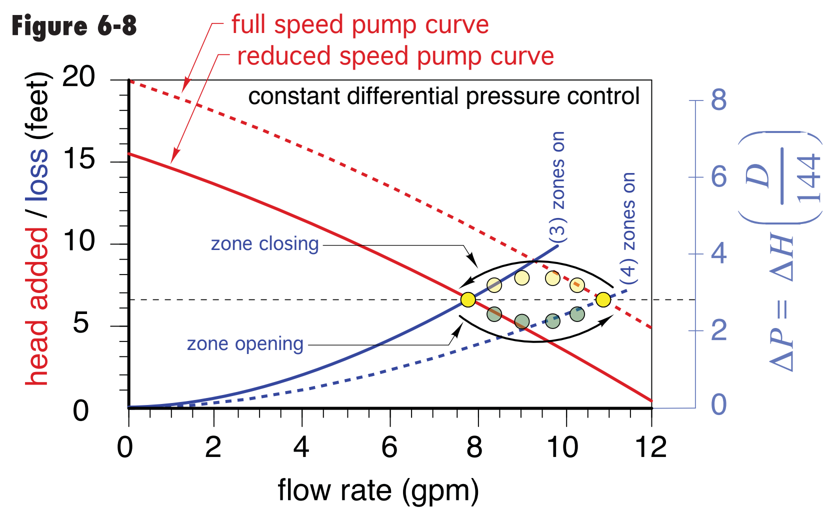


Figure 6-8



differential pressure, and begin to slow the circulator's motor. This causes the pump curve to move downward, and thus the yellow dots representing the progression of the operating point do the same. Within 5 or 6 seconds, the operating point settles to the intersection of the new head loss curve (shown in solid blue) and the reduced speed pump curve (shown in solid red).

When a zone valve opens, the movement of the operating points is represented by the green dots in Figure 6-8. At first, the operating point moves downward and to the right due to the reduced head loss of the distribution system.

Figure 6-9

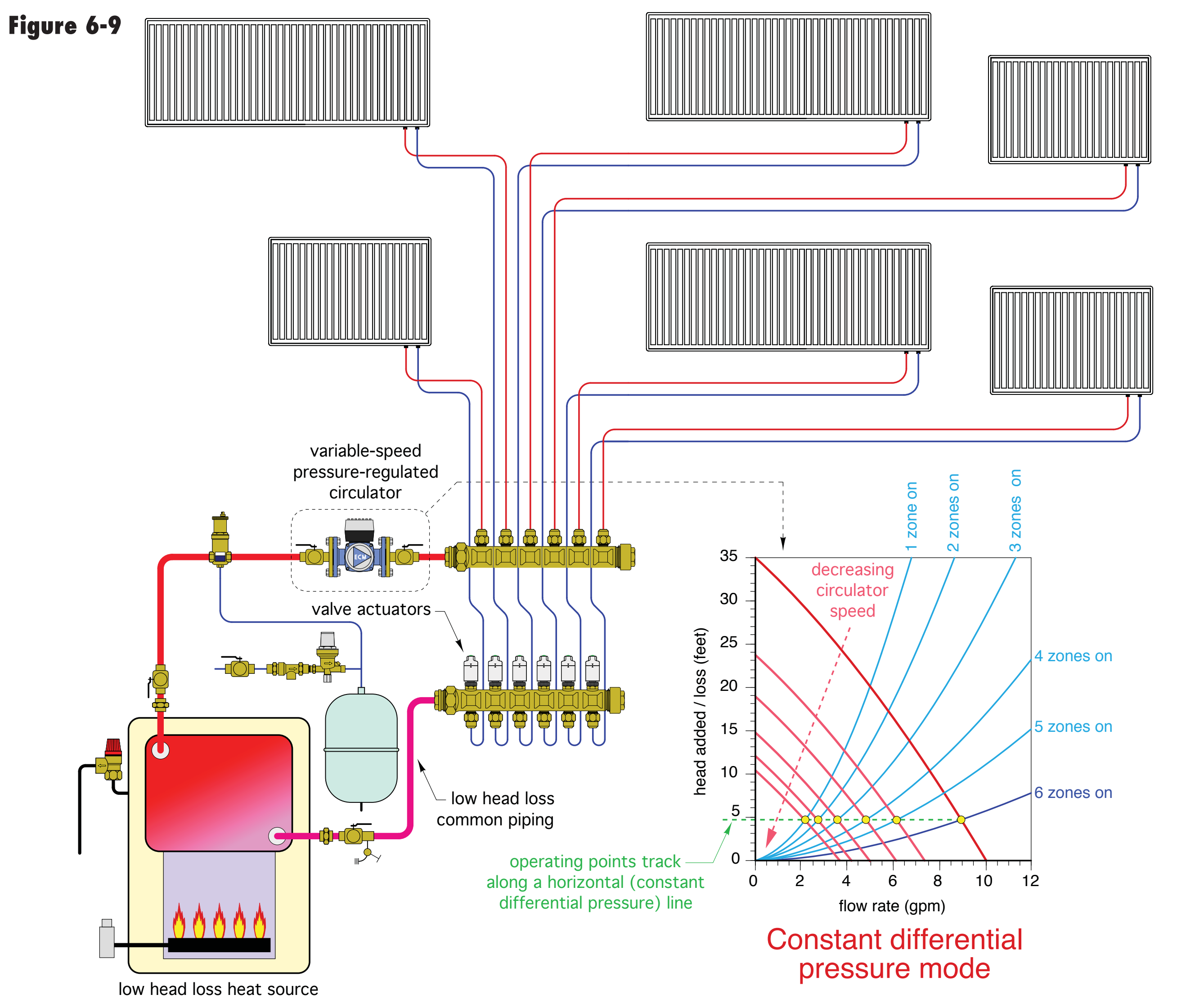
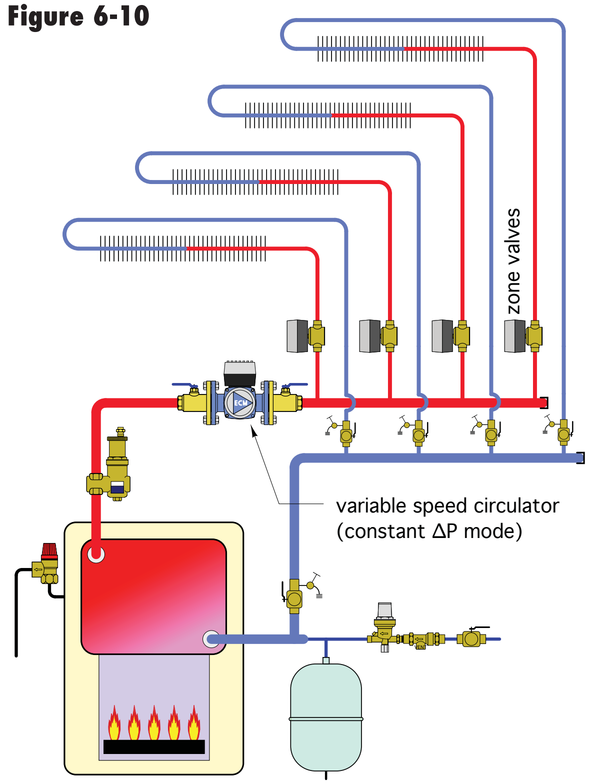


Figure 6-10



This is quickly sensed by the electronics in the circulator, and they respond by increasing the motor speed. This causes the pump curve to rise, and the operating point does the same, eventually settling at the set differential pressure.

Constant differential pressure control mode "mimics" the net effect of using a hypothetical circulator with a perfectly flat pump curve. As such, it is ideally suited to hydronic systems using valve-type zoning, and in which *the majority of the system's overall head loss occurs in the branch piping paths rather than in the common piping path*. A manifold-based system, such as the homerun distribution system shown in Figure 6-9, is a good example of where constant differential pressure control is well applied.

Constant differential pressure control is also well suited to systems with short and generously sized headers and common piping with low flow resistance. Figure 6-10 shows an example.

The very low flow resistance of a cast iron sectional boiler, in combination with short and generously sized headers, gives the "common piping" in this system very low flow resistance. The head loss through this common piping will be very small in comparison to the head loss through any of the branch circuits.

PROPORTIONAL DIFFERENTIAL PRESSURE CONTROL:

Many ECM-based variable-speed circulator can also be configured for "proportional differential pressure control." This control mode is appropriate for distribution systems in which a significant portion of the overall system head loss occurs in the "mains" piping, in comparison to the head loss in the branch piping. A 2-pipe reverse return piping system, such as shown in Figure 6-11, is well suited for proportional differential pressure control mode.

When set for proportional differential pressure mode, the variable-speed circulator establishes a line that the operating points will follow. This line begins where the full-speed pump curve intersects the system head loss curve with all zone circuits fully open. The line slopes downward and ends at one-half the design head when the flow rate is zero. This line minimizes differential pressure variations across the active zone circuits as other zones turn on and off. It compensates for the head loss of the parallel zone crossovers, as well as the significant head loss of the supply and return mains.

Along with pressure regulation, modern variable-speed circulators with electronically commutated motors use significantly less electrical energy when operating at full speed. Their electrical power requirement decreases whenever the operating point of the system moves left or downward, and their motor speed is correspondingly reduced.

Partial-speed operation is possible during much of a typical heating season due to internal heat gains and setback schedules that increase the "off-time" of zones. This allows for significant electrical energy savings relative to systems with fixed-speed circulators and differential pressure bypass valves. Simulations of currently available ECM-based variable-speed circulators predict electrical savings of 60% or more, compared to wet-rotor circulators with permanent split-capacitor motors and equivalent peak flow/head performance.

BEST PRACTICES FOR CIRCULATOR INSTALLATION:

There are several practices that should be observed when designing or installing circulators in hydronic systems:

1. Always provide either 12 diameters of straight pipe upstream of the inlet of a circulator, or use an inlet diffuser. It is important not to create turbulence near the inlet of

Figure 6-11

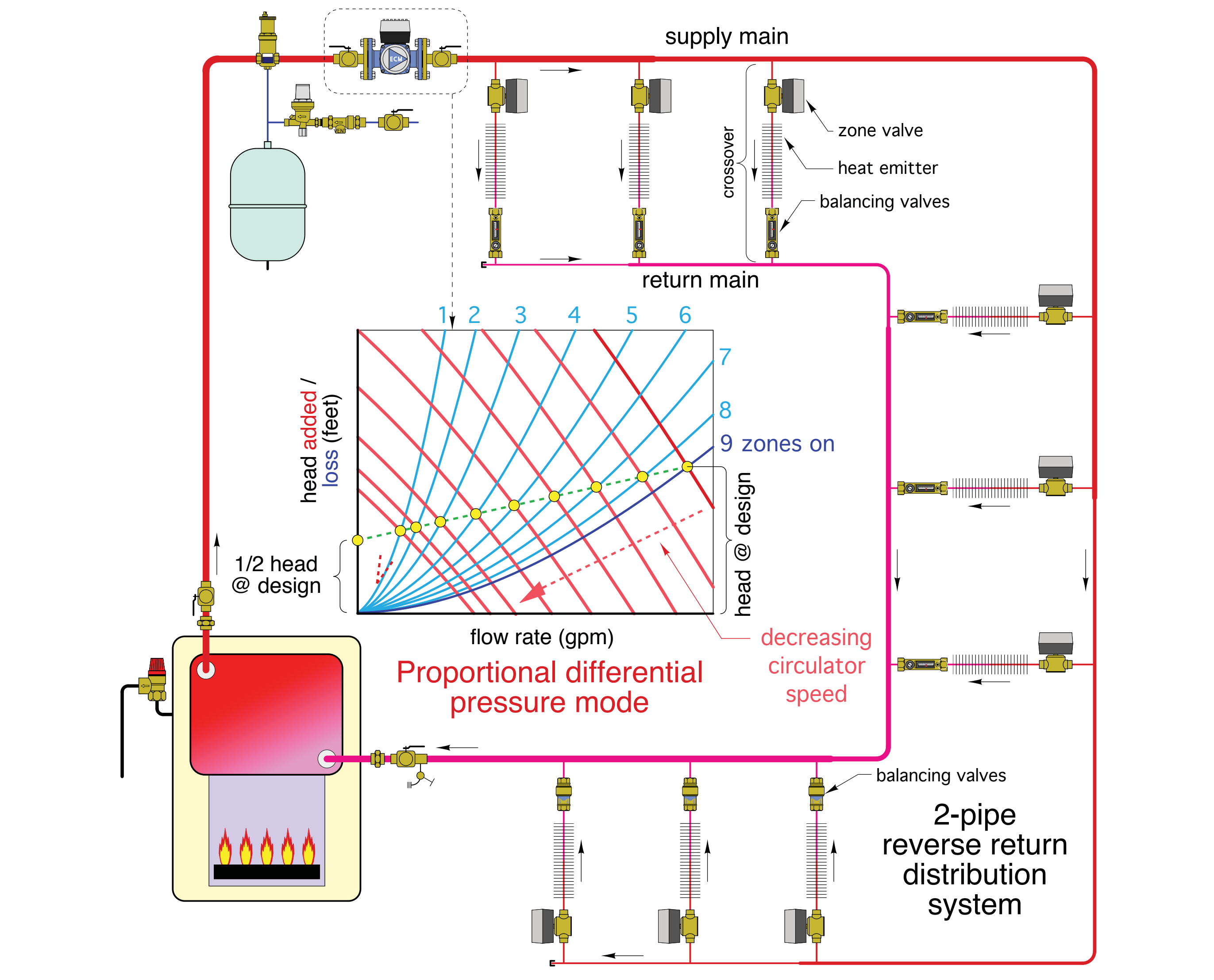


Figure 6-12

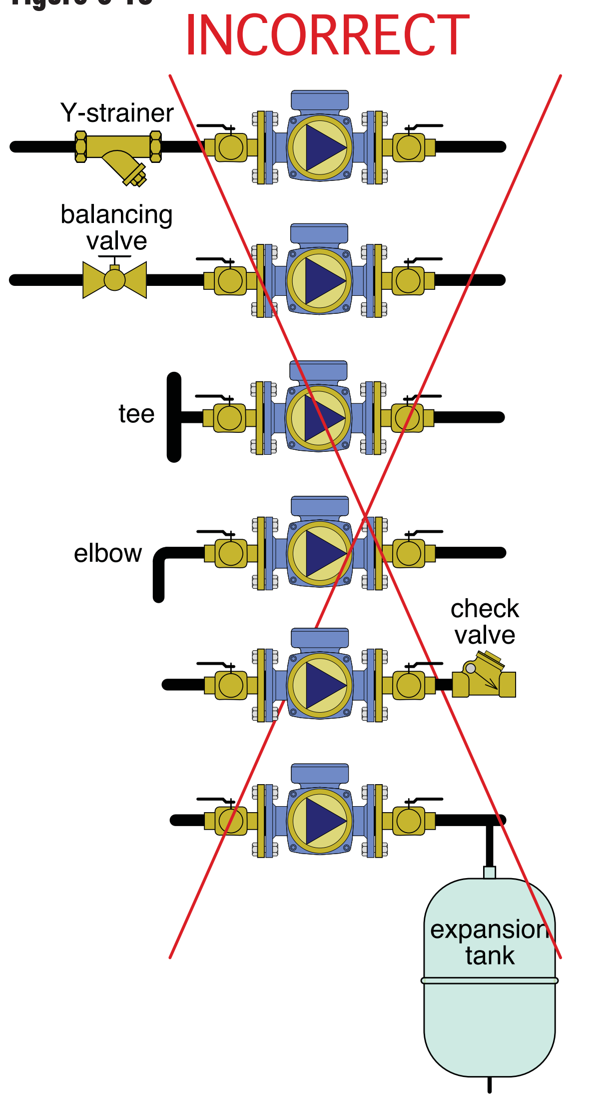
Courtesy of Bell & Gossett

any circulator. Turbulence increases audible noise from the circulator. It can also set up conditions that cause cavitation. For wet-rotor circulators, straight piping on the inlet side of the circulator reduces turbulence. For larger circulators that are mounted to the floor of a mechanical room, an inlet diffuser is often used to help straighten the incoming flow. Figure 6-12 shows an example of an inlet diffuser.

2. Never install throttling devices such as balancing valves near the inlet of circulators. The relatively high pressure drop they might create could set up conditions that cause cavitation at the circulator's inlet (see Figure 6-13).

3. Never install Y-strainers near the inlet of circulators. As they accumulate debris, this type of strainer creates increasingly higher pressure drops.

Figure 6-13



Under the right conditions, this could lead to circulator cavitation (see Figure 6-13).

4. Always remove dirt from the system fluid before that fluid passes through the circulator(s) in the system. Low velocity zone dirt separators should be placed in the piping leading to one or more circulators.

5. Always support piping in close proximity to an inline configuration wet-rotor circulator. Ideally, the piping should be supported by a vibration-absorbing clamp affixed to the piping within a few inches of the circulator (see Figure 6-14).

6. Always install isolation flanges or isolating valves on both sides of every circulator. Eventually every circulator will require maintenance or replacement. This is simple and fast if there are ball valves, gate valves or butterfly valves on each side of the circulator that can be closed to isolate it from the balance of the system.

Figure 6-14

7. Always install at least 12 diameters of straight piping between the outlet of a circulator and any downstream check valve. Failure to do this can cause the flapper in the check valve to rattle when exposed to the most turbulent flow leaving the circulator (see Figure 6-13).

8. Avoid mounting circulators or the piping that supports them to studded partitions separating a mechanical room from occupied space. Any vibration or noise from the circulators could be amplified by the hollow partition. One alternative is to install vertical channel strut that is only fastened to the floor and ceiling, and then secure piping and circulators to this frame. Use vibration-absorbing hardware whenever possible.

9. Always use a check valve in (or downstream of) any zone circulator that is connected to a header with other zone circulators. Failure to do so will result in flow reversal through inactive zone circulators whenever other zone circulators are operating, as shown in Figure 6-15.

10. Always provide hydraulic separation between any circulators in a system that could operate simultaneously. This prevents interference between the circulators, especially when they have very different flow/head ratings. See *idronics* #15 for a complete discussion of hydraulic separation.

11. Always locate circulators so that the connecting point of the system's expansion tank is near the inlet of the circulator(s). This allows the circulator to "pump away" from the point of no pressure change and increase the pressure in the circuit. See the many schematics in this and other issues of *idronics* for examples.

Figure 6-15

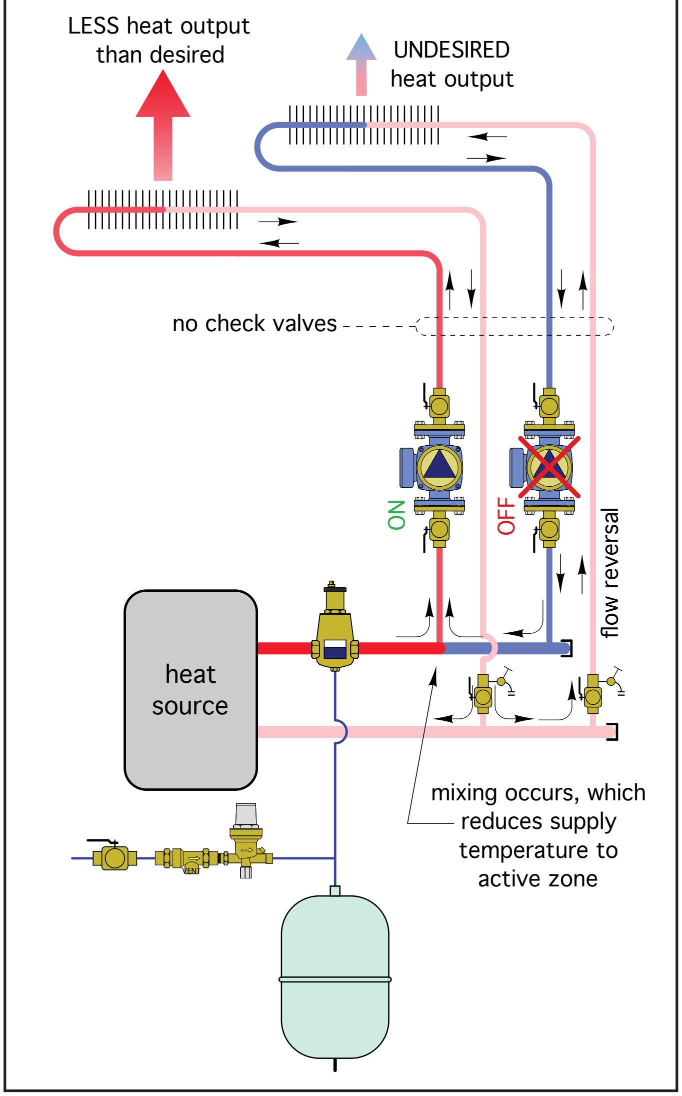
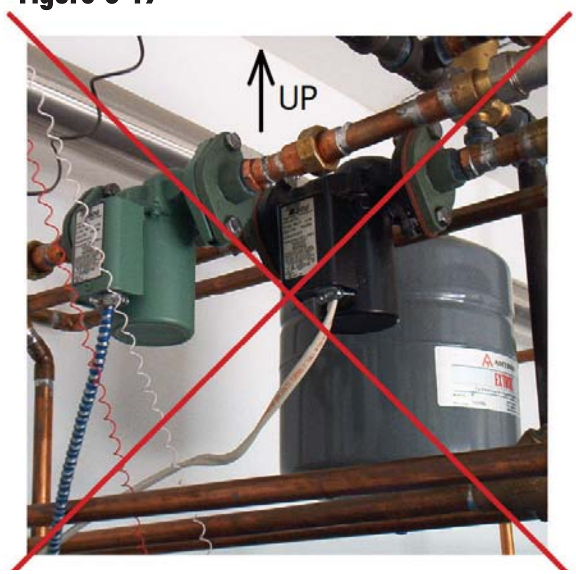


Figure 6-16

Figure 6-17



12. Never use a cast iron circulator in any "open" hydronic circuit. Only circulators made of stainless steel, bronze or engineered polymers are suitable for use in open systems. Cast iron circulators can be severely corroded when used in open systems or in systems that make extensive use of polymer piping without an oxygen diffusion barrier. (See Figure 6-16).

13. Always mount wet-rotor circulators with their motor shaft in a horizontal orientation. This reduces the axial thrust load on the rotor bushings and prolongs their life. Do not mount wet-rotor circulators with the motor shaft in a vertical orientation, as seen in Figure 6-17.

14. Don't operate wet-rotor circulators without fluid in them. Wet-rotor circulators require water for both lubrication of their bushings and heat dissipation. Operating them dry will damage these bushings.

15. Whenever possible, select circulators so that the operating point of the system falls within the middle third of their pump curve. This is the region where the circulator's wire-to-water efficiency is relatively high (see Figure 6-18).

16. Never insulate the <u>motor</u> of a wet-rotor circulator. The surface of the motor is designed to dissipate heat from the circulator. Insulating it could lead to motor failure.

17. Always insulate the volute of a circulator carrying chilled water or cold antifreeze fluids. Failure to do so could cause condensation to form on the volute, as seen in Figure 6-19. Surface rust will quickly follow. Elastomeric foam insulation with low water vapor permeability is the preferred material for this insulation.

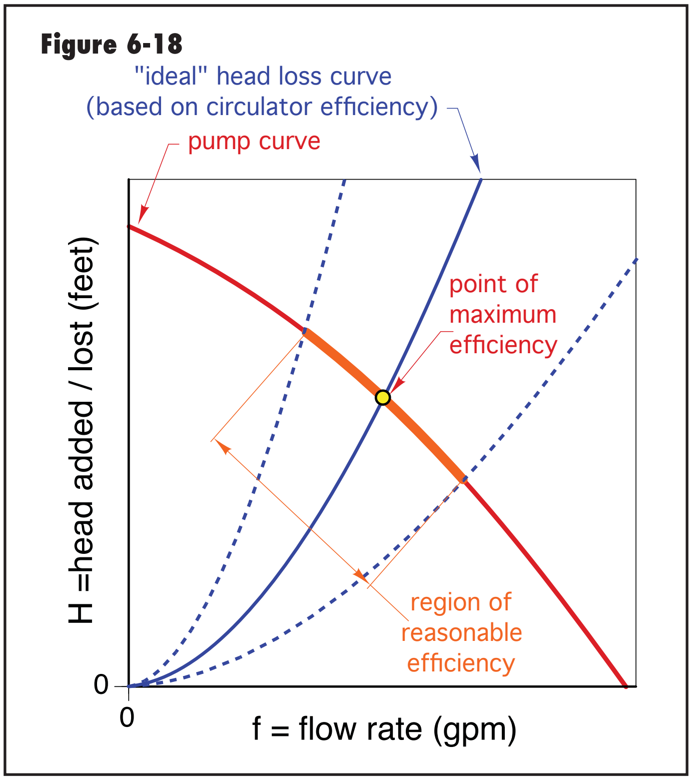
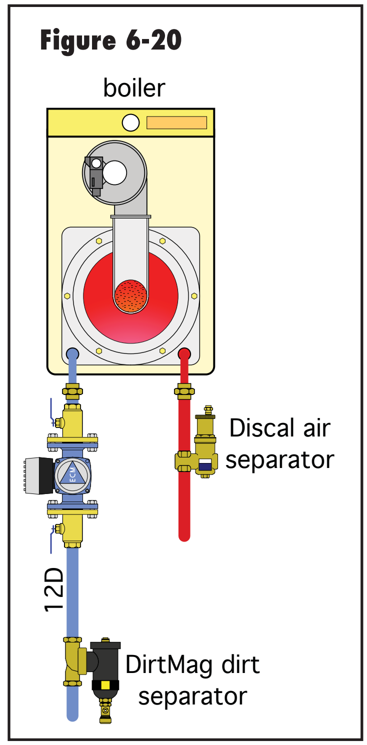


Figure 6-19



18. For circulators without internal check valves, the preferred mounting orientation is in a vertical pipe with upward flow. This allows air bubbles to naturally rise through the circulator's volute and out through the discharge port.

19. For circulators with internal check valves, the preferred mounting orientation is in a horizontal pipe or a vertical pipe with downward flow. This helps prevent air from accumulating under the internal spring-loaded check valve, possibly to an extent where the impeller cannot clear itself of this air.

20. Always use a magnetic dirt separator in systems using high-efficiency ECM circulators. Magnetic dirt separators are more efficient at removing iron oxide particles from the fluid compared to standard dirt separators. These particles, if left to circulate through the system, could eventually attach themselves to the permanent magnet rotor inside the circulator (see Figure 6-20).

21. Use circulators with relatively flat pump curves in systems with valve-type zoning. This minimizes changes in differential pressure across active zone circuits as other zone circuits turn on and off (see Figure 6-3).

22. Always install secondary circulators on the inlet side of secondary circuits connected to a primary loop. This allows the circulator to "pump away" from the system's expansion tank, which is typically connected to the primary loop, as shown in Figure 6-21.

23. Always install a high-efficiency air separator. These separators minimize the dissolved air content of the water, and thus reduce potential for gaseous cavitation of circulators (see Figures 6-20 and 6-21).

24. Locate circulators on the inlet side of devices with high flow resistance. Examples of the latter include mod/con boilers with compact heat exchangers or water-to-refrigerant coil heat exchangers in heat pumps. Locating circulators on the outlet of these devices creates the potential for low pressure and turbulence at the circulator inlet, which can cause cavitation (see Figure 6-20).

25. When supplied from a non-pressurized tank, mount the circulator low in piping. This increases the static pressure on the inlet of the circulator and helps prevent cavitation (see Figure 6-22).

26. Locate circulators and connect wiring to them to minimize the chance of water entering the circulator's wiring compartment. A leak in a component located above a circulator could cause water spillage, as could servicing such a component. Water can also flow downward along electrical cables or conduit and enter the wiring compartment. In general, it's best to avoid locating circulators below other devices that could leak water and to install a "drip curve" in any wiring leading into a circulator, as shown in Figure 6-23. It's also good to rotate the motor house on the circulator so that the wiring knockout is facing downward.

Figure 6-21

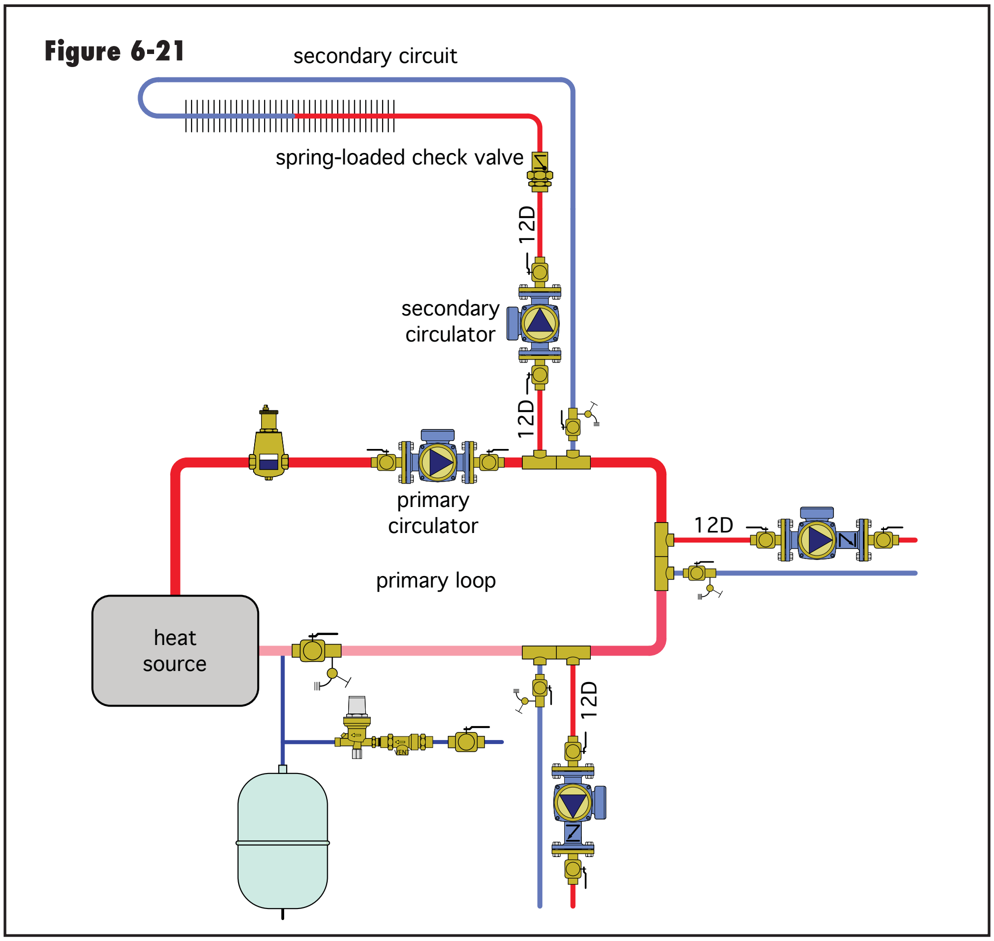
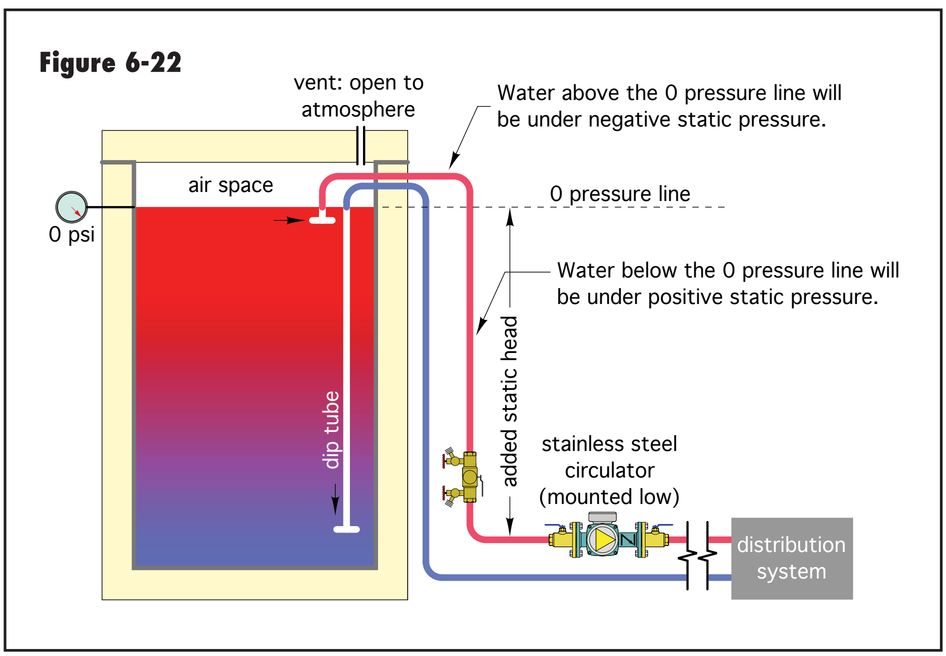


Figure 6-23

Courtesy of Harvey Youker

Figure 6-22



27. Never install circulators in inaccessible locations. Be sure every circulator is installed so that it can be easily accessed for maintenance or replacement if necessary.

28. Always properly ground circulators. Circulators installed in non-metallic piping systems cannot use that piping as an electrical ground. Always connect a properly sized electrical ground conductor to the green grounding screw with the circulator's wiring compartment.

29. Plan mounting locations that allow access to the circulator's wiring compartment with the circulator in place. Having to remove a circulator from the piping or the motor can from the circulator to gain access to its wiring compartment is very poor practice.

30. Do not mount circulators at the very low point of a piping circuit. Over time, any dirt that is not properly captured by a dirt separator could migrate to the low point of the circuit and become lodged within the circulator.

7. DISTRIBUTION EFFICIENCY OF HYDRONIC SYSTEMS

The word "efficiency" always refers to a ratio of desired output divided by the necessary input.

In the case of boilers, the desired output is heat, and the necessary input is fuel. The thermal efficiency of a boiler can be expressed as the instantaneous rate of heat output divided by the instantaneous rate of fuel consumption, where both quantities are expressed in the same units — usually Btu/hr.

Similar definitions of thermal efficiency can be developed for other heat sources such as heat pumps or solar thermal collectors.

While the thermal efficiency of heat sources is important in achieving low operating cost and conserving fuel, this is not the only efficiency that should be considered by heating system designers.

The energy required to *distribute* the heat produced by any heat source, or the cooling effect generated by any cooling source, should also be considered. Systems that use a significant amount of energy to move heat from where it is produced to where it is needed in the building, even when that heat is produced at high *thermal* efficiency, are undesirable. This also holds true for any kind of cooling system.

One way to assess and compare this aspect of system design is to define "distribution efficiency" as follows:

$$n_d = \frac{Q_{delivered}}{w_e}$$

Where:
n_d = distribution efficiency (Btu/hr/watt)
$Q_{delivered}$ = rate of heat delivery (Btu/hr)
w_e = electrical power required by the distribution system (watts)

The higher the distribution efficiency, the lower the operating cost of the distribution system.

For example: Consider a zoned hydronic system with four circulators. Each circulator requires 75 watts power input when operating. At design load, with all four circulators operating, the system delivers 100,000 Btu/hr to the building.

The distribution efficiency of this system at design load conditions would be:

$$n_d = \frac{Q_{delivered}}{w_e} = \frac{100{,}000\,\frac{Btu}{hr}}{4 \times 75\,watt} = 333.3\,\frac{Btu/hr}{watt}$$

The number 333.3 Btu/hr/watt can be interpreted as follows: For each watt of electrical power used by the distribution system, it delivers 333.3 Btu/hr to the building.

However, this number means little without something to compare it to. To provide such a comparison, consider a forced-air furnace with a blower that requires 550 watts while delivering 80,000 Btu/hr to the building. The distribution efficiency of that system is:

$$n_d = \frac{Q_{delivered}}{w_e} = \frac{80{,}000\,\frac{Btu}{hr}}{550\,watt} = 145.5\,\frac{Btu/hr}{watt}$$

In this comparison, the forced-air system has less than half the distribution efficiency of the hydronic system. This implies that the forced-air system will require over twice the electrical power as the hydronic system to deliver the same amount of heat to the load.

The concept of distribution efficiency can be used to compare competing hydronic system designs.

Imagine that the four-zone system cited in the previous example is configured as shown in Figure 7-1.

Each zone is assumed to have an equivalent length of 200 feet of 3/4" copper tubing. Each zone circuit contains 40 feet of fin-tube baseboard and is equipped with a 3-speed circulator operating on high-speed with an electrical power input of 87 watts. The supply water temperature to all zone circuits is 160ºF. Assume the system's heat source and headers have very low head loss, and thus provide good hydraulic separation between the zone circulators.

This system can be simulated to find its thermal and hydraulic equilibrium operating conditions. The results of this simulation indicate the flow rate in each zone will be about 6 gpm, and total heat output of the baseboard in each zone is 12,675 Btu/hr.

The distribution efficiency of this system with all four zones operating is:

$$n_d = \frac{Q_{delivered}}{w_e} = \frac{(4 \times 12{,}675)\,\frac{Btu}{hr}}{(4 \times 87)\,watt} = 146\,\frac{Btu/hr}{watt}$$

If the circulator is changed from its high-speed to low-speed setting, the input power drops to 60 watts, and the zone flow rates drop to about 4 gpm each. The heat output of each zone also drops to 12,310 Btu/hr. The distribution efficiency is now:

$$n_d = \frac{Q_{delivered}}{w_e} = \frac{(4 \times 12{,}310)\frac{Btu}{hr}}{(4 \times 60)watt} = 205\frac{Btu/hr}{watt}$$

In this case, reducing the zone circulator speed from high to low reduced the electrical power required by 31% and heat output by only 2.9%. Thus, the higher flow rate produced a very small gain in heat output. Operating the circulators at their low speed setting also increased distribution efficiency by 40%.

If each zone circulator were set to low speed and operated for 3,000 hours per year, the total electrical energy used by the distribution system would be:

$$4(60watt)\left(\frac{3000hr}{yr}\right)\left(\frac{1kwhr}{1000watt \bullet hr}\right) = 720kwhr/yr$$

This system can also be compared to a system using a single fixed-speed circulator to supply four zones, each regulated by a manifold valve actuator. A schematic for this system is shown in Figure 7-2.

Each zone circuit contains the same 40 feet of 3/4" fin-tube baseboard, as well as 150 feet of 1/2" PEX-AL-PEX tubing. With the single circulator on high speed (87 watts), the heat output of each zone circuit is simulated to be 11,221 Btu/hr and the flow rate through each zone is 1.76 gpm. The distribution efficiency of the system with all four zones operating is:

Figure 7-1

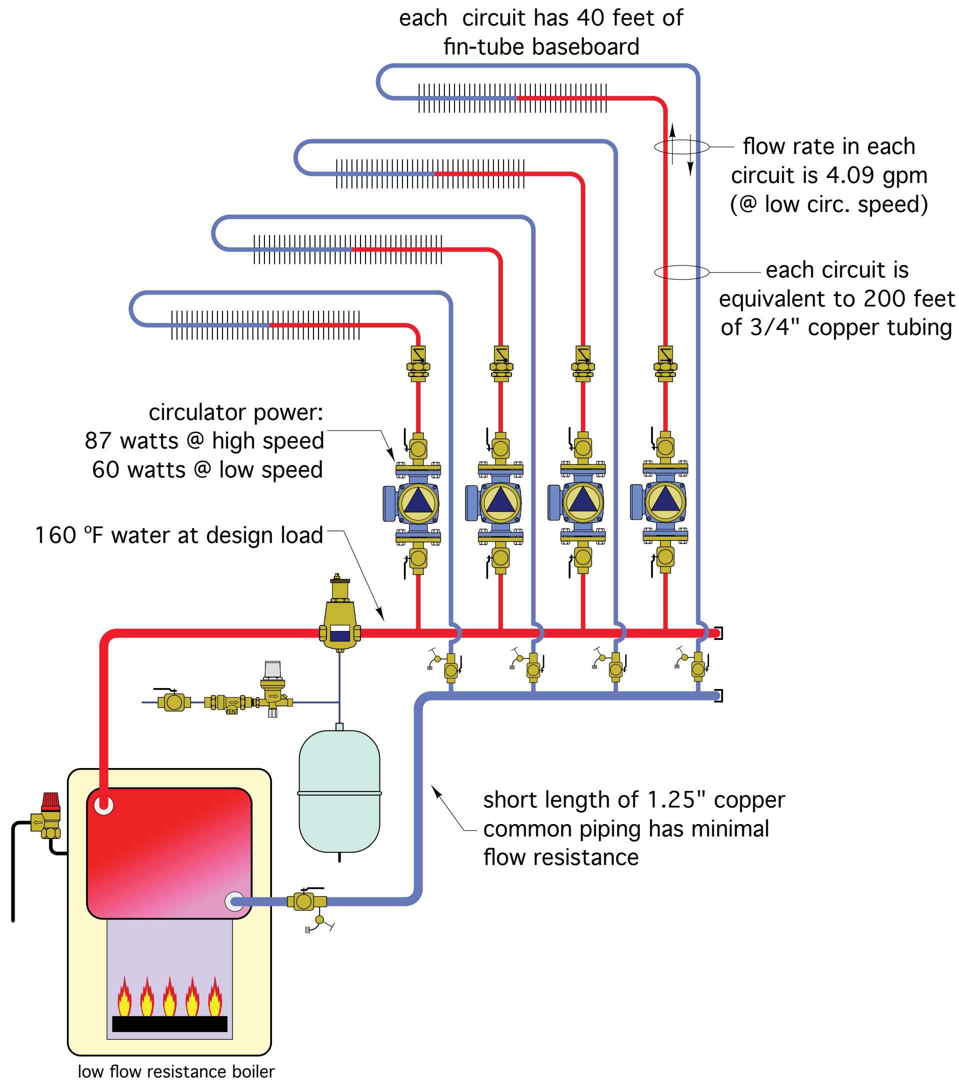
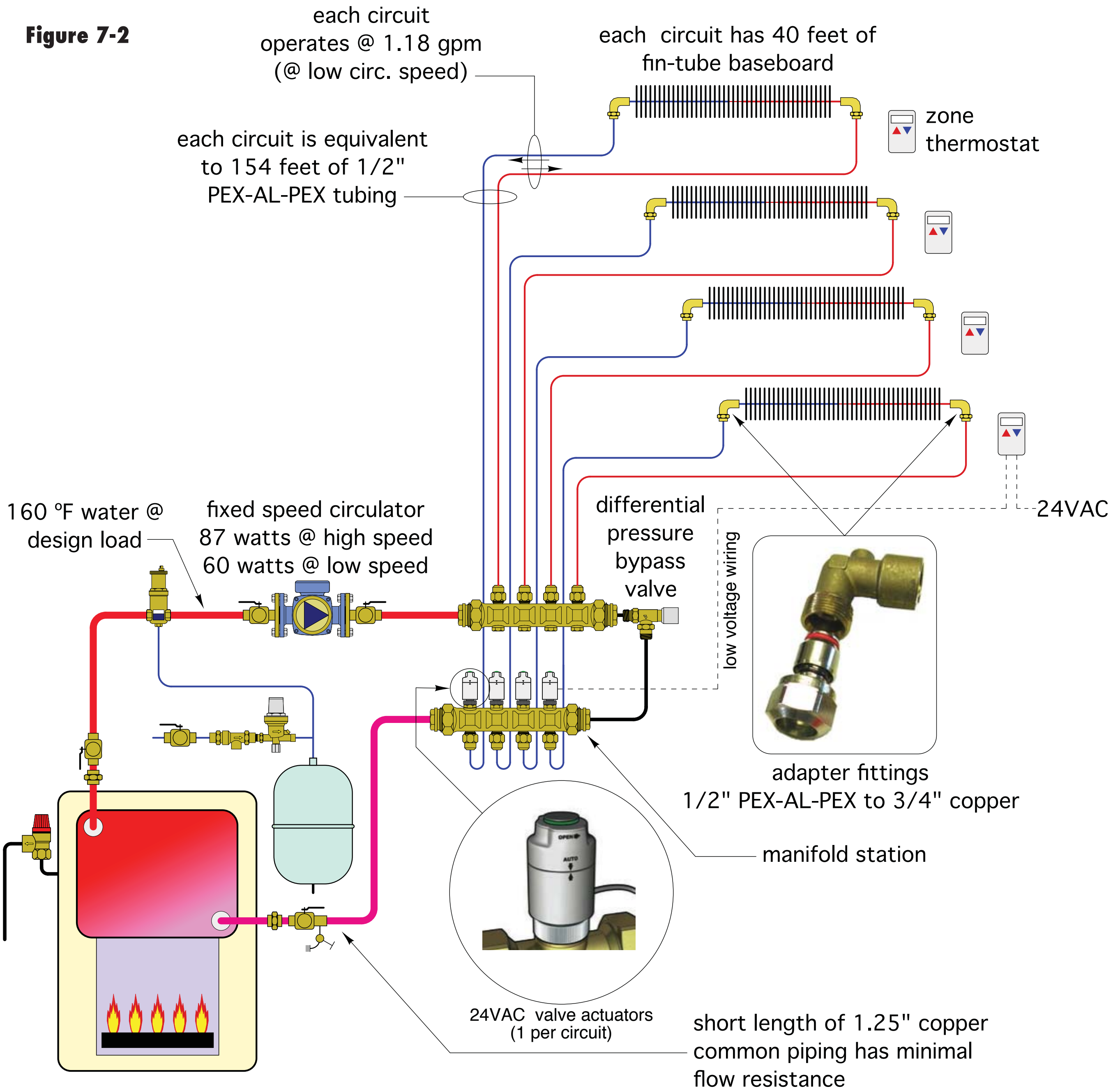


Figure 7-2



$$n_d = \frac{Q_{delivered}}{w_e} = \frac{(4 \times 11{,}221)\frac{Btu}{hr}}{(87)watt} = 516\frac{Btu/hr}{watt}$$

If the circulator is switched to low speed (60 watts), each zone delivers 10,521 Btu/hr, on a flow of 1.18 gpm. The distribution efficiency now becomes:

$$n_d = \frac{Q_{delivered}}{w_e} = \frac{(4 \times 10{,}521)\frac{Btu}{hr}}{(60)watt} = 701\frac{Btu/hr}{watt}$$

To make the comparison fair, the operating hours of the zone valve-based system must be increased so that it delivers the same total seasonal heat output as the system using zone circulators. This is done by multiplying the operating hours of the zone circulator system by the ratio of the design output of a zone circuit in the system using zone circulators, to that of a zone circuit in the system using zone valves. The total electrical energy use of the 60-watt circulator in the system using zone valves is now determined:

$$1(60watt)\left(\frac{3000hr}{yr}\right)\left(\frac{1kwhr}{1000watt \bullet hr}\right)\left(\frac{12,310Btu/hr}{10,521Btu/hr}\right) = 211kwhr/yr$$

In this comparison, the system using zone valves delivers the same total heat using only 29% of the electrical energy required by the system using zone circulators. This demonstrates a distinct advantage of valve-based zoning from the standpoint of distribution efficiency and electrical energy consumption.

If a pressure-regulated circulator with ECM motor were used in combination with the valve-based zoning, and that circulator were operated in a constant differential pressure mode, the estimated electrical energy use would drop by at least 60%. This would put the estimated seasonal electrical energy use of the zone valve system at about 85 kwhr/yr.

At full speed, the power input to an ECM-circulator with the same head/flow characteristics as the previously described 60-watt circulator with PSC motor would be about 50% lower (e.g., about 30 watts). The estimated distribution efficiency of this system under design load conditions would therefore be:

$$n_d = \frac{(4 \times 10,521)\frac{Btu}{hr}}{30watt} = 1402\frac{Btu/hr}{watt}$$

By using state-of-the-art products and careful design, it is possible to create hydronic distribution systems that have distribution efficiencies of at least 3,000 Btu/hr/watt.

Superior distribution efficiency is often the “untold story” regarding use of hydronic heating or cooling. Too often, a comparison of hydronic heating or cooling emphasizes the *thermal* efficiency of the heating or cooling source in comparison to that of a competing system, such as a forced-air furnace or air-to-air heat pump. Currently available condensing-capable forced-air furnaces will usually have *thermal* efficiencies that are equal to or slightly higher than that of mod/con boilers. This is largely due to lower return air temperatures to the furnaces compared to the lowest practical return water temperatures to mod/con boilers or hydronic heat pumps.

Deciding to use a system that includes a heat source with incrementally higher thermal efficiency, at the expense of much lower distribution efficiency, can cause the total operating cost to be higher. Hydronic heating professionals should emphasize the energy savings associated with the low distribution energy requirements of well-designed hydronic systems.

THE WRONG APPROACH:

Although hydronic distribution systems have the *potential* for high distribution efficiency, as demonstrated in the previous examples, they can also be designed in ways that largely negate this potential. One approach that greatly reduces distribution efficiency is overuse of zone circulators, many of which may be oversized for their respective zone flow/head requirements. The system shown in Figure 7-3 is an example of this approach.

Consider the (real) case of a large (10,000 square foot) home with 40 circulators in its hydronic heating distribution system. This system was sized to deliver 400,000 Btu/hr at design load conditions. Assuming these circulators have an average power requirement of 90 watts each, and that they all operate under design load conditions, the distribution efficiency of this system would be:

$$n_d = \frac{400,000\frac{Btu}{hr}}{40 \times 90watt} = 111\frac{Btu/hr}{watt}$$

This poorly conceived hydronic system has a distribution efficiency that is significantly lower than that of the forced-air delivery system discussed earlier (n_d of 111 Btu/hr/watt compared to n_d = 145 Btu/hr/watt for the forced-air system). It consumes about 12.6 times more electrical energy to move heat through its building compared to the previously discussed hydronic system that used a single high-efficiency ECM-based circulator and zone valves. Although the craftsmanship of a “wall full of circulators” system may look impressive, this approach is far from optimal and should be discouraged.

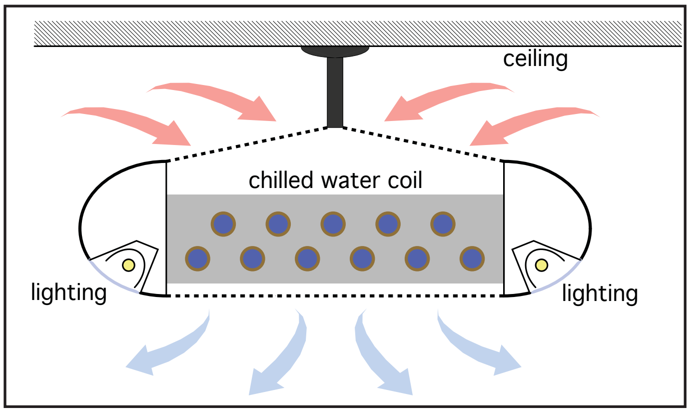
Figure 7-3

DISTRIBUTION EFFICIENCY IN COOLING SYSTEMS:

The importance of high distribution efficiency is even more important in cooling systems. All the electrical energy supplied to move either chilled air or chilled water through the cooling distribution system of a building ultimately ends up as *heat* dissipated within that

Figure 7-4a

Figure 7-4a



building. The total cost of operation includes the cost of electricity to operate the blowers, fans or circulators in the distribution system, and the additional electricity needed by the cooling source to capture and remove the heat added to the building by the blowers, fans or circulators. A well-designed residential hydronic cooling distribution system that uses a single circulator operating on 100 watts of electrical power input adds 341.3 Btu/hr to the cooling load. The blower in a heat pump rated at 3 tons of cooling capacity may require about 1000 watts of electrical input. This adds 3413 Btu/hr (e.g., over 1/4 ton) to the building's cooling load.

The total electrical power requirement to operate the distribution system and dissipate the heat produced by the distribution equipment can be estimated using Formula 7-1:

Formula 7-1

$$P_{total} = P_d\left[1+\frac{3.413}{EER}\right]$$

Where:
P_{total} = total power required to operate the cooling distribution system & dissipate the heat produced by the distribution equipment (watt)
P_d = power required to operate distribution system (watt)
EER = Energy Efficiency Ratio of the cooling equipment (Btu/hr/watt)

For example, assume a well-designed hydronic cooling distribution system could operate with a single 100-watt circulator. The cooling equipment used in the building operates at an EER of 18. The total electrical power required by the overall cooling system to operate its distribution system *and* dissipate the associated heat gain would be:

$$P_{total} = P_d\left[1+\frac{3.413}{EER}\right] = 100\left[1+\frac{3.413}{18}\right] = 119\,watt$$

The higher the power requirement of the cooling distribution system, and the lower the EER of the cooling source, the greater the total power demand required to maintain comfort.

SENSIBLE COOLING USING CHILLED WATER:

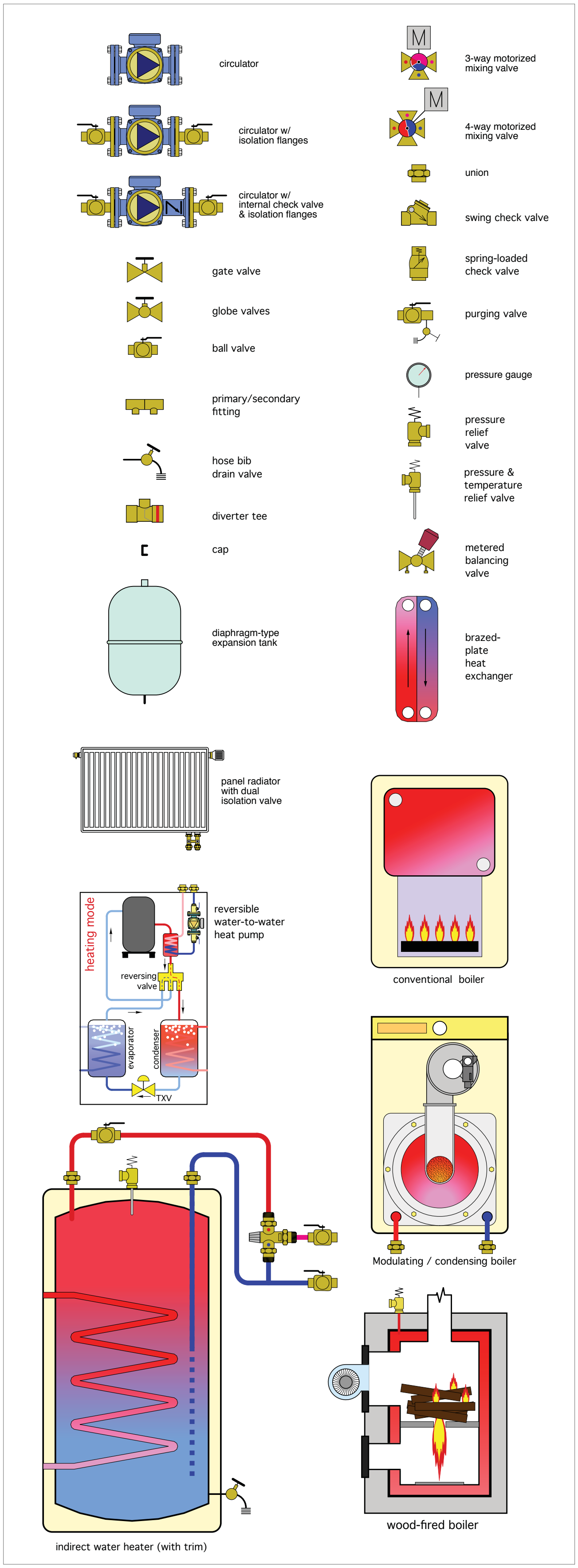
The significantly reduced power requirements of circulators relative to blowers or fans of similar heat conveyance capacity has increased interest in systems that use water rather than air for sensible cooling. These include "chilled beams," such as shown in Figure 7-4, and radiant panel cooling. More information on chilled-water cooling can be found in *idronics* #13.

SUMMARY:

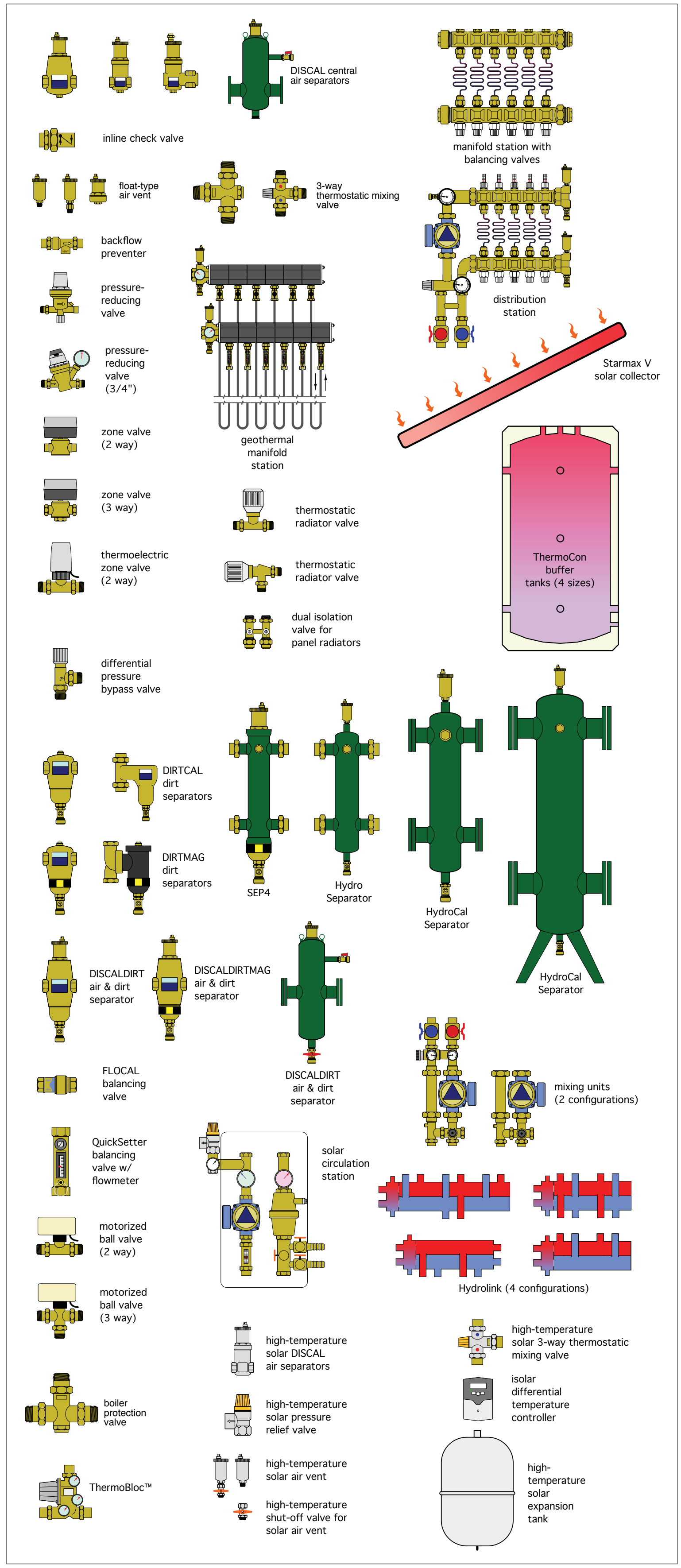
Water flowing through piping and other components is the essence of hydronic heating and cooling systems. Determining proper flow rates and the head losses associated with them is fundamental to proper design of these systems. This issue of *idronics* has presented the tools and principles needed for the latter. It has also discussed the proper selection, installation and programming of circulators to create and manage this flow. Designers are encouraged to calculate the distribution efficiency of all proposed designs to evaluate how they convey heat relative to the electrical power required to operate them.

APPENDIX A: PIPING SYMBOL LEGEND:

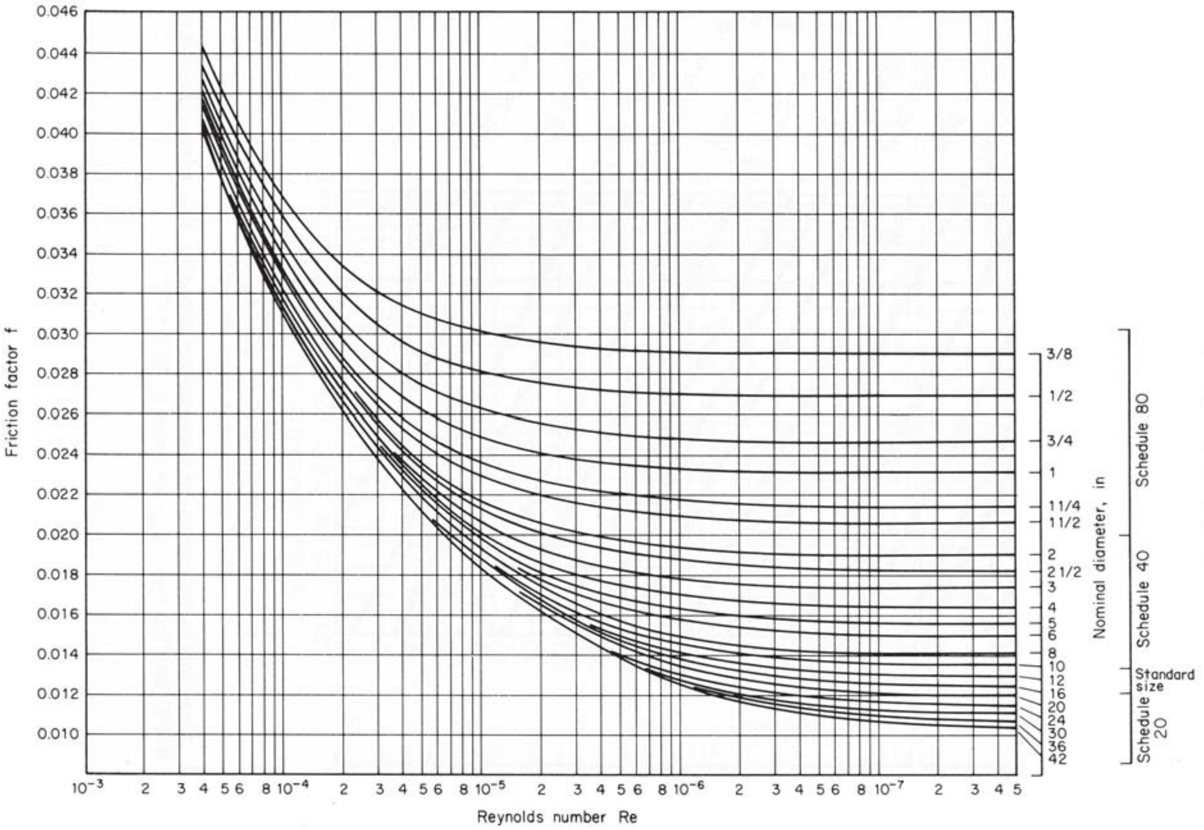
GENERIC COMPONENTS



CALEFFI COMPONENTS



APPENDIX B: MOODY FRICTION FACTOR CHART:



Source: *Hydronic System Design & Operation*, Erwin Hansen, 1985, McGraw Hill, ISBN 0-07-026065-6

Differential pressure by-pass valve

519 series

CALEFFI

519709A

519566A

Function

The differential pressure by-pass valve is used in systems with a fixed speed circulating pump supplying several zones controlled by two way zone valves. This valve ensures that the head pressure of the pump is proportional to the number of two way valves being closed. It will by-pass the differential pressure created by the pump as the zone valves close, thus eliminating water hammer noise.

The 519 series is available with conventional NPT and sweat union connections. Also available for size 3/4", the Presscon™ copper tail-piece with union nut makes installation and maintenance fast, easy and efficient. Special slots in the EPDM O-ring allows fluid to leak during system testing if unpressed and provide a perfect leak proof seal when completely pressed.

Product range

Code 519502A Adjustable differential pressure by-pass valve, flow up to 9 gpm .... connections 3/4" MNPT union inlet x 3/4" MNPT union outlet
Code 519566A Adjustable differential pressure by-pass valve, flow up to 9 gpm ....connections 3/4" press union inlet x 3/4" press union outlet
Code 519599A Adjustable differential pressure by-pass valve, flow up to 9 gpm .... connections 3/4" sweat union inlet x 3/4" sweat union outlet
Code 519600A Adjustable differential pressure by-pass valve, flow up to 40 gpm ....connections 1" FNPT inlet x 1" MNPT union outlet
Code 519609A Adjustable differential pressure by-pass valve, flow up to 40 gpm ....connections 1" FNPT x 1" sweat union outlet
Code 519700A Adjustable differential pressure by-pass valve, flow up to 45 gpm .... connections 1-1/4" FNPT inlet x 1-1/4" MNPT union outlet
Code 519709A Adjustable differential pressure by-pass valve, flow up to 45 gpm ....connections 1-1/4" FNPT inlet x 1-1/4" sweat union outlet

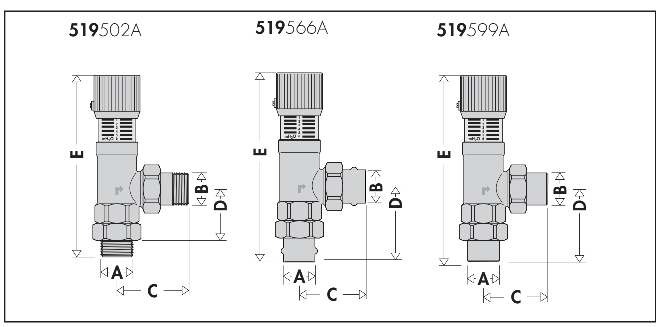
Technical specification

Materials
- body: brass
- valve plug: brass
- valve plug gasket: EPDM
- O-Ring seals: EPDM
- union seals: asbestos free NBR
- control knob: ABS
- spring: stainless steel

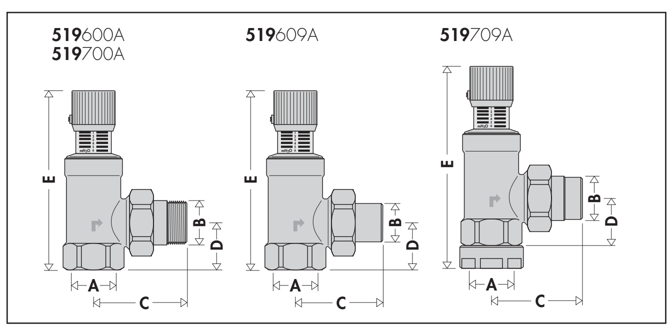
Performance

Suitable fluids: water, glycol solutions
Max. percentage of glycol: 30%
Temperature range: 32 to 230°F (0 to110°C)
Max. working pressure: 150 psi (10 bar)
Flow rates: 3/4" flow up to 9 gpm
1" flow up to 40 gpm
1-1/4" flow up to 45 gpm
Setting range: 1 to 6 m w.g. (2 to 10 psi)

Dimensions



Code	A	B	C	D	E	Wt. (lb.)
519502A	¾" MNPT	¾" MNPT	2¼"	2⅝"	5 11/16"	1
519566A	¾" press	¾" press	2⅜"	2¾"	5 13/16"	**1**
519599A	¾" SWT	¾" SWT	2⅛"	2½"	5 9/16"	1



Code	A	B	C	D	E	Wt. (lb.)
519600A	1" FNPT	1" MNPT	3 13/16"	2 1/16"	6⅝"	1.4
519609A	1" FNPT	1" SWT	2 13/16"	2 1/16"	6⅝"	1.4
519700A	1¼" FNPT	1¼" MNPT	3 15/16"	2 11/16"	7⅜"	1.5
519709A	1¼" FNPT	1¼" MNPT	3"	2 11/16"	7⅜"	1.5

HydroMixer™ direct distribution unit

165 series

CALEFFI

Function

The 165 series HydroMixer™ provides high temperature flow direct to secondary heating circuits, complete with onboard three-speed or variable-speed pump, supply and return temperature gauges, secondary circuit shut-off ball valves and pre-formed insulation shell. Versions are available with either the three-speed Grundfos UPS 15-58 pump or the variable-speed Alpha 25-55U pump.

The unit can be ordered with supply flow and pump on the right or left side, and the unit is field reversible to accommodate changing installation requirements. The adjustable differential pressure by-pass valve, code 519600 and wall bracket code 165001 are optional. The unit comes with male union connections which require separately purchased, 1 inch sweat or NPT, top outlet or bottom inlet fitting kits.

Product Range

Code 165600A	Direct distribution unit with Grundfos UPS 15-58 pump on right side	connections 1” sweat or NPT union*
Code 165610A	Direct distribution unit with Grundfos UPS 15-58 pump on left side	connections 1” sweat or NPT union*
Code 165602A	Direct distribution unit with Grundfos Alpha 25-55U pump on right side	connections 1” sweat or NPT union*
Code 165612A	Direct distribution unit with Grundfos Alpha 25-55U pump on left side	connections 1” sweat or NPT union*

*Fitting sets sold separately

Technical Characteristics

Materials
Connecting pipes
Material: steel

Check valve
Body: brass
Shutter: PPAG40

Shut-off ball valves
Body: brass

Performance

Suitable fluids: water, glycol solutions
Max. percentage of glycol: 30%
Max. working pressure: 145 psi (10 bar)
Max. working temperature: 212°F (100°C)

Connections: male union threads
- top outlets: 1-1/4” male straight
- bottom inlets: 1-1/2” male straight
- inlet/outlet center distance: 5 inches (125 mm)

Pump
Three-speed pump: Grundfos: UPS 15-58-130;
Variable-speed pump: Grundfos: ALPHA 25-55U;

Body material: cast iron
Power supply: 115 V 50/60 Hz
Protection class: Class F
Pump center distance: 5-1/8” (130 mm)

Pump connections: 1 1/2” male straight

Temperature gauge
Dual scale: 32 –176°F (0 – 80°C)

Technical specification of insulation
Material: EPP
Thickness: 1-1/8” (30 mm)
Density: 3 lb/ft^3(45 kg/m^3)
Working temperature range: 25 – 250°F (-5 – 120°C)
Thermal conductivity: 0.256 $BTU \cdot in/hr \cdot ft^2 \cdot$ °F(0.037 W/m·K) at 50°F (10°C)
Fire resistance (UL94): class HBF

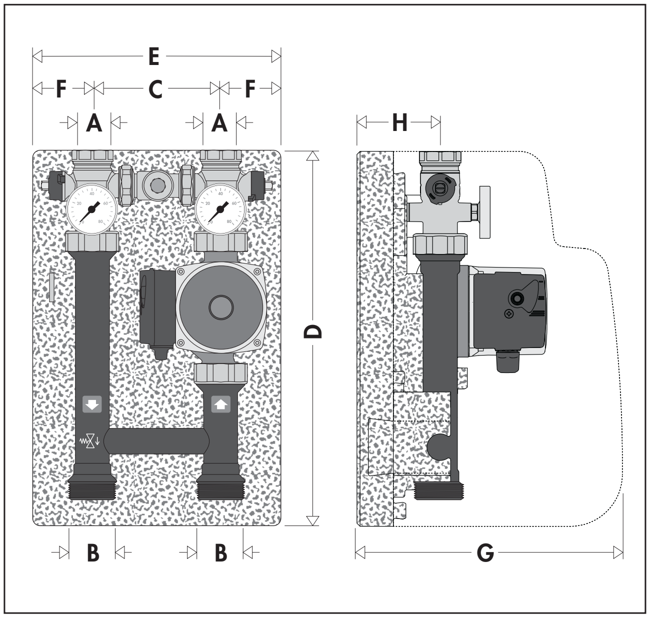
Adjustable Differential by-pass valve code 519006 (optional)
Body: brass
Valve plug: EPDM

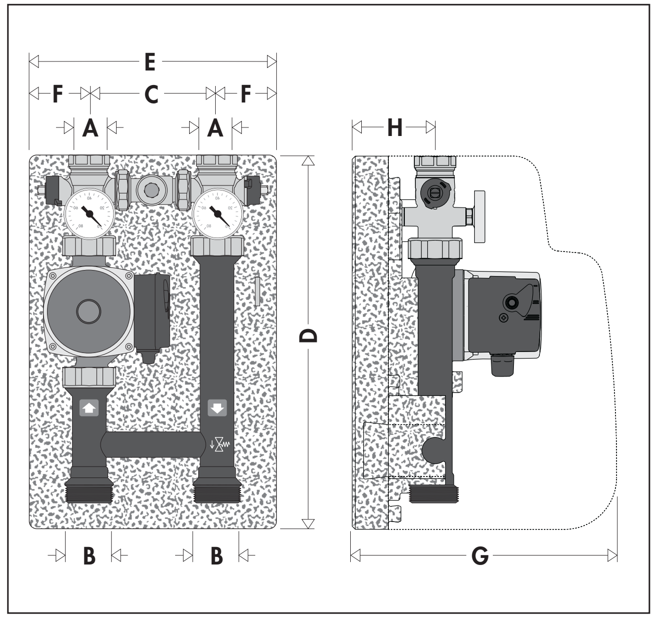
Spring: stainless steel
Seals: EPDM
Max. working pressure: 145 psi (10 bar)
Max. working temperature: 212°F (100°C)
By-pass differential pressure setting range: 0.3 – 4.3 psi (2 – 30 kPa)

Connections: 1” male x 1” male (straight)

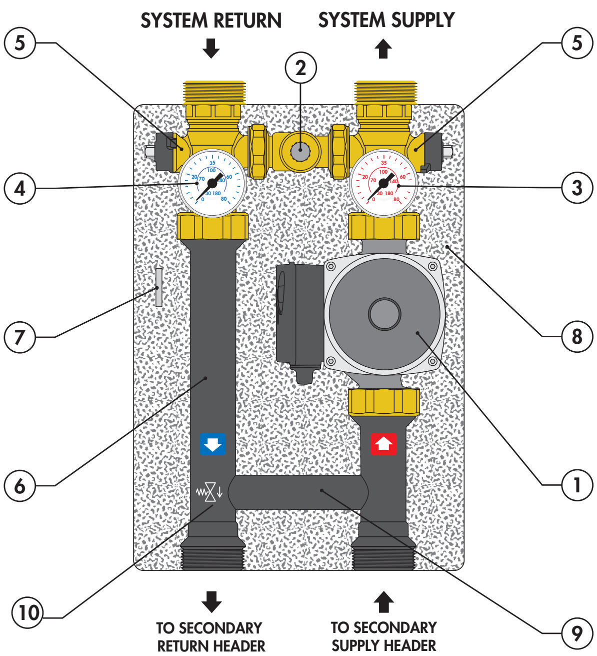
Wall bracket code 165001 (optional)
Material: stainless steel

Dimensions

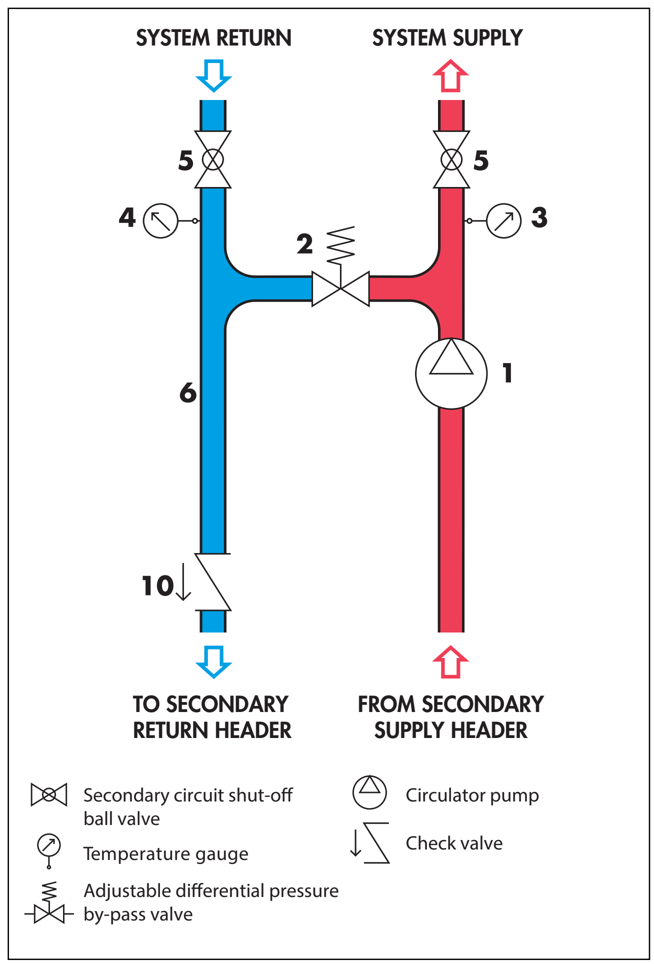




A	B	C	D	E	F	G	H	Weight (lb)
1¼"	1½"	5"	14¼"	9¾"	3"	10"	3¼"	21



Hydraulic diagram



Component Characteristics

1 Variable-speed pump Alpha 25-55U or three-speed pump UPS 15-58
2 Adjustable differential pressure by-pass valve (optional-shown in this view)*
3 Supply temperature gauge
4 Return temperature gauge
5 Secondary circuit shuf-off ball valves
6 Connecting pipes
7 Wrench to adjust secondary circuit shut-off ball valves
8 Insulation
9 Spacer (no flow path)
10 Check valve
*Factory set-up includes a blind spacer (no flow path)

HydroMixer™ thermostatic mixing unit

166 series

Function

The 166 series HydroMixer™ provides a fixed temperature flow to low temperature secondary heating circuits (such as floor radiant panels), complete with onboard three-speed or variable-speed pump, thermostatic three-way mixing valve with built-in temperature sensor, check valve, supply and return temperature gauges, secondary circuit shut-off ball valves and pre-formed insulation shell. Versions are available with either the three-speed Grundfos UPS 15-58 pump or the variable-speed Alpha 25-55U pump.

The unit can be ordered with supply flow and pump on the right or left side, and the unit is field reversible to accommodate changing installation requirements. The adjustable differential pressure by-pass valve, code 519600 and wall bracket code 165001 are optional. The unit comes with male union connections which require separately purchased, 1 inch sweat or NPT, top outlet and bottom inlet fitting kits.

Product Range

Code 166600A Thermostatic fixed temperature mixing unit with Grundfos UPS 15-58 pump on right side............connections 1" sweat or NPT union*
Code 166610A Thermostatic fixed temperature mixing unit with Grundfos UPS 15-58 pump on left side.............connections 1" sweat or NPT union*
Code 166602A Thermostatic fixed temperature mixing unit with Grundfos Alpha 25-55U pump on right side.......connections 1" sweat or NPT union*
Code 166612A Thermostatic fixed temperature mixing unit with Grundfos Alpha 25-55U pump on left side..........connections 1" sweat or NPT union*
*Fitting sets sold separately

Technical Characteristics

Materials

Three-way thermostatic mixing valve

Body:	brass
Shutter:	PSU
Spring:	stainless steel
Seals:	EPDM

Connecting pipes

Material:	steel

Check valve

Body:	brass
Shutter:	PPAG40

Shut-off ball valves

Body:	brass

Performance

Suitable fluids:	water, glycol solution
Max. percentage of glycol:	30%
Max. working pressure:	145 psi (10 bar)
Adjustable temperature range:	80–125°F (25–50°C)
Primary inlet max. temperature:	212°F (100°C)

Connections:	male union threads
- top outlets:	1-1/4" male straight
- bottom inlets:	1-1/2" male straight
- inlet/outlet center distance:	5 inches (125 mm)

Pump

Three-speed pump:	Grundfos: UPS 15-58-130;
Variable-speed pump:	Grundfos: ALPHA 25-55U;
Body material:	cast iron
Power supply:	115 V 50/60 Hz
Protection class:	Class F
Pump center distance:	5-1/8" (130 mm)
Pump connections:	1 1/2" male straight

Temperature gauges

Dual scale:	32 –176°F (0 – 80°C)

Technical specification of insulation

Material:	EPP
Thickness:	1 1/8" (30 mm)
Density:	3 lb/ft^3 (45 kg/m^3)
Working temperature range:	25–250°F (-5–120°C)
Thermal conductivity:	0.037 W/(m·K) at 10°C
Fire resistance (UL94):	class HBF

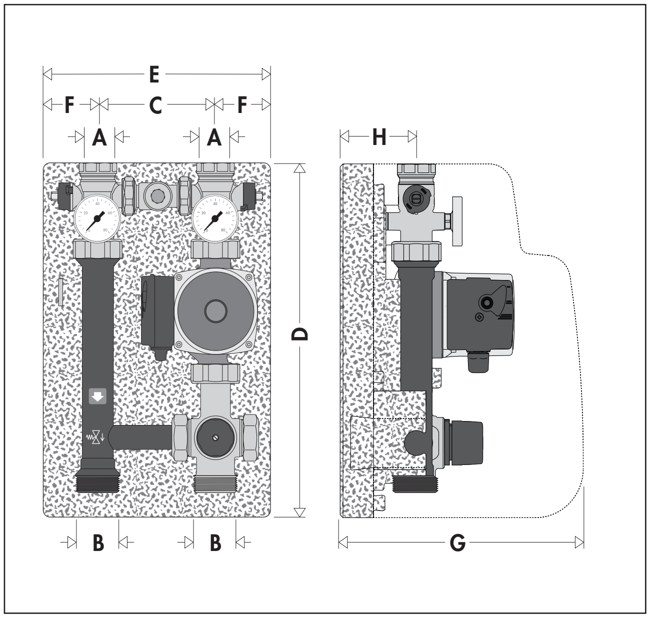
Adjustable Differential by-pass valve code 519006 (optional)

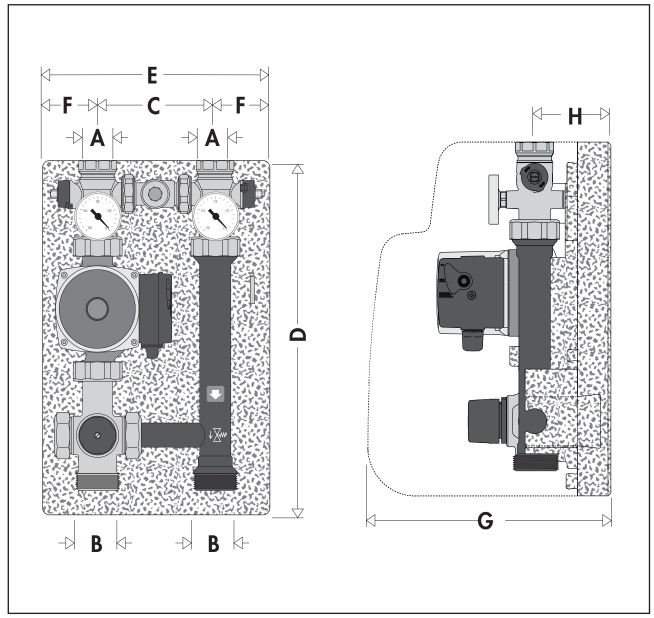
Body:	brass
Valve plug:	EPDM
Spring:	stainless steel
Seals:	EPDM
Max. working pressure:	145 psi (10 bar)
Max. working temperature:	212°F (100°C)
By-pass differential pressure setting range:	0.3 – 4.3 psi (2 – 30 kPa)
Connections:	1" male x 1" male (straight)

Wall bracket code 165001 (optional)

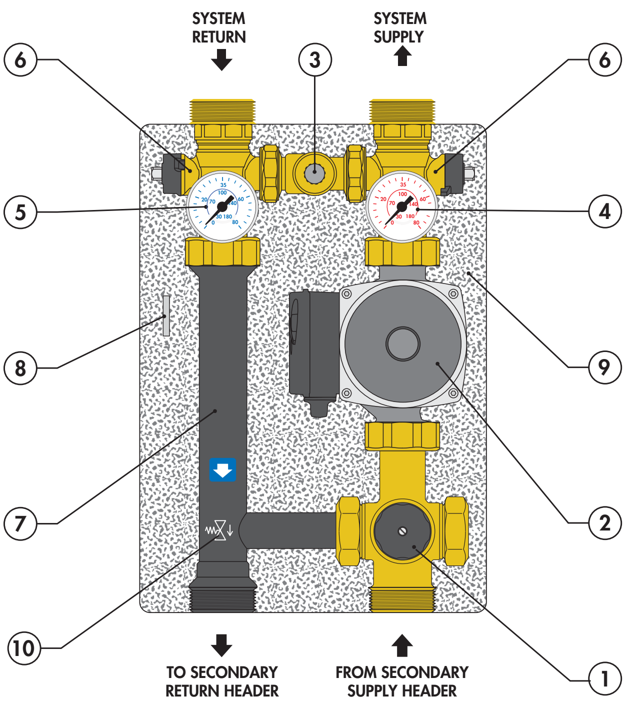
Material:	stainless steel

Dimensions





A	B	C	D	E	F	G	H	Weight (lb)
1¼"	1½"	5"	14¼"	9¾"	3"	10"	3¼"	22

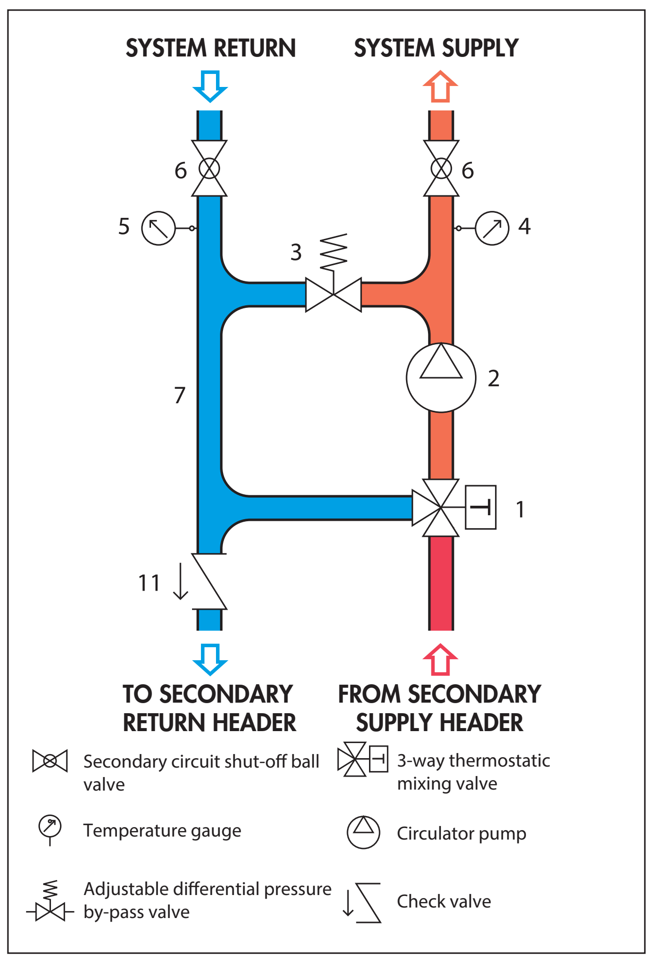


Component Characteristics

1 Three-way mixing valve with built-in temperature sensor
2 Variable-speed pump Alpha 25-55U or three-speed UPS 15-58 pump
3 Adjustable differential pressure by-pass valve (optional - shown in view)*
4 Supply temperature gauge
5 Return pressure gauge
6 Secondary circuit shut-off ball valves
7 Connecting pipes
8 Wrench to adjust secondary circuit shut-off ball valves
9 Insulation
10 Check valve

***Factory set-up includes a blind spacer (no flow path)**

Hydraulic diagram



HydroMixer™ motorized mixing unit

167 series

Function

The 167 series HydroMixer™ provides a modulating three-point floating type non-spring return motorized three-way mixing valve to regulate the fluid temperature in heating and air conditioning systems in response to a separately-sourced outdoor reset controller. It comes complete with onboard three-speed or variable-speed pump, motorized three-way mixing valve, check valve, supply and return temperature gauges, secondary circuit shut-off ball valves and pre-formed insulation shell. Versions are available with either the three-speed Grundfos UPS 15-58 pump or the variable-speed Alpha 25-55U pump.

The unit can be ordered with supply flow and pump on the right or left side. The adjustable differential pressure by-pass valve, code 519600 and wall bracket code 165001 are optional. The unit comes with male union connections which require separately purchased, 1 inch sweat or NPT, top outlet and bottom inlet fitting kits.

Product Range

Code 167600A	Motorized temperature mixing valve with Grundfos UPS 15-58 pump on right side.................	connections 1” sweat or NPT union*
Code 167610A	Motorized temperature mixing valve with Grundfos UPS 15-58 pump on left side...................	connections 1” sweat or NPT union*
Code 167602A	Motorized temperature mixing valve with Grundfos Alpha 25-55U pump on right side.............	connections 1” sweat or NPT union*
Code 167612A	Motorized temperature mixing valve with Grundfos Alpha 25-55U on left side.........................	connections 1” sweat or NPT union*

*Fitting sets sold separately

Technical characteristics

Materials

Motorized three-way valve

Body:	brass
Valve plug:	brass
Seals:	EPDM

Connecting pipes

Material:	steel

Check valve

Body:	brass
Shutter:	PPAG40

Shut-off ball valves

Body:	brass

Actuator (three point floating type non-spring return)

Synchronous motor	
Supply voltage:	24 VAC
Power consumption:	6 VA
Micro-switch contact rating:	6 (2) A (24 V)
Time to rotate 90° both directions:	50 s
Max. ambient temperature:	14 – 130°F (-10°C – 55°C)
Wire lead length:	31 in. (79 cm)

Performance

Suitable fluids:	water, glycol solution
Max. percentage of glycol:	30%
Max. working pressure:	145 psi (10 bar)
Primary inlet temperature range:	40–212°F (5–100°C)
Connections:	male union threads
- top outlets:	1-1/4” male straight
- bottom inlets:	1-1/2” male straight
- inlet/outlet center distance:	5 inches (125 mm)

Pump

Three-speed pump:	Grundfos: UPS 15-58-130;
Variable-speed pump:	Grundfos: ALPHA 25-55U;
Body material:	cast iron
Power supply:	115V 50/60 Hz
Protection class:	Class F
Pump center distance:	5-1/8” (130 mm)
Pump connections:	1 1/2” male straight

Temperature gauges

Dual scale:	32 –176°F (0 – 80°C)

Technical specification of insulation

Material:	EPP
Thickness:	1 1/8” (30 mm)
Density:	3 lb/ft^3 (45 kg/m^3)
Working temperature range:	25–250°F (-5–120°C)
Thermal conductivity:	0.037 W/(m·K) at 10°C
Fire resistance (UL94):	class HBF

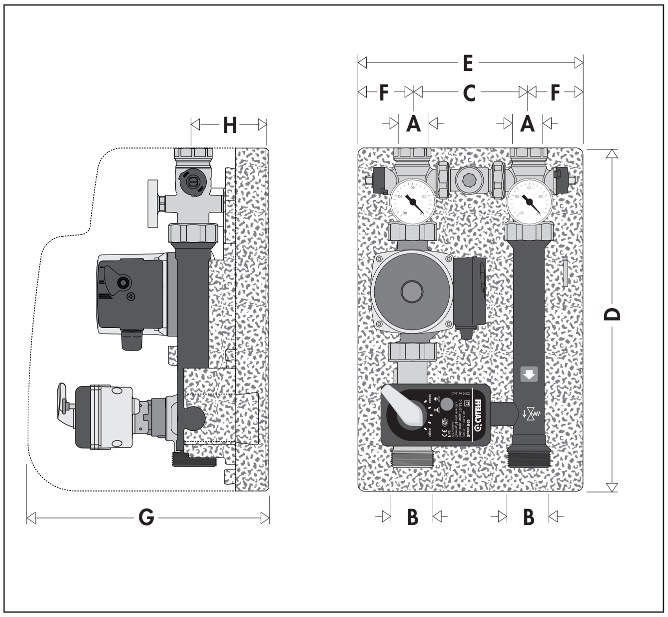
Adjustable Differential by-pass valve code 519006 (optional)

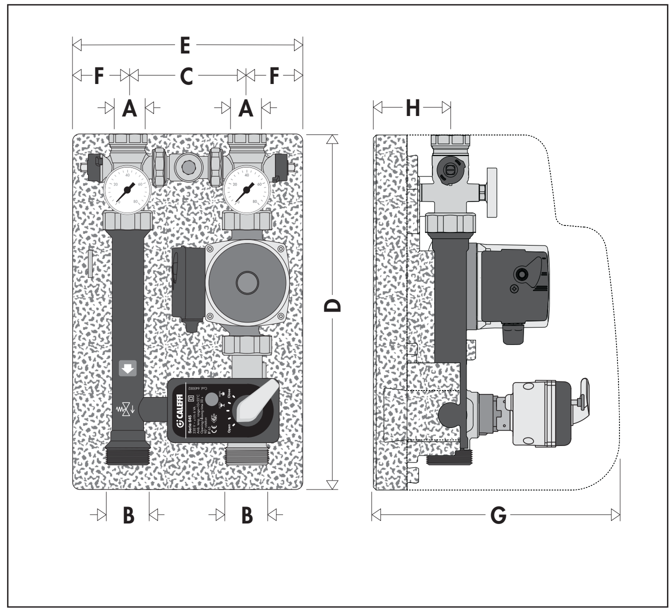
Body:	brass
Valve plug:	EPDM
Spring:	stainless steel
Seals:	EPDM
Max. working pressure:	145 psi (10 bar)
Max. working temperature:	212°F (100°C)
By-pass differential pressure setting range:	0.3 – 4.3 psi (2 – 30 kPa)
Connections:	1” male x 1” male (straight)

Wall bracket code 165001 (optional)

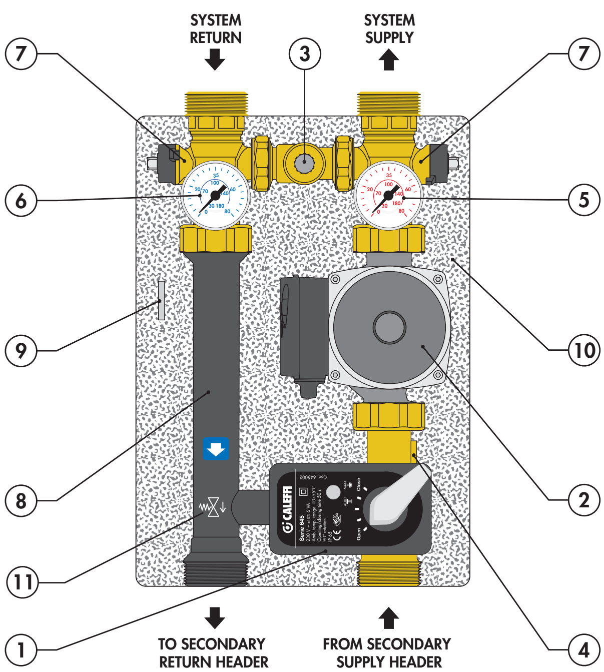
Material:	stainless steel

Dimensions





A	B	C	D	E	F	G	H	Weight (lb)
1¼"	1½"	5"	14¼"	9¾"	3"	10"	3¼"	23

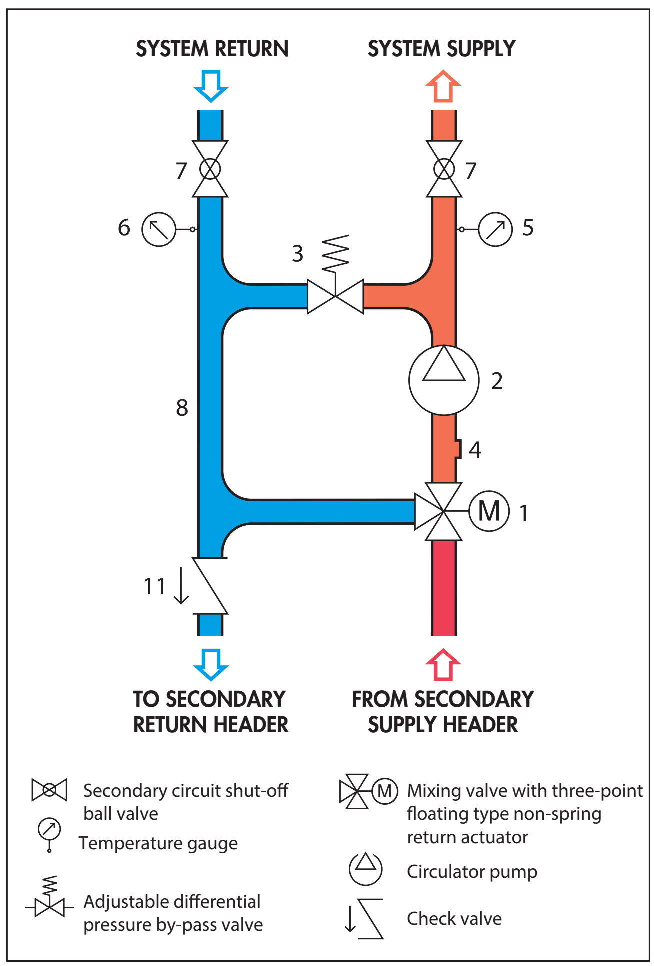


Component characteristics

1 Mixing valve with three-point floating non-spring return actuator
2 Three-speed pump: Alpha 25-554, UPS 15-58
3 Adjustable differential pressure by-pass valve (Optional - shown in this view)*
4 Sensor well
5 Supply temperature gauge
6 Return temperature gauge
7 Secondary circuit shut off ball valves
8 Connecting pipes
9 Wrench to adjust secondary circuit shut-off ball valves
10 Insulation
11 Check valve

***Factory set-up includes a blind spacer (no flow-path)**

Hydraulic Diagram



Manifold motorized mixing station

171 series

Function

The 171 series manifold mixing station is designed for use in manifold-based hydronic distribution systems. The manifold mixing station incorporates a modulating three-point floating actuator to regulate the temperature of the fluid sent to the system flow manifold according to the actual thermal load, in response to a separately-sourced outdoor reset controller. A removable primary circuit hydraulic separator with check valve is also supplied.

The hydraulic separator is essential when there is a primary circuit circulation pump and when radiator circuits or fan coils are controlled by thermostatic or thermo-electric valves. When connecting to a Caleffi HYDROLINK™ or hydraulic separator without a primary pump, the hydraulic separator can be removed and the manifold mixing station can be connected directly.

The 171 station, like the TWISTFLOW™ Series 668S1 distribution manifolds, can be configured with 3 to 13 circuit outlets offering similar benefits with built-in sight flow meters/adjustable balancing valves and optional TWISTOP™ thermo-electric zone actuators.

Product Range

Series 171 Pre-assembled Manifold Mixing Station with flow gauges, floating actuator for use with outdoor reset controller
- Three-speed pump Grundfos UPS 15-58-130 or variable-speed pump Grundfos ALPHA 25-55U
- size 1 ¼” manifold, 3 to 13 outlets ¾” male, ¾” supply and return connections
- supply and return manifolds can be inverted by custom order, so supply and return lines feed from overhead

Technical characteristics

Materials

Three-way mixing valve unit:	
Body:	brass
Bonnet:	brass
Shutter:	stainless steel
Seals:	EPDM
Top elbow with supply temperature gage:	brass
Primary circuit hydraulic separator	
Body:	brass
Check valve:	POM
Spring:	stainless steel
Shut-off valves	
Body:	brass
Ball:	brass, chrome plated
Supply and Return Manifolds	
Body:	brass
Springs:	stainless steel
Seals:	EPDM
End fittings:	brass
Automatic air vent:	brass
Drain valve:	brass

Performance

Suitable fluids:	water, glycol solutions
Max. percentage of glycol:	30%
Control temperature range:	70–170°F (20–78°C)
Primary inlet temperature:	40–210°F (5–100°C)
Max. working pressure:	150 psi (10 bar)
Min. opening pressure for primary circuit check valve:	1.5 psi (10 kPa)
Temperature gauge scale:	30 - 210°F

Connections

- primary circuit:	¾” NPT female
- to mixing valve unit:	1” Female with nut
- manifold circuit outlets:	¾” male
- outlet center distance:	2” (50 mm)

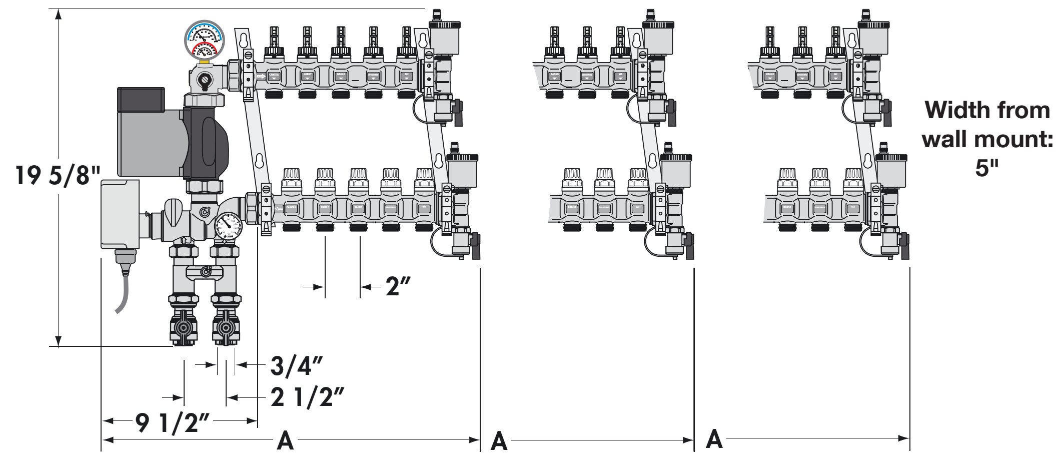
Actuator (three-point floating type)

Supply voltage:	24 VAC
Full stroke time:	50 s (rotation 120°)
Power consumption:	8 VA
Current draw:	0.335 A
Auxiliary switch capacity:	0.8 A, 24 V
Protection class:	IP 44
Max. ambient temperature:	30°F (55°C)
Protective cover:	self-extinguishing VO

Pump

Three-speed pump:	Grundfos: UPS 15-58-130
Variable-speed pump:	Grundfos: ALPHA 25-55U
Body material:	cast iron
Power supply:	115V 50/60 Hz
Max. ambient temperature:	105°F (40°C)
Protection class:	Class F
Pump center distance:	5 1/8” (130 mm)
Pump connections:	1½” male straight

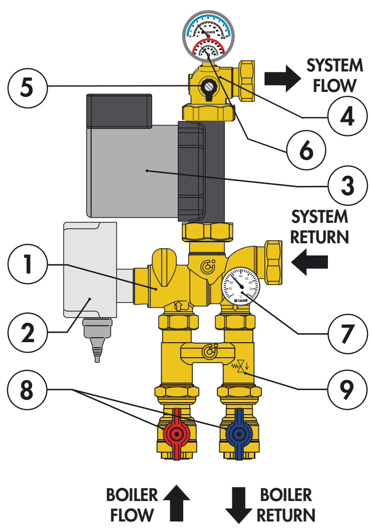
Dimensions

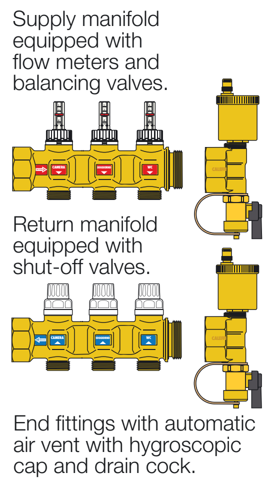


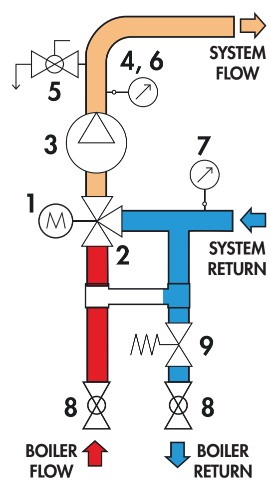
Outlets	3	4	5	6	7	8	9	10	11	12	13
A	19"	21"	23"	25"	27"	29"	32"	34"	36"	38"	40"

Characteristic components / hydraulic diagram

Item	Description	Symbol
1	Three-way mixing valve	
2	Three-point floating actuator	
3	Three-speed circulation pump UPS 15-58	
4	Top elbow with supply temperature and pressure gauge	
5	Purge valve	
6	Supply temperature and pressure gauge	
7	Return temperature gauge	
8	Primary circuit shut-off valves	
9	Primary circuit hydraulic separator with check valve	

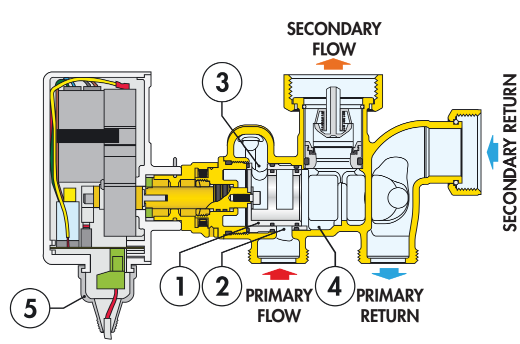






Operating Principle

The fluid temperature is controlled by a three-way mixing valve, operating on a control signal from an outdoor reset controller sensing mixed secondary circuit supply temperature and hot boiler flow temperature to the mixing valve hot inlet port. The flow in the valve is regulated by a shutter (1) that opens and closes the hot water flow port (2) and the water return port from the manifold circuits (3) to adjust the desired system flow temperature. Even if the secondary circuit thermal load or the inlet temperature from the boiler changes, the mixing valve automatically adjusts the flow rates until it obtains the set secondary flow temperature.



Manifold thermostatic mixing station

172 series

Function

The 172 series manifold mixing station is designed for use in manifold-based hydronic distribution systems. The manifold mixing station incorporates a thermostatic actuator with built-in sensor which keeps the flow temperature at a constant set value for use in low temperature systems such as floor radiant panels. A removable primary circuit hydraulic separator with check valve is also supplied.

The hydraulic separator is essential when there is a primary circuit circulation pump and when radiator circuits or fan coils are controlled by thermostatic or thermo-electric valves. When connecting to a Caleffi HYDROLINK™ or hydraulic separator without a primary pump, the hydraulic separator can be removed and the manifold mixing station can be connected directly. The 172 station, like the TWISTFLOW™ Series 668S1 distribution manifolds, can be configured with 3 to 13 circuit outlets offering similar benefits with built-in sight flow meters adjustable balancing valves and optional TWISTOP™ thermo-electric zone actuators.

Product Range

Series 172 Pre-assembled Manifold Mixing Station with flow gauges with thermostatic actuator to provide constant temperature flow.
- Three-speed pump Grundfos UPS 15-58-130 or variable-speed pump Grundfos ALPHA 25-55U
- size 1 ¼” manifold, 3 to 13 outlets ¾” male, ¾” supply and return connections
- supply and return manifolds can be inverted by custom order, so supply and return lines feed from overhead

Technical characteristics

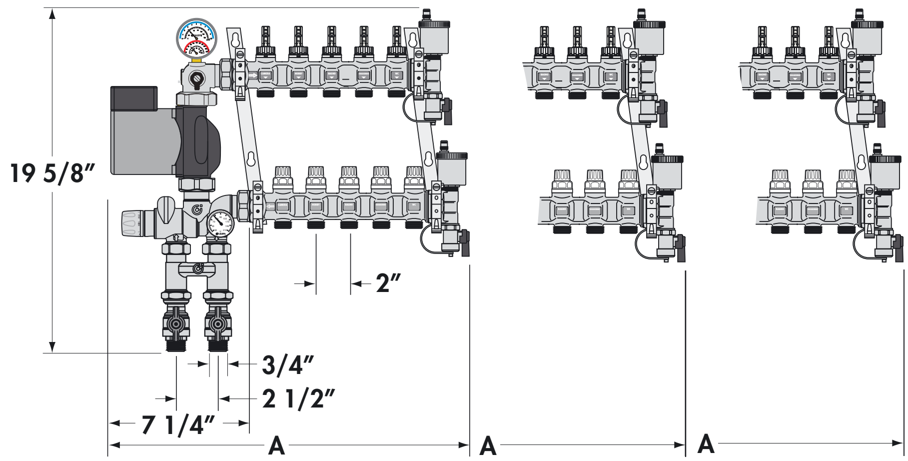
Materials

Three-way mixing valve unit:	
Body:	brass
Bonnet:	brass
Shutter:	PSU
Seals:	EPDM
Top elbow with supply temperature gage:	brass
Primary circuit hydraulic separator	
Body:	brass
Check valve:	POM
Spring:	stainless steel
Shut-off valves	
Body:	brass
Ball:	brass, chrome plated
Supply and Return Manifolds	
Body:	brass
Springs:	stainless steel
Seals:	EPDM
End fittings:	brass
Automatic air vent:	brass
Drain valve:	brass
Performance	
Suitable fluids:	water, glycol solutions
Max. percentage of glycol:	30%
Control temperature range:	80–130°F (25–55°C)
Primary inlet temperature:	40–210°F (5–100°C)
Max. working pressure:	150 psi (10 bar)
Min. opening pressure for primary circuit check valve:	1.5 psi (10 kPa)
Temperature gauge scale:	30 - 210°F

Connections

- primary circuit:	¾” NPT female
- to mixing valve unit:	1” Female with nut
- manifold circuit outlets:	¾” male
- outlet center distance:	2” (50 mm)
Pump	
Three-speed pump:	Grundfos: UPS 15-58-130
Variable-speed pump:	Grundfos: ALPHA 25-55U
Body material:	cast iron
Power supply:	115V 50/60 Hz
Max. ambient temperature:	105°F (40°C)
Protection class:	Class F
Pump center distance:	5 1/8” (130 mm)
Pump connections:	1½” male straight

Dimensions

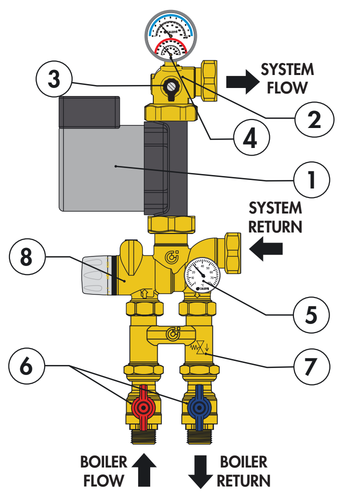


Width from wall mount: 5"

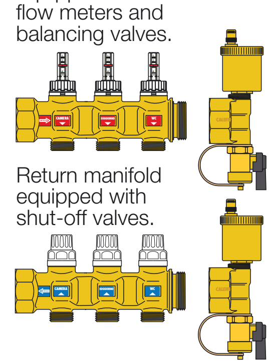
Outlets	3	4	5	6	7	8	9	10	11	12	13
A	17"	19"	21"	23"	25"	27"	30"	32"	34"	36"	38"

Characteristic components / hydraulic diagram

Item	Description	Symbol
1	Circulation pump UPS 15-58 pictured	
2	Top elbow with supply temperature and pressure gauge	
3	Purge valve	
4	Supply temperature and pressure gauge	
5	Return temperature gauge	
6	Primary circuit shut-off valves	
7	Primary circuit hydraulic separator with check valve	
8	Thermostatic three-way mixing valve with built-in sensor	

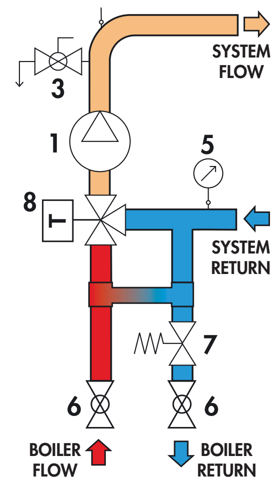


Supply manifold equipped with flow meters and balancing valves.



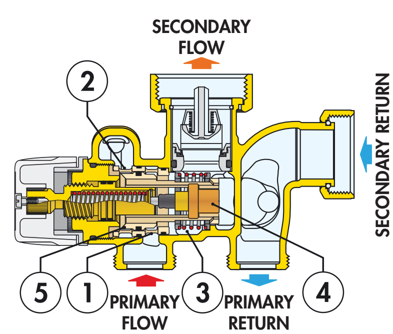
Return manifold equipped with shut-off valves.

End fittings with automatic air vent with hygroscopic cap and drain cock.



Operating Principle

The fluid temperature is controlled by a thermostatic three-way mixing valve regulated by a thermostatic sensor (4) located in the mixed water outlet chamber (3) of the valve. By expanding and contracting, it continuously ensures a correct proportioning of hot water coming from the boiler, and water returning from the manifold circuit. The water intake is regulated by an internal cartridge, consisting of a piston (5) that slides inside a cylinder, located between the hot water flow (1) and the water returning from the circuit (2). Even if the secondary circuit thermal load or the inlet temperature from the boiler changes, the mixing valve automatically adjusts the flow rates until it obtains the set secondary flow temperature.

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