idronics™

CALEFFI Hydronic Solutions

37

Fall 2026

JOURNAL OF DESIGN INNOVATION FOR HYDRONIC AND PLUMBING PROFESSIONALS



Design Errors in Hydronic Systems

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FROM THE CEO

Dear Plumbing and Hydronic Professional,

No two buildings on earth are exactly the same. However, key design fundamentals can be applied to incorporate many lessons learned and set install and service contractors and building operators up for a win. At the end of the day, designers set the bar for the highest potential efficiencies, reliability and serviceability.



This issue of idronics™ discusses the classic design best practices in hydronics systems that will help you put the best foot forward in hydronic piping. There are many cases where system inefficiencies or complexities are inherited in existing buildings. For these projects, I would recommend idronics 32, Troubleshooting Hydronic Systems. The following chapters in this edition will outline the ideal piping diagram details to incorporate while you are still in the pre-construction phase.

Additional topics within our collection of idronics can be found at caleffi.com. We appreciate your continued support of the industry and encourage you to share any feedback by emailing us at marketing.na@caleffi.com.

Tina Gullickson, CEO

Caleffi North America

CALEFFI Hydronic Solutions

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1. INTRODUCTION

This issue of *idronics* begins with an apparent paradox:

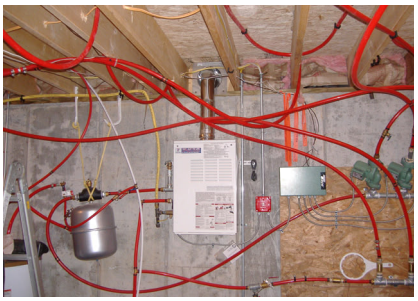
One of the ***best*** things about hydronic heating and cooling systems is that there is virtually no limit to the type of systems that can be created.

One of the ***worst*** things about hydronic heating and cooling systems is that there is virtually no limit to the type of systems that can be created.

Both statements are absolutely true.

In the hands of an informed conscientious designer, hydronic heating and cooling systems can be developed to meet the exact needs of a project. Designs can be "refined" in terms of the hardware, the required control logic, and optimization of both thermal and hydraulic efficiency.

Figure 1-1



On the other hand, choosing components based on lowest cost, most convenient availability or "ballpark" sizing methods is almost certain to lead to disappointing or even dangerous results.

The process of designing a hydronic heating or cooling system should be thought of as skillful rendering of "media." That rendering could be compared to a painter working in watercolor, a musician composing a score, or a skilled carpenter crafting a custom piece of furniture. In all cases, the designer has a palette of materials and methods with which to work. Their skill and experience determine which materials and methods are best suited to the end goal. In most cases, *changes are made during the creative process* as the designer evaluates the various options and progress toward the intended result.

Figure 1-2



An artist planning a painting knows what colors to begin with, what colors to avoid, and how the color selections and the sequence in which they are applied best suits the intended finished work. Most musicians repeatedly try different combinations of notes, rhythms, and combinations of instrumental sounds until they discover the combination that best fits the intended result. A furniture carpenter knows that they won't be cutting finished lumber with a chainsaw or fastening pieces of black walnut together with 16D framing nails.

Very few people who are unfamiliar with the HVAC industry would liken the design of hydronic heating or cooling systems to creations by artists, musicians or custom furniture carpenters. Instead, they are likely to consider such work as utilitarian, uninteresting or unrewarding. However, those who have worked in the hydronics field with open minds and a passion to "craft" systems instead of just choosing components to connect with pipes or wires can understand and embrace the similarities.

Many previous issues of *idronics* were created to expand the skillset of professionals who design custom hydronic heating and cooling systems. Those issues were often based on a deep dive into a specific technical subject. such as zoning, hydraulic separation, balancing or circulation. The material presented focused on the proper way to select and connect hardware to achieve proper performance.

This issue takes a different approach. Rather than just showing the "correct" approach, it discusses many common "incorrect" approaches, explains why they are wrong, describes what undesirable results they can lead to, and shows at least one way to avoid them. The incorrect approaches were selected based on their frequent recurrences, as well as incorrect underlying "myths" on which they are based.

idronics #32 discusses troubleshooting problems within systems that are already installed. In many cases, the corrections for the problems can be expensive in both materials and time. It is far better to identify and correct such problems *at the design stage* and thus avoid costly corrections after the hardware is installed and not functioning properly.

This issue begins with *fundamental principles* that are critically important in understanding how certain design details fail to meet immutable concepts based on physics. Most of these principles are simple concepts, such as knowing the

Figure 1-3



flow rates into and out of a liquid-filled component must always be equal, or that the water temperature at which a system attempts to operate is based on equal rates of heat input and heat dissipation.

The discussion then turns to verifying that these fundamentals are present and properly applied during the design of subsystems, including the heat source, distribution piping and heat emitters. As is customary in *idronics,* the issue concludes with examples systems that demonstrate design errors and their corrections.

Several references are made to past issues of *idronics* that provide more detailed information on specific topics. Readers are encouraged to avail themselves of these past issues, all of which are available in digital form at ***idronics.caleffi.com***

2. APPLYING FUNDAMENTALS

Many design errors associated with hydronic heating or cooling systems can be traced to misunderstanding or attempting to circumvent fundamental principles of thermodynamics, fluid flow or heat transfer. Experienced designers respect these unchanging principals. They learn how to leverage them to create systems that deliver predictable and reliable performance. This section describes several fundamentals that apply to *every* hydronic heating or cooling system regardless of scale, hardware quality or installed cost.

ENERGY INPUTS & OUTPUTS

All hydronic heating systems convert one or more forms of *input* energy into *output* energy. Examples of input energy include the chemical energy in a *primary fuel* such as natural gas, fuel oil or wood pellets. These primary fuels are converted into heat. Some heat sources, such as electric boilers, convert electricity directly into heat by passing current through resistance elements.

Modern hydronic heating systems also require a secondary (electrical) energy input to operate devices, such as the heat source, circulators, zone valves and controllers.

The output energy from a hydronic heating system is heat. Most of it is delivered through heat emitters that have been selected and sized based on specific building requirements.

The rate at which heat is dissipated from heat emitters is approximately proportional to their surface area, and the difference between the water temperature within the heat emitter and the surrounding air, raised to some mathematical power. This relationship can be expressed as formula 2-1:

Formula 2-1:

$$q \approx A \times (T_{water} - T_{air})^n$$

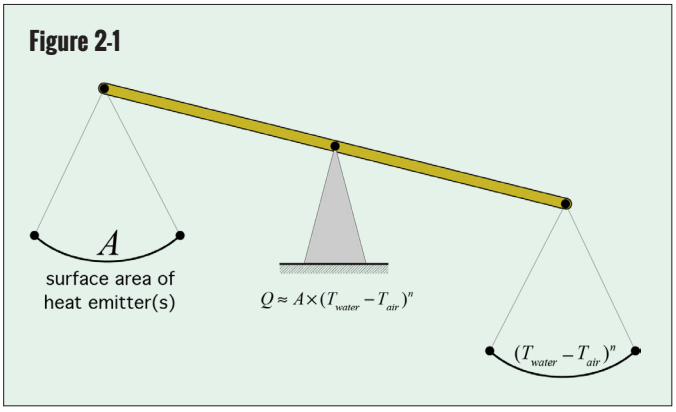
Where:
q = rate of heat output (Btu/hr)
A = effective surface area of the heat emitter (ft^2)
T_{water} = average temperature of the fluid within the heat emitter (ºF)
T_{air} = air temperature surrounding the heat emitter (ºF)
n = an exponent the depends on the type of heat emitter

Finned-tube baseboard	n=1.4
radiators	n=1.3
radiant floor (heating)	n=1.1
radiant ceiling (heating)	n=1

Formula 2-1 implies that, for a given rate of heat output rate (q), any increase in the heat emitter's surface area will lower the required value of the temperature difference (T_{water}-T_{air}), and vice versa. This is a very fundamental relationship that all system designers need to understand and respect.

The "trade off" between heat emitter surface area and the ΔT between water temperature and air temperature is what allows properly installed radiant floors (with

Figure 2-1



large surface area) to operate at much lower water temperatures compared to fin-tube baseboard (with much less surface area).

In addition to the system's heat emitters, a lesser amount of heat is released from the surfaces of piping and other components through which heated water flows. The extent of this secondary heat output depends on the combined surface area of the piping and components, the temperature difference between the water within those components and the surrounding air, and the effectiveness of any insulation placed on the piping and components.

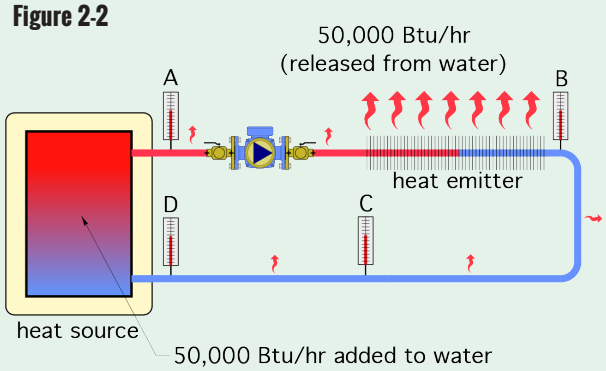
If the piping and components are well insulated, the amount of heat dissipated from them should be very small relative to the heat delivered from the system's heat emitters. Still, from the standpoint of thermodynamics, every pipe, fitting, valve and other component through which heated fluid passes will emit some amount of heat. This is undesirable from the standpoint of efficiency and control but can be minimized through good design and proper insulation.

THERMAL EQUILIBRIUM

All hydronic systems naturally seek to operate at a condition where the rate of heat input to the water at the heat source equals the rate of heat dissipation from the distribution system and heat emitters.

In a hypothetical situation *where all controls were removed from a system*, and that system was put into operation (e.g., heat source(s) and circulators/zone valves turned on), and the system was allowed to operate until it achieved steady state conditions, the water temperatures at all locations in the system would stabilize at values that would allow the total rate of heat input to the system to equal the total rate of heat output. This condition is called thermal equilibrium.

Figure 2-2



Consider the hydronic heating system shown in figure 2-2.

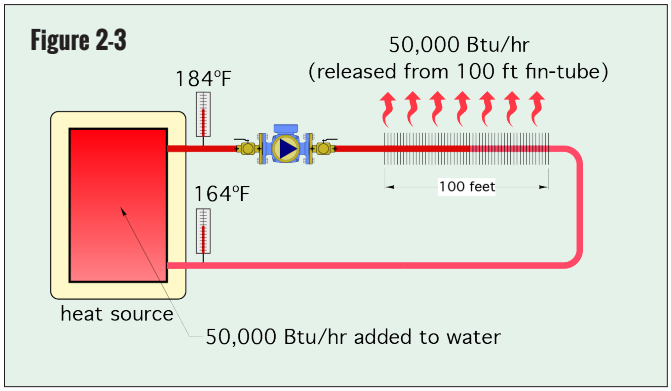
In this system, the heat source is adding heat to the water at a rate of 50,000 Btu/hr. As the water flows through the distribution system, heat is being dissipated from it at the same rate of 50,000 Btu/hr. Under this condition, the system is in "thermal equilibrium," and the water temperature at any location in the circuit (A,B,C,D) remains steady.

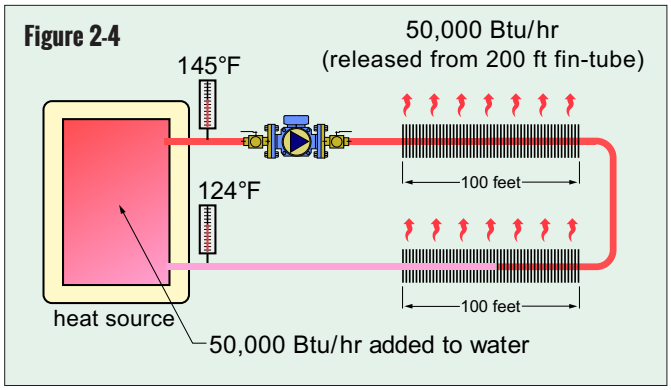
The exact value of the water temperature at these locations depends on the heat dissipating characteristics of the distribution system versus the heat input rate of the heat source. For a given rate of heat input, the higher the heat-dissipating ability of the distribution system, the lower the water temperature will be at all locations around the circuit. Conversely, the lower the heat-dissipating ability of the distribution system, the higher the water temperature will be at all locations. In all cases, water temperature decreases in the downstream direction.

Assume that the system in figure 2-2 contains 100 feet of residential-style fin-tube baseboard. For simplicity, assume all heat dissipation from the distribution system occurs at this baseboard. Assume the heat source can generate water temperatures as high as 210ºF, and that the high-limit controller on the heat source is set for 200ºF. Initially, all system components and the water they contain are at room temperature.

The heat source and circulator are turned on. The heat source is now adding heat to the water at the rate of 50,000 Btu/hr. The water temperature at all locations in the system continues to rise. However, the rate of this temperature rise slows as the water temperature leaving the heat source climbs higher and higher. When the heat source outlet temperature reaches 184ºF there is no further temperature rise, even though the boiler remains in continuous operation. This steady state condition is shown in figure 2-3.

Figure 2-3



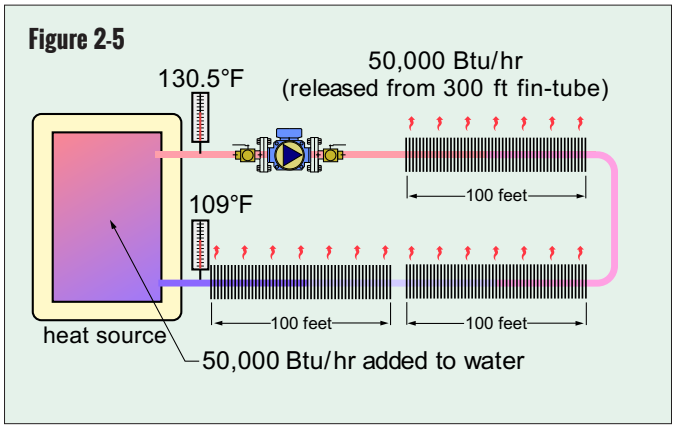


The explanation for this behavior is that when water at 184ºF is supplied to 100 feet of fin-tube baseboard, that baseboard can dissipate 50,000 Btu/hr. The rates of heat input and output are equal at this condition. *There is no reason for the water temperature to climb any higher — and it won't!* The water temperature at all locations around the circuit will also remain stable at this point (albeit, at lowers temperatures farther downstream). The fact that the high-limit controller on the heat source was set at 200ºF makes no difference.

Now consider what would happen if this system were modified so that it contained 200 feet of the same baseboard. No other changes were made. When the system is turned on, the water temperature leaving the heat source rises until it stabilizes at 145ºF, as shown in figure 2-4.

The modified system has again reached thermal equilibrium, where the 50,000 Btu/hr rate of heat input from the heat source is exactly matched by the 50,000 Btu/hr heat dissipation rate from 200 feet of baseboard. Again, the fact that the high-limit controller on the heat source was set at 200ºF makes no difference.

Next, assume another 100 feet of baseboard was added to the system. When the system that now contains 300 feet



of baseboard is turned on, the water temperature leaving the heat source stabilizes at 130.5ºF. It has again achieved thermal equilibrium, but at an even lower temperature.

These scenarios show that increasing the heat dissipation ability of the system by increasing the surface area of the heat emitters lowers the supply water temperature at which thermal equilibrium occurs. Other factors that could increase heat dissipation would include decreasing room air temperature, increasing water flow rate through the circuit, or increasing the air velocity across the heat emitter surfaces.

Lowering the water temperature at which thermal equilibrium occurs is generally acceptable, and even desirable, because it increases the thermal efficiency of modern heat sources such as modulating/condensing (mod/con) boilers and hydronic heat pumps. However, "conventional,"non-condensing boilers have operating temperature limits below which they will experience sustained flue gas condensation. Operating a conventional boiler under such conditions is very *undesirable* because it causes corrosion of the boiler's heat exchanger and vent piping. Fortunately, devices such as thermostatic anti-condensation mixing valves or other mixing assemblies can be used to protect conventional boilers that supply low temperature distribution systems.

If the heating capacity of the heat source is high relative to the total surface area of the heat emitters, the temperature at which thermal equilibrium would occur is often higher than the setting of the heat source's high-limit controller.

For example, if the baseboard length in the system shown in figure 2-2 was *decreased* to only 50 feet, the water temperature leaving the heat source when thermal equilibrium was achieved would be about 265ºF. This is much higher than what is considered safe or efficient for any residential or light commercial system. Fortunately, the heat source's high-limit controller would stop further operation at its set temperature of 200ºF. The distribution circulator would remain on, and the water temperature leaving the heat source would decrease as heat continues to be dissipated by the distribution system. Eventually, the water temperature within the system drops to a temperature where the heat source is turned back on, and the cycle repeats itself. *This is a very common operating mode in many systems, especially during partial load conditions. It can even occur under design load conditions in systems that have an oversized heat source.*

The water temperatures that are established at thermal equilibrium don't necessarily deliver the required amount

of heat where it needs to go. Those temperatures may not allow the system to operate at peak efficiency or create conditions that are "good" for the components. *The conditions established at thermal equilibrium might not even be safe!* Thermodynamics "doesn't care" about these conditions, it only "cares" about a balance between heat input and heat output. *It's up to the designer to create systems that — when operating at or close to thermal equilibrium — deliver the correct amount of heat output, with components operating at efficient and safe conditions.*

HYDRAULIC EQUILIBRIUM

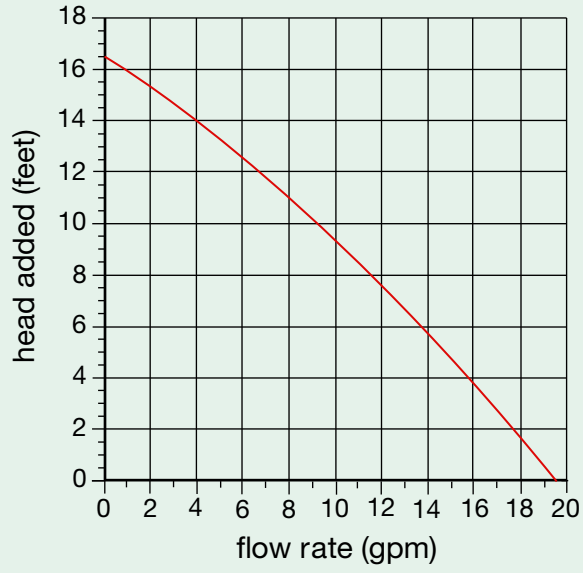
The concept of *equilibrium* between energy input and energy output also applies to the hydraulic (mechanical) aspects of all hydronic systems.

Circulators convert electrical input energy into heat and mechanical energy. The conversion of electricity into heat is an unintentional process within the circulator. Some of the heat generated is caused by friction between the fluid and the internal parts of the circulator. Additional heat is generated by the electrical resistance of the motor windings.

The *intentional* energy conversion process for a circulator is transforming electrical energy into mechanical energy. The latter is called "head." A circulator is the only component in a hydronic circuit that adds head energy to the fluid.

The amount of head energy added to each pound of fluid moving through the circulator depends on the flow rate through the circulator. This relationship is described by a pump curve, an example of which is shown in figure 2-6.

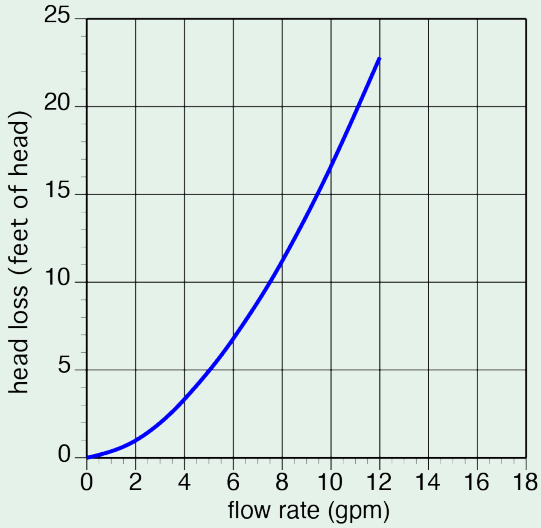
Figure 2-6



Every component in a hydronic circuit — other than an operating circulator — *removes* head energy from the flowing fluid, mostly due to friction.

This dissipation effect depends on several things, including the pipe type and size, the number and types of fittings and valves in the circuit, the hydraulic characteristics of other components in the circuit (heat source, air separator, heat exchanger, etc.), the fluid being circulated, and the temperature of that fluid. These can be mathematically combined into a circuit head loss curve, an example of which is shown in figure 2-7.

Figure 2-7



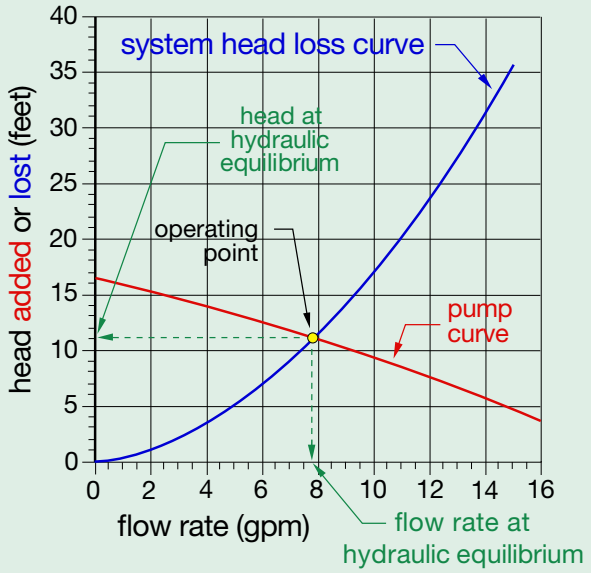
The flow rate that a specific circulator will produce in a specific hydronic circuit is based on an equilibrium condition where the head energy added to the fluid by the circulator equals the head energy dissipated from the fluid by friction within the piping circuit. This condition is called hydraulic equilibrium. In most circuits, this equilibrium is be established within a few seconds after the circulator is turned on.

The flow rate at hydraulic equilibrium can found by plotting the head loss curve for the circuit on the same graph as the pump curve for the circulator, as shown in figure 2-8.

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

Every circulator has at least one pump curve that is published by its manufacturer.
Multi-speed circulators have a separate curve for each speed. *Variable-speed* circulators typically have a maximum speed pump curve and a minimum speed pump curve. In theory, a variable-speed circulator can operate at many difference pump curves between the maximum and minimum curves. *Multi-mode* circulators can have many different pump curves depending on which mode they are set in. Figure 2-9 shows examples of pump curves for multi-speed, variable-speed and multi-mode circulators.

Figure 2-8



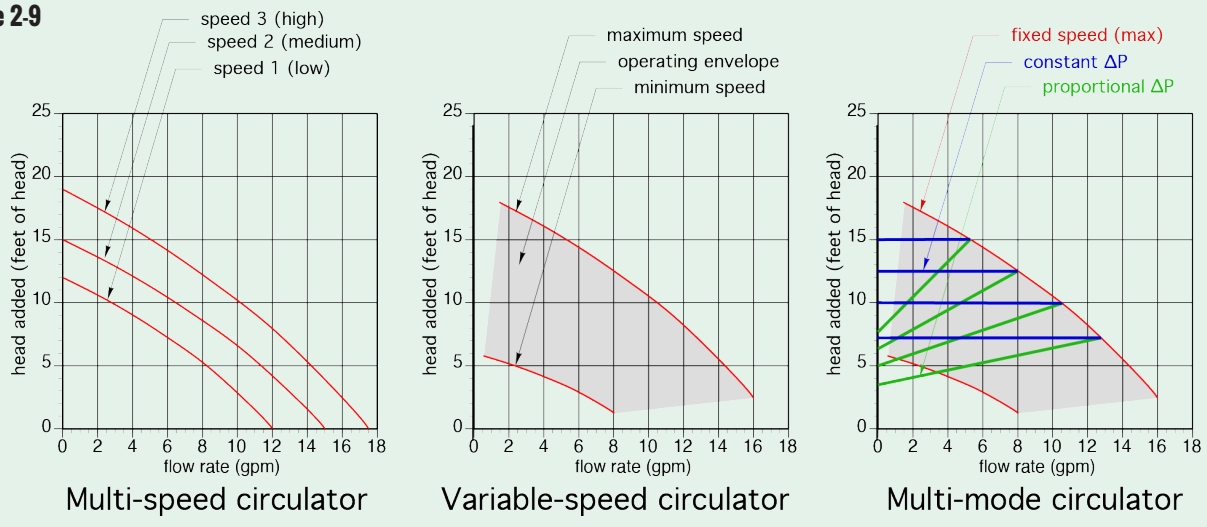
Hydronic system designers should know how to create the head loss curve for any circuit they design. There are numerous manual and computer-based methods for doing so. Likewise, designers should know how to match the circuit head loss curve with the pump curve of a "candidate" circulator to determine what flow rate would occur if that circulator was used in the circuit. Software such as the *Hydronics Design Studio* can quickly determine the flow rates that would occur in a variety of piping circuits matched with a wide range of circulators.

For more information on creating circuit head loss curves and matching them with pump curves, refer to *idronics* #12 and #16.

HEAT TRANSFER FUNDAMENTALS

All hydronic heating and cooling systems involve the transfer of heat from one location to another. In heating applications, this heat transfer takes place across many components, starting at a heat source and ending at the heat emitter(s). Between these "end points" several intermediate heat transfer processes must occur. Examples include heat moving from combustion gases to water flowing through a boiler, heat passing from one fluid to a different fluid through a heat exchanger, and heat passing from water inside a copper tube to aluminum fins attached to the outer surface of that tube, and finally into room air.

Figure 2-9



It's important for designers to understand the physical processes by which heat moves. That knowledge will help them quickly identify factors that may constrain heat movement. Examples include scale formation on a boiler's heat exchanger, heat emitters with insufficient surface area, or a thick rug placed over a heated floor slab.

Figure 2-10





Heat transfer is classified into three processes: Conduction, convection and thermal radiation. The fundamental theory and governing equations for each of these processes are discussed in *idronics #23.*

For more information on heat transfer, see *idronics* #23.

Many design errors that reveal themselves as the inability to maintain interior comfort can be traced to one or more locations where a heat transfer restriction or "bottleneck" occurs.

Since this issue focuses on common design errors, it's appropriate to point out some situations where problems are caused by a misunderstanding of — or perhaps "under-appreciation" of — factors that determine rates of heat transfer. Examples include:

1. *Covering a heat floor slab with one or more materials that have high thermal resistance (e.g., R-value).* Materials such as polyurethane carpet padding and thick carpet, or multiple layers of wood flooring, can severely restrict heat transfer by conduction and thus create a thermal bottleneck that results in low rates of heat transfer and inadequately heated rooms.

2. *Omitting or using inadequate materials to insulate the underside of a heated concrete slab.* This situation is not a heat transfer bottleneck. Quite the opposite, it's a situation where heat can move in an *undesirable direction* (e.g., from the bottom of the slab to the soil below) at an *unacceptably high rate*.

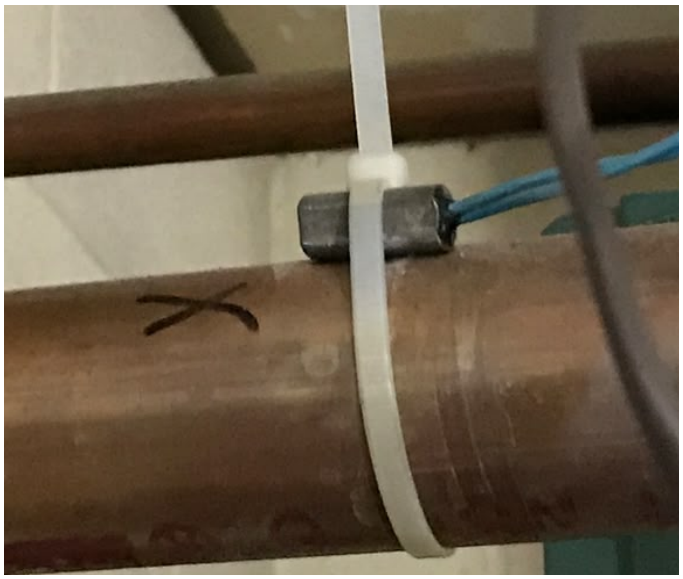
3. *Stapling a PEX tube to the underside of plywood subfloor without tight fitting aluminum plates* and expecting the narrow and spotty contact between the top of the tube and the subfloor to conduct sufficient heat to maintain comfort in the room above the floor.

4. *Assuming that the depth at which a PEX tube is embedded within a concrete slab has no significant effect on the slab's ability to heat the space above*.

5. Assuming that a small internal coil heat exchanger within an indirect water heater can transfer heat into domestic water at the same rate the heat source is producing heat.

6. Connecting a cylindrical temperature sensor to the outside of a copper tube without an insulation sleeve and expecting the sensor to accurately report the temperature of the water flowing through the tube.

Figure 2-11



7. Falsely thinking that "heat rises" and thus it's unnecessary to insulate the underside of a floor that contains tubing and heat transfer plates.

8. Assuming that the heat loss from uninsulated copper tubing is insignificant compared to the rates at which water is conveying heat through that tubing.

Figure 2-12



9. Expecting a temperature sensor that's suspended loosely within a sensor well to accurately report the temperature of the water surrounding the sensor well.

10. Assuming that uninsulated copper tubing connected to a thermal storage tank has no significant effect on the rate of heat loss from the tank.

11. Assuming that a person seated next to a large window on a cold winter day will be comfortable if the air temperature surrounding them is 70ºF.

12. Sizing a brazed plate heat exchanger based on assumed and "optimistic" conditions that will seldom occur in the system.

Several of these heat transfer "bottlenecks" are discussed in more detail in later sections.

FLOW FUNDAMENTALS

Heat transfer "bottlenecks" are not the only potential cause of poorly performing systems. Any design detail that overly restricts flow can be equally problematic. Flow bottlenecks often occur when the designer doesn't account for the flow resistance of all components in the circuit.

One example of a common flow bottleneck is sizing a 3-way thermostatic mixing valve based solely on pipe size, *without considering its Cv value*.

The Cv value of a valve is the flow rate of 60ºF water, measured in gallons per minute, that produces a 1 psi pressure drop through the valve in its fully open position. Formula 2-2 relates the pressure drop, flow rate and Cv of a value over a range of operating conditions.

Formula 2-2:

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

Where:
ΔP = pressure drop through the device (psi)
D = density of the fluid flowing through the device (lb/ft^3)
62.4 = density of 60ºF water (lb/ft^3)
f = flow rate through device (gpm)
Cv = Cv value of the device (fully open)

For example, consider a 3-way thermostatic mixing valve with 1-inch piping connections and a Cv = 3.0. Since the valve is physically able to connect to 1-inch piping, and 1-inch pipe can operate at 10 gallons per minute flow rate, a designer might assume that it's reasonable to use this mixing valve in an application where a flow of 10 gpm is present. However, the corresponding pressure drop, assuming water at 120ºF is flowing through the valve, is:

$$\Delta P = \left(\frac{D}{62.4}\right)\left(\frac{f^2}{Cv^2}\right) = \left(\frac{61.62}{62.4}\right)\left(\frac{10^2}{3^2}\right) = 10.97 \approx 11 psi$$

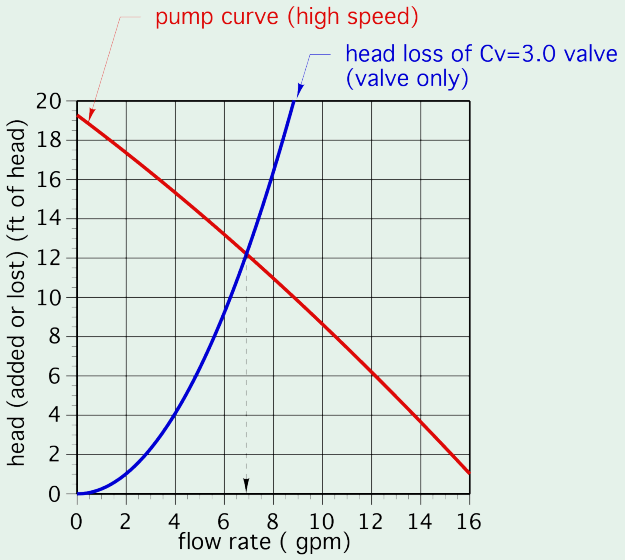
Converting this pressure drop to a corresponding head loss:

$$H_L = \left(\frac{144}{61.62}\right)11 = 25.7\, ft$$

Now consider that the designer, based on "tradition," and without referencing technical data for the product, assumes that a common zone circulator is sufficient to create the necessary 10 gpm flow through the mixing valve and the remainder of the circuit. One common zone circulator used in North America can produce a head of 7.2 feet at 10 gpm, and another common zone circulator can produce a head of 8.6 ft at 10 gpm. Neither of these circulators comes close to producing the required 25.7 feet of head required to maintain 10 gpm through the valve. This insufficiency is likely to be *considerably worse* when the head loss of the remainder of the circuit is factored in.

The result of this "flow bottleneck" can be predicted based on the previously discussed concept of hydraulic equilibrium. The flow rate that would develop — considering *only the valve* and disregarding the head loss of all other components in the circuit — is found by plotting the

Figure 2-13

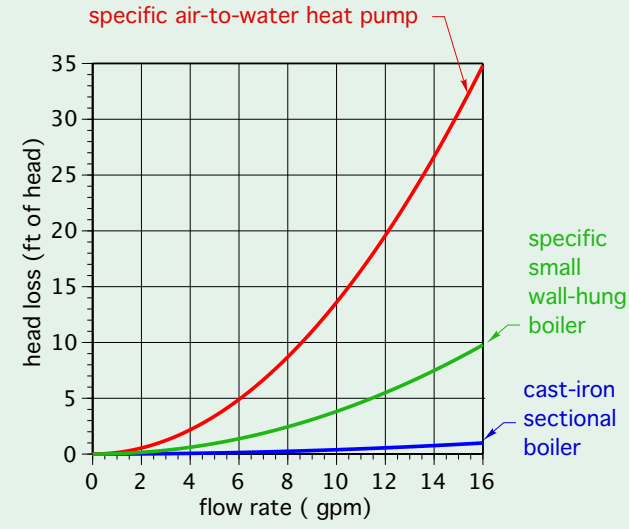


head loss curve of the valve with the pump curve of the circulator, as shown in figure 2-13.

The resulting flow rate is about 6.9 gpm, far below the desired flow rate of 10 gpm. *The flow rate that would develop considering the head loss of the mixing valve combined with the head loss of the other piping and components in the circuit would be signifycantly lower than 6.9 gpm.*

This illustrates a "classic" flow bottleneck, many of which continue to occur when designers select components based solely on pipe size.

Figure 2-14



Another common flow bottleneck can occur when the head loss versus flow rate characteristic of the heat source is not considered. Some mod/con boilers with compact heat exchangers have much higher head loss characteristics compared to conventional boilers with cast iron heat exchangers. Some geothermal heat pumps with coaxial tube-in-tube heat exchangers can also create high head loss. Figure 2-14 shows a comparison.

Proper design requires that the head loss of the heat source is accounted for when selecting a circulator to provide the desired flow rate. Data for this evaluation is usually found in the installation manual or product specifications published by the manufacturer.

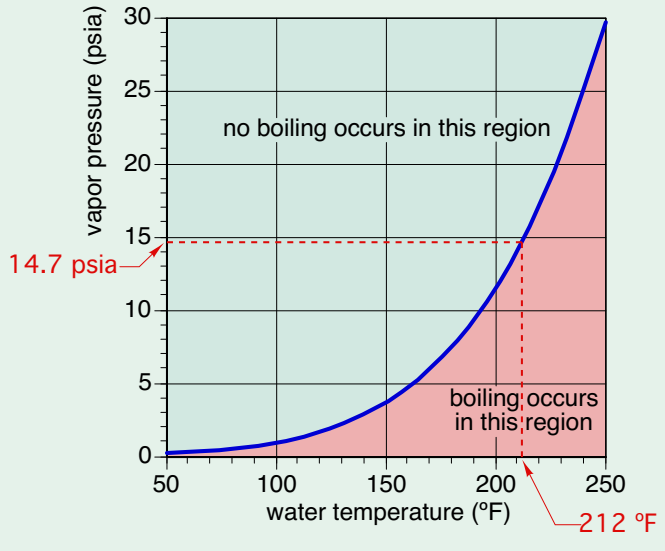
AVOIDING VAPOROUS CAVITATION

Another fundamental concept that must be considered when designing a hydronic system is ensuring that the circulators in the system do not cavitate when operating.

Vaporous cavitation occurs within a circulator's impeller when the absolute pressure drops below the fluid's vapor pressure of the fluid. The fluid instantly "boils" under such a condition, expanding approximately 1,500 times its original volume.

Vapor pressure is a function of fluid temperature. The higher the temperature of the fluid, the higher its vapor pressure, and the "closer" the operating conditions are to vaporous cavitation. Figure 2-15 shows how the vapor pressure of water varies with temperature.

Figure 2-15



The pressure shown on the vertical axis of figure 2-15 is *absolute pressure* (e.g., pressure relative to a pure vacuum).

To verify that vaporous cavitation will not occur, the net positive suction head *available* (NPSHa) at the inlet of a circulator must be higher than the net positive suction head *required* (NPSHr) by the circulator.

The NPSHa at the inlet of a circulator is based on several conditions of the fluid as it enters the circulator. These include the fluid's speed, temperature, pressure and its vapor pressure. Formula 2-3 can be used to calculate the NPSHa.

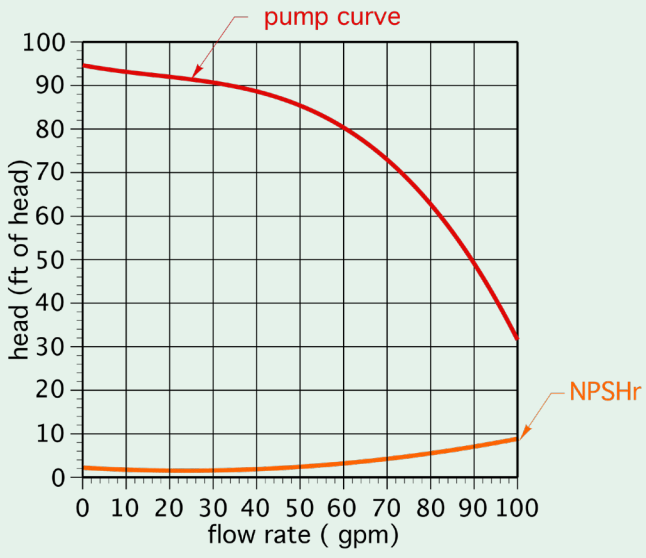
Formula 2-2:

$$NPSHa = \frac{v^2}{64.4} + \left(P_i + 14.7 - P_v\right)\left(\frac{144}{D}\right)$$

NPSHa = net positive suction head *available* at the circulator inlet (ft of head)
v = velocity of the liquid entering the circulator (ft/sec)
P_i = gauge pressure of the moving fluid measured at the circulator entrance (psig)
P_V = vapor pressure of the liquid as it enters the circulator (psi absolute)

The net positive suction head *required* (NPSHr) by a specific circulator depends on the flow rate through that circulator. The higher the flow rate, the higher the NPSHr. This information is typically available from circulator manufactures. Figure 2-16 shows an example of a NPSHr curve co-plotted with a pump curve.

Figure 2-16



There are several design guidelines for avoiding vaporous cavitation. Circuit designs that incorporate these guidelines increase the safety margin against vaporous cavitation. These guidelines strive to make the NPSHa of the circuit as high as possible. However, designers should still use formula 2-2 and compare it to the NPSHr of the circulator to ensure that vaporous cavitation will not occur.

- Keep the static pressure on the system as high as practical.
- Keep the fluid temperature as low as practical.
- Always place the expansion tank near the *inlet* side of the circulator.
- Keep the circulator low in the system to maximize static pressure at its inlet.
- Don't place components with high flow resistance near the inlet of the circulator.
- If the system has a static water level, such as a partially filled tank, keep the inlet of the circulator as far below this level as possible.
- Be especially careful in the placement of high-head circulators or close-coupled series circulator combinations because they create greater pressure differentials.
- Provide a straight length of pipe at least 10 pipe diameters long upstream of the circulator's inlet.

See *idronics* #16 for more information on circulator cavitation.

PRESSURE FUNDAMENTALS

In addition to heat transfer and fluid flow fundamentals, designers need to understand how pressure effects hydronic system operation. In some circumstances, a pressure that's too low can create problems. In other circumstances, a pressure that's too high creates problems. In still other situations, the system will operate without issues over a relatively wide range of pressure.

STATIC PRESSURE

Some of the pressure in a hydronic system is caused by the weight of the fluid above a given location in the piping circuit. This is called *static* pressure. The higher the top of a fluid filled circuit is above a given point, the higher the static pressure at that point. Static pressure can be calculated using formula 2-3.

Formula 2-3:

$$P_{static} = P_{top} + H\left(\frac{D}{144}\right)$$

Where:
P_{static} = pressure at some location (H) feet *below the top of the system* (psi)
P_{top} = pressure at top of the system (psi)
H = vertical distance from top of the system down to the location where static pressure is to be calculated (ft)
D = density of the fluid (lb/ft^3)
144 = a constant needed for the units involved.

For example: Find the static pressure in a piping system filled with 60ºF water at a location 30 feet below the top of the system. Assume that the pressure at the top of the system is 5 psi.

Solution: The density of 60ºF water is 62.4 lb/ft^3. Putting this and the other information into formula 2-3 yields:

$$P_{static} = P_{top} + H\left(\frac{D}{144}\right) = 5 + 30\left(\frac{62.4}{144}\right) = 18\,psi$$

Determining the static pressure at the location where the system's expansion tank is connected is necessary for proper sizing of the expansion tank, and for adjusting the air pressure within the air side of the tank.

Knowing the static pressure at the inlet of a circulator is necessary when calculating the net positive suction head available (NPSHa) to the circulator.

Knowing the static pressure is also important when selecting the pressure relief valve for a system.

Some heat sources sense static pressure and cease operation if that pressure falls below some minimum value.

From the standpoint of troubleshooting, a static pressure read on a pressure gauge that's significantly below the value found using formula 2-3 may indicate that the system is not filled with fluid. It might also be caused by a defective or inaccurate pressure gauge.

DYNAMIC PRESSURE

The pressures in a piping circuit change when the circulator is operating. The pressure at all points in a circuit when its circulator is operating is called *dynamic* pressure. That pressure is the combined result of that static pressure, the pressure increase due to head energy being added to the circuit by the circulator, and the head loss due to friction effects through all piping and components in the flow path.

In many systems, the change from static pressure to dynamic pressure will not create operational problems. For example, if the static pressure at some location in the system is 10 psi, and this changes to a dynamic pressure of 12 psi when the circulator is operating, it's unlikely that any operational problem would occur.

However, if the dynamic pressure at the location of the pressure relief valve happens to approach or reach the rated pressure of that valve, it can start discharging fluid.

If the dynamic pressure at the location of an air separator or float-type vent drops below atmospheric pressure, air will be sucked into the system through these components.

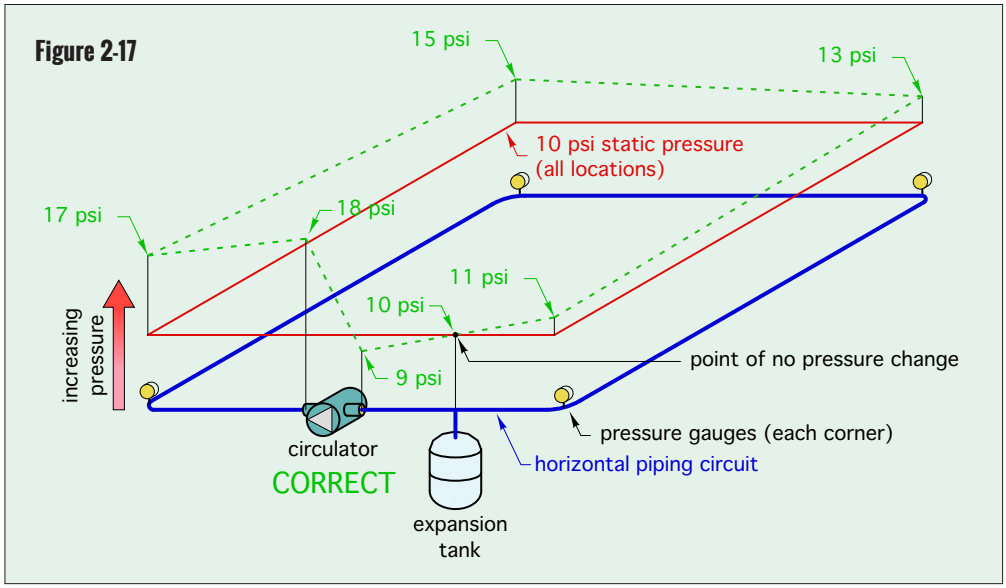
IMPROPER LOCATION OF AN EXPANSION TANK

One of the most important pressure-related fundamentals that applies to all closed-loop hydronic circuits is the location where the expansion tank connects to the system. *The expansion tank connection should always be close to the inlet side of the circulator.*

This allows the pressure differential created by the circulator to *add* to the static pressure in the system. Doing so makes the dynamic pressure at all locations around the circuit — up to the connection point of the expansion tank — higher than the static pressure.

An increase in pressure when the circulator is operating helps eject air bubbles from the system at vent locations. It also increases the "safety margin" between the conditions at the circulator's inlet and conditions that would cause vaporous cavitation.

Figure 2-17



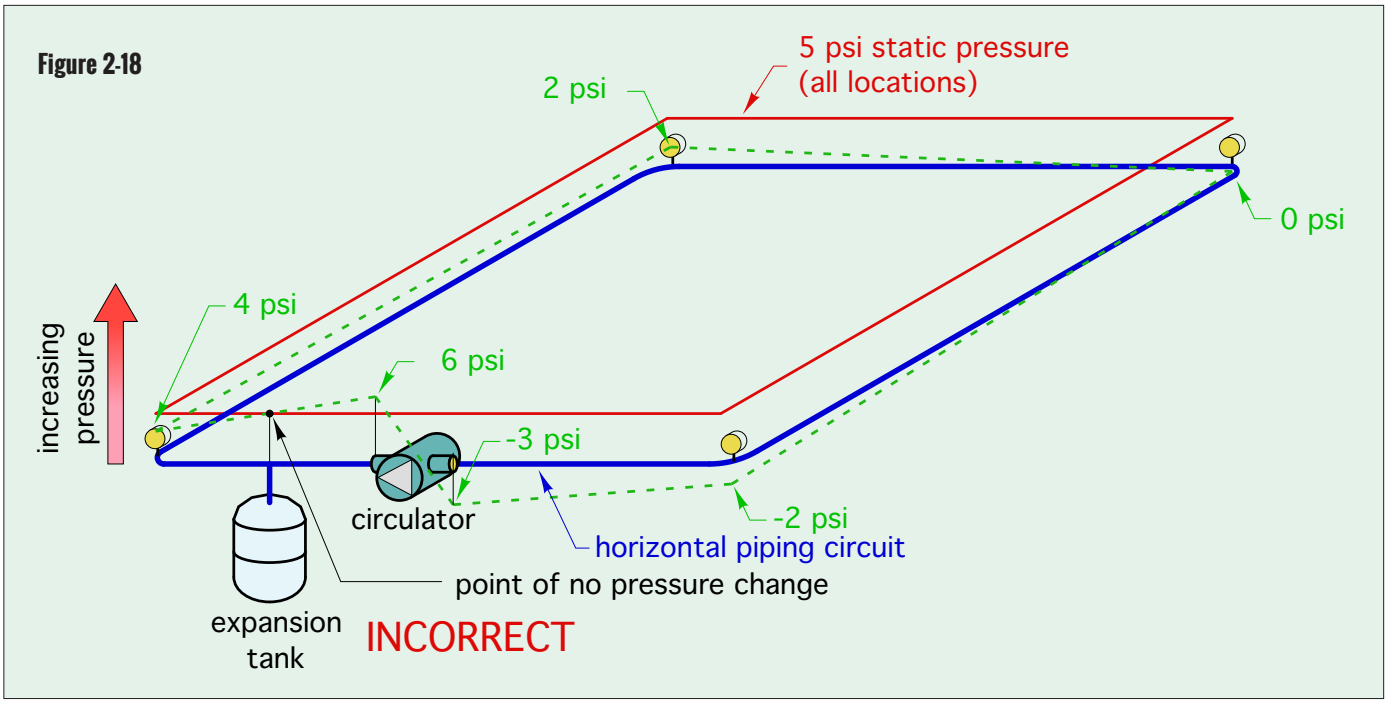


Figure 2-17 illustrates the proper location of the expansion tank connection relative to the circulator.

The blue lines represent a piping circuit located within a horizontal plane. The red lines that "float" above the piping circuit represent a static pressure of 10 psi. Since the height of the piping circuit is the same at all locations, the static pressure will also be the same 10 psi at all locations.

The green dashed lines represent the dynamic pressure that occurs when the circulator is operating. Note that the dynamic pressure is the same as the static pressure at the location where the expansion tank connects to the system. That point is appropriately named the "*point of no pressure change.*"

The dynamic pressure on the discharge side of the circulator is the highest pressure in the circuit. The slight downward slope of the dashed green line represents pressure decrease due to head loss as fluid flows around the circuit. In all cases, the dynamic pressure is above atmospheric pressure, which allows any venting device to eject air while the circulator is operating.

The desirable situation shown in figure 2-17 can be changed into a very undesirable situation by moving the connection point of the expansion tank to the discharge side of the circulator and dropping the circuit's static pressure to 5 psi, as shown in figure 2-18.

Although the static pressure at all locations in the horizontal circuit is 5 psi above atmospheric pressure, the dynamic pressure profile represented by the dashed green lines has changed significantly. The point of no pressure change is now located on the discharge side of the circulator and the pressure differential across the circulator remains at 9 psi. These conditions cause the dynamic pressure to drop to 0 psi at the upper right corner of the piping circuit and further decrease to sub-atmospheric pressures from that point back to the circulator's inlet. Any venting device located where there's sub-atmospheric pressure will suck air into the circuit when the circulator operates. This operating condition also increases the potential for vaporous cavitation of the circulator. Designers need to anticipate and avoid such conditions.

3. DESIGN ERRORS RELATED TO PIPING LAYOUT

The introduction to this issue of *idronics* began with a paradox, a portion of which stated that *there is virtually no limit to the type of systems that can be created.* This versatility is a powerful concept for an experienced and articulate designer. It can also lead to very disappointing results when haphazardly applied without regard to the fundamentals.

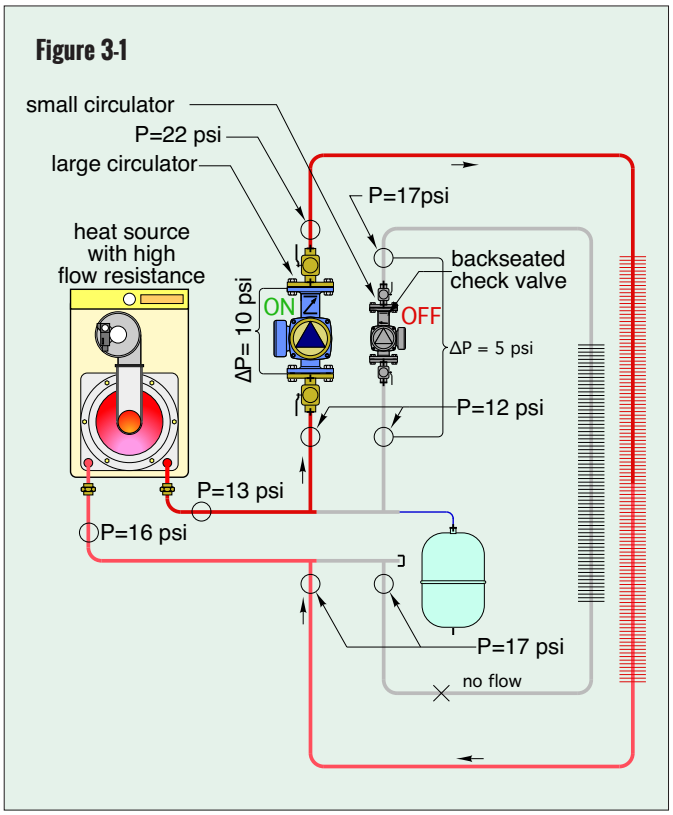
This section discusses several common piping errors, explain why they create problems, and shows at least one way to avoid problems through alternate design.

LACK OF HYDRAULIC SEPARATION

Most modern hydronic systems have more than one circulator. Whenever multiple circulators in the same system operate at the same time, the *potential* for undesirable interaction between them exists. The pressure dynamics created by one circulator can interfere with the pressure dynamics created by another circulator. This undesirable interaction can lead to problems such as insufficient or unpredictable flow rates under certain conditions, cavitation, and even "deadheading" of a small circulator due to the influence of a larger circulator. These problems are caused by a lack of a desirable effect called hydraulic separation.

The system in figure 3-1 is an example of what can happen when two circulators are not hydraulically separated. For simplicity, assume that the piping circuits are all in the same horizontal plane.

Figure 3-1

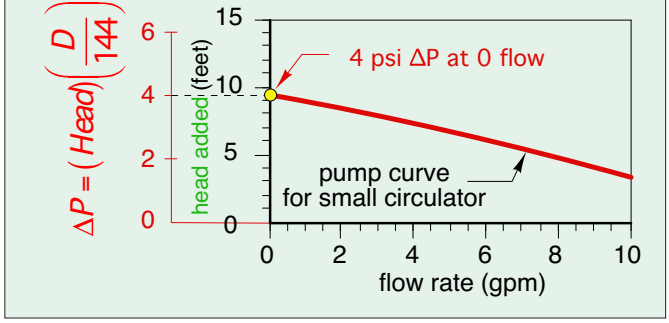


In this system, the larger circulator is sized to move sufficient flow through the higher flow-resistance circuit including a heat source having high-flow resistance. When operating, the flow created by the larger circulator creates a pressure drop of 5 psi between the (gray) supply and return headers connected to the heat source.

The pressures shown in Figure 3-1 are those established when the larger circulator is operating and the smaller circulator is off. Because there is no flow in the circuit containing the small circulator, the 17 psi pressure at the return end of this circuit is present around the circuit and thus present at the discharge port of the smaller circulator. This creates a *reverse* pressure differential of (17-12) = 5 psi across the small circulator, which forces its internal check valve closed.

Figure 3-2 shows the pump curve for the smaller circulator. Notice that the maximum possible differential pressure this circulator can create is 4 psi.

Figure 3-2



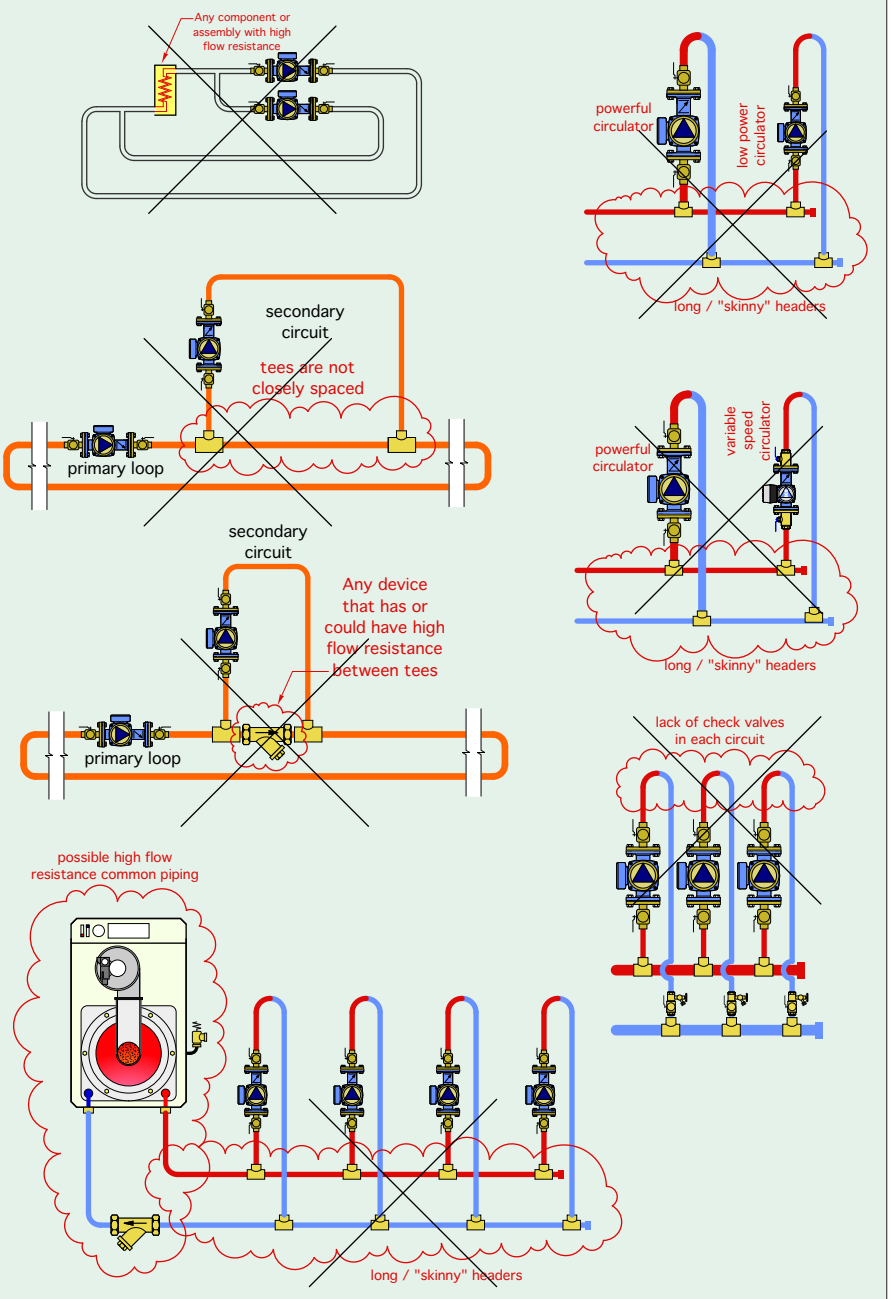
Because the reverse differential pressure of 5 psi is greater than the maximum forward differential pressure of 4 psi, the smaller circulator cannot create flow, even though its impeller is spinning at normal speed. Under this condition, the internal check valve in the smaller circulator remains closed, and the circulator is said to be "deadheaded." There is no flow through the smaller circulator. It will dissipate its full electrical input power as heat. This heat will be absorbed by the water in the circulator's volute, as well as be dissipated by the circulator's body. Although this is not a condition that should be allowed by proper design, small wet-rotor circulators can usually withstand such operation for several hours. This undesirable and likely unanticipated operating condition can be avoided by ensuring that hydraulic separation exists between the circulators in the system.

PIPING DETAILS THAT *PREVENT* HYDRAULIC SEPARATION

The following design details are *INCORRECT*. If present in systems, these details can compromise or eliminate *desirable* hydraulic separation.

- Any piping detail or component that creates high head loss through piping that is shared by two or more circuits, each having a circulator, and where the circulators could operate simultaneously.

Figure 3-3



- The lack of two closely spaced tees when coupling a secondary circuit to a primary circuit in a primary/secondary piping system.

- Installing any device that creates head loss between two tees that are supposed to be installed as close as possible to each other where a secondary circuit connects to a primary loop.

- Long/undersized headers that supply multiple zone circulators.

- High flow resistance heat sources that constitute a common flow path for two or more circuits each of which have circulators capable of operating simultaneously.

- A large (more-powerful) circulator supplied by the same undersized header as a smaller (less-powerful) circulator.

- A large (more-powerful) circulator supplied by the same undersized header as a variable-speed circulator.

- The lack of check valves in two or more circuits with independently operated circulators.

Figure 3-3 shows several configurations that lack hydraulic separation and thus should be avoided.

ACHIEVING HYDRAULIC SEPARATION

A hypothetical system that provided "*perfect*" hydraulic separation of multiple circulators would allow each circulator to create a pressure dynamic that is *completely unaffected by the operation of any other circulator in the system.* Each circuit would operate as if it was completed isolated from any other circuit in the system.

Unfortunately, *perfect* hydraulic separation is not possible in any real system. However, it is possible

to provide *sufficient* hydraulic separation of the circulators so that any interaction between them is of no practical consequence to the overall system performance.

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 interfere with each other.*

See *idronics* #15 for a complete discussion on methods for achieving hydraulic separation.

LACK OF DIFFERENTIAL PRESSURE CONTROL

Many hydronic heating and cooling systems use valves for zoning. Some use electrically operated valves, others use thermo-electric zone valves, and still others use non-electric thermostatic valves. Regardless of the type of valve used, it's important to control changes in differential pressure across the supply and return headers as the valves open and close.

The operating condition of the circulator in a system with zone valves can vary widely depending upon how many zone circuits are present and how many zones are active.

Consider the simple 4-zone system shown in figure 3-4.

The pump curve of the fixed-speed circulator is shown in red on the graph.

When all four zone valves are open, the flow resistance of the distribution system is relatively low, represented by the dark blue head loss curve. This curve crosses the red pump curve at a point that corresponds to a flow rate of approximately 9 gpm.

When one of the zone valves closes, the head loss of the distribution system increases, as represented by the light blue curve labelled "3 zones on". The intersection between this curve and the circulator's pump curve (e.g., the circuit's operating point) has moved to the left and *upward* relative to the previous condition when all four zones were operating.

The movement to the left implies that the flow rate through the circulator has decreased. The upward movement indicates that the head at which the circulator operates has increased. The latter also implies that the differential pressure across the circulator has increased. This higher differential pressure is exerted on the three active zones. *This higher differential pressure causes an increase in flow rate through each active zone.*

A similar response occurs when another zone valve closes, as represented by the light blue curve labelled "2 zones on". The differential pressure across the circulator increases more, and so does the flow rate in each of the two active zones.

When only one zone is active, the differential pressure across the circulator is even higher.

These increases in differential pressure as zone valves close can lead to problems, including:

Figure 3-4

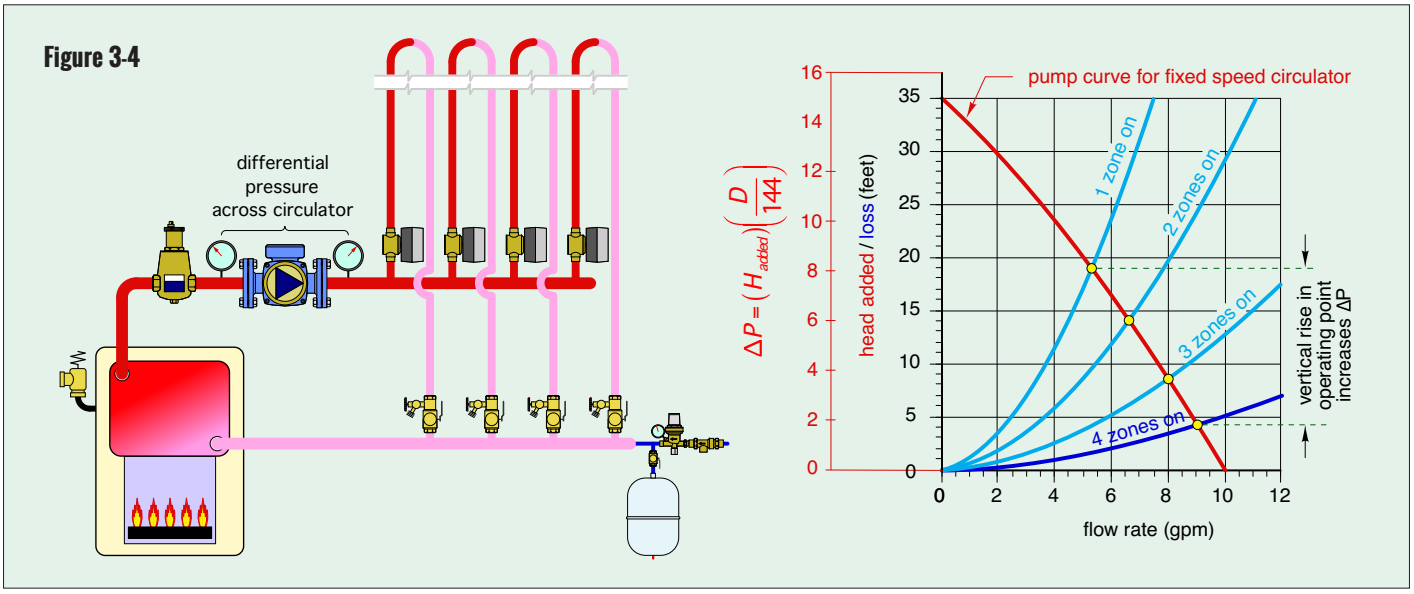


Figure 3-5a

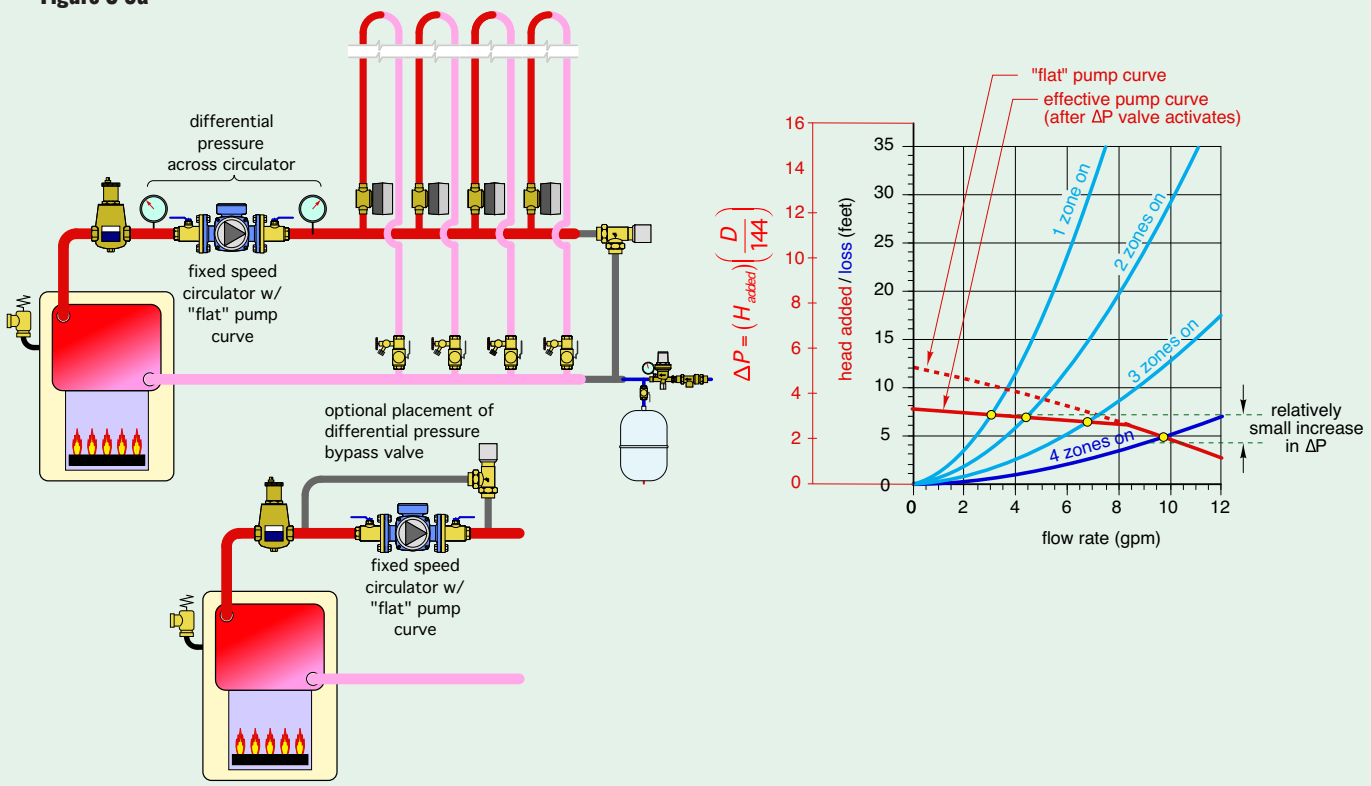
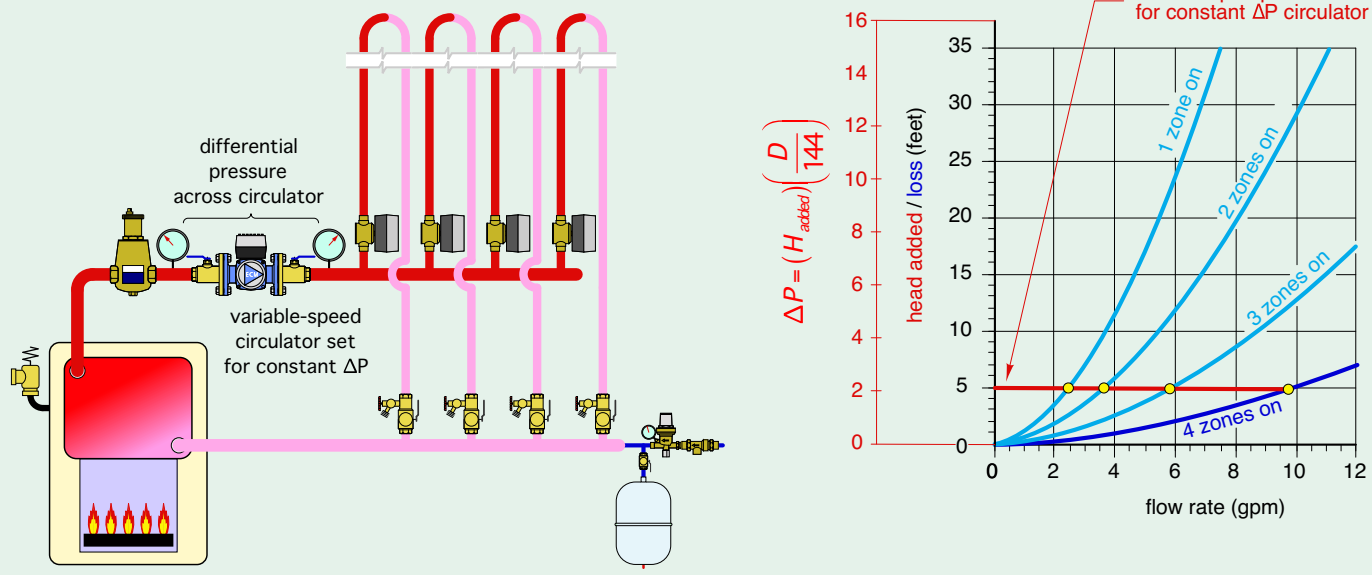


Figure 3-5b



• Increased flow noise in active zone circuits due to high flow velocity.

• Potential erosion of metal in pipe fitting or valves due to high flow velocity, such as over 5 feet per second for small sizes of copper tubing.

• The possibility that the spring inside an *inactive* zone valve cannot hold the valve's paddle fully closed, allowing flow through one or more zones that are supposed to be off.

• Potential cavitation of the circulator depending upon system static pressure, water temperature, and the location of the expansion tank relative to the circulator.

The large and undesirable increase in differential pressure occurs because the design doesn't include a provision to limit or prevent it. There are several ways to do this:

1. If a fixed speed circulator is used, select one that has a relatively "flat" pump curve and install a differential pressure bypass valve as shown in figure 3-5a. The differential pressure bypass valve should be set 0.5 to 1 psi higher than the differential pressure across the circulator when all zones are operating (e.g., 0.5 to 1 psi higher than the lowest differential pressure across the circulator).

2. Use a multi-mode variable-speed circulator set for constant differential pressure mode (e.g., constant ΔP), as show in figure 3-5b.

See *idronics* #36 for a detailed discussion on differential pressure control in systems with valve-based zoning.

LACK OF BALANCING PROVISIONS

Many hydronic heating and cooling systems have multiple branch (or zone) circuits supplied by a common circulator. If all the branch circuits had identical construction, *and* if the headers that connect the branches (or zone circuits) were configured for reverse return flow, then the flow rate through each branch would be similar — *but not necessarily identical.*

Such hypothetical systems would be the exception to typical design requirements. Most systems don't have identical branch or zone circuits. There will be differences in pipe size, pipe length, fittings used, components installed with each branch, etc. Each branch is also likely to require a different flow rate.

Given the range of physical variations for branch or zone circuits, and different flow rate requirements, most systems require a means of adjusting the flow resistance of each branch or zone so that the intended flow rates are achieved. This process is called balancing.

Unbalanced systems can create insufficient heat delivery from some circuits and excess heat delivery from other circuits. This condition

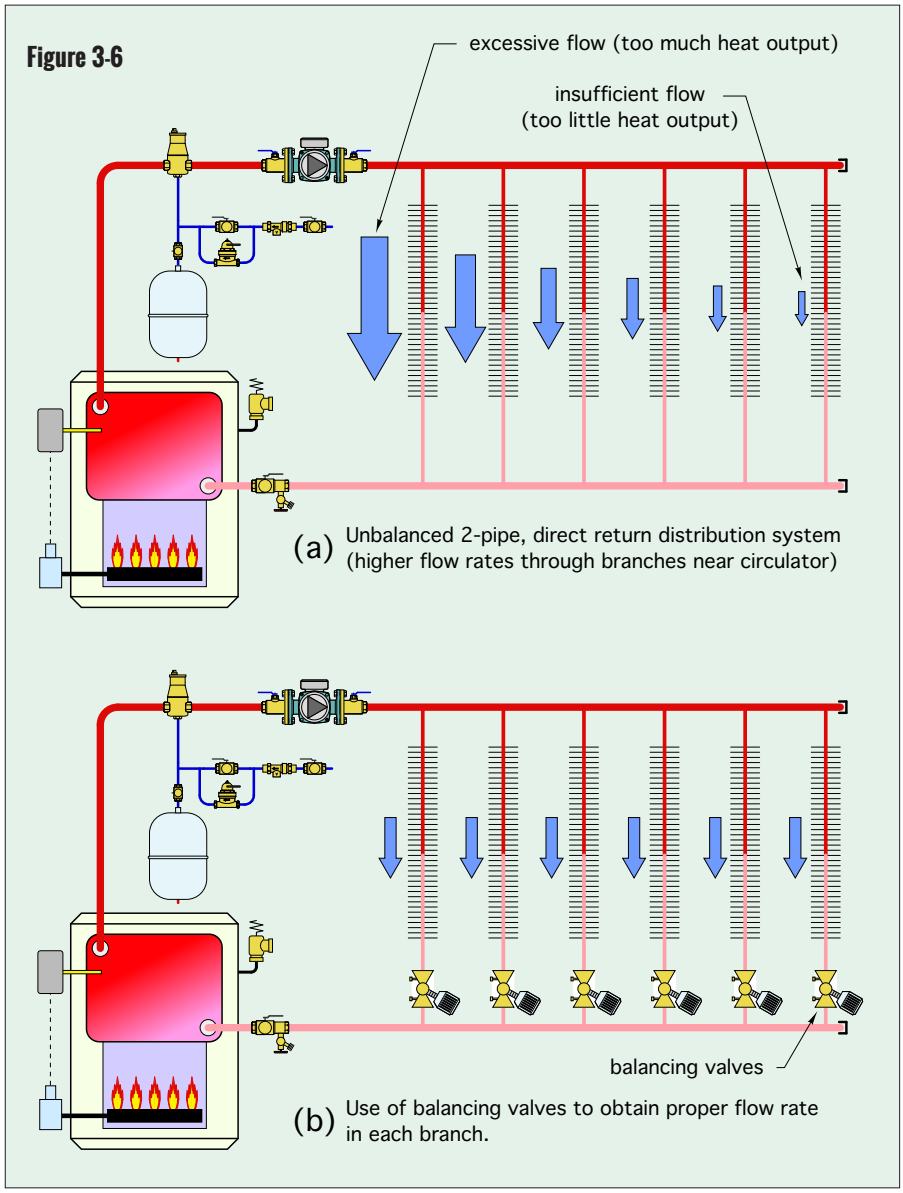


Figure 3-6

is relatively common, especially if the designer has not included balancing devices within the system. These devices range from simple manually set valves to devices that automatically adjust their flow resistance to maintain a specific flow rate.

Figure 3-6 shows a comparison of a simple system in an unbalanced as well as a balanced state.

Assume that the intent is to create equal flow rates in each branch. In the unbalanced state, the arrows indicate wide differences in flow rate among the branches, with higher flow rates through branches near the circulator and progressively lower flow rates in branches that are farther removed from the circulator.

The flow resistance of each balancing valve is adjusted by setting it to a specific Cv value, or by reading the flow rate directly on the valve. When the balancing process is complete, the excess flow through branches close to the circulator is reduced, while flow through the branches farther from the circulator is increased. In this scenario, the goal of achieving equal flows through all branches is accomplished.

There are several approaches available for balancing depending on the complexity of the system, the type of balancing devices used, and the engineering specifications for the system.

See *idronics* #8 for hydronic balancing fundamentals, and *idronics* #34 for balancing valve options.

INCORRECT PRIMARY/SECONDARY PIPING DETAILS

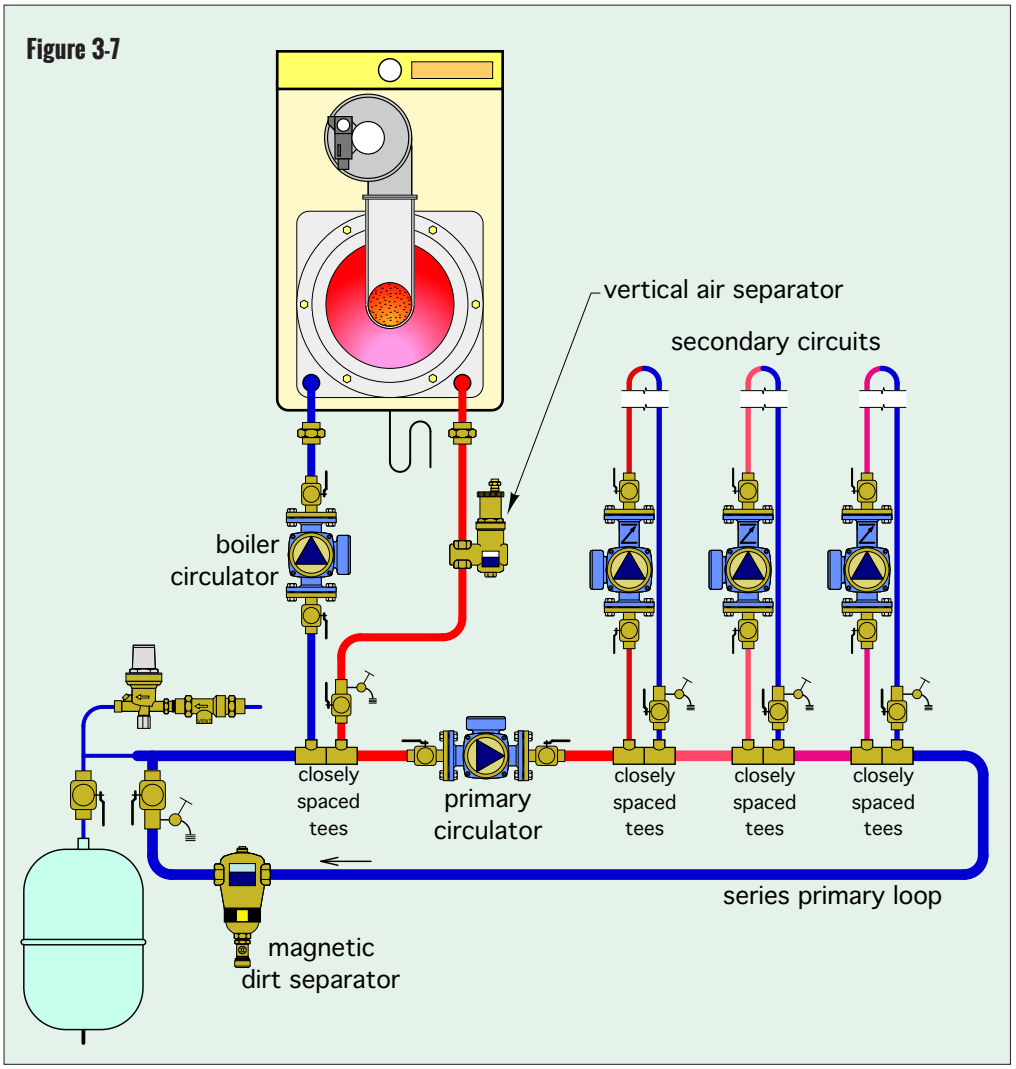
Primary/secondary piping has been used for decades in both large and small hydronic heating and cooling systems. The underlying concept is to have two or more secondary circuits, each with their own circulator, supplied from a common primary loop with its own circulator. The heat source(s) or chillers(s) for the system are connected to the primary loop and thus can provide heated or chilled water to any of the secondary circuits connected to that primary loop.

Figure 3-7 shows an example of a *series* primary/secondary piping system.

The distinctive design detail in a primary/secondary system are the pairs of closely spaced tees used to join each secondary load circuit, as well as the boiler, to the primary loop. By installing the two tees as close to each other as possible, the head loss (and pressure drop) between them approaches zero. This greatly reduces (but doesn't eliminate) the potential for flow through the secondary circuit when its circulator is off.

Although the fundamentals of using closely spaced tees to hydraulically uncouple secondary circuits from the

Figure 3-7



primary loop are valid, there are related design details that must be considered to maximize both the hydraulic and thermal separation of the circuits.

The following design considerations should be verified:

Design detail 1: *The tees that make up each pair should be installed as close together as possible.* The closer the side ports of the tees are to each other, the lower the head loss between them, and the lower the incentive for undesirable flow to develop in the secondary circuit. The center-to-center measurement of the tees should not exceed four pipe diameters. *Any intentional separation between the tees that make up the pair should be avoided. Installing a ball valve between the tees for use in purging should also be avoided.*

Design detail 2: There should be a minimum of 8 pipe diameters of straight pipe upstream of the first tee in a pair, and at least 4 pipe dimeters of straight pipe downstream of the second tee. These distances help suppress turbulence that degrades the hydraulic separation function of the tees.

Design detail 3: The circulator in each secondary circuit should push flow *into* the secondary circuit. This allows the pressure differential created by the secondary circulator to add to the static pressure in the secondary circuit.

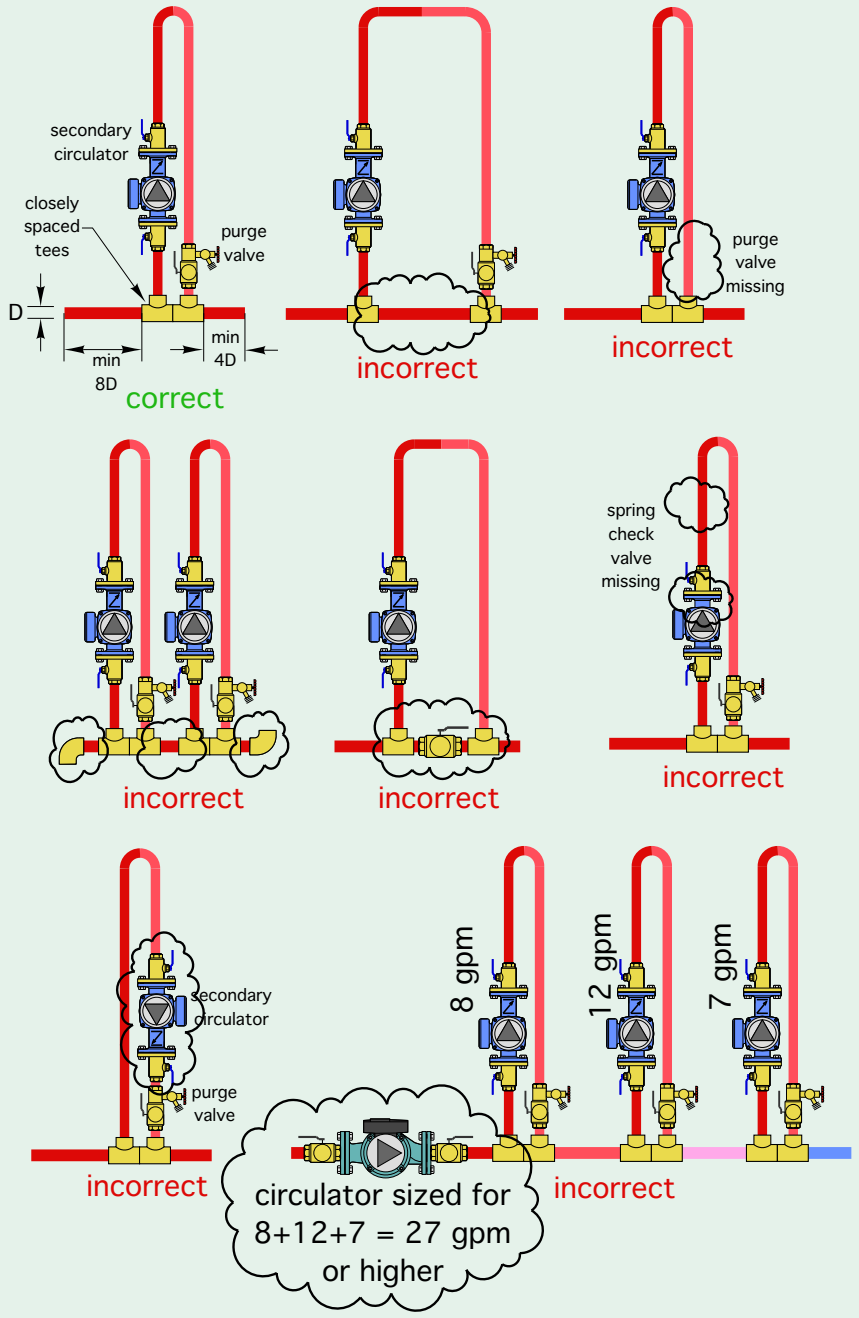
Design detail 4: There should be a spring-loaded check valve on the supply side of each secondary circuit to prevent undesirable thermosiphoning through the circuit when the secondary circulator is off. A spring-loaded check valve *within the secondary circulator* can provide this detail. So can a spring-loaded check valve mounted at least 10 pipe diameters downstream of the secondary circulator.

Design detail 5: There should be a purging valve on the return side of each secondary circuit near the downstream tee.

Figure 3-8 illustrates one correct secondary circuit design that incorporates these details. It also shows several piping configurations that are ***incorrect***.

Additional design errors that can occur with primary/secondary systems include:

Figure 3-8



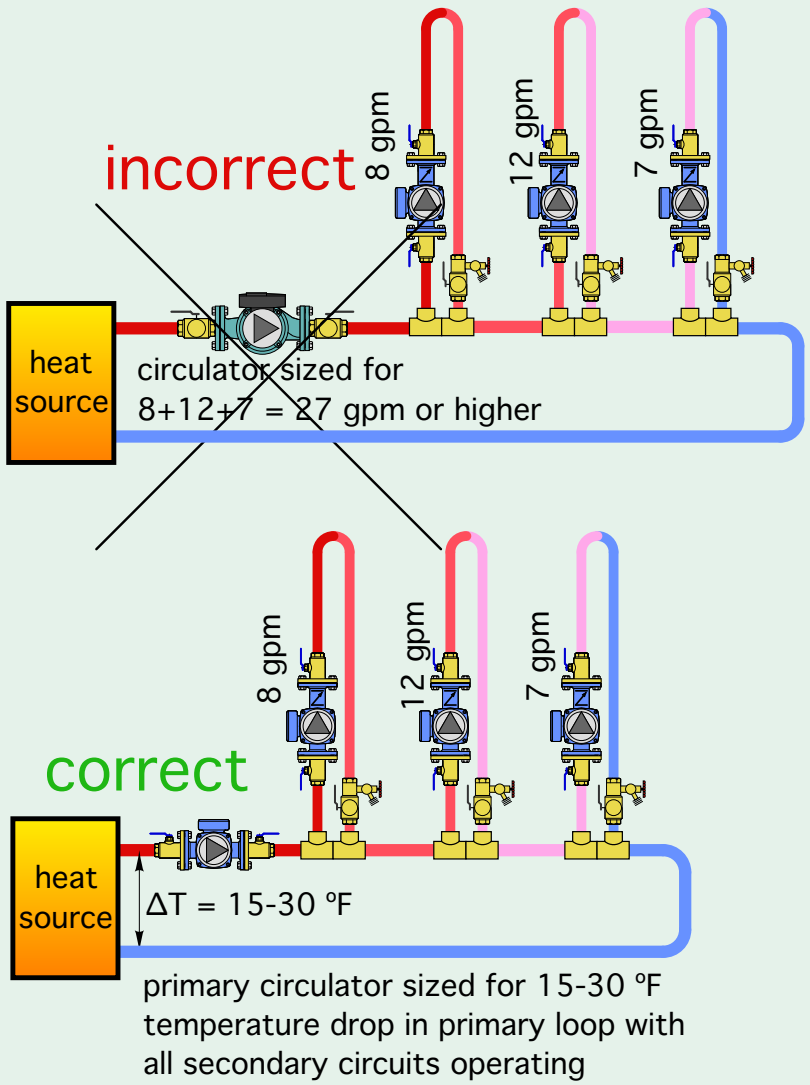
Design error 1: *Assuming that the flow rate in the primary circuit must be at least as great as the sum of the flow rates in all the secondary circuits.* This is not true and often leads to oversized circulators, adding to both installation and operating cost. The primary circulator should be sized so that the overall temperature drop around the primary loop is in the range of 15-30ºF when all secondary circuits are operating at design load conditions. The larger overall temperature drop is appropriate when the secondary circuits have a wider range of required supply water temperature. The smaller temperature drop should be used when all the secondary circuits have similar supply temperature requirements.

The required primary loop flow rate for a given primary/secondary system can be determined using formula 2-4:

Formula 2-4:

$$f_{primary} = \frac{Q_{total}}{(8.01Dc)(\Delta T)}$$

Figure 3-9



Where:

$f_{primary}$ = required flow rate in primary loop (gpm)
Q_{total} = total heat dissipated by all secondary circuits on the loop (Btu/hr)
ΔT = chosen temperature drop around the primary loop with all secondary circuits operating (typically 15-30ºF) (ºF)
D = density of the fluid (lb/ft^3)
c = specific heat of the fluid (Btu/lb/ºF)
8.01 = constant required by the units used

For water at approximately 140ºF, the value of (8.01DC) can be taken as 491. If antifreeze is used, or water at temperatures more than 20ºF above or below 140ºF is used, look up the density and specific heat of the fluid at the approximate average temperature of the primary loop.

For example; a primary/secondary system includes 4 secondary circuits having design load heat outputs of 20,000, 12,000, 6,000 and 7,000 Btu/hr each. The system operates with water at an average primary loop temperature of 140ºF. Determine the primary loop flow rate that will yield a temperature drop of 25ºF when all four secondary circuits are operating.

Solution: The total load on the primary loop is 20,000+12,000+6,000+7,000 = 45,000 Btu/hr. The value of (8.01Dc) for 140ºF average water temperature is 491. Putting these values into formula 2-4 yields:

$$f_{primary} = \frac{Q_{total}}{(8.01Dc)(\Delta T)} = \frac{45{,}000}{(491)(25)} = 3.7\,gpm$$

This primary loop flow rate is quite low, and is the result of selecting a relatively "wide" (25ºF) ΔT for the primary loop. It's likely that a small primary loop circulator could provide this flow rate assuming that the primary loop piping is selected and sized for a relatively low head loss.

Design error 2: *Failure to account for the temperature drop along the primary loop from one operating secondary circuit to another.* This temperature drop is caused by heat being released from each operating secondary circuit. If this temperature drop is not accounted for, the supply water temperature to secondary circuits farther along the primary loop could be significantly lower than the temperature leaving the heat source. This temperature drop can be calculated using formula 2-5.

Formula 2-5:

$$\Delta T = \frac{q_s}{(8.01Dc)f}$$

Where:

ΔT = temperature drop across the closely spaced tees of an operating secondary circuit (ºF)

q_s = the rate of heat transfer to the secondary circuit (Btu/hr)

D = density of the fluid (lb/ft^3)

c = specific heat of the fluid (Btu/lb/ºF)

f = flow rate along primary circuit (gpm)

8.01 = constant required by the units used.

Example: A primary loop supplies water to a set of closely spaced tees leading to a secondary circuit. The upstream tee receives water at 140ºF and a flow rate of 5 gpm. When operating at design load, the secondary circuit dissipates 20,000 Btu/hr. Determine the temperature drop across the closely spaced tees, and the supply water temperature to the next secondary circuit located downstream.

Solution:

$$\Delta T = \frac{q_s}{(8.01Dc)f}\frac{20{,}000}{(491)5} = 8.15^{\circ}F$$

The supply temperature to the next set of tees is 140-8.15 = 131.9ºF

When a primary loop supplies several secondary circuits in series, the temperature drop across each set of tees continues to reduce the supply temperature to any downstream set of tees. *Failure to account for this drop can lead to underheating in downstream secondary circuits.*

Design error 3: *Failure to place higher temperature secondary loads near the beginning of the primary loop, and lower temperature secondary loads near the end of the primary loop.* Since there will be a sequential temperature drop along the primary loop, it is logical to connect secondary circuits in an order that "cooperates" with this temperature drop, as shown in figure 3-10. **However, arranging the loads from higher to lower temperature along the primary loop is *NOT* a substitute for installing proper mixing hardware for lower temperature loads.**

Design error 4: *Not protecting a conventional boiler from sustained flue gas condensation.* Depending on the operating temperature of the heat source and the number of secondary circuits connected to it, there is a possibility that the water returning to the boiler could be below the dewpoint of the flue gases. This would lead to sustained flue gas condensation within the boiler, a condition that must be avoided.

If the temperature drop calculations under design load conditions, and with all secondary circuits operating, imply that the water temperature returning to the boiler is less than 130ºF, there needs to be an anti-condensation detail included in the system.

One of the simplest methods of protecting conventional boilers from sustained flue gas condensation is to include a thermostatic anti-condensation mixing valve such as the Caleffi ThermoProtec™ valve, as shown in figure 3-11.

Figure 3-10

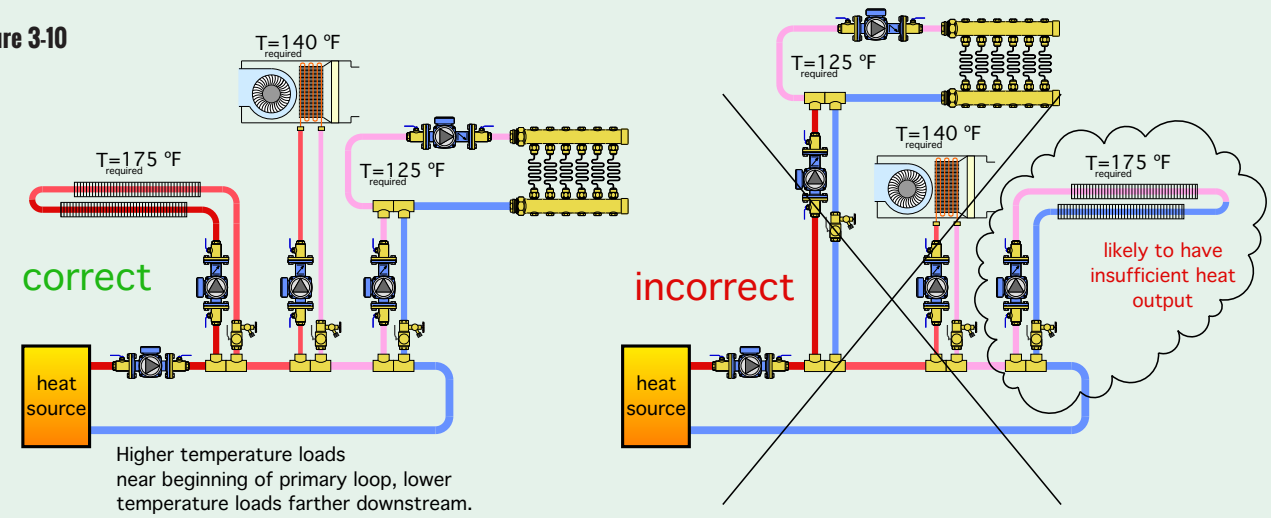
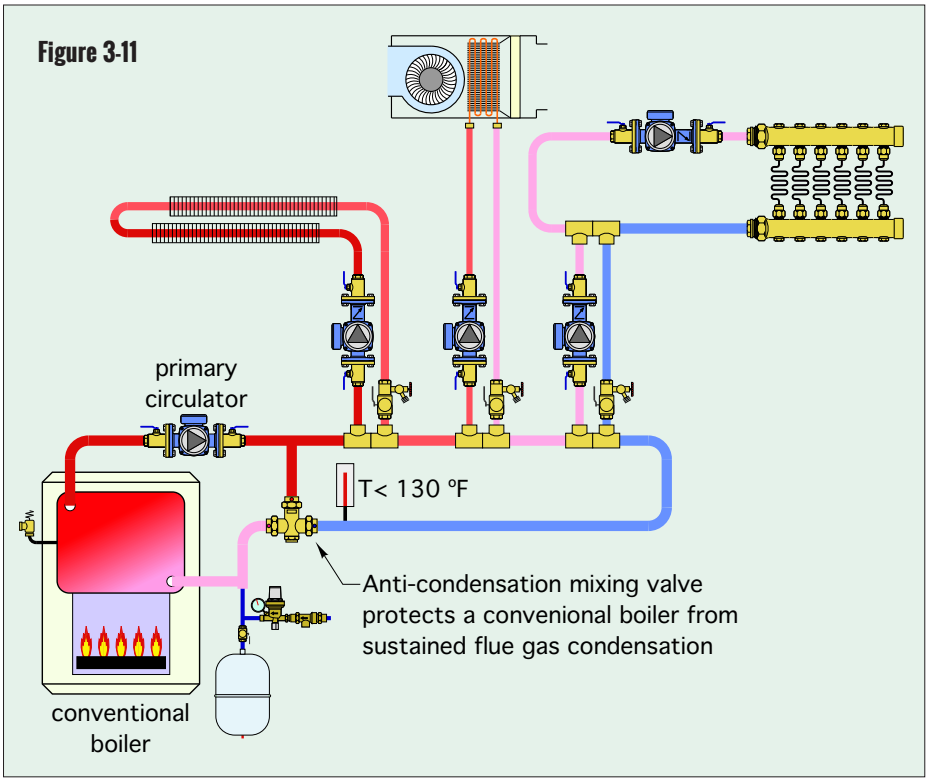


Figure 3-11



The ThermoProtec valve blocks cool water (typically <130ºF) returning from the primary loop from flowing directly into the boiler. At the same time, it "recirculates" warm water leaving the boiler back into the boiler. This allows the boiler to quickly reach a temperature above the dewpoint of the flue gases. At that point, the ThermoProtec valve starts to allow cool water into the valve, where it mixes with hotter water leaving the boiler. If the boiler inlet temperature continues to rise, more return water enters the valve, and an equal flow of hot water flows into the primary loop.

When using an anti-condensation mixing valve, the primary circulator needs to be installed between the boiler and the valve, as shown in figure 3-11.

Design error 5: *Not connecting each secondary circuit with a pair of closely spaced tees.* Designers who understand that sequential temperature drops occur along a primary loop may try to compensate by grouping the tees that supply each secondary circuit together near the beginning of the primary loop, while grouping the return tees from each secondary circuit farther along the primary loop, as shown in figure 3-12.

The designer's *intent* is to supply each secondary circuit with the same (hottest) water temperature before that temperature was reduced by mixing with return flow from one or more secondary circuits. This can work, but only under a specific condition: *The primary loop flow rate must be equal to or greater than the total of all secondary circuit flow rates.* Although this condition is possible, it's often wasteful both in terms of oversized component and operating cost.

When this specific condition is not met, there will be unintentional mixing

Figure 3-12

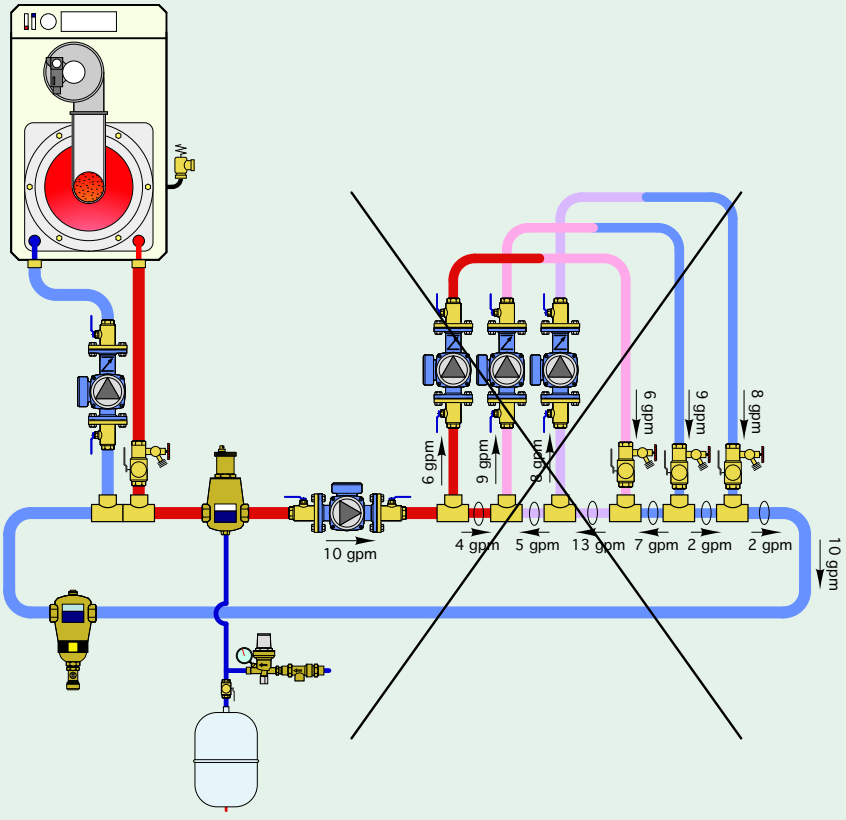
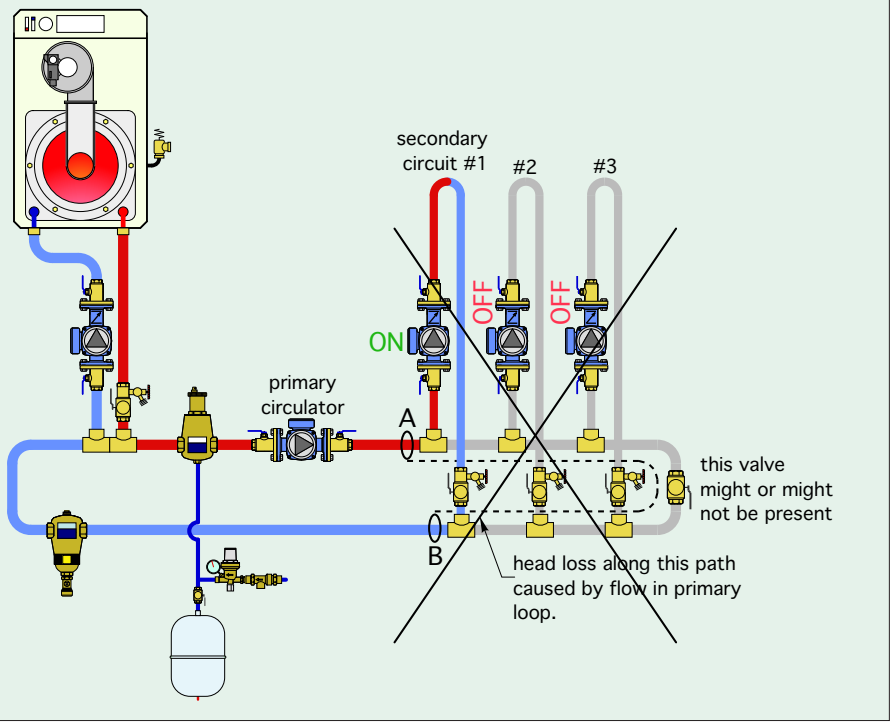


Figure 3-13



in the tees connected to the primary loop ,as shown in figure 3-12. The flow rates shown are arbitrary, but their sum is lower than the flow rate in the primary loop. The result is mixing, which can greatly reduce the supply temperature to some (or all) of the secondary circuits.

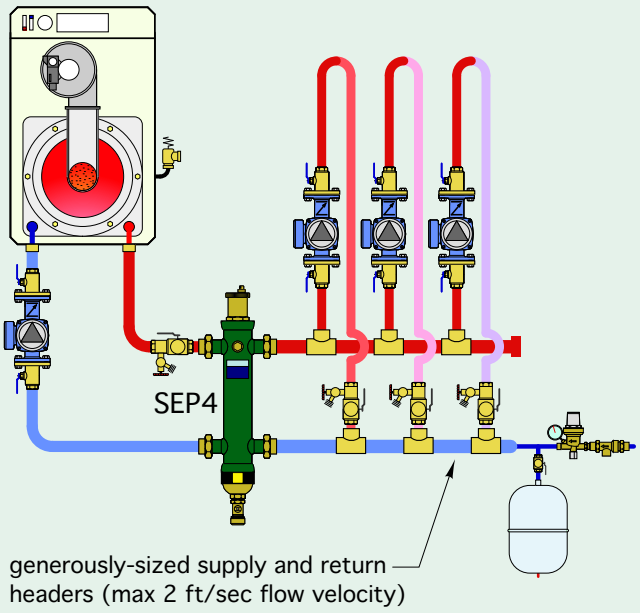
The principal that applies is simple: *The total flow entering any tee must equal the total flow leaving that tee.* Applying this principal to each of the circuits shown in figure 3-12 shows where flow reversal and mixing will occur. The temperatures that result from this mixing will vary depending on which secondary circuits are operating and what the water temperatures in each circuit are at any time. It's a situation that will produce a wide range of flow and temperature variations and should always be avoided. A verification test for a proposed primary/secondary system is simple: *If the tees that connect each secondary circuit to the primary loop are not adjacent to each other, the resulting piping configuration is NOT a primary/secondary system.*

Design error 6: *Mixing a primary/secondary approach with a 2-pipe header approach.* The piping configuration shown in figure 3-13 is neither a primary/secondary system, nor a 2-pipe header system.

This piping arrangement likely results from the designer's intent of keeping equal supply water temperature to each secondary circuit. In that regard, it is subject to the same variations in flow and mixing as the system in figure 3-12. If the primary flow rate was always at least as high as the sum of the flows for the operating secondary circuits, this problem would be alleviated.

However, another problem will occur. Consider a situation where the primary circulator and the circulator for secondary circuit #1 are operating. There will be flow in the primary loop from point A to point B. This flow creates head loss and pressure drop between these points (e.g., the pressure at point A will be higher than the pressure at point B). The value of this pressure difference depends on the piping size and length, number of fittings, and any valves present between points A and B, as well as the primary loop flow

Figure 3-14



rate. This pressure difference will be "felt" by the other two secondary circuits that are not operating. If the pressure differential between points A and B is greater than the forward opening pressure of the spring check valve in the secondary circuits, which is typically about 0.5 psi, some flow will "push through" the check valves and proceed through the secondary circuits that are supposed to be off. This leads to unintentional heat output in these secondary circuits, as well as possible mixing and supply temperature drop in secondary circuit #1. ***This is another piping configuration that must be avoided.***

AN ALTERNATIVE TO PRIMARY/SECONDARY PIPING

Although primary/secondary piping systems can work if they are carefully designed based on the previously discussions, there are simpler ways to achieve equal or better results by using a hydraulic separator rather than pairs of closely spaced tees. Figure 3-14 shows an example.

This system uses a Caleffi SEP4™ hydraulic separator combined with short and generously sized headers. The SEP4 constitutes the "common piping" for all 4 circuits in this system (e.g., three load circuits and the boiler circuit). The pressure drop though the SEP4 is very small, and thus it provides excellent hydraulic separation of all the circulators. It also provides air, dirt and magnetic particle separation for the system, eliminating the need for a dedicated air separator and dedicated dirt/magnetic particle separator. It also eliminates the need for the primary circulator shown in figure 3-13.

The headers on the load side of the hydraulic separator *provide the same supply water temperature to each load circuit.* These headers should be sized for a maximum flow velocity of 2 feet per second. They should also be as short as possible. These details complement the low pressure drop through the hydraulic separator.

Compare the simplicity of the system shown in figure 3-14 to some of the previously discussed systems using primary/secondary piping. Fewer components, and multi-function components, reduce both installation time and cost, while providing equal supply water temperatures to each load circuit.

See *idronics* #15 for more information on hydraulic separators.

PREVENTING THERMOSIPHONING

A fundamental principal of fluid mechanics is that hot water is less dense than cool water. Stated another way, hot water is more "buoyant" than cool water. Prior to the invention of circulators, this principal was used to create flow in hydronic systems. Water heated within a boiler would rise upward through the piping system, cool as

Figure 3-15

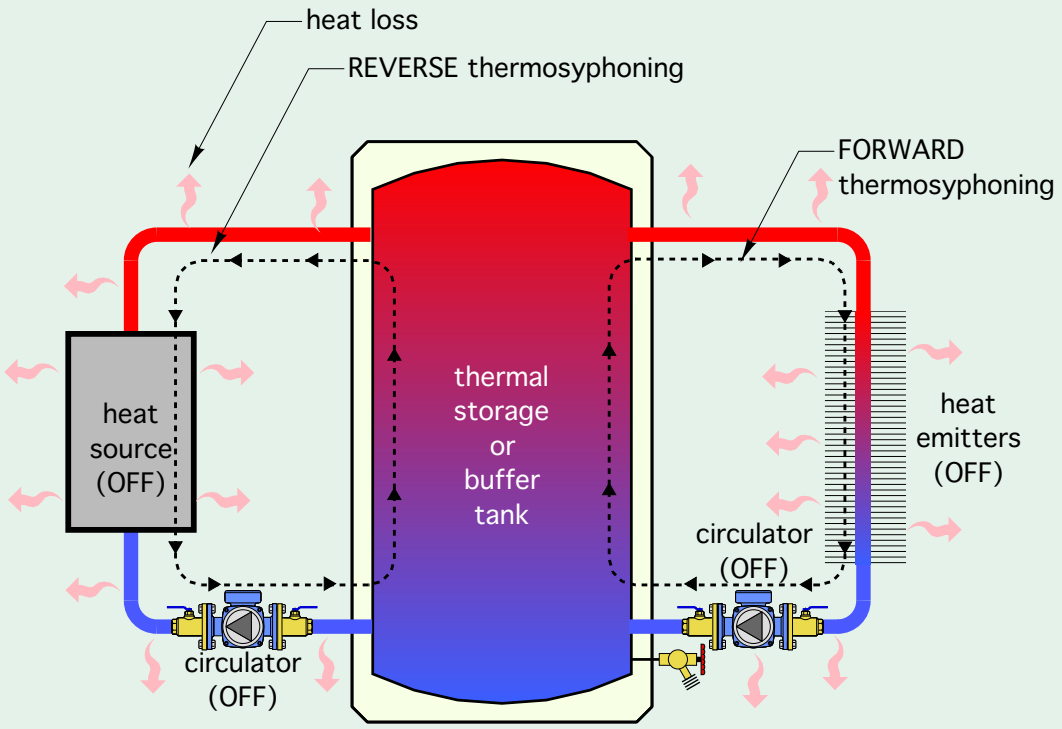
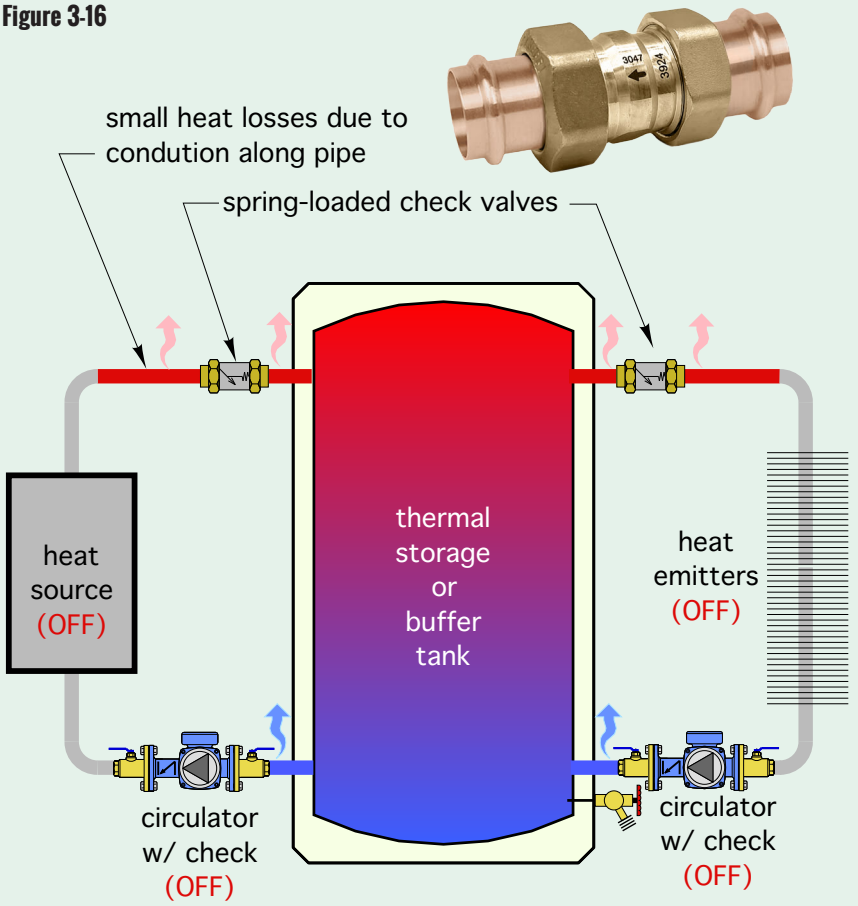


Figure 3-16



it released heat at radiators, and then "sink" downward through piping leading back to the boiler's inlet. System's that used this principle were examples of good design given the resources available at the time.

Modern systems use circulators to create flow. However, the principle of buoyancy still applies, and when ignored, it can create unintentional and undesirable flow and heat transfer.

One common situation where buoyancy-driven flow can create problems is when a thermal storage tank is present, such as shown in figure 3-15.

When the heat source is off, any unblocked piping path from the upper portion of the tank to the lower portion allows a weak but *persistent* flow to develop. This is called *reverse* thermosiphoning since the direction of flow is opposite that of the intended flow when the circulator is operating. This flow is caused by the increasing density of water at it cools through piping, fittings and the inactive heat source. Although the actual flow rate is low compared to the flow when the circulator is on, over time reverse thermosiphoning can dissipate significant amounts of heat from the tank.

A similar situation will occur through any unblocked piping path through the heat emitters. This is called *forward* thermosiphoning because the direction of flow is the same as when the circulator is operating. Forward thermosiphoning can also drain heat from the tank and dissipate it in areas of the building where it is not expected or needed.

The higher the system's piping rises above the source of the heated water, the stronger the thermosiphon effect can become.

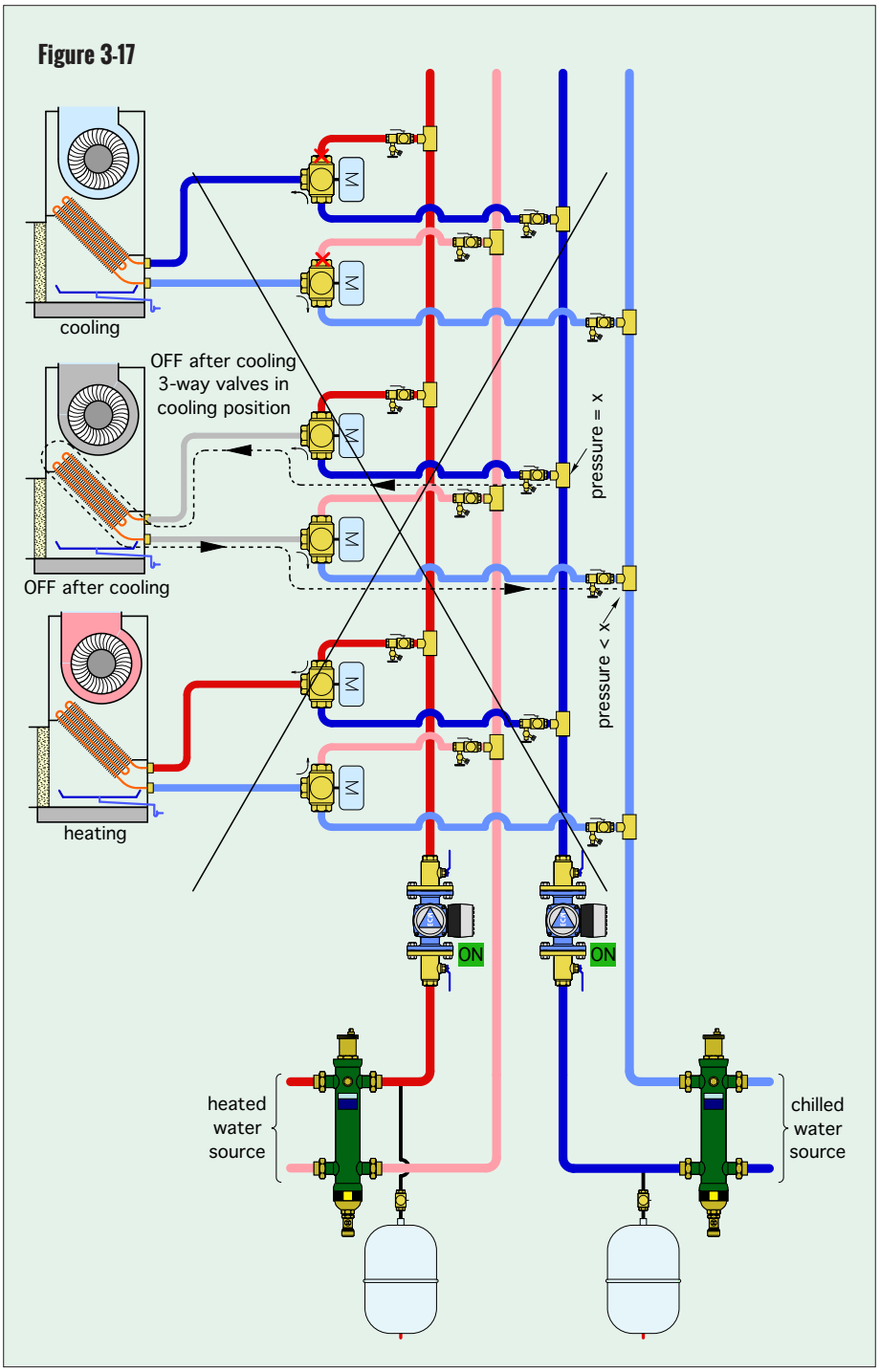
Forward and reverse thermosiphoning can be prevented by devices that block the flow path when the circulator for that path is inactive. A spring-loaded check valve is often used for this since it can block both reverse and forward thermosiphon flow. In the latter case, the spring in a typical spring check can provide 1/3 to 1/2 psi forward opening resistance, which is usually sufficient to stop forward thermosiphoning in low-rise systems. Figure 3-16 shows where spring-loaded check valves can be installed to block both reverse and forward thermosiphoning.

Even with the flow paths blocked, there will still be some heat loss from the tank due to conduction heat transfer along the piping. Placing the check valve close to the tank will reduce (but not eliminate) this effect. If the circulators are installed as shown in figure 3-16, they should also include integral check valves that help impede conduction heat loss along the piping. Always install the check valves with the arrow on the valve body pointing in the direction of intended flow.

Tank heat loss should also be minimized by insulating the piping and components in the circuit. This reduces standby heat loss, which in turn reduces the buoyancy effects that create thermosiphoning.

UNANTICIPATED FLOW IN 4-PIPE HEADER SYSTEMS

Some hydronic distribution systems, especially those used in commercial buildings, provide heating and cooling. When the designer



wants each air handler to operate independently in either heating or cooling, a 4-pipe distribution system is commonly used. Two of the four pipes are for heated water supply and heating return. The other two pipes are for chilled water supply and return. Flow through the heated water circulation system is typically handled by a variable-speed circulator. Another variable-speed circulator provides flow through the chilled water circulation system.

Each air handler connected to a 4-pipe distribution system will be in one of three modes: Heating, cooling or off. *It's important to prevent flow through any air handler that is off.* One approach that does NOT provide such isolation is a combination of two 3-way diverter valves, as shown in figure 3-17.

The problem with this approach is that the common port in a 3-way diverter valve (e.g., the AB port) is always open to one of two ports, the A port or the B port, and thus cannot block flow through an air handler that's off.

One design that ***does*** block flow through air handlers that are off, while still allowing other units to independently operate in heating or cooling, is shown in figure 3-18.

This system uses two 2-way valves and one 3-way diverter valve for each air handler. When an air handler is off, both associated 2-way valves are closed, which blocks flow from the hot and cold risers. The 3-way diverter valve "steers" return flow to the appropriate return riser. This valve combination also ensures a pressure equalization path back to an expansion tank when the terminal unit is in any of the three modes.

Since flow is blocked through inactive air handlers, the variable-speed circulators for both heating and cooling distribution can reduce speed when some of the air handlers are off. A typical approach with this piping arrangement uses a pressure-regulated ECM circulator set for proportional pressure mode.

It's also possible to use this valve combination when multiple heat pumps are being used to independently supply heated or chilled water to a 4-pipe header system. Figure 3-19 shows this scenario along with another plausible valve combination (four

Figure 3-18

One 3-way valve and two 2-way valves provide complete isolation

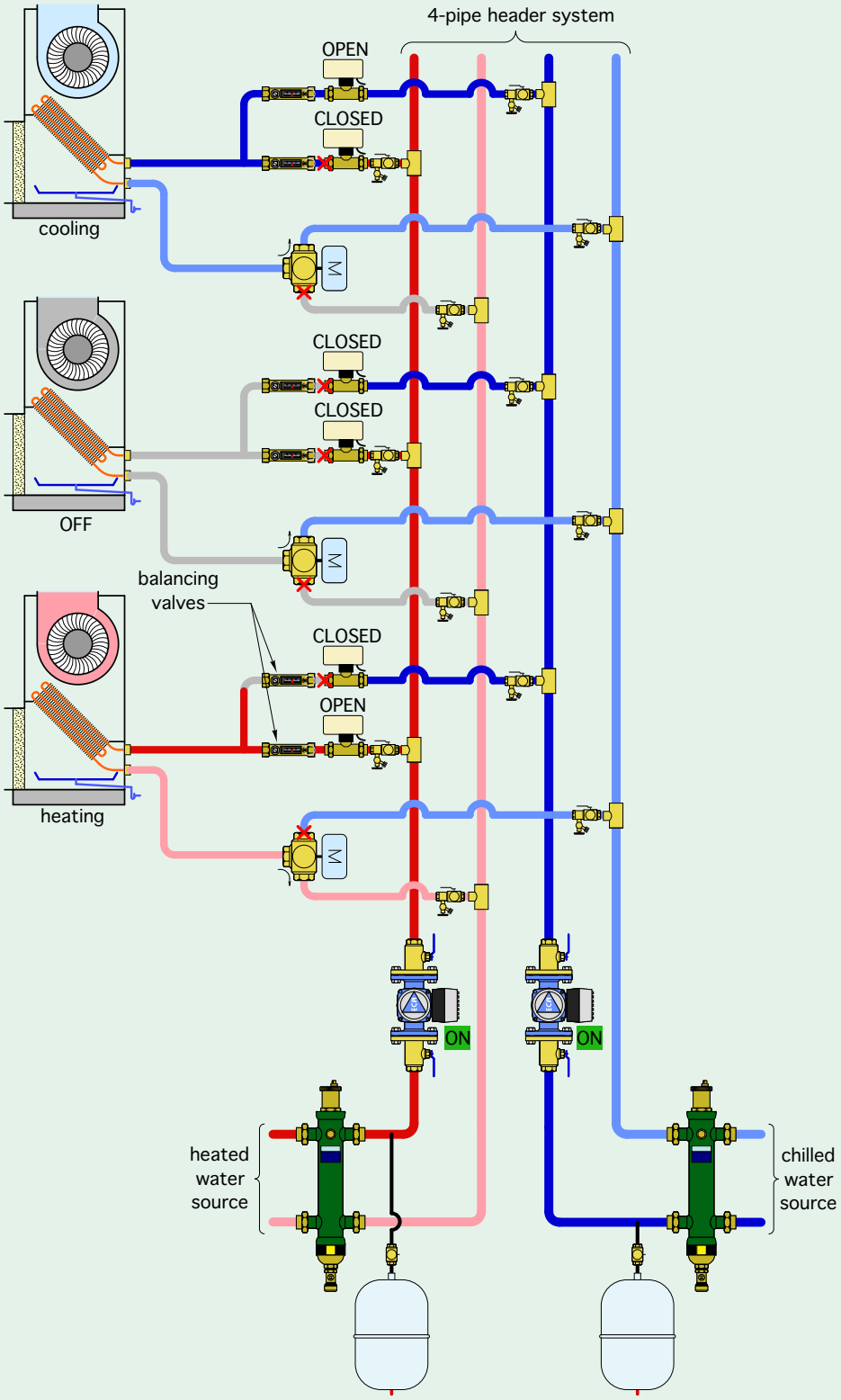
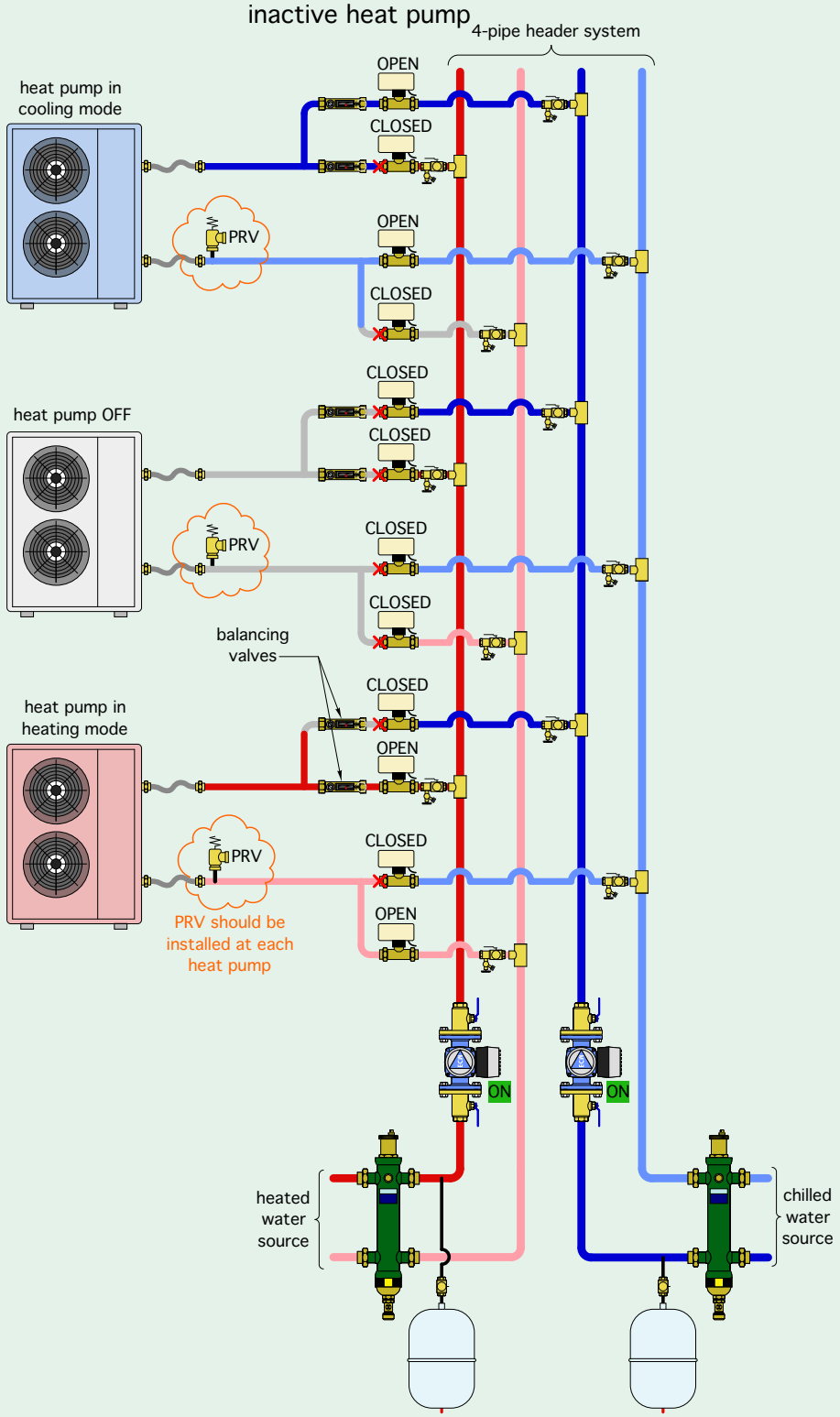


Figure 3-19

Four 2-way valves at each unit provide complete isolation of any inactive heat pump



2-way valves) that can isolate any inactive heat pump.

Although the four 2-way valves combination does isolate inactive heat pumps from either the hot or cold risers, *it also isolates a portion of the overall system at each air handler when that unit is off.* This is generally not a problem for air handlers that *deliver* heating or cooling. However, if this piping and valve combination is used to connect *heat pumps rather than air handlers*, a pressure-relief valve should be installed at each heat pump to prevent over-pressure if the heat pump is unintentionally operated in heating mode when all four 2-way valves associated with that heat pump are closed.

Some air-to-water heat pumps may have internal pressure-relief valves. Some may also have internal expansion tanks, which could absorb possible water expansion if the heat pump was operated with all four 2-way valves closed.

4. DESIGN ERRORS RELATED TO HEAT SOURCES

There are many devices that can serve as heat sources in hydronic systems. They include several types of boilers operating on different fuels, as well as air-to-water and water-to-water heat pumps and solar thermal collectors.

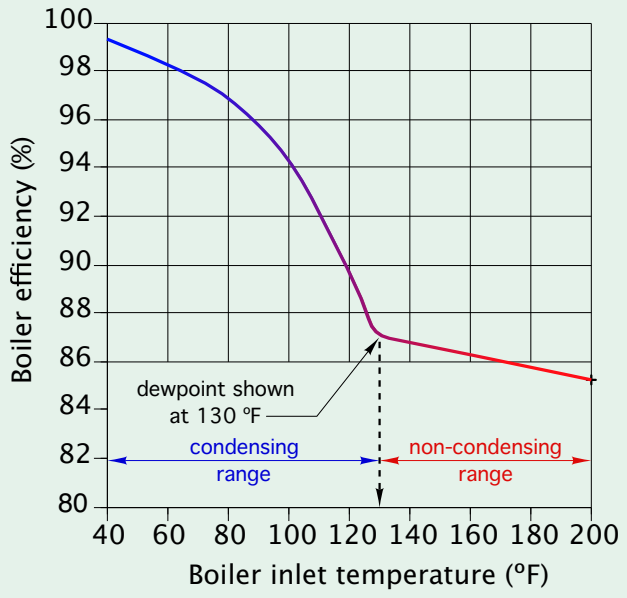
When designing a system, it's important to understand both the thermal and hydraulic characteristics of the heat source and ensure that the remainder of the system provides conditions that allow that heat source to operate at high efficiency over a long service life.

MISAPPLICATION OF MODULATING/CONDENSING (mod/con) BOILERS

Prior to the 1990s, most boilers used for hydronic heating in North America were intended to operate at conditions that did *not* allow for sustained condensation of their flue gases. These "conventional" boilers had heat exchangers made of cast iron, welded steel and finned copper tubing. When combined with heat emitters such as fin-tube baseboard, cabinet convectors, and cast iron or panel radiators, the water returning from the distribution system was usually above the dewpoint temperature of the boiler's flue gases. This allowed the flue gases to pass out of the boiler with condensing.

During the 1990s, a new type of boiler design emerged in the North American market that was largely designed to achieve high thermal efficiency when operating on gaseous fuels such as natural gas or propane. The new design was intended to *promote* flue gas condensation. *When properly matched to low-temperature hydronic distribution systems,* these "condensing" boilers could achieve thermal efficiencies significantly higher than conventional boilers (typically 90-95%versus 80-86% for conventional boilers). The new boilers were also capable of varying their fuel input rate and thus their heat output rates. The combined characteristics of intentional condensing mode operation and heat output modulation lead to the term "mod/con" boiler.

Figure 4-1



The degree to which flue gases can condense within a boiler is largely dependent on the water inlet temperature, as shown in figure 4-1.

When *any* boiler is operated at water temperatures above the dewpoint of its flue gases, condensation does not occur. However*, there is no such thing as a boiler that cannot condense its flue gases given the right operating conditions.* There is no such thing as a "non-condensing" boiler.

Thermodynamic and chemical analyses of combustion demonstrate that thermal efficiencies up to about 87% are possible without flue gas condensation. These conditions tend to occur when the water temperature entering the boiler is approximately 130ºF or higher. This range of operation is called "non-condensing" mode.

When inlet water temperatures are below approximately 130ºF, some amount of flue gas condensation occurs, and with it comes an increase in thermal efficiency. This boost comes from recovering some of the latent heat associated with condensing water vapor produced during combustion back into liquid water.

The lower the water inlet temperature to the boiler, the higher its thermal efficiency, as shown in figure 4-1. For example, at a boiler inlet temperature of 100ºF, figure 4-1 shows a corresponding thermal efficiency of 94%. At a boiler inlet temperature of 80ºF, the boiler's thermal efficiency increases to 97%.

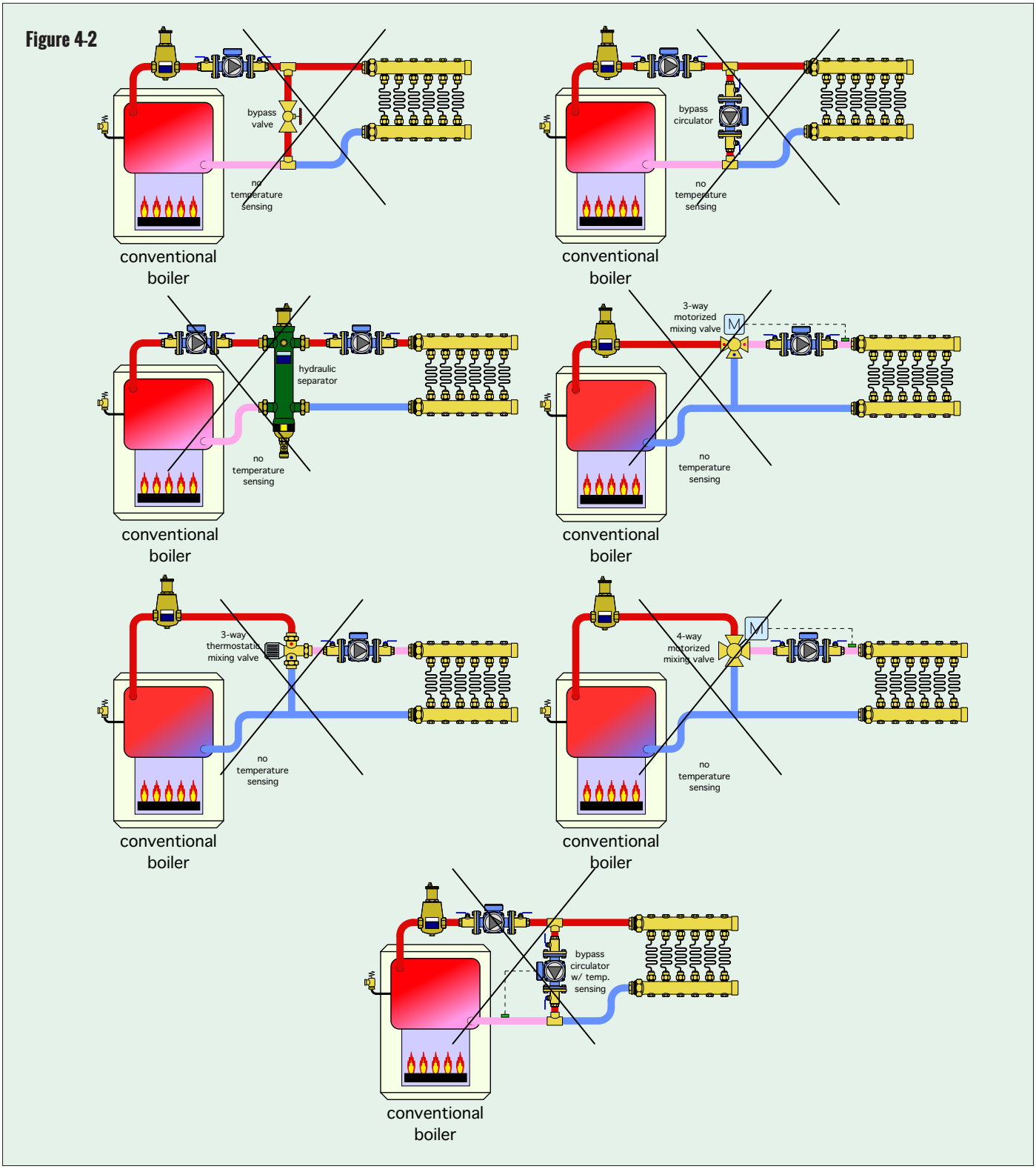
Operating a mod/con boiler at conditions that keep its inlet water temperature high enough to prevent sustained flue gas condensation is possible. However, doing so wastes the potential performance ability of the mod/con boiler and often increases installation cost relative to conventional boilers, especially if the venting system for the conventional boiler already exists.

MISAPPLICATION OF CONVENTIONAL BOILERS

Just as the use of mod/con boilers to supply consistently high temperature loads is poor design practice, so too is the use of "unprotected" conventional boilers to supply consistently low temperature loads. Operating a conventional

boiler at low water temperatures can cause sustained flue gas condensation. The condensate is acidic and will quickly corrode both the combustion side of the boiler's heat exchanger, as well as common galvanized steel vent pipe. Flue gases that manage to make it out of the boiler and through the vent connector can eventually condense within a chimney, leading to degradation of masonry materials and potential structural failure over time.

Figure 4-2



A general design guideline is to ensure that inlet water temperatures to conventional boilers remain above 130ºF *during most the boiler's operating time*. When a conventional boiler operates from a cold start, there will be intermittent flue gas condensation. However, proper design makes this "transient" condition as short as possible.

Conventional boilers should never be *directly connected* to low-temperature distribution systems, such as slab-type floor heating or snowmelting.

There are also long-standing "misunderstandings" about what design details protect a conventional boiler from sustained flue gas condensation. Figure 4-2 shows several piping configurations that "attempt" to provide protection from sustained flue gas condensation but fail to do so under certain operating conditions.

Although some of these piping configurations can boost boiler inlet temperature under specific conditions (i.e., conditions that are at or close to steady state), they all fail to provide anti-condensation protection under transient conditions (e.g., when the system is warming up). *They all lack hardware and control logic that:*

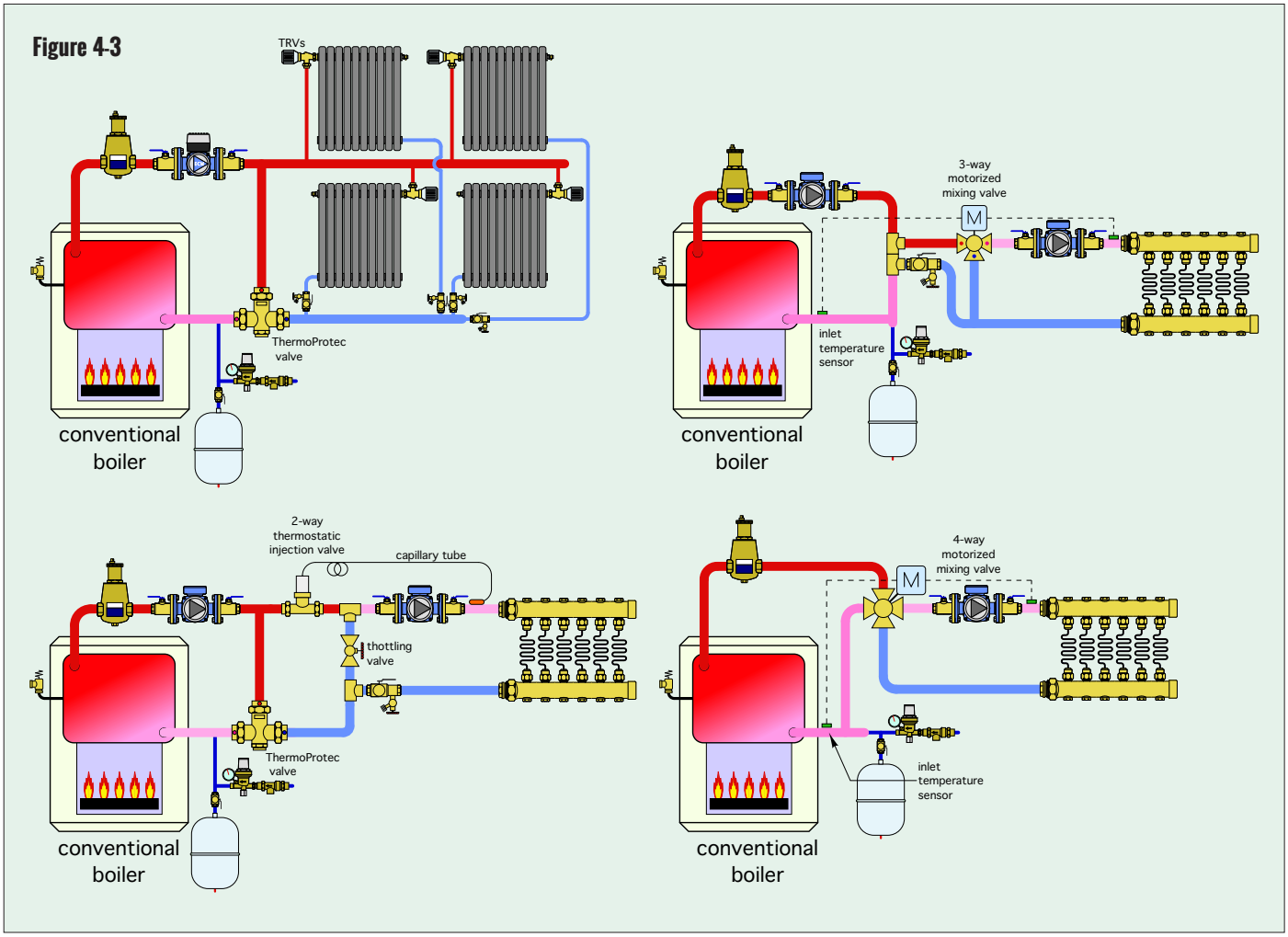
1. Allows the boiler inlet temperature to be continually measured.

2. Can thermally disconnect the boiler from the distribution system when necessary.

Many of these insufficient piping details are based on the premise of steady-state operating conditions, where the boiler is *assumed* to reach some elevated temperature above the dewpoint of its flue gases. Some designers may assume that setting the boiler's high limit control to an elevated setting well above this dewpoint, along with bypassing some flow from the boiler's outlet to its inlet, is sufficient to prevent flue gas condensation. *It's NOT.*

It's important to remember that distribution systems, such as concrete slab floor heating or pavements equipped with

Figure 4-3



hydronic snowmelting, have very high thermal mass. Under startup conditions, that thermal mass can absorb heat much faster than the boiler can produce heat. Under these conditions, thermal equilibrium, *and not the boiler high limit setting*, determines the water temperatures within the system. That temperature will only climb to a value where the heat output of the boiler is matched by heat absorption and dissipation of the distribution system. In many cases, this will be well below the dewpoint of the flue gases.

Figure 4-3 shows several piping details that *DO* provide protection against sustained flue gas condensation. They all have hardware devices and control logic that 1) allows the boiler inlet temperature to be continually measured, and 2) are capable of thermally disconnecting the boiler from the load when and as necessary to keep the inlet water temperature above the dewpoint of the flue gases.

OPERATING HYDRONIC HEAT PUMPS AT HIGH TEMPERATURES

Current-generation hydronic heat pumps (air-to-water or water-to-water) operating on R-32 or R-454B refrigerant are capable of water temperatures up to about 150ºF. However, at these temperatures, the coefficient of performance (COP) is not very high (typically 1.2 to 1.8, depending on other operating conditions).

Matching these heat pumps to heat emitters that were designed to operate at high water temperatures (180ºF+) under design load, in applications where the heat pump is the sole heat source, will not satisfy design load requirements.

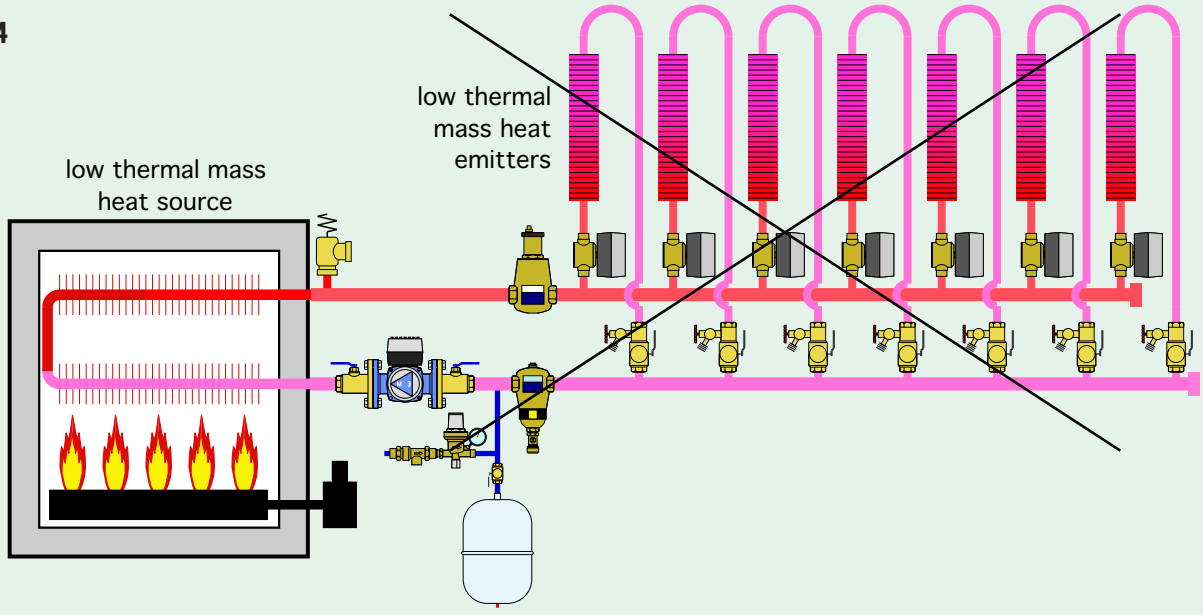
If the heat pump has a supplemental electric heating element, or if the heat pump is paired with a boiler for supplemental heating, and if the supplemental heat source(s) is properly sized, the combined output of the heat pump and supplemental heat source can meet design heating load. However, it is important to control the system so that the heat pump is not forced to inadvertently operate at inlet temperatures above the manufacturer's specified limit. This functionality may be provided by the heat pump's internal controls. If not, the designer needs to provide a controller that monitors the water temperature entering the heat pump and disables heat pump operation if that temperature exceeds the manufacturer's specific limit.

OPERATING HEAT PUMPS OR CERTAIN BOILERS AT LOW FLOW RATE

Most cast iron boilers can be operated at very low — even zero — flow rate. The latter was common when the boiler fired to maintain the temperature of an internal "tankless" water-heating coil. Other types of hydronic heat sources, such as mod/con boilers, copper tube boilers, electric boilers and hydronic heat pumps, do not have the metal or water content of cast iron sectional boilers. Their low thermal mass is an asset in terms of rapid rise to nominal operating temperature following a cold-start condition. However, low thermal mass also implies that heat needs to be removed from the heat source at rates very close to the rate of heat production.

Most mod/con boilers and hydronic heat pumps have minimal flow rates specified by their manufacturer. The balance of the system that connects to these heat sources

Figure 4-4



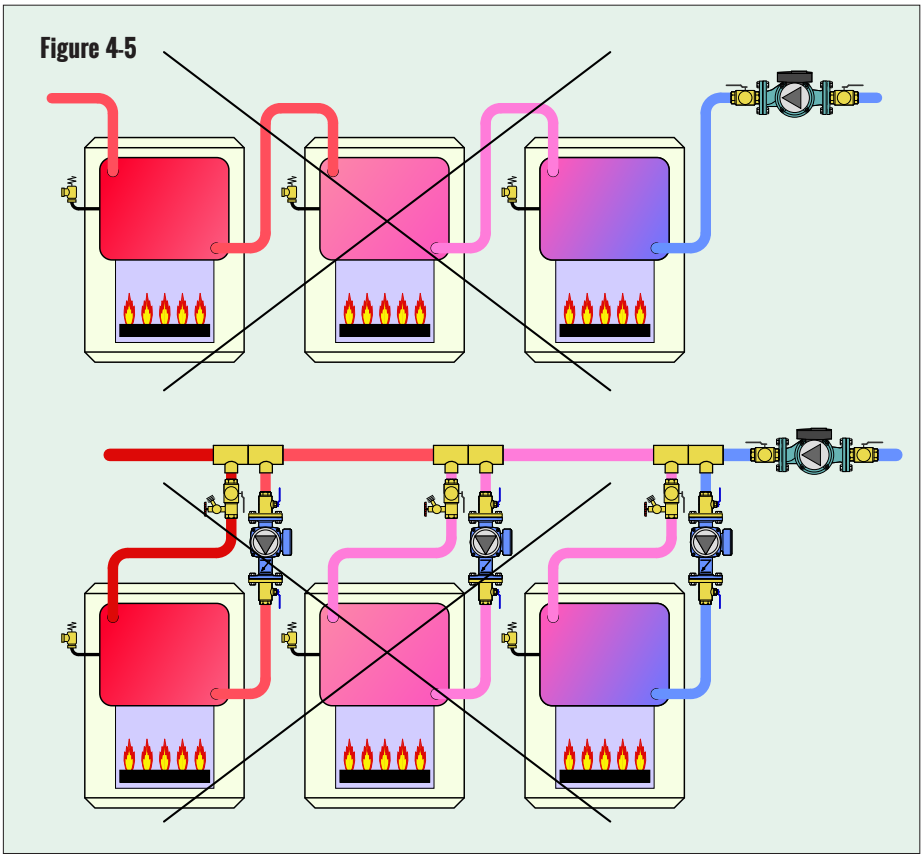
must ensure that these minimum flow rates are met. Although seemingly simple, this requirement can be compromised by inadvertently creating a distribution system with extensive zoning and low thermal mass. Figure 4-4 shows one example.

When only one or two zones are operating, the flow rate through the boiler may be very low. This can lead to high mechanical stress on the boiler's heat exchanger. In extreme situations, it could cause the water passing through the boiler to actually boil, which would lead to opening of the pressure relief valve, and a mechanical room full of steam. Low flow rates caused by extensive zoning can also cause on/off boilers to short cycle.

INCORRECT PIPING OF MULTIPLE HEAT SOURCES

Figure 4-5 shows multiple heat sources piped in ways that do not allow for optimum efficiency or serviceability.

Figure 4-5



Multiple heat sources need to be piped so that several conditions are met:

1. The piping should allow each heat source to receive the same inlet water temperature. This implies that multiple heat sources should be piped in *parallel*, not in series.

2. There should be *no flow through inactive heat sources*. Allowing such flow allows the inactive heat source to dissipate heat coming from the other operating heat sources.

3. The piping should allow any of the heat sources to be isolated from the remainder of the system if necessary for service or replacement, without the need to shut down the system.

4. The control system should detect a fault condition at any of the heat sources that renders it temporarily inactive and prevent the staging controller from sending an enabling signal to that heat source until the fault condition is corrected.

5. There should be minimal, if any, changes in flow rate through a given heat source in a multi-heat source system based on which of the other heat sources is operating. This implies proper hydraulic separation if multiple circulators are used.

6. Each heat source must be equipped with a pressure-relief valve that cannot be isolated from that heat source through any valve closure.

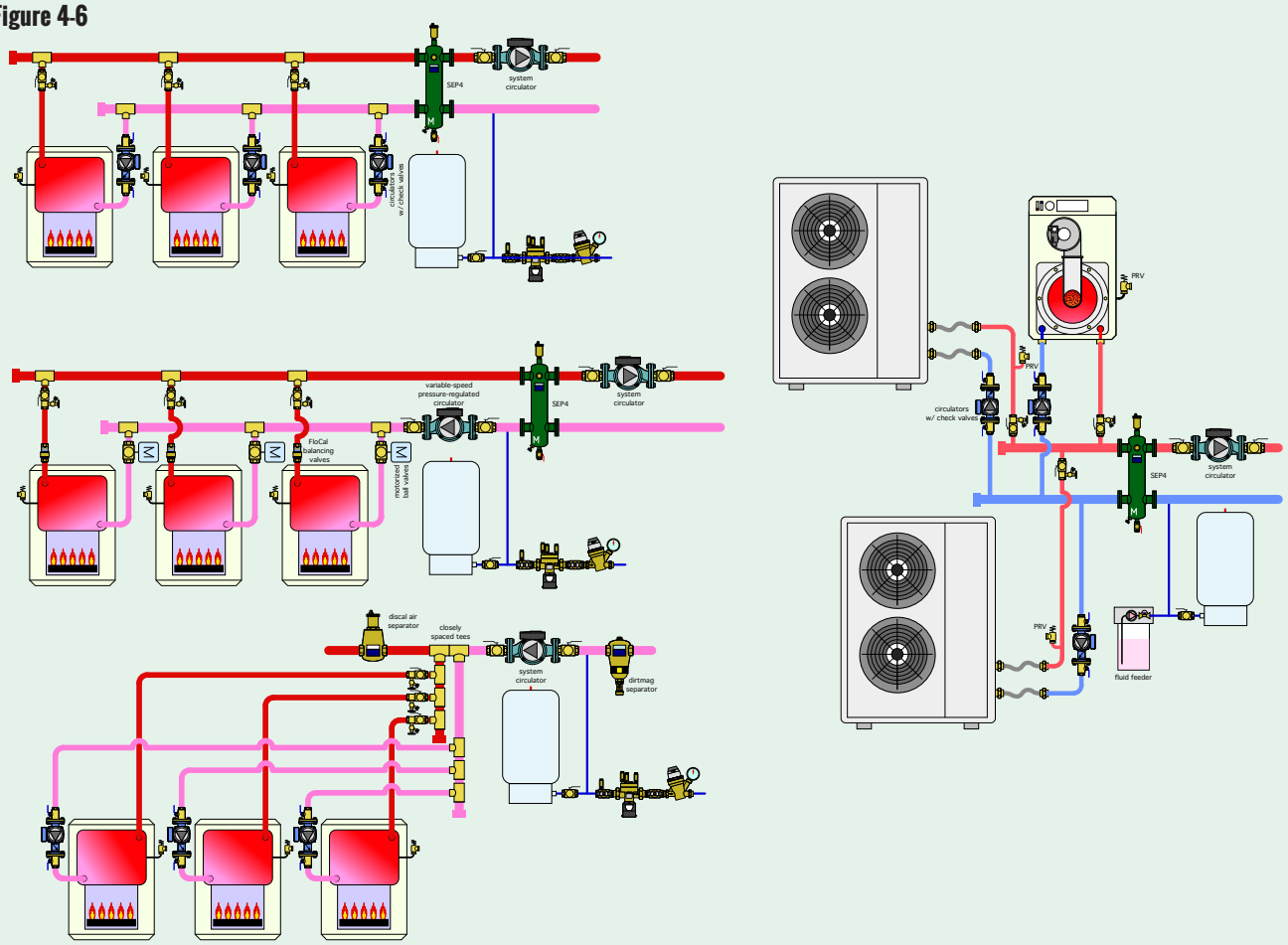
7. Each heat source should be equipped with a low-water cutoff and manual high-limit controller as required by relevant codes.

Figure 4-6 show several piping configurations for multiple boilers, as well as combinations of boilers and air-to-water heat pumps, that meet these design requirements.

In each piping configuration, the heat sources can be individually isolated from the remainder of the system. Each heat source receives water at the same temperature from the return header. Each heat source has a pressure-relief valve. When individual circulators are used for each heat source, those circulators contain integral spring-loaded check valves to prevent reverse flow. Another option is to use external spring-loaded check valves at each heat source.

Three of the four piping assemblies use a Caleffi SEP4 hydraulic separator to isolate the pressure dynamics of the heat source circulators from the main distribution circulator. The SEP4 also provides air, dirt and magnetic particle separation for the system.

Figure 4-6



AVOIDING SHORT CYCLING

The ability to easily create multiple zones has always been an incentive to use hydronic heating or cooling. Several approaches are possible depending on the hardware the designer chooses to use.

***idronics* #36 covers methods and materials for hydronic zoning in detail.**

One of the consequences of a highly zoned system is that the load imposed by a zone with a low heat delivery requirement, especially when it's the only zone operating, can be much lower than the heat output rate of the heat source. This is especially true for on/off heat sources.

In a system with a modulating heat source, the minimum output rate of that heat source should be compared against the heating load of the smallest zone. When there is a reasonable match between the minimum heat output and the load of the smallest zone, it's likely that the thermal mass of the fluid and the metal in the system provides adequate buffering between the heat source and load, eliminating the need for a buffer tank. This is especially true for systems with high thermal mass heat emitters, such as heated concrete slabs.

The greater the difference between the minimum heat output rate of the heat source and the heat output rate of the system's smallest zone, and the lower the thermal mass of the distribution system/heat emitters, the greater the need for a buffer tank.

Figure 4-7 shows a simplified piping configuration for a heat source, buffer tank and zoned distribution system.

This system uses a "3-pipe" buffer tank configuration. When the heat source and one or more zones are operating

Figure 4-7

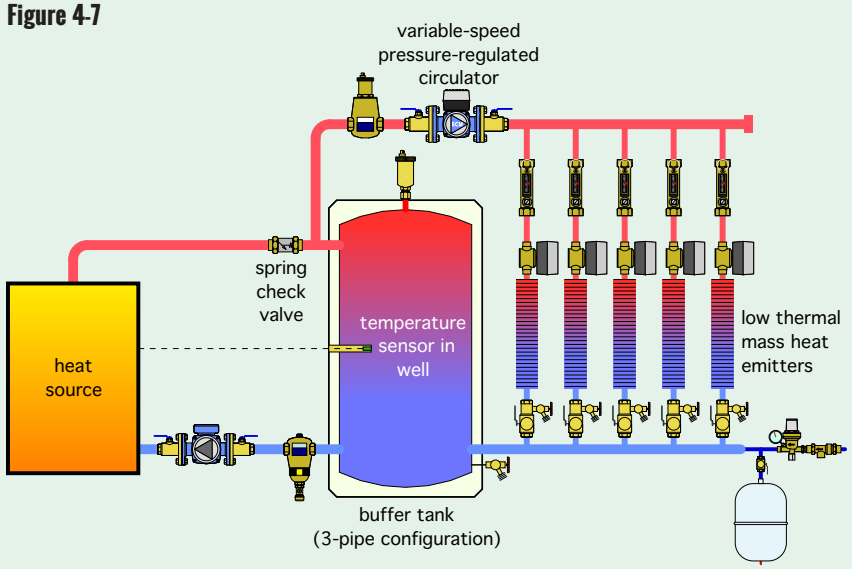
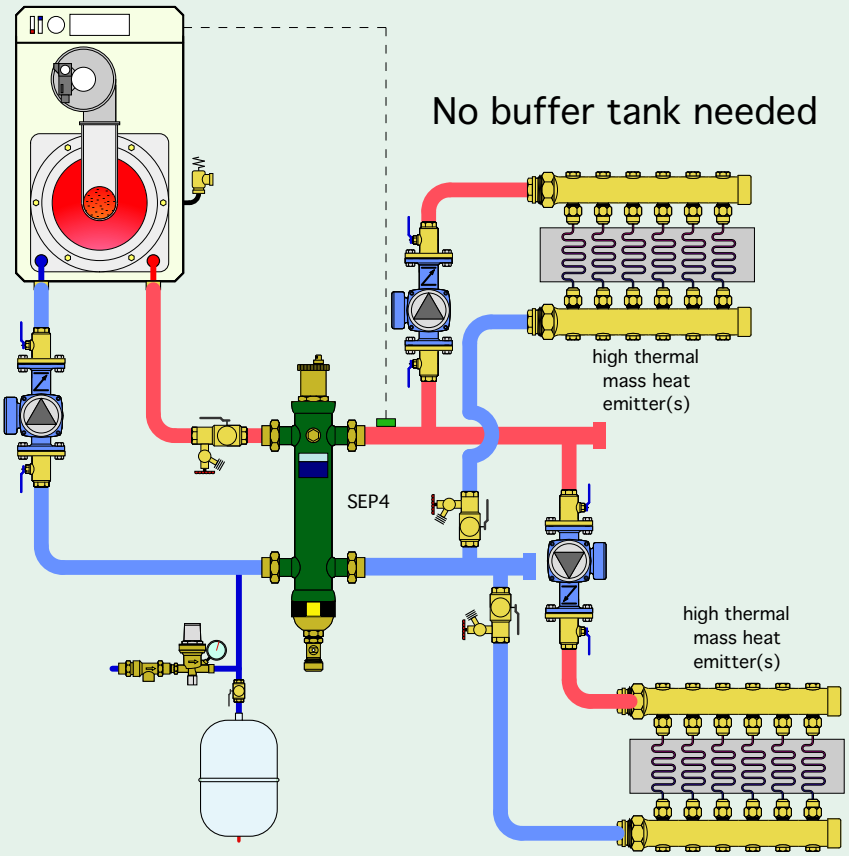


Figure 4-8



at the same time, this arrangement allows some flow to pass directly from the heat source to the heat emitters *without first passing through the upper portion of the buffer tank.* The heat source simply maintains some target water temperature within the buffer tank, which could be based on outdoor reset or a temperature setpoint.

Other buffer tank piping configurations are possible.

See *idronics* #17 for more information on buffer tank piping options

The required size of the buffer tank can be estimated using formula 4-1.

Formula 4-1:

$$V_{buffer} = \frac{t\left(q_{HSmin} - q_{Lmin}\right)}{500(\Delta T)}$$

Where:

V_{buffer} = required volume of the buffer tank (gallons)

t = minimum run time of the heat source (minutes)

q_{HSmin} = minimum heat source output (Btu/hr)

q_{Lmin} = load of smallest zone operating at the same time as the heat source (Btu/hr)

ΔT = temperature rise in the buffer tank during minimum heat source run time (ºF)

500 = constant required by the units

Example: A modulating boiler has a minimum output of 10,000 Btu/hr. The load of the smallest zone that could be operating at the same time as the heat pump is 2,000 Btu/hr. The desired minimum run time for the boiler at its minimum output rate is 10 minutes. During this 10-minute runtime, the buffer tank temperature is allowed to increase by 20ºF. What is the required buffer tank volume?

$$V_{buffer} = \frac{t\left(q_{HSmin} - q_{Lmin}\right)}{500(\Delta T)} = \frac{10\left(10{,}000 - 2{,}000\right)}{500(20)} = 8 gallons$$

This result might be surprisingly small by comparison to older systems using fixed output boilers. For example, if an on/off boiler with an output of 80,000 Btu/hr were substituted for the small mod/con boiler, the required buffer tank volume would be:

$$V_{buffer} = \frac{t\left(q_{HSmin} - q_{Lmin}\right)}{500(\Delta T)} = \frac{10\left(80{,}000 - 2{,}000\right)}{500(20)} = 78 gallons$$

These calculations are "conservative" because they do not include the buffering effect of the thermal mass of the remaining water or the metal in the system, only the water content in the buffer tank.

As building loads continue to decrease, so does the required capacity of its hydronic heat source. As the modulation ability of hydronic heat sources increases (e.g., turndown ratios of 5:1 progress toward ratios of 10:1), the required size of a buffer tank will decrease. *In some cases, no buffer tank will be required.* An example of the latter would be a system having high thermal mass heat emitters, and perhaps 1 to 3 zones. In these situations, the heat source can be connected to the high thermal mass distribution system using a hydraulic separator, as shown in figure 4-8.

This simple configuration provides ample thermal mass to prevent short cycling even if a single zone is operating.

Design issues that can *cause* short cycling of the system's heat source include:

• On/off heat source coupled to many independently controlled zones using low thermal mass heat emitters and no buffer tank.

• A grossly oversized heat source relative to design load.

• Undersized heat emitters relative to heating capacity of the heat source.

• Scale formation within the boiler's heat exchanger.

• Scale formation on the domestic water side of the coil heat exchanger within an indirect water heater.

• Inadequate flow relative to the heating capacity of the heat source. Typical flow rates allow for 8-20ºF temperature rise across the heat source at full capacity.

• Dirt, scale, slug or other fouling factors on internal or external surfaces of the heat emitters.

• One or more zones having very low load relative to the *minimum* heating capacity of the modulating heat source.

• Narrow differential temperature setting between the ON vs. OFF status of the heat source.

• Clogged Y-strainer in line with the heat source.

5. DESIGN ERRORS RELATED TO HEAT EMITTERS

Heat emitters are the "caboose" of any hydronic heating system. They create *intentional* heat transfer from the circulating water to the space being heated. When properly selected, sized and supplied, they deliver unsurpassed comfort to their assigned spaces with minimal, if any, detectable sound, odor or other compromises.

Designers need to understand that every heat emitter is fundamentally a heat exchanger. The same principles that govern the thermal and hydraulic performance of any heat exchanger apply to all heat emitters. These principles include:

1. The output of a heat emitter is approximately proportional to the temperature difference between the average water temperature and the air temperature of the space being conditioned. The higher the average water temperature within the emitter, the greater its rate of heat output.

2. The larger the surface area of the heat emitter, the lower the required average water temperature for a given rate of heat output.

3. Any scale, dirt, slug or other "fouling" materials on the water side or room side of the heat emitter reduce its ability to release heat at a given rate and when operating at a given average water temperature.

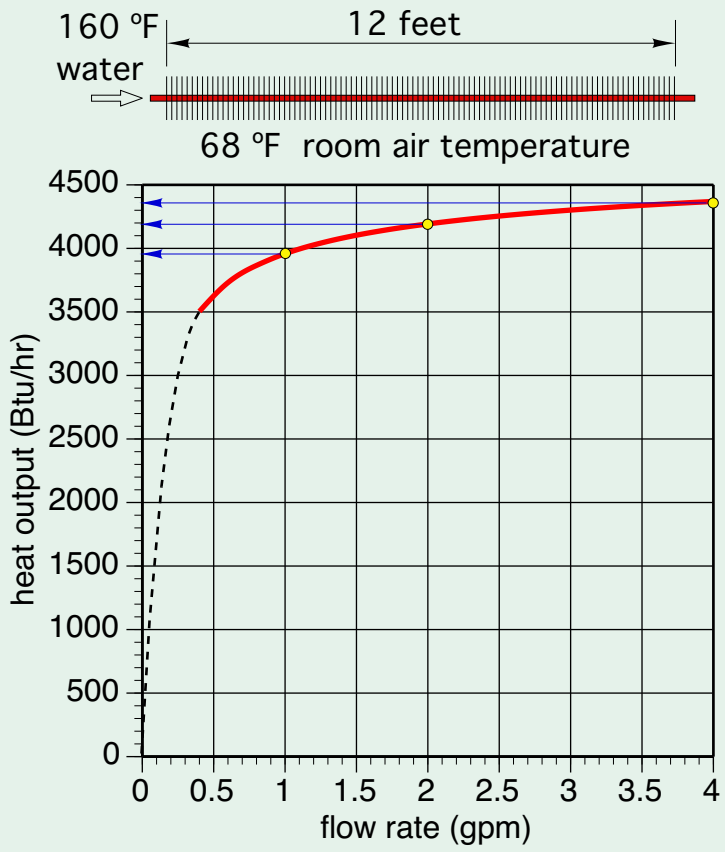
4. The head loss and associated pressure drop of any heat emitter needs to be factored into the hydraulic design of the system.

RELATIONSHIP BETWEEN FLOW RATE AND HEAT OUTPUT

One misunderstanding regarding heat emitters is that the rate of heat output is *proportional* to the flow rate through the emitter. *This is not true.* While it is true is that the rate of heat output increases with the flow rate, that relationship is not proportional.

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

Figure 5-1



Assume that water enters a baseboard that conforms to the relationship shown in figure 5-1. The water flow rate is 4 gpm and it enters the baseboard at 160ºF. Room air enters the bottom of the baseboard at 68ºF. Under these conditions, the baseboard's heat output is about 4,350 Btu/hr.

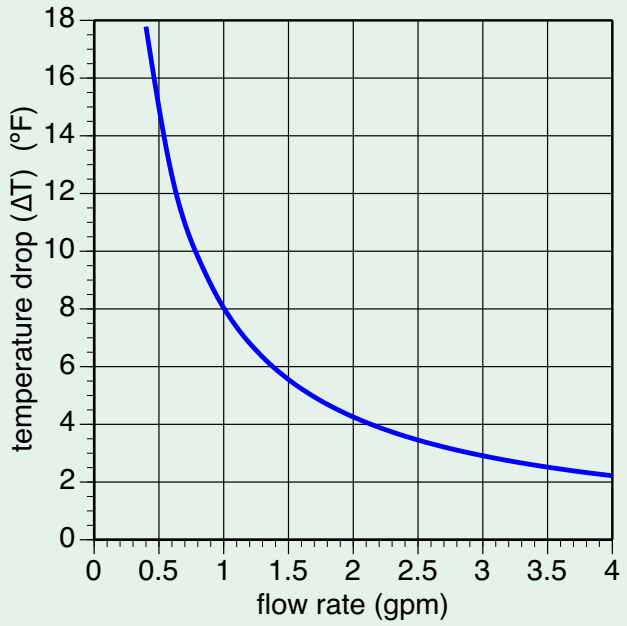
Consider what happens if the same water and air inlet temperatures are maintained, but the flow rate through the baseboard is reduced from 4 to 2 gpm. The baseboard's heat output drops to about 4,200 Btu/hr — a drop of only about 3.5%. If the flow is cut in half again, from 2 to 1 gpm, the heat output drops to about 3,900 Btu/hr, a drop of about 10.3% from the heat output at 4 gpm flow rate. This behavior demonstrates that heat output from the baseboard decreases slowly at higher flow rates but drops off quickly at low flow rates.

The red curve in Figure 5-1 doesn't go below flow rates of 0.4 gpm because at flow rates in the range of 0.3 gpm, the water passing through the ¾" copper fin-tube transitions from turbulent to laminar flow. This will cause a significant drop in heat output. Although the exact output at very low flow rates is hard to predict, it must drop to 0 output at 0 flow rate.

Figure 5-2 shows what happens to the

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

Figure 5-2



At 4 gpm, the temperature drop across this baseboard is only about 2.2ºF. At 2 gpm, it's about 4.2ºF, and at 1 gpm, the ΔT is about 8ºF. If the flow gets down to 0.4 gpm, the ΔT is just under 18ºF. The temperature drop along the baseboard changes as the flow rate through the baseboard changes.

The non-linear relationship between flow rate and heat output is not limited to fin-tube baseboard. It applies to all types of heat emitters.

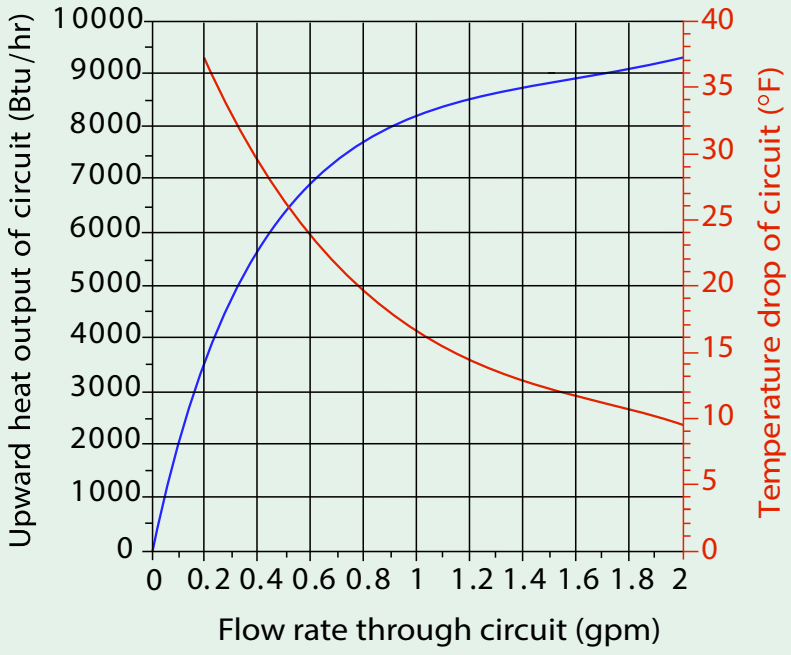
Figure 5-3 shows an example of how the upward heat output and temperature drop of a typical slab heating circuit consisting of 300 feet of 1/2" PEX tubing embedded in a bare 4-inch-thick concrete slab and supplied at a *constant water temperature of 110ºF* varies with changes in flow rate. The room temperature above the slab is 70ºF.

Notice how the rate of heat output increases rapidly at relatively low flow rates, but is much slower at higher flow rates. Increasing the flow rate from 1 to 2 gpm only increases heat output from about 8,200 Btu/hr to 9,300 Btu/hr, or about 13.4%.

The temperature drop along the circuit changes from about 17ºF at 1 gpm to 9.5ºF at 2 gpm.

If the circuit were to operate at 2 gpm rather than 1 gpm, the theoretical increase in pumping power would be a factor of 8. The higher installation and operating cost of a circulator capable of the higher flow rate is likely not justified based on the small increase in heat output.

Figure 5-3



One myth that occasionally surfaces is that increasing the flow rate through a heat emitter somehow *reduces* its heat output. The supposed rationale is that high flow rates don't allow the heat "sufficient time" to transfer from the water to the wetted internal surfaces of the heat emitter. *This is not true*.

One simple way to demonstrate it is to look at the heat output charts for fin-tube baseboard that show thermal output for flow rates of 1 gpm and 4 gpm. For a given model, and given water temperature, the heat output is slightly (but consistently) higher at 4 gpm compared to 1 gpm flow rate.

Another way to think about this situation is to consider what happens to the *average water temperature* in a given length of baseboard as the flow rate increases, but

the entering water temperature remains constant. This is illustrated in figure 5-4.

As the flow rate increases, the temperature drop along the baseboard decreases, *but the average water temperature within the baseboard increases.* Higher average water temperatures imply higher overall heat output.

From the standpoint of heat output only, there will always be an increase in output at higher flow rates. However, designers should also consider operating conditions such as head loss, required pumping power, flow noise and potential for erosion corrosion of copper tube and fittings at high flow rates. These factors often limit flow rates to ranges specified by the heat emitter manufacturer. The flow velocity in copper tubing supplying any heat emitter should also be kept no higher than 4 feet per second to minimize flow noise.

It's also important to keep the flow rate high enough to maintain turbulent flow within the heat emitter. For fin-tube baseboard, the Reynolds number of the flow should be above 4000. For panel radiators, the heat output at low flow rates should be estimated using the European standard EN442.

RELIANCE ON ASSUMED CIRCUIT TEMPERATURE DROP

One conjecture that has long been present in the North American hydronics market is that all hydronic heating circuits should operate at a temperature drop of 20ºF. The origin of this number is unknown. Perhaps it came from the concept that ***if*** a system using water operates with a 20ºF temperature drop, ***then*** the rate of heat delivery would be approximately 10,000 Btu/hr per gpm of flow rate. While this is a close approximation to the rate of heat delivery, it all hinges on the words "if" and "then." More specifically, ***if*** the system operates with a 20ºF temperature drop, ***then*** each gpm of water flow rate would transport about 10,000 Btu/hr.

There is nothing special or advantageous about operating a hydronic heating circuit with a 20ºF temperature drop. In specific applications, circuits can perform well using much lower and much higher ΔTs.

For example, if a floor-heating circuit operated at a very high flow rate, its ΔT might only be 2 to 4ºF. That would make the average water temperature in the circuit only 1 to 2ºF lower than the supply water temperature. It would also make the floor area served by the circuit relatively uniform in surface temperature. So, what's wrong with operating a circuit under such a low ΔT? From the standpoint of heat transfer, and achieving relatively uniform floor surface temperatures, there's nothing wrong. However, maintaining the high flow rate necessary for this very low ΔT would require a larger circulator that would likely use significantly more electrical energy compared to that required to operate the circuit at a more typical ΔT of 10 to 15ºF at design conditions.

It's also possible to operate hydronic circuits at temperature drops much higher than 20ºF. In Europe, some panel radiator systems are designed for a 20ºC (36ºF) temperature drop at design load conditions. Using

Figure 5-4

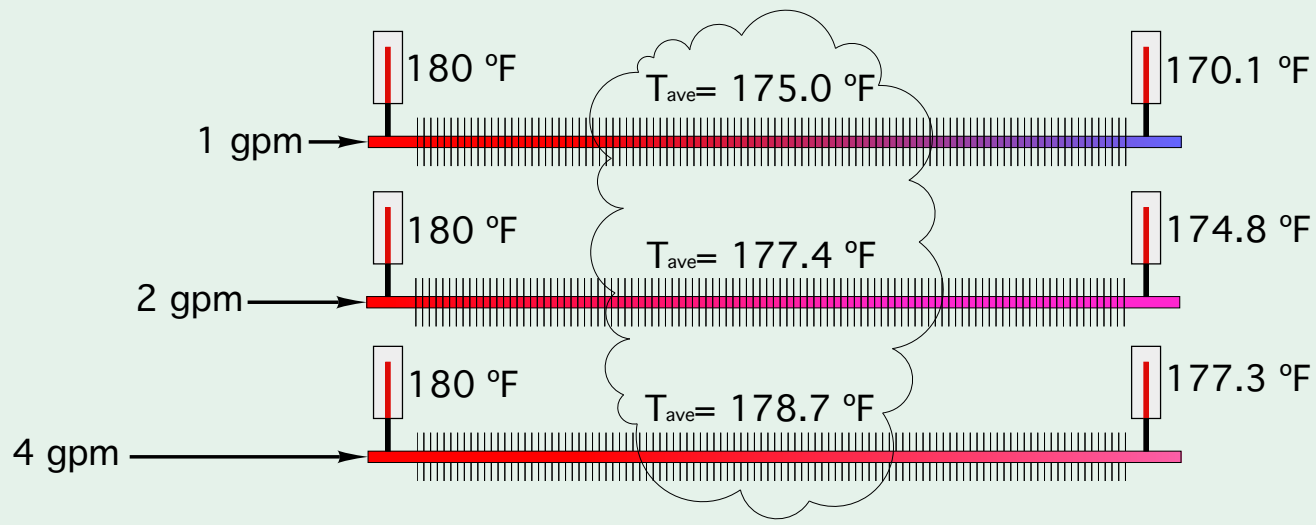
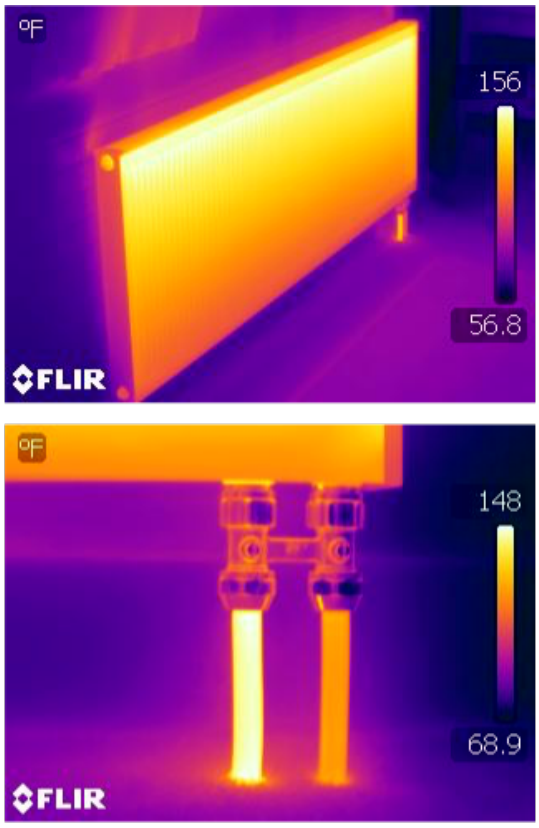


Figure 5-5



the higher ΔT allows the flow rates to each panel radiator to be reduced. The lower flow rate may also allow smaller tubing to be used to supply the radiator, as well as the possibility of a smaller circulator with lower power input requirements.

Figure 5-5 shows a thermograph of a panel radiator operating with an estimated 40ºF ΔT between its supply and return water temperature. The ΔT was estimated based on the temperature scale and colors on the supply and return piping.

Notice the relatively even temperature gradient from the top to the bottom of the panel radiator. This indicates even flow distribution among the panels' vertical flow channels.

USE OF FIXED ΔT CONTROL

There are design methods that use two temperature sensors and associated control logic to maintain a fixed, user-set temperature drop around a heating circuit. This approach is applicable when ***all*** the following conditions are met:

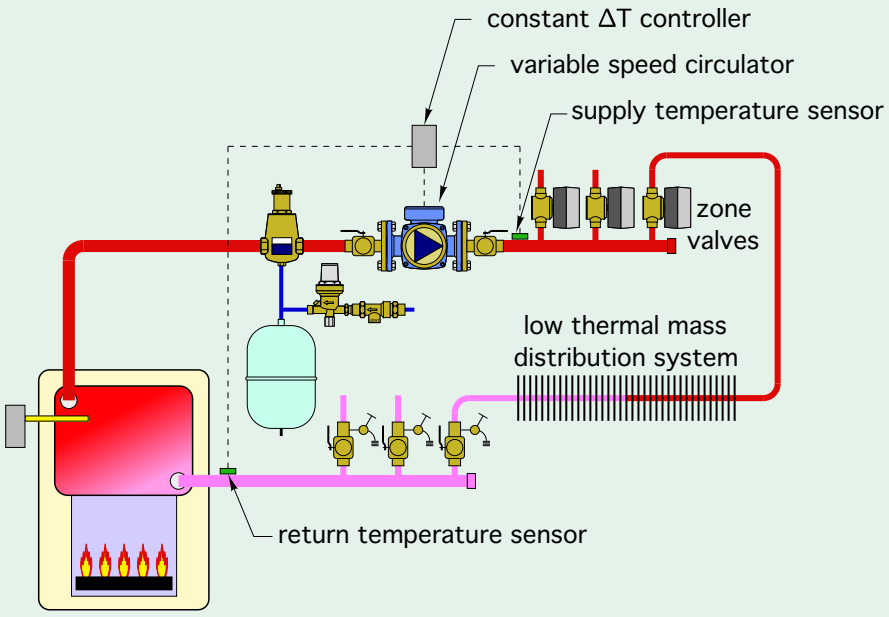
1. Multiple heating zones are controlled using:

    a. on/off zone valves
    or
    b. zone circulators on a header supplied from a hydraulic separator where flow from the heat source into the separator is varied to maintain a constant ΔT.

2. The distribution system and heat emitters have low thermal mass.

3. The heat source provides a constant supply water temperature, at the design value, whenever any zone is calling for heat.

Figure 5-6

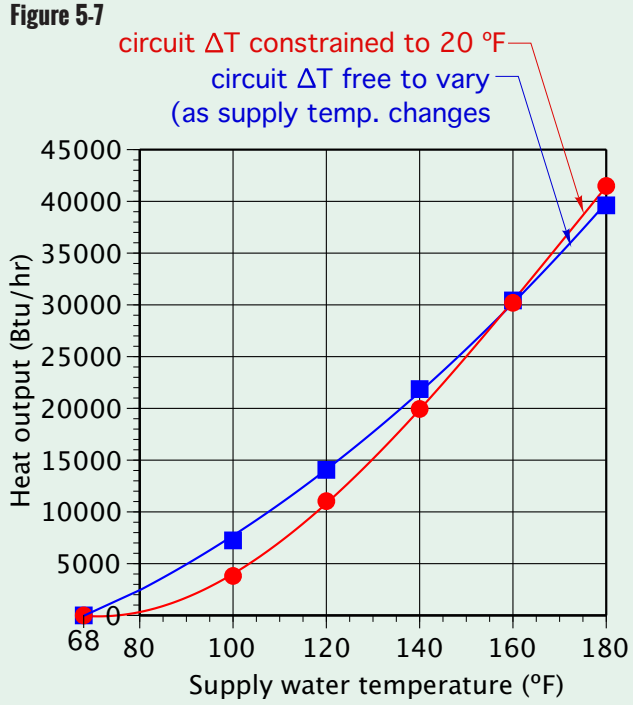


Consider a low thermal mass hydronic heating system that supplies design load heat output when all zones are active and the supply water temperature remains constant at the design load value. An example of such a system is shown in figure 5-6.

If the heat source is not oversized for the design load, all zones would, in theory, remain on until the design load condition subsided (or other factors such as internal gains or intentional thermostat setbacks began influencing the zone loads).

When the thermostat in a zone is satisfied and that zone turns off, less

Figure 5-7



heat would be removed from the distribution system. This change would reveal itself as an *increase in return water temperature* (assuming that the supply water temperature remains constant). The ΔT of the circuit would decrease. A controller that senses this decrease could respond by reducing the speed of a circulator so that the design ΔT was reestablished for the zones that remain active. This process would repeat when another zone turned off. This method of control reduces circulator energy use during partial load conditions.

When a zone turns on, *and the supply water temperature remains fixed at the design value*, the return water temperature decreases because more heat is being removed from the distribution system. A controller could sense this increase in ΔT and respond by increasing circulator speed to reestablish the design ΔT for all the active zones.

The requirement that the distribution system have low thermal mass implies that the temperature changes on the return side of the system would occur quickly as zones turn on and off. A high thermal mass distribution system, such as a heated concrete floor slab, could significantly delay these temperature changes due to heat being absorbed into or released from the thermal mass.

A control method that maintains a fixed supply temperature, based on design load conditions, forces the active portion of the system to operate as if design load conditions were always present. In effect, it directs "pulses" of heat into each zone whenever the associated zone thermostat calls for heat. Each pulse of heat delivery occurs at the design load rate. The duration of each pulse is the time that the zone valve or zone circulator is on. The heat transfer rate multiplied by the elapsed time of the heat input pulse determines the total heat added to the space during that time. This "pulsed" method of heat delivery has been used in millions of North American hydronic systems. It is generally acceptable if the thermostat differential is reasonable.

Not all hydronic systems meet the three previously stated constraints. Many modern systems use outdoor reset control to vary the water temperature supplied to the distribution system. When this method of water temperature control is combined with a controller that varies flow rate to maintain a *fixed* ΔT, the heat output from the distribution system decreases differently compared to a system without any imposed control over ΔT.

Figure 5-7 shows the heat output for a 100-foot-long segment of residential fin-tube baseboard as a function of supply water temperature. The assumed room air temperature is 68°F.

The **blue** curve shows the heat output of the baseboard as the water temperature is decreased from 180 to 100°F. The flow rate in the circuit varies from about 3.2 gpm at 180°F supply temperature down to 2.93 gpm at 100°F. The slight decrease in flow rate is the result of increasing viscosity at lower water temperatures. At a supply temperature of 68°F, the heat output of the circuit is 0. The ΔT of the circuit adjusts on its own.

The **red** curve is the heat output of the same 100-foot length of fin-tube baseboard across the same range of supply water temperature (180°F down to 100°F), but with the flow rate adjusted to yield a fixed temperature drop (ΔT) of 20°F at all supply water temperatures.

Although the two heat output curves are similar, the curve for the circuit operating with a fixed ΔT of 20°F releases less heat at lower supply water temperatures. Hence the reason for suggesting that this method be used with a constant supply water temperature at the design load value.

INADEQUATE HEAT OUTPUT FROM RADIANT FLOOR PANELS

There are many ways to incorporate small-diameter flexible tubing such as PEX, PERT or PEX-AL-PEX into various floor constructions. One example is tubing fully

Figure 5-8

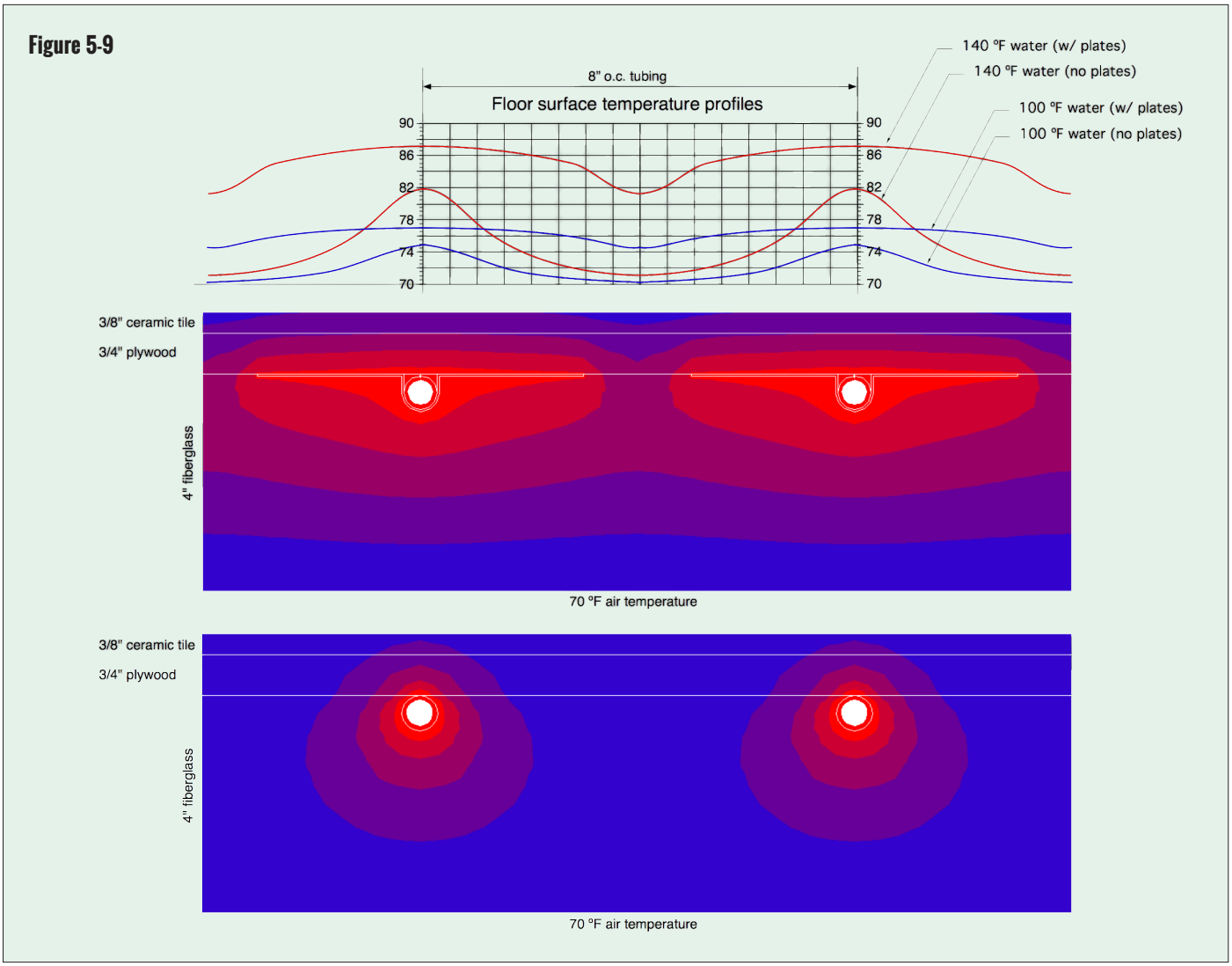


embedded in cementitious materials such as concrete or poured gypsum underlayment. Another example is tubing in contact with preformed aluminum plates that are stapled to the underside of a wood subfloor.

From the standpoint of thermal performance, *it is critically important that the materials adjacent to the tubing are capable of conducting heat away from the tubing and dispersing it laterally across the materials between adjacent tubes.* Think of these materials collectively as "thermal wicks" that pull heat away from the tube and spread it across the floor construction.

If the materials used do not have reasonably good thermal conductivity, or if the materials don't make good contact with the tubing or other materials within the floor, the thermal performance of the panel as a whole will be disappointing.

Figure 5-9



One example that demonstrates the results of poor heat transfer is the "plateless" staple-up technique shown in figure 5-8.

The tubing in this type of installation has very limited contact with the floor, especially when heated water causes it to expand and droop between adjacent staples. Because of this poor contact, there is very limited heat conduction from the tubing to the surrounding materials. Convection and thermal radiation from the tube will be present, but very limited due to the small temperature differences involved. *In short, this installation method should NEVER be used.*

Figure 5-9 shows a thermal model for a floor heating panel using PEX tubing spaced 8 inches on center.

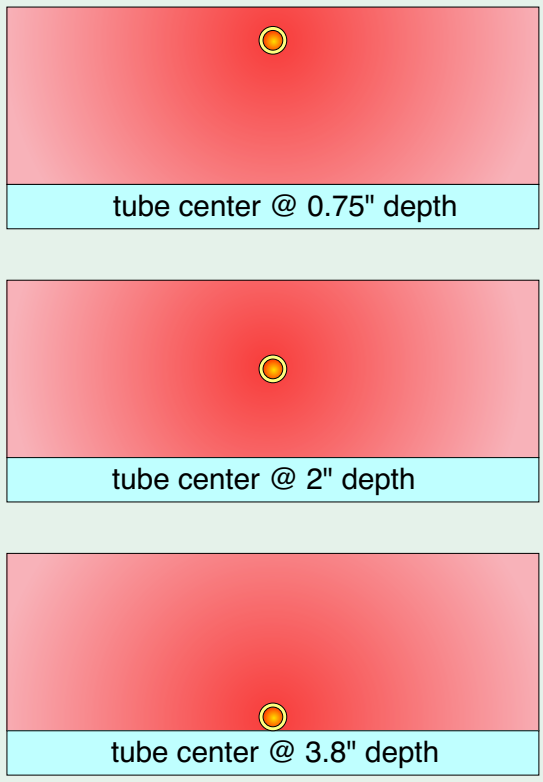
The upper cross section image includes 6-inch-wide aluminum plates in tight contact with the PEX tubing. It also includes 4 inches of fiberglass insulation under the tube and plates. The lower image has the same construction as the upper image, but without the aluminum plates. The tubes are assumed to be in contact with the subfloor. The surface temperature profiles for both constructions are shown above the cross sectional images. *The system using plates provides about three times the upward heat flux compared to the "plateless" system when operated at the same water temperature*. It's also likely that this is a "generous" comparison because it still assumes the tubing without the plates is making contact with the subfloor. A careful look at figure 5-8 show that this is not the case for much of the tubing. This lack of contact further reduces upward heat output from the plateless assembly.

NOT CONSIDERING TUBING DEPTH IN A HEATED SLAB

Another design issue that effects the thermal performance of slab-on-grade floor with embedded tubing is the depth of the tubing within the slab. Some practitioners may assume that this depth makes no difference in thermal performance. Others might claim that there's no way to control the depth of the tubing, and thus it should be left at the bottom of the slab.

Figure 5-10 shows a thermal model of a 4-inch-thick bare concrete slab with 1/2" PEX tubing spaced at 12-inch centers. The tubing is carrying water at 100ºF. The room temperature above the slab is 70ºF.

Figure 5-10



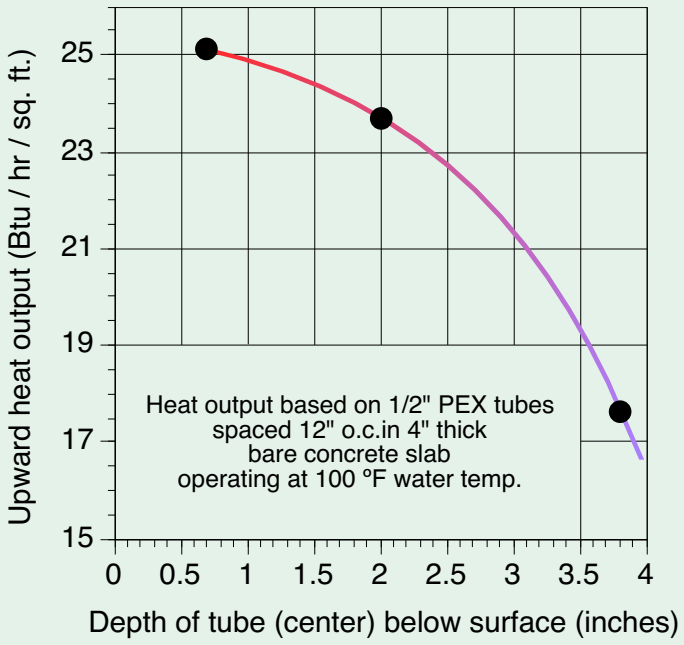


Figure 5-11

Upward heat output	Average circuit water temperature with tubing at ½ slab depth	Average circuit water temperature with tubing at bottom of slab
15 Btu/hr/ft²	95°F	102°F
30 Btu/hr/ft²	120°F	134°F

The only difference between the three cross sections is the depth of the tubing within the slab. The graph shows the upward heat flux from the slab surface based on the depth of the tubing. It's very apparent that upward heat output decreases as the tubing depth below the top surface increases.

It would be possible to compensate for the lower upward heat flux of the deeper tubing by increasing the average water temperature in the tubing. The required change in average water temperature for two different rates of upward heat output is shown in figure 5-11.

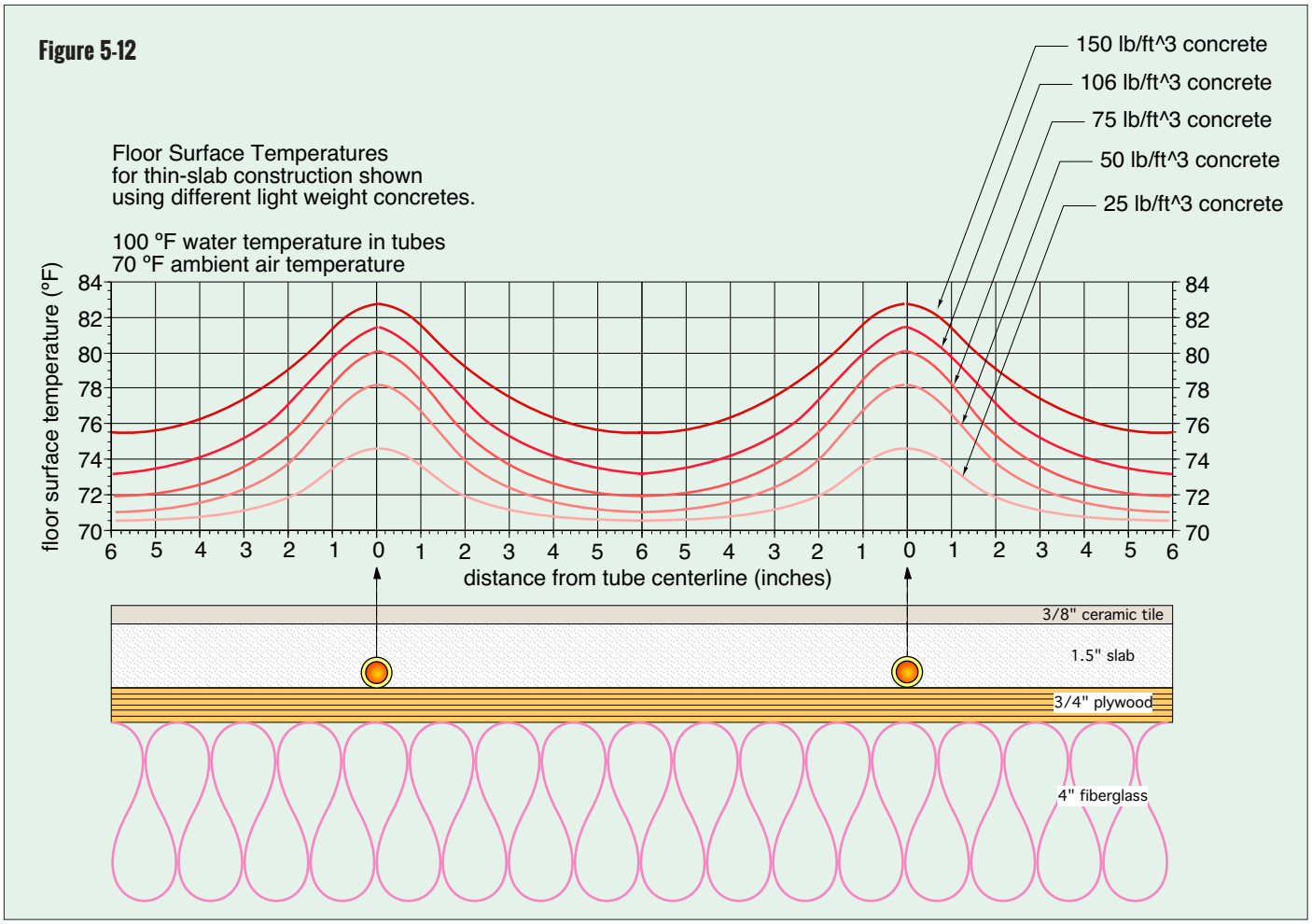
Higher water temperatures decrease the efficiency of modern heat sources such as mod/con boilers and hydronic heat pumps. Once the concrete is placed, the relationship between required water temperature and tubing depth is *set for the life of the building*. It only makes sense to minimize the need for higher water temperatures by limiting the depth of the tubing.

A practical guideline based on thermal performance, as well as structural issues such as the need for control joints in the slab, is to keep the tubing approximately one half of the slab depth for slabs 4 to 6 inches thick. Thicker slabs could also limit tubing depth to 2 to 3 inches below the upper surface.

THERMAL PERFORMANCE LOSS FROM LOW DENSITY CONCRETE

One method of creating a heated floor is to fasten tubing directly to the top of the subfloor and pour a "thin

Figure 5-12



slab" over the tubing. The thin slab might range from 1.5 to 2 inches thick and could be made of poured gypsum underlayment or a specific concrete mix design.

One concrete option that might seem plausible is called lightweight concrete. The mix design typically includes Portland cement and sand, but the coarse aggregate may be vermiculite or polystyrene beads rather than traditional crushed stone. These aggregates decrease the weight of the concrete, in some cases by a factor of 6. They also significantly decrease the thermal conductivity of the slab which encases the tubing. *This can drastically reduce the slab's ability to "wick" heat away from the tubing and disperse it across the slab.*

Figure 5-12 shows a comparison of the surface temperature profiles for a thin slab system with 1/2" PEX tubing spaced 12 inches on center, where the only variation is the density of the concrete used for the slab.

The mathematical area under the surface temperature profile is proportional to the upward heat output. A slab using ultra-lightweight concrete with a density of 25 lb/ft^3 will only release about 22% of the output attainable if the slab used standard concrete with a density of 150 lb/ft^3. This drastic drop in output implies that *lightweight concrete is not a good material for thin slab heated floors.*

INSUFFICIENT BACKSIDE, UNDERFLOOR OR UNDERSLAB INSULATION

All radiant panels should be constructed to "steer" heat output in the intended direction (e.g., toward the occupied space). This is done by including insulation materials on the back side or underside of the radiant panel. The lack of such insulation or using materials that don't provide adequate R-value can result in high backside or underslab heat loss that increases operating cost. It may prevent the radiant panel from maintaining the desired temperature in the occupied space.

Figure 5-13

Panel type	MINIMUM R-value of underside insulation
heated floor slab (w/covering R-value ≤1.0°F•ft²•hr/Btu)	R-10
heated thin slab (heated space below) (w/covering R-value ≤1.0°F•ft²•hr/Btu)	R-11
heated thin slab (unheated space below) (w/covering R-value ≤1.0°F•ft²•hr/Btu)	code R-Value x 1.5
above floor tube & plate (w/covering R-value ≤1.0°F•ft²•hr/Btu) (heated space below)	R-11
above floor tube & plate (w/covering R-value ≤1.0°F•ft²•hr/Btu) (unheated space below)	code R-Value x 1.5
below floor tube & plate (w/covering R-value ≤1.0°F•ft²•hr/Btu) (heated space below)	R-19
below floor tube & plate (w/covering R-value ≤1.0°F•ft²•hr/Btu) (unheated space below)	code R-Value x 1.5
radiant wall on exterior wall	code R-Value x 1.5
radiant wall (heated space behind)	R-11
radiant ceiling (unheated space above)	code R-Value x 1.5
radiant ceiling (heated space above)	R-13

A reasonable guideline is that the back side or underfloor insulation should have a *minimum* R-value of 10 times the thermal resistance of the materials on the room side of the panel. Thus, a heated slab covered with one or more layers of underlayment and finish flooring having a total R-value of 1.0ºF•ft^2•hr/Btu should have underslab insulation with a minimum R-value of 10ºF•ft^2•hr/Btu. Even higher underfloor R-values may be justified depending on the energy efficiency goals of the project. Higher R-values may also be required by building codes.

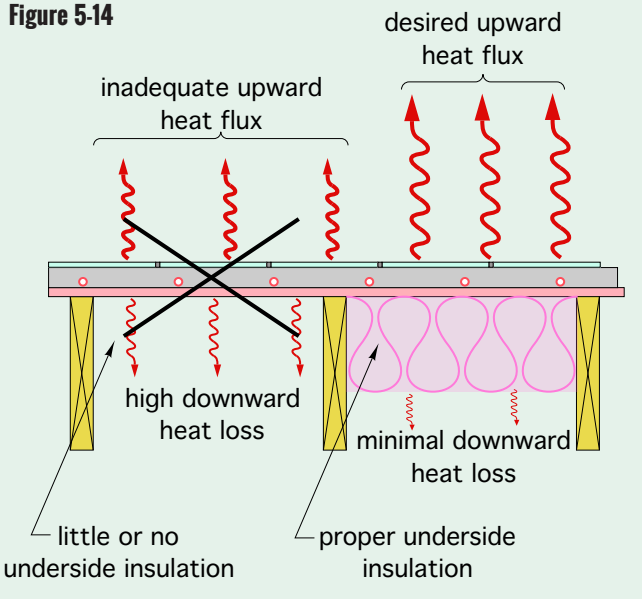
The following *minimum* R-values are suggested for backside or underfloor insulation in various radiant panel constructions. Designers should verify that these suggested minimum R-values comply with current building or energy codes.

An *undesirable* consequence of inadequate underslab insulation is higher losses to the soil under the slab. This is especially problematic in locations with rock ledge under the slab or where a high water table is present. Both situations can absorb large amounts of heat with very little increase in temperature and thus are excellent heat sinks. They are exactly what *isn't wanted* under a heated slab.

Another situation is missing or very inadequate insulation under a wood-framed floor with either underfloor tube & plate heating or thin slab heating. The latter is illustrated in figure 5-14.

Some designers might think that omitting underfloor insulation in a situation such as shown in figure 5-14 allows

Figure 5-14



heating of the space above the floor as well as the space below the floor. While this is fundamentally true, there are sound reasons NOT to do this:

1. There is no independent control of upward versus downward heat flow. For example, the space below the heated floor may require heating at the same time that internal gains are adequately heating the space above the floor. Operating the floor heating system under this condition would needlessly overheat the space above the floor. Not operating floor heating would allow the space below the floor to drop below the desired temperature.

2. There is no established design method for predicting the *proportions* of upward versus downward heat flux for a given floor assembly. For example, if the floor is covered with a material having relatively high R-value, the upward flux might be very low compared to the downward heat flux. If space below the floor has many cool surfaces, the downward heat flux could be higher than the upward heat flux.

RADIANT PANEL OR SERIES BASEBOARD CIRCUITS THAT ARE TOO LONG

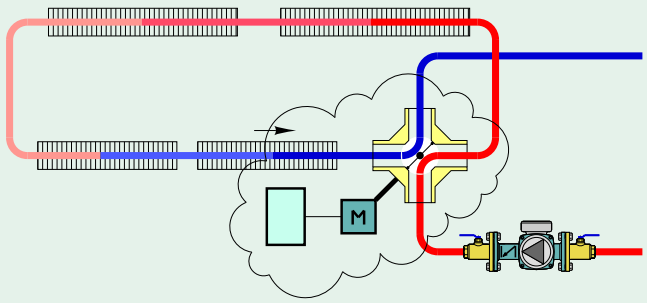
1/2" PEX tubing is commonly used for residential radiant panels, and often sold in 1,000-foot-long coils. There have been situations where installers have unwittingly installed the entire 1,000-foot coil as a single circuit. This creates very high head loss and low flow rates, as well as a very high temperature drop along the circuit. Correcting this situation after installation by finding intermediate locations along the circuit, cutting the circuit and reconnecting the segments into parallel circuits can be very expensive. If the tubing is embedded in a slab, the correction may be virtually impossible. 1/2" PEX circuits should be limited to 300-foot lengths. Even shorter lengths are preferred when practical because they operate with lower temperature drops and thus produce less variation in floor surface temperature from the beginning to the end of the circuit.

A similar situation can occur when many fin-tube baseboards are connected in series to form a single long circuit. The long circuit will operate at lower flow rates and higher temperature drops. A classic symptom of such an installation is underheating of rooms near the end of the long circuits. Correcting this situation by dividing up the long circuit into multiple parallel circuits may be possible but will also be expensive.

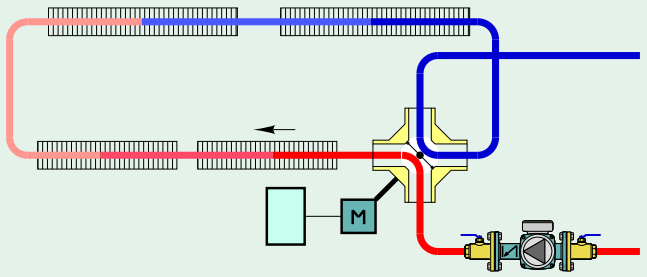
One possible corrective action for excessively long radiant panel or baseboard circuits is to periodically reverse the flow direction. One way to do this is shown in figure 5-15.

This approach uses a 4-way value, such as those used for mixing in other hydronic systems, but now functioning

Figure 5-15



4-way valve for periodic flow reversal
24 VAC spring return actuator controlled by
repeat-cycle time delay relay



as a "reversing" valve. The valve shaft is operated by a 2-position spring-return actuator. When 24 VAC is applied to the actuator, it rotates the valve's shaft 90 degrees, and flow follows the path shown in the upper image of figure 5-15. When power is removed from the actuator, its internal spring rotates the shaft in the opposite direction by 90 degrees, and flow follows the path shown in the lower image of figure 5-15.

The time during which the actuator is powered versus unpowered is controlled by a repeat cycle time delay relay. These relays are readily available from several sources. Some will have equal on-time versus off-time settings. Others allow the on time versus off time to be individually set. An example of the latter would be an on time of 20 minutes followed by an off time of 30 minutes. The longer time would allow water at the higher water temperature to be supplied to the portion of the circuit where the room(s) is consistently under-heated.

Designers should treat this detail as a corrective action to an existing system rather than a detail to be incorporated into a new system. It is preferable to prevent the need for this detail through conservative piping design that limits circuit length and/or limits the number of heat emitters connected in series.

COMBINING HIGH AND LOW THERMAL MASS EMITTERS IN THE SAME SYSTEM

Some hydronic systems combine high thermal mass slab heating on the ground level of a building with low thermal mass emitters in upper floors or other areas. Examples of the latter are air handlers, fan-coils and panel radiators.

Although this approach can work, it also creates the potential for unanticipated and undesirable performance glitches when both the high and low thermal mass emitters operate simultaneously.

One example of such a system would be floor heating in a basement or garage slab floor combined with console fan-coils in other areas of the building. Assume that the fan-coils were sized to provide design load output at a supply water temperature of 120ºF, and that the heated slab areas provide design load output when supplied with 100ºF water.

Assume that the system controls are configured to operate the heat source with a leaving water temperature of 120ºF when one or more of the fan-coils calls for heat, and the floor slab is off. These controls also provide 100ºF leaving water when the floor slab calls for heat and the fan-coils are off. If both the floor slab and fan-coil call for heat simultaneously (a likely scenario), the heat source is configured to supply 120ºF water, and a mixing device is used to lower the supply water temperature to 100ºF for the floor circuits.

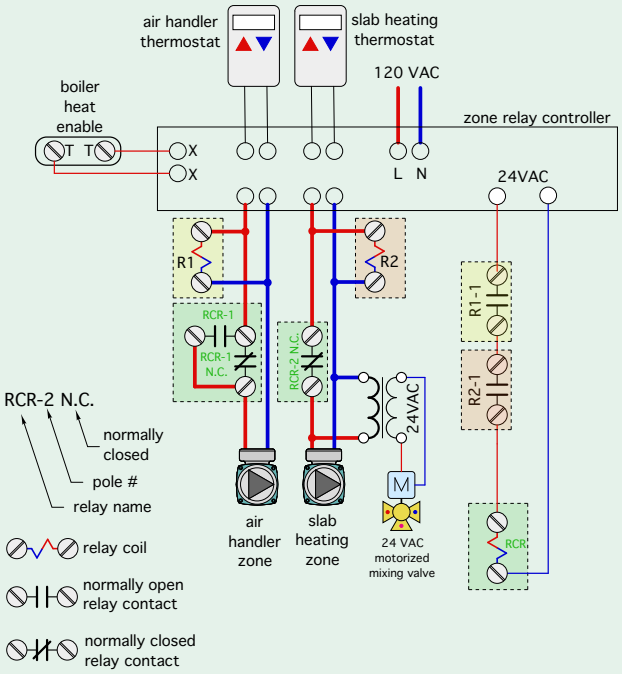
Assuming that the heat source capacity can match the total design heating load of the building, and *assuming that the floor slab is maintained at or close to steady state,* this approach can work. However, any situation that causes the slab to have to increase in temperature will impose transient conditions on the system. Examples include recovering the space heated by the slab from a setback condition, turning on the slab from a "cold start" at the beginning of the heating season, or restoring the temperature in the space heated by the slab following a power outage of several hours. Under these conditions, the temporary load imposed on the heat source by the slab can be two to three times the design load because the slab's thermal mass is capable of absorbing heat at a high rate.

This condition will quickly lower the leaving water temperature from the heat source to a temperature dictated by thermal equilibrium (e.g., where the output of the heat source matches the current total load of the heat emitters, including the rate of heat absorption by the slab). That water temperature is likely to be well below the water temperature at which the fan-coils were sized, and thus the

Figure 5-16



Macromatic TR 6512U repeat cycle time delay relay



When this condition is detected by the controls, a *repeat cycle time delay relay* can be used to operate valves or circulators that "toggle" the heat source output between the two loads. One load would operate for a set time (such as 30 minutes) and then turn off while the other load operates for a set time (such as 20 minutes). When only one load is operating, the time delay relay is inactive. A schematic for one approach is shown in figure 5-16.

The repeat cycle time delay relay is what "toggles" the boiler (or other heat source) between the two loads. *This relay is only active when both loads are calling for heat simultaneously*.

rate of heat output from the fan-coils will be significantly lower and very gradually increase as the slab warms and the transient conditions subside. During this time, the heat source should operate continuously since the distribution system is accepting all the heat the heat source can deliver.

This condition can be dealt with several ways. One is to simply tolerate the depressed output of the low thermal mass heat emitters when transient conditions occur. This should be discussed with owners to be sure that they understand and are willing to accept the transient system performance.

Another option would be to have separate heat sources for the slab and the low mass heat emitters. This would separate the transient conditions imposed by the heated slab from the fan-coils. A multiple boiler, multiple heat pump or dual fuel system that combines a heat pump with a boiler are all possible solutions.

Another less preferable but possible approach would be to significantly oversize the heat source so that it's capable of supplying the full transient load the slab can impose following a cold start. This could be costly both in terms of hardware cost and reduced operating efficiency. If a single large on/off boiler was used, it would spend most of its operating time short cycling.

It is also possible to design the system so that the heat source's output is "shared" whenever the high thermal mass and low thermal mass loads call for heat simultaneously.

The specific relay shown in figure 5-13 is a Macromatic TR 6512U. It has double pole double throw (DPDT) contacts with the poles designated at RCR-1 and RCR-2. Each pole has a normally open and normally closed contact. Two additional relays with 24 VAC coils (R1) and (R2) are used to verify that their respective zones are operating. When the coils in relays (R1) and (R2) are powered at the same time (e.g., when both the high mass and low mass zones are operating simultaneously), their normally open contacts (R1-1) and (R2-1), which are wired in series, are closed to pass 24 VAC to the repeat cycle time delay relay.

The motorized mixing valve shown in figure 5-16 only operates when the slab heating zone is active.

6. MISCELLANEOUS DESIGN ERRORS

Figure 6-1



This section describes several design errors not covered in previous sections.

MISAPPLIED VALVES

The common valves used in plumbing and hydronic systems fall into two categories based on their intended function:

1. Component isolation
2. Flow regulation

Although it is possible to install valves intended for component isolation in situations where flow regulations are required, and vice versa, *this is not advisable*. It can lead to serious problems, such as high head loss, low flow rates, erosion of metal surfaces within the valve, noise, vibration and eventual leakage.

Valves intended for component isolation should have very low head loss characteristics when fully open. It's highly likely that these valves will spend most of the system's life in a fully open position. They would only be closed if and when necessary to isolate a component from the remainder of the system for service or replacement.

The three valves specifically developed for component isolation are:

1. Gate valves
2. Ball valves
3. Butterfly valves

Examples of all three valves are shown in figure 6-1.

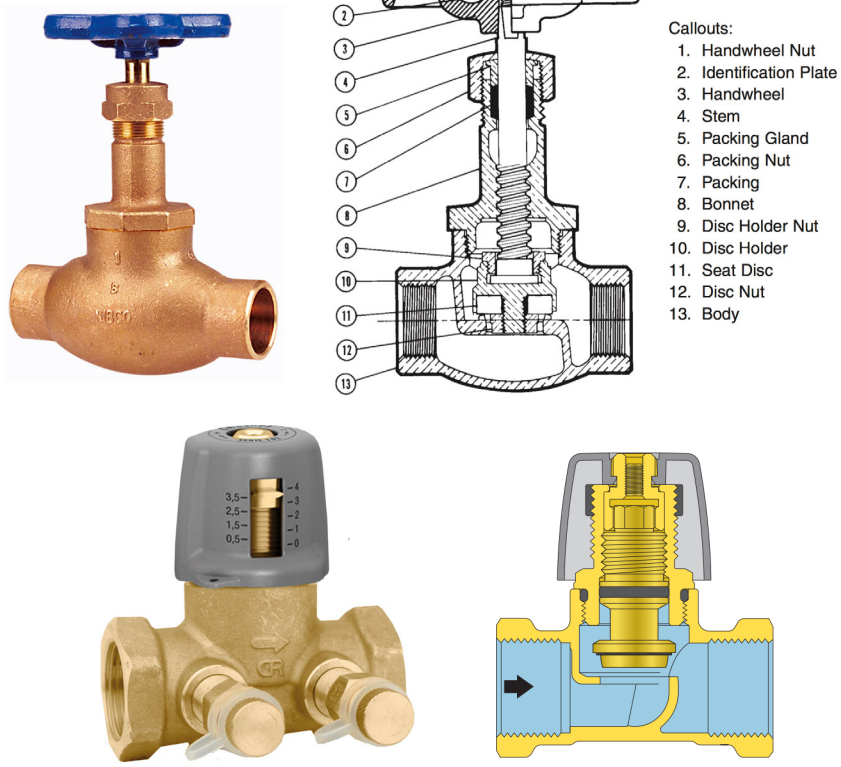
Of these, ball valves have largely taken over the market from gate valves in most residential and light commercial hydronic systems. Butterfly valves are used mostly in commercial systems, when the installation space along a piping path is minimal.

Gate valves are still available, but should only be used for component isolation, and should always be in a fully open or fully closed position. Leaving a gate valve in a partially open position will result in rattling sounds created by the valve's disc, and eventual deterioration of the valve's internal surfaces.

Butterfly valves are typically fitted to piping using flanges. They take up very little linear space along the piping path. As with gate valves, they should only be fully open or fully closed and never used for flow regulation.

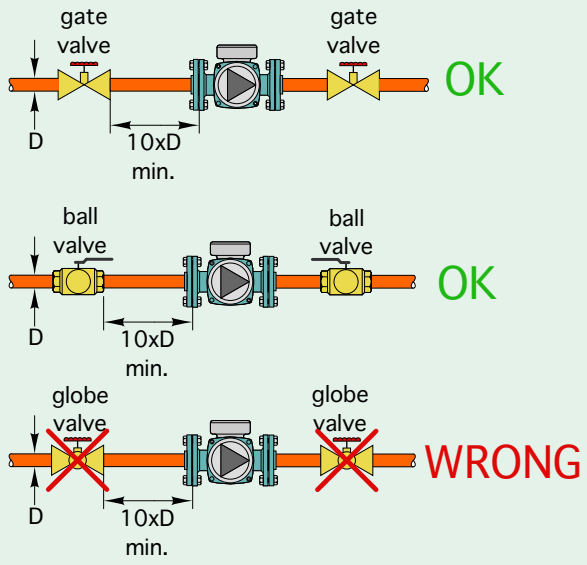
Valves intended for flow regulation have internal designs that are very different from valves intended for component isolation. *These valves are designed to induce head loss to the flow stream, which is necessary to reduce flow rate.* It's helpful to think of flow-regulating valves as "brakes" on the fluid stream.

Figure 6-2



Many valves intended for flow regulation have internal design that forces flow through a highly curved path, as well as through a gap between the valve's disc and its seat. This geometry is called a globe-type valve. Figure 6-2 shows examples of valves intended for flow regulation.

Figure 6-3



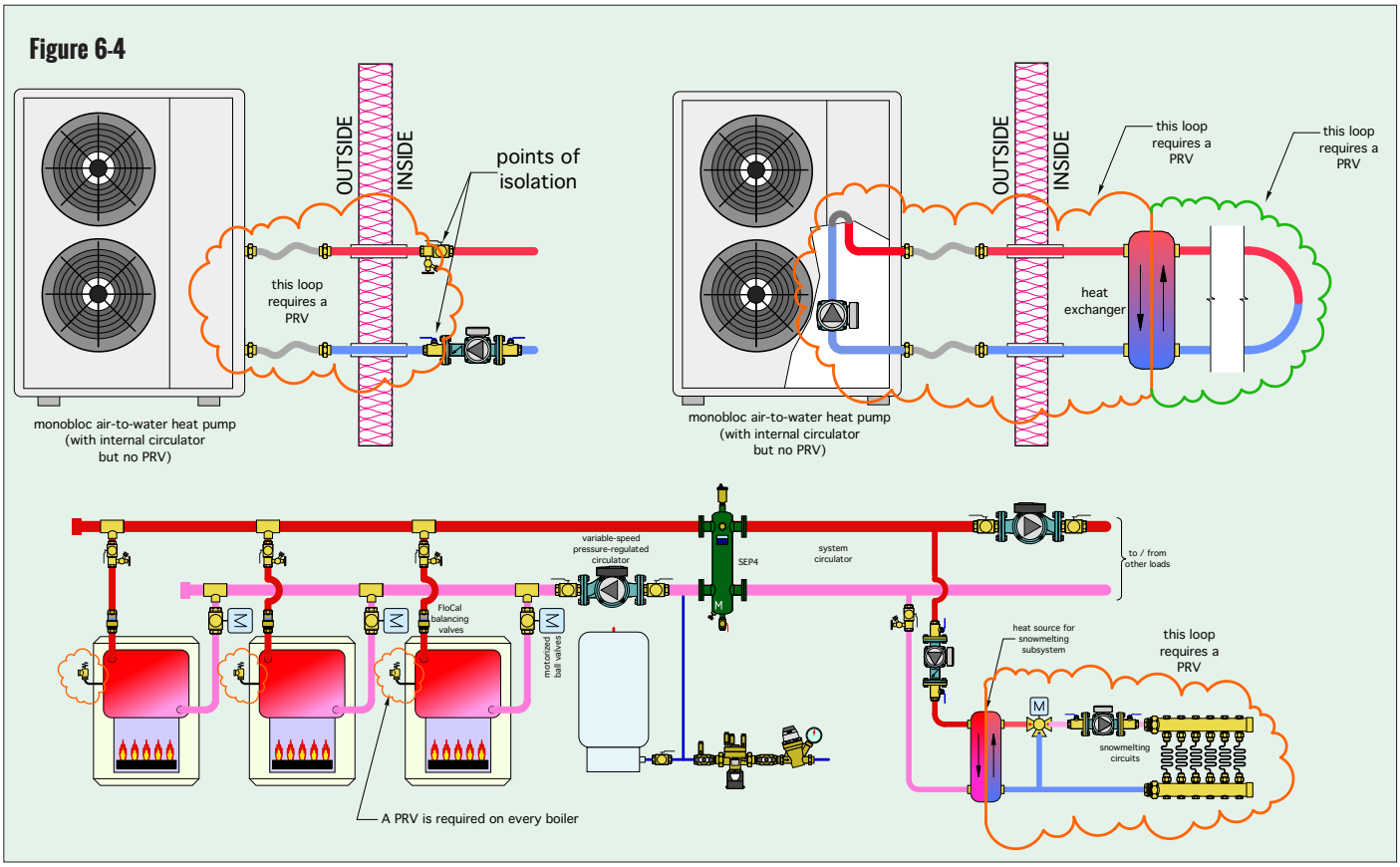
Because flow-regulating valves are designed to create and sustain head loss in the circuit, they should never be installed as a substitute for a component isolation valve. Doing so would impose much higher head loss on the circuit whenever there's flow through it, which would be most of the operating hours over the service life of the system. That head loss increases the required power input to a circulator to maintain a given flow rate. *Using a globe-type valve for component isolation is like driving a car with the brakes partially applied at all times.*

Flow-regulating valves using a globe-type internal design should always be installed so that flow passes upward through the valve's seat and then through the gap between the valve's seat and its disc. The correct flow direction is typically shown as an arrow cast onto the valve's body. Installing a globe-type valve backwards will cause noise, unsteady flow, and possibly flow cavitation that eventually damages the valve.

INADVERTENT ISOLATION OR LACK OF A PRESSURE-RELIEF VALVE

Any closed hydronic circuit that contains a heat source must have a pressure-relief valve. These situations constitute a rigid closed container filled with an incompressible fluid (water or an antifreeze solution). Any increase in

Figure 6-4



temperature creates rapidly rising pressure. Without a suitable pressure-relief valve, such situations could end with the rupture of the weakest component in the closed loop.

In relatively simple systems, this requirement is routinely met by installing a pressure-relief valve on the boiler. Millions of hydronic systems have this type of protection. However, there are systems with multiple heat sources, or designed in ways that isolate one closed-loop circuit from another through a heat exchanger. In these systems, designers need to identify any portions of a system that could be isolated and also contain a device capable of adding heat to that portion of the system. Examples of such heat sources include boilers, heat exchangers, heat pumps and electric heating elements.

Figure 6-4 shows examples of systems in which a closed loop that includes a heat source could be formed when certain valves are closed, or when certain components such as a heat exchanger are used. All such piping assemblies must be equipped with a pressure-relief valve that, as a minimum, meets code requirements, and secondly has a rated opening pressure lower than that of the component in the isolated loop with the lowest pressure rating.

It's also recommended that every *intentionally* created closed-loop subsystem be equipped with an expansion tank. This tank will reduce pressure variations in the closed loop, which reduces the likelihood that the pressure-relief valve will open when the closed loop is heated.

INADVERTENT ISOLATION OF AN EXPANSION TANK

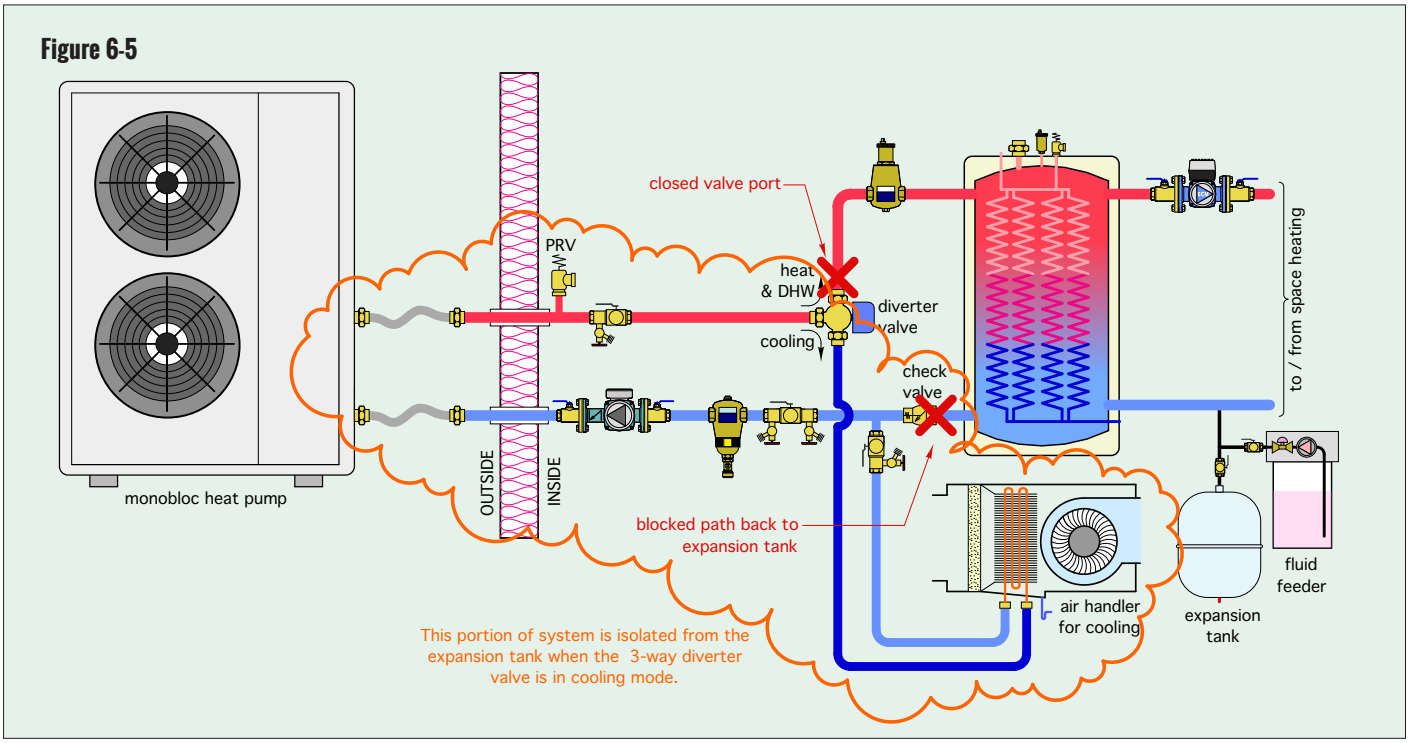
Another design review detail is the ability of a single expansion tank to "communicate" with the entire fluid volume in a closed-loop system in every possible operating mode. On simple systems, this is easily accomplished by installing the expansion tank in the piping that's common to all branches in the system. It could be at or near the heat source, at the central air separator, or teed into the return header that collects flow from all zones.

The person reviewing the design should look for an unblocked pathway between any fluid in the system and the point of connection of the expansion tank in every operating mode. *It's especially important to ensure that this path exists in systems containing check valves and/or motorized valves.* Figure 6-5 shows an example of where the combined use of a motorized diverter valve and a check valve can isolate a portion of the system from the expansion tank during one operating mode of the system.

The check valve in figure 6-5 is intended to prevent cool water migration back into the reverse indirect water heater, which is heated throughout the year to provide domestic hot water.

When the diverter valve is in the cooling position, the fluid's pressure equalization path to the expansion tank that passes through the upper portion of the reverse indirect tank is blocked. The check valve also prevents any fluid

Figure 6-5



expansion in the cooling portion of the system from reaching the expansion tank through the lower portion of the indirect tank. When the cooling cycling is operating, the fluid in this portion of the system is chilled and may slightly contract. A small amount of fluid might pass from the expansion tank into the cooling port of the system. However, when the cooling cycle ends, and the fluid in this portion of the system slowly warms up, the check valve will backseat, preventing the expanding fluid from reaching the expansion tank. This will lead to "dribbling" (slow and minor volume leakage) at the pressure-relief valve. Although the amount of fluid lost during each of these cycles is small, the fluid feeder will replenish the fluid, and fluid losses through the relief valve will accumulate over time.

There are several possible remedies to prevent this problem:

1. If the heat pump contained a small internal expansion tank, this scenario would not occur.

2. If the check valve was not present, this scenario would not occur.

3. Adding a small expansion tank somewhere within the cooling circuit would also prevent this issue.

4. Setting up controls that would automatically return the diverter valve to the heating position at the end of each cooling cycle would allow pressure equalization.

UNDERSIZED HEAT EXCHANGERS

Heat exchangers are very useful components in many types of hydronic heating and cooling systems. They allow heat to move from one fluid to another without any contact between the two fluids, and in situations where the two fluids are at different pressures.

Heat exchangers impose a "thermal penalty" whenever they are used. To transfer heat from one fluid to another, there must be a temperature difference between those fluids. The fluid supplying the heat must be at a higher temperature than the fluid receiving the heat.

The higher temperature fluid supplying the heat is typically coming from a heat source such as a boiler, heat pump or solar thermal collector. The presence of the heat exchanger forces any of these heat source to operate at a higher temperature than would be necessary if no heat exchanger were present.

The temperature difference between the fluid supplying the heat into the heat exchanger and the fluid carrying that heat away from the heat exchanger is called the "approach temperature difference." If cost and physical size of the heat exchanger were of no concern, designers could select large heat exchangers that would make the approach temperature difference almost zero. This would also make the thermal penalty of having the heat exchanger approach zero. However, in real systems, cost and size are important considerations when selecting a heat exchanger.

In situations where a heat pump is supplying the heat, a suggested maximum approach temperature difference is 5ºF. Even lower approach temperature differences are desirable when economically viable.

See *idronics* #29 for more information on sizing heat exchangers.

Design errors associated with heat exchangers used in hydronic systems also include:

1. Selecting a heat exchanger that's too small to transfer a given rate of heat at the desired approach temperature difference.

2. Not piping the heat exchanger for counterflow (e.g., where the two fluid pass in *opposite directions* through the heat exchanger.

3. Failure to protect the heat exchanger from internal scaling due to dissolved minerals or other contaminants in either fluid.

4. Failure to keep the internal surface of the heat exchanger clean from dirt or magnetite in the flow streams.

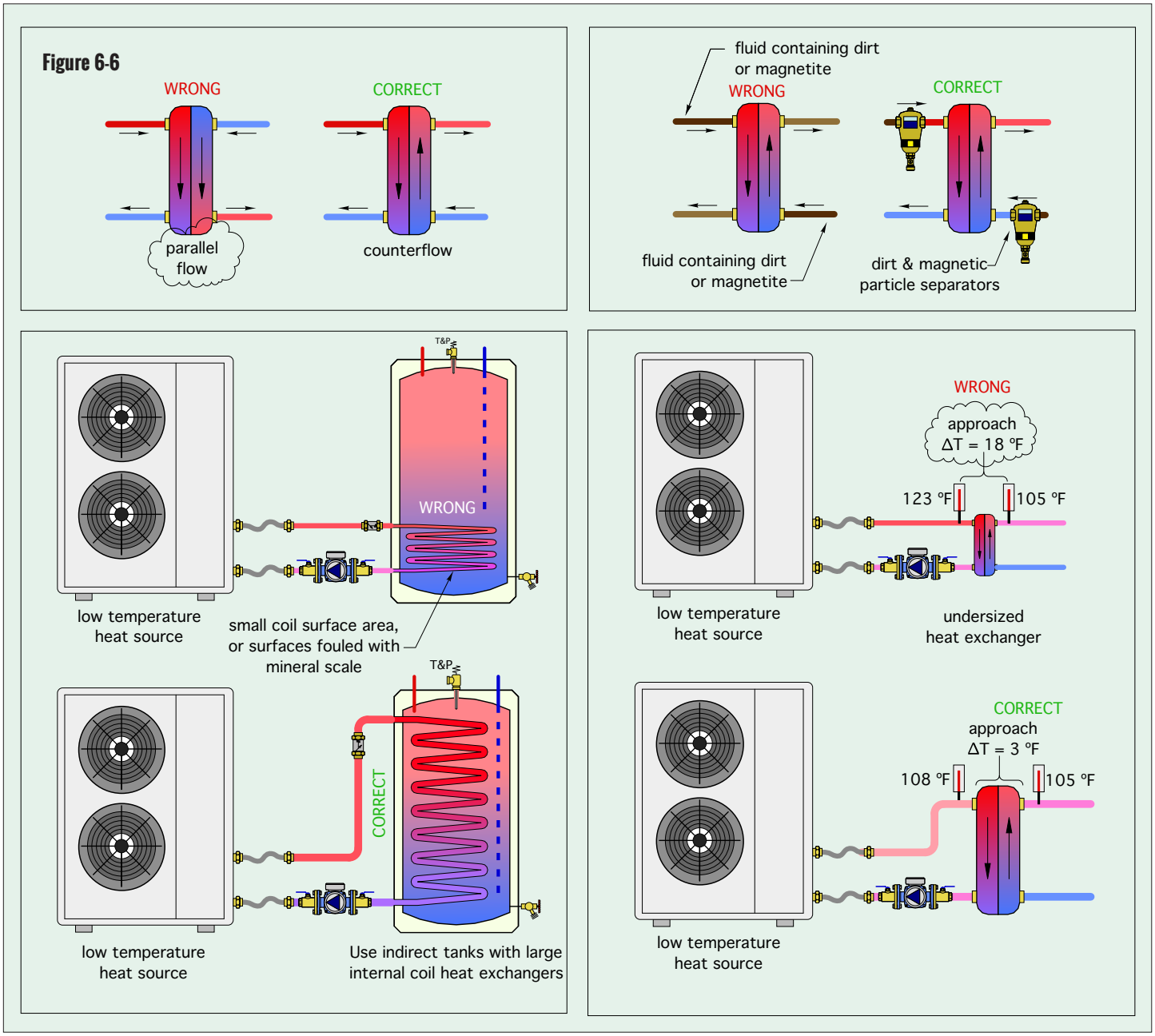
5. Failure to provide a means of internally cleaning all surfaces of a heat exchanger whenever possible.

6. Overestimating the ability of natural convection heat transfer to remove heat from the external surface of coil heat exchangers immersed in water within a tank (i.e., indirect water heaters, or copper coils immersed in non-pressurized thermal storage tanks).

7. Failure to account for the head loss and pressure drop created by flow through the heat exchanger.

Figure 6-6 illustrates several incorrect design details associated with heat exchangers.

Figure 6-6


WRONG
CORRECT
parallel flow
counterflow
fluid containing dirt or magnetite
WRONG
CORRECT
fluid containing dirt or magnetite
dirt & magnetic particle separators
T&P
WRONG
low temperature heat source
small coil surface area, or surfaces fouled with mineral scale
T&P
CORRECT
low temperature heat source
Use indirect tanks with large internal coil heat exchangers
WRONG
approach ΔT = 18 °F
123 °F
105 °F
low temperature heat source
undersized heat exchanger
CORRECT
approach ΔT = 3 °F
108 °F
105 °F
low temperature heat source

7. EXAMPLE SYSTEMS

This section presents systems that contain multiple design errors, all of which were discussed in previous sections. A system schematic containing design errors is shown, the errors it contains are described, and a reconfigured system with the same functionality but using alternate design details is presented.

SYSTEM #1 : (Figure 7-1a)

The designer's intent was to use a single-speed monobloc air-to-water heat pump for heating and cooling. Heating on the 1st floor would be handled by six zones of low thermal mass floor heating. Heating and cooling for the second floor would be handled by a ducted air handler. An electric boiler would be installed for supplemental and backup heating. All the distribution piping is 1" bare copper. Examine the schematic for design errors.

The design errors in system #1 include:

1. The heat pump and boiler are in *series*. If either is operating while the other is off, the latter acts as a heat dissipator, needlessly releasing heat into the mechanical room or outside. If either heat source must be removed for service, the entire system must remain off. The head loss of both heat sources is additive regardless of their operating status. If the boiler was the sole heat source, flow returning from the heat emitters would have to pass outside the building and through the heat pump prior to returning to the boiler. This would increase heat losses outside of conditioned space.

2. There are two valves that can isolate the heat pump from the remainder of the system. There is no pressure-relief valve inside the heat pump. If these valves were closed and the heat pump was unintentionally operated in heating mode, the pressure within the heat pump would rise rapidly with no means of accommodating the fluid expansion. Eventually, some component within the heat pump could rupture due to excess pressure. The pressure-relief valve on the electric boiler cannot protect the heat pump if the two isolation valves are closed.

3. The floor heating manifold station is highly zoned. The heat pump is an on/off (single speed) unit. If only one or two floor-heating circuits were active, the heat pump would likely short cycle.

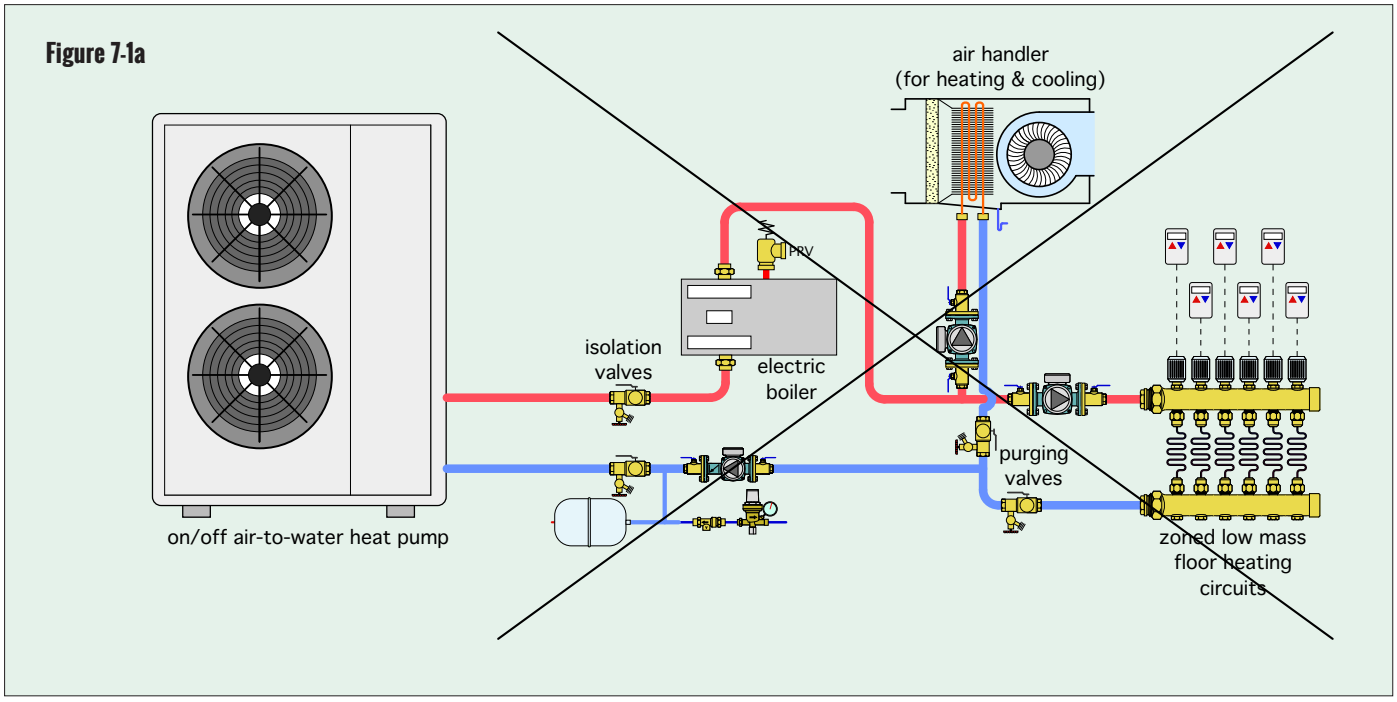
4. The circulator for the heat pump and boiler is pumping *toward* (rather than away from) point of connection of the expansion tank.

5. Expansion tanks should not be mounted horizontally due to the possibility of over-stressing and breaking the connection to the tank shell — especially if the tank became waterlogged.

6. As shown, this system operates with water, and there's no means of freeze protection for the heat pump during a power outage of long duration.

7. The backflow preventer should always be mounted upstream of the pressure-reducing valve.

Figure 7-1a



8. The piping to and from the coil in the air handler does not create a counterflow heat exchanger between the water flow and air flow. This reduces the heat transfer rate of the coil.

9. The two distribution circulators are — to some extent — in series with the heat pump/boiler circulator. There is no means of hydraulic separation between the circulators.

10. The two distribution branches lack check valves to prevent reverse flow when one branch is operating and the other is off.

11. The purging valves for the two branches are both mounted backwards (e.g., the hose-bib portion of the purge valve must be upstream of the inline ball valve).

12. The system lacks a central air separator.

13. The system lacks a central dirt and magnetic particle separator.

14. The circulator for the air handler is mounted very close to a tee. There should be a minimum of 10 pipe diameters of straight pipe upstream of any circulator to reduce turbulence at the inlet of the impeller.

15. There are no flex connectors between the rigid piping and the heat pump. The rigid piping could transfer vibration from the heat pump into the remainder of the system. Slight movement of the heat pump due to settling of its stand or strong wind gusts can also increase mechanical stress on the rigid piping.

16. The uninsulated copper tubing, as well as other uninsulated piping components — such as the electric boiler — that are in the path of chilled water flow from the heat pump to the air handler, will quickly develop condensation and begin dripping that condensate when the system operates in cooling mode.

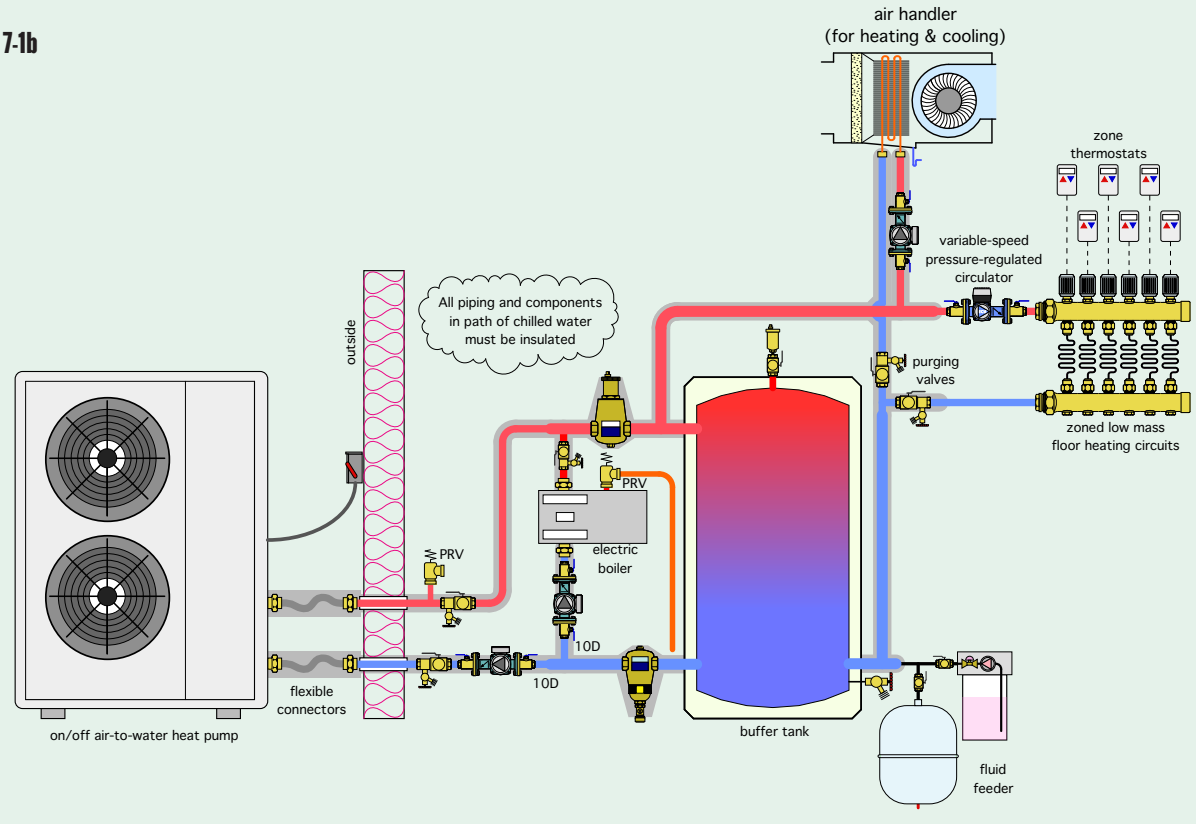
17. 1" copper tubing may be too small to accommodate the combined heat output and associated flow requirement when the two heat sources operate simultaneously.

18. There is no means of differential pressure control for the zoned floor-heating circuits.

19. There is no discharge piping on the pressure-relief valve. Most codes require this pipe to end within 6 inches of the floor.

Figure 7-1b shows one way to eliminate these design issues.

Figure 7-1b



This design uses a properly sized buffer tank to prevent the heat pump from short cycling during periods of low demand. The heat pump and boiler are piped in parallel, each with their own circulator, both of which contain an internal check valve. Either heat source could be isolated from the remainder of the system if necessary for service, and without shutting off the other heat source. A pressure-relief valve is included on the heat pump side of the isolation valve to prevent over-pressure if the heat pump operates when the isolation valves are closed. Flexible connectors are used between the heat pump and rigid piping to reduce noise and vibration transfer.

The buffer tank has been piped in a 3-pipe configuration. This allows heat from either heat source to flow directly to the load when either heat source is operating at the same time as a load circuit.

The headers have been "generously" sized so that the maximum flow velocity is 2 feet per second. This, in combination with the buffer tank, provides hydraulic separation of all circulators in the system.

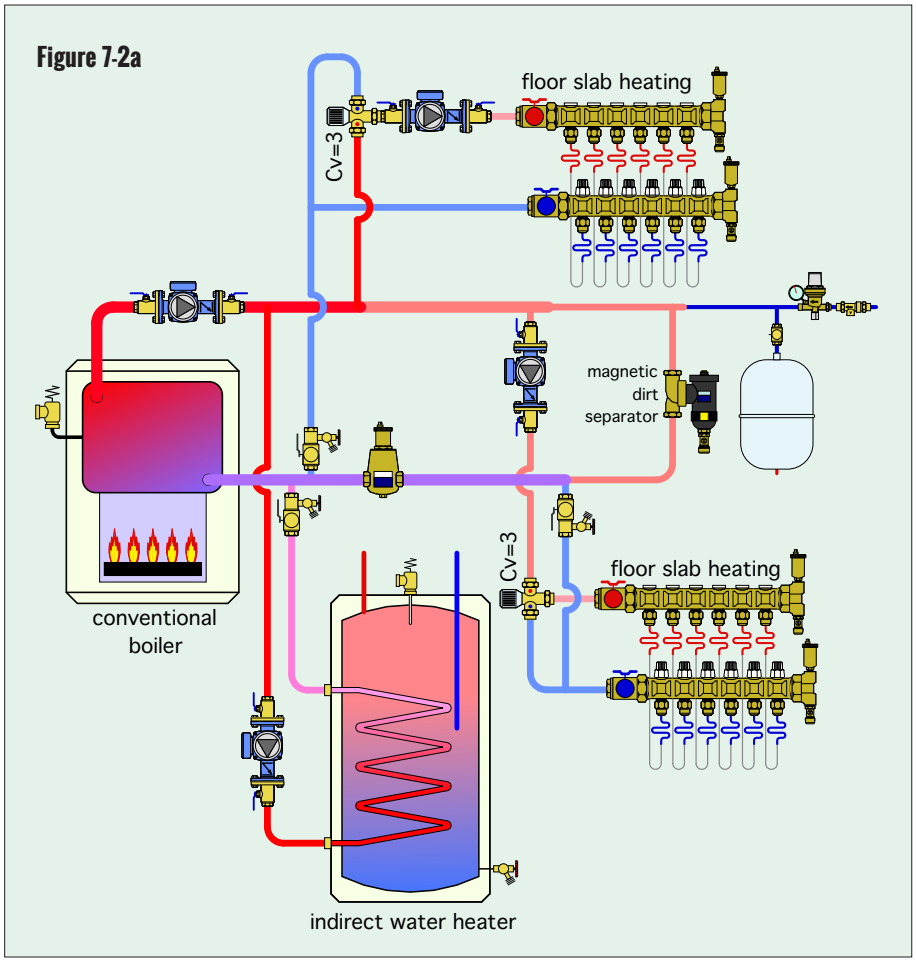
The air handler and floor-heating manifold station are also piped in parallel. Both are supplied by a circulator with an internal check valve. *All heat emitters are sized to operate at the same temperature to eliminate the need for a mixing assembly.* A variable-speed pressure-regulated circulator is used for the floor-heating manifold station. This circulator automatically adjusts its speed to maintain a constant differential pressure as the individual manifold valve actuators open and close.

A fluid feeder is used to maintain system pressure resulting from very minor losses over time.

Piping to and from the air handler's coil has been configured so that the coil is a counterflow heat exchanger.

All piping and components in the path of chilled water flow to the air handler are insulated and vapor sealed to prevent condensation.

Figure 7-2a



SYSTEM #2 : (Figure 7-2a)

The designer's intent was to create a system with two independent zones of floor slab heating, *both operating at the same supply water temperature*. The system will also have an indirect domestic water heater. All heat will be supplied from a gas-fired cast iron boiler. 3-way thermostatic mixing valves with Cv=3.0 rating are used to reduce the water temperature to both manifold stations. The expected flow rate through each floor-heating circuit is 1 gpm. The designer plans to use a bypass circulator (seen above the boiler) to boost the return water temperature to the boiler. Examine the schematic for design errors.

The design errors in system #2 include:

1. *The boiler is not protected against sustained flue gas condensation.* Although the "bypass circulator" will boost inlet temperature somewhat, it cannot thermally disconnect the load circuits from the boiler when necessary. When the slabs are warming up (e.g., transient conditions), they will absorb heat significantly faster than the boiler can produce it. Thermal equilibrium will dictate the temperature leaving the boiler, and it is likely to be significantly lower than the boiler inlet temperature required to prevent flue gas condensation.

2. The distribution system morphs the concept of a primary loop with a 2-pipe header system. *The result is an undefined piping assembly.* Assuming that the bypass circulator is wired to operate whenever any load circuit operates, it will create a differential pressure between the upper and lower portions of this loop. Any flow resistance through the dirt separator will increase this differential pressure. It's likely that the differential pressure will be high enough to push flow past the spring-loaded check valves in the other (not operating) zone circuits. This will lead to heat migration into spaces that are not calling for heat.

3. There is no provision for hydraulic separation of the circulators. Given the unanticipated differential pressure conditions that could develop, it's likely that there will be interference among simultaneously operating circulators.

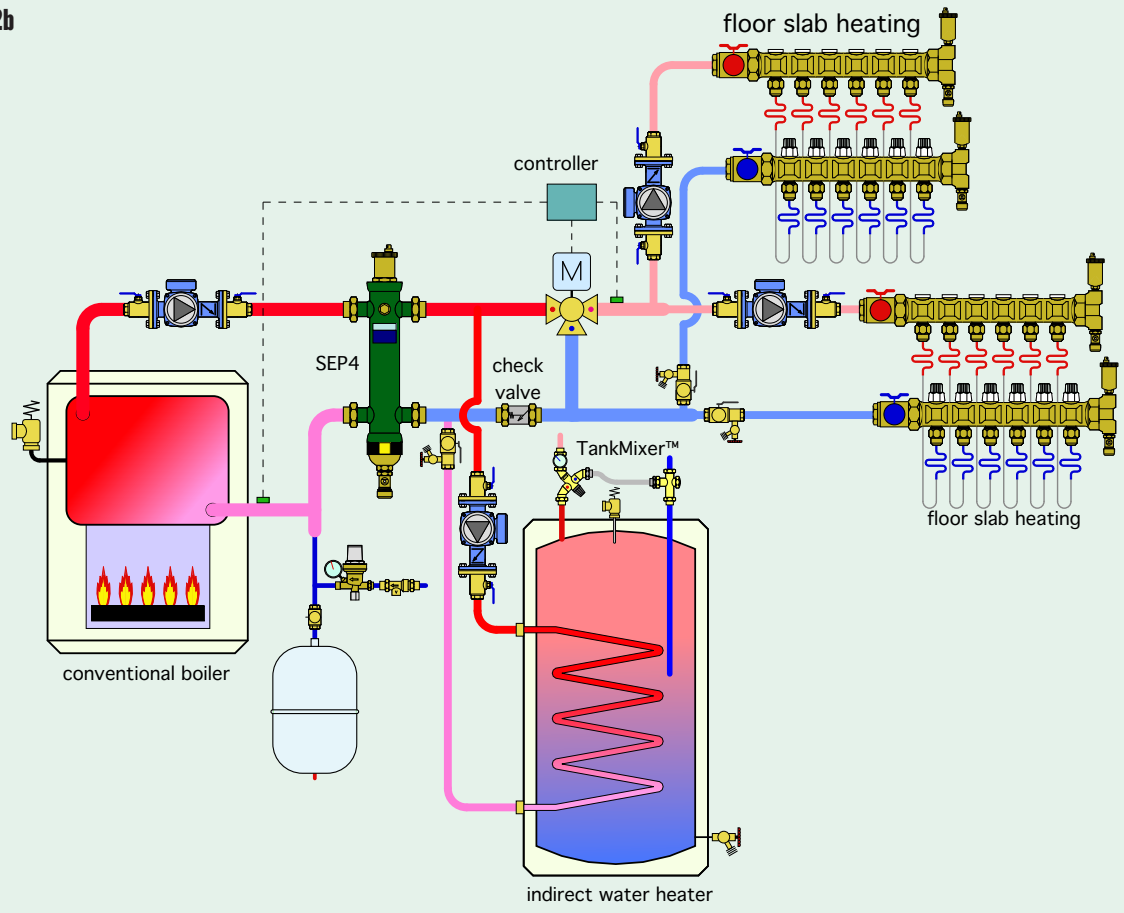
4. *The coil of the indirect water heater is piped in reverse.* The hottest fluid should enter the top of the coil and flow downward through the coil to create counterflow heat exchange conditions that maximize the rate of heat transfer from the system water to the domestic water.

5. Although *air and dirt separators are present, they are not in locations that optimize their function.* The air separator should be located where the water is hottest. This reduces the solubility of dissolved air molecules in the water, allowing microbubbles to form and be subsequently captured. The dirt separator should be upstream of any components or circuits containing hardware that could be adversely affected by dirt or magnetite accumulation. In this system, that would be upstream of the boiler.

6. The two thermostatic mixing valves have relatively flow coefficients (e.g., Cv=3.0). This will cause higher than necessary pressure drop when flows in the range of 6 gpm are anticipated through each floor-heating manifold station. Furthermore, neither of these valves can protect the boiler against sustained flue gas condensation.

7. The circulator serving the lower floor-heating manifold is improperly located in two respects: A) It must be located between the mixing valve and the manifold station to allow proper mixing. B) its inlet is directly against the side port of the tee, and thus it doesn't have the recommended 10 pipe

Figure 7-2b



diameters of straight pipe upstream of its inlet. This will increase turbulence at the circulator inlet.

8. The circulator serving the upper floor-heating manifold is also mounted with its inlet very close to a tee and thus is subject to turbulent flow.

9. The indirect water heater is not equipped with an anti-scald mixing valve. This is an essential detail to prevent scalding should the temperature of the water in the tank exceed 120ºF.

10. The return piping from the upper and lower manifold stations should be equipped with a spring-loaded check valve upstream of the purging valve in each circuit. The check valve prevents the possibility of reverse flow through the porting in the mixing valves that may not be fully closed when the associated circuit is off.

11. The purging valve on the return side of the circuit servicing the coil of the indirect water heater is mounted upside down.

Figure 7-2b shows an alternate design that corrects these design errors.

Since both radiant floor-heating manifolds are intended to operate at the same supply temperature, they can be supplied through a single (properly selected) mixing valve. In this case, it's a motorized 3-way mixing valve with an electronic controller. The controller monitors both supply water temperature and *boiler inlet temperature*. When the latter drops below a minimum return temperature (typically about 130ºF), the mixing valve begins restricting the hot water inlet to the 3-way valve as necessary to restore the boiler inlet to its minimum temperature and thus prevent sustained flue gas condensation. The motorized mixing valve would be selected to have a Cv at least as high as the sum of the flow rates going to each manifold station at design load.

A single spring-loaded check valve prevents the possibility of flow reversal through the 3-way mixing valve when only the indirect water heater circuit is operating.

The coil of the indirect water heater is piped to provide counterflow heat exchange.

A thermostatic point-of-distribution mixing valve (Caleffi TankMixer™) has been added to the domestic water connections on the tank to limit the delivery temperature of domestic hot water.

Air, dirt and magnetic particle separation, as well as hydraulic separation of the circulators, is provided by the Caleffi SEP4 separator.

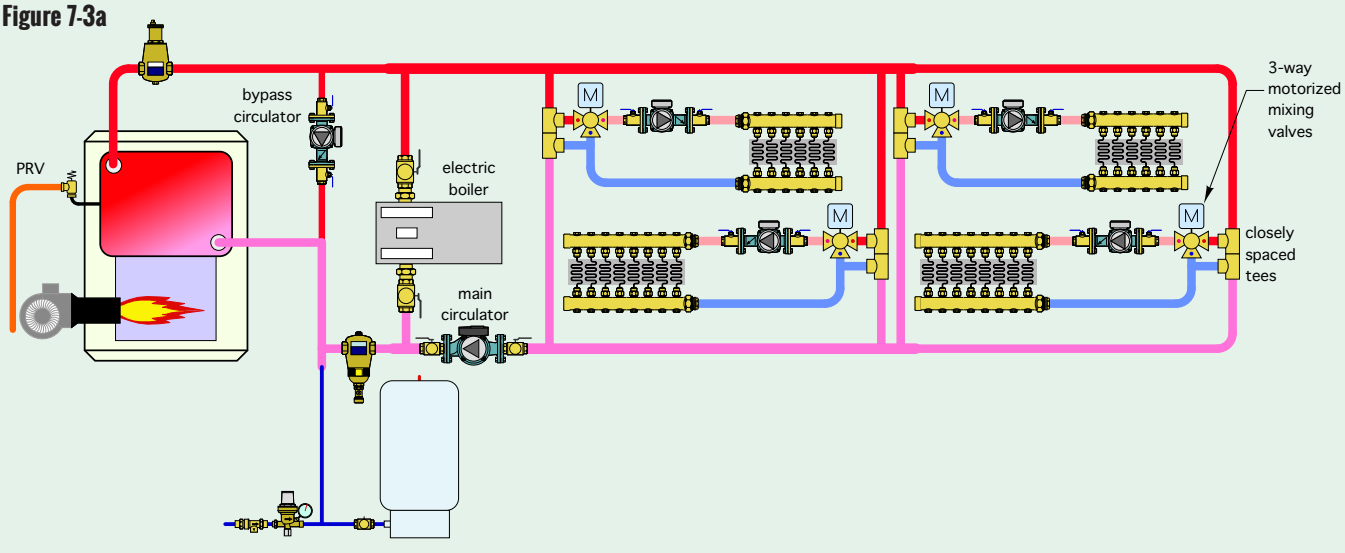
All purging valves are located to allow forced water purging of the entire piping system.

SYSTEM #3 (Figure 7-3a)

This system is based on the combined use of an oil-fired boiler and electric boiler as heat sources for four zones of low-temperature floor heating. The electric boiler would be used during periods of low-cost electricity, as well as act as a supplemental heat source if needed.

The system has four zones of slab heating, all of which are designed to operate at the same supply

Figure 7-3a



water temperature. The designer realizes that the cast iron boiler would operate with sustained flue gas condensation if the water returning from the floor-heating circuits went directly into the boiler. To prevent this, the designer decides to operate the oil-fired boiler at a high water temperature (180ºF) and install a bypass circulator to boost boiler inlet temperature. Because this temperature is much higher than needed by the floor circuits, each of the four zones is equipped with a 3-way motorized mixing valve and circulator. Each of the four zones connects to crossover piping using a pair of closely spaced tees. This was done to provide the same (high) supply water temperature to each mixing valve. The closely spaced tees hydraulically separate the zone circulators from the main circulator that creates flow through both boilers and the header system. Examine the schematic for design errors.

The design errors in system #3 include:

1. A boiler bypass circulator alone cannot provide *consistent* anti-condensation protection of the oil-fired boiler. When any of the slab zones are warming up (e.g., transient conditions), the load they impose on the system can be several times the design heating load. The only way to ensure consistent anti-condensation protection is to use a device that reacts to temperature changes and limits heated water flow to the load when necessary. A Caleffi ThermoProtect valve can provide this requirement.

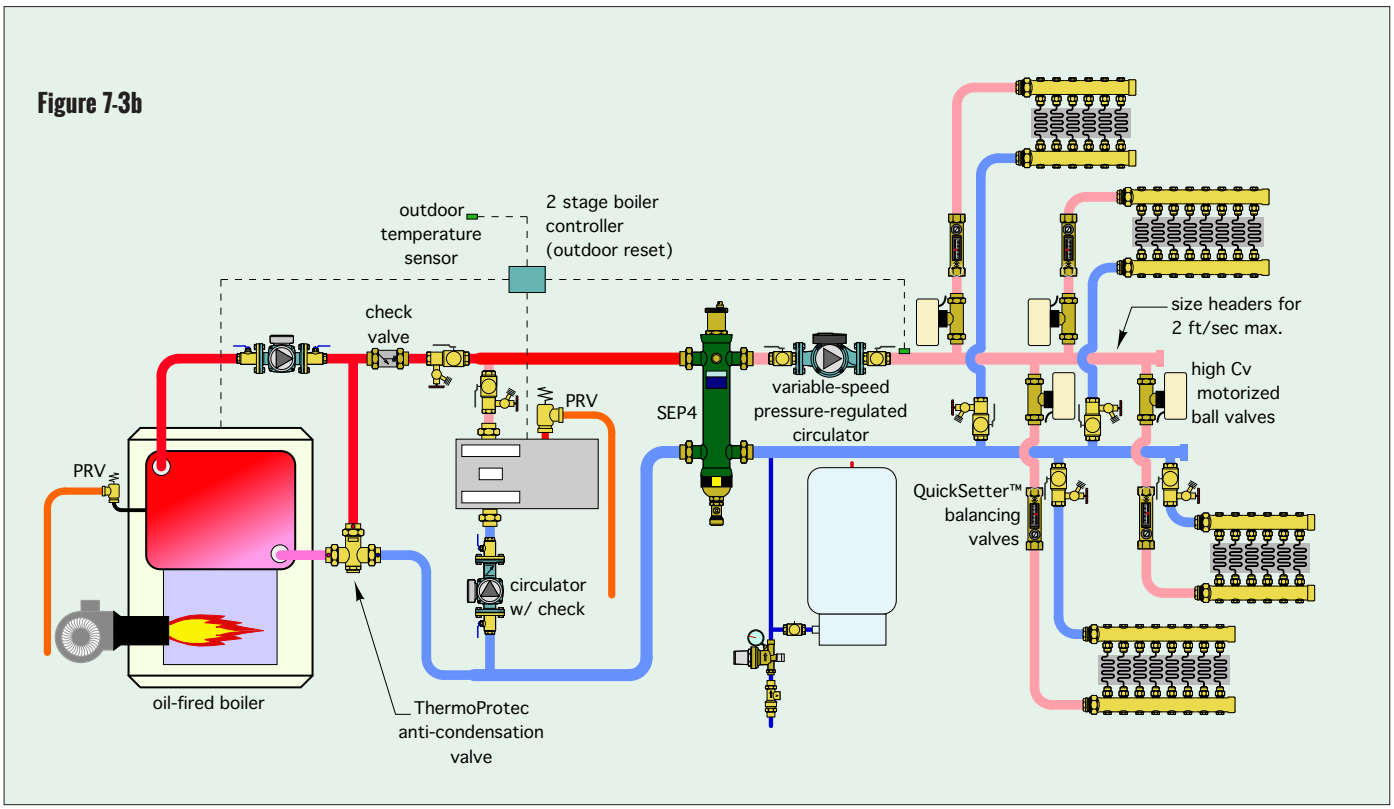
2. The "main circulator" creates flow through the oil-fired boiler and the electric boiler whenever they are operating. If one of the boilers is off, it is acting as a heat sink for the other operating boiler. Each heat source should have its own means of flow control (e.g., circulator or motorized valve) so that flow only passes through that heat source when it is operating.

3. The "main circulator" is pumping toward rather than away from the expansion tank's connection point. This will cause a drop in system pressure on the inlet side of the main circulator, possibly leading to cavitation.

4. The dirt separator, mounted as shown, is protecting the oil-fired boiler but not the electric boiler. It should be mounted upstream of both boilers.

5. The cost of the four motorized mixing valves is hard to justify given that all zones were designed to operate at the same supply water temperature. Even if the zones had different upward heat output requirements, design measures, such as adjusting the tube spacing in each zone to bring the supply temperatures close to the same value and thus allow a single mixing assembly, are possible. This would lower cost compared to using four motorized mixing valves.

Figure 7-3b



6. There are no balancing valves on the crossovers that supply each set of closely spaced tees. This will cause higher flow rates through crossovers closer to the main circulator, and vice versa.

7. The are no purging valves on the individual zones. Creating forceful purging into an assembly connected to the crossovers by closely spaced tees is very difficult (due to the hydraulic separation the tees create).

8. There is no pressure-relief valve on the electric boiler.

9. There is no device capable of capturing magnetic particles entrained with flow.

Figure 7-3b shows one approach that corrects for these design errors.

The oil-fired boiler and the electric boiler are now piped in parallel, each with their own circulator. The oil-fired boiler is equipped with a Caleffi ThermoProtec anti-condensation valve. This valve allows the water to "recirculate" through the boiler without being released to the load until the boiler inlet temperature is high enough to prevent sustained flue gas condensation. A check valve is installed downstream of the tee connecting to the hot port of the ThermoProtec valve. It prevents any possibility of reverse flow through the oil-fired boiler when the electric boiler is operating, or if the shuttle inside the ThermoProtec valve doesn't completely block the path between the valve's hot and cold ports.

The boilers are controlled by a 2-stage outdoor reset control. The operating order of the boilers is determined based on the electrical rate. When rates are low during off-peak hours, weekends, and certain holidays, the electric boiler is the first to operate. When electrical rates are higher, the oil-fired boiler is the first to operate. The firing order is "exchanged" using a 4PDT relay operated based on a timer programmed for the electrical rate schedule. The 2-stage outdoor reset control continually calculates the "target" supply water temperature for the floor heating circuits. It operates the boilers as needed to maintain the temperature at the supply sensor as close as possible to this target value, but still allowing for an on/off differential that prevents either boiler from short cycling

The SEP4 hydraulic separator serves as a mixing chamber where hot water from the boilers is combined with return water from the four manifold stations. It also provides air, dirt and magnetic particle separation for the system.

A single variable-speed, pressure-regulated circulator, set for constant differential pressure mode, provides flow for the distribution system. Flow into each zone is controlled by a high Cv zone valve or motorized ball valve. If the outdoor reset schedule in the boiler controller is set properly, the average supply water temperature will be just high enough to meet the current heating load. This keeps the zone valves open most of the time.

This system is much simpler that the original design, and its installed cost should also be lower.

SUMMARY

We conclude this issue by restating its opening paradox:

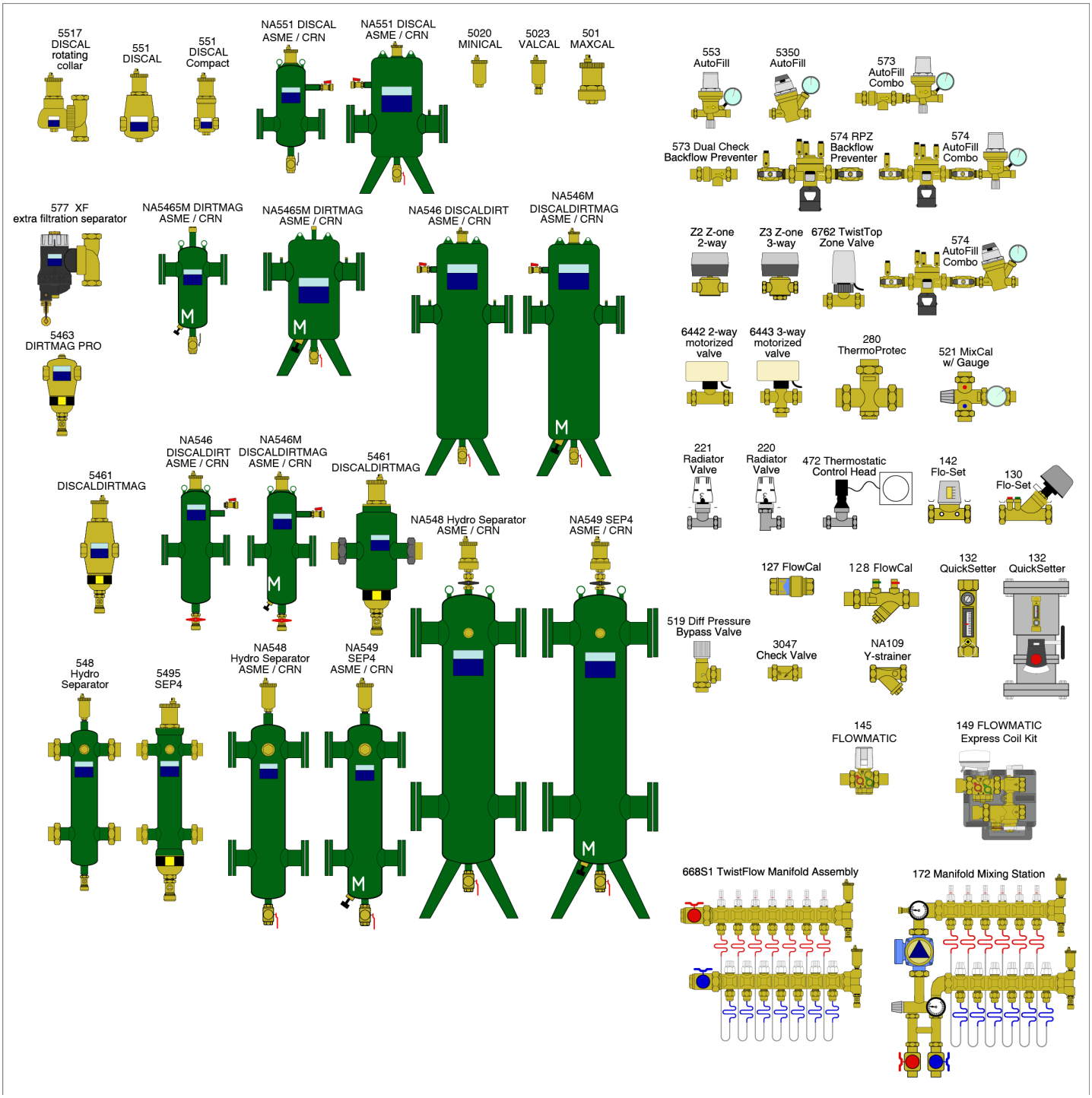
One of the ***best*** things about hydronic heating and cooling systems is that there is virtually no limit to the type of systems that can be created.

One of the ***worst*** things about hydronic heating and cooling systems is that there is virtually no limit to the type of systems that can be created.

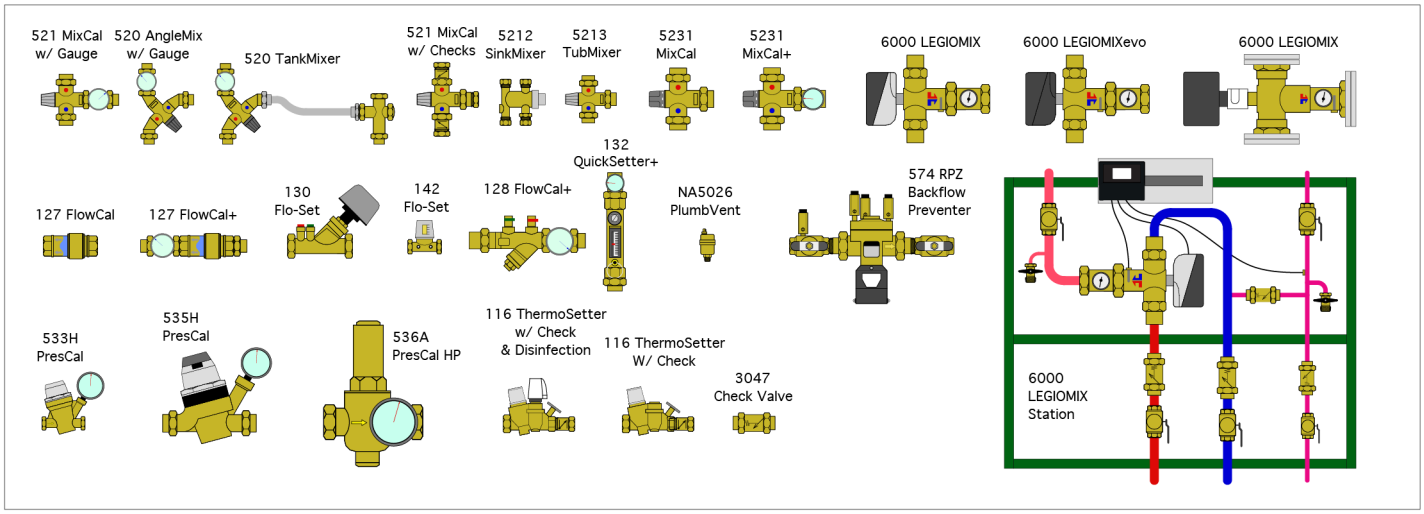
This issue has identified many common design-related errors that tend to reoccur in the North American hydronics industry. By recognizing these errors during the design review process, understanding their root cause, and taking steps to correct them prior to installation, designers can eliminate the high cost and customer frustrations associated with correcting errors on site.

Fundamental concepts such as hydraulic separation, proper valve selection, thermal equilibrium, hydraulic equilibrium, and proven piping configurations don't change, regardless of the scale of the system. Professional designers understand these concepts and respect their immutability. They also leverage them to refine their designs into the simplest, most reliable, and lowest cost configurations that meet project requirements.

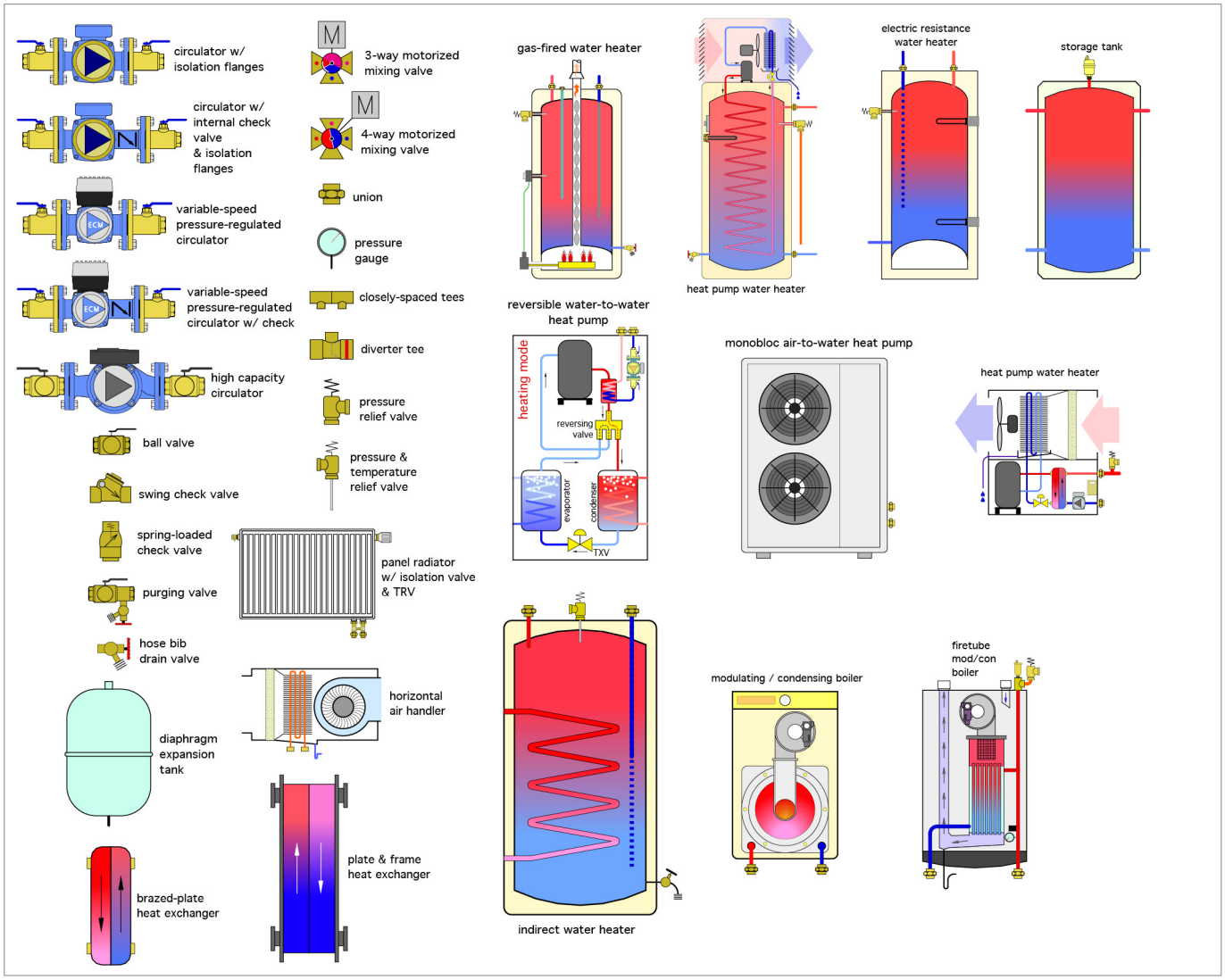
APPENDIX A: CALEFFI HYDRONIC COMPONENTS



APPENDIX B: CALEFFI PLUMBING COMPONENTS



APPENDIX C: GENERIC COMPONENTS



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