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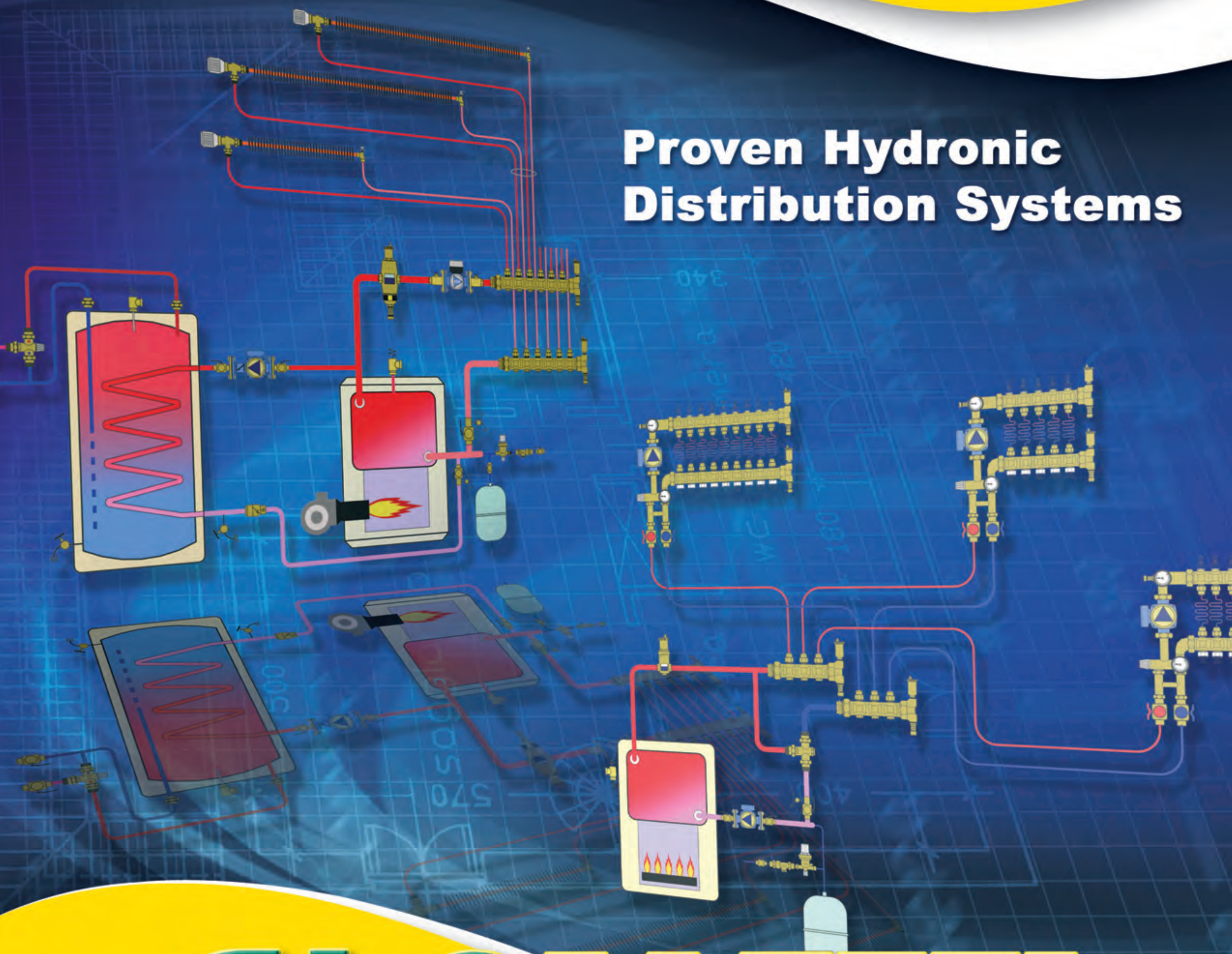
JOURNAL OF DESIGN INNOVATION FOR HYDRONIC PROFESSIONALS

Proven Hydronic Distribution Systems



CALEFFI

A Technical Journal
from
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Dear Hydronic and Plumbing Professional,

Caleffi North America offers an on-the-road class entitled: “Near Boiler Piping”. This popular presentation discusses components located close to the boiler, how they should be arranged and why they are vital to good system performance.

This issue of idronics extends this discussion to complete hydronic distribution systems. It explores the interactions of components within properly designed systems. It presents several proven distribution system layouts, along with their strengths and limitations. It explains interactions between the distribution system, the heat source and the heat emitters. It helps readers decide which type of distribution system is best for a given application.

After presenting proven distribution system layouts, the focus turns to what not to do. Section 7 identifies common errors related to distribution system design, explains what’s wrong and presents at least one alternative that eliminates the problem.

Section 8 wraps up with several complete system examples.

The information presented in this issue is based on decades of experience, and covers both “traditional” and contemporary approaches. References to previous issues of idronics are provided so readers can find additional details on specific topics.

We hope you enjoy this 19th issue of idronics and encourage you to send any feedback by e-mail to idronics@caleffi.com.

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Mark Olson

Mark Olson

General Manager & CEO

INDEX

Proven Hydronic Distribution Systems

1. INTRODUCTION

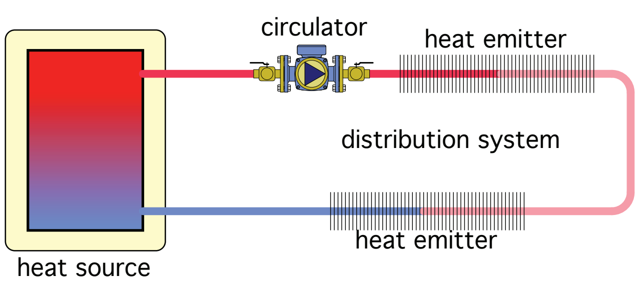
Hydronic systems use water as a "conveyor belt" for heat.

In a heating application, thermal energy (e.g., heat) is created at a heat source, "loaded" onto a stream of water, carried by that water to where it is needed within the building, and "unloaded" from the water into the space at one or more heat emitters. The water is neither the source of the heat, nor its final destination — it's only a means of transport.

In a cooling application, heat is absorbed into a stream of chilled water at one or more heat absorbers, carried back to a mechanical room, transferred to a medium at higher temperature, and rejected to the atmosphere, a body of water, or into the earth. Again, water serves only as the conveyance medium for the heat.

The portion of a hydronic system that lies between the heat source (or the chiller in a cooling system) and the heat emitters (or heat absorbers in a cooling system) is called the *distribution system*. Its function is to distribute heated water or chilled water to all areas of the building when and where heating or cooling is needed. Figure 1-1 shows the simplest concept for a hydronic heating distribution system.

Figure 1-1



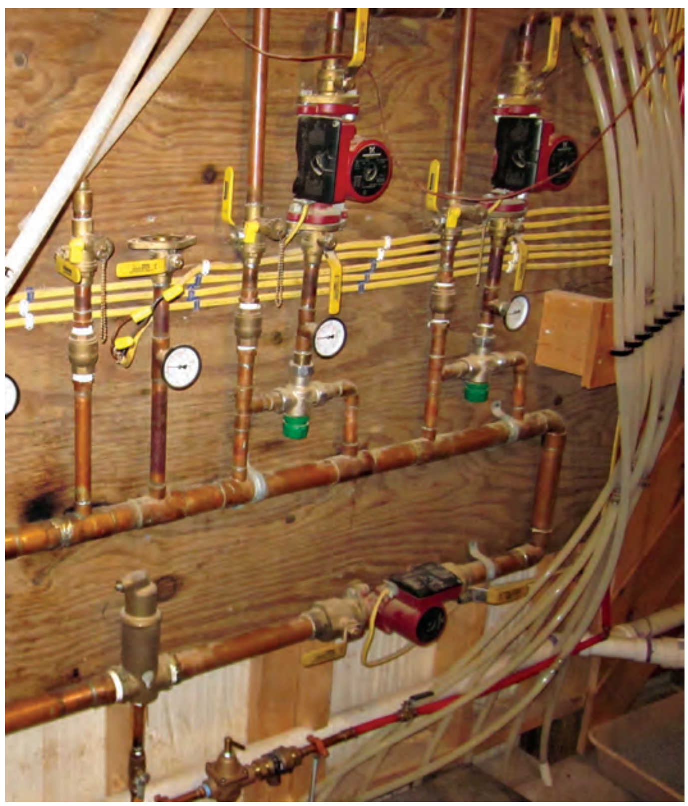
Unfortunately, the attention given to planning hydronic distribution systems is often less than that given to selecting a heat source or chiller. This could be compared to designing a race car by focusing mostly on engine selection, while overlooking opportunities to improve the aerodynamics of the body, the efficiency of the transmission, or the rolling resistance of the tires. The resulting vehicle will be less than optimal, even though the "best" engine was used.

A *poorly designed distribution system can ruin the performance of any hydronic system.*

Sadly, the vast majority of complaints about insufficient or poorly proportioned heat delivery or cooling ability are traceable to poorly designed or installed distribution systems, rather than improperly performing heat sources or chillers.

The hydronic distribution systems used in North America often deviate from established and proven approaches. In some cases, a distribution system ends up being a "morphing" of two or more standard design approaches. While such an approach can — with skilled design —

Figure 1-2

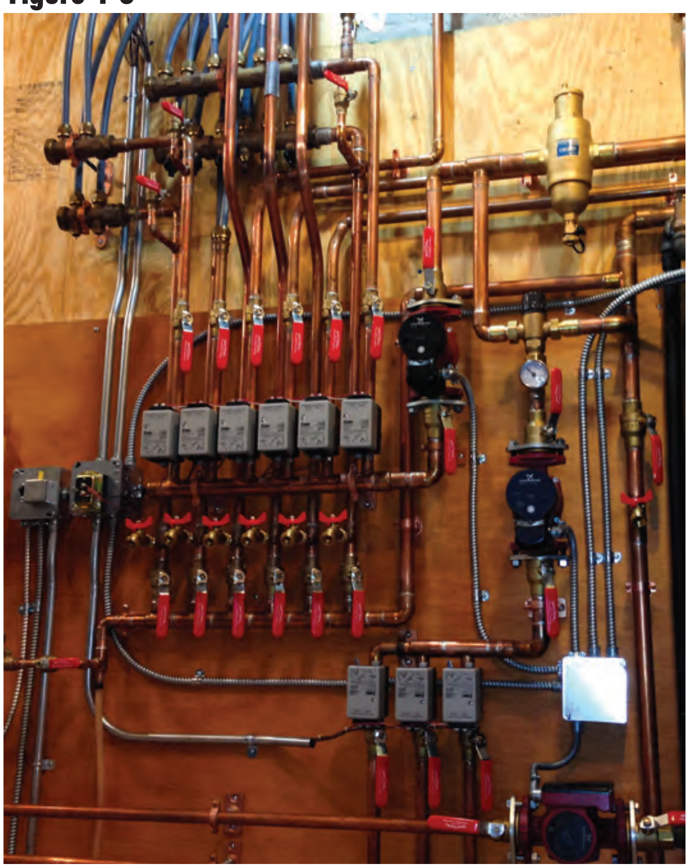


provide a unique solution in some situations, it is much more likely to produce disappointing results, especially if the designer is not absolutely sure how the deviations will perform. Furthermore, these "one-up" distribution systems often lack proper documentation, which makes them difficult to understand and difficult to service by those not familiar with the original installation.

The preferred approach, whenever possible, is to stay with proven design concepts that properly and reliably address issues such as hydraulic separation, differential pressure control, head loss through valves, and control of supply water temperature.

This issue of *idronics* focuses on basic and *proven* distribution system designs that have been successfully used in a wide range of hydronic heating and cooling applications. It discusses the strengths and limitations of each approach, and provides design tips that help optimize each type of system. It also discusses many common errors related to design or detailing of hydronic distribution systems, and presents relevant solutions. Frequent reference is also made to previous issues of *idronics* that can provide additional information on specific subjects.

Figure 1-3



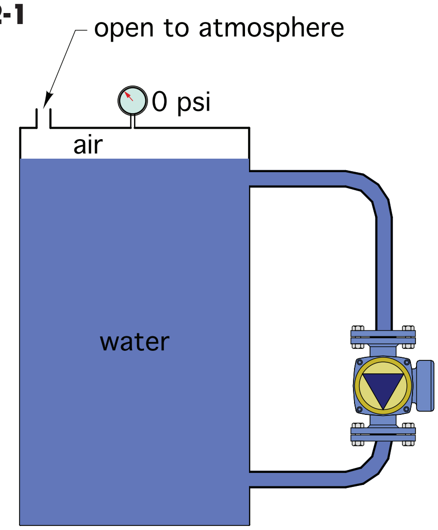
2. BASIC CONCEPTS & DETAILING

There are several fundamental concepts that need to be understood prior to designing a hydronic distribution system. This section presents them. It also provides references to previous issues of *idronics* in which the concept is discussed in more detail.

OPEN HYDRONIC SYSTEMS

Any hydronic system in which the water contacts the atmosphere outside the system, even through a small opening, is an "open" hydronic system. An example is shown in figure 2-1.

Figure 2-1

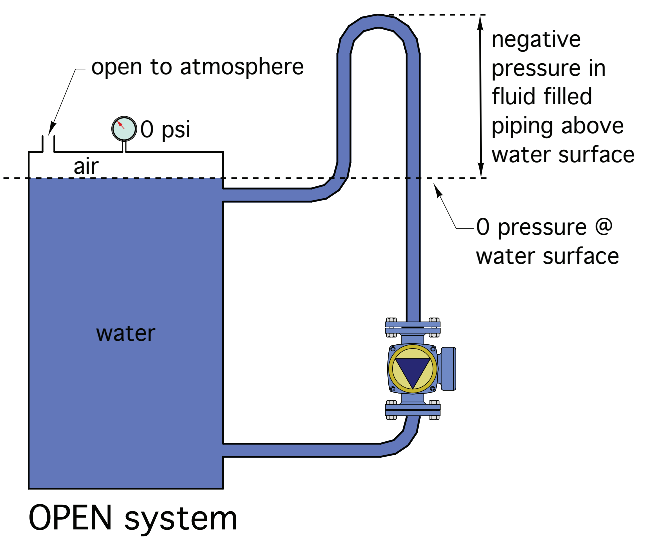


Open hydronic systems have several characteristics that must be respected.

First, the point at which the water contacts the atmosphere will remain at atmospheric pressure at all times. This is not necessarily a problem in systems where piping only rises a few feet above the level where water contacts the atmosphere. However, the pressure in all piping above the water line level will be under negative pressure relative to the atmosphere, as shown in figure 2-2.

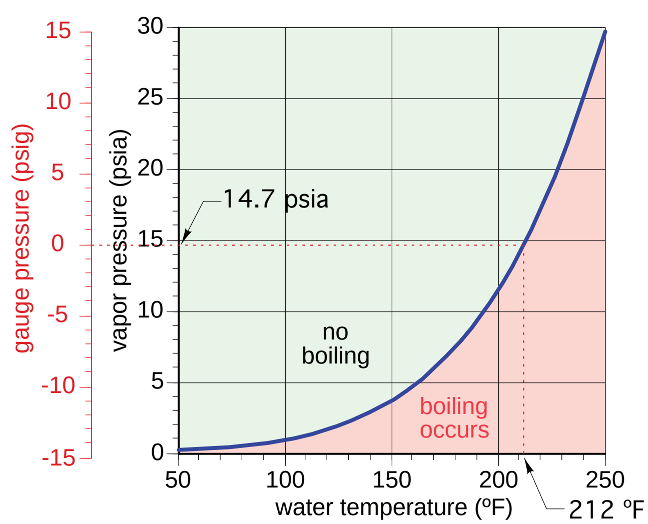
If the sub-atmospheric (e.g., negative) pressure of the water drops to or below the water's vapor pressure, immediate boiling occurs. The vapor pressure of water is highly dependent on its temperature. A commonly known correlation is a vapor pressure of 14.7 psi absolute pressure at a temperature of 212ºF (or 100ºC). The earth's atmosphere exerts an absolute pressure of approximately 14.7 psia at sea level, and thus an open container of water at sea level will boil if the water reaches 212ºF.

Figure 2-2



Water will boil at temperatures well below 212ºF under negative pressure. Figure 2-3 shows the vapor pressure of water over a range of absolute pressures, as well as the more commonly used "gauge" pressure scale.

Figure 2-3



Boiling must be avoided in hydronic systems. This requires that the combination of temperature and pressure of the water remain safely within the green shaded area of figure 2-3. It is prudent to provide a margin of safety against the water approaching the blue vapor pressure curve in figure 2-3, on the basis of temperature or pressure.

This generally requires that any open-loop hydronic system not have piping that is routed more than a few feet above the water line in the system. Ideally, most, if not all of the piping in an open system should be below the water line.

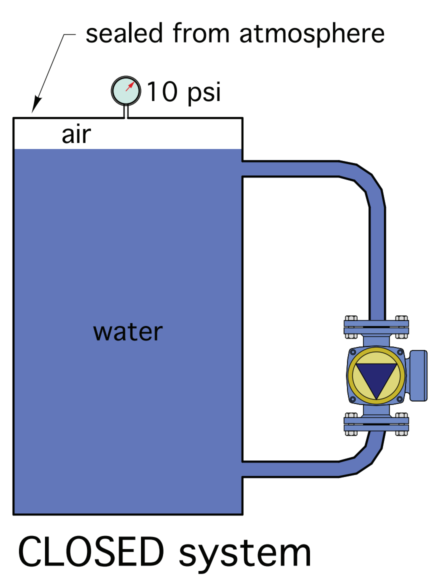
CORROSION POTENTIAL IN OPEN SYSTEMS

Another concern with open hydronic systems is the ability of water to reabsorb oxygen molecules from the location where the water in the system contacts the atmosphere. As water cools, its ability to absorb these gases increases. As water is heated, its ability to hold oxygen and nitrogen molecules in solution with H_20 molecules decreases. The repetitive heating and cooling of water in a hydronic system, in combination with the water/atmosphere interface, allows the water to transport oxygen molecules throughout the system. If this oxygen contacts ferrous materials, such as steel or cast iron, corrosion will occur. Over time, this corrosion can cause premature failure of any ferrous metal components, such as boilers, circulators, valves, panel radiators or expansion tanks. *Thus, ferrous metal components should never be used in open hydronic systems.* Components made of stainless steel, copper, brass, bronze or engineered polymers such as PEX are generally required in open systems.

CLOSED HYDRONIC SYSTEMS

Any system in which all contact between the fluid within the system and the atmosphere is blocked is classified as a "closed" hydronic system. The majority of residential and commercial hydronic systems currently used in North America fall into this category.

Figure 2-4



Because they are sealed from the atmosphere, properly designed and maintained closed hydronic systems experience very little entry of oxygen, and thus have far lower potential for oxygen-based corrosion than do open systems.

To avoid the previously described oxygen-based corrosion, boilers made of carbon steel or cast iron should only be used in closed systems. Likewise, steel panel radiators, steel expansion tanks, cast iron circulators, or any valves and fittings made of steel or cast iron should only be used in closed systems.

Closed hydronic systems can operate at positive pressure relative to the atmosphere. Many residential and light commercial hydronic systems operate with positive pressures in the range of 5 to 25 psi in the lower portion of the system.

In large industrial hydronic systems, the ability to operate at positive pressures allows water temperatures well above 212ºF. Although possible, most of the modern hydronic systems used in residential and light commercial buildings seldom need to (or should) operate at temperatures exceeding 200ºF. Even lower water temperatures are preferred when they are compatible with the system's heat emitters.

All closed hydronic systems containing any source of heat generation must be equipped with a pressure relief valve. This valve automatically opens at a specific pressure rating to prevent any higher pressure from developing within the system. Most residential and light commercial hydronic heating systems are equipped with pressure relief valves that open at 30 psi pressure.

HEAT SOURCE CHARACTERISTICS

There are hundreds of devices available that can serve as a heat source for a hydronic system. They include boilers fueled by natural gas, fuel oil, propane, wood and electricity. They also include a range of hydronic heat pumps that gather low temperature heat from air or water, and transfer that heat to a stream of water at higher temperatures. Both flat plate and evacuated tube solar thermal collectors can also serve as hydronic heat sources.

Some hydronic systems also use two or more independently operated heat sources. For example, a residential system may use a boiler in combination with an air-to-water heat pump to provide space heating, cooling and domestic hot water.

From the standpoint of compatibility with specific distribution systems, heat sources can be classified as follows:

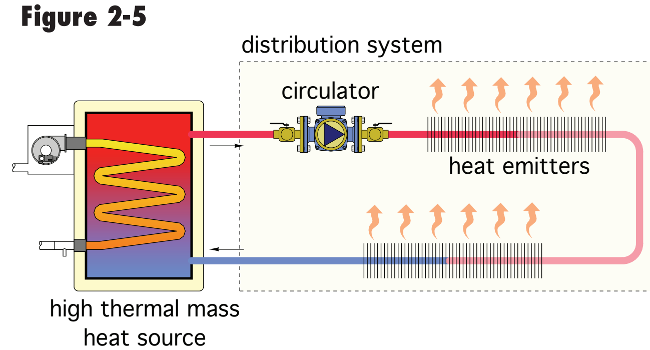
1. High thermal mass versus low thermal mass
2. High flow resistance versus low flow resistance
3. Conventional versus condensing

HIGH THERMAL MASS VS. LOW THERMAL MASS HEAT SOURCES

A high thermal mass heat source contains significantly more metal or water relative to a low thermal mass heat source. Examples of high thermal mass heat sources include cast iron boilers, steel fire-tube boilers, and high water-content, "tank-type" modulating/condensing hydronic heating appliances.

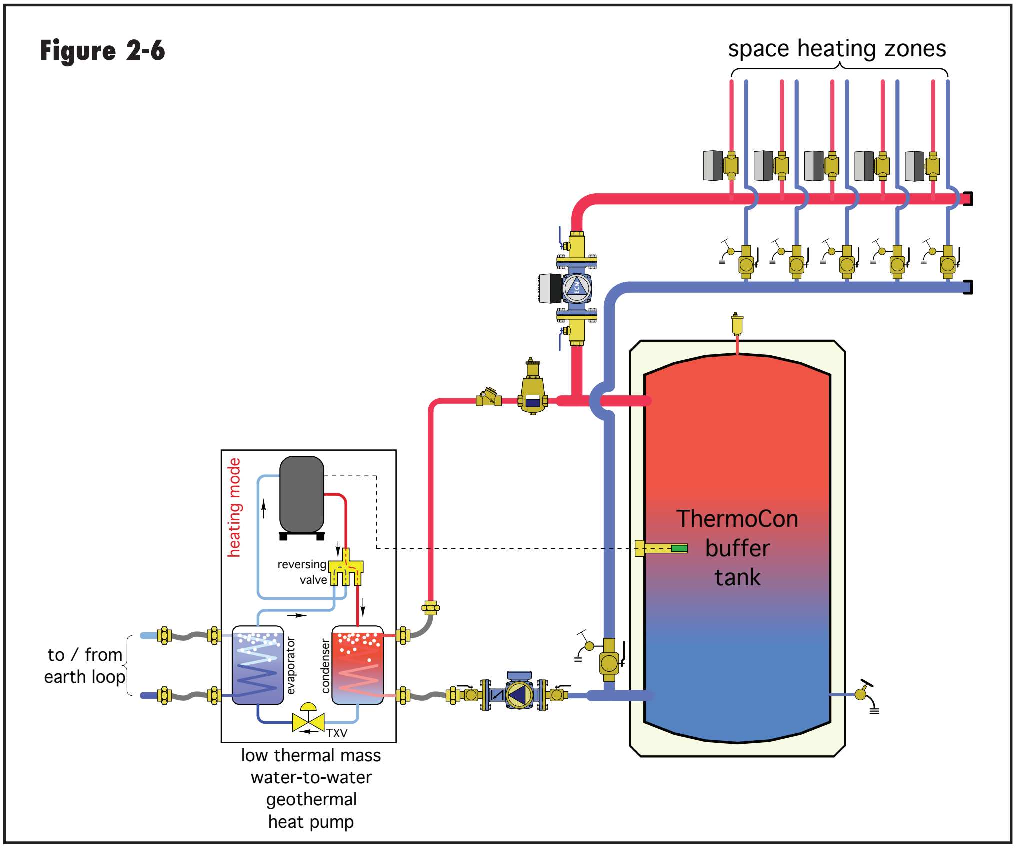
High thermal mass provides stability in a system that contains many zones. It protects the heat source from short cycling under partial load conditions. For many common applications, one can treat high thermal mass heat sources as "self-buffering." This term implies that no additional buffer tank is needed.

Figure 2-5



Other hydronic heat sources contain minimal amounts of metal and water. They can be categorized as low thermal mass heat sources. Examples include modulating/condensing boilers with compact heat exchangers, electric boilers, and hydronic heat pumps with coaxial heat exchangers.

Low thermal mass heat sources can quickly heat up to normal operating temperatures when called to operate. Their low thermal mass also implies that very little residual heat remains in the heat source following shutdown. These heat sources are relatively light in comparison to high mass cast iron and steel boilers, and in some cases can be mounted to a wall rather than placed on a concrete floor.



The red curves are pump curves for two different circulators. The higher pump curve is for a more powerful circulator, while the lower red curve is for a lower power circulator.

Assume that the objective is to create a flow of 10 gallons per minute (gpm) through the heat source. Figure 2-7 shows the pump curve of the more powerful circulator intersecting the head loss curve of the high flow resistance heat source at a corresponding flow rate of 10 gpm. Notice, however, that the same 10 gpm flow rate can be achieved using a lower power circulator in combination with the low flow resistance heat source. Lower power circulators have the potential to save hundreds of dollars in operating cost over the life of the system, and thus are preferred whenever they can create the necessary flow rate.

Low thermal mass heat sources are not self-buffering. When combined with zoned distribution systems, they often require use of an external buffer tank to add stabilizing thermal mass to the system, and prevent undesirable short cycling. Figure 2-6 shows an example of this approach in which a low thermal mass water-to-water heat pump is connected to a buffer tank in a system that has several zones.

HIGH FLOW RESISTANCE VS. LOW FLOW RESISTANCE HEAT SOURCES

Many heat sources that would be categorized as *high thermal mass* also have *low flow resistance*. This is beneficial because it reduces the electrical power required by the circulator that creates flow through the heat source. Low flow resistance heat sources typically do not require a dedicated heat source circulator. Conversely, many low thermal mass heat sources have higher flow resistance, and often require a dedicated circulator.

Figure 2-7 illustrates the differences between high flow resistance heat sources and low flow resistance heat sources. The head loss versus flow rate curves (shown in blue) represent this effect. The steeper blue curve represents a high flow resistance heat source. The shallow blue curve is for a low flow resistance heat source.

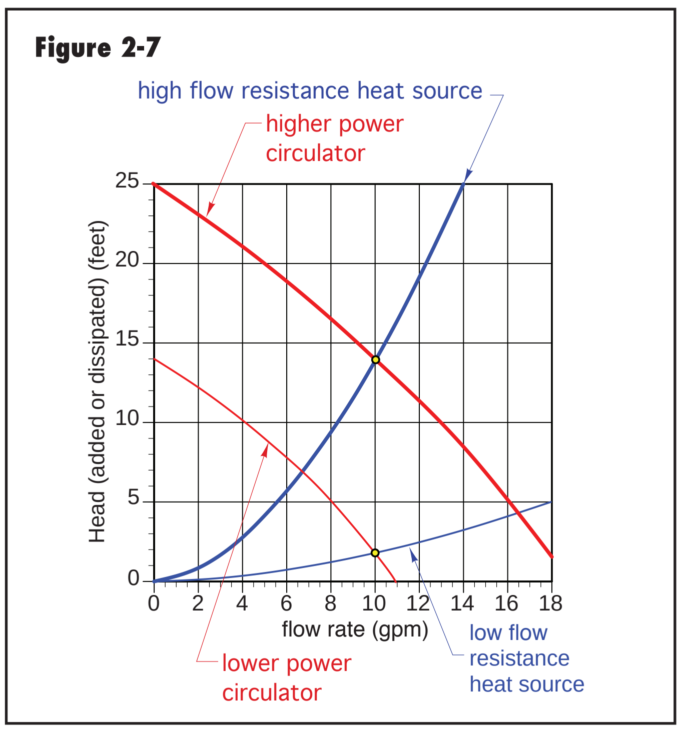
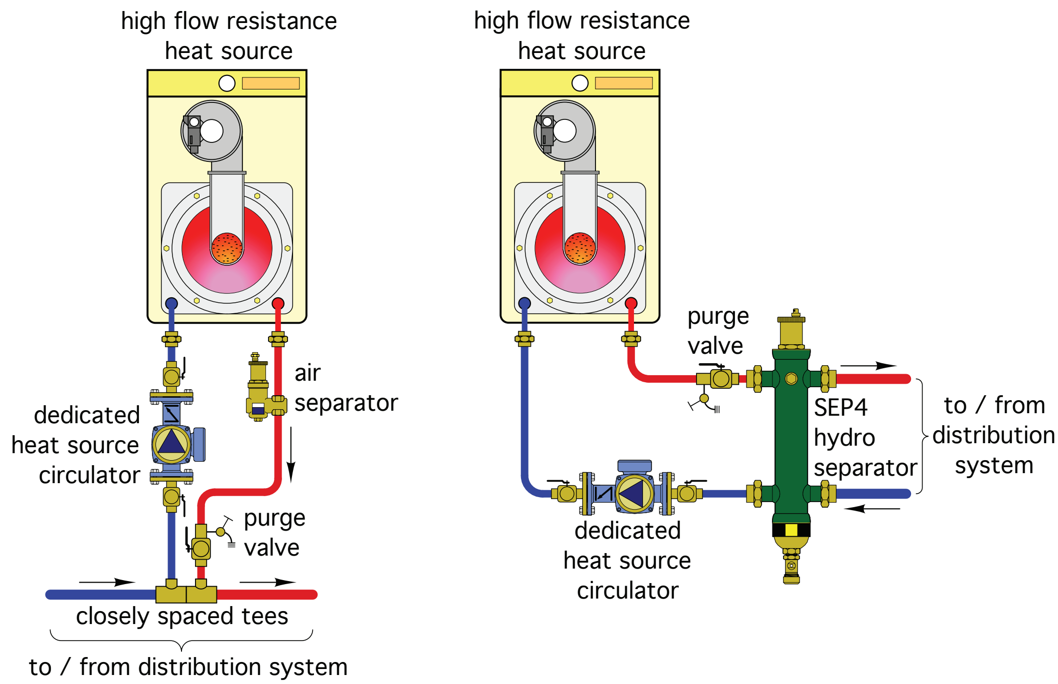


Figure 2-8



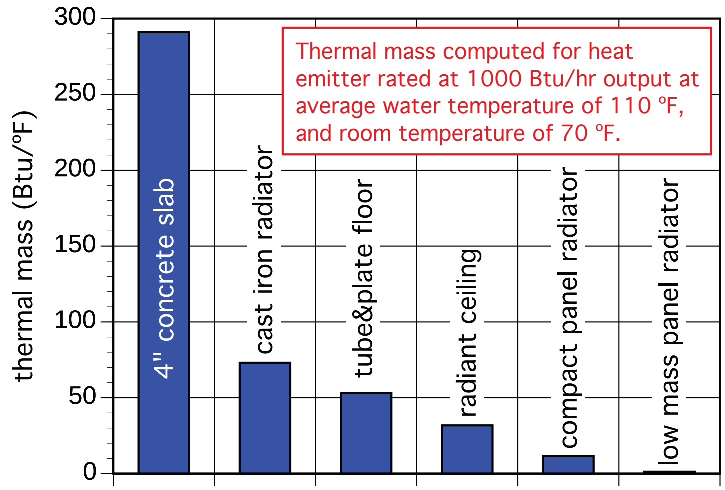
The high flow resistance of some heat sources requires that they be connected to the distribution system in a manner that uses a separate circulator just to create the necessary flow through the heat source. Figure 2-8 illustrates how this can be done using either closely spaced tees or a hydraulic separator.

Both of these piping methods provide hydraulic separation between the dedicated heat source circulator and any other circulators in the system.

HEAT EMITTER CHARACTERISTICS

As is true with heat sources, hydronic distribution systems must also be well matched with the system's heat emitters. This matching requires knowledge of the heat emitter characteristics.

Figure 2-9



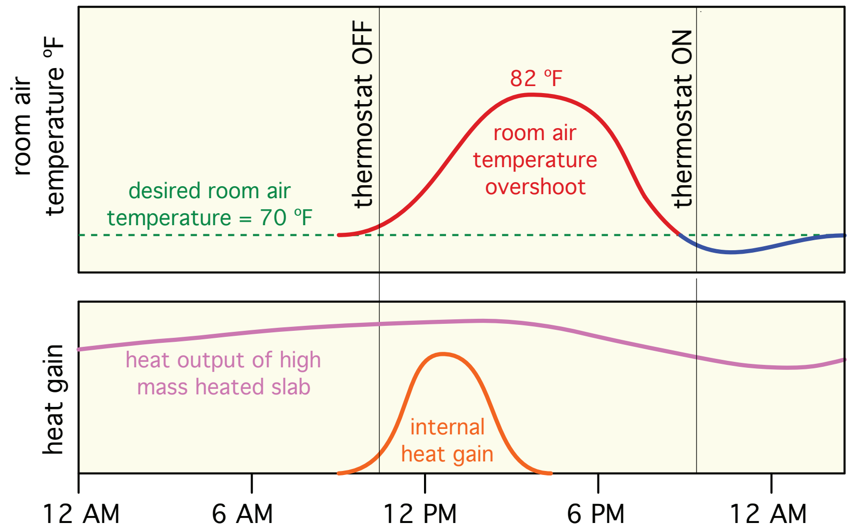
One characteristic that plays a critical role is the thermal mass of the heat emitter(s) used in the system. Some hydronic heat emitters have very high thermal mass. An example would be a 4-inch-thick heated concrete floor. Other heater emitters have very little thermal mass, an example of which would be a fin-tube baseboard convector. Figure 2-9 compares the thermal mass of several common hydronic heat emitters.

Notice that the thermal mass of the 4-inch-thick heated concrete floor is over 100 times that of the low mass panel radiator.

HIGH MASS HEAT EMITTERS

High thermal mass heat emitters can store large amounts of heat in the materials present between the water flowing through them, and the room to which they are delivering heat. This characteristic can create significant differences between the time and rate at

Figure 2-10



which heat is absorbed into the heat emitter from the water passing through it, and the time and rate at which that heat is delivered to the space being heated. The greater the thermal mass of the heat emitter(s), the greater the thermal lag effect can be.

This can cause significant over-shooting and under-shooting of the desired room air temperature. Figure 2-10 illustrates an example.

Consider an interior space heated by a high mass heat emitter such as a heated concrete slab. Assume that the desired goal is to maintain a constant interior temperature of 70ºF. On a cold winter night, there is a continuous, but slowly increasing heat input rate to the slab so that its heat output keeps pace with the heat loss of the space. By morning, the slab has been “fully charged” with heat, and its surface temperature is in the low 80ºF range.

Now assume that the heated slab just described is in a building with large south-easterly facing windows. By mid-morning, there is a significant internal heat gain caused by solar radiation through these windows. Other internal heat gains are also created by occupants and use of interior equipment. This causes interior air temperature to rise. The thermostat senses this and turns off any subsequent heat delivery to the slab when the air temperature rises 1ºF above its 70ºF setpoint.

However, the slab surface is still well above the desired 70ºF indoor temperature and continues to release heat into the space. The result is a significant temperature overshoot as the residual heat “drains” from the slab over several hours. Such overheating is undesirable. Occupants are likely to deal with it by opening windows for ventilation, or even turning on the building’s cooling system. Comfort is comprised and energy is wasted.

This thermal lag effect can be minimized by:

1. Not using high mass heat emitters in spaces that are subject to significant and/or unanticipated internal heat gains from people, sunlight or interior equipment.

2. Using outdoor reset control to keep the supply water temperature to the heat emitter just high enough to meet the current prevailing heating load. Properly applied, outdoor reset control produces nearly continuous circulation through the system’s heat emitters, and thus helps equalize the rate of heat input to the emitter with the rate the emitter releases heat to the space.

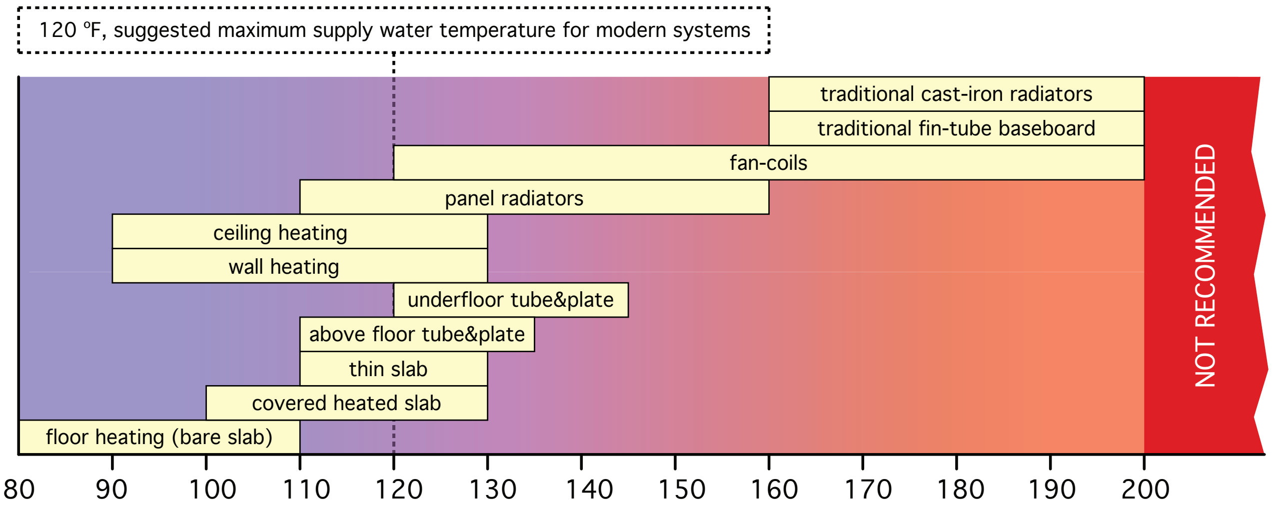
idronics #7 provides a detailed discussion of outdoor reset control.

In the right application, high thermal mass heat emitters can also provide *beneficial* effects. For example, consider the use of a heated concrete floor in a garage-type facility, such as a vehicle maintenance building, fire station or aircraft hangar. When a large door in such a facility is opened on a cold day, there is an influx of cold outside air over the surface of the heated slab. Because heat is stored in the slab, it responds with an immediate “surge” of heat output to counteract the effect of the cold air. This surge can be sustained for several minutes and helps restore comfort very quickly after the large door is closed. Thus, high thermal mass floor heating is an excellent choice in many garage-type facilities.

LOW MASS HEAT EMITTERS

Low thermal mass heat emitters such as most fin-tube baseboard convectors, panel radiators and fan-coils have the advantage of fast thermal response. Many can begin releasing heat to their respective spaces within seconds of having heated water passing through them. This allows for better matching between the building’s

Figure 2-11



varying heating load and the heat output from the heat emitters. It also minimizes the potential for temperature overshoot when internal heat gains occur. Low thermal mass heat emitters store very little residual heat, and thus stop emitting heat very quickly after the flow of heated water through them stops.

Low mass heat emitters are recommended in the following situations:

1. When a building is subject to significant and unscheduled internal heat gains.
2. When a building is operated with daily temperature setback and recovery schedules.
3. When frequent changes in room air temperature are required.
4. When fast recovery to normal room temperatures following a setback is expected.

TYPICAL WATER TEMPERATURE REQUIREMENTS FOR HEAT EMITTERS

When selecting heat emitters, one should distinguish between "traditional" supply water temperatures and "advisable" supply water temperatures.

Traditional supply water temperatures are the result of decades of application with little motivation to consider alternatives. Many of these temperature ranges were established when fuel was inexpensive, and the principle driver of system design was minimizing installation cost.

A common example is the *traditional* supply water temperature for fin-tube baseboard. It is typically 180 to 200ºF under design load conditions. It is even possible find heat output ratings for fin-tube baseboard at water temperatures as high as 230ºF!

This is not to suggest that fin-tube baseboard can't be designed around lower supply water temperatures. Indeed, it can. In some buildings, it may even be possible to size fin-tube baseboard for a design load supply water temperature of perhaps 120ºF. In an average house, this would require much more linear footage of baseboard compared to sizing around a more traditional water supply temperature of 180 to 200ºF. It might be impractical from the standpoint of having sufficient wall space to mount the required length of baseboard. However, in a low heating load home, where heating requirements are often less than 1/3 that of average homes, designing a fin-tube baseboard system for such a low supply water temperature may be possible given the available wall space. *Traditional* design practice simply ignores the latter as a possibility.

Figure 2-11 shows a range of *traditional* supply water temperatures for several types of heat emitters *under design load conditions.*

Heat emitters sized for the upper end of the temperature range are difficult (sometimes impossible) to match with modern hydronic heat sources. Examples of such heat sources include mod/con boilers, solar thermal collectors, hydronic heat pumps, and systems that rely on deeply cycled thermal storage. All of the latter heat sources perform at higher efficiency when operated at relatively low water temperatures. Some, such as a typical water-to-water heat pump, cannot *consistently* produce supply water temperatures over 120ºF.

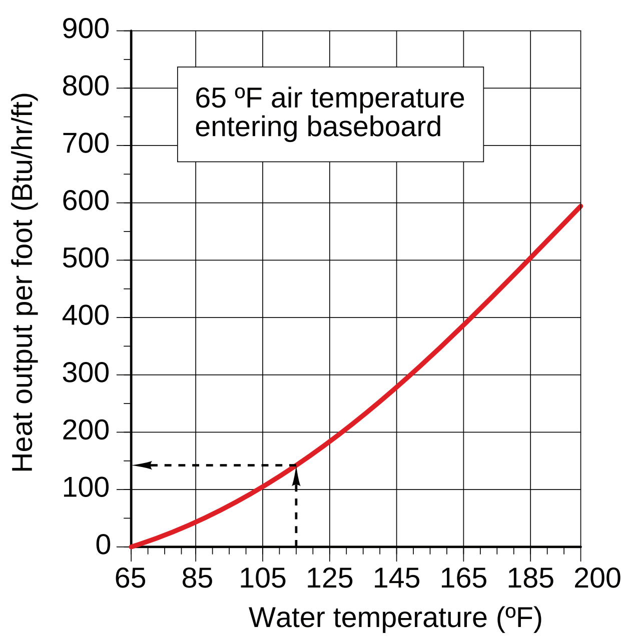
The limited ability, or complete inability, of some modern hydronic heat sources to consistently produce high water temperatures has serious implications for distribution systems and heat emitters.

To achieve the full performance potential of modern hydronic heat sources, distribution systems and the heat emitters they serve must be compatible with relative low supply water temperatures, even during design load conditions.

A suggested criterion that addresses this issue is to design all hydronic heating distribution systems so that they can deliver full design load output using supply water temperatures no higher than 120ºF.

Beyond the compatibility issue with modern hydronic heat sources, this criterion recognizes that well-planned and properly maintained hydronic distribution systems will last many decades, far outliving their original heat source. It also anticipates that future hydronic heat sources, whatever they might be, are more likely to be compatible with lower water temperatures. Thus, considering that a hydronic distribution system installed today is likely to have two, three, or perhaps even more heat sources connected to it over its useful service life, it is prudent to design for compatibility with those future heat sources. Simply put, designing for relatively low supply water temperatures helps "future-proof" hydronic distribution systems.

Figure 2-12



Designers should also keep in mind that heat emitters don't necessarily have to be selected based on "traditional" operating water temperatures. For example, standard residential fin-tube baseboard, which is traditionally sized based on an average circuit temperature of 180ºF, can still operate at much lower temperatures, albeit at much lower heat outputs. Figure 2-12 shows how the heat output of standard residential fin-tube baseboard varies based on the average water temperature in the fin-tube element.

At an average water temperature of 115ºF, the baseboard is still releasing about 140 Btu/hr per foot of active element length. While this is probably not sufficient to maintain comfort in an older energy-inefficient house, it might be a possibility in an energy-efficient house, provided there is sufficient wall space to accommodate the required baseboard length.

Output derating curves are also available for other types of heat emitters. Designers should consider the possibilities before dismissing a given type of heat emitter based on "traditional" water temperature selections. In the case of fan-coils, a suggestion is to ensure that the water temperature supplied to the coil can maintain a discharge air temperature of at least 100ºF. Forced-air distribution systems operating with relatively low supply air temperatures need to be carefully designed to avoid creating drafts.

CHANGES IN WATER DENSITY

Good hydronic system design requires an understanding of several characteristics of water. One that is especially relevant to the design of hydronic distribution systems is how the density of water varies with temperature, and the resulting implications.

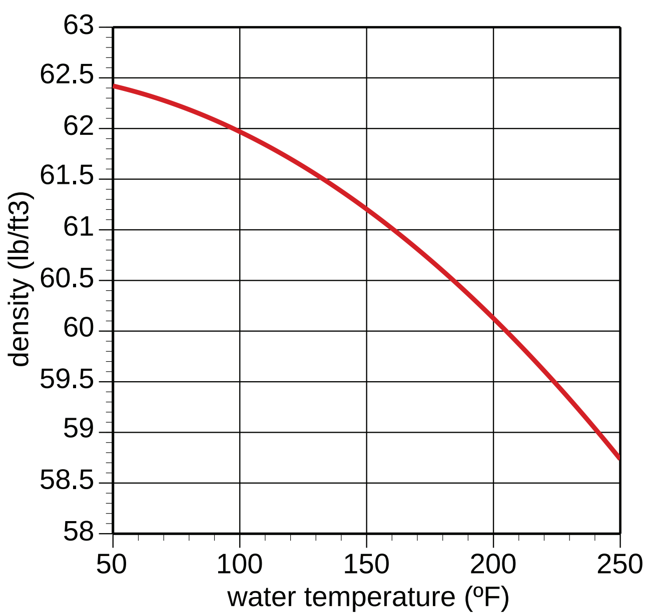
The density of water varies considerably with temperature, as shown in figure 2-13. This variation also occurs with water-based antifreeze solutions.

When water is heated, it expands and becomes less dense, and vice versa. One can also think of heated water as being "lighter" than cooler water.

The reduced density of heated water induces a tendency for it to rise upward in the system relative to the location of cooler water. This characteristic can be beneficial in some circumstances, but very undesirable in others.

Before circulators were available, the weak pressure differential caused by differences in the density of hotter and cooler water provided the sole means of moving

Figure 2-13



water through early hydronic heating systems. Hot water would rise upward from a boiler and pass through heat emitters where it would cool. The cooler water, due to its increased density, would flow downward through other piping, eventually reaching the lower connection on the boiler where the cycle would repeat.

Although this effect was very useful in early hydronic systems, its presence in modern systems is usually unwelcome. Any unblocked piping circuit containing a source of heated water, as well as vertical piping, has the potential to allow a slow but persistent buoyancy-induced flow to develop that will dissipate heat from that circuit, as illustrated in figure 2-14.

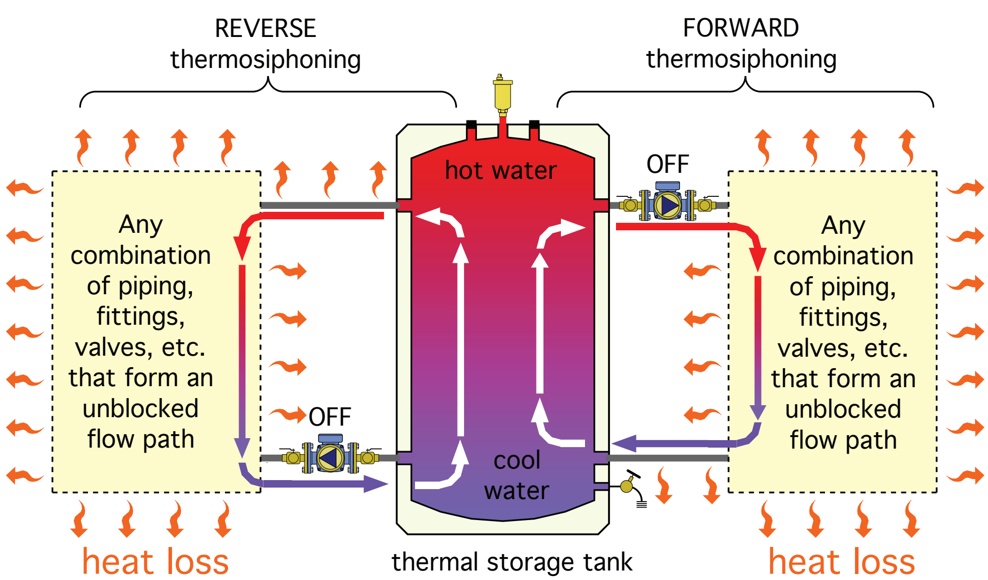
This unintentional flow has been referred to by several names, including reverse thermosiphoning, forward thermosiphoning, ghost flow and heat migration. From the standpoint of thermodynamics, it is nature's way of increasing the entropy of the thermal energy in the system.

Forward thermosiphoning is buoyancy-induced flow through a circuit in the *same direction* as when the circulator is operating. Reverse thermosiphoning is buoyancy-induced flow through a circuit in the *opposite direction* as when the circulator is operating.

Undesirable heat dissipation caused by either forward or reverse thermosiphoning can be prevented in several ways. The most common is to install a device that blocks the flow that would otherwise develop in the piping circuit.

A common swing check valve, such as shown in figure 2-15, can stop *reverse* thermosiphon flow. However, it offers very little forward opening resistance, and thus cannot block forward thermosiphon flow. This characteristic can be useful in situations where only reverse thermosiphon flow needs to be blocked.

Figure 2-14



When installing swing check valves, it is important to provide at least 12 pipe diameters of straight pipe upstream of the valve. This reduces turbulence entering the valve, and helps prevent "rattling" noises from the metal flapper inside the check valve.

It's also important to only mount swing check valves in *horizontal* piping with the bonnet of the valve facing up. Swing check valves are not designed to function in vertical piping, or if mounted upside down.

Figure 2-15

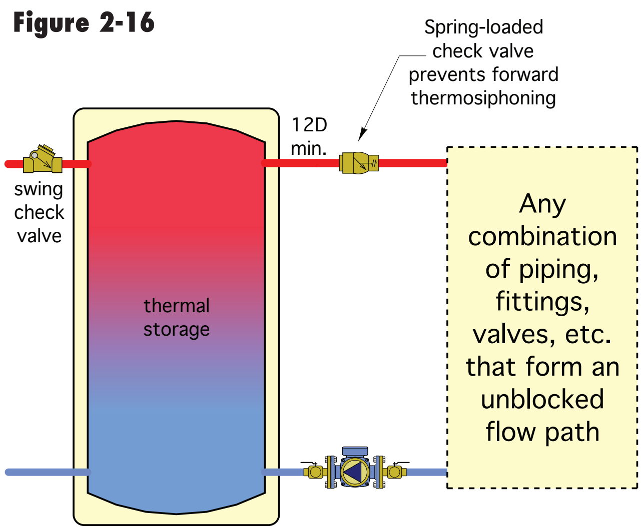
Courtesy of NIBCO

When *both* forward thermosiphoning and reverse thermosiphoning are to be prevented, a check valve with a slight forward opening pressure is required. Two types of valves can provide this characteristic: 1) A spring-loaded check valve, and 2) A weighted plug flow check valve.

A spring-loaded check valve contains a mechanism in which a small stainless steel spring holds the "plug" of the valve against its seat. When the plug in the spring check is against the valve's seat, no flow can pass backwards through the valve.

The internal spring in most spring-loaded check valves creates a forward "threshold pressure" of 0.3 to 0.5 psi. This is generally sufficient to prevent forward thermosiphon flows from developing in systems that are not installed in tall buildings.

Figure 2-16



Many small circulators are now supplied with internal spring-loaded check valve mechanisms. These mechanisms prevent reverse flow through an inactive zone, as well as stop forward thermosiphoning into an inactive zone circuit.

Figure 2-16 shows where a spring-loaded check valve would be installed to prevent forward thermosiphon flow between a thermal storage tank and any unblocked piping path.

Because it relies on a spring rather than gravity, a spring-loaded check valve can be installed in any orientation. However, it should also be installed with a minimum of 12 pipe diameters of straight piping upstream of its inlet to help prevent rattling due to turbulent flow.

AIR SEPARATION DETAILING

Hydronic systems perform best when the water they contain is free of air and dirt. Fortunately, eliminating both air and dirt from hydronic systems is easy with the proper hardware and correct detailing. The discussion that follows summarizes placement of air and dirt separating devices within typical hydronic systems.

Air separation can be divided into two categories:

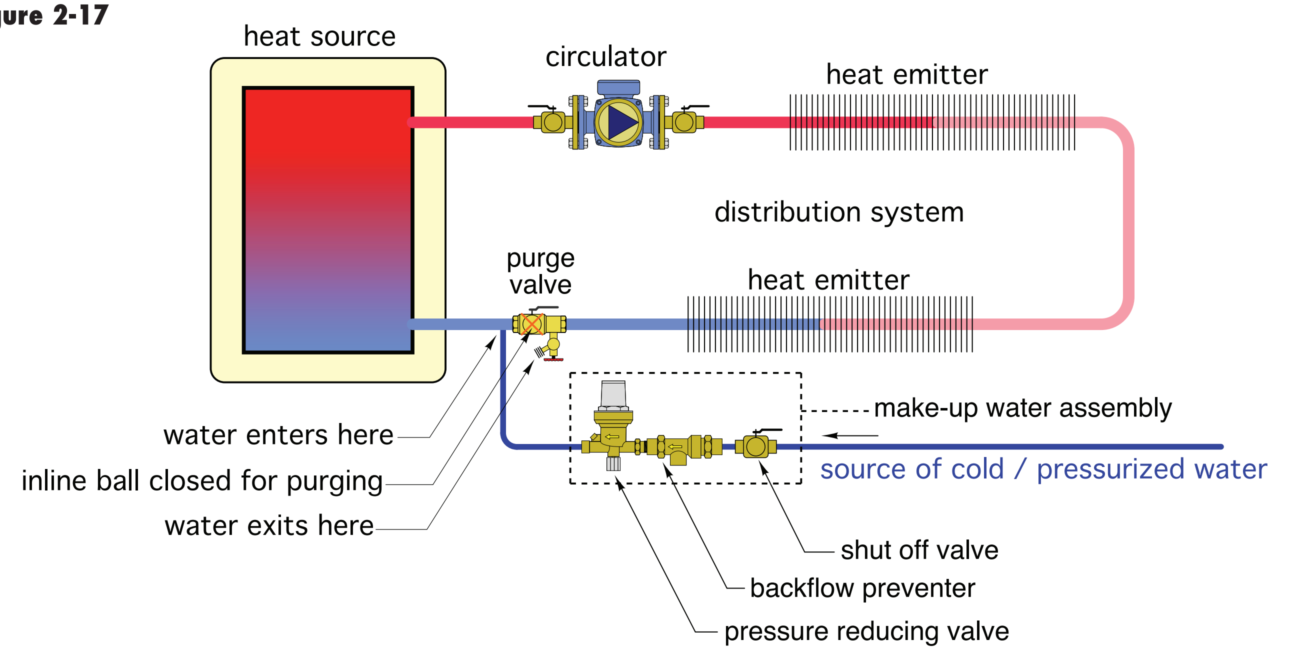
1. Eliminating bulk air from a hydronic system when it is first filled (e.g., purging the system)

2. Providing continuous collection and elimination of small air bubbles that form when the water in the system is first heated or that may be present after a component is serviced and air has entered the system.

Bulk air removal is called "purging" the system. It is usually done by introducing water into the system at a high flow rate, and forcing it to move through one or more circuits in a specific direction. The fast-moving stream pushes air along the piping like a piston pushes gases ahead of it as it moves through a cylinder. The air, and the stream of water pushing it, exit the circuit at one or more specific locations, typically through a special valve called a "purge valve."

The details needed for purging are based on planning where water will enter the system and how it will flow from that point through one or more piping paths, until it can exit through a purge valve placed close to the water entry point. Figure 2-17 shows this concept for a single circuit.

Figure 2-17



A typical purging valve consists of two valve mechanisms in a common body: an inline ball valve, and a side-port ball valve. The inline ball valve is closed to block flow in the piping circuit during purging. The side-port ball valve is opened during purging to provide an exit point for air and some of the water pushing it along.

To purge the system shown in figure 2-17, the inline ball valve is closed, and the side port of the purge valve is opened. The side-port valve should also be connected to a hose that can carry the discharge to a drain or bucket. The make-up water assembly, consisting of a pressure-reducing valve, backflow preventer, and isolating ball valve, is turned on to allow a rapid flow of cold water from the building's plumbing system to enter the hydronic circuit just downstream of the purging valve. Because the inline ball in the purging valve is closed, this water is forced to flow around the entire circuit, pushing air ahead of it. Air will start exiting the side port of the purge valve as soon as water flows into the system. Within a few moments, a mixed stream of air and water will exit the side port of the purge valve. Once the exiting stream is running free of visible air, the side port of the purging valve is closed. Water will enter the system until the set pressure at the pressure-reducing valve is achieved. The inline ball within the purge valve can then be opened. The system should now be purged of bulk air.

In multiple-zone distribution systems, it is good practice to install a purging valve at the return of each zone or other branch circuit.

Although forced-water purging removes most of the bulk air initially in the system, molecules of nitrogen and oxygen remain dissolved in solution with water molecules. A modern "micro-bubble" air separator is designed to capture this dissolved air and eject it from the system.

See idronics #15 for a complete discussion on micro-bubble air separation and dirt separation.

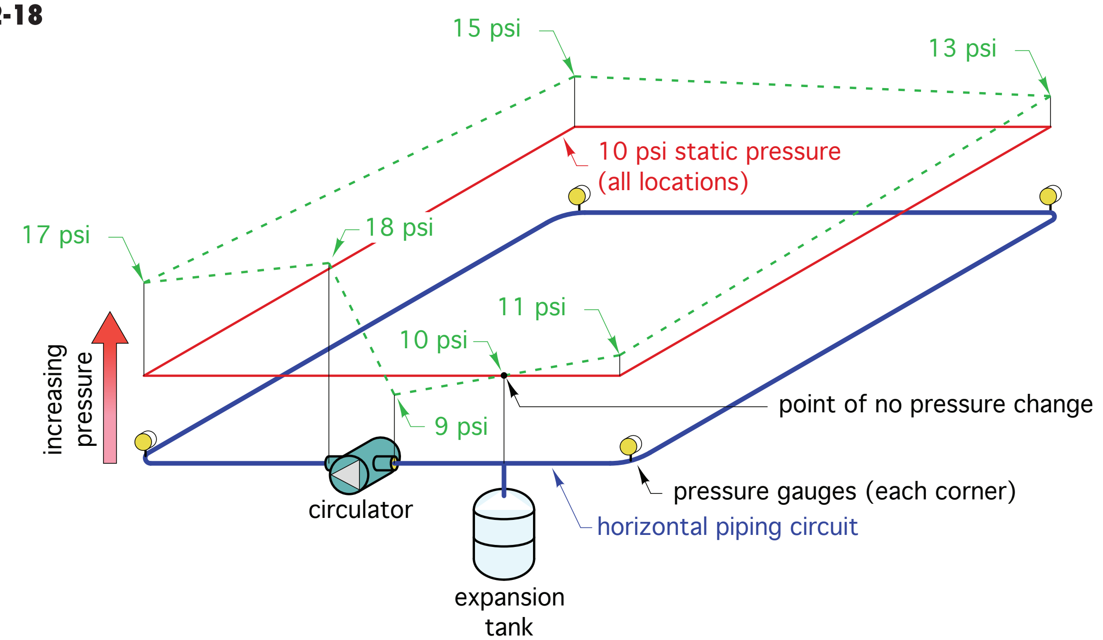
EXPANSION TANK PLACEMENT

All closed-loop hydronic systems require an appropriately sized expansion tank. The captive air volume within this tank provides a "cushion" against which the system fluid will expand when heated. A properly sized expansion tank allows the system fluid to expand and contract while only creating minor pressure variations within the system.

Most modern systems use a diaphragm-type expansion tank. This type of expansion tank contains an elastomeric diaphragm that flexes up and down within the tank shell as water enters and leaves the tank. The diaphragm totally separates the system water from the air contained within the tank shell.

The placement of the expansion tank relative to the circulator significantly affects the pressure distribution in the hydronic system when it operates.

Figure 2-18



The point where the expansion tank connects to the circuit is called the "point of no pressure change" (PONPC). As the name suggests, the pressure at this location doesn't change when the circulator is turned on. However, the pressure at all other locations within the system *will* increase or decrease depending on where the PONPC is located. The most desirable results are attained when the pressure at other locations in the system goes up when the circulator is turned on. This result is achieved when the expansion tank is located *close to the inlet side of the circulator*, as shown in figure 2-18.

When the circulator is turned on, it immediately creates a pressure differential between its inlet and outlet ports. However, the pressure at the point where the expansion tank is connected to the circuit remains the same. The combination of the pressure differential across the circulator, the flow resistance of the piping, and location of the PONPC gives rise to the new dynamic pressure distribution shown by the dashed green line in figure 2-18.

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 helps keep dissolved air in solution and minimizes the potential for cavitation at the circulator inlet. The short segment of piping between the expansion tank connection point and the inlet port of the circulator experiences a very slight drop in pressure due to flow resistance in the piping. The numbers used for pressure in figure 2-18 are only examples. The actual numbers will depend on flow rates, fluid properties and pipe sizes.

LOADS SUPPLIED THROUGH HEAT EXCHANGERS

There are situations where a heated liquid within a hydronic system needs to pass heat to another liquid without physically contacting that other liquid. This task is routinely handled using a heat exchanger.

Common examples of where heat exchangers are used in hydronic systems include:

- Separation of domestic water from system water
- Separation of antifreeze in solar collector circuits from water in thermal storage tanks
- Separation of antifreeze from system water in snow melting systems
- Separation of swimming pool or space water from system water

The most commonly used type of heat exchanger in modern hydronic systems is called a brazed plate heat exchanger. It consists of a "stack" of specially formed stainless steel plates that have been brazed together at their perimeter. The resulting component allows one fluid to flow through all the even-numbered channels between these plates, while the other fluid flows through the odd-numbered channels. A brazed plated heat exchanger provides a large internal surface area for good heat exchange, but contains that area within a relatively small component. Figure 2-19 shows an example of a brazed plate stainless steel heat exchanger.

Figure 2-19

Courtesy of Kelvion
(GEA Flat Plate)

There is a wide variety of brazed plate heat exchangers available in North America. Most companies offering them provide sizing and selection services, including software that allows several design variables such as flow rates and entering temperatures to be simultaneously evaluated.

General application considerations for heat exchangers include:

1. All heat exchangers require a temperature difference between the entering fluid carrying heat, and the exiting fluid to which the heat is transferred. This difference, known as the "approach temperature difference," is determined by the design of the heat exchangers and the temperature, specific heat, and flow rates of the two fluid streams exchanging heat. One can think of the approach temperature difference as the *thermal penalty* associated with having a heat exchanger in the path of heat flow. The smaller this temperature difference is, the lower the thermal penalty. Where energy efficiency is the primary consideration, heat exchangers should be selected for a maximum approach temperature difference of 5ºF. Even small values are possible using larger heat exchangers.

2. Be sure the material used in the heat exchanger is compatible with both liquids that will be passing through it. This is especially true for highly treated water, such as that used in swimming pools and spas.

3. Be sure the head loss (or pressure drop) of the heat exchanger is factored into the overall circuit head loss when selecting a circulator. Manufacturers generally provide head loss or pressure drop information as a function of flow rate.

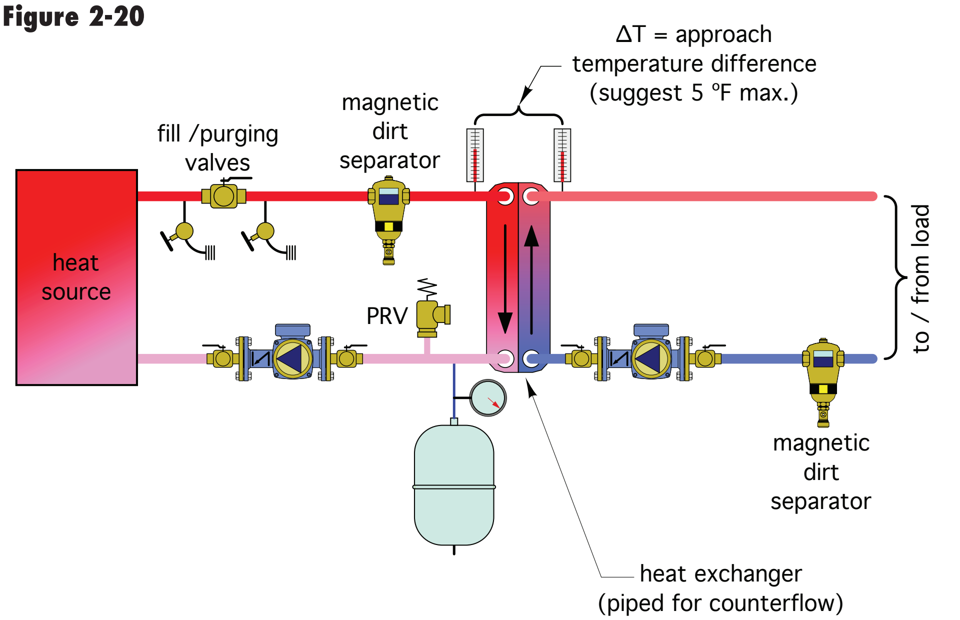
4. Be sure to pipe the heat exchanger for *counterflow*. This means that the two fluids exchanging heat should pass through the heat exchanger in *opposite directions*. This increases the average temperature difference between the two sides of the heat exchangers, and maximizes the rate of heat transfer, all other conditions being equal.

5. Recognize that heat exchangers completely isolate two piping circuits. If both circuits connected to the heat exchanger are closed loops, each should be equipped with an expansion tank, pressure relief valve, air separator and fill/purging valves.

6. Any dirt or debris that lodges inside a heat exchanger will reduce its thermal and hydraulic performance. To prevent this, a high-quality dirt-separating device should be present in each fluid stream flowing *into* the heat exchanger. If one fluid stream is domestic water, be sure the dirt-separating device is lead-free. It is also good practice to install isolation/flushing valves on each side of a heat exchanger flow path carrying domestic water. These valves allow that side of the heat exchanger to be isolated, and provide a simple means of circulating a chemical cleaning solution through the heat exchanger to remove scaling.

Figure 2-20 shows a heat exchanger piped for counterflow, along with suggested "trim," including a magnetic dirt separator on each fluid stream entering the heat exchanger.

Figure 2-20



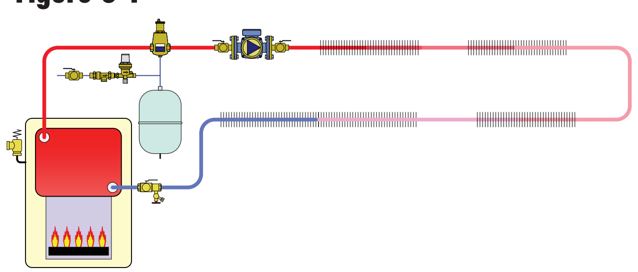
3. SINGLE-CIRCULATOR DISTRIBUTION SYSTEMS

There are several hydronic distribution systems that can be operated by a single circulator. They range from simple series loop systems, to multi-zone systems using valves to control flow through each zone. This section presents the proper piping layout for these systems, and describes their individual strengths and limitations.

SINGLE SERIES LOOP DISTRIBUTION SYSTEMS

The simplest hydronic distribution system is a single series loop. A single loop of piping begins at the heat source, passes through several heat emitters, and ends back at the heat source. System operation is usually controlled by a single room thermostat in the heated space. An example of such a system with multiple fin-tube heat emitters is shown in figure 3-1.

Figure 3-1



Single series circuits are appropriate for small buildings in which all rooms experience similar changes in heating load, and hence can be controlled as a single zone. Single series loop distribution systems can be designed around one type of heat emitter, or use a combination of heat emitters. When different types of heat emitters are used, they should have comparable thermal mass, and thus comparable response times. They should also have comparable water temperature requirements.

Because heat input to the entire building is regulated based on the temperature at one thermostat location, overheating or under-heating of rooms other than where the thermostat is located is possible. It is therefore critical that all heat emitters be sized for the load of their respective rooms, *as well as the water temperature at their location within the piping circuit.* The latter condition is often ignored in favor of sizing heat emitters based on the estimated *average* water temperature within the circuit.

When sized based on the average water temperature in the circuit, heat emitters near the beginning of the series circuit tend to be oversized, and thus release heat at a rate higher than necessary. Heat emitters near the end of the circuit tend to be undersized, and therefore release heat at a rate lower than required.

To *prevent this, designers should keep track of the fluid temperature within the circuit as it cools while passing from one heat emitter to the next.* There are various calculations that can be used for this. They range from relatively simple estimates to detailed mathematical models.

A relatively simple approach uses formula 3-1 to calculate the temperature drop across each heat emitter.

Formula 3-1

$$\Delta T = \frac{Q}{500 \times f}$$

Where:
ΔT = temperature drop across the heat emitter (°F)
Q = rate of heat output from the heat emitter (Btu/hr)
f = flow rate in the circuit (gpm)
500 = a constant based on properties of water. Change the 500 to 485 if a 30% glycol solution is used in the circuit, or to 450 if a 50% glycol solution is used.

Once the ΔT is determined for a given heat emitter, it is subtracted from the inlet temperature of that emitter to get the outlet temperature from that emitter. This outlet temperature becomes the *inlet* temperature for the next heat emitter in the series circuit.

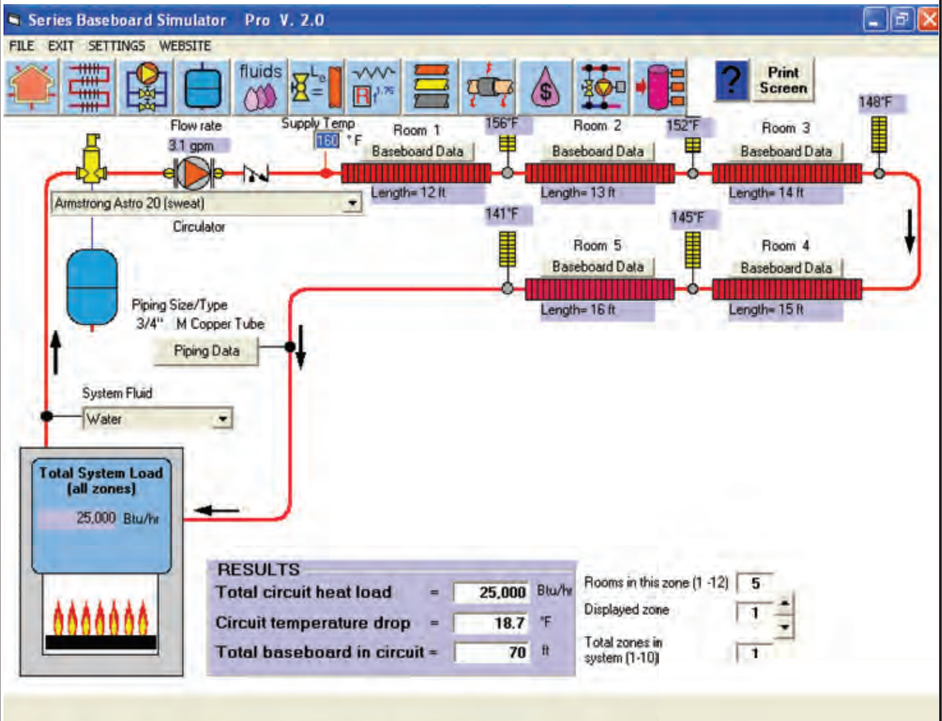
Use of formula 3-1 requires a value for the circuit flow rate. This can be determined through standard hydraulic analysis of the circuit, and the pump curve of the circulator used in the circuit.

***idronics* #16 describes methods and data for estimating flow rates in hydronic circuits.**

In addition to circuit flow rate, designers must access data that gives the heat output of the heat emitters based on water temperature and flow rate. This data is typically published by heat emitter manufacturers. Figure 2-12 can be used to estimate the heat output of standard residential fin-tube baseboard at various average water temperatures.

There is also software that can simulate the combined hydraulic and thermal characteristics of single series circuits that use fin-tube baseboard heat emitters. One example is the Hydronics Design Studio. Figure 3-2 shows an example of this software.

Figure 3-2



This software analyzes a user-specified single serial loop containing up to 12 fin-tube baseboard heat emitters. The sizes of each baseboard, as well as the fluid temperature at locations within the circuit are reported.

The limiting factors in designing single series loop systems are temperature drop and flow resistance. In North America, series circuits have *traditionally* been designed around a temperature drop of approximately 20°F under design load conditions. However, *there is nothing special about a 20ºF temperature drop*. Systems can be designed to operate properly with circuit temperature drops greater than 20ºF, as well as less than 20ºF. In Europe, hydronic distribution systems are routinely operated with temperature drops of 30°F to 40°F.

The advantage of designing around higher temperature drops is that the flow rate can be relatively low. This allows use of small-diameter piping and a smaller, less energy-consuming circulator. The disadvantage of a high temperature drop system is that heat emitters will need to be larger, especially those near the end of the series circuit. This is necessary to meet the load requirement with the lower temperature water. When the circuit is supplied from a conventional boiler, the water temperature returning from the last heat emitter needs to be high enough to prevent sustained flue gas condensation.

Systems that operate with low temperatures drops usually have relatively high flow rates. This may require increased pipe sizes to keep the flow velocity under a suggested limit of 4 feet per second. Systems designed around low temperature drop may also require larger circulators. However, because the heat emitters will operate at higher *average* temperatures, their size can usually be slightly reduced compared to sizes required in systems operating at higher temperature drops.

It is impossible to say what the *optimum* temperature drop is for a single series circuit. The answer depends on the cost of piping, circulators, heat emitters, electricity and several other factors. Designers should investigate different possibilities based on various combinations of supply water temperature and circuit flow rate.

Single series loops that contain one or more heat emitters with high flow resistance must be carefully designed to avoid problems. A single heat emitter, or other component, that has high flow resistance, can *restrict flow through the entire series circuit*, limiting its total heat output. A high head circulator with a larger power requirement may be required. The installation and life cycle operating cost of such a circulator should be considered against other modifications that could eliminate the flow restriction in the system. *In most cases, a distribution system that uses a parallel arrangement of heat emitters will prove more appropriate for use with heat emitters having high flow resistance characteristics.*

The use of high head circulators in single series loop systems can also cause excessive flow velocity through the small tubes or valves in the restrictive heat emitter(s). This can create noise and erosion inside some components.

Another drawback of a single series loop is that heat output control is limited to the control features present on the heat emitters. In the case of fin-tube baseboard, heat output can be reduced up to about 50% by closing the dampers on the enclosure. When fan-coils are used, the blower speed can be reduced to slightly limit heat output. Both these methods require manual adjustments to the heat emitters in response to changing load conditions. Most occupants either do not know these adjustments can be made or quickly tire of making them.

DIVERTER TEE DISTRIBUTION SYSTEMS

There are ways to retain the general concept of a single loop system while also providing a means of individual flow control through each heat emitter. One approach is to create a *diverter tee* distribution system.

This piping arrangement involves the use of special fittings called diverter tees. From outward appearance, these fittings appear as standard tees. However, they have internal detailing that is specially designed to divert a portion of the water flowing in the main piping circuit through a branch circuit that includes at least one heat emitter. Figure 3-3 shows the outward appearance and internal cross section of a diverter tee.

Figure 3-4

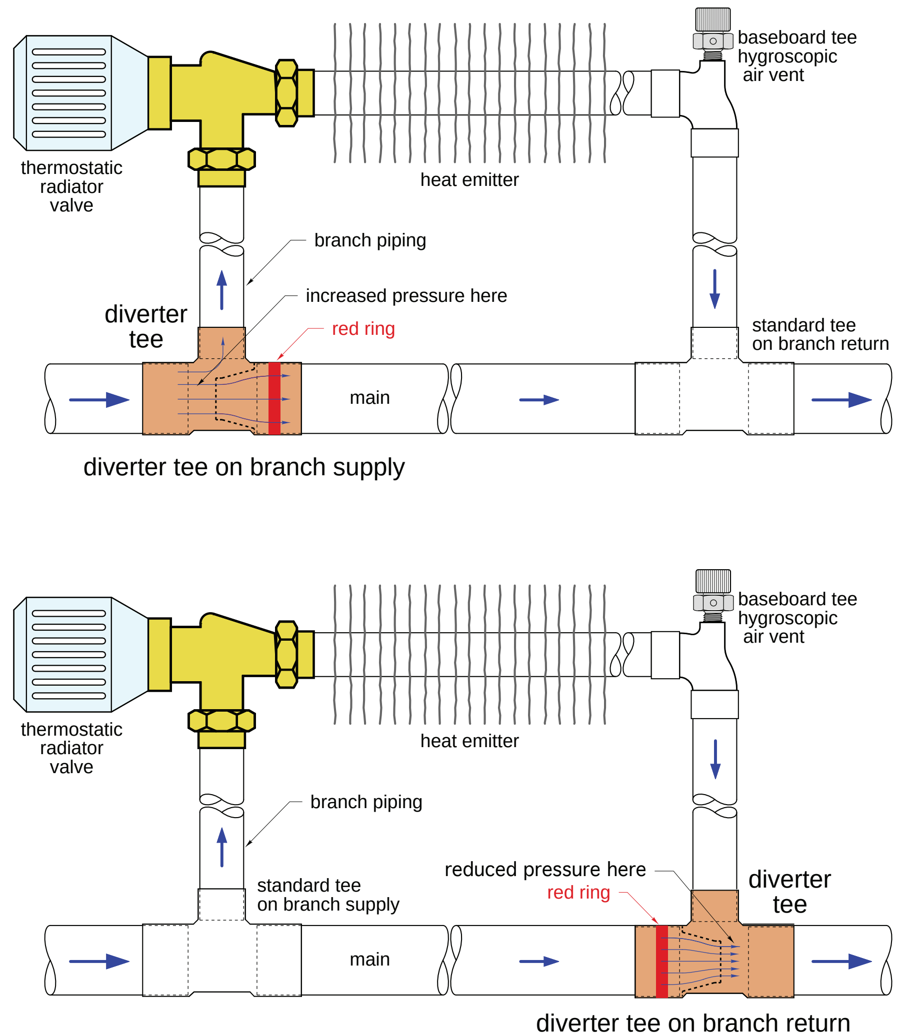


Figure 3-3

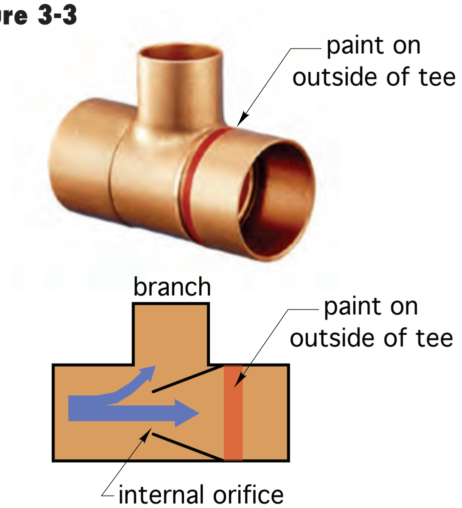


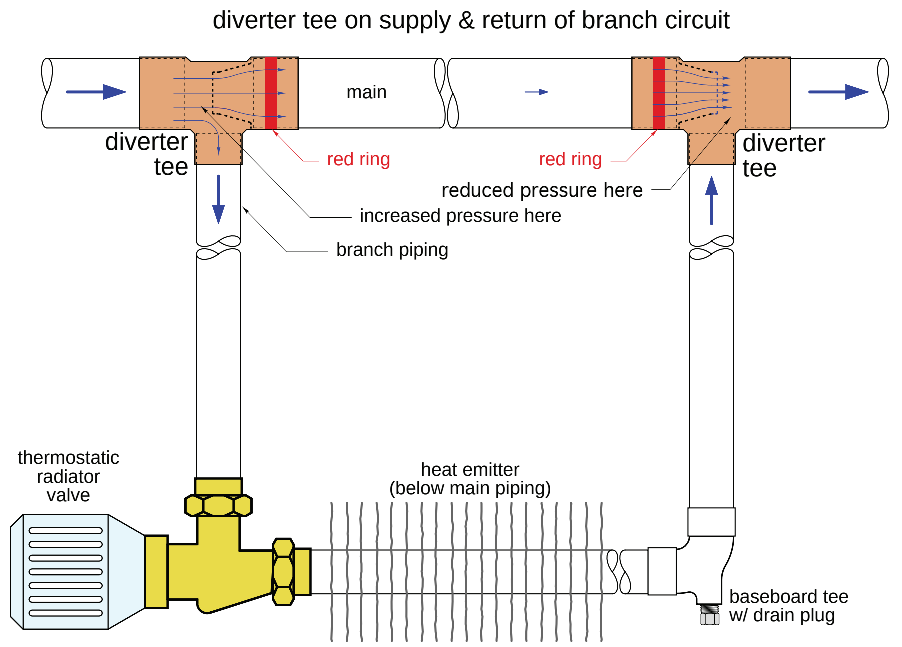
Photo courtesy of Bell & Gossett

The truncated cone inside a diverter tee creates a pressure differential along the length (e.g., "run") of the tee. This pressure differential is what induces flow through the branch circuit.

Diverter tees often have a "red ring" painted on the outside of the fitting, as shown in figure 3-3. This ring indicates the side of the fitting that must face toward the piping between this tee and the other tee that connects the branch back to the main piping circuit, as shown in figure 3-4.

Diverter tees can be installed in either the "upstream" location, as shown by the upper image in figure 3-4, or in the "downstream" location, as shown by the lower image in figure 3-4. The pressure differential created in either location, under the same operating conditions, is approximately the same.

Figure 3-5



A single diverter tee is usually sufficient to create flow through low resistance heat emitters such as a few feet of fin-tube baseboard or a panel radiator, *assuming the heat emitter is above the piping loop containing the diverter tee.*

Two diverter tees may be needed on heat emitters with high flow resistance. When two diverter tees are used, they create a greater pressure differential across the branch. This creates a higher flow rate through that branch. Two diverter tees are also suggested to overcome buoyancy forces when the heat emitter is located several feet below the distribution circuit, as shown in figure 3-5.

When two diverter tees are used, there should be at least one foot of pipe between the tees to allow turbulence created by the upstream tee to dissipate before the flow enters the downstream tee.

Figure 3-6 shows how multiple branches can be supplied from a single loop system powered by a single circulator.

Each branch includes a thermostatic radiator valve that can modulate flow through that branch based on the set room temperature. Flow through a given branch can be completely stopped if necessary. There will be slight changes in a given branch flow rate as other branches open, close or modulate flow rate.

Any heat emitter on what would otherwise be a single series circuit can be selected to have independent temperature *limiting* control by piping it into the circuit using one or two diverter tee(s) and some type of flow-regulating valve.

Figure 3-6

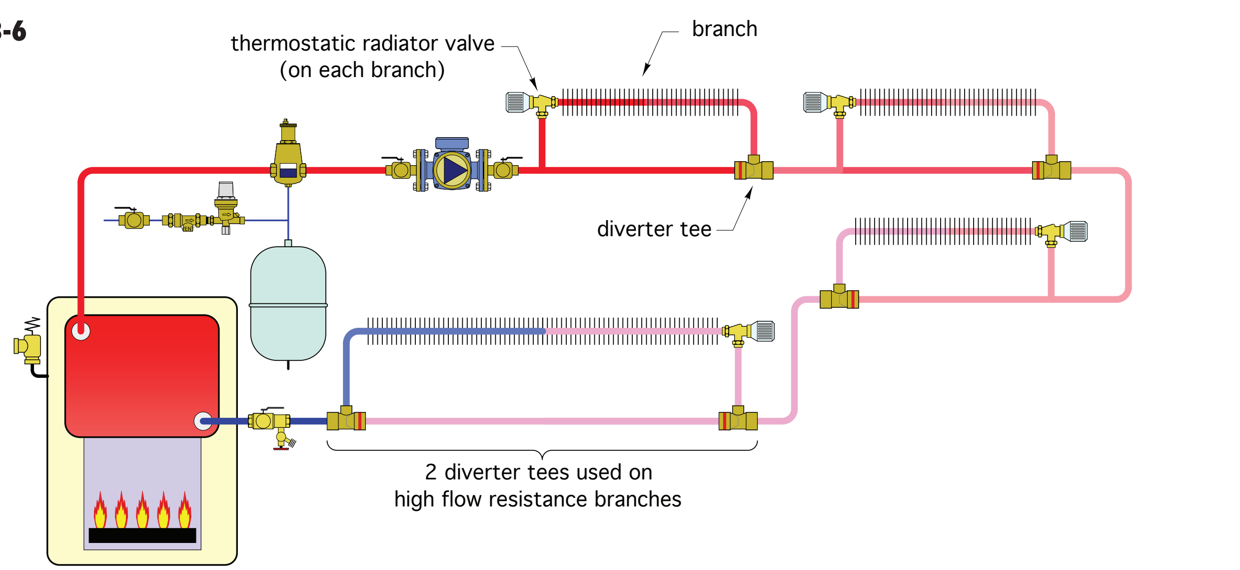


Figure 3-7

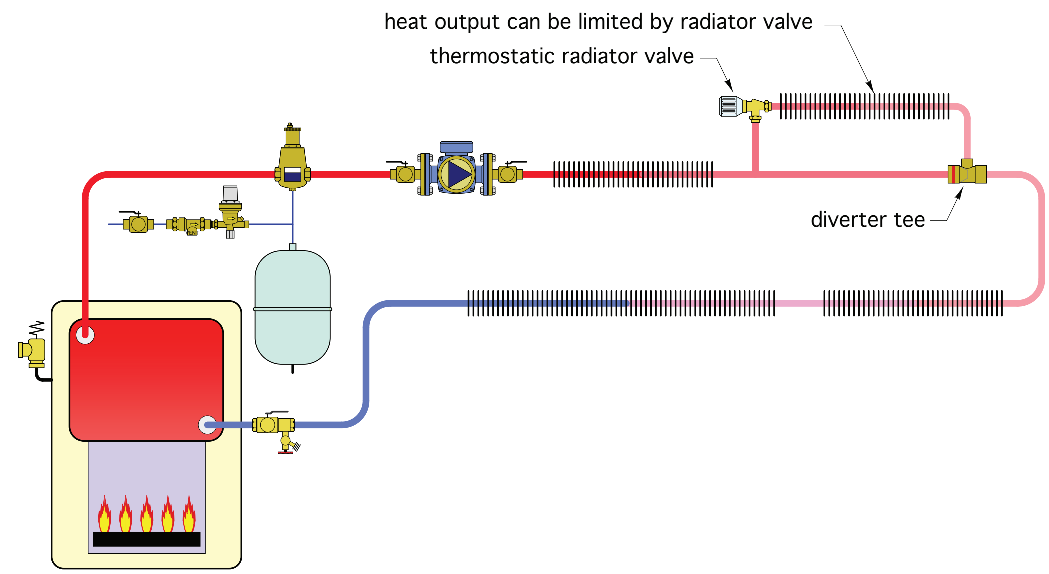


Figure 3-7 shows a series circuit where one heat emitter is connected using a diverter tee arrangement, while the other heat emitters remain in series. This is a convenient way to limit the heat output of one heat emitter without significantly affecting the remainder of the heat emitters. This arrangement could be used, for example, in a guest bedroom that only needs to be heated to normal comfort temperatures a few days each year. Keep in mind, however, that the branch connected to the system using the diverter tee *cannot independently call for heat*. There must be flow in the main circuit in order to have flow through the branch. If independent on/off control of several heat emitters is needed, it is better to use a parallel distribution system.

When non-electric thermostatic radiator valves are used for individual room temperature limiting, the designer must provide controls to operate the heat source and distribution circulator whenever the building might need heat. One approach is to equip the system with an outdoor reset control that automatically turns on the heat source and distribution circulator when the outdoor temperature falls below a preset value. When such a control system is used, the thermostatic valves on the heat emitters serve as temperature-limiting devices for their respective rooms. Since this approach requires constant circulation in the main distribution circuit, all piping should be insulated to minimize heat loss.

Whenever diverter tees and thermostatic radiator valves are used to reduce or totally stop water flow to a heat emitter, it is crucial that the heat emitter and the piping leading to it are protected against freezing during cold weather. Some thermostatic radiator valves have a freeze-proof minimum setting that only allows a trickle of heated water flow through the branch to provide this protection.

SERIES BYPASS DISTRIBUTION SYSTEMS

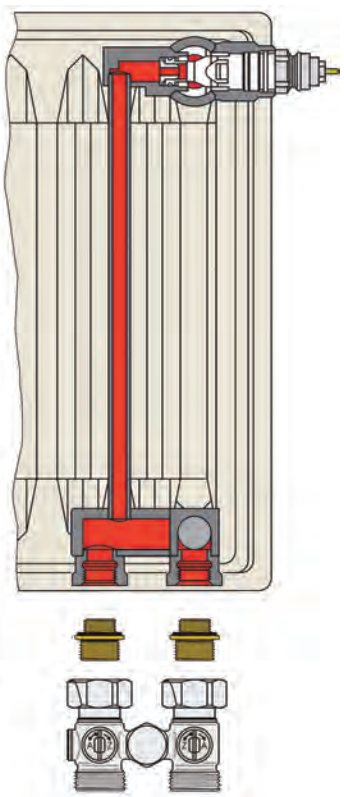
There is another way to construct what appears to be a series piping circuit, but retain the ability to regulate flow through each heat emitter. It is called a series bypass system, and is commonly used with panel radiators that are equipped with thermostatic radiator valves.

A series bypass system uses a special “H-pattern” bypass valve at each panel radiator. An example of such a valve is shown in figure 3-8.

Figure 3-8

This valve is designed to connect to the bottom of a panel radiator that has standard supply and return connections spaced 50 mm apart, as shown in figure 3-9.

Figure 3-9



The H-pattern valve is actually an assembly of three separate valves. Two of those valves are located in the two vertical sections of the valve. They are ball valves that are designed to be fully open when the panel radiator is in service, or fully closed if the panel radiator needs to be completely isolated from the remainder of the system. The latter might be necessary if the panel radiator has to be removed to redo the wall finish behind it, or if the panel radiator developed a leak and had to be replaced. The slot in the front-facing portion of each ball valve is parallel with the fluid passage through the internal ball. This slot can be rotated using a slotted screwdriver. When the panel radiator is in service, both slots should be vertical. When the panel is to be isolated, both slots should be horizontal.

The third valve is located in the connector between the two vertical segments of the assembly. It is a flow-regulating valve. Its setting determines the percentage of the flow entering the left vertical portion of the valve that is diverted through the panel radiator to which the H-valve is attached, *assuming the valve at the top of the radiator is fully open.*

The Caleffi H-pattern diverter valves are supplied with this flow-regulating valve pre-adjusted so that 35% of the entering flow will be diverted upward through the panel radiator, while the remaining 65% of entering flow passes through the bypass. This setting is fully adjustable. These percentages are appropriate when three panel radiators of approximately the same size are connected in the arrangement shown in figure 3-10.

The H-pattern diverter valve allows up to three panel radiators to be configured for individual flow control. However, as is true with a diverter tee system, as well as a single series circuit, there is a drop in water temperature as it passes each active radiator. Designers should calculate this temperature drop based on the flow in the circuit, and the rate of heat release from each panel radiator. Formula 3-1 can be used for this calculation.

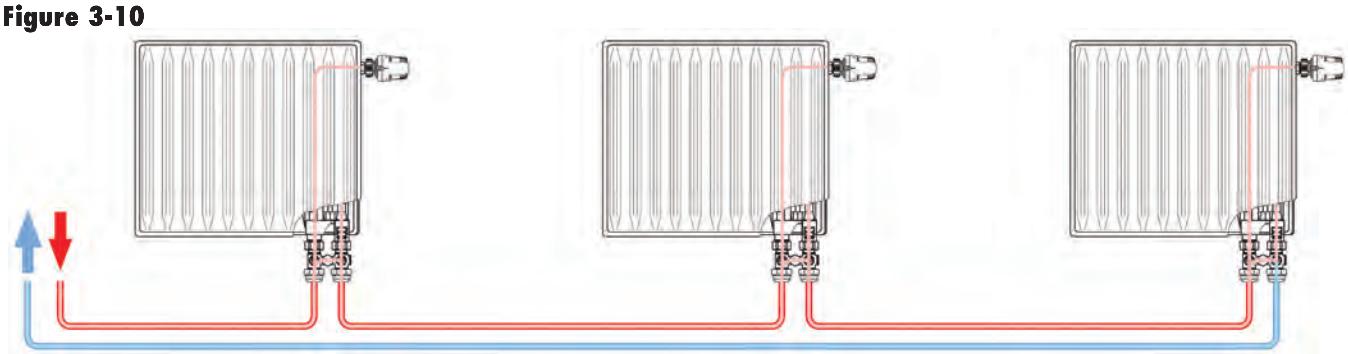
The circuit should be designed assuming that all radiators are active simultaneously. Each panel radiator should be sized based on its entering water temperature, which decreases along the circuit as the flow passes each panel.

PARALLEL DISTRIBUTION SYSTEMS

All of the distribution systems discussed thus far have heat emitters connected in a series or "quasi-series" arrangement.

The disadvantage of these arrangements is that the water temperature supplied to each heat emitter will be lower than that supplied to the upstream heat emitter — assuming all heat emitters are operating. Each time a heat emitter in a diverter tee or series bypass system goes from inactive to active, the water temperature to the downstream heat emitters changes.

Figure 3-10



This is "tolerable" when accompanied by accurate design calculations, but it is not ideal.

Parallel distribution systems do not experience this temperature drop from one heat emitter to the next. In most cases, it's safe to assume that the water temperature supplied to each heat emitter in a parallel distribution system is the same. The exception being where the piping lengths to one heat emitter are significantly longer, or subject to significantly higher heat loss, compared to the piping supplying the other heat emitters.

There are several types of parallel distribution systems that can be supplied by a single circulator. They will be discussed in this section. There are also parallel distribution systems that use multiple circulators, which will be discussed in the next section.

SYSTEMS WITH ZONE VALVES

One of the most common parallel distribution systems uses a separate circuit to heat (or cool) each zone of a building. Flow through each zone circuit is allowed or prevented by an electrically operated zone valve. A typical piping layout is shown in figure 3-11.

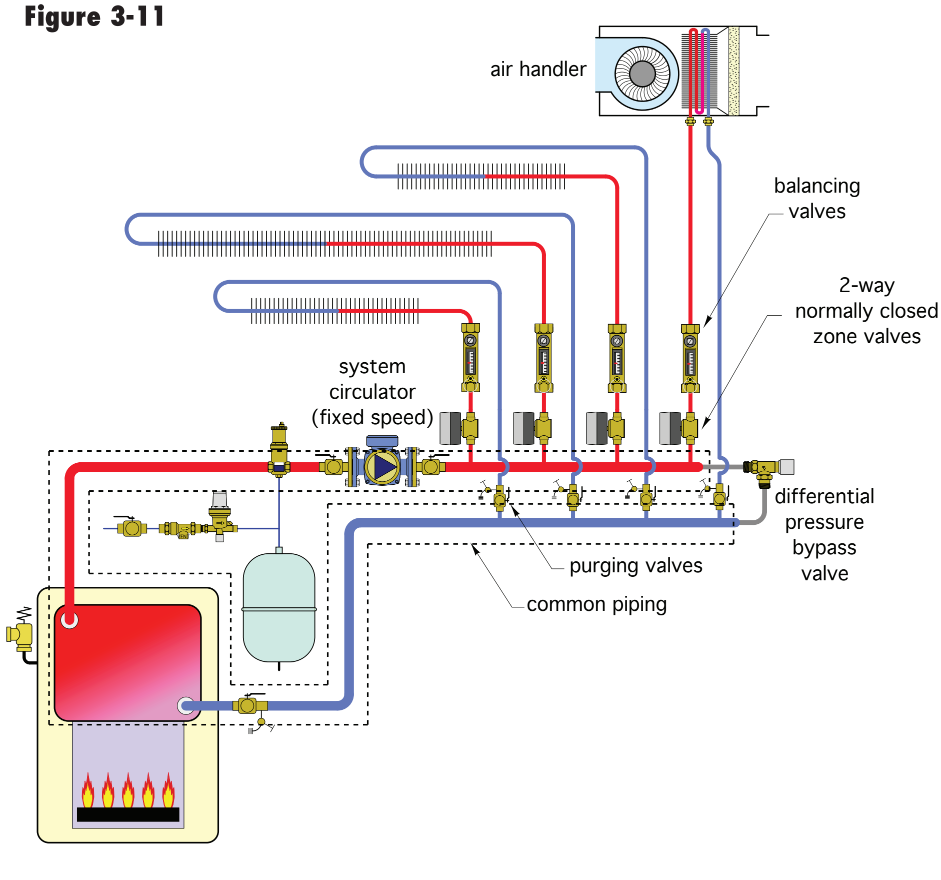
This system uses a single *fixed-speed* circulator to create flow in any zone circuit that has an open zone valve. Since all zones are supplied from a common header, they have the same supply water temperature. The heat emitters used in each zone circuit can be different, but they should be sized based on the supply water temperature produced by the heat source at design load conditions.

This system also includes flow-balancing valves on each zone circuit. This allows flow rates within each parallel circuit to be adjusted in proportion to the percentage of the total load supplied by that circuit. The flow-balancing valves can also be completely closed to isolate the zone valves below if servicing is needed.

The zone valves are located on the supply side of each zone circuit. This placement is preferred because it prevents heat migration into the zone circuit when the zone valve is closed.

Each parallel circuit also includes a purging valve where it connects to the return header. This placement allows bulk air to be quickly purged from each circuit when the system is commissioned. It also allows each circuit to be fully isolated if servicing is needed.

Figure 3-11

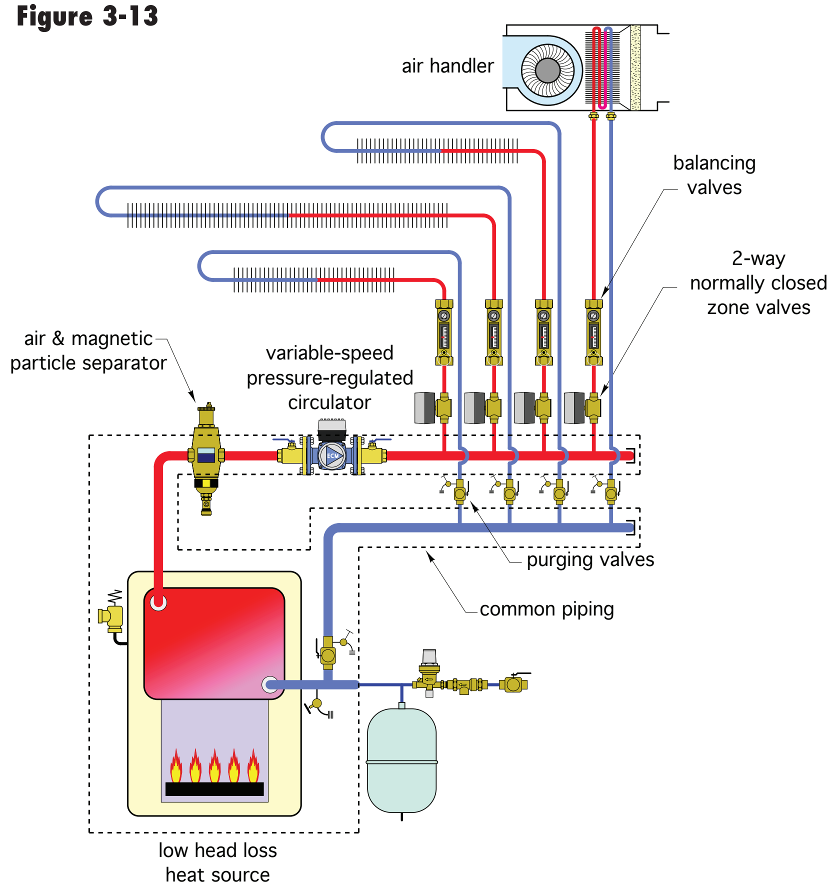


The piping within the dashed lines in figure 3-11 is called the "common piping." It is the piping through which all system flow passes. *To provide optimal performance, the common piping should be selected to provide a very low head loss.* Doing so will minimize change in flow rate through a given zone circuit when another zone valve turns on or off.

A differential pressure bypass valve is connected between the right ends of the supply and return headers. Its function is to prevent excessively high differential pressure from developing between these headers when only one or two of the zones are on. It should be set to a pressure that is 0.5 to 1 psi above the differential pressure across the headers when all zone valves are open. A differential pressure bypass valve should always be used when valve-based zoning is used in combination with a *fixed-speed* circulator.

***idronics* #5 provides a more complete description of how to size and select differential pressure bypass valves.**

Figure 3-13



In systems where a fixed-speed circulator is used in combination with zone valves, the circulator used should have a relatively “flat” pump curve. Figure 3-12 illustrates the difference between such a pump curve compared to a “steep” pump curve.

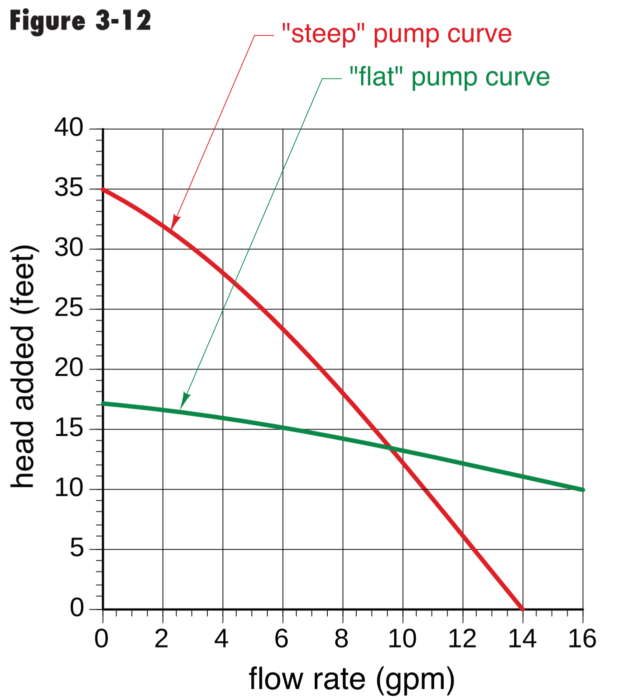
Fixed-speed circulators with flat pump curves reduce the change in differential pressure experienced by operating zone circuits when another zone circuit turns on or off. This, in turn, reduces changes in flow rate, and thus helps maintain consistent heat output from all active zone circuits.

Systems using zone valves are also well-suited to high-efficiency variable-speed pressure-regulated circulators. These circulators can be set to maintain approximately *constant differential pressure* between the supply and return manifolds, regardless of how many zones are active at any given time. Figure 3-13 shows an example.

Notice that the differential pressure bypass valve shown in figure 3-11 is no longer present. It is not needed when a pressure-regulated circulator is used. If a variable-speed pressure-regulated circulator is retrofitted to a system that has a differential pressure valve, that valve should be completely closed or removed.

Systems of this configuration, and using variable-speed pressure-regulated circulators, should also have low flow resistance common piping. The variable-speed circulator should be set for *constant differential pressure* mode. The differential pressure “setpoint” should be adjusted to that required when all zone valves are open (e.g., under design load conditions). Once set, this circulator will automatically adjust speed in an attempt to maintain the differential pressure between the supply and return headers approximately constant. When a zone valve closes, the circulator will reduce speed, which also

Figure 3-12



reduces electrical power input. When a zone valve opens, the circulator will increase speed.

It is also advisable to include a magnetic dirt separator in any system using an ECM-type variable-speed circulator. These circulators have powerful permanent magnets inside their rotors. Iron oxide particles are attracted to these magnets. Use of a magnetic dirt separator will reduce the presence of such particles in the system, and thus reduce any potential for them to lodge within the circulator.

idronics #5 provides a more complete description of how variable-speed pressure-regulated circulators can be used in hydronic systems.

SYSTEMS WITH MANIFOLDS

It is also possible to create a parallel distribution system using manifolds, as shown in figure 3-14.

Figure 3-14

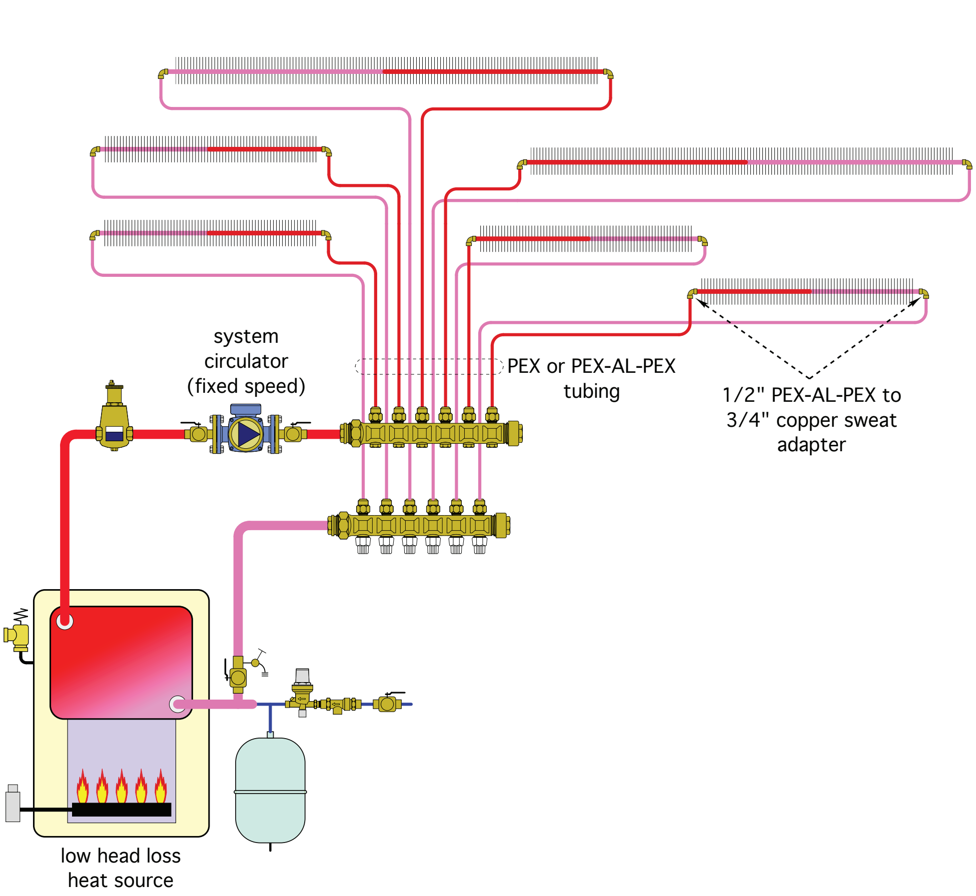


Figure 3-15

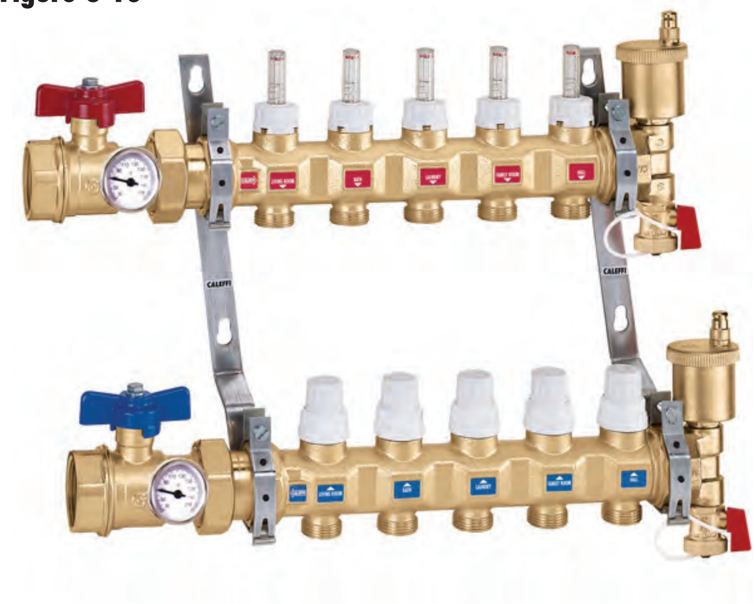
The manifold replaces the need for site-built headers. It consolidates multiple zone connections into a common assembly. This type of distribution system is often referred to as a “homerun” system. The name stems from each piping path making a complete round trip from the supply manifold, to the heat emitter, and back to the return manifold.

Manifold distribution systems are most commonly used in combination with flexible PEX or PEX-AL-PEX tubing. The small flexible tubing is easy to route within common frame construction in both new and retrofit applications, as shown in figure 3-15. It eliminates the need for any pipe joints other than at the manifold and the heat emitter.

Special fittings are available to adapt between common flexible tubing, such as 1/2” PEX and either 1/2” or 3/4” copper tubing, such as that used in fin-tube baseboards.

As is true in systems using zone valves, the heat emitters used in each zone circuit can be different, but they should all be selected and sized based on the supply water

Figure 3-16



temperature produced by the heat source under design load conditions.

Manifolds are available with integrated flow-balancing valves and flow meters. These replace the need for external balance valves such as those shown in figure 3-13. Figure 3-16 shows an example of a 5-circuit manifold station that includes flow-balancing valves and flow meters for each circuit. This manifold station also provides main isolation valves, air vents, drain valves and thermometers.

The benefits of a manifold distribution system are further enhanced when individual flow control is added to each zone circuit. This can be done in two ways:

1. By using a thermostatic radiator valve at each heat emitter.
2. By adding manifold valve actuators to each circuit.

Figure 3-17

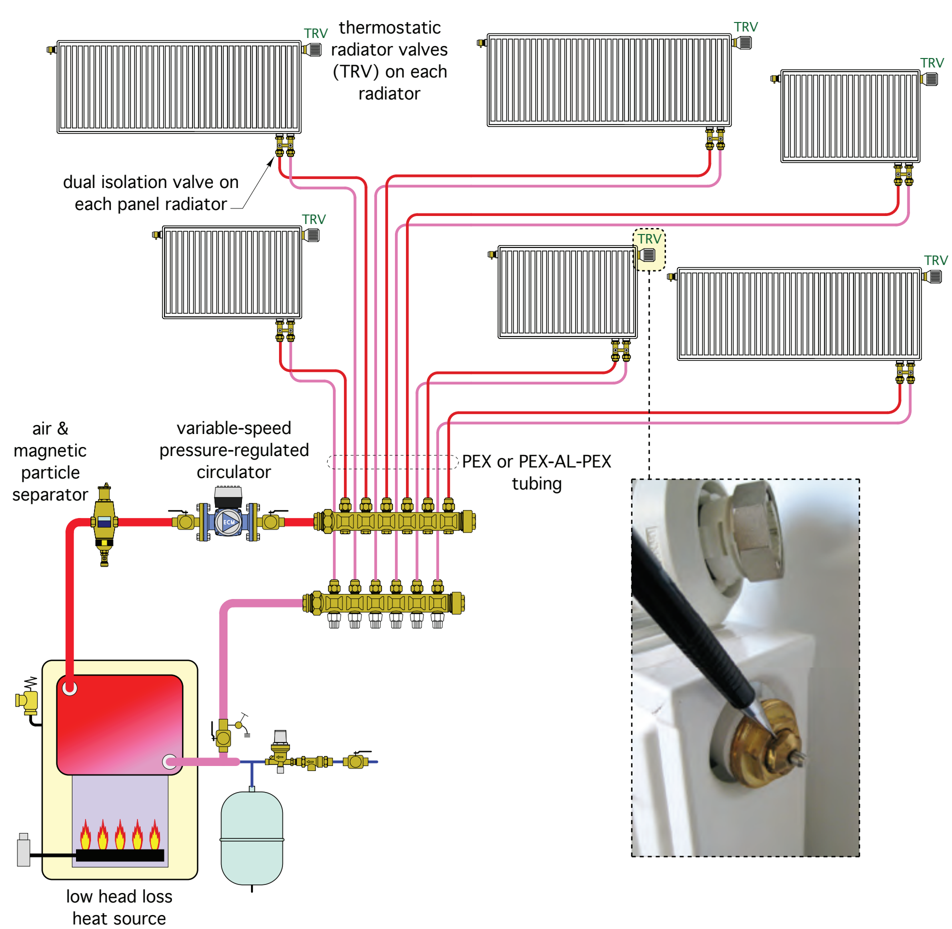


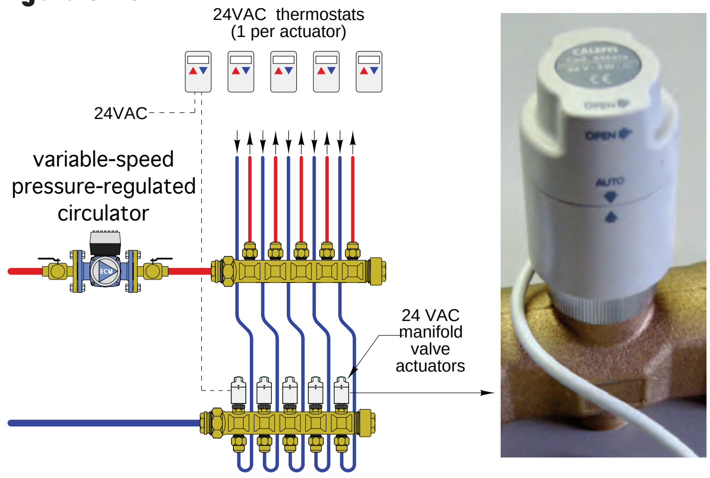
Figure 3-17 shows how multiple panel radiators, each equipped with a thermostatic radiator valve, can be supplied by a manifold distribution system.

The flow rate through each panel radiator is regulated by the thermostatic radiator valve, which responds to changes in room air temperature. Thus, each panel radiator creates a separate zone within the building.

Some radiators, such as those depicted in figure 3-17, are supplied with the body of the radiator valve integrated into the radiator. The installer just screws on a thermostatic actuator to allow automatic actuation of the valve's stem.

A variable-speed pressure-regulated circulator is ideal for this type of system. It would be set for constant differential pressure mode, and would automatically speed up or slow down as thermostatic valve

Figure 3-18



Some manifold valve actuators are equipped with an isolated end switch. This switch closes when the actuator reaches its fully open position. A low power circuit through the end switch is used to signal that the circulator and heat source need to operate.

***idronics* #5 provides more details on how to wire manifold valve actuators with other electrical components in the system.**

2-PIPE DIRECT RETURN DISTRIBUTION SYSTEMS

Another common parallel piping method, one this is especially common in commercial buildings, is called a 2-pipe direct return distribution system. Figure 3-19 shows an example.

actuators allowed more flow or reduced flow. The circulator could be turned on using a master thermostat within the building, or whenever the outdoor temperature dropped below some threshold at which heating was deemed necessary.

The manifolds may be valveless, or they may be equipped with isolation valves. The latter has the advantage that each circuit can be isolated at the manifold station if service is necessary.

If zoning control will be accomplished using electric thermostats, the manifold station can be equipped with manifold valve actuators, as shown in figure 3-18.

When a manifold valve actuator is screwed onto the threaded valve body above a manifold piping connection, it compresses the spring-loaded valve shaft to its closed position. This prevents any flow through that circuit.

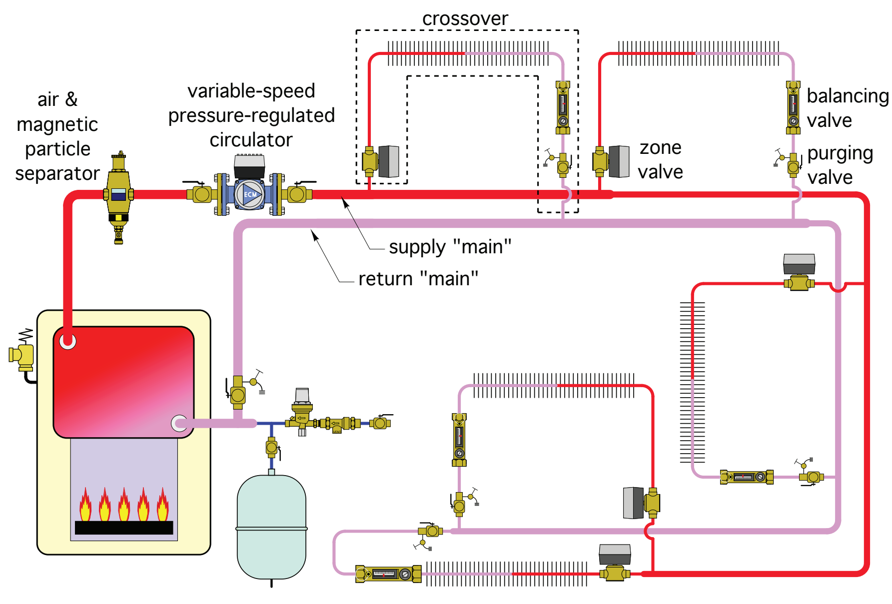
When 24 VAC electrical power is switched to the valve actuator, it retracts its stem over a period of 2 to 3 minutes, allowing the spring-loaded valve in the manifold to open.

This is another good application for a variable-speed pressure-regulated circulator set for constant differential pressure mode.

Each heat emitter is located within a “crossover” connected to the supply and return “mains.” If the supply main is properly insulated, heat loss will be minimal, and each heat emitter will be supplied with water at approximately the same temperature.

Notice that the pipe size of the mains decreases as one moves from the heat source toward the farther end of the distribution system. This is possible because the flow in the mains decreases as the mains pass outward beyond each crossover.

Figure 3-19



"2-pipe" direct return distribution system

The heat emitter closest to the circulator on the supply main is also closest to the circulator on the return main. The next heat emitter connected to the supply main has a greater length of piping between it and the circulator. The heat emitter farthest away from the circulator has the longest overall piping path length.

If each crossover in the system has the same flow resistance, the highest flow rate will be through the shortest piping path. If uncorrected, this situation reduces the flow rate through heat emitters located farther away from the circulator.

This situation can be remedied through use of properly set balancing valves on each crossover. These valves can be adjusted so that the flow rate through each crossover is in proportion to the rate of heat release from the heat emitters in that crossover as a percentage of the total rate of heat release from the system.

idronics #8 provides details on how to balance a 2-pipe direct return distribution system.

Flow through each crossover can be allowed or prevented by installing zone valves, as shown in figure 3-19. The flow rate through each crossover can also be modulated when a modulating valve is used in lieu of a zone valve. Either configuration allows each heat emitter to be independently controlled, and thus room-by-room comfort control is possible when the system is properly configured.

As with other parallel distribution systems, different types of heat emitters can be used on the crossovers provided they are all sized for the same supply water temperature.

The system shown in figure 3-19 uses a variable-speed pressure-regulated circulator. This is the same type of variable-speed circulator described in previous parallel distribution systems. However, when used in a 2-pipe direct return system, this circulator should be set for *proportional differential pressure* mode.

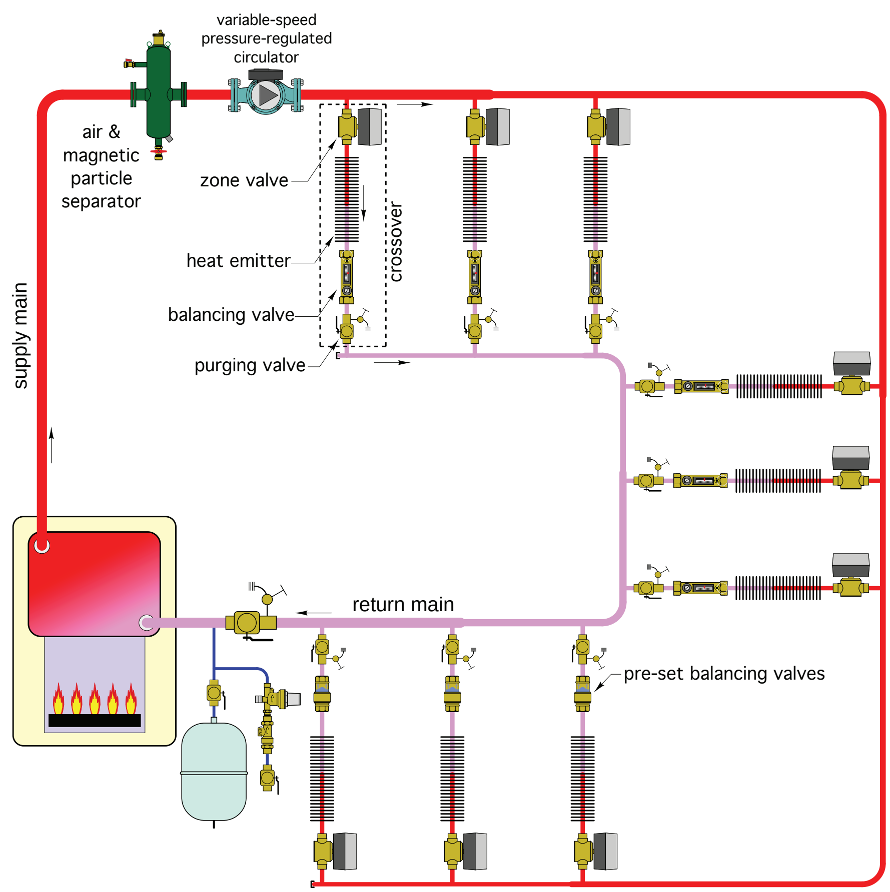
In this mode, the differential pressure created by the circulator decreases in a pre-determined manner as the flow rate decreases.

Proportional differential pressure control accounts for the head loss in the supply and return mains. In a 2-pipe direct return system, this head loss is a significantly higher percentage of total head loss compared to the head loss in the common piping of a manifold-type distribution system. Proportional differential pressure control approximates the ideal scenario of maintaining constant differential pressure across each crossover, regardless of which crossovers are active or inactive. Most circulators that offer constant differential pressure control can also be set for proportional differential pressure control.

2-PIPE REVERSE RETURN SYSTEMS

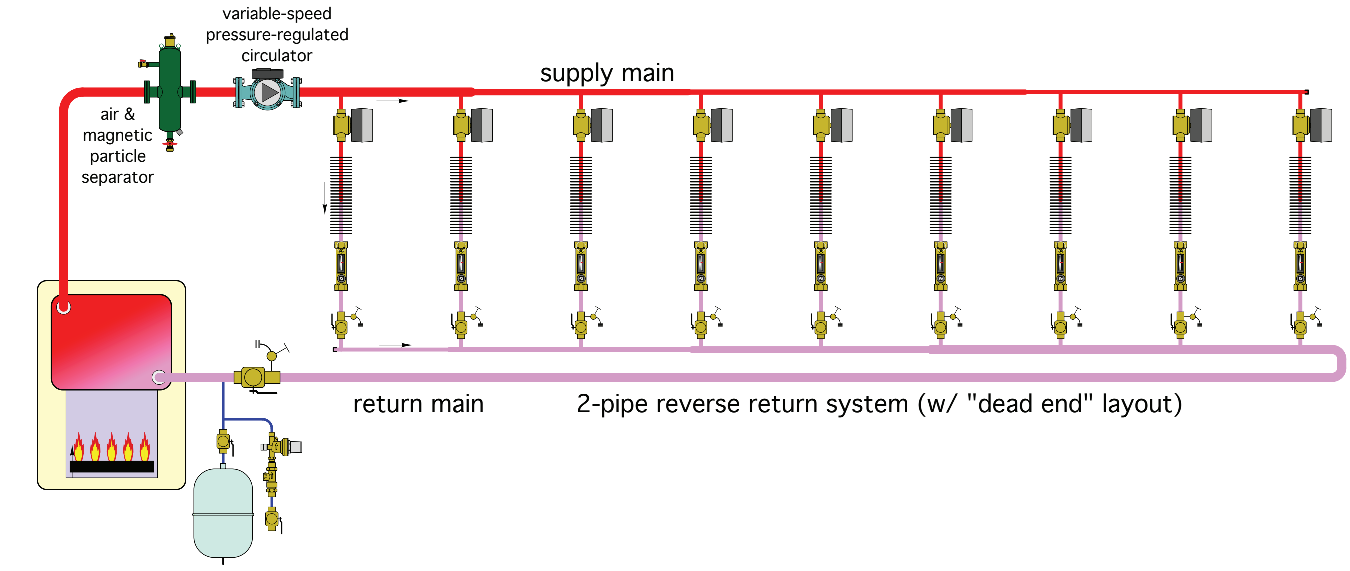
Another variation of the "2-pipe" approach is known as a 2-pipe *reverse* return system. An example is shown in figure 3-20.

Figure 3-20



2-pipe reverse return distribution system

Figure 3-21



Like a 2-pipe direct return system, this system connects multiple crossovers, each with a heat emitter and valve trim, between common supply and return mains. Thus, each crossover is supplied with water at approximately the same temperature.

Notice that the crossover closest to the circulator along the supply main is now farthest from the circulator on the return main. This is what distinguishes a 2-pipe *reverse* return system from a 2-pipe *direct* return system.

Reverse return piping attempts to make the flow resistance of each flow path *approximately* equal, beginning at the circulator discharge, passing out along the supply main, through a crossover, back along the return main, and finally back to the circulator inlet. Emphasis is on the word *approximately*.

An "ideal" reverse return system, in which each crossover had the same piping, heat emitter, and valve trim, and with the ability to maintain the same head loss per unit length along the mains piping, could yield exactly the same flow resistance along each flow path. Such a system would be "self-balancing." There would be no need of balancing valves in each crossover.

Unfortunately, this is unrealistic from a number of standpoints. First, not all systems will have identical crossovers. Some heat emitters in the system may be different from others in both heating output and flow resistance. The amount of piping within a given crossover may also be different from that in other crossovers. Second, there are only a finite number of pipe sizes that can be used for the mains piping. Although changing pipe size based on the flow rate at different locations along the mains is possible, it is impossible (or highly impractical) to create *exactly* the same head loss per unit length of piping.

Still, even though any real reverse return system can only approximate ideal conditions, it will be closer to "self-balancing" compared to a direct return system. The amount of balancing required is usually less, and this implies that less head energy will be throttled away in balancing valves. Less throttled head energy suggests that a smaller, lower power circulator could likely be used.

Reverse return systems are well-suited to applications where the supply and return mains run side by side and make a loop from the mechanical room around the building and back to the mechanical room. They are not well suited to "dead end" layouts such as shown in figure 3-21.

"Dead end" layout requires a long length of the largest size piping to return all flow from the location of the farthest crossover. While this will work, it can add significant cost to the system relative to reverse return systems that "loop" around the building, as depicted in figure 3-20.

A variable-speed pressure-regulated circulator, set for proportional differential pressure control, is also appropriate for 2-pipe reverse return systems.

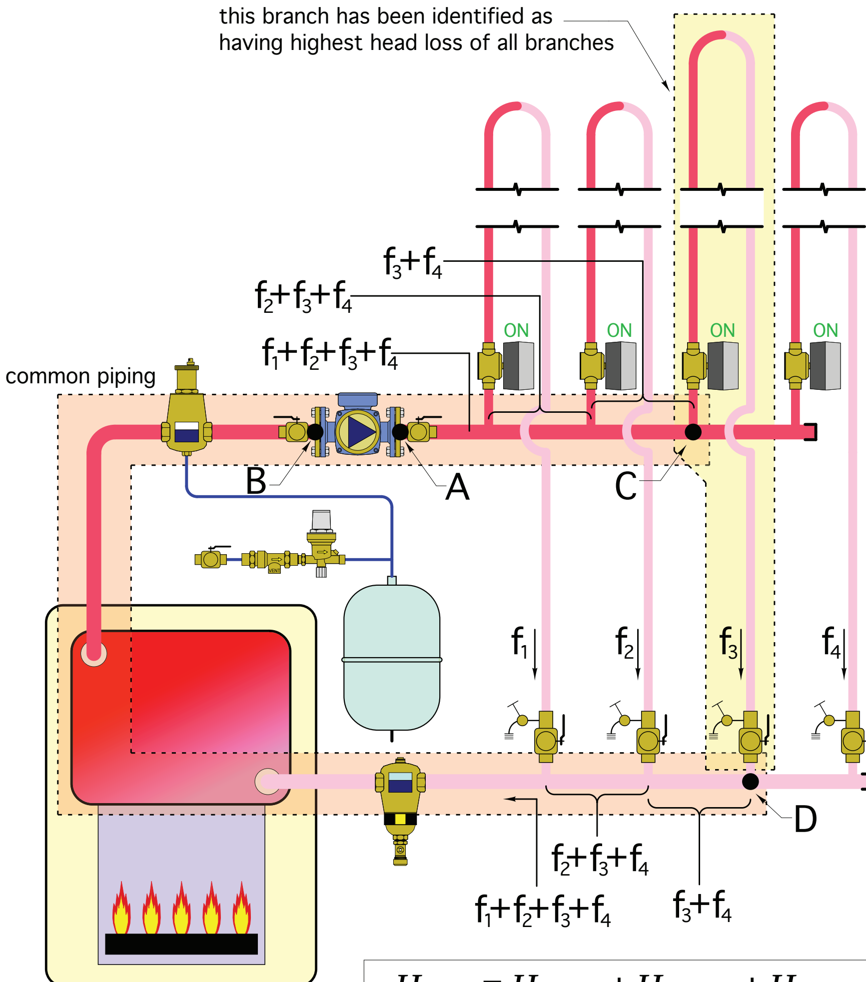
FLOW RATES AND CIRCULATOR SELECTION FOR PARALLEL DISTRIBUTION SYSTEMS

The target flow rates within each branch of a parallel distribution system should be proportional to the design load heat output of that branch as a percentage of the total design heat output of the distribution system. Thus, if the heat emitter in a given branch is sized to release 15% of the total heat output of the system, that branch should operate at approximately 15% of the total system flow rate at design load conditions. This criterion keeps the temperature drop across each branch approximately equal.

The circulator used in a parallel distribution system is typically sized by identifying the branch that would have the highest head loss, assuming all branches are active. This branch might have to be determined by trial and error, in which several possible branches are identified, and the head loss through each branch is calculated and compared with that of the other branches.

When calculating the head loss of each branch, the design flow rate should be assumed in each piping segment between the circulator discharge and the circulator inlet. Different pipe sizes will likely be involved along with different flow rates. The head loss of each piping segment at its associated flow rate must be calculated. Likewise, the head loss through all common piping components, (e.g., heat source, air separator, etc.) should also be calculated at whatever flow rate would be present with all branches on. Once the head loss of the individual branch is calculated, it can be added to the head loss of the common piping to determine the total head loss of the complete flow path. This is illustrated in figure 3-22.

Figure 3-22

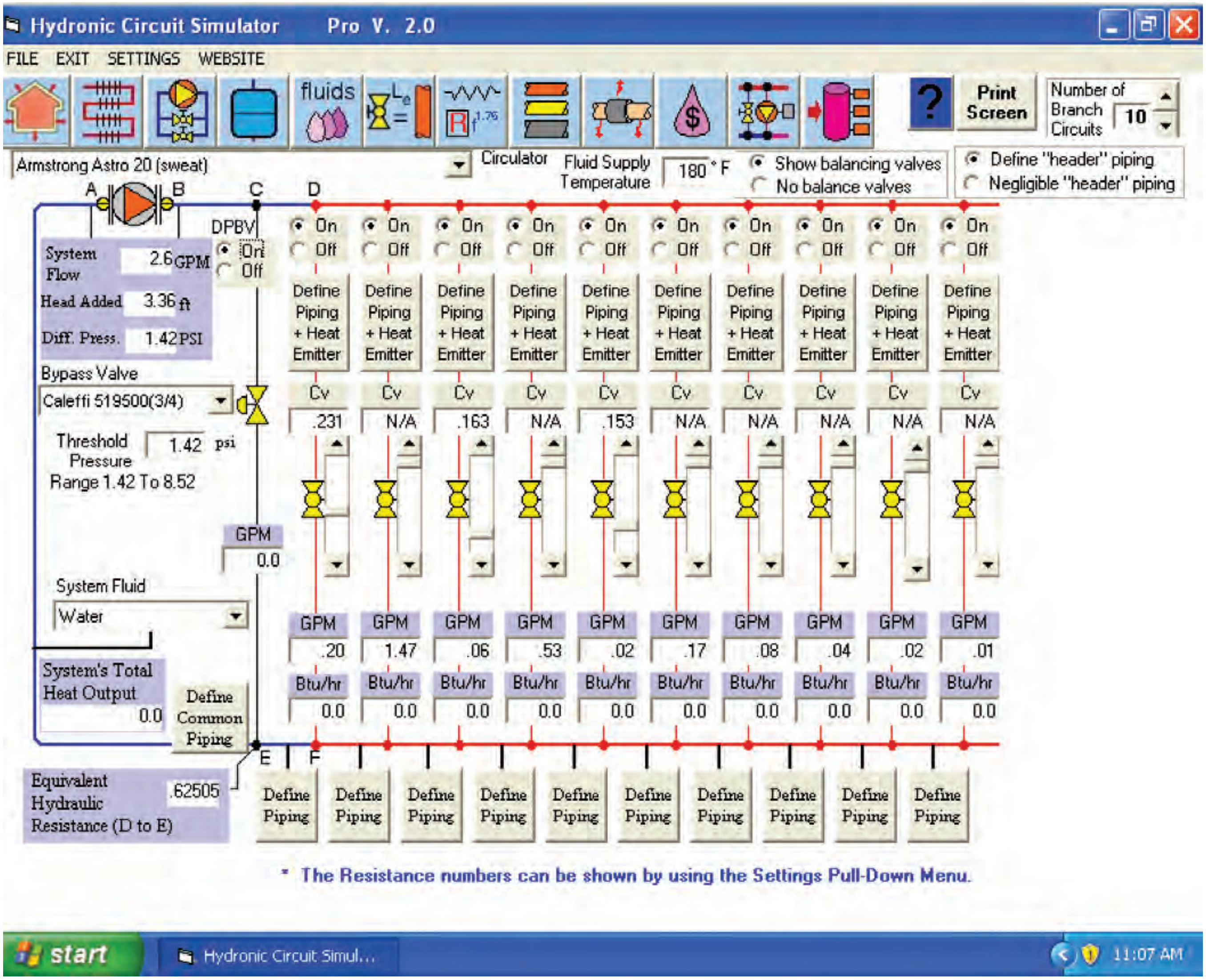


$$H_{A\to B} = H_{C\to D^*} + H_{D\to B^{**}} + H_{A\to C^{**}}$$

* = calculated at brach flow rate
** = calculated at flow rate present in each piping segment, assuming all branches are operating

idronics #12 and #16 discuss methods for calculating head loss in systems using smooth piping such as copper or PEX tubing. When steel piping is used, head loss can be determined using the Darcy-Weisbach formula, which is also discussed in idronics #16.

Figure 3-23



Once the path with the greatest head loss is determined, that head loss, combined with the ***total*** system flow rate (e.g., the sum of the flow rates through all crossovers operating simultaneously at design load conditions), will set the required "duty point" for a circulator. Pump curves from candidate circulators can then be examined to find a suitable model.

Parallel direct return piping systems with up to 12 crossovers can also be simulated using the Hydronics Design Studio software. Figure 3-23 shows an example of a system with 10 user-defined crossovers, including balancing valves, and use of a differential pressure bypass valve.

4. MULTIPLE-CIRCULATOR DISTRIBUTION SYSTEMS

The use of multiple circulators allows design flexibility that is not possible in single circulator systems. This section discusses both “traditional” and novel approaches for systems with multiple circulators. The discussion begins with a design goal that is desirable in *any* system containing two or more independently controlled circulators. That goal is hydraulic separation.

When two or more circulators operate simultaneously in the same system, each attempts to establish differential pressures based on their own pump curves. *Ideally, each circulator will establish a differential pressure and flow rate that is unaffected by the presence of another operating circulator in the system.* When this desirable condition is achieved, the circulators are said to be *hydraulically separated* from each other.

Conversely, the lack of hydraulic separation can create very *undesirable* operating conditions in which circulators interfere with each other. The resulting flows and rates of heat transport within the system can be greatly affected by such interference, often to the detriment of proper heat delivery.

The degree to which two or more operating circulators interact with each other depends on the head loss within the piping path they have in common. This piping path is called the common piping, since it is shared by both circuits. *The lower the head loss of the common piping, the less the circulators will interfere with each other.*

Consider the system shown in figure 4-1. In this system, both circuits share common piping. The “spacious” geometry of this common piping creates very low flow velocity through it. As a result, very little head loss can occur across it.

Figure 4-1

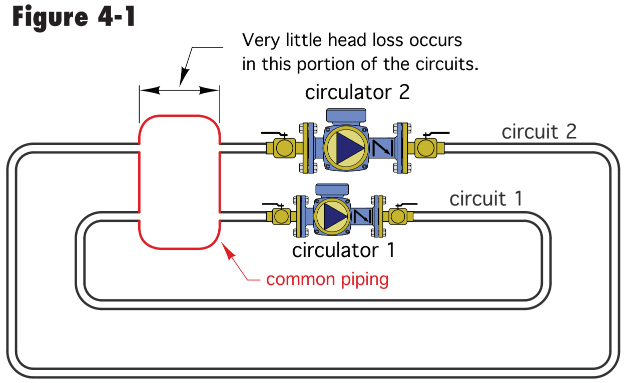


Figure 4-2

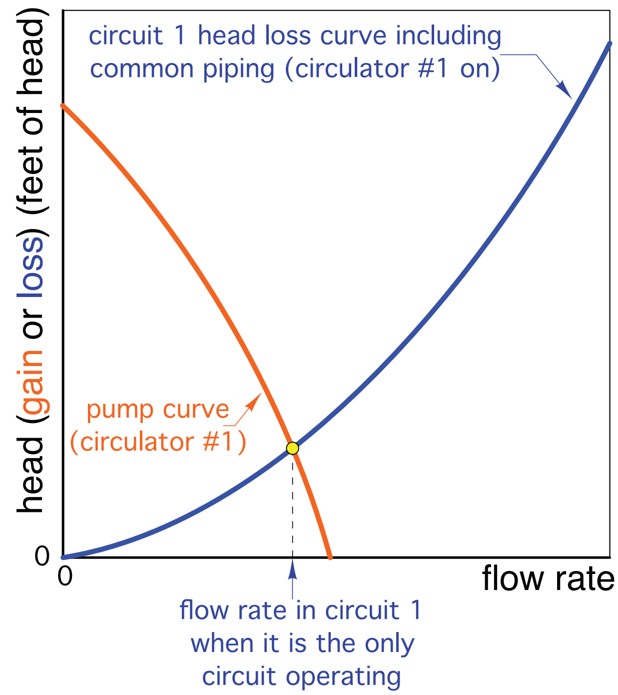
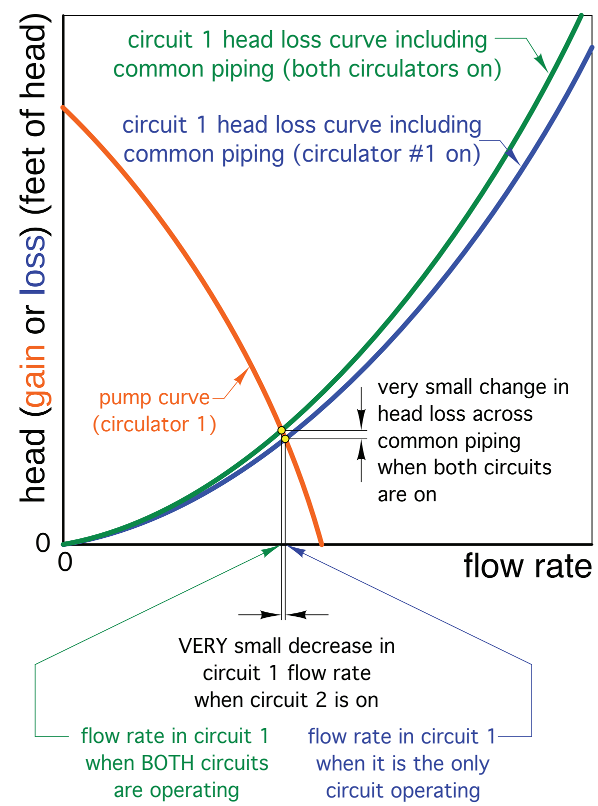


Figure 4-3



Assume that circulator 1 is operating, and circulator 2 is off. The blue circuit head loss curve shown in figure 4-2 applies to this situation. The point where the blue circuit head loss curve crosses the orange pump curve for circulator 1 establishes the flow rate in circuit 1.

Next, assume circulator 2 is turned on, while circulator 1 remains on. The flow rate through the common piping increases, and so does the head loss across it. However, because of its spacious geometry, the increase in head loss across the common piping is very small. The system head loss curve that is now "seen" by circulator 1 will steepen, but very slightly. It is shown as the green curve in figure 4-3.

The operating point of circuit 1 has moved very slightly to the left and slightly upward. This implies that the flow rate through circuit 1 has decreased very slightly. This very small change in flow rate is indicated in figure 4-3. Such a small change in flow rate will have virtually no effect on the ability of circuit 1 to deliver heat. Thus, the interference created when circulator 2 turned on is of no consequence. This arrangement provides acceptable hydraulic separation between the two circulators.

One can think of (and design) circuits that are known to have a high degree of hydraulic separation as if they were *completely independent of each other*, as illustrated in figure 4-4.

The required hydraulic performance of each circuit can be determined as if it were a standalone circuit, unaffected by the other circuits in the system. This is a very powerful concept that simplifies design and troubleshooting.

To summarize: By keeping the head loss of the common piping shared by two or more circuits, each with their own circulator, as low as possible, the system achieves a high degree of hydraulic separation between the circulators. This point will be reinforced and repeated as several multiple circulator systems are discussed.

idronics #15 describes several methods for achieving hydraulic separation in multi-circulator systems.

ZONING WITH CIRCULATORS

One common method of zoning hydronic systems in North America is by using a separate circulator for each zone circuit. Figure 4-5 illustrates the concept.

Figure 4-5

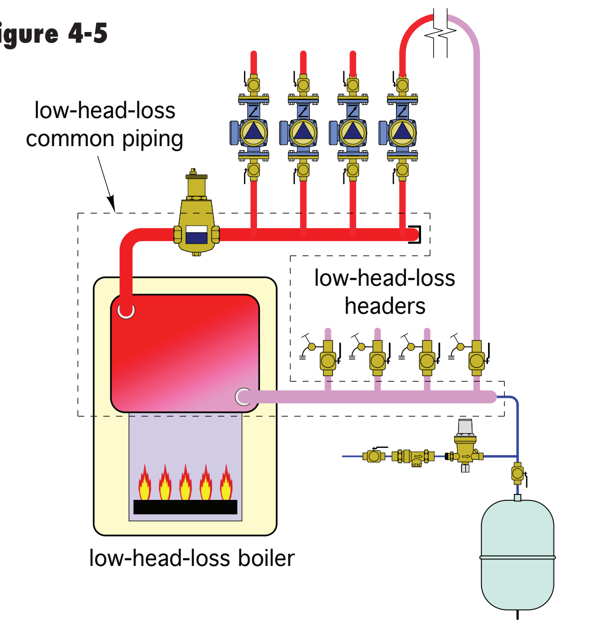
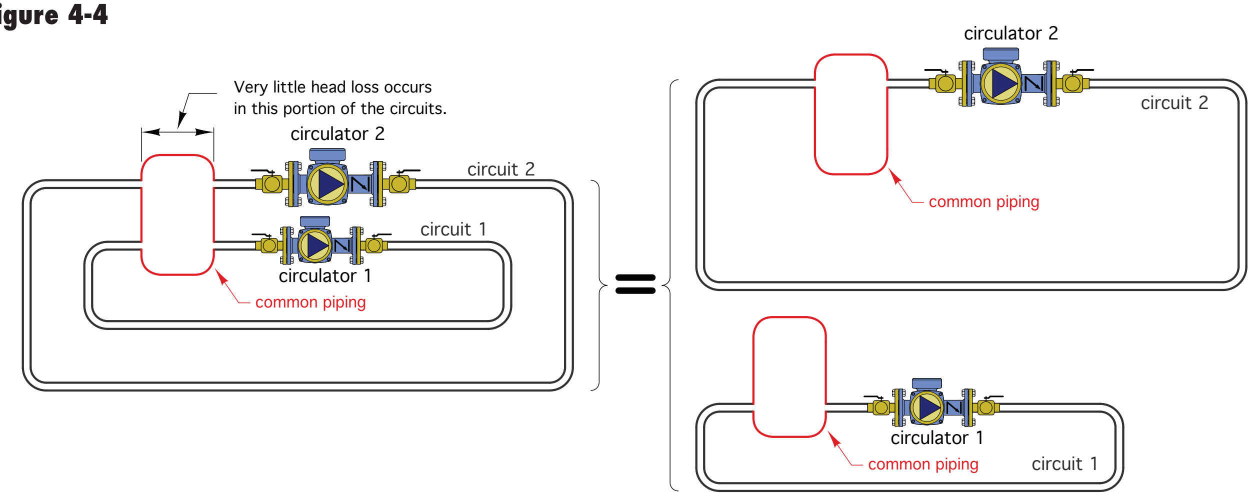


Figure 4-4



Each of the 4 zone circuits contains a circulator, and each of those circulators contains an internal spring-loaded check valve assembly.

When the thermostat in a given zone calls for heat, its associated zone circulator is turned on, and the heat source is enabled to operate. Heated water is routed to the active zone and comes back to the return header after delivering heat to the heat emitters in that zone.

The dashed lines seen in figure 4-5 identify the common piping in the system. The head loss of this common piping should be as low as possible to provide good hydraulic separation of the circulators. The headers should be short and generously sized.

A suggested guideline is to size headers for a flow velocity in the range of 2 to 4 feet per second when all the circulators supplied by the header are operating. Low flow velocity creates minimum head loss.

Figure 4-6 lists the flow rates corresponding to flow velocities of 2 feet per second and 4 feet per second for type M copper tubing in sizes from 1-inch to 4-inch, and in schedule 40 steel for larger pipe sizes.

Figure 4-6

Tube/pipe size	Flow rate at 2 ft/sec	Flow rate at 4 ft/sec
1” M copper	5.5 gpm	10.9 gpm
1.25” M copper	8.2 gpm	16.3 gpm
1.5” M copper	11.4 gpm	22.9 gpm
2” M copper	19.8 gpm	39.6 gpm
2.5” M copper	30.5 gpm	61.1 gpm
3” M copper	43.6 gpm	87.1 gpm
4” M copper	75.9 gpm	152 gpm
5” M copper	118 gpm	236 gpm
6” schd. 40 steel	180 gpm	361 gpm
8” schd. 40 steel	312 gpm	624 gpm
10” schd. 40 steel	492 gpm	984 gpm
12” schd. 40 steel	699 gpm	1397 gpm

The head loss of the boiler shown in figure 4-5 is assumed to be low. This is typical for cast iron sectional boilers and boilers with steel fire tube heat exchangers.

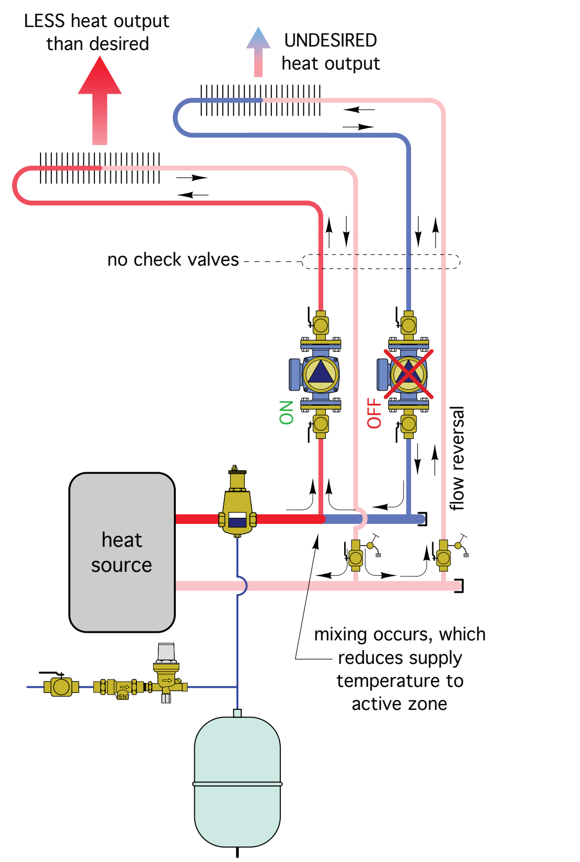
The spring-loaded check valve within each zone circulator does two things:

1. It prevents thermosiphoning of hot water from the boiler into an inactive zone circuit.
2. It prevents flow reversal through inactive zones when other zones are operating.

The spring-loaded check valve used in most circulators has a forward opening pressure requirement of 0.3 and 0.5 psi. This is sufficient to prevent the relatively weak pressure differential created by buoyancy differences between the hot and cool water from establishing an undesirable thermosiphon flow within an inactive zone circuit.

For the internal spring-loaded check valve to be effective, the circulator must be mounted on the supply side of the zone circuit, as shown in figure 4-5. If the circulators used

Figure 4-7

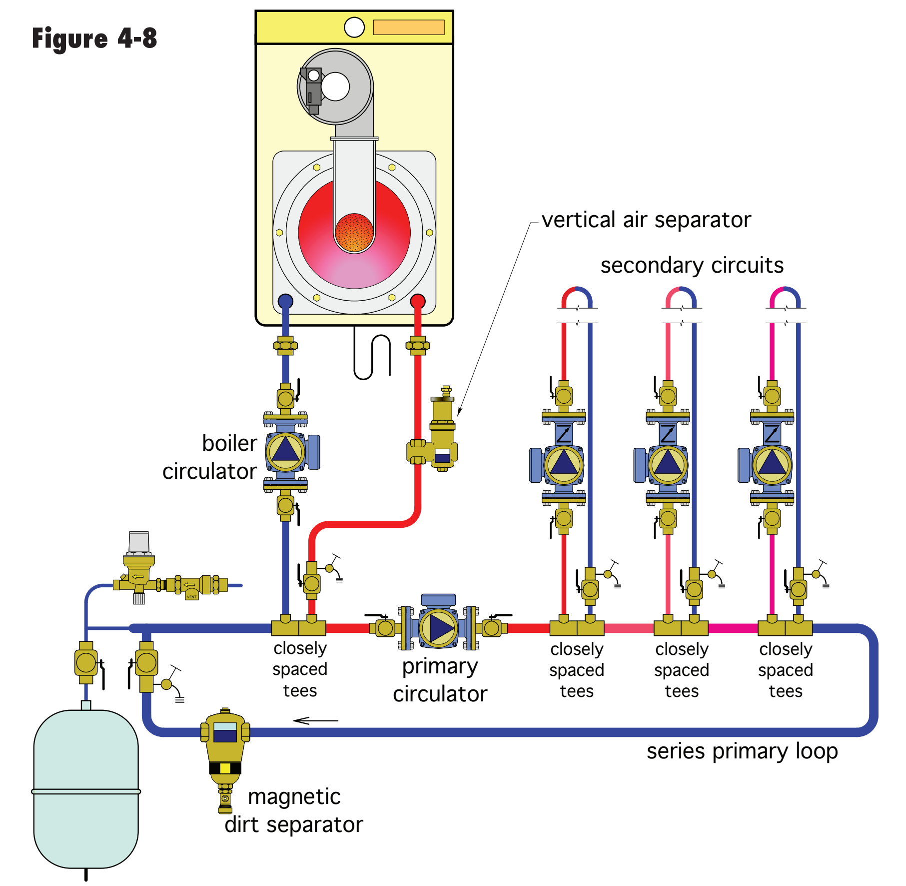


do not have internal check valves, an external spring-loaded check valve with comparable forward open pressure can be used. It should be mounted at least 12 pipe diameters downstream of the zone circulator outlet to minimize any potential for rattling due to turbulent flow exiting the circulator.

The spring-loaded check valve's ability to prevent flow reversal through an inactive zone circuit is equally important. Figure 4-7 illustrates the potential flow reversal path in a system with two zone circuits, neither of which has a check valve, assuming that one zone circulator is operating while the other is not.

Notice that the flow returning to the lower header from the active zone "splits" at the tee below the purging valve. Some of it passes back into the heat source as desired. The rest passes backward through the other (supposedly inactive) zone circuit.

Figure 4-8



Keep in mind that return flow in some systems can be quite hot, perhaps even 160ºF. Water at elevated temperatures flowing backward through an inactive zone circuit will dissipate heat through the heat emitters in that circuit. Such heat output is unnecessary and undesired. Be sure that check valves are in place to prevent such heat migration.

PRIMARY/SECONDARY SYSTEMS

In addition to zoning, many hydronic systems serve multiple heating loads, each of which can independently call for heat. An example of a multi-load system would be one that provides some space heating through fin-tube baseboard, additional space heating through low temperature radiant panels, provides domestic water heating, and also provides snow melting for the entry area of a building

One approach to piping such a system is to use a circulator for each load and connect all the loads to a common piping loop. This is called a *series* primary/secondary system. An example of such a system is shown in figure 4-8.

In this system, each of the load circuits (also referred to as "secondary circuits") connects to the primary loop using a pair of closely spaced tees. Because these tees are as close as possible, there is very little pressure drop between them. Thus, there is very little tendency for flow to be induced within any *inactive* secondary circuit, even though there is flow in the primary loop.

Each pair of closely spaced tees provides hydraulic separation between the primary loop and its associated secondary circuit.

When a given secondary circuit needs to deliver heat, its circulator is turned on. The secondary circuit draws heated water from the primary loop, sends it through the secondary circuit, and returns the cooler flow downstream of the heat emitters back to the primary loop.

The high flow resistance mod/con boiler shown in figure 4-8 is also connected to the primary loop using a pair of closely spaced tees. Those tees provide hydraulic separation between the boiler's circulator and the primary loop circulator.

Although hydraulic separation exists between all circulators, so does an undesirable effect; *a drop in supply water temperature from one secondary circuit to the next whenever two or more secondary circuits are operating simultaneously.* Although there are situations in which this temperature drop doesn't present a problem, it does add complications that designers must compensate for after assessing.

The temperature drop across each set of closely spaced tees can be calculated using formula 4-1 (repeated below).

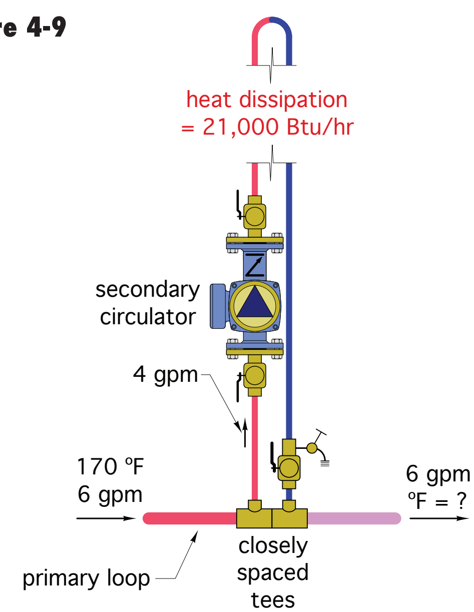
Formula 4-1

$$\Delta T = \frac{Q}{500 \times f}$$

Where:
ΔT = temperature drop across the closely spaced tees of an active secondary circuit (ºF)
Q = rate of heat output from the secondary circuit (Btu/hr)
f = flow rate in the *primary loop* (gpm)
500 = a constant based on properties of water. Change the 500 to 485 if a 30% glycol solution is used in the circuit, or to 450 if a 50% glycol solution is used.

Example: Figure 4-9 shows a situation with water at 170ºF and 6 gpm flowing along a primary loop. A secondary circuit is operating at a flow rate of 4 gpm, and dissipating heat at a rate of 21,000 Btu/hr. The fluid in the system is water. Determine the temperature drop across the closely spaced tees.

Figure 4-9



Solution: Putting the stated operating conditions into formula 3-1 and solving yields:

$$\Delta T = \frac{Q}{500 \times f} = \frac{21,000}{500 \times 6} = 7^o F \qquad T_{out} = (T_{in} - \Delta T) = (170 - 7) = 163^o F$$

Discussion: Notice that the flow rate in the *primary* loop (and not in the secondary circuit) was used in this calculation. The temperature drop depends only on the primary loop flow rate and the rate of heat dissipation by the heat emitters in the secondary circuit. The water temperature passing to the next set of closely spaced tees in the primary loop will be approximately 163ºF, assuming that the primary loop piping is well insulated. This calculation can be repeated for each operating secondary circuit in the system to determine the total temperature drop around the primary loop. Assuming that the flow rate in the primary loop doesn't change, the total temperature drop around the primary loop can vary considerably depending on which secondary circuits are operating at any time.

The sequential temperature drop across each secondary circuit connection is a consequence of the series arrangement of the closely space tees. It is not a desirable condition.

PARALLEL PRIMARY LOOPS

One way to overcome the sequential temperature drop effect associated with *series* primary loops is to create a *parallel* primary loop, as shown in figure 4-10.

A parallel primary loop is divided into two or more "crossover bridges." A pair of closely spaced tees within each crossover bridge provides hydraulic separation between each secondary circuit and the parallel primary loop.

Unlike a system with a series primary loop, a system with a parallel primary loop provides the same supply water temperature to each secondary circuit, regardless of which secondary circuits are operating. *However, this benefit is achieved through more complicated and costly piping.* Notice that each crossover bridge contains a flow-balancing valve. These valves are needed to set the flow through each crossover bridge in proportion to the thermal load served by the secondary circuit supplied from that bridge. If these valves are not present and properly adjusted, there may be problems such as inadequate flows through the crossover bridges located farther away from the primary circulator.

Another important consideration is that both series and parallel primary/secondary systems require a primary circulator. This circulator has to operate whenever any one or more of the secondary circuits is active.

The primary loop circulator adds to the installed cost of the system. More importantly, it adds to the system's operating cost over its entire life. Even one small primary loop circulator can have operating costs that total $1,000 or more over a typical 20-year system design life. Larger primary loop circulators can have life cycle operating costs of several thousand dollars.

Example: Consider a primary loop circulator that must produce a flow rate of 50 gpm, with a corresponding head of 15 feet (which is evidenced by a pressure gain of 6.35 psi across the circulator). Assume the circulator is a typical wet-rotor design and has a wire-to-water efficiency of 25% at these operating conditions. The estimated input power to operate this circulator is:

$$W = \frac{0.4344 \times f \times \Delta P}{0.25} = \frac{0.4344 \times 50 \times 6.35}{0.25} = 552 watts$$

If this primary loop circulator were to operate for 3000 hours each year, and the local cost of electrical energy was \$0.10/kwhr, the annual operating cost would be:

$$\text{1st year cost} = \left(\frac{3000hr}{yr}\right)\left(\frac{552w}{1}\right)\left(\frac{1kwhr}{1000whr}\right)\left(\frac{\$0.10}{kwhr}\right) = \$165.60$$

Furthermore, if the cost of electricity were to inflate at 4% each year, the total operating cost of this circulator over a 20-year period would be:

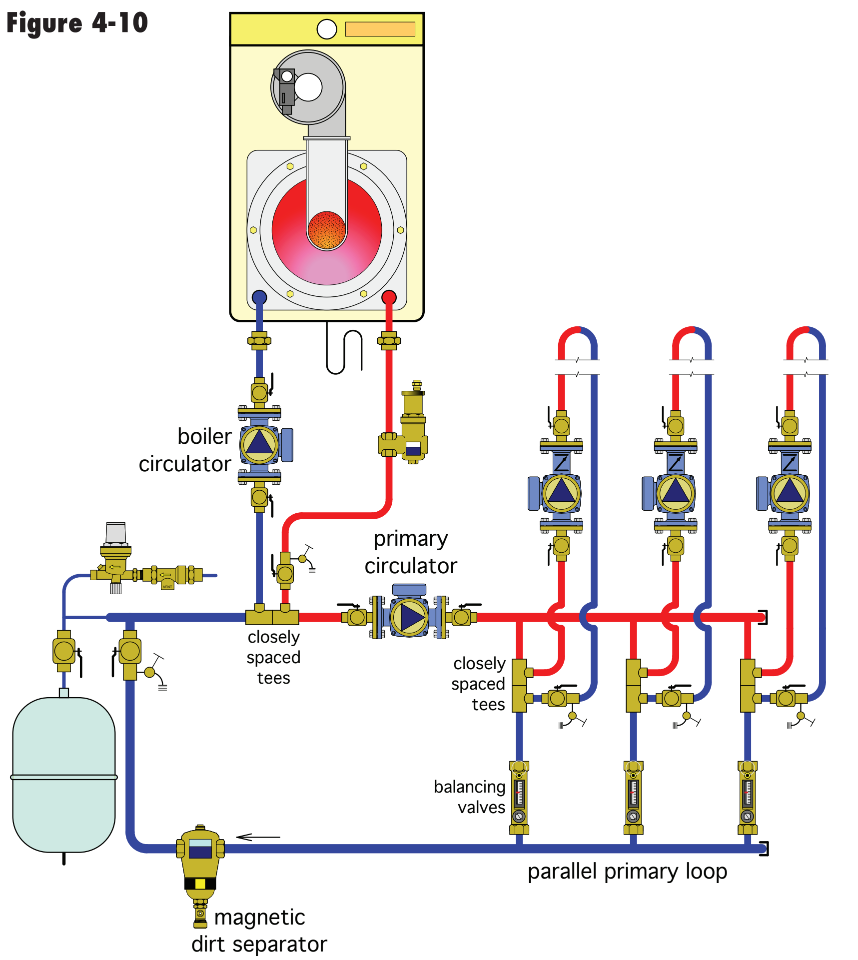
$$c_T = c_1 \times \left(\frac{(1+i)^N - 1}{i}\right) = \$165.60 \times \left(\frac{(1+0.04)^{20} - 1}{0.04}\right) = \$4,931$$

This cost is only for *operation* of the primary loop circulator. It does not include purchase, installation or maintenance of the circulator over time.

BEYOND PRIMARY/SECONDARY SYSTEMS:

It is possible to retain the hydraulic separation benefits of primary/secondary piping, and deliver equal supply water temperature to each load circuit *without having to construct a parallel primary loop, or use a dedicated primary loop circulator.*

Figure 4-10



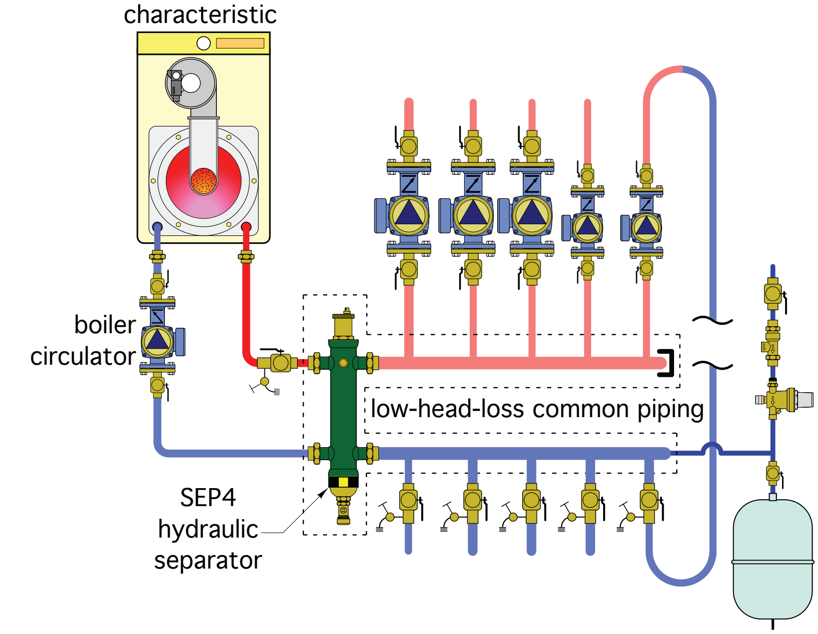
USE A HYDRAULIC SEPARATOR:

One way to provide hydraulic separation between multiple circulators and provide equal supply water temperature to each load is by installing a device called a hydraulic separator, as shown in figure 4-11.

The shape of a hydraulic separator minimizes any dynamic pressure drop through it in either the vertical or horizontal direction. When combined with short and generously sized headers, the common piping in the system, identified by the dashed lines in figure 4-11, has very low head loss. The result is excellent hydraulic separation between all circulators in the system.

Some hydraulic separators contain coalescing media that allow them to provide high-performance air and dirt separation. As such, they can eliminate the need to install separate air and dirt separating devices in the system, as illustrated in figure 4-12.

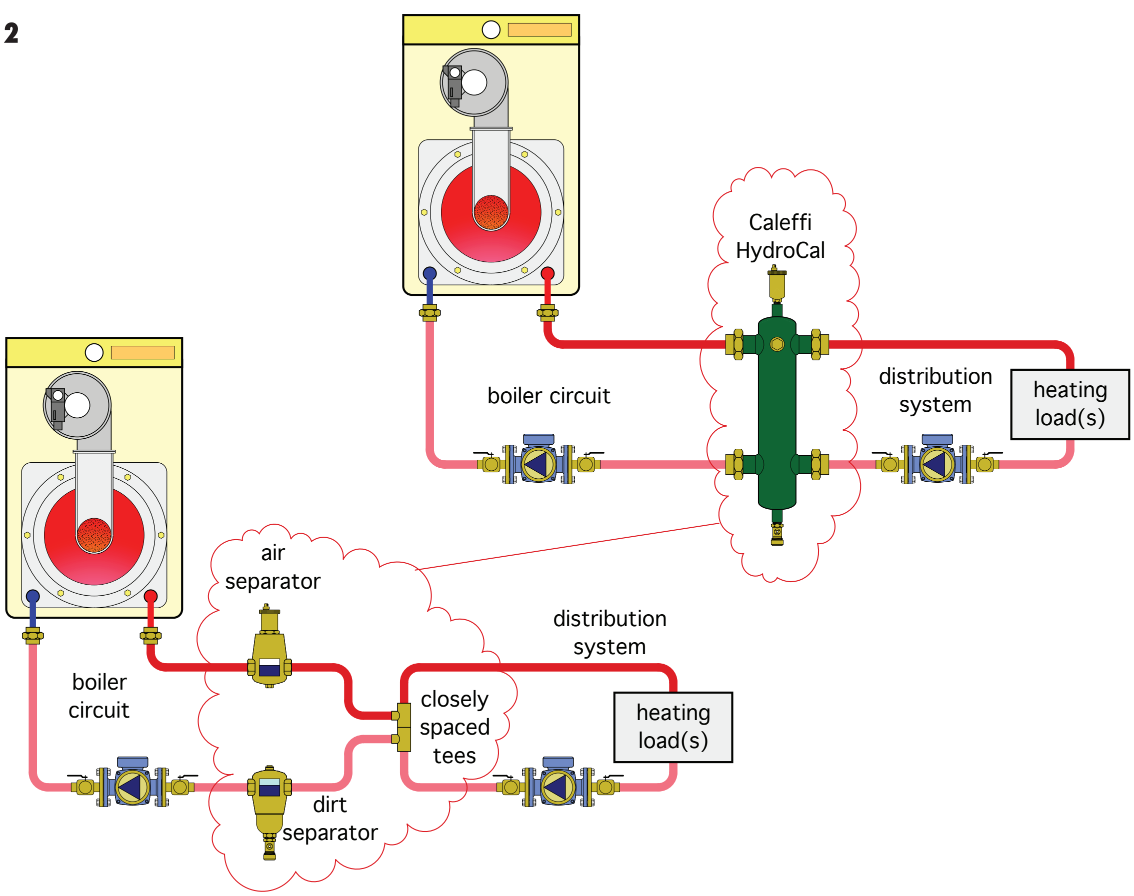
Figure 4-11



The use of a hydraulic separator eliminates the need to create a primary loop. With no primary loop, there is no need of a primary loop circulator. This can significantly reduce installation cost, as well as operating cost over the life of the system.

idronics #15 provides a more detailed discussion of hydraulic separators.

Figure 4-12



5. MULTIPLE-TEMPERATURE SYSTEMS

Many modern hydronic distribution systems require two or more supply water temperatures. At certain times, these different supply water temperatures must be created simultaneously, but remain independently controlled.

The most common approach is to select a heat source capable of heating water to the *highest* required temperature for the system, and combine this with one or more mixing assemblies that can reduce that temperature as required in other portions of the system. An example would be a system that supplies 180ºF water to fin-tube baseboard on one level of a home, while simultaneously supplying 100ºF water to radiant floor-heating circuits in the basement slab.

These multi-temperature systems are easy to create using modern hydronic hardware and control methods. In the scenario just described, a boiler would be set to produce water at 180ºF. Some of that water would be routed directly to the fin-tube baseboard portion of the system without any mixing. In another portion of the system, a mixing assembly would be configured to supply 100ºF water to the radiant panel circuits. It would do so by blending a portion of the cooler water *returning* from the radiant panel circuits with some of the 180ºF water from the boiler.

idronics #7 provides a more detailed discussion of mixing theory.

CONCEPT OF A MIXING ASSEMBLY

It is helpful to visualize a hydronic system that involves mixing as two circuits connected by a mixing assembly, as shown in figure 5-1.

One circuit is called the "heat source circuit." It includes one or more heat sources, as well as a circulator, and usually some "trim" such as an air separator, make-up water assembly and purging valve. The heat source circuit is responsible for creating the highest water temperature in the system and delivering it to the mixing assembly.

The other circuit is called the "load circuit." It contains piping for one or more heating zones, one or more circulators, and the heat emitters. The load circuit is responsible for delivering water received from the mixing assembly to all heat emitters in the system.

These two circuits are connected by a "bridge" which is called the *mixing assembly*. There are several hardware devices and arrangements that could form the mixing assembly. They will be discussed later. For now, focus on the mixing assembly as the link between where heated water is created, and where that heat needs to be delivered. No heat can pass from the heat source circuit to the load circuit without passing across the mixing assembly. Thus, by controlling the mixing assembly, one can completely regulate heat transfer from heat source to load from zero up to the maximum heating capacity of the heat source. The ability of the mixing assembly to completely control the rate of heat transfer from source to load will reveal itself as critically important in many situations.

Figure 5-1

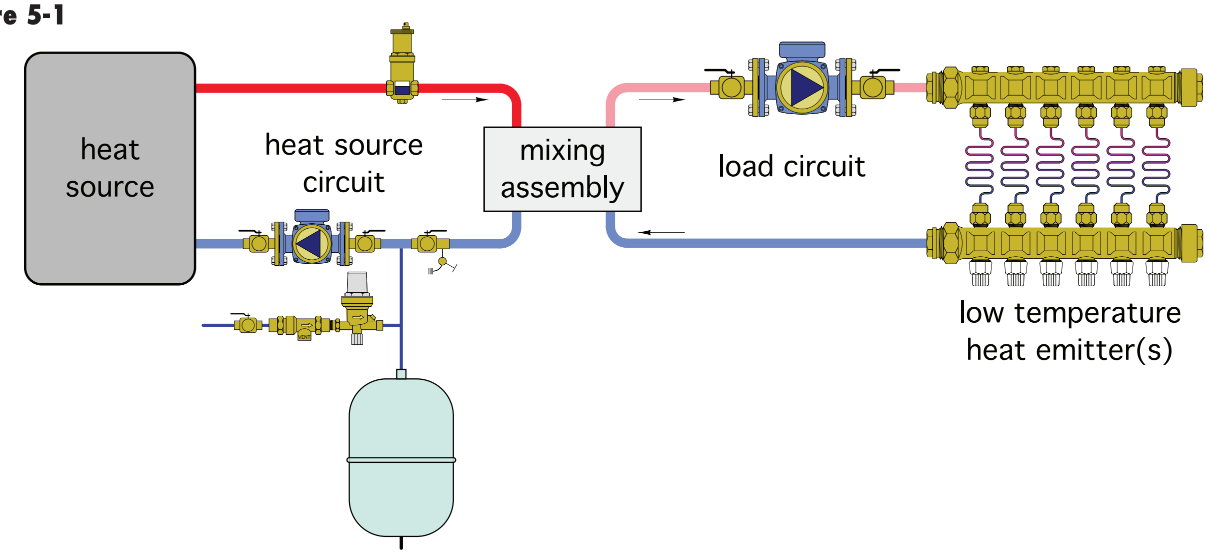
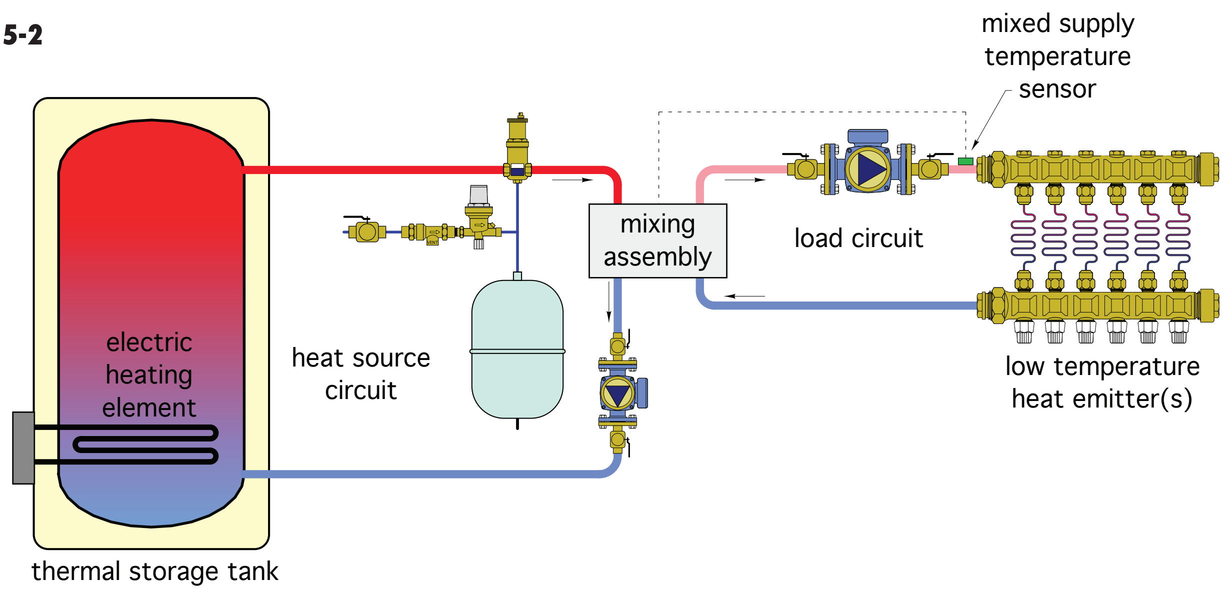


Figure 5-2



In some systems, the only function of the mixing assembly is to create the necessary *supply* water temperature to the load circuit. For example, imagine a system in which the heat source for a low temperature distribution system is a thermal storage tank that has been heated by an electric element, as shown in figure 5-2.

In this system, the mixing assembly's only responsibility is to monitor the temperature of the water supplied to the low temperature heat emitters and adjust the proportions of hot water from the thermal storage tank with cooler water returning from the heat emitters, so that the supply temperature remains at the desired value.

PREVENTING SUSTAINED FLUE GAS CONDENSATION

Many hydronic systems use "conventional" boilers burning a hydrocarbon-based fuel such as natural gas, propane, fuel oil or wood. These boilers have been designed to operate at water temperatures that do not create sustained condensation of the flue gases within the boiler or its exhaust system. In these systems, the mixing assembly must provide a second function: Protecting the conventional boiler from operating with sustained flue gas condensation.

Maintaining the boiler's inlet water temperature above a specified value prevents sustained flue gas condensation. That value varies with the design of the boiler and the type of fuel being burned. Boiler manufacturers should be consulted for specific minimum inlet water temperature requirements. In the absence of manufacturer recommended values, a minimum inlet water temperature of 130ºF is usually sufficient to prevent sustained flue gas condensation.

One of the most reliable methods of controlling boiler inlet water temperature is through use of a properly controlled mixing assembly. *However, to be effective, that mixing assembly must measure and react to changes in boiler inlet temperature.* Otherwise the mixing assembly is "blind" to what is happening at the boiler inlet, and cannot assure the boiler is protected against sustained flue gas condensation.

Figure 5-3 shows a modified version of figure 5-1. The heat source is now a conventional boiler. A temperature sensor that measures boiler inlet water temperature has been added. That sensor provides feedback to the mixing assembly.

Whenever the mixing assembly in figure 5-3 determines that the boiler inlet temperature is at or below a user-set minimum value, it reacts by reducing the rate of hot water flow from the boiler to the mixing assembly. *This control action allows the full heat production of the boiler to pass to the distribution system, but does not allow the distribution system to extract heat from the water circulating through it at a rate greater than the rate of heat production.* This action also "lifts" the combustion side of the boiler's heat exchanger above the dewpoint of the exhaust gases.

idronics #7 pro led discussion of protecting conventional boilers against sustained flue gas condensation.

Figure 5-3

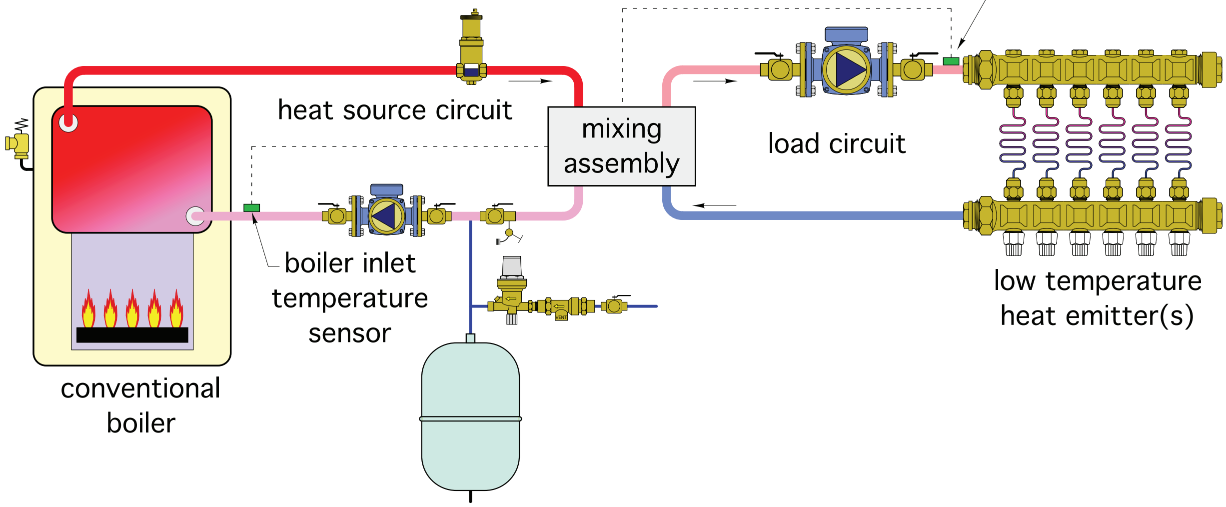
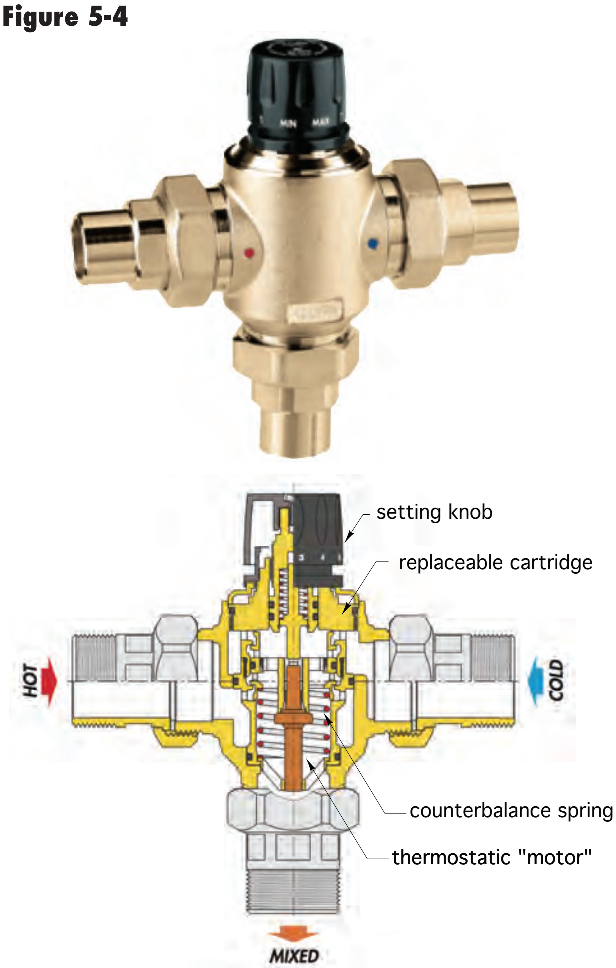


Figure 5-4



MIXING WITH 3-WAY THERMOSTATIC VALVES

One device that is often used for mixing is a 3-way thermostatic valve. Figure 5-4 shows an example of such a valve.

All 3-way mixing valves have two inlet ports, one for hot water and the other for cooler water. The inlet flows merge within the valve to create a single outlet flow at the mixed temperature condition.

3-way thermostatic valves use a non-electric thermostatic element to modulate the two incoming flows in an attempt to create an outlet flow at the temperature set using the valve's knob.

If the mixed fluid temperature flowing across the thermostatic element decreases below the setting, the valve's hot port opens farther as its cold port moves toward (but not necessarily to) its closed position. If the mixed fluid temperature flowing across the thermostatic element rises, the hot port begins to close as the cool port opens. The valve constantly monitors and responds to the fluid temperature leaving its mixed port. High-quality thermostatic mixing valves can respond to temperature changes within a few seconds.

ALWAYS CHECK THE VALVE'S Cv VALUE

When selecting a 3-way thermostatic valve for a hydronic mixing application, it is important to consider its flow resistance. The head loss across the valve at a given flow rate can be calculated by knowing the valve's flow coefficient (also known as its Cv).

The relationship between head loss, flow rate and valve Cv is given by formula 5-1:

Formula 5-1

$$H_{loss} = \left(\frac{2.308}{C_v^{\,2}}\right) f^2$$

Where:
H_{loss} = head loss across valve (ft)
C_v = flow coefficient of valve
f = flow rate through mixed port of valve (gpm)

For example, the head loss experienced across a 3-way thermostatic valve with a C_v of 3.5, while passing 6 gpm through its mixed port would be:

$$H_{loss} = \left(\frac{2.308}{C_v^{\,2}}\right) f^2 = \left(\frac{2.308}{3.5^2}\right) 6^2 = 6.8\,ft$$

This head loss is equivalent to about 98 feet of 3/4-inch type M copper tubing operating at the same 6 gpm flow rate, or 343 feet of 1-inch type M copper tubing operating at 6 gpm. This is a relatively high head loss that could significantly reduce flow through the distribution system.

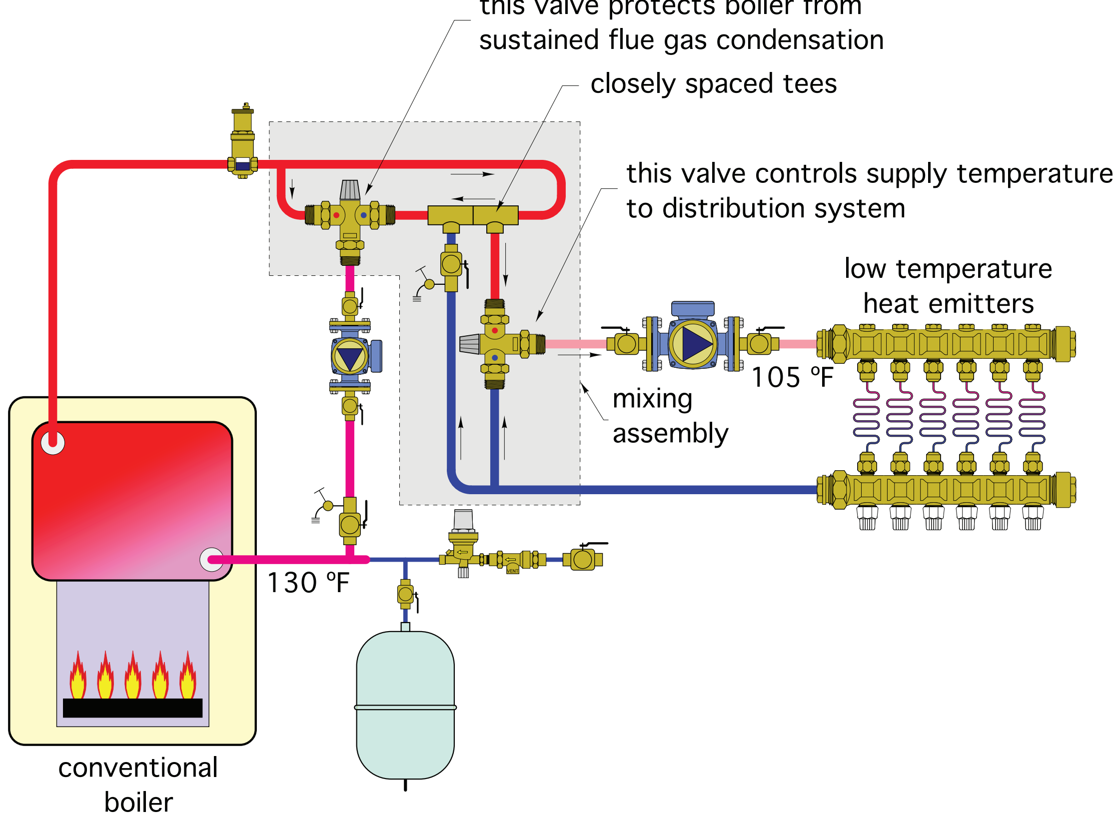
A guideline in selecting 3-way thermostatic valves is to match the C_V of the valve with the full-load flow rate through the valve (e.g., with all downstream zones operating). Sizing the valve to this guideline will result in a nominal 1 psi pressure drop and a nominal 2.3-foot head loss across the valve. *Do not assume that just because the valve has piping connections that match the piping where it will be installed, it has the proper flow resistance characteristics.*

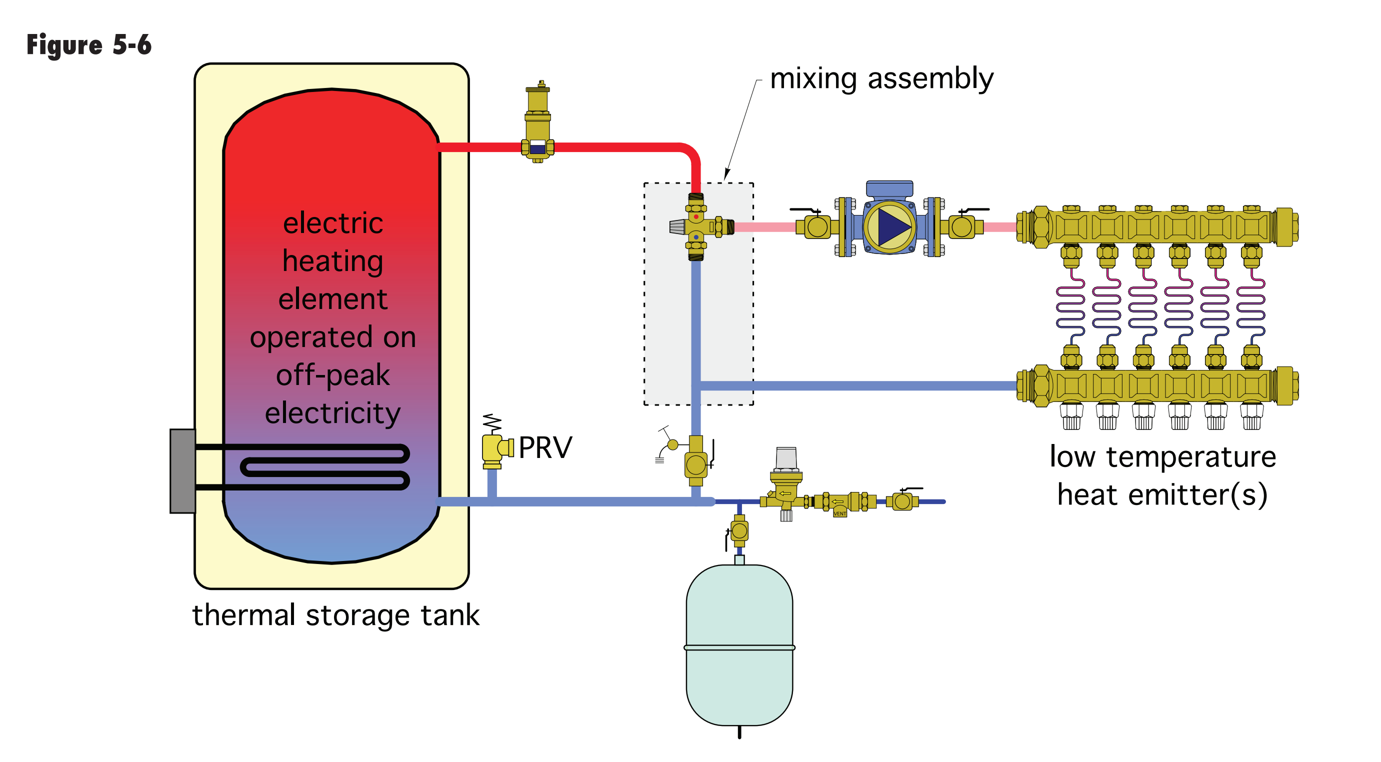
PIPING FOR 3-WAY THERMOSTATIC MIXING VALVES

The piping configuration used for 3-way thermostatic mixing valves depends on the heat source used. If that heat source is a conventional boiler, and thus requires protection against sustained flue gas condensation, it is necessary to install *two* 3-way valves, as shown in figure 5-5.

One of the valves in figure 5-5 provides the mixing point to create the required *supply water* temperature for the low temperature distribution system. The other 3-way thermostatic valve creates the second mixing point needed to boost boiler inlet temperature high enough to avoid sustained flue gas condensation.

Figure 5-5





Assume the upper valve in figure 5-5 is set for 130ºF, and the lower valve is set for 105ºF. When the boiler is first fired, the water temperature leaving the boiler is much lower than 130ºF. Under this condition, the hot port of the upper 3-way valve is fully open, while its cold port is fully closed. This routes all water coming from the boiler's outlet back to the boiler's inlet. None of this water is "released" to the remainder of the system.

The lower thermostatic valve also has its hot port fully open and its cold port fully closed when the system first turns on. All flow returning from the manifold station is routed to the left side of the closely spaced tees. Since no flow is entering the right-side tee, the cool water entering the right tee makes a "U-turn" and heads back to the hot port of the lower mixing valve. From there it passes out to the distribution system, albeit without picking up any heat for the time being.

Because no heat is being released to the load, the boiler is now warming up as fast as possible. When the temperature of the water leaving the upper mixing valve reaches 130ºF, it begins opening its cold port and closing its hot port in an attempt to keep the mixed water temperature close to 130ºF. Heated water can now pass around the upper right side of the schematic, enter the right side of the closely spaced tees, and pass down to the hot port of the lower 3-way valve. The lower valve will adjust as necessary to maintain a 105ºF supply temperature to the manifold station.

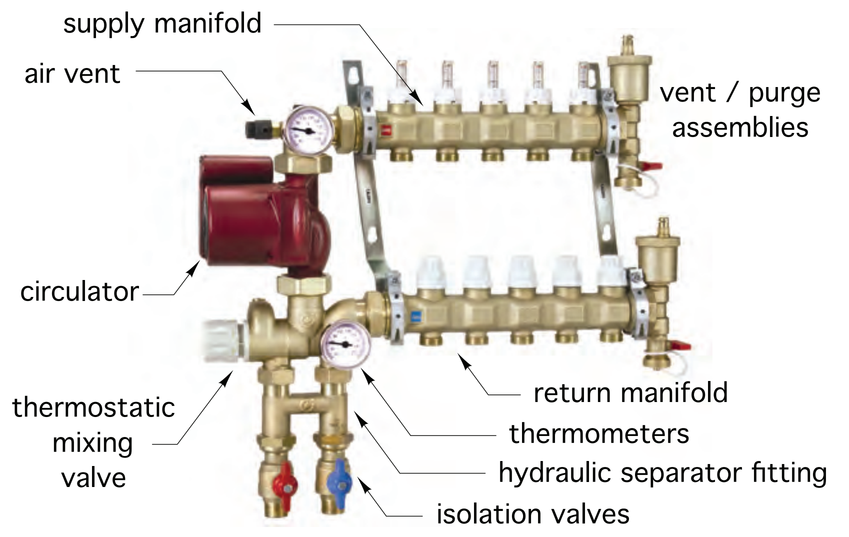
The combination of two valves prioritizes protection against sustained flue gas condensation and provides stable supply water temperature to the distribution system.

When the heat source in the system does not require protection against sustained flue gas condensation, and is not sensitive to low flow rates, a single 3-way thermostatic valve can be used to provide regulation of supply water temperature. An example of such an application is shown in figure 5-6.

In this system, the function of the 3-way thermostatic mixing valve is to protect the low temperature heat emitters against potentially high temperature water in the thermal storage tank.

Assume that the heat emitters require a supply water temperature of 105 ºF. Whenever water from the thermal storage tank exceeds this temperature, the 3-way thermostatic valve mixes in a portion of the cooler water returning from the low temperature heat emitters to maintain supply temperature close to 105ºF.

Figure 5-7



DISTRIBUTION STATIONS USING 3-WAY THERMOSTATIC VALVES

The combination of a 3-way thermostatic mixing valve, circulator and manifold station is an often-used piping assembly. Caleffi has combined these components, along with a means of hydraulic separation, into a pre-engineered mixing station, as seen in figure 5-7.

This distribution station reduces installation time and space requirements. Its integral thermostatic mixing valve reduces incoming water temperature to that required by the manifold circuits. The distribution station also provides hardware for venting and purging, and indication of both supply and return temperatures.

Figure 5-8

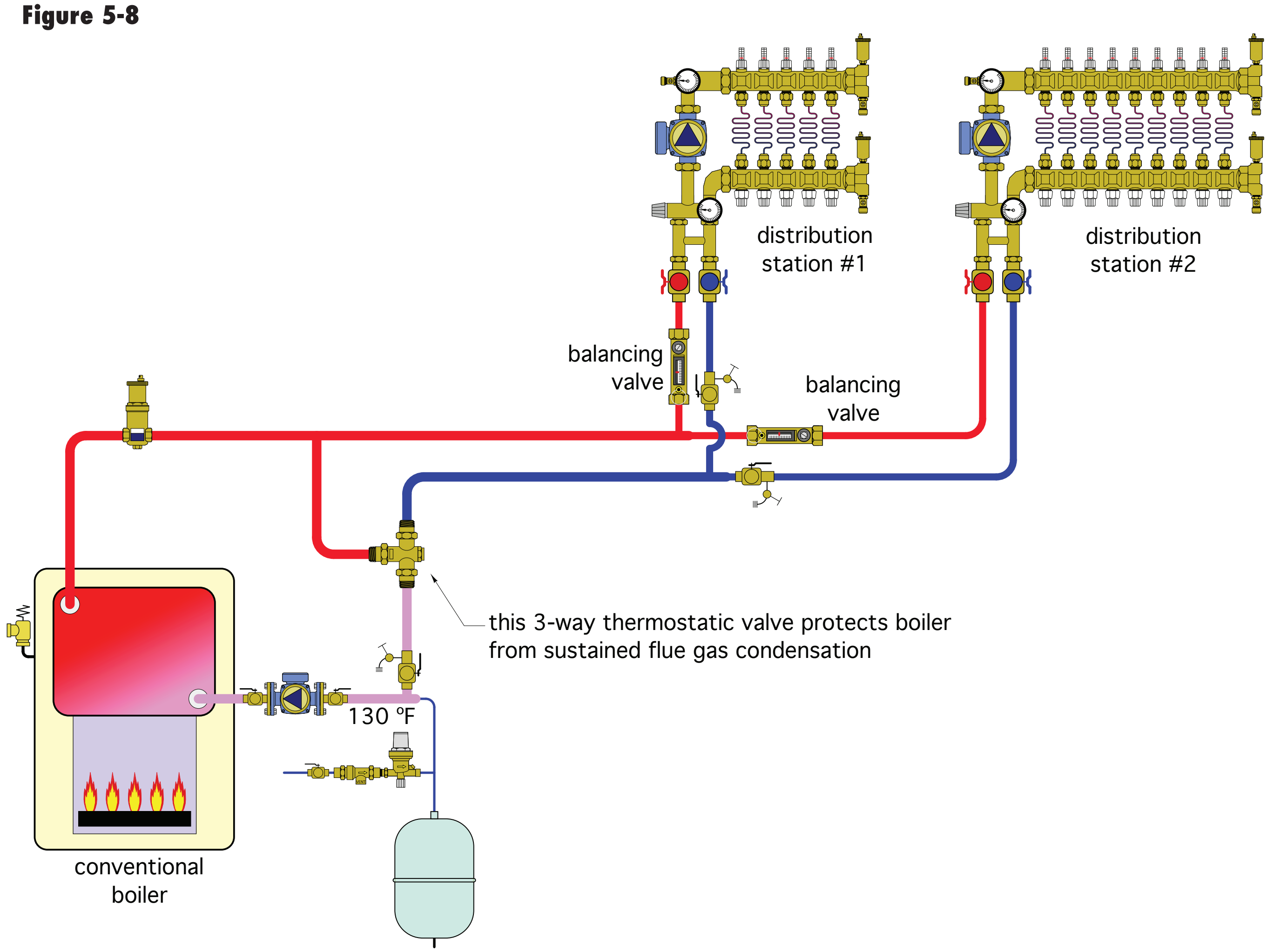


Figure 5-8 shows how two of these distribution stations could be supplied from a single conventional boiler.

Each distribution station can be set up with different numbers of circuits, and each can operate at a different supply temperature. The boiler must supply water at or above the highest of these supply temperatures. A separate 3-way thermostatic valve is installed to protect the boiler from sustained flue gas condensation.

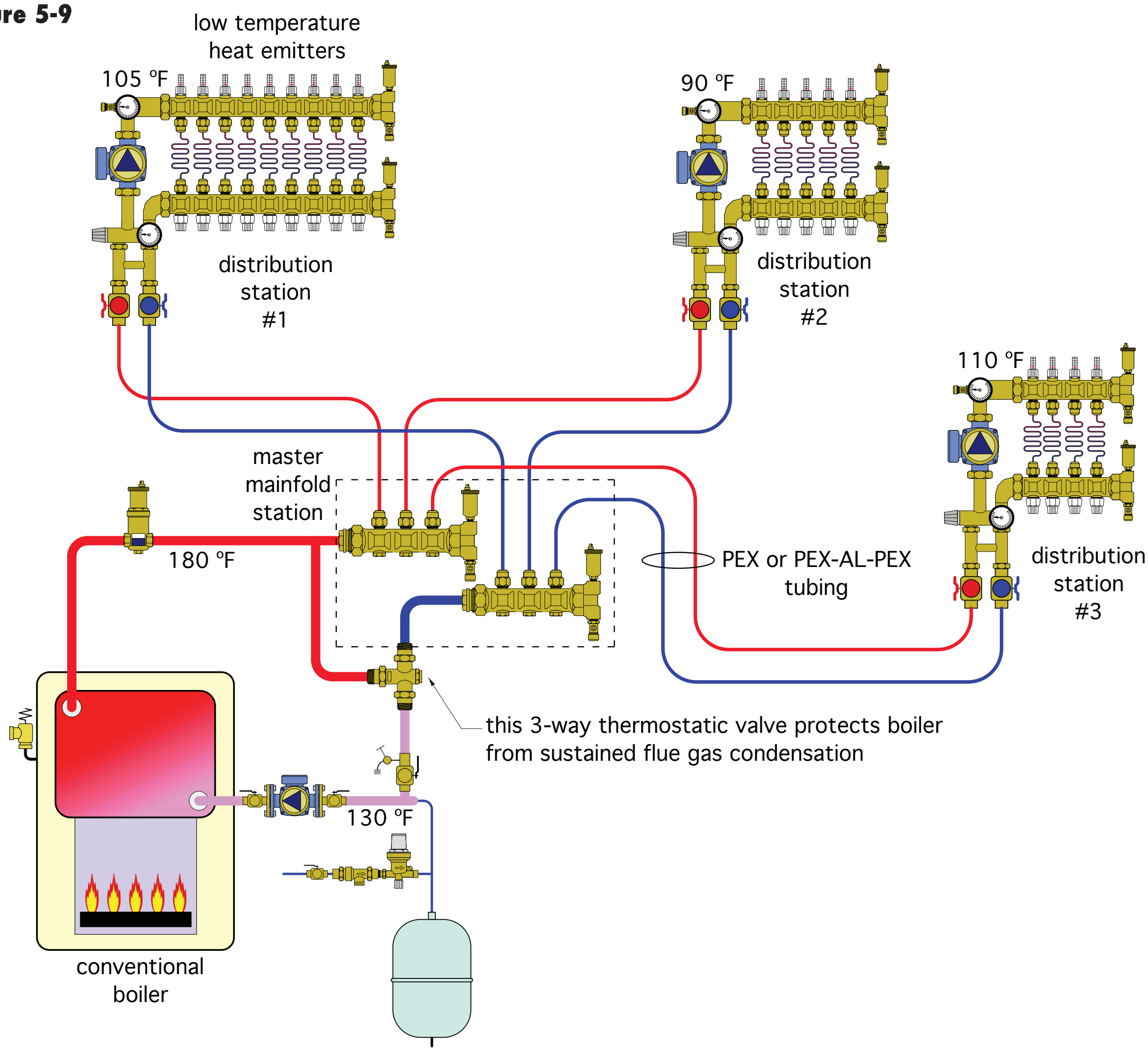
The bypass built into each distribution station provides hydraulic separation between the circulator in the distribution station, and the boiler circulator. The boiler circulator is responsible to move water to each distribution station, *but not through the radiant panel circuits connected to that distribution station.*

When the water temperature leaving the boiler is relatively high in comparison to the water temperature on the return side of the distribution stations, the flow rate supplied to each distribution station can be quite low. For example, if the boiler supplied 180ºF water under design load conditions, and the return water temperature from a distribution station supplying a 20,000 Btu/hr load is 95ºF, the flow rate that needs to be delivered to the distribution station is:

Formula 5-2:

$$f = \frac{Q}{490 \times \Delta T} = \frac{20,000}{490 \times (180 - 95)} = 0.48 gpm$$

Figure 5-9



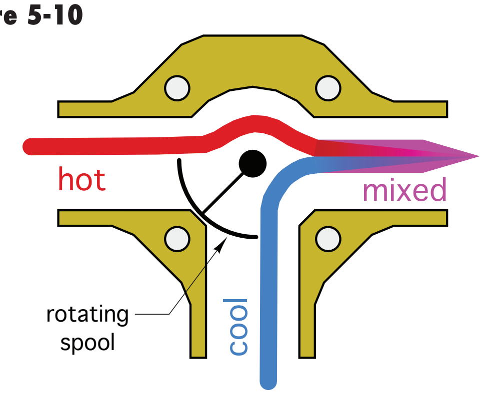
This relatively small flow rate could easily be conveyed through 1/2-inch PEX-AL-PEX tubing. This approach can simplify the installation and reduce cost relative to systems that use rigid tubing between the mechanical room and each distribution station. Figure 5-9 shows an example of this concept. A “master manifold” is used to divide flow from the boiler into each of three distribution circuits made of flexible PEX or PEX-AL-PEX tubing.

MIXING WITH 3-WAY MOTORIZED VALVES

3-way mixing valves can also be combined with motorized actuators that are operated by electronic controllers.

The type of 3-way valve body commonly used for motorized operation is different from that used for thermostatic operation. Rotary valve bodies are commonly used for motorized 3-way valves, in contrast to the linear shaft movement used in most thermostatic 3-way mixing valves. The cross section of a typical 3-way rotary mixing valve is shown in figure 5-10.

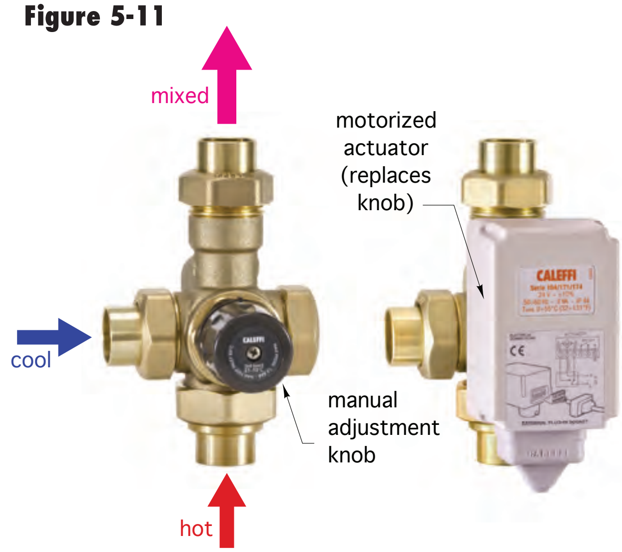
Figure 5-10



The rotating spool within the valve is a segmented cylinder. Its angular position within the body determines how much hot versus cool water enters the valve. Clockwise rotation increases cool water entry while simultaneously decreasing hot water entry, and vice versa. The two flow streams come together within the valve. Turbulence within the valve creates good mixing.

An example of a Caleffi 3-way mixing valve with motorized actuator is shown in figure 5-11.

Figure 5-11



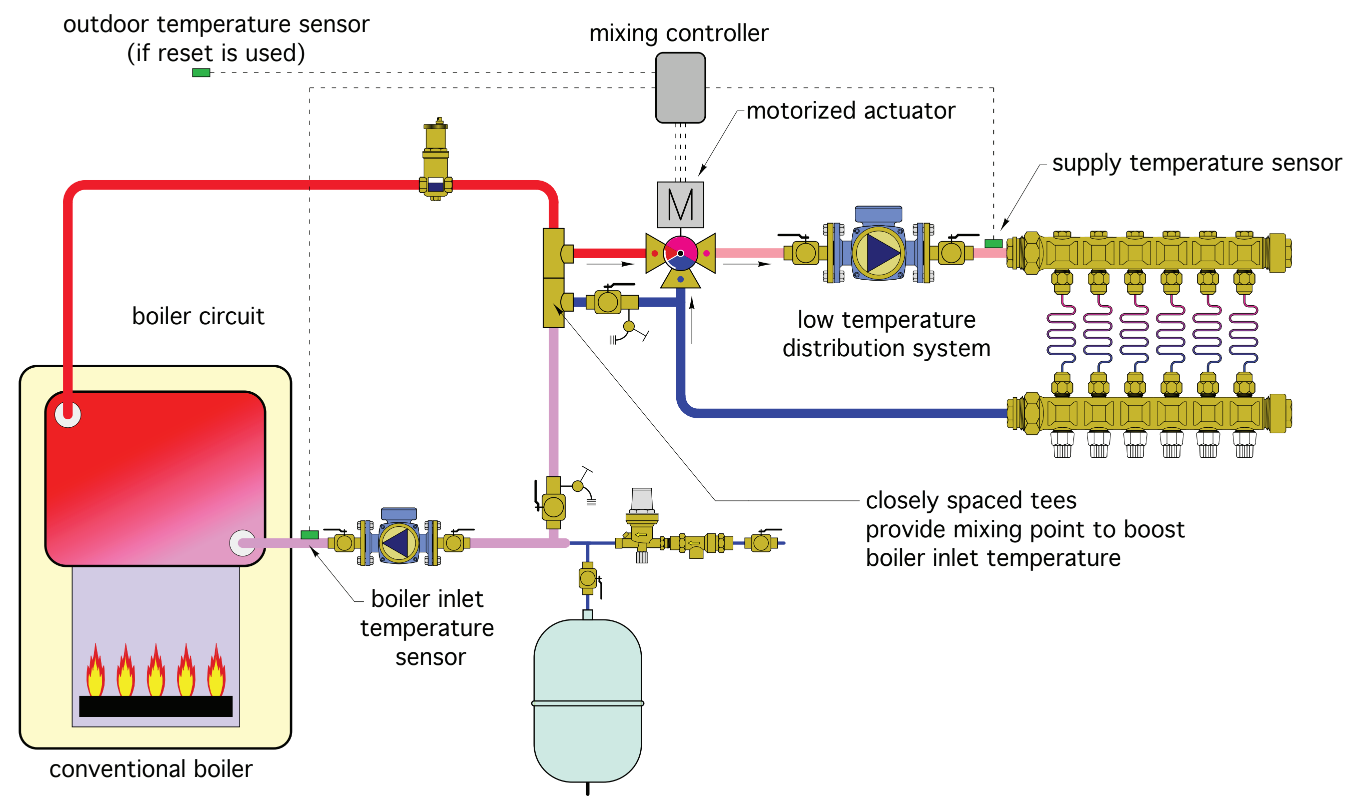
The mixing valve body is supplied with a knob, which allows it to be operated manually. For motorized applications, this knob is removed and replaced with a motorized actuator, which fastens to the valve body, as seen on the right side of figure 5-11. The actuator’s shaft is coupled to the valve’s shaft. The gear motor assembly inside the actuator provides very slow rotation of the valve’s shaft. Slow rotation prevents rapid temperature changes and improves control stability. A typical motorized actuator can require two to three minutes to rotate the valve’s shaft by 90 degrees.

Figure 5-12 shows typical piping for a 3-way motorized mixing valve that is supplying a low temperature distribution system using heat supplied by a conventional boiler.

The actuator motor driving a 3-way motorized mixing valve must be electronically controlled. When energized, the mixing controller continuously queries one or more temperature sensors, and uses this information along with its settings to operate the motorized actuator. This rotates the valve shaft one way or another, or if no signal is sent to the actuator, leaves the shaft in its current position.

Most modern mixing controllers that are designed to operate motorized valves are equipped with a sensor to measure boiler inlet temperature. The controller can then respond to this temperature by automatically reducing hot water flow into the mixing valve when boiler inlet temperature is at or below a user-selected minimum temperature that prevents sustained flue gas condensation. This feature allows a *single* 3-way motorized mixing valve to protect the boiler and regulate supply water temperature. Most mixing valve controllers also have the option of providing either a fixed setpoint or full outdoor reset of the water temperature supplied to the distribution system.

Figure 5-12



The boiler loop and boiler circulator shown in figure 5-12 are essential for creating a second mixing point (within the lower of the two closely spaced tees). The mixing point boosts boiler inlet temperature as necessary to prevent sustained flue gas condensation within the boiler.

6. DISTRIBUTION EFFICIENCY OF HYDRONIC SYSTEMS

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

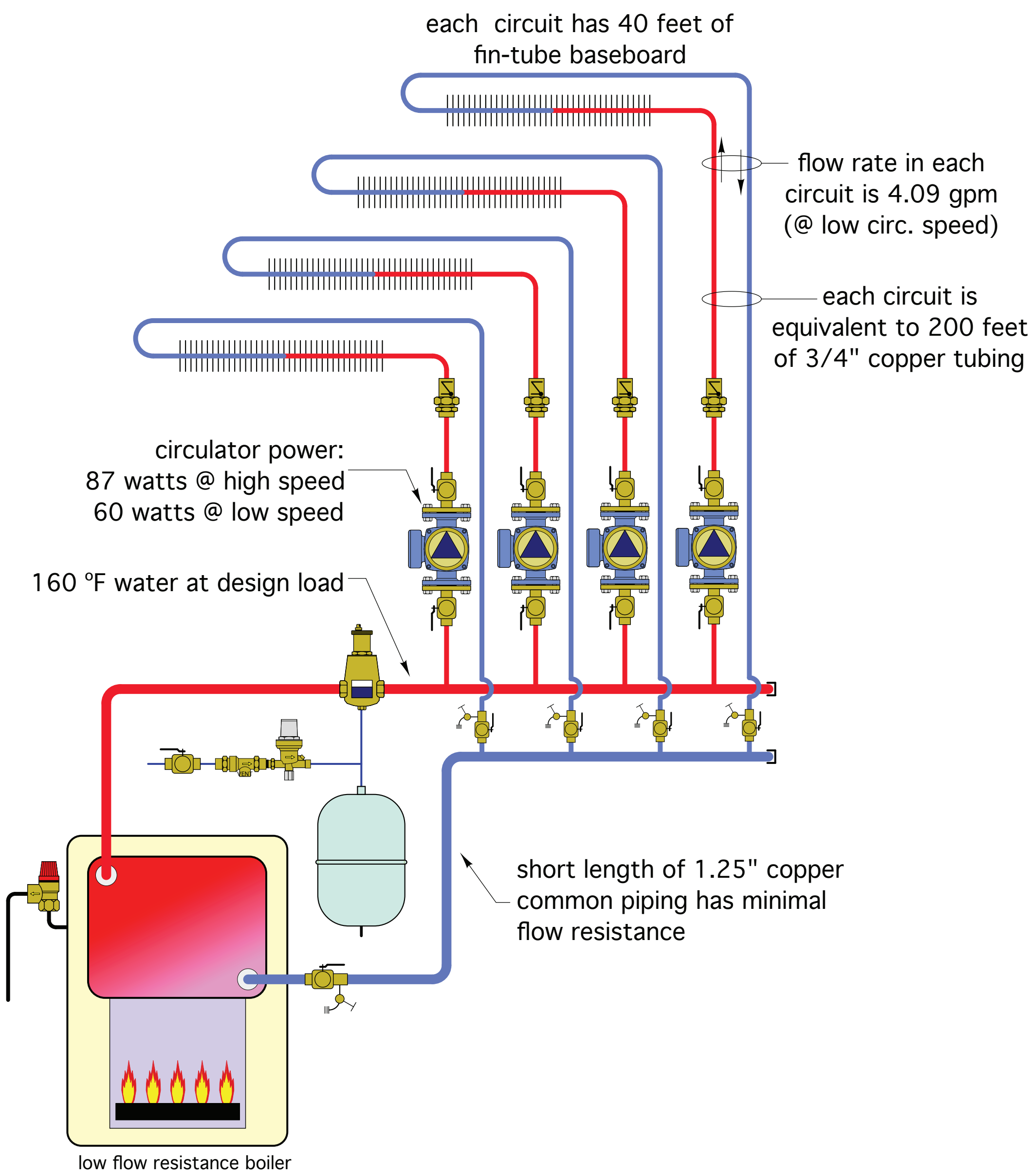
In the case of boilers, the desired output is heat, and the necessary input is fuel. The thermal efficiency of a boiler can be expressed as the instantaneous rate of heat output divided by the instantaneous rate of fuel consumption, where both quantities are expressed in the same units — usually Btu/hr.

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, it 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

Figure 6-1



use a significant amount of electrical 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:

Formula 6-1:

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

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-watt 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 supplied to the distribution system, that system delivers 333.3 Btu/hr to the building.

However, this number means little without something to compare it to. To provide 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 energy 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 6-1.

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:

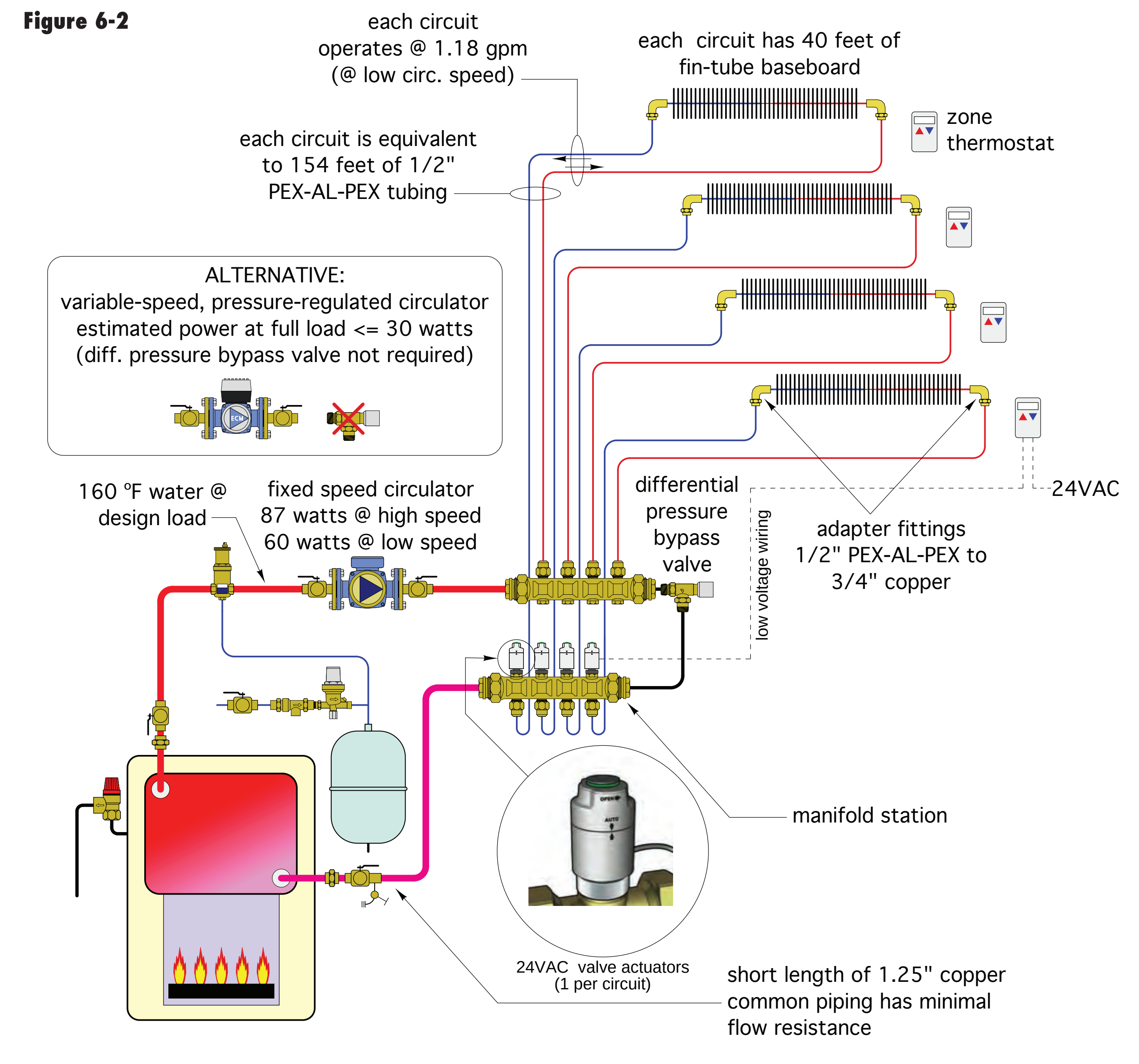
$$n_d = \frac{Q_{delivered}}{w_e} = \frac{(4 \times 12{,}310)\frac{Btu}{hr}}{(4 \times 60) watt} = 205\frac{Btu/hr}{watt}$$

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. In this example, operating the circulators on their low-speed setting increased distribution efficiency by 40%.

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

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

Figure 6-2



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 6-2.

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:

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

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:

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

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 was used in combination with the valve-based zoning, and that circulator was 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.

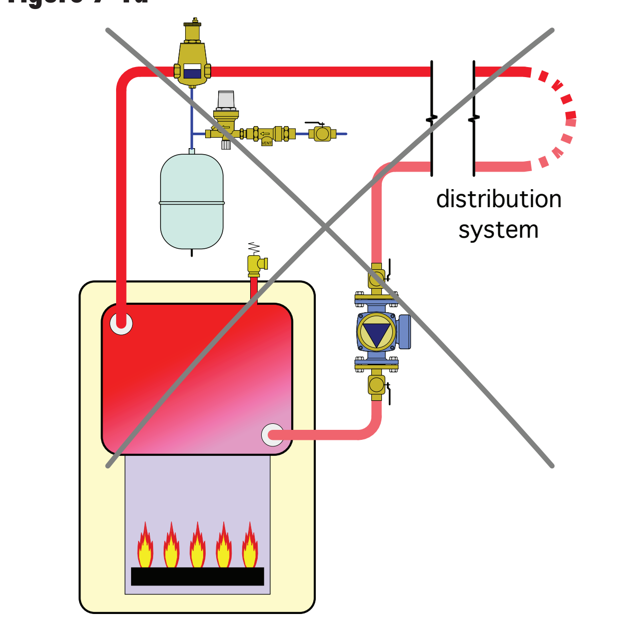
Deciding to use a system that includes a heat source with incrementally higher thermal efficiency, at the expense of much lower distribution efficiency, can cause the total operating cost to be higher. Hydronic heating professionals should emphasize the energy savings associated with the low distribution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

7. COMMON HYDRONIC PIPING ERRORS

The previous sections have discussed several preferred piping details for hydronic distribution systems. While it's good to fill a designer's toolbox with such details, it's also good to create a "blacklist" of details that should be avoided. This section presents several such details, discusses the problem, and presents at least one way to correct each deficiency.

Error 1: Circulator located upstream of expansion tank

Figure 7-1a



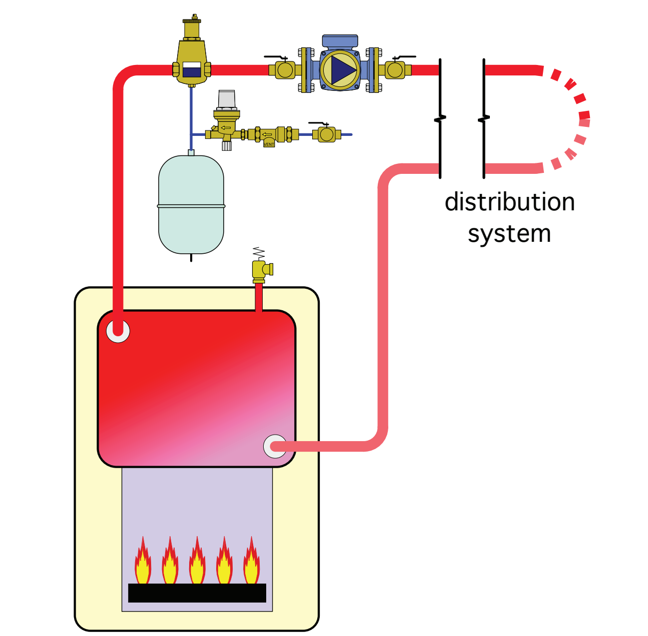
Discussion: This is a common error found in many existing hydronic systems. It stems partially from the traditional practice of installing a circulator on the return side of a boiler. This was usually done to reduce the size of shipping containers, rather than as a result of best hydronic design practice.

Another traditional practice was to attach the system's expansion tank to the bottom of the air separator, which was located on the outlet side of the boiler, as shown in figure 7-1a. This arrangement puts the point of no pressure change (PONPC) near the *discharge* of the circulator. This causes the pressure in most areas of the distribution system to drop when the circulator turns on. If the pressure at some point in the distribution system drops below atmospheric pressure, and if there are any components such as float-type air vents or valves with loose packings present at these locations, air can be sucked into the system each time the circulator runs.

This problem usually presents itself as the inability to rid the system of air. The system might be purged to a point where the service person believes air has been eliminated. However, the air problem reappears in a short time through the process described above.

Solution: Move the circulator so that its inlet port is close to the point of no pressure change (PONPC), as shown in figure 7-1b.

Figure 7-1b

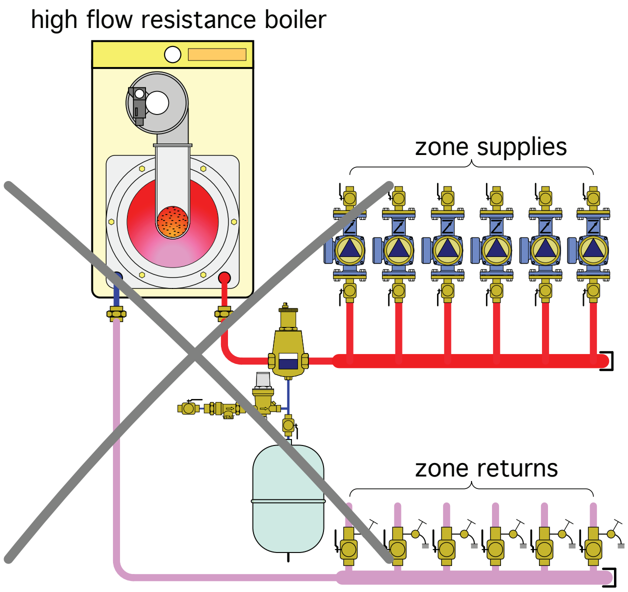


Error 2: Routing total system flow through high flow resistance heat source

Discussion: There are several types of hydronic heat sources and chillers that have high flow resistance, especially in comparison to early generation boilers. A good example is a mod/con boiler with a compact heat exchanger. Another is a water-to-water heat pump with a coaxial heat exchanger. Problems arise when such devices are piped so that all system flow must pass through them, as shown in figure 7-2a.

A high flow resistance heat source or chiller, piped as shown in figure 7-2a, can significantly increase the flow resistance of the circuit in which it is located. If this higher flow resistance is not eliminated or otherwise compensated for through use of a more powerful circulator(s), the flow rate through the system will be much lower than anticipated. The result will be inadequate heat delivery, or in the case of a chilled-

Figure 7-2a



water system, inadequate cooling. The arrangement shown in figure 7-2a also fails to provide hydraulic separation between the circulators.

Solution: Install the high flow resistance heat source in a *hydraulically separated circuit*, such as shown in figure 7-2b.

This system uses a dedicated boiler circulator in combination with a pair of closely spaced tees to hydraulically separate the high flow resistance heat source, and provide sufficient flow through it. This arrangement will not affect flow in the remainder of the distribution system. This same technique could be used for chillers.

Another option is to install a hydraulic separator between the heat source and distribution system, along with the dedicated circulator, as shown in figure 7-2c. This option provides hydraulic separation, as well as high-performance air and dirt separation, for the system.

Figure 7-2b

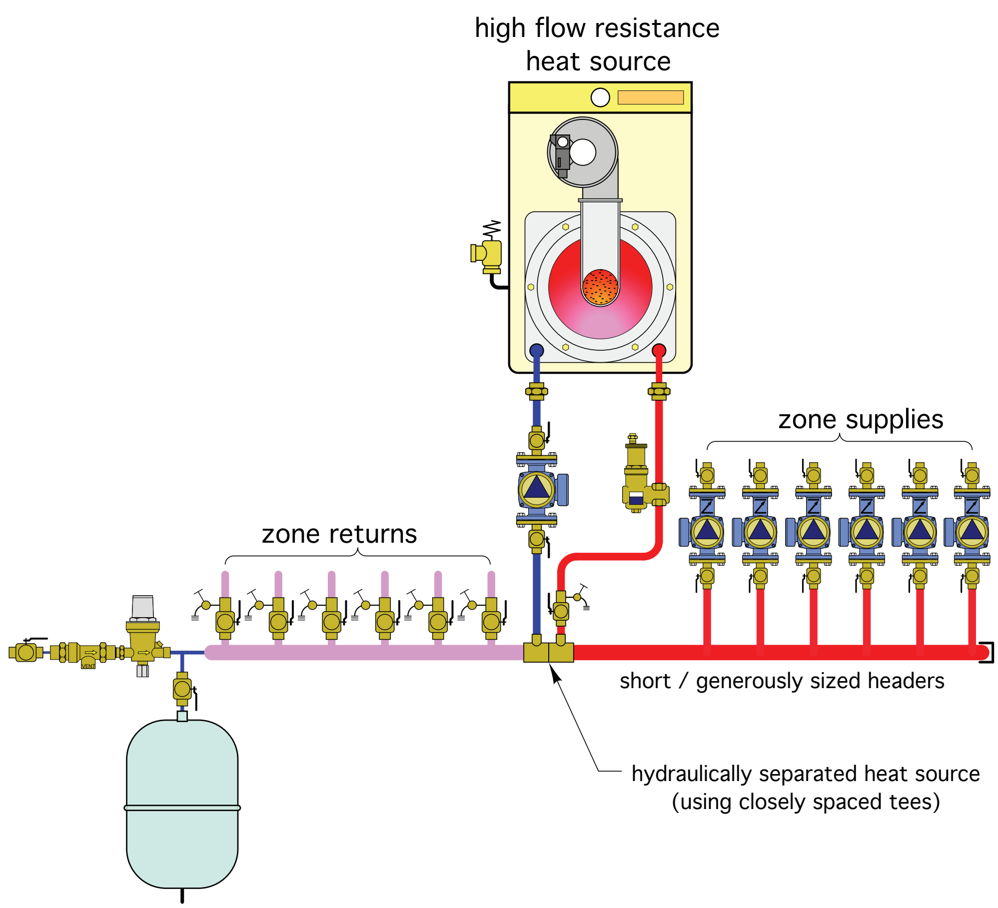


Figure 7-2c

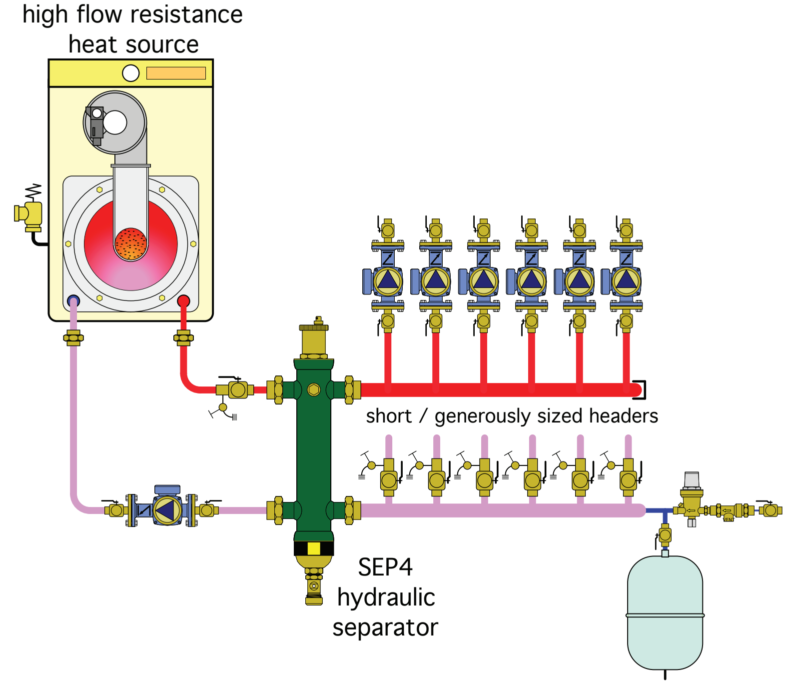
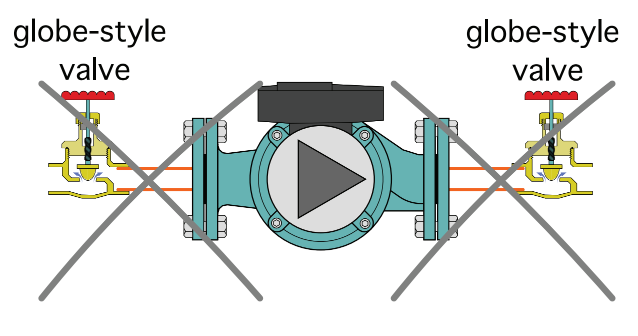


Figure 7-3b

Error 3: Using high flow resistance valves for component isolation

Discussion: Globe valves, or derivatives of globe valves, such as stop & waste valves, or any kind of valve that creates a flow path with significant and sharp changes in flow direction, should never be used for component isolation, as shown in figure 7-3a.

Figure 7-3a



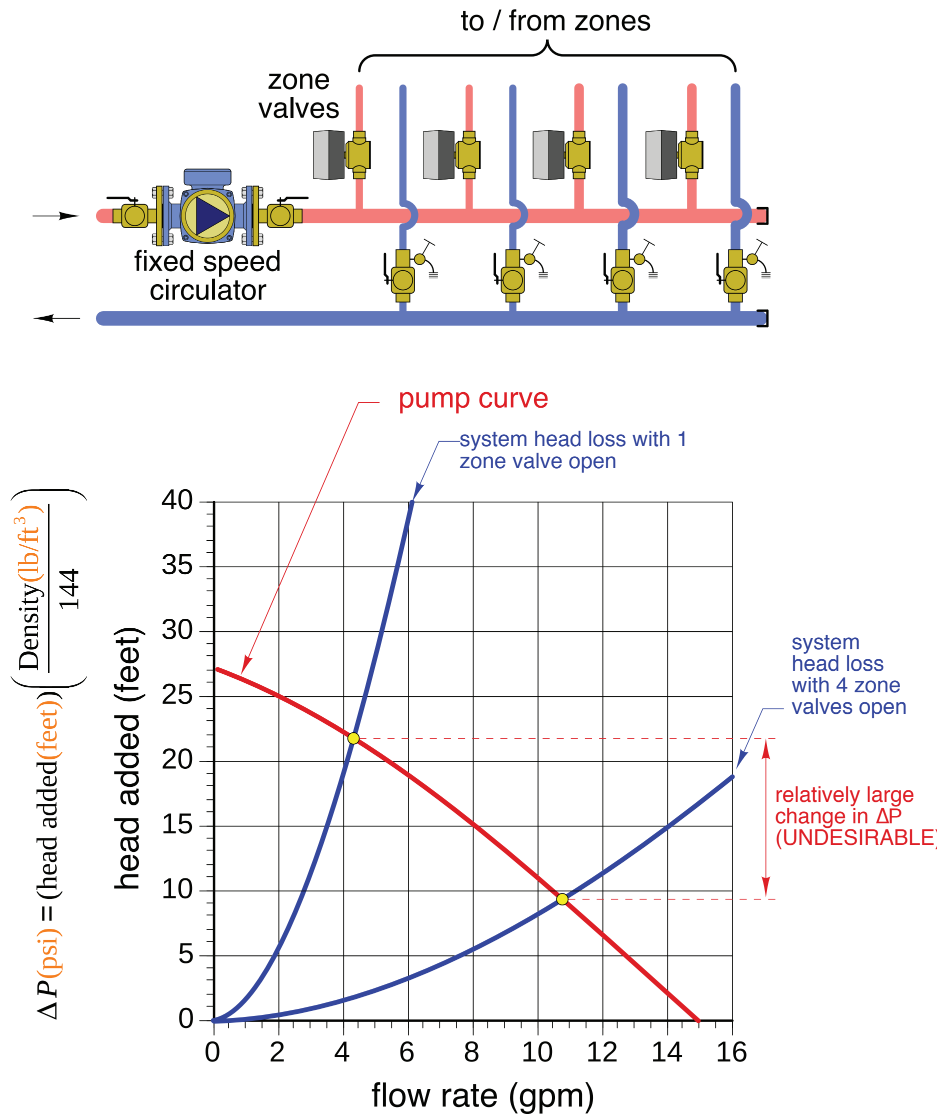
Good practice advocates placing a means of isolating major components within a hydronic system. This includes boilers, circulator, heat exchangers and tanks. The valves used for isolating components need to provide a pressure-tight seal when closed. However, *they should also create very little head loss when they are open*, which *they should be over the vast majority of their service life.* The more head loss an open isolation valve creates, the more pumping power is required to maintain a given flow rate in the system. One could think of using higher flow resistance isolation valves as equivalent to keeping the brakes on a car slightly engaged at all times. Both are a needless waste of mechanical energy.

Solution: Several types of valves can provide acceptable service as isolation valves. These include gate valves, full-port ball valves and butterfly valves. The latter are typically used in flange-connected piping systems, as shown in figure 7-3b.

Error 4: Lack of differential pressure control in systems using valve-based zoning

Discussion: Many hydronic systems use valve-based zoning. These include systems with electrically operated zone valves, manifold valve actuators or thermostatic radiator valves. The circulators in these systems are selected based on providing the necessary flow rate and corresponding head loss of the distribution system when all valves that control flow to the zones are open. When one or more zones turn off, and a *fixed-speed* circulator is providing flow, the differential pressure created by the circulator increases, as shown in figure 7-4a.

Figure 7-4a



The increased pressure differential between the supply and return headers causes the flow in the active zone circuits to increase. In some cases, this leads to flow noise, or even internal erosion of components if the flow velocities are excessive.

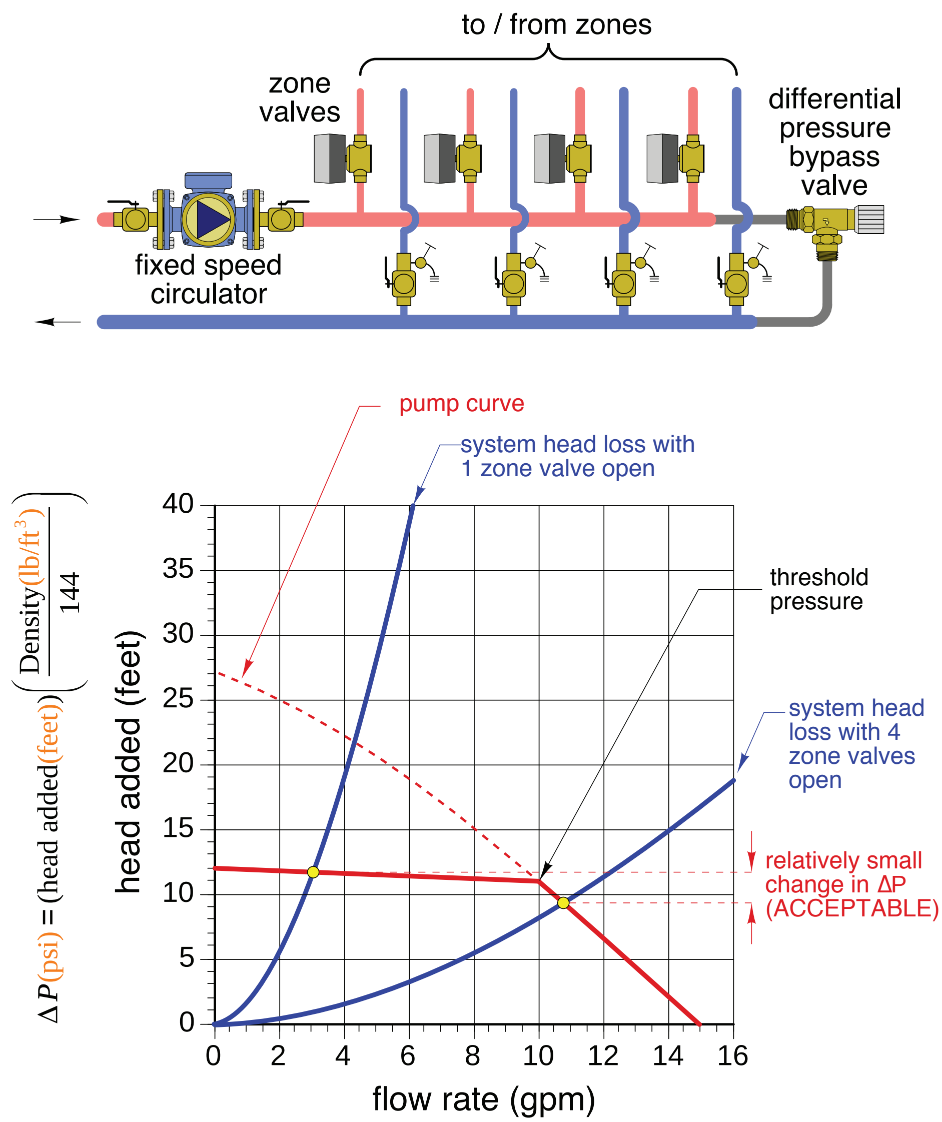
Solutions: The ideal system would maintain the same differential pressure between the supply and return headers regardless of the number of active zones. This can be closely approximated in two ways:

1. Use of a properly set differential pressure bypass valve

2. Use of a variable-speed circulator configured for constant differential pressure control

Figure 7-4b shows how a differential pressure bypass valve would be added to the piping. The threshold opening pressure of this valve should be set to at 0.5–1 psi above the pressure differential across the circulator when all zones are open.

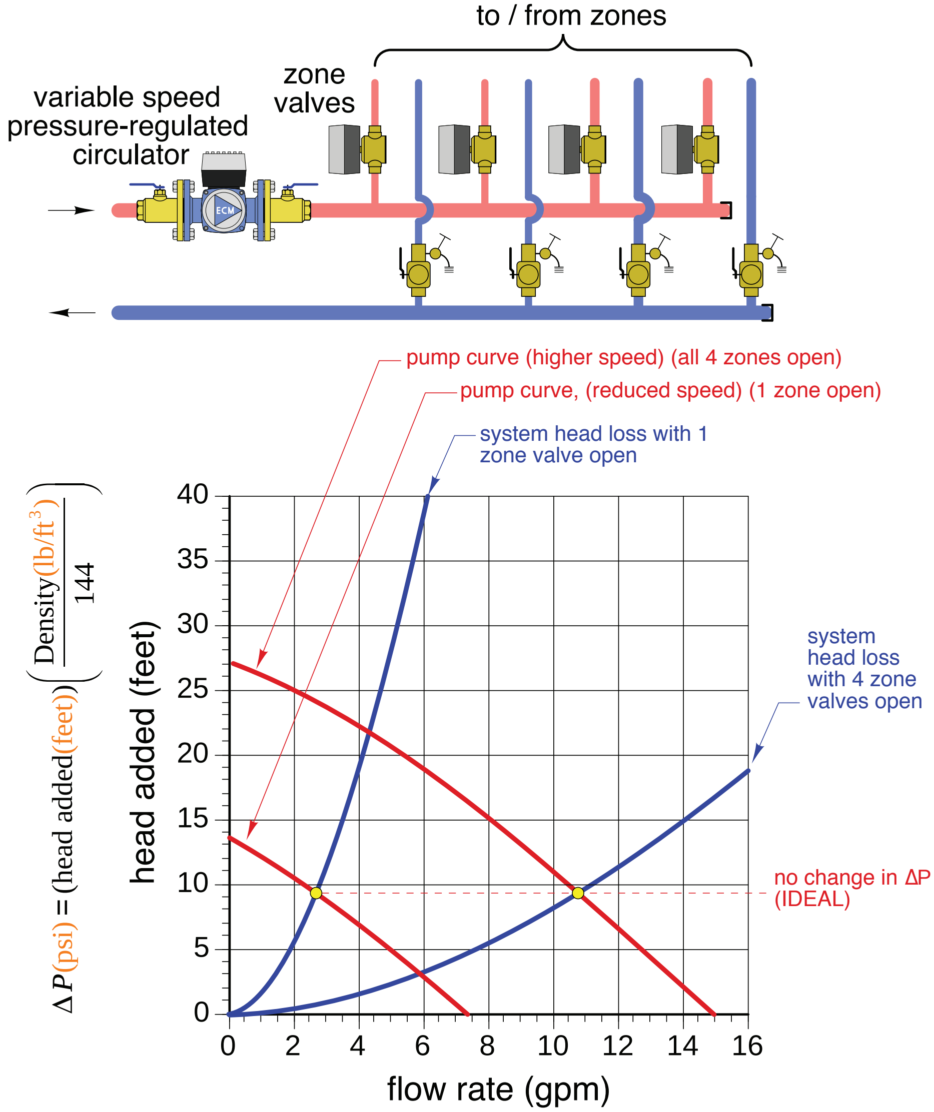
Figure 7-4b



When the pressure across the differential pressure bypass valve exceeds it threshold pressure, some flow passes through it from the supply header to the return header. This minimizes further increases in differential pressure, as shown in figure 7-3b.

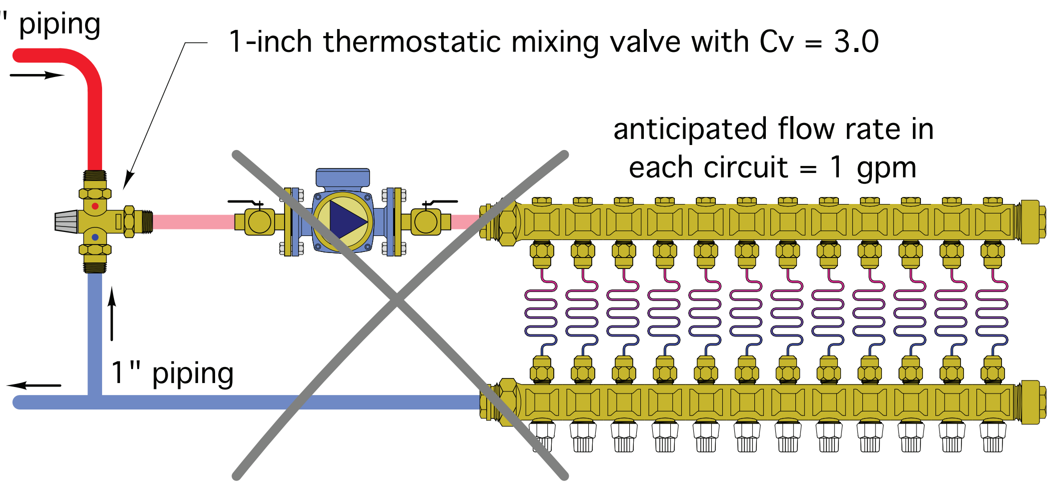
Figure 7-4c shows how a variable-speed circulator configured for *constant differential pressure* would maintain approximately constant differential pressure across the headers supplying the zones. This allows the flow rate in each zone to remain stable regardless of what zones are operating.

Figure 7-4c



When a zone valve closes, the circulator running at its current speed attempts to increase the pressure difference between its inlet and outlet. However, a variable-speed pressure-regulated circulator quickly senses this tendency and responds by decreasing its speed. As speed decreases, the circulator's pump curve shifts down and left, as shown in figure 7-4c. Within a few seconds, the circulator will settle to a speed where the user-set differential pressure exists. The net effect is that active zones remain at the same flow rate as they were before one of the zones closed. This process also works in reverse as another zone valve opens.

Figure 7-5a



Error 5: Flow-restricting thermostatic mixing valves

Discussion: It's may seem intuitive to select a 3-way thermostatic mixing valve based on its pipe size. For example, if a piping circuit is built using 1-inch copper tubing, it would seem that a mixing valve with 1-inch piping connections is the proper selection, as shown in figure 7-5a. Unfortunately, this often leads to significant flow restriction through the valve.

The underlying reason is that some 3-way thermostatic mixing valves are not specifically designed for use in hydronic systems. They are designed for use as anti-scald devices in domestic hot water systems. In the latter application, a pressure drop of a few psi as water flows through the valve is generally not a problem. However, in a small hydronic system, a pressure drop of a few psi may represent most of the pressure differential that a small circulator can create.

Mixing valves should be selected based on their Cv values (a.k.a. flow coefficient) rather than the size of their piping connections. A Cv of 3.0 indicates that a flow rate of 3 gallons per minute of 60ºF water through the valve will result in a pressure drop of 1 psi across the valve. This pressure drop increases based on the following formula:

$$\Delta P = \left(\frac{D}{62.4}\right)\left(\frac{f}{Cv}\right)^2$$

Where:
ΔP = pressure drop across valve
D = density of fluid (lb/ft^3)
f = flow rate through valve (gpm)

Thus, a valve with a Cv of 3.0 would have a pressure drop of 1 psi at a flow rate of 3 gpm, a pressure drop of 4 psi at 6 gpm, and a pressure drop of 16 psi at 12 gpm. Notice that pressure drop increases rapidly with increasing flow rate.

If a mixing valve with a Cv of 3.0 was placed in a circuit that was expected to operate at 10 gpm, and the large pressure drop of the valve at this flow rate was overlooked when the circulator was selected, the flow rate in the circuit would be far less than expected, and so would the rate of heat transfer by the circuit. Unfortunately, this is a common error in many North American hydronic systems.

Solution: Use a mixing valve with a Cv value that is approximately equal to the intended flow rate through the valve. Thus, if a circuit is supposed to operate with a 10 gpm flow rate, it would be appropriate to select the valve with a Cv of approximately 10.

3-way motorized mixing valves that are designed to be operated by an electrical actuator typically have higher Cv values than valves designed for domestic water-mixing applications.

idronics #7 provides a description of how to size and select several types of mixing valves.

COMMON ERRORS IN PRIMARY/SECONDARY DISTRIBUTION SYSTEMS

The relationship between the primary and secondary circuits in a primary/secondary system is important. Unfortunately, this relationship is sometimes compromised by trying to combine other piping techniques in an attempt to circumvent issues such as the sequential temperature drop from one set of closely spaced tees to another in a series primary loop.

Error 6: An undefined piping assembly

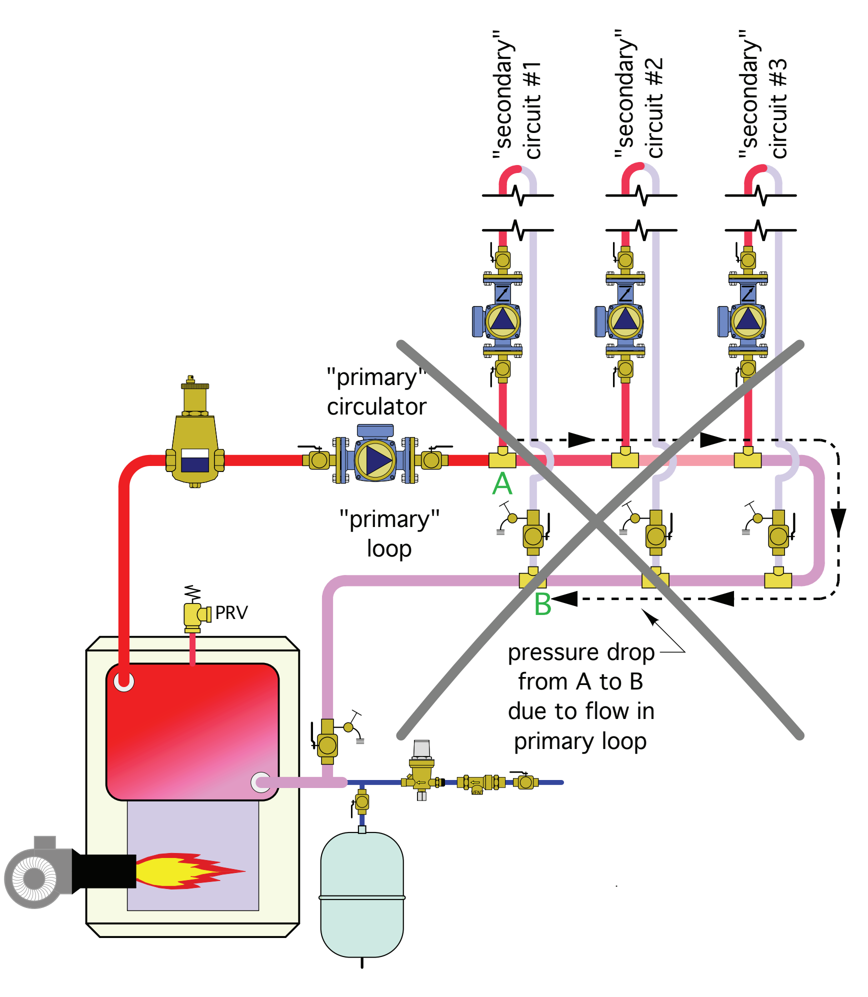
Discussion: One common error associated with attempts at creating primary/secondary piping is shown in figure 7-6a.

This piping arrangement is the likely result of someone beginning to design a system based on use of a primary loop, but in the process, reverting to the concept of "headers" when connecting the load circuits to that primary loop. The result is neither a primary/secondary system, nor any other acceptable piping topology.

The problem with this piping arrangement is the pressure differential created by head loss along the primary loop. This pressure differential can induce flow through one or more of the load circuits when those circuits are *not* supposed to be delivering heat.

For example: Consider the pressure drop from point A to point B along the primary loop shown in figure 7-6a. This pressure drop is due to head loss created by flow moving along the primary loop. Once this pressure drop exceeds the forward opening threshold of the spring-loaded check valves in a given circulator, flow will be pushed through that circulator and its associated circuit. This will deliver heat to the space served by that circuit when there is no call for such heat.

Figure 7-6a



Solution: Either design a proper primary/secondary system using closely spaced tees, or design a different (but established) piping arrangement. Do not "morph" the two concepts together, as represented in figure 7-6a.

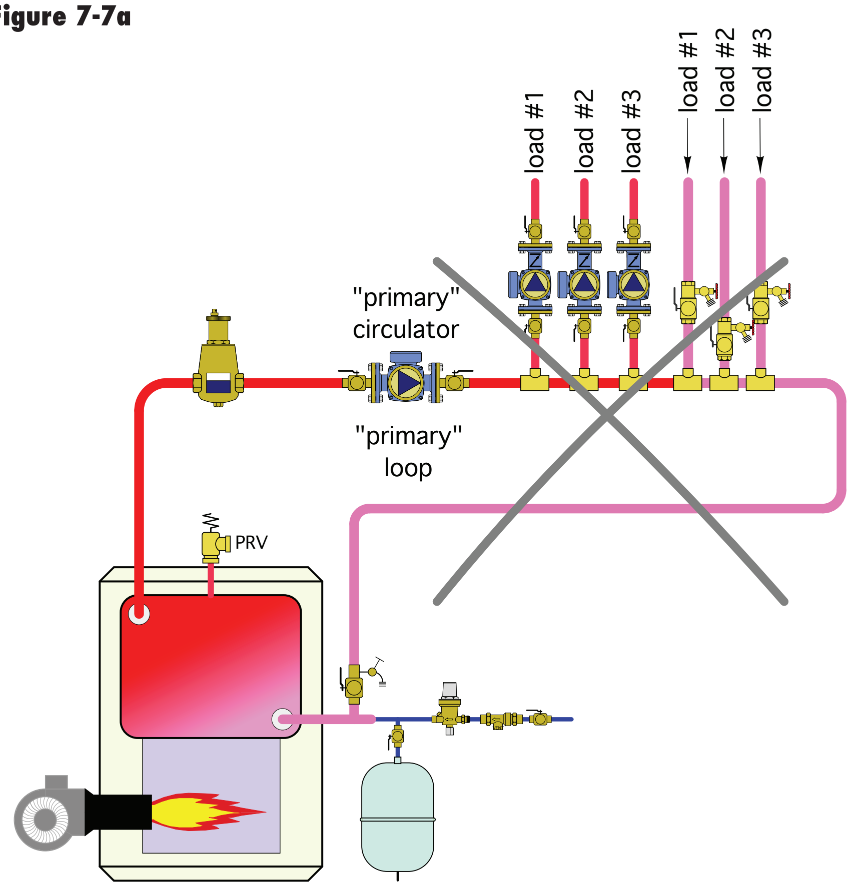
Error 7: Incorrect placement of "closely spaced" tees

Discussion: Another "intuitive" but flawed attempt at eliminating the sequential temperature drop in a series primary loop is shown in figure 7-7a.

In this system, the upstream tees for each of the three "secondary circuits" have been placed upstream of the three return tees. The rationale is that this allows each secondary circuit to operate at the same supply water temperature. The tees remain "relatively" close, so the head loss between them is "relatively" small.

This scenario will only function as intended when the flow rate in the primary loop is greater than or equal to the sum of the flow rates in all active secondary circuits. If this condition is not met, there will be a flow reversal in the pipe between the upstream group of three tees and the downstream

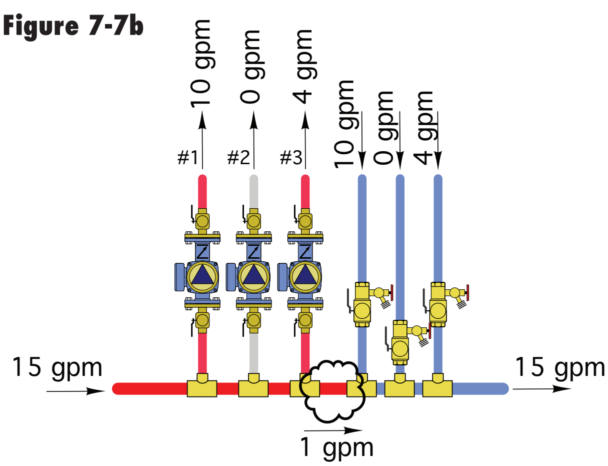
Figure 7-7a



group of tees. This flow reversal will create a mixed water temperature condition at the upstream tees.

Assume the first zone circulator was operating at a flow rate of 10 gpm, the second zone circulator was off and the third zone circulator was operating at a flow rate of 4 gpm. Also assume that the flow rate created by the primary

Figure 7-7b



circulator was 15 gpm. This would produce the result shown in figure 7-7b. All zones would be receiving water at the same temperature. So far, so good.

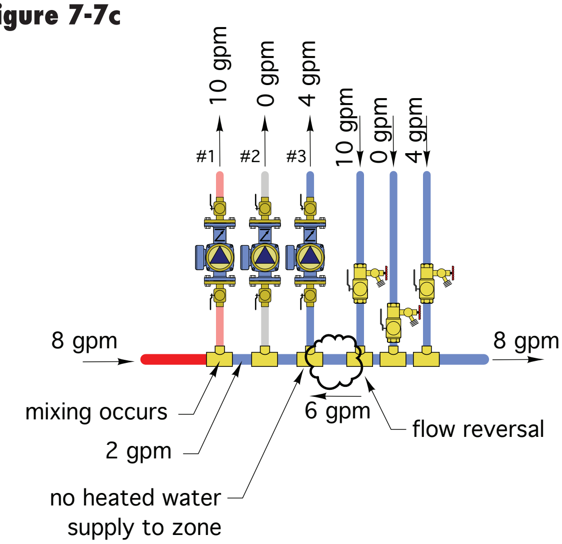
Now assume all zone flow rates remain the same, but reduce the flow rate in the primary loop from 15 to 8 gpm. The result is shown in figure 7-7c.

In this scenario, the flow between the 3rd and 4th tee has reversed. This is the only possibility, since the flow entering any portion of the system has to be the same as the flow leaving that portion of the system.

There is no heated water entering zone #3, and the flow sent into zone #1 will be mixed (8 parts heated water with 2 parts return water). This will obviously have detrimental effects on heat delivery to these zones.

There are many other possibilities for what might happen based on assumed primary loop flow rate and the combined zone flow rates. However, this piping arrangement will only produce the expected equal supply temperature to each zone circuit if the flow rate in the primary loop is always greater than or equal to the sum of the zone flow rates.

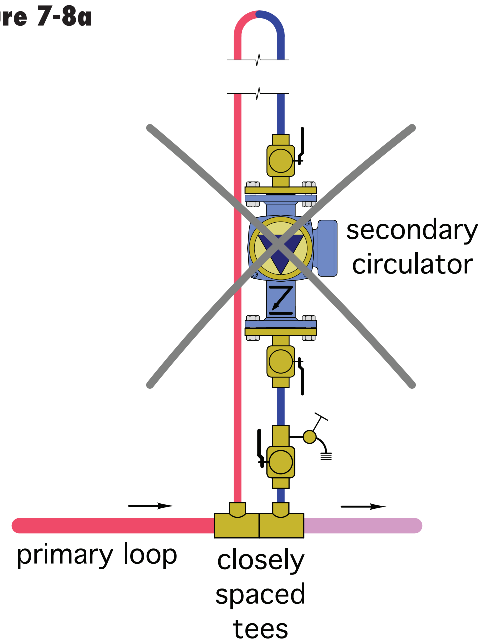
Figure 7-7c



Solution: One could argue that use of a large circulator could ensure that the primary loop flow rate is always greater than the sum of the secondary circuit flow rates. However, a larger circulator would have a high installation and operating cost. The best solution is to avoid this piping configuration and stick with proven piping arrangements.

Error 8: Incorrect placement of secondary circulator

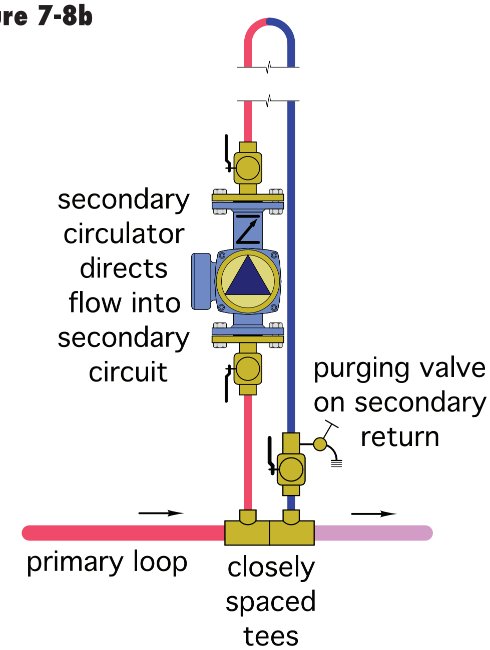
Figure 7-8a



Discussion: The expansion tank in most primary/secondary systems is connected to the primary loop. This makes the primary loop the pressure "reference point" for the secondary circuit. From the perspective of the secondary circulator, one can think of the connection between the primary loop and secondary loop as if it were a point of no pressure change. If the secondary circulator is installed on the *return* side of the secondary loop, as shown in figure 7-8a, its pressure differential will produce a drop in pressure within the secondary circuit. *This is not desirable*. If the pressure at any point in the secondary circuit drops below the vapor pressure of the fluid being circulated, that fluid will boil. If the pressure drops below atmospheric pressure at a location where a float-type air vent or a valve with a loose packing nut is located, air will be drawn into the circuit.

Solution: For these reasons, *it is best to locate the secondary circulator on the supply side of a secondary circuit*, as shown in figure 7-8b.

Figure 7-8b



This placement allows the pressure differential created by the secondary circulator to *increase* the pressure within the secondary circuit. In effect, the secondary circulator is now "pushing" fluid though the secondary circuit, rather than "sucking" fluid through that circuit.

Error 9: Lack of protection against thermosiphoning

Discussion: Hot water is less dense, and therefore "lighter" than cool water. Whenever a hydronic circuit contains a source of hot water, vertical piping, and no means for blocking flow, thermosiphoning will develop. Hot water will rise from the heat source, cool as it passes through the piping and heat emitters, and "sink" downward through the return side of the circuit. This is illustrated in figure 7-9a.

Secondary circuits located above a primary loop, and lacking a means of thermosiphon prevention, will experience thermosiphon flow. This can deliver heat into areas of the building where it is neither needed or desired. For example, consider a situation where a primary loop is delivering hot water to an indirect water heater during a hot summer day. Any unprotected secondary circuit with upward routed piping will allow some heat to migrate to the heat emitters in that circuit, perhaps on a day when that building's air conditioning is operating. Such a situation must be avoided.

Figure 7-9a

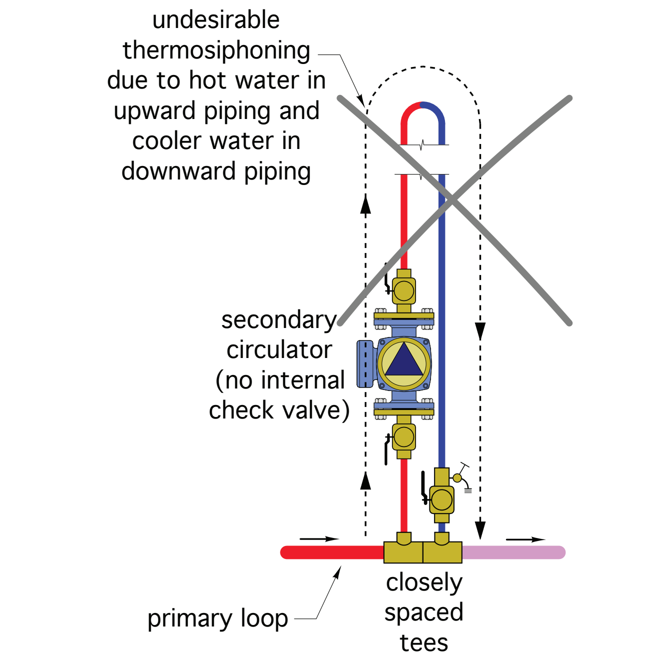
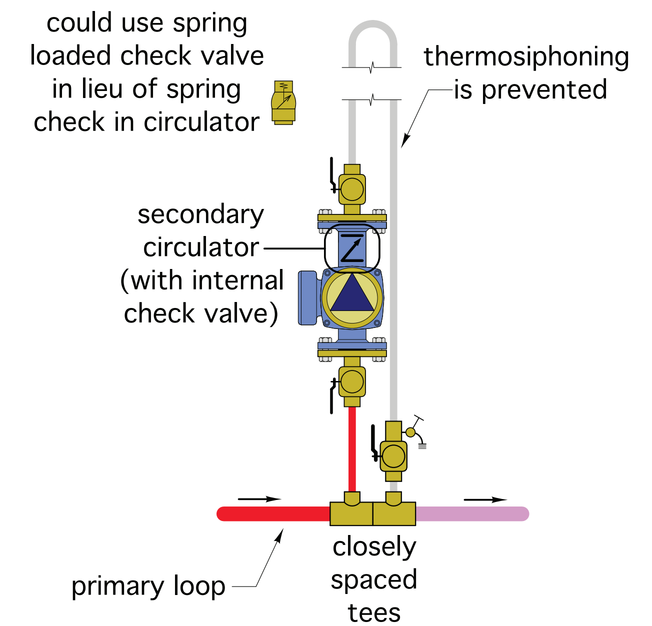


Figure 7-9b



Solution: To prevent thermosiphoning, every secondary circuit should contain a device that provides forward opening resistance sufficient to block thermosiphon flow. Examples include spring-loaded check valves mounted within the circulator volute, an external spring-loaded check valve or a weighted-plug low check valve.

Figure 7-9b shows proper piping using either of the spring-loaded check valve options.

All of these devices require a slight forward opening pressure of 0.3 to 0.5 psi, which is generally sufficient to block the weak pressure differentials created by buoyancy differences between heated supply water and cooler return water in hydronic systems with no more than 25 feet of vertical rise. This vertical rise is based on the possibility of water at 180ºF in the primary loop, water at 70ºF on the return side of the secondary circuit, and a forward opening resistance of 0.3 psi in the spring-loaded check valve.

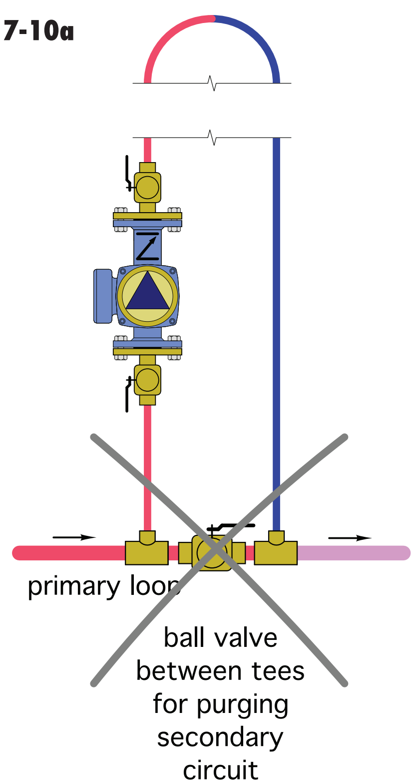
Error 10: Excessive pressure drop between tees.

Discussion: The rationale for using closely spaced tees is to minimize the pressure drop between them. Anything that adds flow resistance between the tees will reduce their ability to provide hydraulic separation.

One common practice that can be a problem is placing a ball valve between the two tees, as shown in figure 7-10a.

The valve is only used for routing purging flow from the primary loop through the secondary circuit during system commissioning, or following servicing.

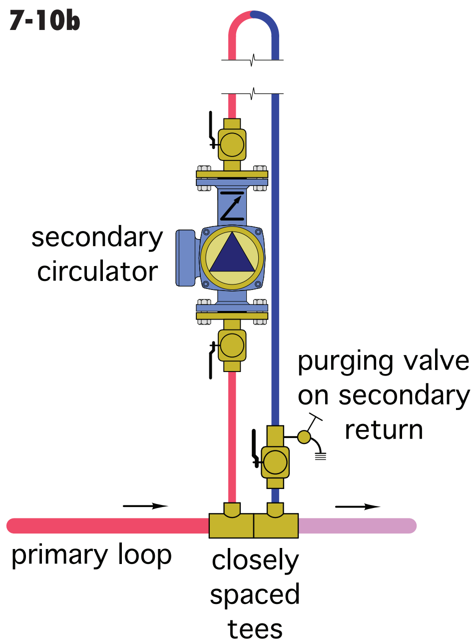
Figure 7-10a



Although the pressure drop through a “full-port” ball valve is small compared to other types of valves, it may still represent the equivalent of 1-2 feet of tubing. A “standard-port” ball valve will create even higher pressure drop.

Solution: A preferred detail is to install a purging valve *at the end of each secondary circuit,* as shown in figure 7-10b.

Figure 7-10b



This detail allows the tees to remain as close as possible to provide the best possible hydraulic separation, as well as providing a way to effectively purge each secondary circuit.

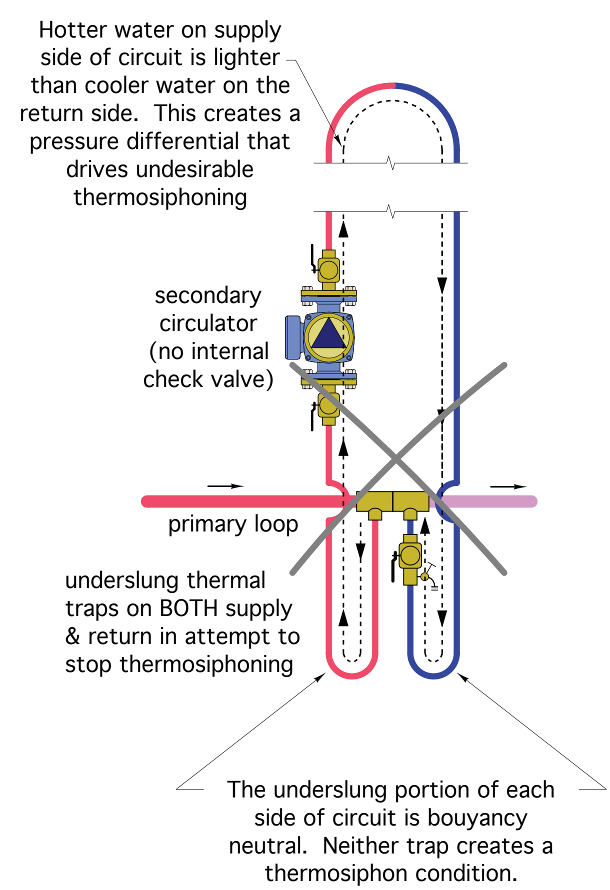
Error 11: Use of two underslung thermal traps

Discussion: Even in systems where forward thermosiphon flow is prevented using a spring-loaded check valve or flow check valve, it is possible for a slight thermosiphon flow to develop in the *return side* of a secondary circuit that is located above the primary loop. This is the result of weak 2-directional flow within the return pipe. Hot water passes upward through the “core” of the pipe, while cooler water against the pipe wall flows downward. Although this is a weak flow compared to forward thermosiphoning, it can be prevented.

One of the “classic” approaches to stopping thermosiphoning in the *return* pipe of a secondary circuit is by installing a thermal trap that is at least 12 inches deep.

One might assume that if a thermal trap is effective at preventing thermosiphoning on the *return* side of a secondary circuit, it should be equally effective on the *supply* side of the circuit, and thus eliminate the need for a spring-loaded check valve or flow check on that side of the circuit. This is not true, as illustrated in figure 7-11a. The underlying cause is that each thermal trap is buoyancy neutral. Neither trap induces a tendency for flow to develop in the secondary circuit. However, once that circuit is operating, the supply side of the circuit above

Figure 7-11a



the primary loop is filled with heated water, while the return side is filled with cooler water. The buoyancy differences between these columns of water will sustain a forward thermosiphon flow after the secondary circuit turns off. This flow will continue until the primary loop cools off, which denies the supply side of the circuit more hot water. **Solution:** *Do not use a thermal trap on the supply side of the secondary circuit.* Use a spring-loaded check valve or weighted-plug flow check, as previously discussed.

Figure 7-12a

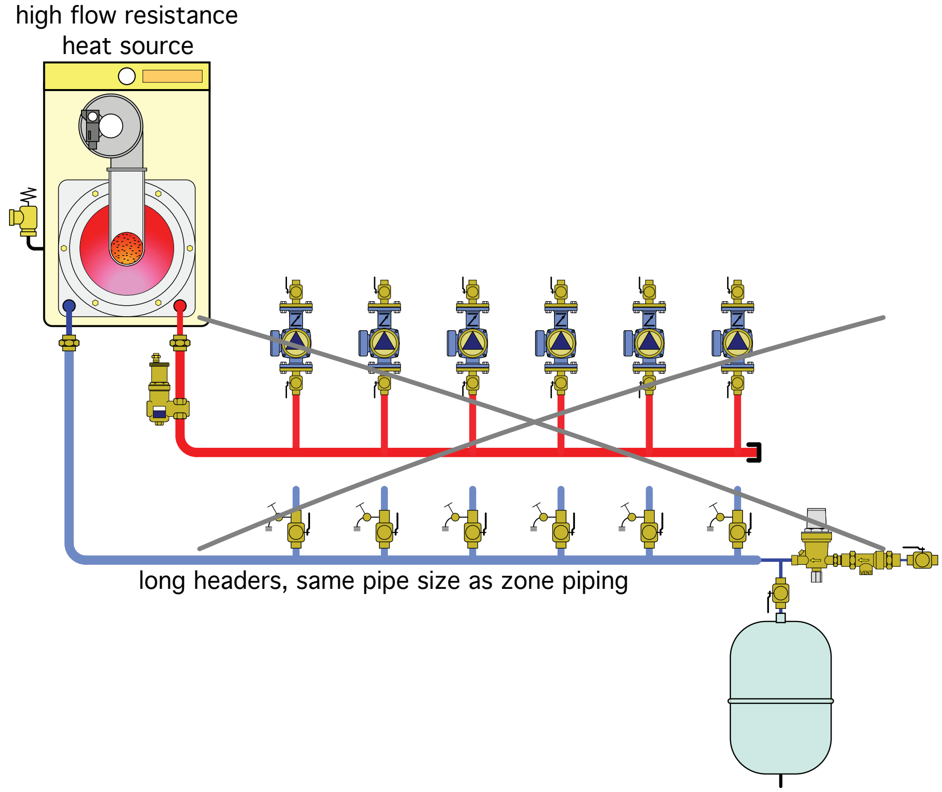
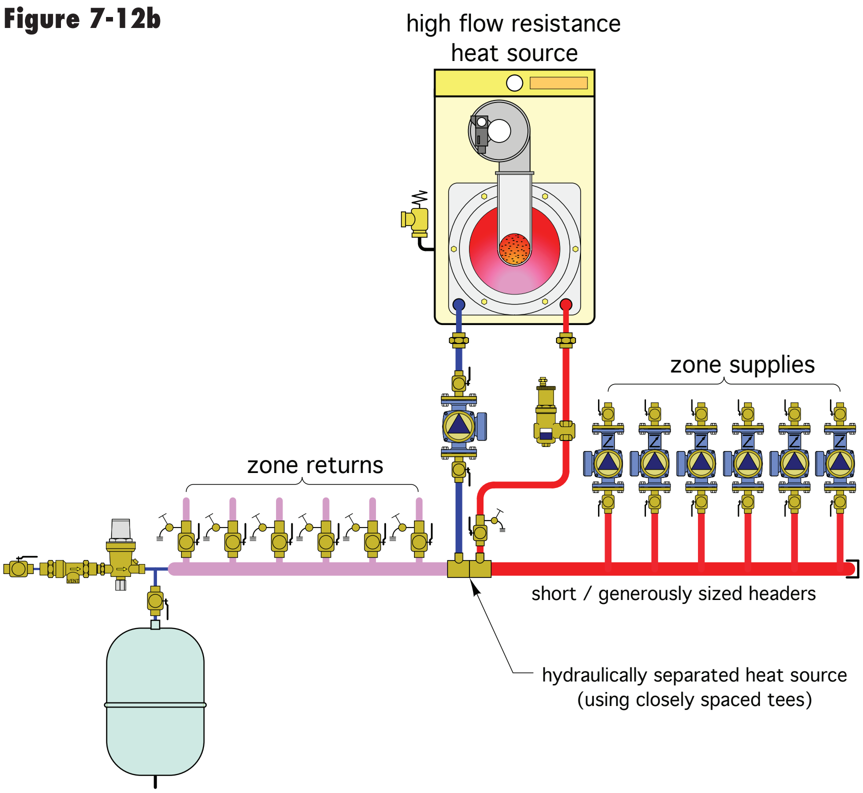


Figure 7-12b



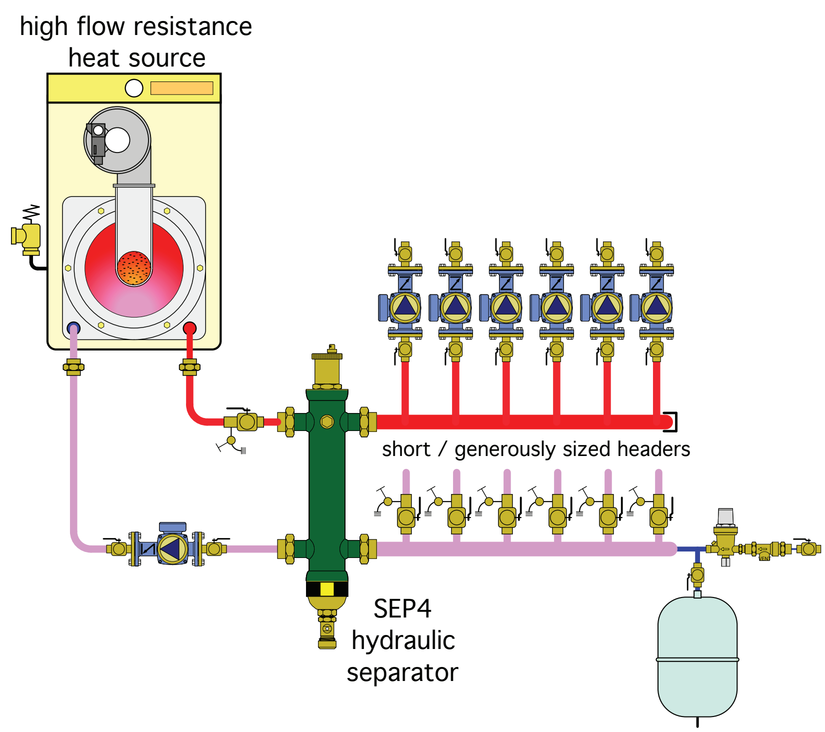
COMMON ERRORS IN MULTI-CIRCULATORS SYSTEMS

Error 12: Lack of hydraulic separation

Discussion: One common error in many multi-circulator systems is a lack of detailing to provide hydraulic separation between the circulators. This may be the result of routing the combined system flow through a high flow resistance heat source. It can also result from undersized or excessively long header piping. Although both errors can occur individually, figure 7-12a combines them into the same system.

Solution: The high flow resistance heat source should be hydraulically separated from the distribution circulators and equipped with its own circulator. The distribution headers should be sized for a flow velocity of 2 to 4 feet per second,

Figure 7-12c



and kept as short as possible. Both of these details are shown in figure 7-12b.

Another solution is to install a hydraulic separator between the high flow resistance heat source and the distribution headers, as shown in figure 7-12c. The Caleffi SEP4 hydraulic separator also eliminates the air separator shown in figure 7-12b, and provides magnetic particle and dirt separation for the system.

Error 13: Unnecessary use of reverse return piping

Another detail that, while not necessarily "wrong," remains nonessential, is shown in figure 7-13a.

Discussion: The use of reverse return header piping as a means to provide equal flow through identical devices (in this case boilers), *each with their own circulator*, is unnecessary.

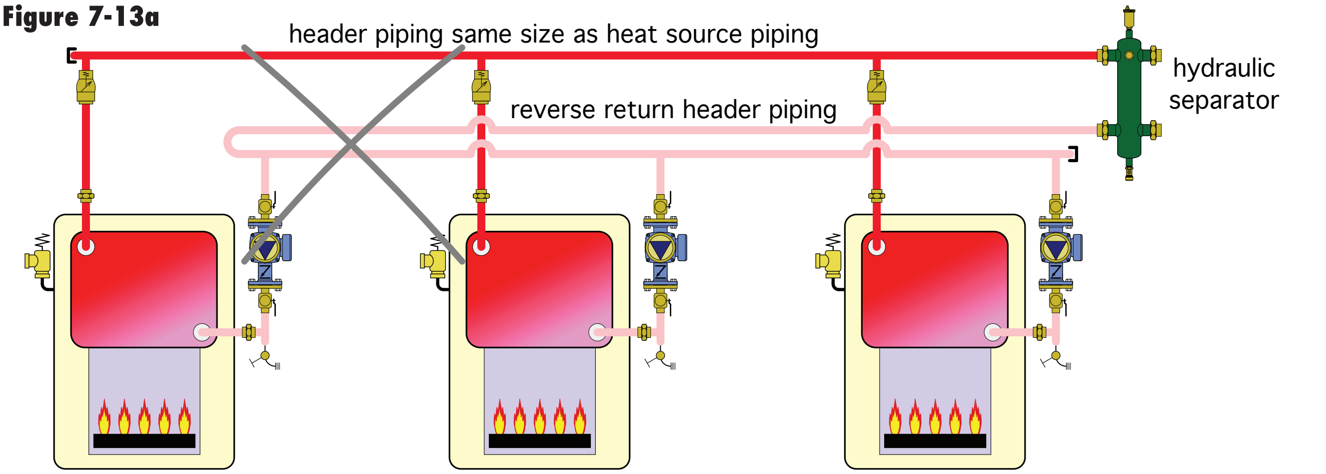
Solution: A simpler approach to providing approximately equal flows is to keep the header as short as possible, and generously sized, as shown in figure 7-13b.

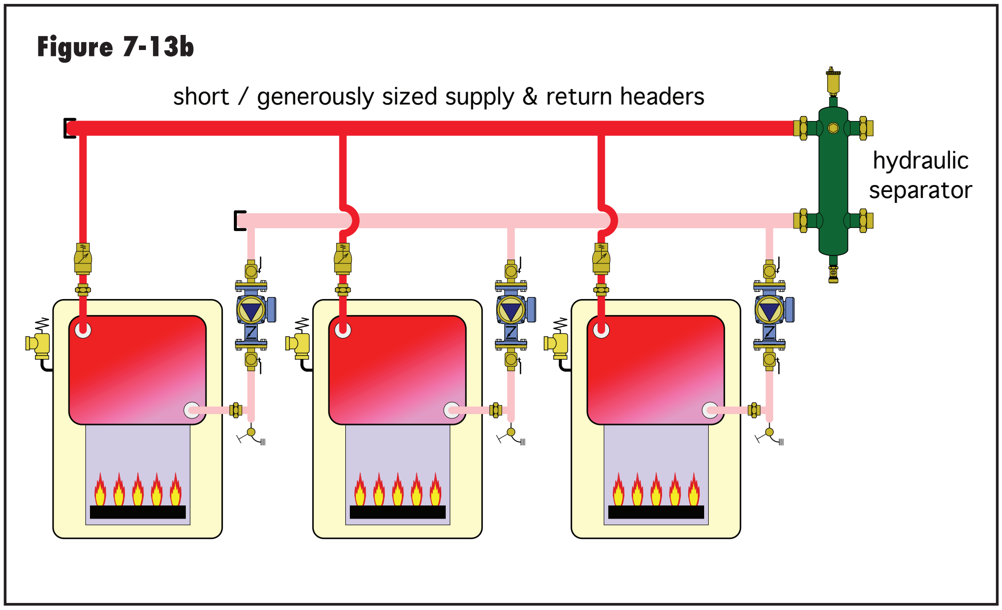
The header piping in combination with the hydraulic separator is providing the hydraulic separation between the three boiler circulators. As such, it should have a low head loss. A suggested criterion is to size the headers for a flow velocity of 2 to 4 feet per second when all circulators are operating.

In most cases, it is not essential to have *exactly* the same flow rate through identical heat sources. A slight variation in flow will only yield a slight variation in temperature rise. These variations are typically eliminated when the flows from simultaneously operating identical heat sources are combined at a hydraulic separator or other device.

If the flow rate must be identical, the same piping as shown in figure 7-13b can be used with an individual flow

Figure 7-13a





rate of heat production at the boiler equals the rate of heat dissipation from the distribution system. This occurs when the water temperature supplied to the heat emitters reaches 105ºF. There is no need for the supply water temperature to climb and higher, and thus the return water temperature will also remain at (or in some cases less than) 90ºF.

This arrangement is sure to allow sustained flue gas condensation, leading to corrosion of the boiler and vent connector piping, and damage to masonry chimneys. This system also violates the previously discussed prerequisite of having *two* mixing points when a conventional boiler is paired with a low temperature distribution system. The only mixing point in this system is the one within the 3-way thermostatic mixing valve, which controls supply water temperature.

Solution: Always provide a means of boosting boiler inlet temperature when a conventional boiler is used to supply a lower temperature distribution system. The solution must include hardware that senses and reacts to boiler inlet temperature, as well as two mixing points, one for supply temperature control and the other to

measuring/balancing valve added to each heat source circuit. All circulators can be turned on, and each flow-balancing valve adjusted until the desired flow through each heat source is attained.

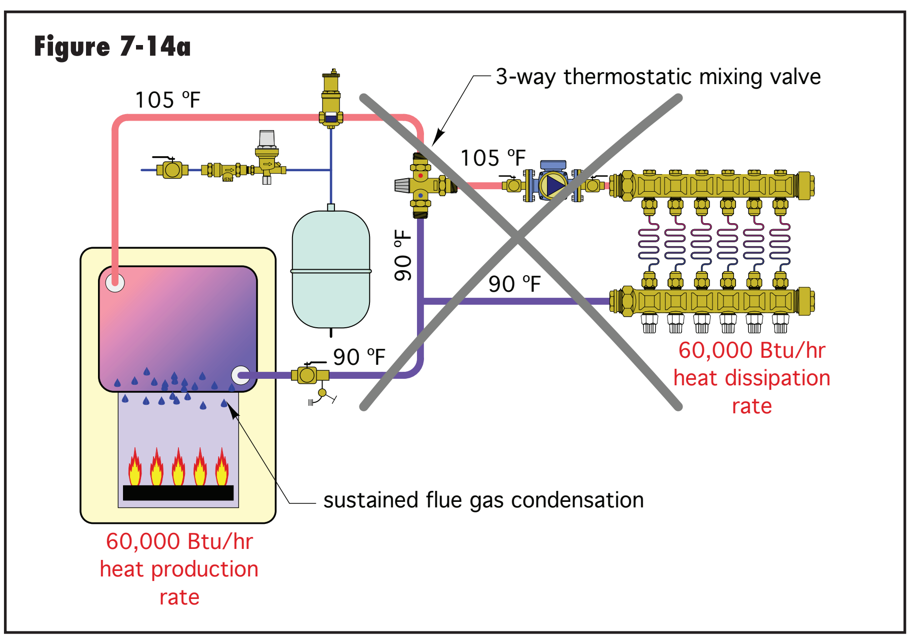
COMMON ERRORS IN MIXING APPLICATIONS

There are several errors that commonly occur when integrating mixing into hydronic distribution systems.

Error 14: Lack of boiler protection detailing

Discussion: A common error found in many systems that combine a conventional boiler with a lower temperature distribution system is not protecting a conventional boiler against sustained flue gas condensation when it is coupled to a low temperature distribution system. The schematic in figure 7-14a shows a typical (but incorrect) arrangement.

The 90ºF water returning from the low temperature heat emitters divides up between the cool port of the 3-way thermostatic mixing valve and the pipe leading back to the boiler. There is no detail that senses or reacts to boiler inlet temperature. Furthermore, the return water temperature may never increase to a nominal 130ºF, even after hours of sustained operation. That's because the low temperature distribution system has achieved thermal equilibrium with the boiler. The



boost boiler inlet temperature. It must also allow the boiler to be completely uncoupled from the distribution system when necessary to raise boiler inlet temperature as quickly as possible.

idronics #7 describes several options for providing boiler protection.

Error 15: Expecting a "bypass circulator" to provide boiler protection

Discussion: To properly protect a conventional boiler from operating with sustained flue gas condensation *under all conditions*, there must be a way to completely "uncouple" heat transfer from the heat source to the load. This is especially critical when the load has high thermal mass and operates at low water temperatures.

The system shown in figure 7-15a uses a variable-speed "bypass circulator" to monitor the inlet temperature to a pellet-fueled boiler. The load is a large thermal storage tank that, at times, may be near room temperature.

The variable-speed circulator is set to speed up when the boiler inlet temperature is below some minimum boiler inlet temperature, such as 130ºF. The intent is to drive a significant flow of supposedly hot boiler water into the tee below the variable-speed circulator. This hot water would then mix with cooler water returning from the lower portion of the thermal storage tank, to keep the boiler inlet water temperature high enough to prevent sustained flue gas condensation.

Although well intended, this method of boiler protection is flawed.

Consider a situation where the water coming back from the lower portion of the thermal storage tank is 90ºF. This is well below the dewpoint temperature of the flue gases in the boiler, so the variable-speed circulator is likely to be running at or close to full speed. The circulator carrying flow to the tank is also running. The combined flow rate of these two circulators will be passing through the boiler. Under such a condition, the temperature rise across the boiler, even at maximum boiler output, is likely to be far less than 20ºF. Thus, the anticipated "hot" water needed to bring the blended flow entering the boiler above 130ºF is simply not present. No amount of "bypass flow" can remedy this situation. Under such conditions, the boiler will operate with sustained flue gas condensation. Furthermore, this situation could last for hours, based on the volume and initial water temperature of the thermal storage tank. Fundamentally, this detail fails to provide a means to totally prevent heat flow from the boiler to the load.

Figure 7-15a

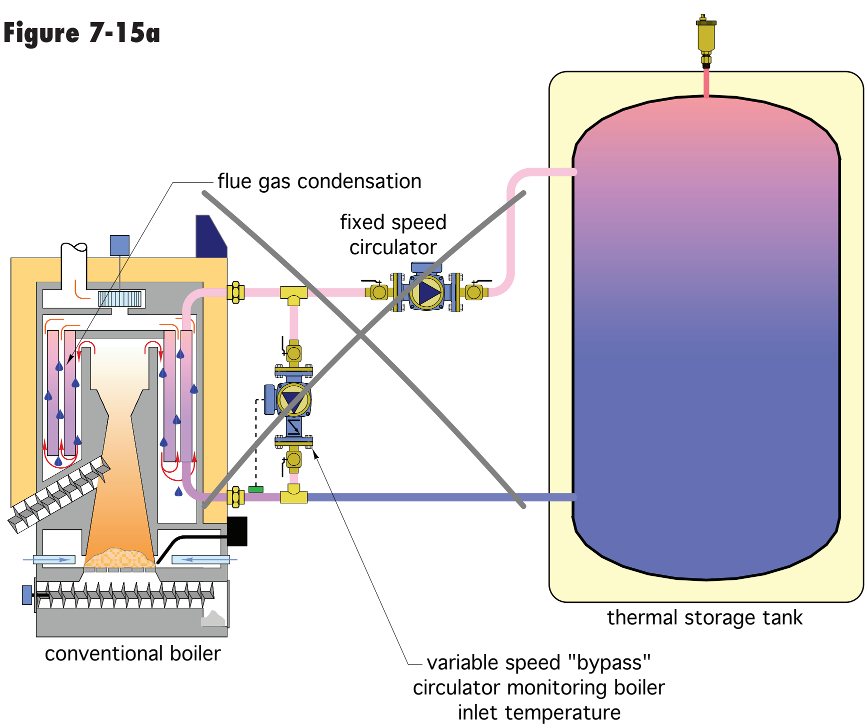


Figure 7-15b shows another scenario in which a high thermal mass heated floor slab is supplied through a 3-way thermostatic mixing valve, and boiler protection is intended to be provided by the variable-speed bypass circulator.

Although there is mixing to control the *supply* water temperature, the inability of this arrangement to totally stop heat flow between the boiler and load when necessary removes its ability to properly protect the boiler from sustained flue gas condensation.

Solution: Figure 7-15c shows how the system of figure 7-15a could be modified to provide proper boiler protection.

Figure 7-15b

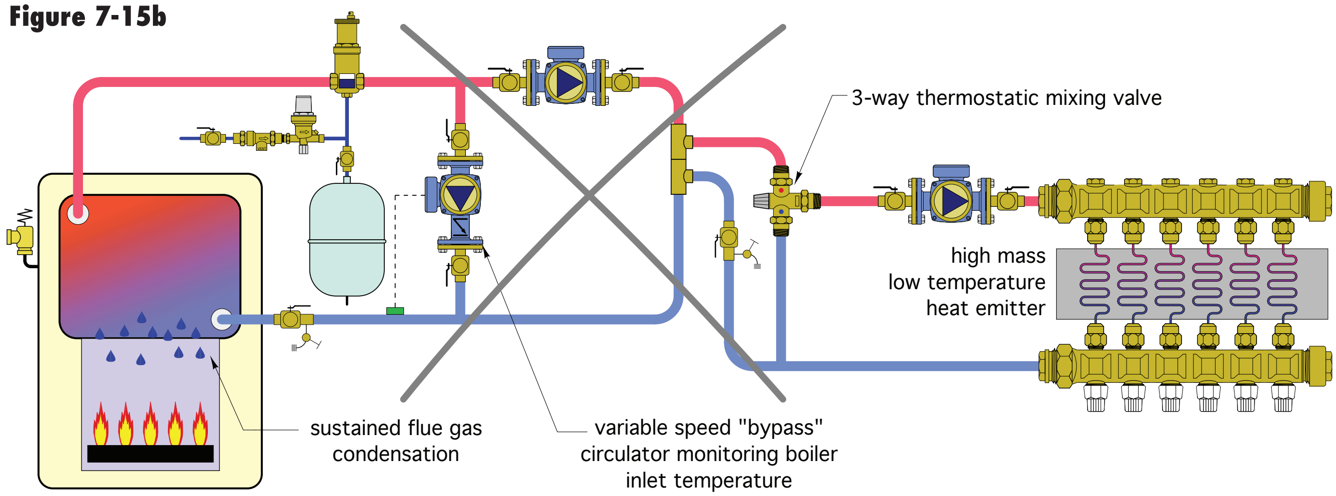
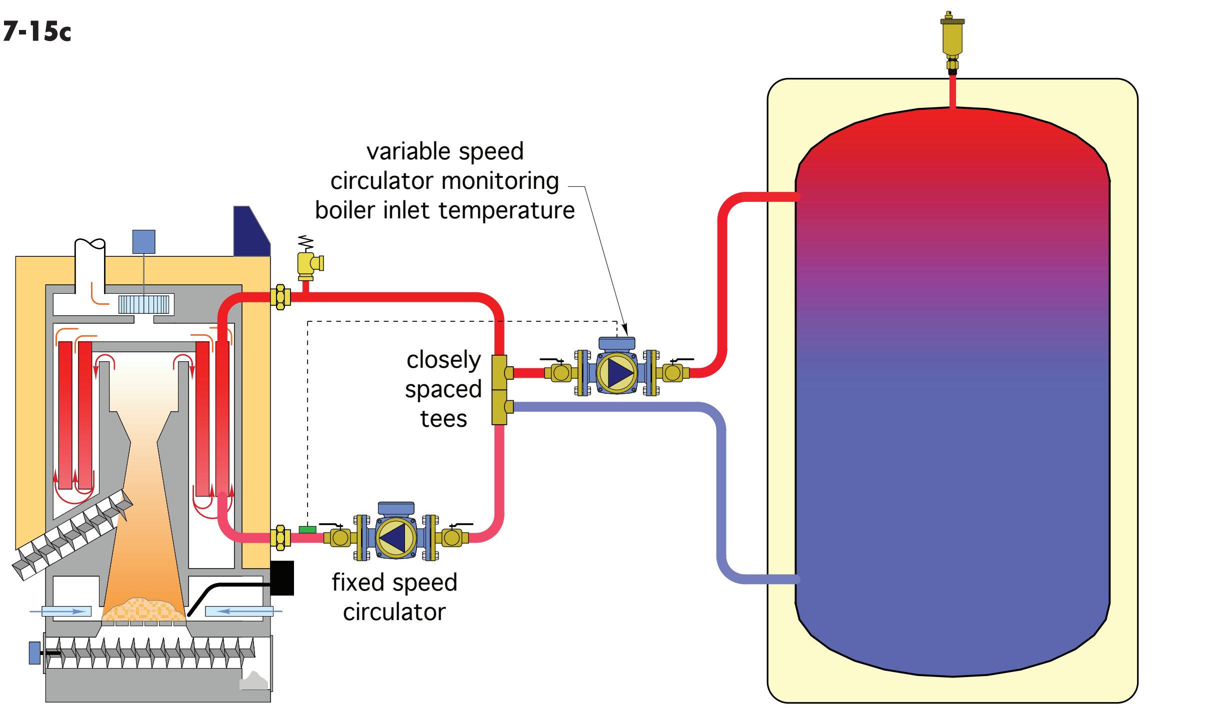
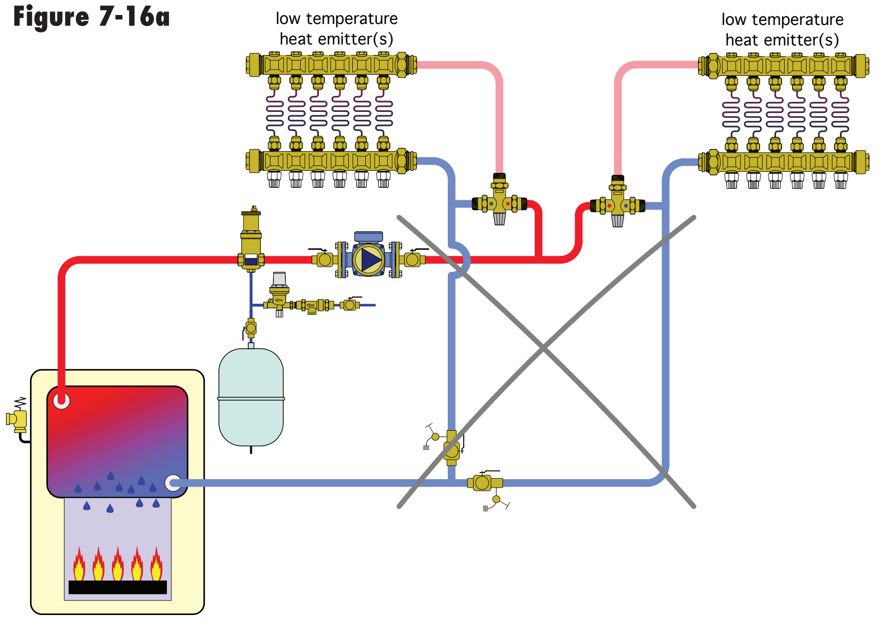


Figure 7-15c



The variable-speed setpoint circulator is now located where it can completely stop any heat transfer between the boiler and thermal storage tank. The closely spaced tees provide hydraulic separation between the fixed-speed boiler circulator and the variable-speed boiler protection circulator. The variable-speed circulator remains off until the boiler inlet temperature climbs to some minimum value (typically 130ºF) that can prevent sustained flue gas condensation. The variable-speed circulator then increases speed, but continues to monitor boiler inlet temperature. If that temperature decreases, the variable-speed circulator decreases speed to reduce the rate of heat transfer from the boiler to the thermal storage tank. If the boiler inlet temperature continues to rise, so does the speed of the variable-speed circulator. The variable-speed circulator typically reaches full speed when the boiler inlet temperature is above the minimum boiler inlet temperature setting by 5 to 10ºF.

Figure 7-16a

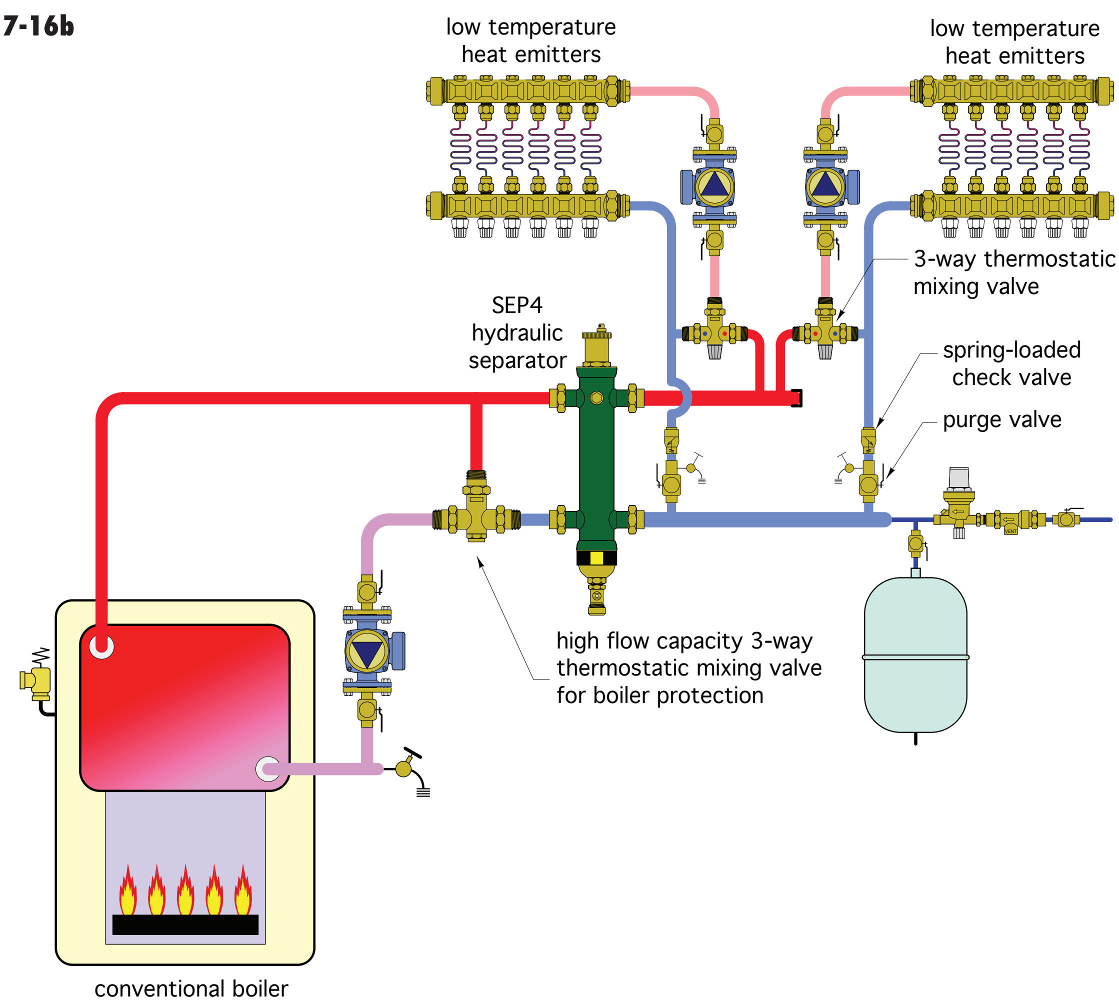


Error 16: Incorrect placement of circulator relative to mixing valve

Discussion: There must be a circulator between a mixing valve and the heat emitters served by that mixing valve. The schematic shown in figure 7-16a fails to provide the required circulators.

Water discharged from the circulator shown in figure 7-16a will favor the path of least resistance when returning to the circulator's inlet. This will not allow proper mixing within the 3-way thermostatic valves, and therefore will not deliver heat at the required rate or supply water temperature to the low temperature heat emitters. The schematic in figure 7-16a also fails to

Figure 7-16b



provide boiler protection, and thus allows sustained flue gas condensation within the boiler.

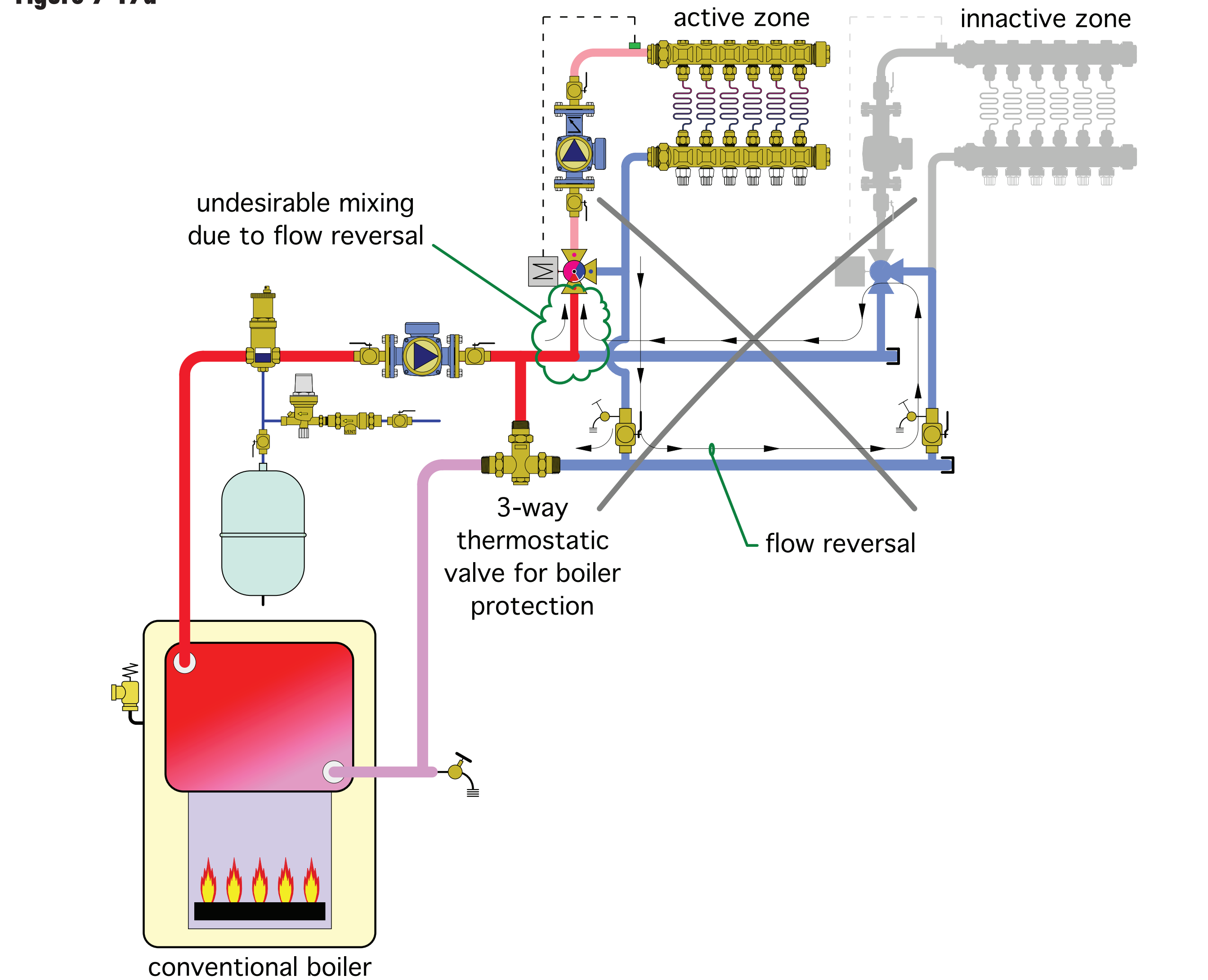
Solution: The system shown in figure 7-16b places a circulator downstream of each mixing valve. It also includes a high flow capacity 3-way thermostatic valve for boiler protection.

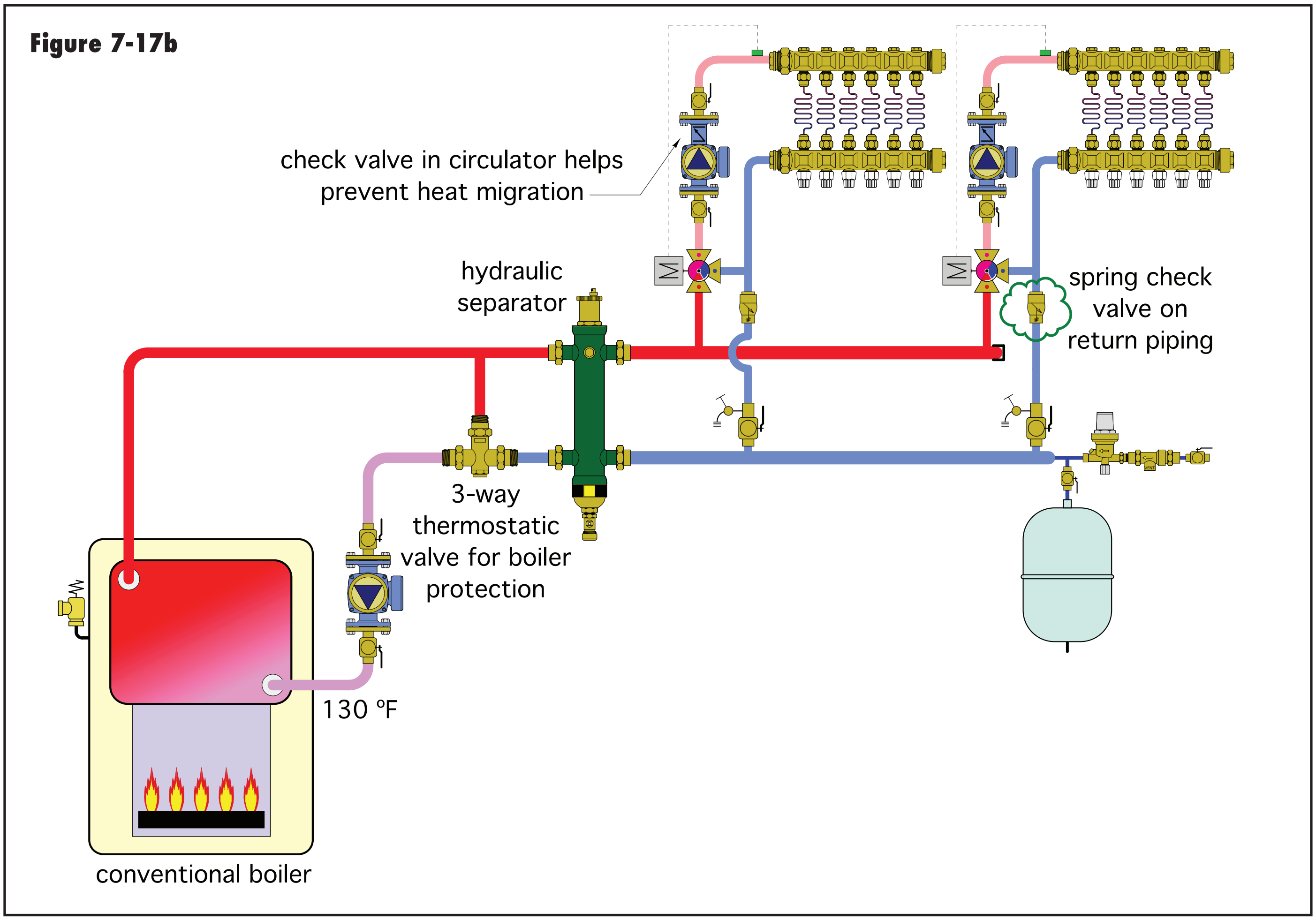
Error 17: Lack of hydraulic separation, and possible flow reversal

Discussion: The system shown in figure 7-17a embodies two significant errors.

The first error is a lack of hydraulic separation between the boiler loop circulator and the two zone circulators. When the boiler has reached a stable operating temperature above the dewpoint of the flue gases, the hot port of the boiler protection mixing valve will be closed. Under this condition, the boiler loop circulator will create a significant pressure differential between the supply and return headers to which the zone circuits are connected. This pressure differential is imposed across both zone circuits and will push flow into both circuits. If one of the zones is off, the spool within the 3-way motorized mixing valve will remain in the last position it had when operating. This could allow a flow path through the 3-way mixing valve from the hot inlet port to the mixed outlet port. The imposed pressure differential will allow hot (unmixed) water to flow into the inactive zone, leading to possible space overheating, as well as a possible over temperature condition in the low temperature heat

Figure 7-17a





emitters (i.e., imagine 170ºF water being pushed into a radiant floor zone).

The other problem with the system in figure 7-17a is lack of check valves that can prevent any reverse flow from an active zone through an inactive zone. A 3-way rotary valve that is not operating may present a flow path from the cool inlet port to the hot inlet port, and thus allow flow reversal through the valve. To stop this, a check valve must be located between the header connections and the mixing valve. *A spring-check valve within the circulator will not be able to stop possible flow from the cool port to the hot port of the mixing valve.* Some 3-way thermostatic valves contain internal check valve cartridges that could prevent such flow reversal, but 3-way rotary mixing valves do not contain any check valve cartridges.

Solution: One solution that solves both problems is shown in figure 7-17b.

This system uses a hydraulic separator to isolate the pressure dynamics of the boiler loop circulator from that of either zone circulator. The Caleffi SEP4 hydraulic separator also provides central air and dirt separation, as well as the ability to capture ferrous metal particles within the system.

A spring-check valve has been installed on the return side of each zone circuit between the mixing valve and return header. This check valve prevents flow reversal through either zone when it is inactive and the other zone is active.

When spring-check valves are located on the return side of the zone circuits, as shown in figure 7-17b, it is still good practice to use zone circulators with internal spring-checks. These internal valves provide a forward opening pressure requirement of 0.3 to 0.5 psi, which is generally sufficient to prevent heat migration into the supply side of the zone circuit.

8. EXAMPLE SYSTEMS

Having discussed many aspects of distribution system design, it's time to combine several details into systems. This section will present several such systems.

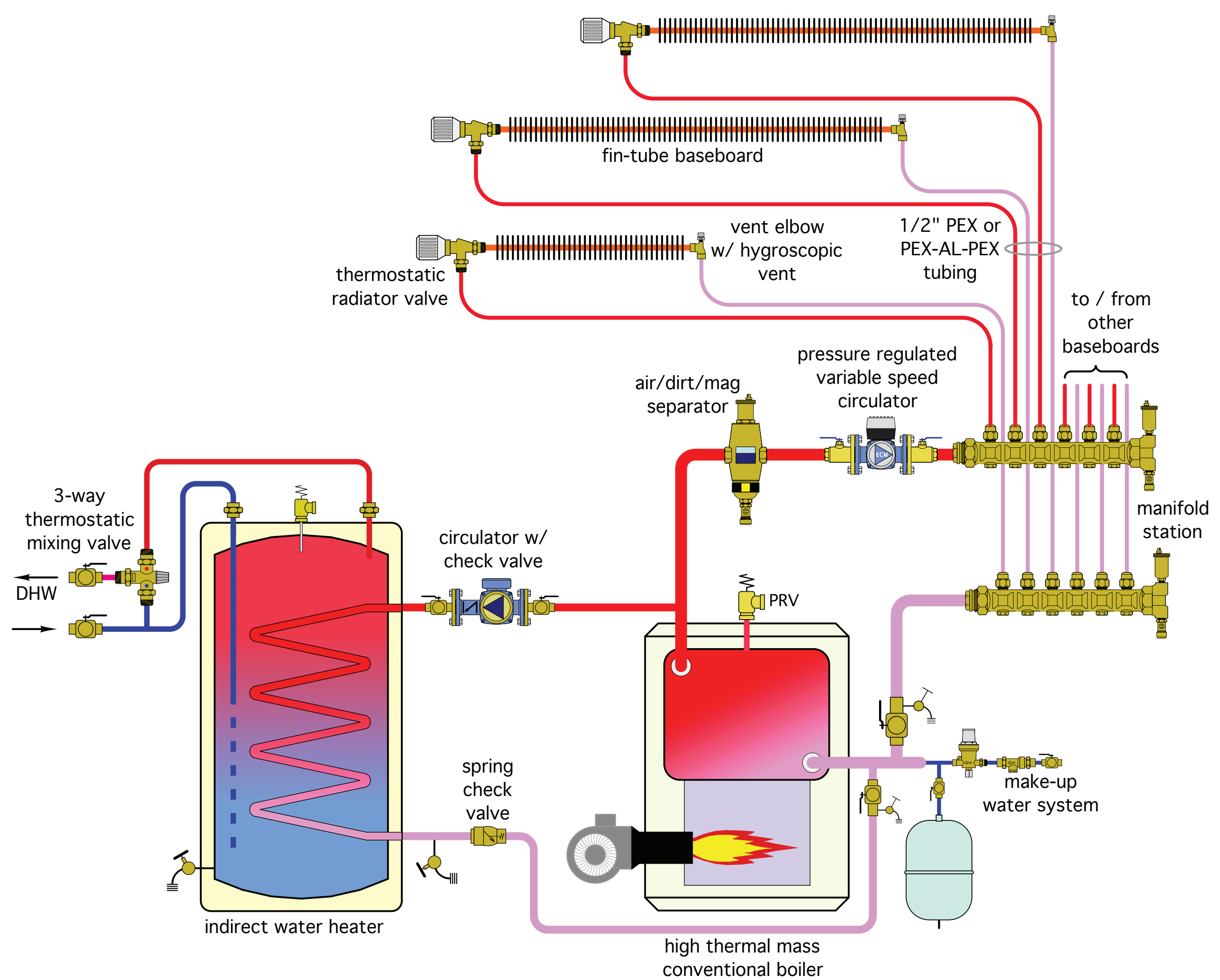
System #1: Baseboard system using manifold distribution + indirect domestic water heating

Fin-tube baseboard is one of the most common hydronic heat emitters used in North American hydronic systems. Traditionally, multiple fin-tube baseboards have been connected into a series circuit using rigid copper piping. In smaller houses, a single series loop has often been used. In larger houses, there may be two or more series loops based on zoning.

Although this traditional approach works, it carries the previously discussed limitations associated with series loop systems (e.g., temperature drop from one baseboard to the next, higher head loss, lack of room-by-room zoning). This approach was also developed when rigid copper tubing was typically the only practical piping material available for such systems.

Today there are hardware selections available that can significantly improve both the installation and performance of traditional fin-tube baseboard heat emitters. The system shown in figure 8-1 is one example. It retains fin-tube baseboard as the heat emitters, but changes the piping arrangement from series circuits to a homerun system.

Figure 8-1



Each baseboard is served by supply and return tubing made of flexible 1/2-inch PEX or PEX-AL-PEX tubing. This tubing is much easier and faster to install compared to rigid copper tubing, especially in retrofit applications.

All baseboard circuits are now in parallel and connected to a central manifold. Each baseboard will receive water at the same temperature. Parallel piping also allows the flow rate to each baseboard to be independently controlled. The use of thermostatic radiator valves on the supply side of each baseboard allows for room-by-room zoning.

The variable-speed pressure-regulated circulator is set for constant differential pressure mode and adjusted to provide sufficient differential so that design flow rates can be supplied to all six baseboards operating simultaneously. This circulator will automatically reduce its speed (and its electrical energy usage) as the thermostatic valves modulate to reduce or stop flow through each distribution circuit. The circulator is turned on whenever the temperature at a master thermostat drops below its setpoint. If none of the thermostatic radiator valves are open, the circulator drops into a "sleep" mode where power consumption is less than 10 watts. As a radiator valve begins to open, the circulator automatically senses the attempted drop in differential pressure and responds by increasing its speed.

A magnetic air/dirt separator is installed upstream of the variable-speed circulator. Its coalescing media will strip out and eject microbubbles from the system. The lower portion of the separator will capture both ferrous and non-ferrous particles before they can enter the circulator.

The indirect water heater is connected as a separate zone with its own circulator. Domestic water heating is controlled as a priority load. During a call for domestic water heating, the circulator supplying the manifold station is temporarily turned off. This allows the full heat output of the boiler to be directed to domestic water heating, ensuring the fastest possible recovery to the tank's setpoint temperature. A 3-way thermostatic valve ensures that domestic water leaving the tank does not exceed a safe delivery temperature.

This system uses an oil-fired, cast iron, sectional boiler. Such boilers generally have sufficient thermal mass to stabilize heat delivery to a zoned distribution system, and thus a buffer tank is not needed.

System #2: Multiple water temperature system supplying panel radiators and radiant panels, in combination with "on-demand" domestic water heating.

The versatility of hydronics makes it easy to build a distribution system that requires a range of water temperatures to supply different types of heat emitters. The system shown in figure 8-2 allows for three simultaneous supply water temperatures. They range from a low of 95ºF for the basement floor heating circuits, to 120ºF for the panel radiators. Another area of radiant panel heating requires a supply water temperature of 110ºF.

All heat for the system comes from a low thermal mass modulating/condensing boiler.

A Caleffi ThermoCon buffer tank is used to protect this boiler from short cycling when minimal loads are present on the extensively zoned distribution system. The boiler fires to maintain the temperature at the mid-height sensor location between 120 and 140ºF. This allows the mod/con boiler to operate in condensing mode most of the time. The buffer tank has been sized to allow the boiler to operate for a minimum of 10 minutes when there is no concurrent load on the system.

The buffer tank is piped in a "2-pipe" configuration. This reduces the flow velocity of heated water entering the upper sidewall connection of the tank when the distribution system is operating at the same time as the boiler. Reduced inlet flow velocity encourages better temperature stratification within the tank. The short and generously sized headers at points A and B also allow the tank to provide hydraulic separation between the boiler circulator (P1) and the variable-speed distribution circulator (P2).

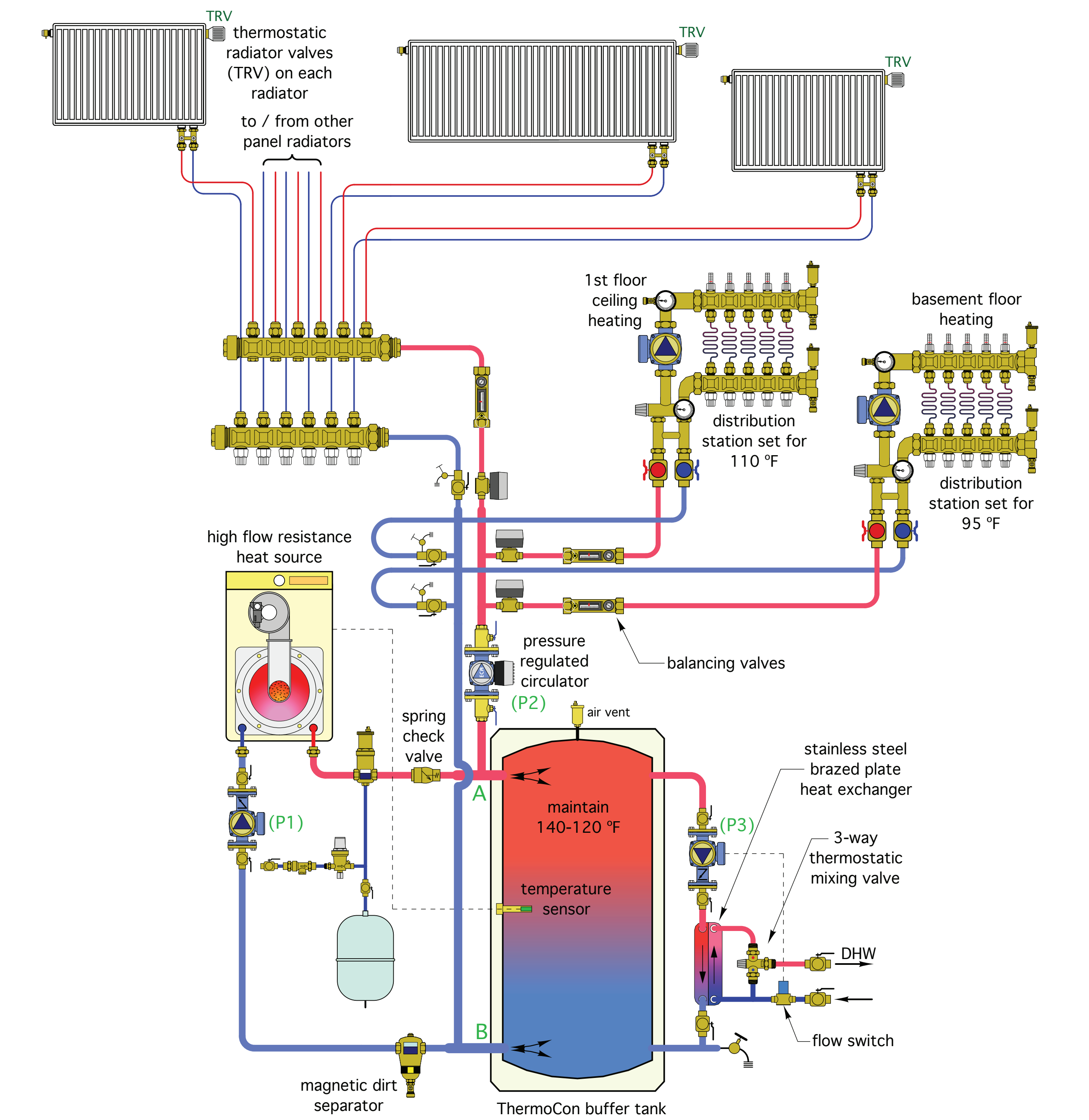
A spring-loaded check valve near the upper tank header, as well as an internal spring-loaded check valve in the boiler circulator, prevent hot water in the tank from reverse thermosiphoning through the boiler circuit when the boiler is off.

A single pressure-regulated variable-speed circulator (P2) supplies flow to two Caleffi distribution stations, each set at an appropriate supply temperature for the radiant panel circuits they serve. This circulator also provides flow to a manifold station that supplies 4 panel radiators, which are piped as a homerun subsystem.

Zone valves open and close to determine which of the three load circuits receives hot water. The bypass built into each distribution station provides hydraulic separation between the variable-speed distribution circulator and the fixed-speed circulator in each distribution system. Each of the three load circuits is equipped with a flow-balancing valve. This allows the proper flow proportions to be set, assuming all three loads are operating simultaneously under design load conditions. The variable-speed pressure regulated circulator (P2) ramps its speed up or down as necessary to maintain approximately constant differential pressure between the supply and return headers.

Domestic water is heated "on demand" by the stainless steel heat exchanger seen to the right of the ThermoCon tank. Whenever there is a draw of hot water, a flow-activated switch turns on circulator (P3) to send hot

Figure 8-2

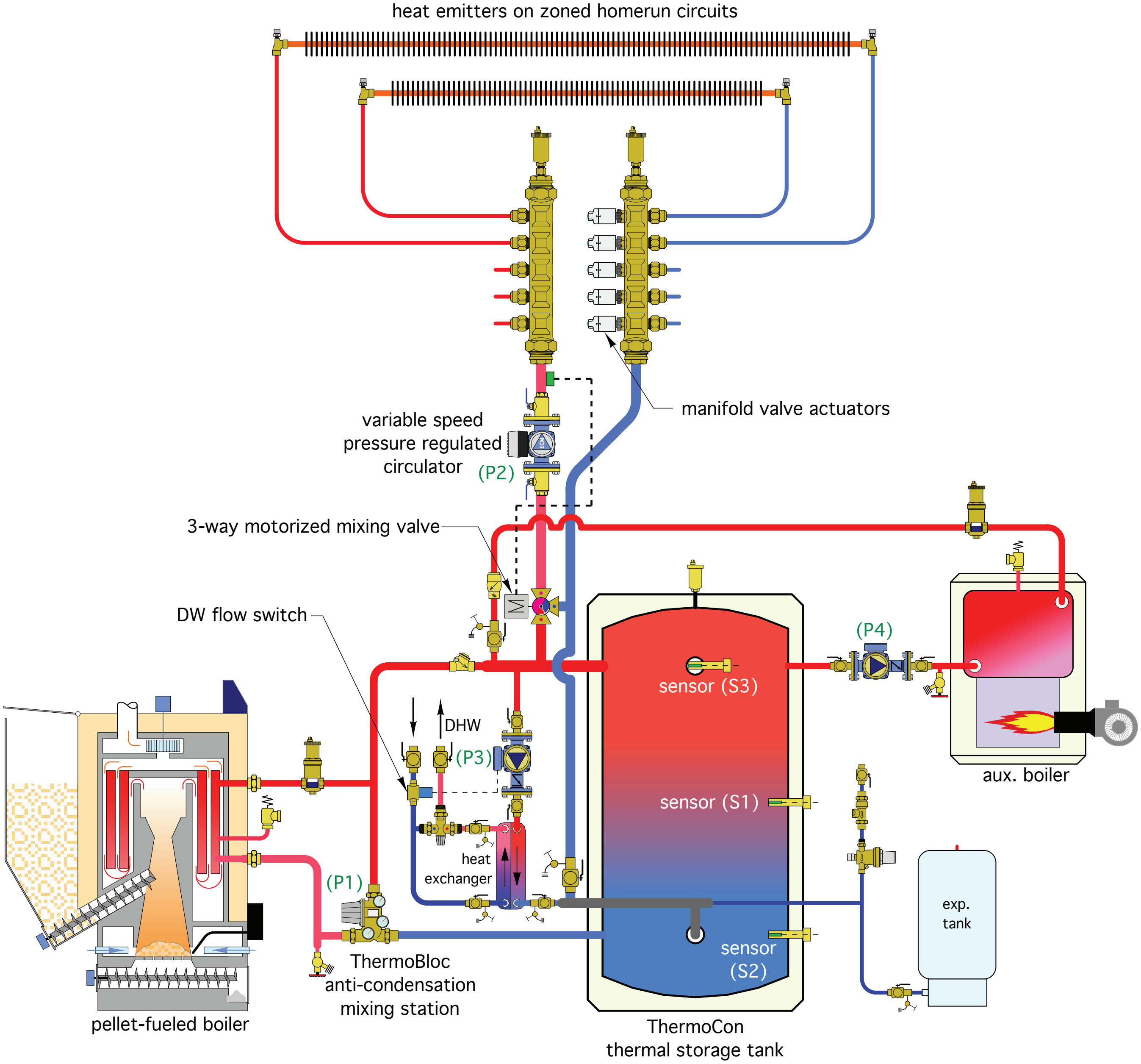


water from the top of the ThermoCon tank through the primary side of the heat exchanger. Domestic cold water passes through the other side of the heat exchanger and absorbs heat. A 3-way thermostatic mixing valve provides anti-scald protection for the domestic hot water delivery. This configuration leverages the thermal mass of the ThermoCon tank to stabilize domestic hot water production, and eliminates the need for an indirect water heater. Because the top of the tank is always at or above 120ºF, domestic water is always available.

System #3: Space heat and domestic hot water supplied from a pellet-fueled boiler with an auxiliary boiler

The system shown in figure 8-3 uses a pellet-fueled boiler as the primary heating source, and an oil-fired boiler as the auxiliary heat source. For good thermal efficiency and low emissions, the pellet boiler is sized to 75% of the building's design heating loss. This allows it to provide about 96% of the total seasonal space-heating energy needs, and helps prevent short cycling under partial loads.

Figure 8-3



The auxiliary boiler fires only when the pellet-fueled boiler is unable to keep pace with the total load on the system.

A Caleffi ThermoCon tank serves as the central thermal mass component of the system. It can accept heat from either boiler and deliver heat to both space-heating and domestic water-heating loads. The multiple connection ports on this tank allow for both piping connections and sensor placements that coordinate with the overall system control scheme.

The pellet-fueled boiler is turned on when storage sensor (S1) drops to 130ºF. It remains on until the lower tank sensor (S2) reaches 175ºF. This operating logic allows for long burn cycles and good thermal performance. The auxiliary boiler is fired only if sensor (S3) in the upper portion of the ThermoCon tank drops to or below 130ºF. Once fired, the auxiliary boiler remains on until the upper tank sensor (S3) reaches 140ºF. These temperatures are high enough that anti-condensation protection is not needed by the auxiliary boiler.

The pellet-fueled boiler is protected against sustained flue gas condensation by the ThermoBloc mixing station. This mixing station also creates flow between the pellet-fueled boiler and the header on the left side of the ThermoCon tank. If a power failure occurs, the ThermoBloc allows passive thermosiphon flow between the pellet-fueled boiler and ThermoCon tank. This safely dissipates residual heat from the boiler into the tank.

The ThermoCon tank is set up in a “2-pipe” configuration with short 2” pipe size headers at the upper and lower left side connections. This configuration allows the tank to absorb any surplus heat created by either heat source that is not being used by the loads. It also allows the tank to provide hydraulic separation between all the circulators.

Space heating is provided by multiple homerun circuits of 1/2” PEX or PEX-AL-PEX tubing routed from the manifold station to each of several fin-tube baseboard heat emitters. These emitters have been sized to deliver design load output at a supply water temperature of 130ºF. Flow to each heat emitter is controlled by manifold valve actuators on the return manifold.

A 3-way motorized mixing valve controls the water temperature suppled to the heat emitters. This valve is regulated based on outdoor reset control. It reduces what could be relatively high temperature (180+ºF) water from the thermal storage tank to a value that allows the heat emitters to supply the current heating load of the building.

Domestic water is heated “on demand” by the stainless steel heat exchanger seen to the left of the ThermoCon tank. Whenever there is a draw of hot water, the flow switch turns on circulator (P3) to send hot water from the top of the ThermoCon tank through the primary side of the heat exchanger. Domestic cold water passes through the other side of the heat exchanger and absorbs heat. A 3-way thermostatic mixing valve provides anti-scald protection for the domestic hot water delivery. This configuration leverages the thermal mass of the ThermoCon tank to stabilized domestic hot water production, and eliminates the need for an indirect water heater. Because the top of the tank is always at or above 130 ºF, domestic water can always be produced. If the pellet-fueled boiler is not operating, the upper portion of the ThermoCon tank is maintained at a minimum acceptable temper by the auxiliary boiler. This configuration also provides buffering for the auxiliary boiler to prevent short cycling.

System #4: Space Heating, cooling & DHW from air-to-water heat pump

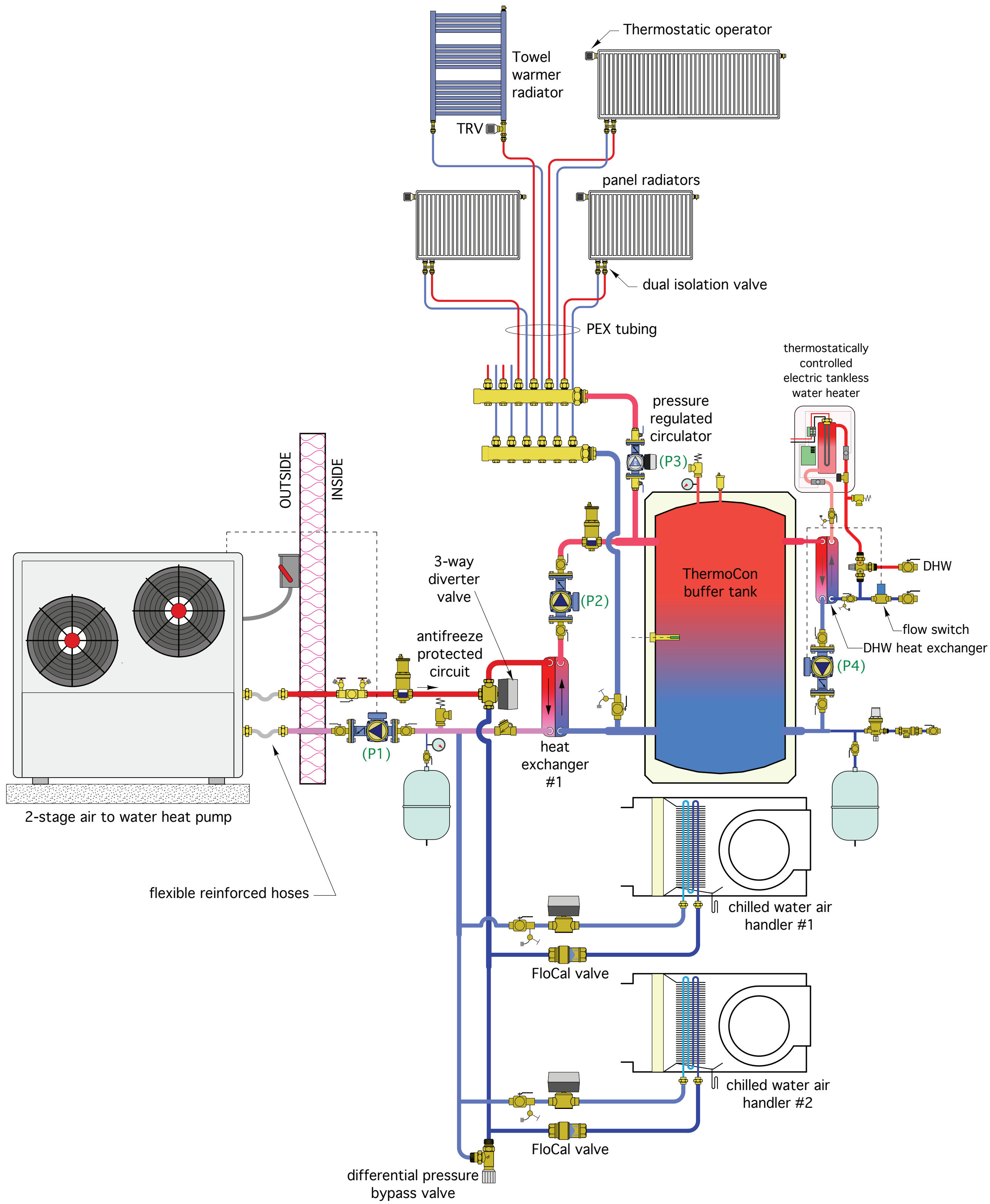
Most hydronic heat pumps can provide heated water for space heating and domestic hot water production, as well as chilled water for cooling. The system shown in figure 8-4 takes advantage of this ability to provide room by room zoned heating, two zones of cooling, and domestic water heating.

This system uses a 2-stage air-to-water heat pump. Heating or cooling capacity can be switched between 100 % of maximum rated output and 50% of that output. This is particularly important for the zoned cooling distribution system. If only only one air handler is operating, the cooling capacity of the heat pump can be reduced to prevent short cycling.

The heat pump is connected to an antifreeze protected circuit. In heating mode, heated antifreeze flows through a 3-way diverter valve to heat exchanger #1, where heat is transferred to water. This heat exchanger has been sized so that the approach temperature difference (e.g., the temperature difference between the incoming antifreeze solution and the leaving water) is only 5 ºF under full design heat transfer rate. Heated water is then routed to a generously sized header at the top left connection of the ThermoCon buffer tank. If one or more of the panel radiators is active, some of the heated water is routed to the load. The remainder goes into the buffer tank.

A variable speed pressure-regulated circulator, set for constant differential pressure mode, automatically adjusts speed based on the flow demands of the homerun distribution subsystem serving the panel radiators. Each

Figure 8-4


Towel warmer radiator
TRV
Thermostatic operator
panel radiators
dual isolation valve
PEX tubing
pressure regulated circulator
(P3)
thermostatically controlled electric tankless water heater
OUTSIDE
INSIDE
3-way diverter valve
(P2)
ThermoCon buffer tank
DHW
flow switch
DHW heat exchanger
(P4)
antifreeze protected circuit
heat exchanger #1
(P1)
2-stage air to water heat pump
flexible reinforced hoses
chilled water air handler #1
FloCal valve
chilled water air handler #2
FloCal valve
differential pressure bypass valve

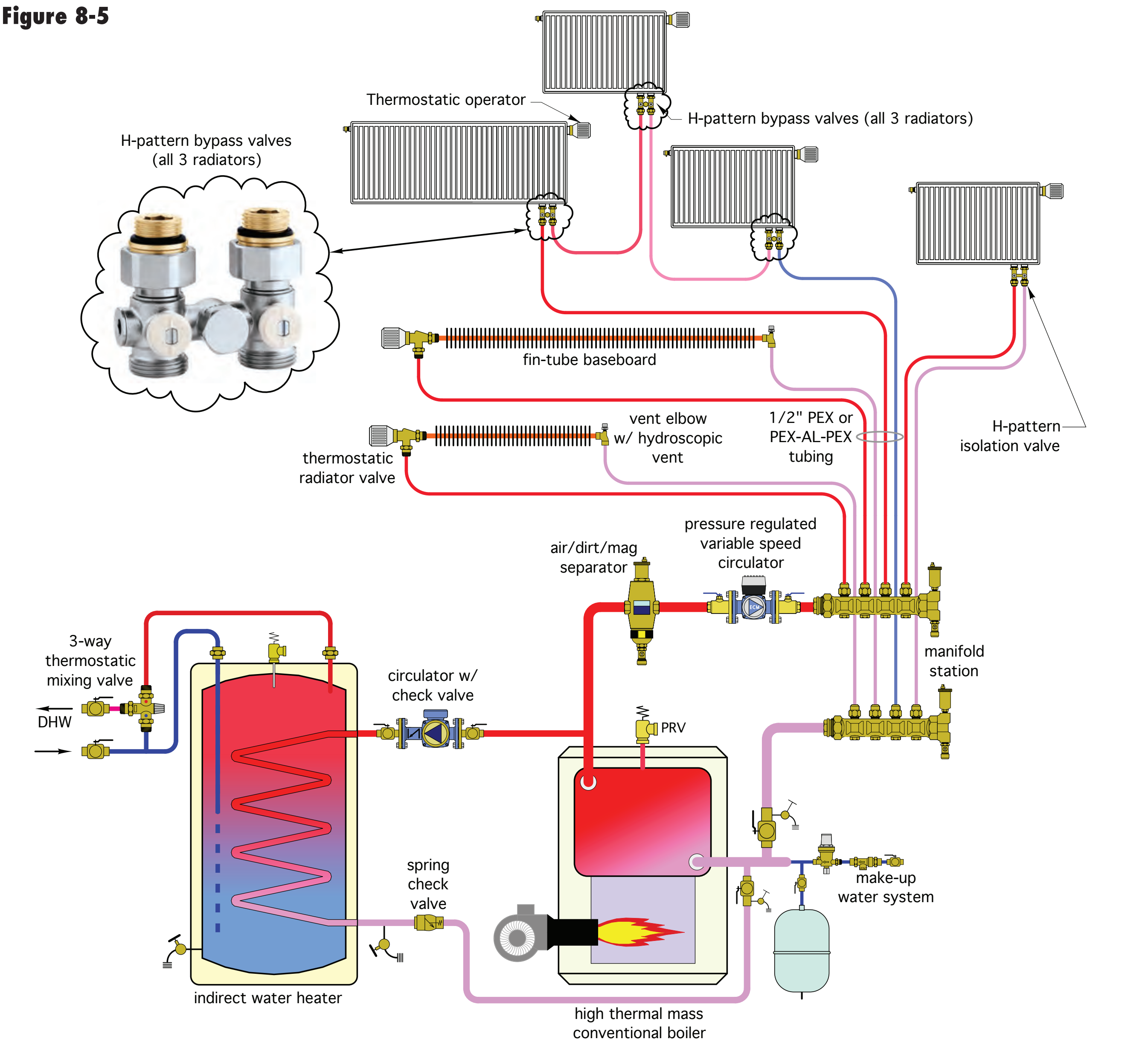
radiator is equipped with a thermostatic radiator valve to allow room-by-room comfort control. Each radiator also has a dual isolation valve at its lower connections.

During heating mode operation, the temperature of the thermal storage tank is regulated based on outdoor reset control. Under design load conditions, the tank is heated to 125 °F at the mid-height sensor. Under partial load conditions the temperature to which the tank is heated decreases with increasing outdoor temperature. The minimum tank temperature is set to 90 °F. Reducing the tank temperature under partial load conditions allows the heat pump to achieve higher COPs and higher heating capacities.

The cooling subsystem is a 2-pipe direct return configuration. It connects "upstream" of heat exchanger #1. When the 3-way diverter valve is energized, all incoming chilled antifreeze is routed to the active air handler(s). The heat exchanger is not involved in cooling mode operation. This eliminates any thermal penalty, allowing for optimal cooling performance.

Flow through each air handler is controlled by a zone valve. Circulator (P1), which drives flow through the heat pump, also drives flow through the air handlers. A differential pressure bypass valve is installed at the end of the 2-pipe distribution system to prevent over-pressure when only one

Figure 8-5



air handler is operating. A Caleffi FloCal valve is installed in the piping to each air handler. This valve ensures stable flow through each air handler regardless of the status of the other air handler. During cooling mode operation, the temperature of the antifreeze solution delivered to the air handlers varies between 40 and 60 ºF.

Domestic water is heated "on demand" by the stainless steel heat exchanger connected to the ThermoCon tank. Whenever there is a draw of hot water, a flow switch turns on circulator (P4) to send hot water from the top of the ThermoCon tank through the primary side of the heat exchanger. Domestic cold water passes through the other side of the heat exchanger and absorbs heat. The temperature of the domestic water leaving the heat exchanger depends on the temperature of the water in the upper portion of the buffer tank. In this system that temperature varies from a high of 125 ºF under design space heating load conditions, to a minimum of 90 ºF under partial space heating load. This implies that there will be times when the domestic water leaving the heat exchanger is only partially heated. Therefore, a thermostatically controlled electric tankless water heater is provided to "boost" the domestic water temperature to the value deemed necessary. A 3-way thermostatic mixing valve provides anti-scald protection for the domestic hot water delivery. This configuration leverages the thermal mass of the ThermoCon tank to stabilize domestic hot water production, and eliminates the need for a separate water heating tank.

The system's controls are configured to allow the buffer tank to be heated, as a priority load, whenever the system is set for cooling mode operation, and the tank temperature drops to 120 ºF or less. This ensures that most of the heat delivered for domestic water heating will be sourced from the heat pump rather than from the tankless electric heater. If the tank drops to 120 ºF or less, any active cooling load is temporarily suspended while the tank temperature is increased to 130 ºF, after which cooling resumes.

System #5: Space Heating and DHW using mixed heat emitters

The system shown in figure 8-5 is a modification of the system shown in figure 8-1.

This system combines fin-tube convectors and panel radiators within a modified homerun system. The two fin-tube baseboards are individually supplied from the manifold station. Each is equipped with a thermostatic radiator valve to allow individual room temperature control. Likewise, the panel radiator at the far right of the system is connected using a homerun distribution circuit and is regulated by a thermostatic radiator valve.

The three panel radiators at the top of the schematic are located in a remote area of the building, perhaps 75 feet away from the manifold station. They appear to be connected in a series circuit. However, each is equipped with an H-pattern bypass valve. This allows flow through each panel to be individually controlled. This series bypass arrangement was selected to reduce the amount of tubing that would otherwise be required to provide individual homerun circuits to each of these panel radiators.

Domestic water is heated by an indirect water heater operated as a priority load.

SUMMARY

This issue of *idronics* has examined a wide range of proven piping layouts for hydronic heating and cooling systems. These systems have been built around the basic concepts discussed in Section 2. The strengths and limitations of these distribution systems have been discussed. Section 7 has also presented several common errors to avoid when designing distribution systems.

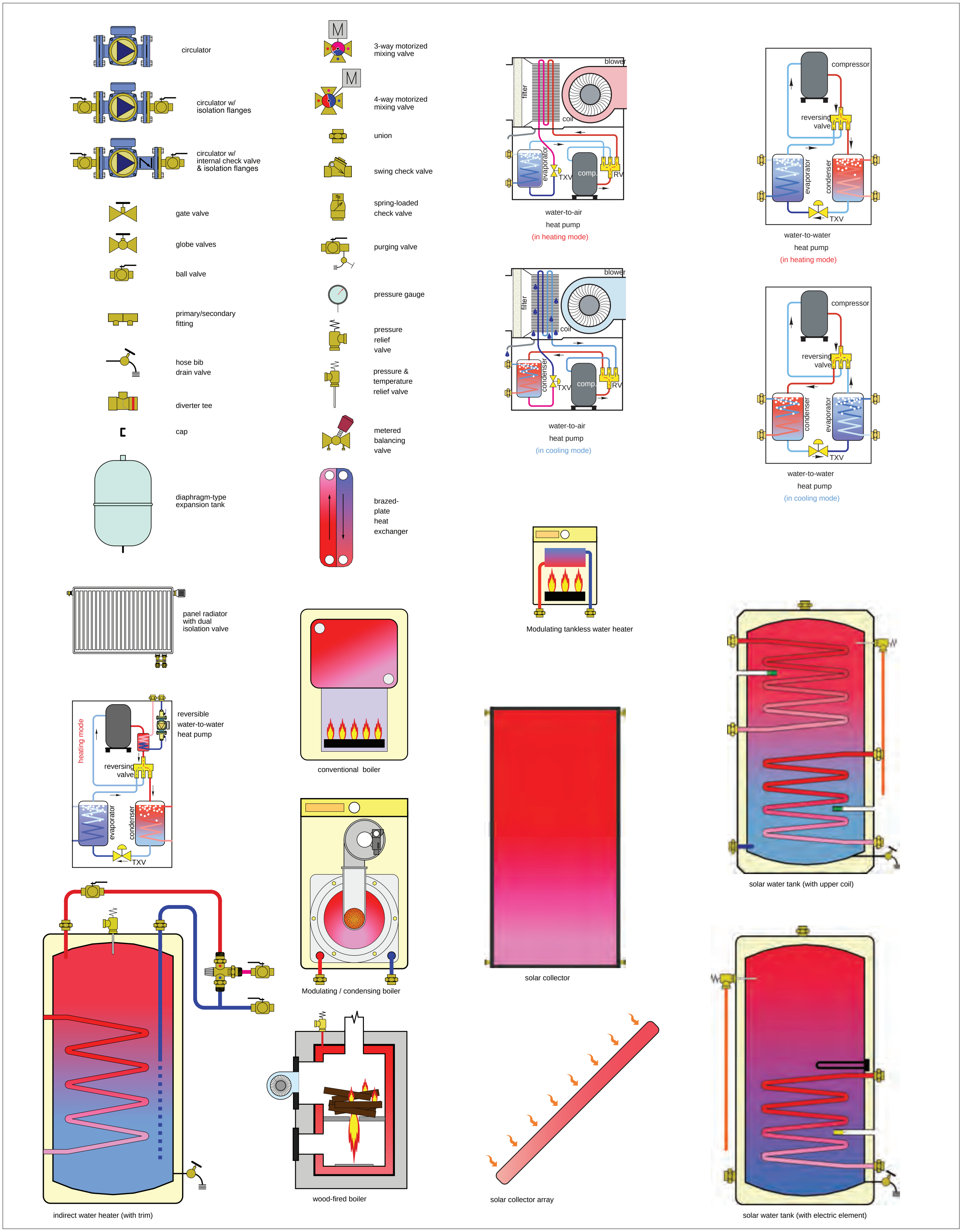
Heating professionals are encouraged to study both the proper piping layouts and the errors presented to further develop both their design and troubleshooting skills.

Section 8 presented several example systems that range from simple concepts to those which integrate two or more distribution "subsystems" into an overall system with specific design objectives. Readers are encouraged to "dissect" all of the systems presented to identify details that provide functions such as purging, mixing, zoning, air/dirt elimination, hydraulic separation, boiler protection and differential pressure control.

Several references to previous issues of *idronics* have also been made that provide a more detailed discussion of these topics.

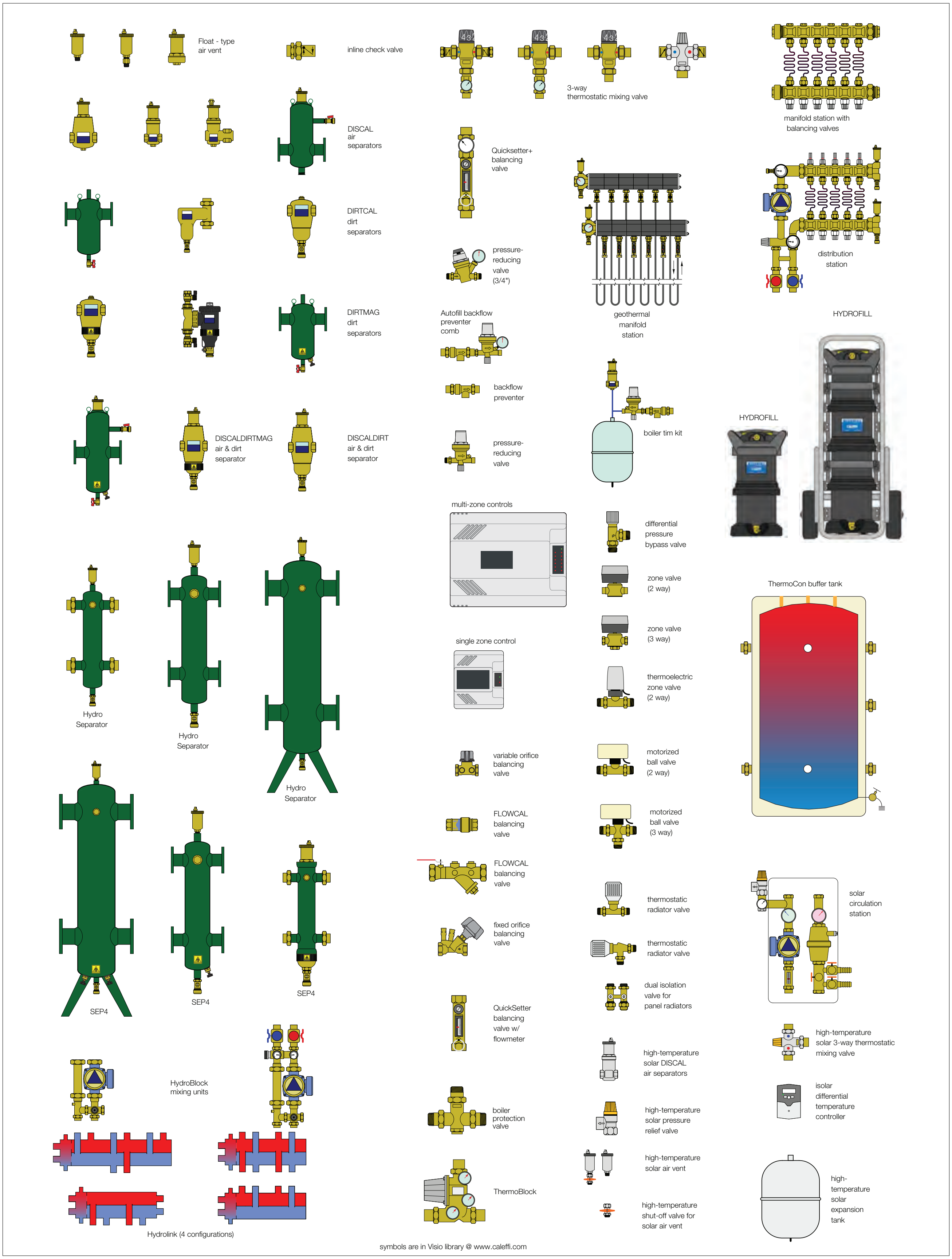
APPENDIX A: PIPING SYMBOL LEGEND

GENERIC COMPONENTS



APPENDIX A: PIPING SYMBOL LEGEND

CALEFFI COMPONENTS



SEP4™ combination hydraulic, air, dirt and magnetic separator

549 series

Function

The Caleffi SEP4 combination separator is a device that incorporates four critical functions for hot or chilled water systems. It incorporates high performance air and dirt (magnetic and non-magnetic) removal into the hydraulic separation function which makes the primary and secondary circuits connected to it hydraulically independent. The SEP4 features an internal coalescing element that continuously and automatically eliminates air micro-bubbles with the simultaneous removal of dirt particles as tiny as 5 microns. The air discharge capacity is very high, with the capability of automatically removing all the air present in the system down to the micro-bubble level. The 4-in-1 high performance functionality of the SEP4™ saves system installation and maintenance costs as there is no need to include separate air and dirt separators. In addition to removing solid impurities in the system, the added powerful removable external magnet belt (union) or probe (flanged) around or in the lower body removes up to 100% of the ferrous impurities, including magnetite, that can form in a hydronic system. The SEP4 has 2½ times the ferrous impurities removal performance of standard air and dirt separators.

Product range

5495 series	SEP4™ hydraulic, air, dirt and magnetic separator in steel with union connections, drain and insulation..connections 1" to 2" sweat, NPT female and press
NA549_M series	SEP4™ hydraulic, air, dirt and magnetic separator in steel with flanged connections, drain and insulation, ASME and CRN..connections 2" to 4" ANSI
NA549_M series	SEP4™ hydraulic, air, dirt and magnetic separator in steel with flanged connections and drain, ASME and CRN (5-6", other sizes pending)......................................connections 5" to 14" ANSI

Technical specifications

Threaded, sweat and press union connections

Materials

	- body:	epoxy resin coated steel
	- air vent body:	brass EN 12165 CW617N
	- air vent hydraulic seal:	EPDM
	- air vent float:	PP
	- air vent float linkages:	stainless steel
	- air vent float guide pin:	stainless steel
	- int. element:	HDPE
	- drain valve body:	brass EN 12165 CW617N
	- magnet:	neodymium rare-earth
	- insulation:	closed-cell expanded PE-X

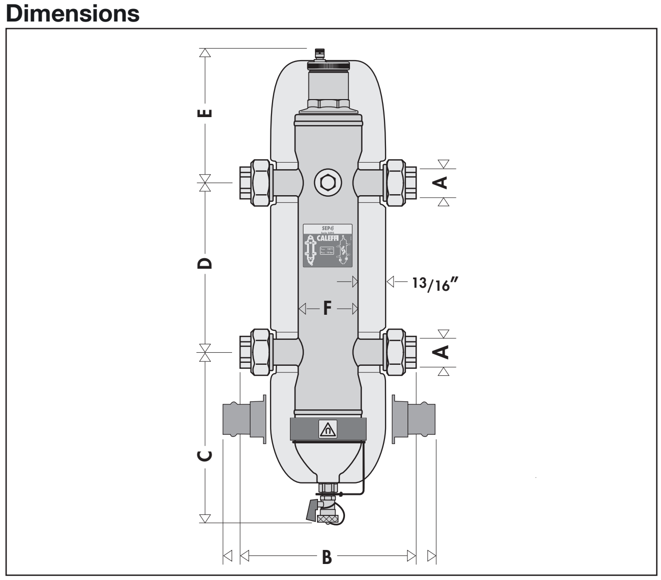
Performance

Suitable fluids:	water, glycol solution
Max. percentage of glycol:	50%
Max. working pressure:	150 psi (10 bar)
Temperature range:	without insulation 32–230°F (0–110°C) with insulation 32–210°F (0–100°C)
Particle separation capacity:	to 5 μm (0.2 mil)
Air separation efficiency:	100% removal to microbubble level
Ferrous impurities separation efficiency:	up to 100% removal

Connections

Main connections:	1", 1-¼", 1-½", 2" NPT female with unions 1", 1-¼", 1-½", 2" sweat with unions 1", 1-¼", 1-½", 2" press with unions
Thermowell tap connection:	½" female straight thread
Lay length (press connection):	size 1 inch: 8-¾" size 1-¼ inch: 9-¾" size 1-½ inch: 11-⅝" size 2 inch: 12-½"
Drain valve:	¾" garden hose connection

Dimensions



Code*	A	B swt	B npt	C	D	E	F	Wt. (lbs.)	Wt. (kg)
549506A/96A	1"	8¾"	8½"	7"	8⅝"	6"	3"	15	6.8
549507A/97A	1¼"	9¾"	9"	7"	9½"	6"	3½"	19	8.6
549508A/98A	1½"	11"	10½"	8½"	10¼"	7¼"	4½"	27	12.2
549509A/99A	2"	12⅜"	11½"	8½"	11⅞"	7¼"	5¾"	29	13.1
549566A	1"	10½"	--	7"	8⅝"	6"	3"	15	6.8
549567A	1¼"	11¾"	--	7"	9½"	6"	3½"	19	8.6
549568A	1½"	14¼"	--	8½"	10¼"	7¼"	4½"	27	12.2
549569A	2"	15½"	--	8½"	11⅞"	7¼"	5¾"	29	13.1

***54950:** NPT female connections; **54959:** sweat connections; **54956:** press connections.

Technical specifications

Flanged connections

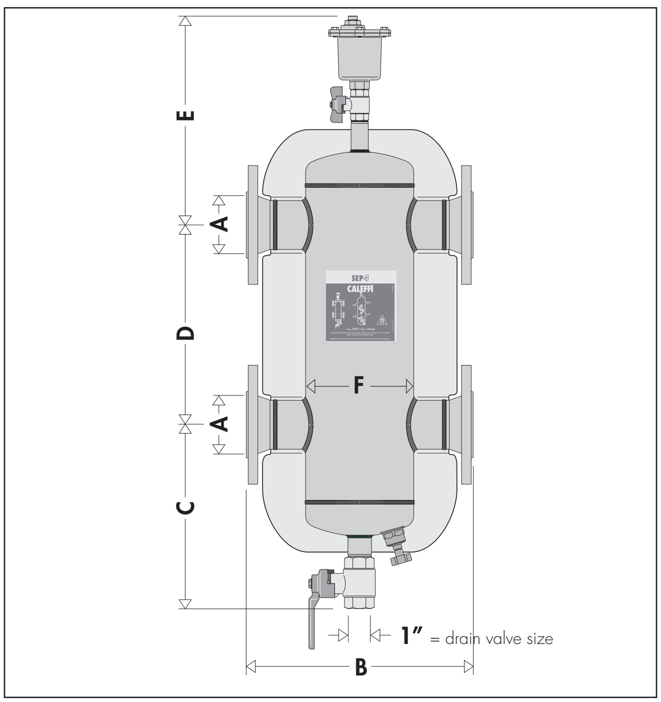
Materials
- separator body: epoxy resin painted steel
- air vent body: brass
- shut off and drain valve body: brass
- internal element: 300 series stainless steel
- air vent seal: VITON
- air vent float: stainless steel
- magnet: neodymium rare-earth
- magnet probe: brass

Performance

Suitable fluids: water and non-hazardous glycol solution up to 50%
Max. operating pressure: 150 psi (10 bar)
Max. connection velocity: 4 feet per second (1.2 m/s)
Temperature range: -with insulation: 32–220°F (0–105°C)
-without insulation (vessel) 32–270°F (0–132°C)
Particle separation capacity: 5 μm (0.2 mil)
Air separation efficiency: 100% removal to micro-bubble level
Ferrous impurities separation efficiency: up to 100% removal

Connections
- main: 2"—14"ANSI B16.5 150 CLASS RF
- drain valve: 2" — 6": 1" NPT female
  8" — 14": 2" NPT female
- thermo well tap (8" — 14" only):
  - front center: ¾" NPT female
  - inlet/outlet flanges: ½" NPT female

Dimensions



Code	A	B	C	D	E	F	Wt. (lbs.)	Wt. (kg)
NA549052AM	2"	13¾"	13"	13"	13½"	6⅝"	76	35
NA549062AM	2½"	13¾"	13"	13"	13½"	6⅝"	81	37
NA549082AM	3"	18⅜"	15"	17¾"	15¼"	8⅝"	111	50
NA549102AM	4"	18½"	15"	17¾"	15½"	8⅝"	120	55
NA548120AM*	5"	25"	23¹⁄₁₆"	22"	18¹⁄₁₆"	12¾"	223	101
NA548150AM*	6"	25"	23¹⁄₁₆"	22"	18¹⁄₁₆"	12¾"	234	106

*Without insulation.

Agency approval

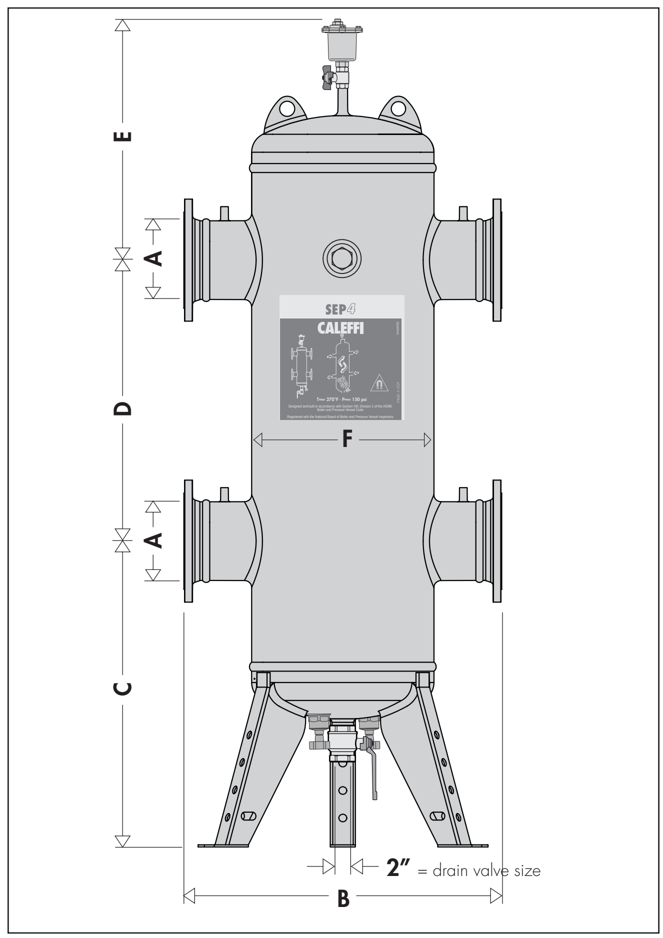
Series NA549_M is designed and built in accordance with Section VIII, Division 1 of the ASME Boiler and Pressure Vessel Code and tagged and registered with the National Board of Boiler and Pressure Vessel Inspector, and CRN registered, and stamped for 150 psi (10 bar) working pressure, with ASME U stamp. 8"-14" is CRN pending, contact Caleffi.

Technical specifications of insulation, flanged versions to 4"

Internal part
Materials: rigid closed cell expanded polyurethane foam
Thickness: 2 3/8" (60 mm)
Density: 2.8 lb/ft^3 (45 kg/m^3)
Thermal conductivity: 6 BTU·in/hr·ft^2·°F (0.023 W/(m·K))
Temperature range: 32–220°F (0–105°C)

Outer part
Materials: embossed aluminum
Thickness: 7.0-mil (0.7 mm)
Reaction to fire (DIN 4102): class 1

Head covers
Heat formed materials: PS



Code	A	B	C	D	E	F	Wt. (lbs.)	Wt. (kg)
NA549200A	8"	35½"	34"	39⅜"	25⅜"	20"	530	236
NA549250A	10"	41¾"	36⁵⁄₁₆"	43⁵⁄₁₆"	27⁵⁄₁₆"	26"	740	331
NA549300A	12"	46½"	37⅞"	47¼"	29⅜"	30"	1,110	500
NA549350A	14"	52"	38¹⁄₁₆"	58⅞"	34½"	36"	1,410	635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SEP
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DISCAL
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MAX