january 2013 12

idronics™

JOURNAL OF DESIGN INNOVATION FOR HYDRONIC PROFESSIONALS

Hydronic Fundamentals

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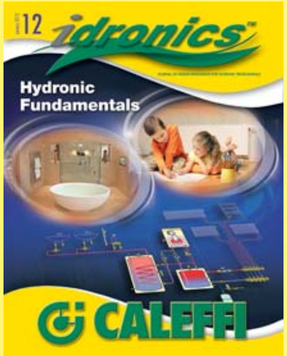

A Technical Journal
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Dear Hydronic and Plumbing Professional,

Students returning to school after summer break can attest that it's good for the professor to start with a review of the basics. Jumping into Thermodynamics 401 without a refresher can make grasping more advanced topics difficult. It's in this spirit that we created this 12th edition of idronics entitled Hydronic Fundamentals.

Feedback from our face to face training programs, as well as those offered online, frequently asks for explanation of basic design principles, or why certain installation practices are followed. So, like a professor giving a review at the start of a semester, this issue of idronics was created to review, refresh, and reinvigorate your understanding of the fundamentals that every hydronic system relies on. With a solid grasp of these fundamentals you'll be able to design efficient and reliable systems that deliver unsurpassed comfort. You'll also be better prepared for the more specialized topics that will be coming in future issues of idronics

This issue also makes frequent reference to past issues that have dealt with specific topics in more detail. We encourage you to get these previous issues by going to the idronics link at www.caleffi.us, and downloading the PDF files. While there, you can also register to receive hard copies of future issues.

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

Mark Olson

Mark Olson

General Manager & CEO

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Hydronic Fundamentals

1. INTRODUCTION

This is a special issue of idronics. Unlike past issues, which have focused largely on one topic, this issue broadly describes the fundamentals of hydronic systems. These fundamentals are the foundation of good system design and proper installation. When clearly understood and applied, they allow designers to create systems that deliver superior comfort, energy efficiency and long-term reliability. When ignored, the results are likely to be inefficient operation, wasted materials, excessive callbacks and unhappy customers.

This issue was outlined based on suggestions from many hydronic heating professionals. They indicated the need for an issue directed at those new to hydronics, as well as one that could refresh and clarify basic concepts to those already working with hydronic systems. This issue fills that need.

Starting from scratch, a thorough discussion of hydronics technology would require hundreds of pages. Fortunately, previous issues of idronics have already discussed many specific topics related to hydronic system design in considerable detail. This issue will leverage that information.

While reading through this issue, you will find frequent references to previous issues. Watch for the idronics icon shown below, followed by reference to one or more previous issues.

References to previous issues of idronics

All of these previous issues are freely available as downloadable PDF files under the idronics archive link at www.caleffi.us. You are encouraged to download all of them, and keep them available for reference.

2. BENEFITS OF HYDRONIC SYSTEMS

Modern hydronics technology is a “media” that skilled designers can use to create an appropriate heating or cooling solution for almost any building. The versatility of modern hydronics technology is unmatched by any other means of heating or cooling buildings.

This section highlights the benefits of modern hydronic systems, both to those designing and installing systems, and to the occupants of buildings that are heated or cooled using hydronics technology. It is essential that heating and cooling professionals understand the basis for these benefits and effectively communicate them to those who are trying to select a means of heating or cooling for their building.

COMFORT:

Every heating system affects the health, productivity and general well-being of numerous people over many years. The ability of that system to provide thermal comfort is of paramount importance and should be the primary objective of any heating system designer or installer.

Figure 2-1

Few people don't appreciate a warm comfortable house on a cold winter day. A place that helps them forget about snow, ice and wind as they walk in the door. A place that encourages a sense of well-being and relaxation.

Still, the average North American homeowner spends little time thinking about his or her heating system. Many view such systems as necessary but uninteresting. When construction budgets are strained, it is often the heating system that gets compromised to save money for other, seemingly more important amenities. In many cases, it is the homebuilder rather than the homeowner who selects the heating system. Builders often have different priorities from those who will live with, maintain and pay for the operation of that system.

In many cases, people who have lived with uncomfortable heating systems don't realize what they have been missing. Many would welcome the opportunity to live or work in truly comfortable buildings and would willingly spend more money, if necessary, to do so. Successful heating professionals take the time to discuss with potential clients the full range of benefits of hydronic systems as well as the price.

Contrary to common belief, comfort during the heating season is not solely determined by indoor air temperature. Comfort is achieved and maintained by controlling how the body loses heat. When interior conditions allow heat to leave a person's body at the same rate as it is generated, that person feels comfortable. If heat is released faster or slower than the rate it is produced, some degree of discomfort is experienced.

Figure 2-2

A normal adult engaged in light activity generates heat through metabolism at a rate of about 400 Btu/hr. The body releases this heat through several processes, including convection, radiation, evaporation and conduction.

For indoor environments in colder weather, thermal radiation and convection typically account for almost 75% of the total heat output from the body. Heat loss by thermal radiation alone can be 50% to 60% of the total heat loss, especially within buildings that have cold wall, floor or ceiling surfaces.

Properly designed hydronic systems control the air temperature as well as the surface temperature of rooms to maintain optimal comfort. A hydronically heated floor or ceiling can raise the average surface temperature of rooms. Since the human body is especially responsive to radiant heat loss, these warm surfaces significantly enhance comfort. Proper indoor humidity is also easier to maintain in buildings using hydronic heating due to reduced air leakage.

Factors such as activity level, age and general health determine what a comfortable environment is for a given individual. When several people are living or working in a common environment, any one of them might feel too hot, too cold or just right. Heating systems that allow different zones of a building to be maintained at different temperatures can adapt to the comfort needs of several individuals. Hydronic heating systems are easy to zone using several different approaches.

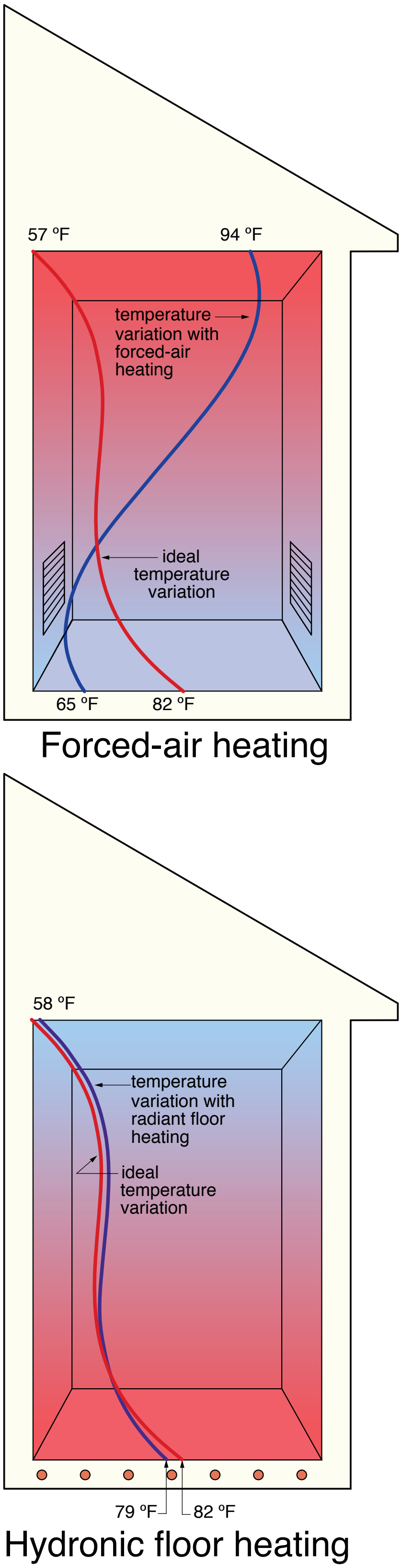
See idronics #5 for a complete discussion on how to zone hydronic heating systems.

ENERGY SAVINGS:

Ideally, a building's rate of heat loss would not be influenced by how that heat is replaced. However, evidence has shown that otherwise identical buildings can have significantly different rates of heat loss based on the types of heating systems installed. Buildings with hydronic heating systems have demonstrated a tendency for lower heat loss compared to equivalent structures with forced-air heating systems.

There are several reasons for this reduced heat loss. One is that hydronic systems, unlike forced-air systems,

Figure 2-3



have very little effect on room air pressure while operating. The more significant changes in room air pressure created by forced-air systems can increase the rate of outside air infiltration or inside air exfiltration through the exterior surfaces. The greater the rate of air leakage, the greater the rate of heat loss.

Another factor affecting building energy use is air temperature stratification (e.g., the tendency of warm air to rise toward the ceiling while cool air settles to the floor). Stratification tends to be worsened by high ceilings, poor air circulation and heating systems that supply air into rooms at high temperatures. Maintaining comfortable temperatures in the occupied areas of rooms plagued with a high degree of temperature stratification leads to significantly higher air temperatures near the ceiling. This increases heat loss through the ceiling.

Hydronic heat emitters that transfer the majority of their heat output by thermal radiation reduce, and in some cases eliminate, undesirable air temperature stratification. This reduces heat loss through ceilings and allows comfort to be maintained at lower air temperatures.

Figure 2-3 shows a comparison between the undesirable air temperature stratification created by forced-air systems operating with poorly placed registers versus the desirable "reverse" stratification created by hydronic radiant floor heating.

LOW DISTRIBUTION ENERGY:

Properly designed hydronic systems use significantly less energy to move heat from where it is produced to where it is needed. A well-designed hydronic system, using a modern high-efficiency circulator, can deliver a given rate of heat transport using less than 10% of the electrical energy required by the blower of a forced-air heating system transporting heat at the same rate. This is a very important advantage of hydronic systems—one that can save thousands of dollars in operating cost over the design life of a typical residential heating system. This savings is often overlooked or not presented with sufficient emphasis by those who only consider the energy use associated with *producing* heat.

Figure 2-4

VERSATILITY:

Modern hydronics technology offers virtually unlimited potential to accommodate the comfort needs, usage, aesthetic tastes and budget constraints of almost any building. A single system can be designed to supply space heating, domestic hot water and specialty loads such as snowmelting or pool heating. These "multi-load" systems reduce installation costs because redundant components are eliminated. They also improve the efficiency of boilers and reduce fuel usage relative to systems where each load is served by its own heat source.

Hydronic systems can combine a variety of heat emitters for space heating. For example, hydronic radiant floor heating may be used to maintain comfort in a basement, while the first and second floors are heated by panel radiators.

A wide variety of hydronic heat sources are available. They include boilers fired by gas, oil, electricity, firewood and pellets; geothermal and air-source heat pumps; solar collectors and devices that adapt to special circumstances or opportunities, such as "time-of-use" electrical rates or waste heat recovery.

Figure 2-5

Figure 2-6

Figure 2-7

Courtesy of Gary Todd

CLEAN OPERATION:

A common complaint about forced-air heating is its propensity to move dust and other airborne particles such as pollen and smoke throughout a building. In buildings where air-filtering equipment is either of poor quality or improperly maintained, dust streaks around ceiling and wall diffusers are often evident, as seen in Figure 2-8.

Figure 2-8

Figure 2-9

Eventually duct systems and other air-handling equipment require internal cleaning to remove dust and dirt that has accumulated over several years of operation, as seen in Figure 2-9.

In contrast, few hydronic systems involve forced-air circulation. Those that do create *room* air circulation rather than building air circulation. This reduces the dispersal of airborne particles and microorganisms. This is a major benefit in situations where air cleanliness is imperative, such as for people with allergies, health care facilities or laboratories.

QUIET OPERATION:

Many people view their home as an escape from the noise and stress associated with modern life. As such, the home should provide "acoustical comfort" as well as thermal comfort. When it's time to relax, noise from any heating or cooling system is certainly unwelcome.

Figure 2-10

A properly designed and installed hydronic system can operate with virtually no detectable noise in the occupied areas of a home. These characteristics make hydronic heating ideal in sound-sensitive areas such as home theaters, reading rooms or sleeping areas.

NONINVASIVE INSTALLATION:

Consider the difficulty encountered when installing adequate and properly sized ducting within a typical wood-framed house. If the house happens to use open web floor trusses rather than traditional floor joists, the ducting may be able to be installed as shown in Figure 2-11.

However, most homes are *not* framed in this manner. Instead, they have solid floor joists. Making the cuts necessary to accommodate properly sized ducting in this type of framing would destroy its structural integrity.

Figure 2-11

Source: Earth Advantage Institute

This situation leads to ducts being installed in unconditioned attics or "hidden" behind interior soffits, as seen in Figure 2-12. Such situations can also result in undersized ducts or inadequate placement of ducts. Occupants eventually pay the price in terms of reduced comfort, noise, increased heat loss or compromised aesthetics.

Figure 2-12

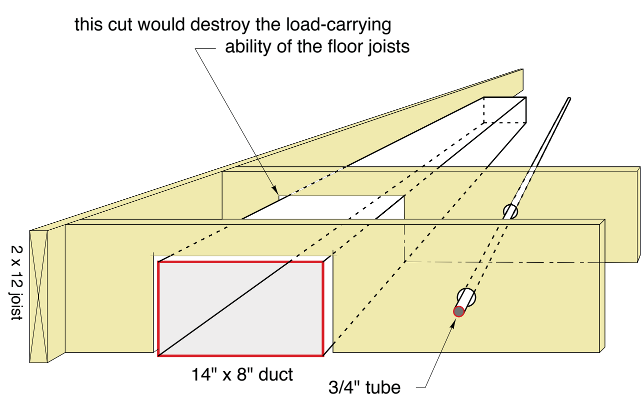
Source: http://angthebuilder.blogspot.com

Source: U.S. Department of Energy

By comparison, modern hydronic systems are easy to integrate into buildings without compromising their structure or the aesthetic character of the space. The reason for this is the high heat capacity of water. A given volume of water can absorb almost 3,500 times more heat as the same volume of air for the same temperature change. This means that the volume of water that must be moved through a building to deliver a given amount of heat is only about 0.03% that of air! This greatly reduces the size of the distribution "conduit."

For example, a 3/4-inch diameter tube carrying water at 6 gallons per minute around a hydronic system operating with a 20ºF temperature drop transports as much heat as a 14" by 8" duct carrying 130ºF air at 1,000 feet per minute. Figure 2-13 depicts these two options, side by side, in true proportion.

Figure 2-13



Notching into 2 x 12 floor joists to accommodate the 14" by 8" duct would destroy their structural integrity. However, small-diameter flexible tubing is easily routed through the center of the same framing with negligible effect on structure.

Figure 2-14

Hydronic systems using small diameter flexible tubing are also much easier than ducting to retrofit into existing buildings. The tubing can be routed through closed framing spaces much like electrical cable, as shown in Figure 2-14.

For buildings where utility space is minimal, small wall-hung boilers can often be mounted in a closet or small alcove, as seen in Figure 2-15. In many cases, these compact boilers can supply the building's domestic hot water as well as its heat. The entire "mechanical room" may have a footprint of less than 10 square feet.

Figure 2-15

ABILITY TO ZONE:

The purpose of any heating system is to provide comfort in all areas of a building throughout the heating season. Doing so requires systems that can adapt to the lifestyle of the building occupants, as well as the constantly changing thermal conditions inside and outside of a building.

Imagine a building in which all rooms have the same floor area. Someone not familiar with heating system design might assume that each room, because they are all the same size, requires the same rate of heat input. Although such an assumption might be intuitive and simplistic, it fails to recognize differences in the thermal characteristics of the rooms. Some rooms might have only one exposed

Figure 2-16

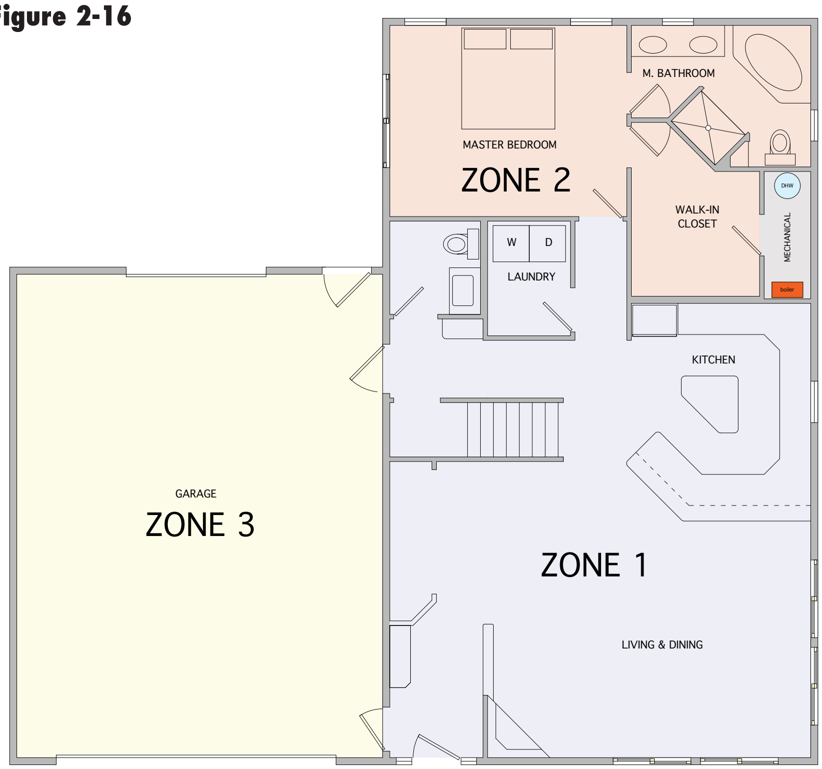
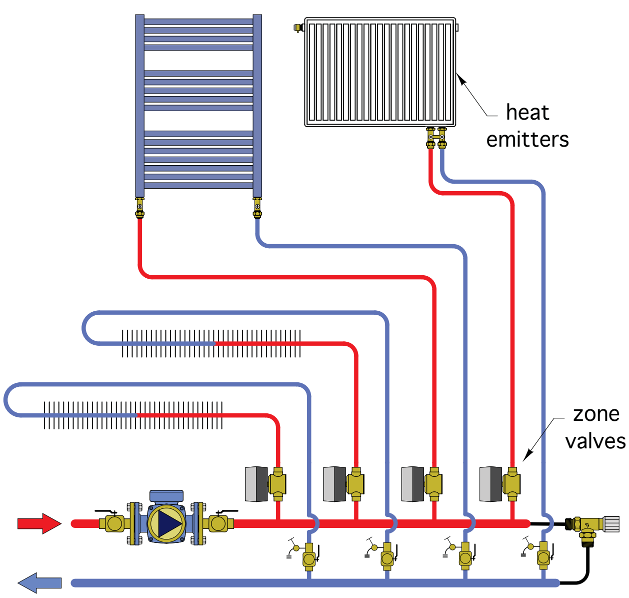


Figure 2-17



wall, while other rooms might have two or three exposed walls. Some rooms might have a generous window area, while other rooms have no windows. The air leakage and potential for sunlight to enter through windows may also vary from one room to another.

Beyond these differences are conditions imposed by the occupants. One person might prefer sleeping in a room maintained at 63ºF, while another feels chilled if their bedroom is anything less than 72ºF. One occupant might prefer a living room maintained at 70ºF while relaxed and reading, while another wants the temperature in the exercise room at 65ºF during a workout.

The combination of room thermal characteristics, outdoor conditions and occupant expectations presents a complex and dynamic challenge for the building's heating system as it attempts to maintain comfort.

One method that has long been used to help meet this challenge is dividing the building's heating system into zones. *A zone is any area of a building for which indoor air temperature is controlled by a single thermostat (or other temperature-sensing device).* A zone can be as small as a single room, or it may be as large as an entire building.

Figure 2-16 shows a zoning plan for the first floor of a house. The living room, kitchen, half-bath and laundry are combined into a single zone. The master bedroom, master bathroom and closet form another zone. The garage is also treated as a separate zone.

There are several ways to create zoned hydronic heating and cooling systems. All of these approaches are simpler and less expensive than equivalent zoning of a forced-air system. Figure 2-17 shows part of a hydronic piping schematic in which zone valves are used to control the flow of heated water to each heat emitter.

Proper zoning accounts for differences in the activities that occur in different areas of a building, as well as differences in preferred comfort levels, interior heat gain and the desire to reduce energy use through reduced temperature settings (e.g., "setback").

***idronics* #5 provides a more in-depth discussion of modern zoning methods for hydronic systems.**

3: A BRIEF HISTORY OF HYDRONICS

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

Although steam systems were effective, and some are still in use today, they impose several fundamental limitations on how the system must be designed and installed.

First, in a low-pressure steam heating system, the boiler needs to be located at the bottom of the distribution system. This is necessary to allow the condensate (steam that has cooled and changed back to liquid water) to flow back to the boiler.

Second, steam boilers generally need to operate at relatively high temperatures. There are exceptions to this (e.g., vacuum-based steam systems), but they are not common. The high temperatures needed to produce steam essentially eliminate contemporary heat source options such as solar thermal collectors or heat pumps.

Third, steam systems require carefully planned metal piping systems. The piping must be pitched for condensate return to the boiler. Fittings on steam mains must be properly oriented. The temperatures at which most steam heating systems operate preclude the use of polymer piping such as PEX or polypropylene. Pipe sizes also tend to be larger than those required for water-based distribution systems. The installation cost of the iron or steel piping required in steam systems has significantly increased over the last decade.

Most steam heating systems use cast iron radiators as their heat emitters. An example of such a radiator is shown in Figure 3-1. While such radiators can be very effective, and are often elegant in detail, they are very heavy. This significantly increases the cost of transportation and installation. The manufacturing procedures used for cast iron have also become more difficult to sustain, especially in North America, due to increasing energy cost and environmental regulations.

Finally, the iron and steel piping used in steam heating systems at various times contains steam, liquid water and air. The combination of water, oxygen from air and iron creates iron oxides within the piping and radiators. These oxides eventually form a sludge that reduces heat transfer in the radiators and must be periodically "blown down" from steam producing boilers.

Figure 3-1

Steam heating deserves distinct recognition within the evolution of North American heating systems. Thousands of innovative products have been produced to improve the performance of steam heating systems over several decades. Still, given the resources and installation details they require, the current worldwide costs of energy and the availability of many competing methods of heat generation and delivery, very few new steam heating systems are being installed. The present market for steam heating is almost entirely based on replacement of older steam boilers, radiators and related hardware. *Water, in liquid form, is the future of hydronic heating.*

EARLY WATER-BASED SYSTEMS:

Water-based central heating systems were originally developed in the late 1800s. During that time, and for several decades to follow, there were no electrically powered circulators available. The only "propulsion effect" available to move water through systems was the differential pressure created by the simultaneous presence of hot and cool water. Hot water in the boiler is slightly lighter than cooler water in the return side piping. This creates a slight weight imbalance that causes hot water to rise while cool water descends, and thus circulation occurs within the system. Early designers quantified this effect, as shown in Figure 3-2, which dates from an 1896 heating manual.

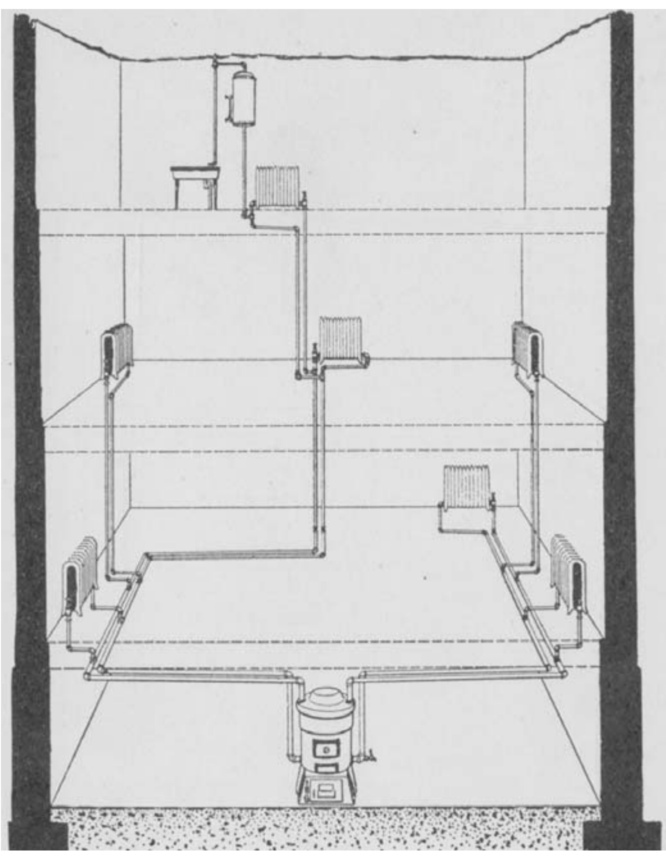
Figure 3-2

DIFFERENCE IN WEIGHT OF TWO COLUMNS OF WATER, EACH ONE FOOT HIGH, AT VARIOUS TEMPERATURES. (Hood.)

Difference in Temperature of the two Columns of Water, in degrees of Fahrenheit's scale.	Difference in Weight of two Columns of Water contained in different sized Pipes each ONE FOOT in height.				
	1 in. diam.	2 in. diam.	3 in. diam.	4 in. diam.	Per square inch.
°	grains.	grains.	grains.	grains.	grains.
2	1·5	6·3	14·3	25·4	2·028
4	3·1	12·7	28·8	51·1	4·068
6	4·7	19·1	43·3	76·7	6·108
8	6·4	25·6	57·9	102·5	8·160
10	8·0	32·0	72·3	128·1	10·200
12	9·6	38·5	87·0	154·1	12·264
14	11·2	45·0	101·7	180·0	14·328
16	12·8	51·4	116·3	205·9	16·392
18	14·4	57·9	131·0	231·9	18·456
20	16·1	64·5	145·7	258·0	20·532

These early hot water systems were often referred to as "gravity" hot water systems. An example of the piping used in these early hydronic systems is shown in Figure 3-3.

Figure 3-3



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

Many of these early systems contained an expansion tank at the high point in the system, often in the attic of a house. This tank was usually sized to 1/36th of the total system volume, and equipped with an overflow pipe leading to a drain or outside the building.

When heated, the water in the system expanded, possibly to the point where some would spill out of the overflow pipe from the expansion tank. As the system cooled, the water would decrease in volume, and the water level within the expansion tank would drop.

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

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

Many of these limitations were eliminated when electrically powered circulators became available in North America starting in the early 1930s. One example of an early generation (1931) "booster" circulator is shown in Figure 3-4.

Figure 3-4

Courtesy of Bell & Gossett

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

Figure 3-5 shows a 1949 drawing of a two-zone hydronic heating system using circulators. This system used a special fitting called a “Monoflo® tee” to induce flow from the distribution mains through each cast iron radiator. These fittings are still available, and their application is discussed in section 9.

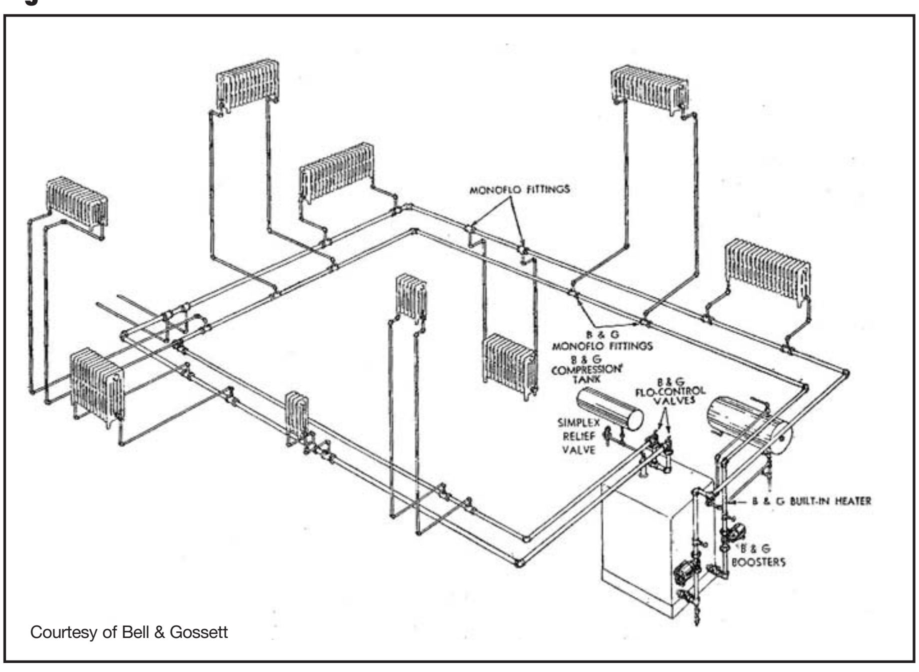
Systems of this vintage were entirely constructed of steel or iron piping, typically with threaded cast iron fittings. As time progressed, copper tubing became an increasingly popular alternative, especially in residential systems.

In 1958, Taco introduced the first “wet rotor” circulator to the North American hydronics market (shown in Figure 3-6).

Wet rotor circulators slowly gained acceptance as an alternative to traditional 3-piece circulators. By 1980, they had become the predominant type of circulator used in new residential and light commercial hydronic systems.

The availability of circulators also allowed for distribution systems and heat emitters very different from those required by “gravity” flow systems. One example is the hydronic radiant floor panel system shown in Figure 3-7, during its installation in the 1940s.

Figure 3-5



Courtesy of Bell & Gossett

Figure 3-6

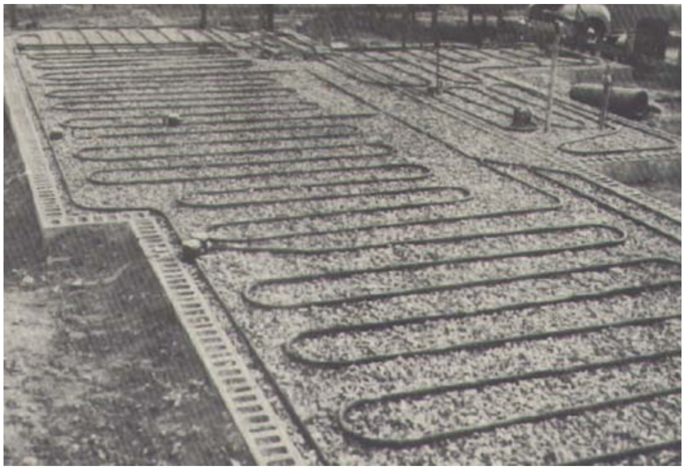
Courtesy of Taco

The first generation hydronic radiant panel systems used “grids” and “coils” of steel or wrought iron piping embedded in concrete slabs. After World War II, soft-temper copper tubing became the preferred material for such systems due to its availability in smaller sizes and easier bending characteristics. During the later 1940s and early 1950s, radiant floor and ceiling heating systems using copper tubing became increasingly popular.

Although these radiant panels delivered exceptional comfort for their time, and some are still in use today, others failed prematurely due to chemical reactions between copper tubing and certain materials used in concrete. Repetitive mechanical stresses due to the heating and cooling also caused some failures. Such issues eventually tarnished the reputation of hydronic radiant panel heating. By the 1970s, new installations of copper-based radiant panel heating systems were almost nonexistent.

Most residential hydronic systems of the 1960s through 1970s shifted to cast iron baseboard or copper fin-tube baseboard. Pressure-rated boilers with heat exchangers made of cast iron sections or steel fire tubes and fueled by natural gas and fuel oil became common. Hydronic systems of this era were relatively simple, typically having 1 to 3 zones and basic electromechanical controls.

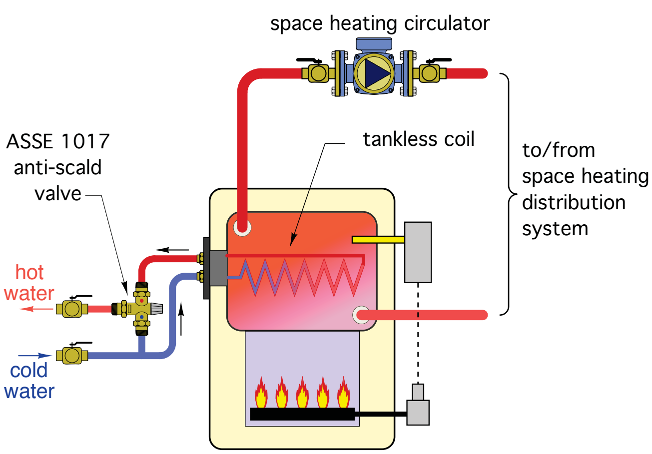
Figure 3-7



They were marketed as the modern, safe and fully automatic approach to centralized heating.

Some of the boilers used during this time were equipped with "tankless coil" water heaters, as shown in Figure 3-8.

Figure 3-8



Source: http://epb.apogee.net

These devices allowed the boiler to provide domestic hot water as well as space heating. They also required the boiler to remain at an elevated temperature of at least 140ºF for the entire year. Although acceptable at the time, this method of domestic water heating is now viewed as very inefficient.

In the early 1980s, crosslinked polyethylene tubing (e.g., PEX) made its way to North America after several years of successful use in Europe. PEX, and other polymer-based tubing, revolutionized the installation of hydronic radiant panel heating, providing fast installation and a long, reliable life. The availability of PEX tubing was the spark that rekindled interest in hydronic radiant panel heating in North America during the 1980s. Since then, thousands of new products have been introduced into the North American hydronics market. These, in combination with contemporary design methods, ushered in the era of modern hydronics in North America.

MODERN HYDRONICS:

Today, much of the technology available for hydronic heating is a quantum leap above that available in the mid 1900s. Examples include the following:

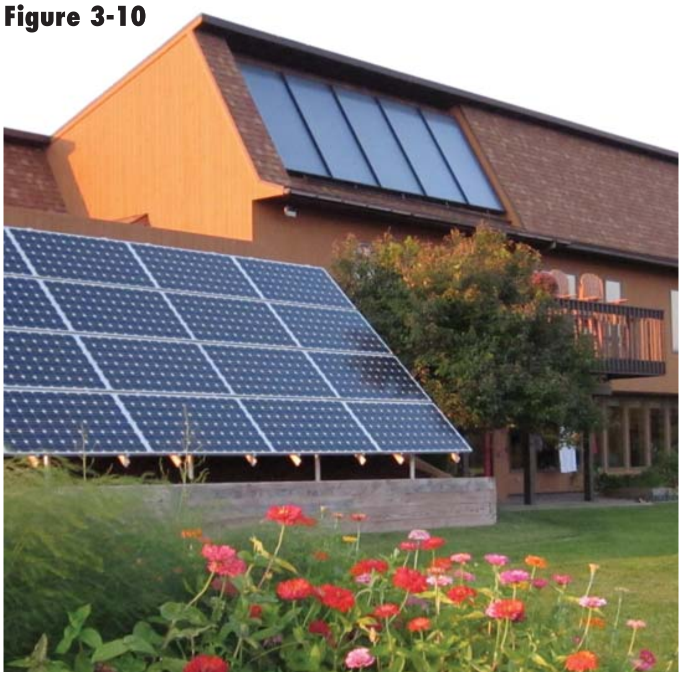
• Compact wall-hung condensing boilers, such as the one shown in Figure 3-9, when properly applied, can operate with thermal efficiencies in range of 95%.

Figure 3-9

Courtesy of Triangle Tube

• Renewable energy heat sources, such as solar thermal collectors, heat pumps and biomass boilers, can be integrated into hydronic systems.

Figure 3-10



• Multiple water temperatures within the system can be precisely controlled using both electronic and thermostatic devices.

Figure 3-11

Source: bonjourlife.com

• Durable polymer tubing can be quickly and easily installed within building structures to create radiant floor, wall and ceiling panels. With proper design and controls, these panels can be used for both heating and cooling.

Figure 3-12

• A wide assortment of heat emitters is available to match heating loads and aesthetic preferences.

Figure 3-13

• Systems can heat buildings as well as supply domestic hot water, melt snow on outdoor surfaces and warm swimming pools.

• Heat sources, mixing devices and circulators can be intelligently controlled using microprocessors.

Figure 3-14

• Systems can be designed that use a fraction of the fuel and electrical energy required by earlier systems, yet provide superior comfort.

Figure 3-15

• System operation can be monitored and altered if necessary using a laptop, tablet or smart phone from any location with Internet access.

Figure 3-16

Courtesy of ecobee"

The remainder of this issue will introduce many state-of-the-art components and design concepts. It will also show you how to combine them into contemporary hydronic systems that deliver superior comfort, high efficiency and reliability.

4: THE BASIC HYDRONIC CIRCUIT

Nearly all hydronic systems have several common components. This section introduces those components and briefly describes their function. Later sections describe this hardware in more detail and show how to size and assemble it into complete systems.

The vast majority of modern hydronic heating systems are closed-loop systems. This means that the piping components form an assembly that separates the fluid within the system from contact with the atmosphere. If the system's fluid is exposed to the atmosphere at any point, that system is classified as an open-loop system.

Closed-loop circuits have several advantages over "open-loop" systems. First, because they are well-sealed from the atmosphere, there is very little fluid loss over time. Any slight fluid loss that occurs is usually attributable to weepage at valve packings or gaskets. Such minor losses do not create problems and can be automatically corrected with components discussed later in this section.

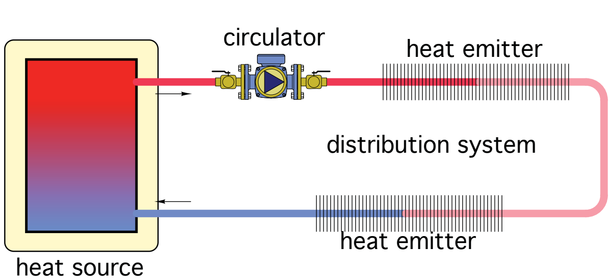
Second, properly designed closed-loop systems allow very little oxygen to enter the system. This greatly reduces the potential of corrosion, especially in systems containing iron or steel components.

Third, closed-loop systems operate under slight pressure. This helps in eliminating air from the system. It also helps in suppressing boiling or circulator noise within the system.

THE BASIC CLOSED CIRCUIT:

Figure 4-1 shows a basic closed-loop hydronic circuit. The box representing the heat source could be a boiler, a heat pump, a thermal storage tank heated by solar collectors or some other heat-producing device. Whatever the case, it is a *closed* device, sealed from the atmosphere and capable of operating under some pressure.There are hydronic heat sources that are not closed. An example would be some outdoor wood-burning furnaces, which are not designed to operate under pressure.

Figure 4-1



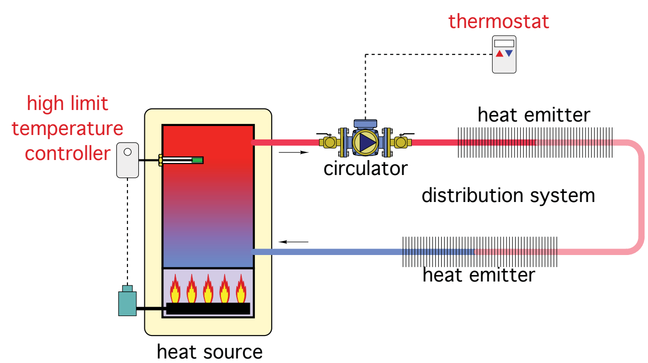
The fluid within a hydronic circuit serves as a *conveyor belt for heat*. That heat is transferred into the fluid at the heat source, carried through the distribution system by the fluid and dissipated from the fluid at one or more heat emitters. The fluid then returns to the heat source to absorb more heat and repeat the process. A circulator, when operating, maintains flow through the system. The same fluid remains in the closed-loop system, often for many years. It never loses its ability to absorb, transport or dissipate heat.

BASIC CONTROLS:

In an ideal system, the rate at which the heat source produces heat would always match the rate at which the building, or domestic water, needs heating. This is seldom the case with any real system, and thus it is necessary to provide controllers that manage both heat production and heat delivery.

One of the most basic controllers is a thermostat that measures room air temperature and turns the circulator on and off based on how the room's temperature compares to the thermostat's setpoint. The thermostat's "goal" is to keep the room air temperature at, or very close to, its setpoint temperature. If the room's air temperature begins rising above the setpoint temperature, the thermostat turns off the circulator to stop further heat transport to the heat emitters. When the room temperature drops below the thermostat's setpoint, it turns on the circulator to resume heat transport.

Figure 4-2



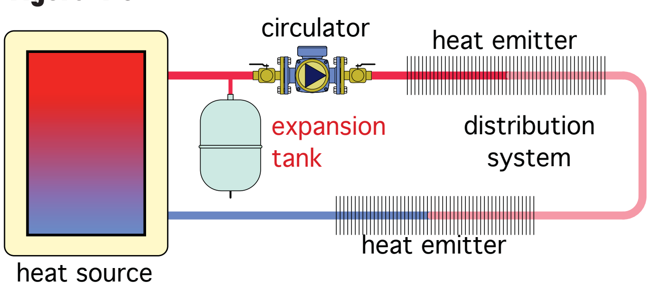
Another common controller is called a high-limit controller. Its function is to turn the heat source on and off so that the temperature of the fluid supplied to the distribution system remains within a narrow and usable range.

Figure 4-2 shows a basic hydronic system supplied by a fuel-burning heat source. The circulator is assumed to be controlled by a room thermostat. The water temperature supplied by the heat source is limited by the high-limit controller.

EXPANSION TANK:

Water expands when it is heated. This is a very powerful but predictable characteristic that must be accommodated in any type of closed-loop hydronic system. Figure 4-3 shows a diaphragm-type expansion tank added to the system. This tank contains a captive volume of air. As the heated water within the system expands, it pushes into the tank and slightly compresses the captive air volume. This causes the system pressure to rise a small amount. As the system's water cools, its volume decreases, allowing the compressed air to expand, and the system pressure returns to its original value. This process repeats itself each time the system heats up and cools off. Expansion tank selection and sizing are covered in section 6.

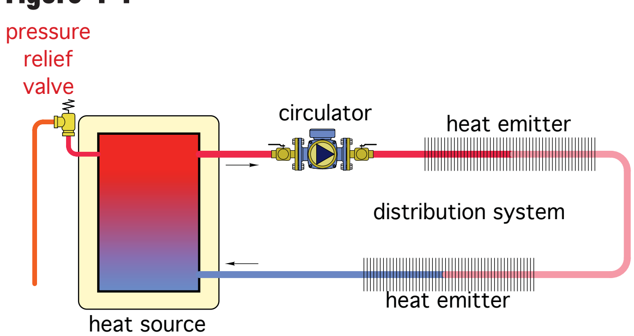
Figure 4-3



PRESSURE-RELIEF VALVE:

Consider what could happen to a closed-loop hydronic system in which a defective high-limit controller fails to turn off the heat source after its upper temperature limit has been reached. As the water gets hotter, system pressure steadily increases due to the water's expansion. This pressure could eventually exceed the pressure rating of the weakest component in the system. The consequences of a system component bursting at high pressures and temperatures could be devastating. Thus, all closed-loop hydronic systems must be protected by a pressure-relief valve. This is a universal requirement of all mechanical codes in North America. Figure 4-4 shows a pressure relief valve installed on the heat source.

Figure 4-4

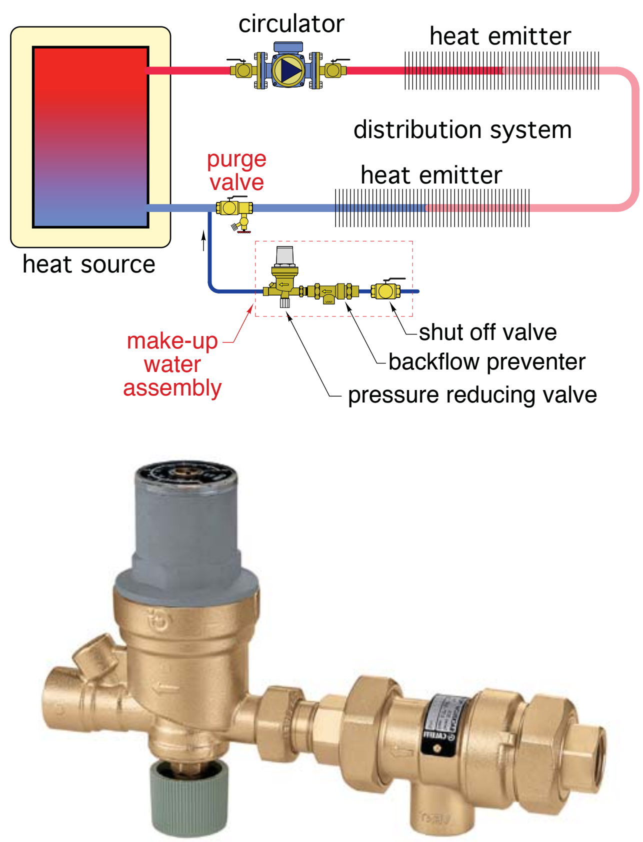


MAKE-UP WATER SYSTEM:

Most closed-loop hydronic systems experience very minor water losses over time due to weepage from valve packings, pump seals, air vents and other components. These losses are normal and must be replaced to maintain adequate system pressure.

A common method for replacing this water is through an automatic make-up water assembly consisting of a pressure-reducing valve, backflow preventer and shutoff valve (shown in Figure 4-5).

Figure 4-5



Because the pressure in a building water system is usually higher than the pressure-relief valve setting in a hydronic system, the latter cannot be *directly* piped to the building water system. However, a pressure-reducing valve can be placed between the building water system and hydronic system to limit pressure in the latter. This valve is used to reduce and maintain a constant minimum pressure within the hydronic system. It lets water enter the hydronic system whenever the pressure in the system drops below the valve's pressure setting. Most pressure-reducing valves have an adjustable pressure setting.

The backflow preventer stops any water that has entered the system from returning and possibly contaminating the potable water supply system. Most municipal codes require such a device on any heating system connected to a public water supply.

The shutoff valve in the make-up water assembly is installed to allow the system to be isolated from its water source.

A purging valve is also shown in the schematic of Figure 4-5. It allows most of the air within the circuit to exit as the system is filled with water. Purging valves consist of a ball valve that is inline with the distribution piping and a side-mounted drain port. To purge the system, the inline ball valve is closed and the drain port is opened. As water enters the system through the make-up water assembly, air is expelled through the drain port of the purge valve.

idronics #2 describes proper procedures for purging hydronic systems.

AIR SEPARATOR:

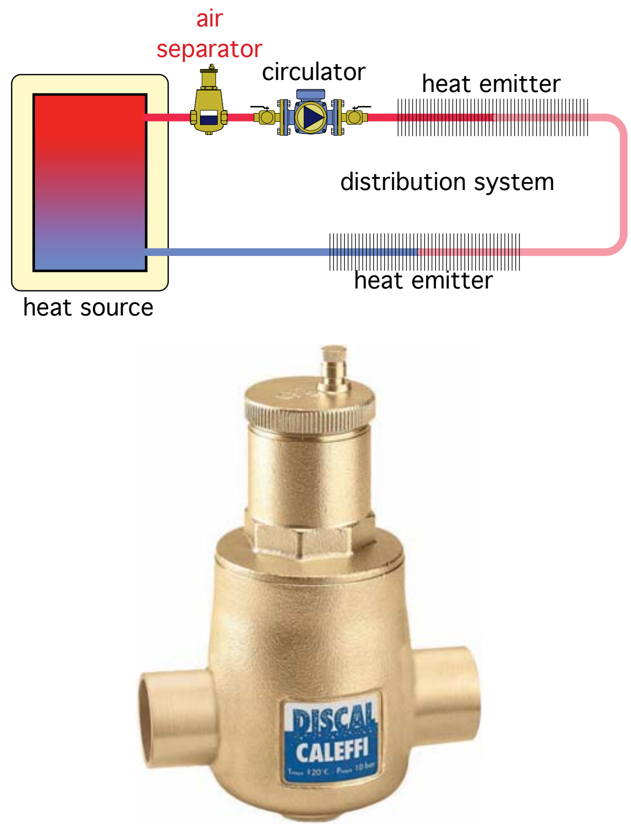
An air separator is designed to separate air from water and remove that air from the system. Modern air separators create regions of reduced pressure as water passes through. The lowered pressure causes dissolved gases in the water to form into bubbles. These bubbles are guided upward into a collection chamber where an automatic air vent expels them from the system. For best results, the air separator should be located where fluid temperatures are highest. This is typically in the supply pipe from the heat source, as shown in Figure 4-6.

idronics #2 describes how high performance air separators operate and how to apply them.

PUTTING IT ALL TOGETHER

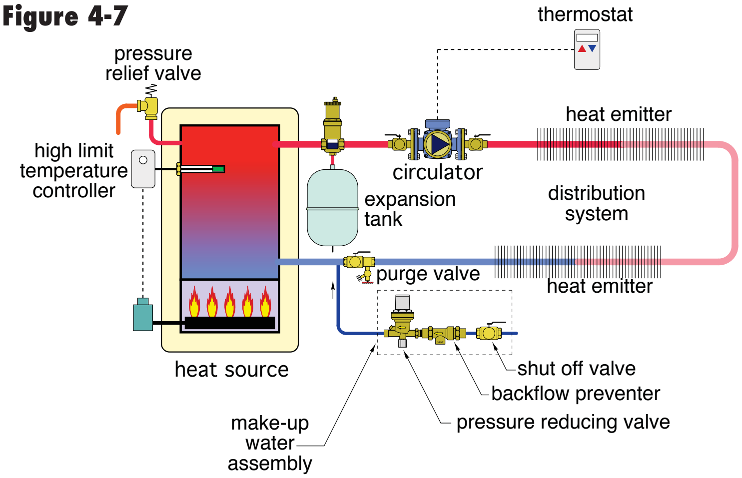
Figure 4-7 shows all the components previously discussed in their proper positions relative

Figure 4-6



to each other. By assembling these components, we have built a simple hydronic heating system. However, just because all the components are present doesn't guarantee that the system will function properly. The components must be selected and sized properly so that they work together as a system. Later sections will discuss how to do this.

Figure 4-7



5: PHYSICAL OPERATING CONDITIONS IN HYDRONIC SYSTEMS

Water is the essential ingredient in any hydronic system. Its physical properties make it possible to deliver heating or cooling throughout a building using relatively small "conduits" and minimal amounts of electrical energy.

Figure 5-1

The performance of all hydronic systems is intimately tied to the physical properties of water—in particular, the process by which water absorbs heat, conveys it and releases it. This section discusses the fundamental relationships that govern those processes. It discusses properties that affect both the thermal and hydraulic performance of all hydronic systems. It clarifies the differences as well as the interrelationships between physical quantities such as temperature, heat content, rate of heat transfer, flow rate, flow velocity, head energy and pressure drop.

Understanding these physical properties and how they are related is essential for anyone who wants to design efficient and reliable hydronic systems. Such an understanding is also important for those who diagnose and adjust hydronic systems for optimal performance.

SENSIBLE HEAT & LATENT HEAT:

The heat absorbed or released by a material while it remains in a *single phase* is called sensible heat. The word sensible means that the presence of the heat can be *sensed by a temperature change in the material*. When a material absorbs sensible heat, its temperature must increase. Likewise, when a material releases sensible heat, its temperature must decrease. Under normal operating conditions, all the hydronic systems discussed in this and other issues of *idronics* operate with water that remains in liquid form. Thus, all heat and heat transfer quantities discussed are sensible heat.

If a material *changes phase* while absorbing heat, its temperature does not change. The heat absorbed in such a process is called latent heat. For water to change from a liquid to a vapor, it must absorb approximately 970 Btu/lb. This is a very large amount of heat to be conveyed by a single pound of material. It is also the physical characteristic that is leveraged by a steam heating system.

SPECIFIC HEAT & HEAT CAPACITY:

One of the most important properties of water relevant to its use in hydronic systems is specific heat. The specific heat of any material is the number of Btus required to raise the temperature of one pound of that material by one degree Fahrenheit (ºF). The specific heat of water is 1.0 Btu per pound per degree F. To raise the temperature of 1 pound (lb) of water by 1 degree Fahrenheit (ºF) will require the addition of 1 Btu of heat.

This relationship between temperature, material weight and energy content also holds true when a substance is cooled. For example, to lower the temperature of 1 pound of water by 1 degree F will require the removal of 1 Btu of energy.

The specific heat of any material varies slightly with its temperature. However, for water, this variation is very small over the temperature range used in residential and light commercial hydronic systems. Thus, for the calculations involved in designing these systems, the specific heat of water can be considered to remain constant at 1.0 Btu/lb/ºF.

The specific heat of water is high in comparison to other common materials. In fact, water has one of the highest specific heats of any known material. Only ammonia has a higher specific heat (which ranges from 1.12 to 1.6 Btu/lb/ºF, depending on temperature).

Another index used to quantify a material's ability to contain sensible heat is called heat capacity. It is the number of Btus required to raise the temperature of one cubic foot of the material by one degree Fahrenheit. The heat capacity of a material can be found by multiplying the material's specific heat by its density.

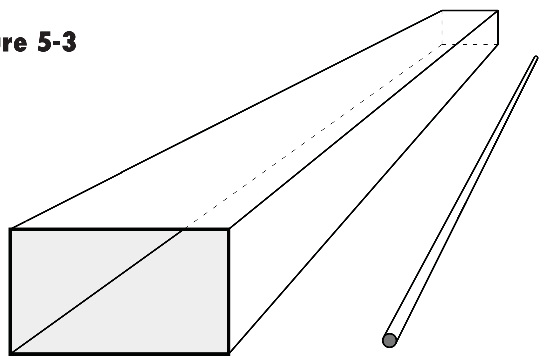
Figure 5-2 lists the specific heat, density and heat capacity of some common materials.

Figure 5-2

Material	Specific heat (Btu/lb/ºF)	Density (lb/ft^3)	Heat capacity (Btu/ft^3/ºF)
Water	1.00	62.4	62.4
Concrete	0.21	140	29.4
Steel	0.12	489	58.7
Wood (fir)	0.65	27	17.6
Air	0.24	0.074	0.018
Sand	0.1	94.6	9.5

By comparing the heat capacity of water and air, it can be shown that *a given volume of water can hold almost 3,500 times as much heat as the same volume of air* for the same temperature rise. This allows a given volume of water to transport vastly more heat than the same volume of air. This is the underlying (and very significant) advantage of using water rather than air for moving sensible heat. Figure 5-3 illustrates this advantage by comparing two heat transport systems of equal heat-carrying ability. The 1" diameter tube carrying water can convey the same amount of heat as a 10" by 20" duct conveying air when both systems are operating at typical temperature differentials and flow rates.

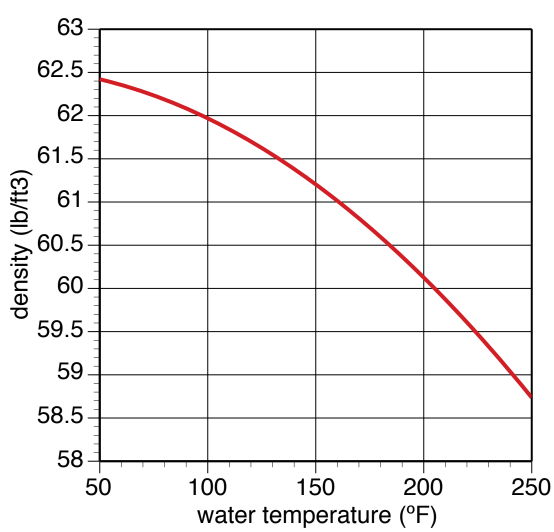
Figure 5-3



DENSITY:

In this and other issues of *idronics*, the word "density" refers to the number of pounds of the material needed to fill a volume of one cubic foot. For example, it would take 62.4 pounds of water at 50ºF to fill a 1 cubic foot container, so its density is said to be 62.4 pounds per cubic foot, (abbreviated as 62.4 lb/ft^3).

Figure 5-4



The density of water varies with its temperature. Like most substances, as the temperature of water increases, its density decreases. As temperature increases, each molecule of water requires more space. This molecular expansion is extremely powerful and can easily burst metal pipes or tanks if not properly accommodated. Figure 5-4 shows the relationship between temperature and the density of water.

The change in density of water or water-based antifreeze solutions directly affects the size of the expansion tank required on all closed-loop hydronic systems. Proper tank sizing requires data on the density of the system fluid both when the system is filled and when it reaches its maximum operating temperature.

The change in water's density based on its temperature affects several other aspects of hydronic system design and operation. For example, because hot water is less dense than cool water, it will rise to the top of a thermal storage tank. The difference in density between warm and cool water in different parts of the system was also responsible for creating circulation in early hydronic systems, as discussed in section 3.

SENSIBLE HEAT *QUANTITY* FORMULA:

It is often necessary to determine the quantity of heat stored in a given volume of water that undergoes a temperature change. Formula 5-1, known as the sensible heat quantity formula, can be used for this purpose:

Formula 5-1

$$h = 8.33v(\Delta T)$$

where:

h = quantity of heat absorbed or released from the water (Btu)
v = volume of water (gallons)
ΔT = temperature change of the material (ºF)

Example: Assume a storage tank contains 500 gallons of water initially at 70ºF. How much heat must be added to raise this water to a temperature of 180ºF?

Solution: The amount of heat involved is easily determined using Formula 5-1:

$$h = 8.33v(\Delta T) = 8.33 \times 500 \times (180 - 70) = 458{,}000\,Btu$$

SENSIBLE HEAT *RATE* FORMULA:

Hydronic system designers often need to know the rate of heat transfer to or from a fluid flowing through a device such as a heat source or heat emitter. This can be done using the sensible heat rate formula, given as Formula 5-2:

Formula 5-2

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

where:

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

When using Formula 5.2, the density and specific heat should be based on the average temperature of the fluid during the process by which the fluid is gaining or losing heat.

For *cold water only*, Formula 5-2 simplifies to Formula 5-3:

Formula 5-3

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

where:

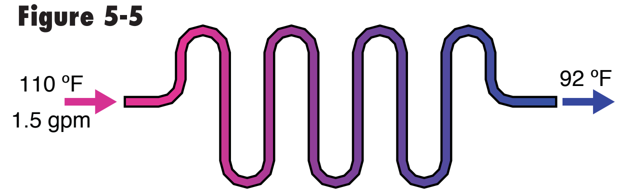
Q = rate of heat transfer into or out of the water stream (Btu/hr)
f = flow rate of water through the device (gpm)
500 = constant rounded off from 8.33 x 60
ΔT = temperature change of the water through the device (ºF)

Formula 5-3 is technically only valid for cold water because the factor 500 is based on the density of water at approximately 60ºF. However, because the factor 500 is easy to remember, Formula 5-3 is often used for quick mental calculations for the rate of sensible heat transfer involving water. While this is fine for initial estimates, it is better to use Formula 5-2 for final calculations because it accounts for variations in both the density and specific heat of the fluid. Formula 5-2 can also be used for fluids other than water.

Example: Water flows into a radiant panel circuit at 110ºF and leaves at 92ºF. The flow rate is 1.5 gpm, as shown in Figure 5-5. Calculate the rate of heat transfer from the water to the heat emitter using:

a. Formula 5-2
b. Formula 5-3

Figure 5-5



Solution: Using Formula 5-2, the rate of heat transfer from the circuit is:

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

To use Formula 5-3, the density of water at its average temperature of 101ºF must first be estimated using Figure 5-4:

D = 61.96 lb/ft^3

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

Putting these numbers into Formula 5-2 yields:

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

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

Formula 5-2 is generally accepted in the hydronics industry for quick estimates of heat transfer to or from a stream of water. However, Formula 5-3 yields slightly more accurate results when the variation in density and specific heat of the fluid can be factored into the calculation.

Formulas 5-2 and 5-3 can also be rearranged to determine the temperature drop or flow rate required for a specific rate of heat transfer.

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

Solution: Just rearrange Formula 5-2 and put in the numbers.

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

Example: What is the flow rate required to deliver 100,000 Btu/hr in a distribution system that operates with water and a temperature drop of 30ºF?

Solution: Again, Formula 5-2 can be rearranged to determine flow and the known values factored in.

$$f = \frac{q}{500(\Delta T)} = \frac{100{,}000}{500(30)} = 6.7\,gpm$$

Formula 5-2 and the slightly more accurate Formula 5-3 are perhaps the most important formulas in hydronic system design. They are constantly used to relate the rate of heat transfer to fluid flow rate and the temperature change of the fluid.

VAPOR PRESSURE AND BOILING POINT:

The vapor pressure of a liquid is the minimum pressure that must be applied at the liquid's surface to prevent it from boiling. If the pressure on the liquid's surface drops below its vapor pressure, the liquid will instantly boil. If the pressure is maintained above the vapor pressure, the liquid will remain a liquid and no boiling will occur.

The vapor pressure of a liquid is depends on its temperature. The higher the liquid's temperature, the higher its vapor pressure. Stated another way, the greater the temperature of a liquid, the more pressure must be exerted on its surface to prevent it from boiling.

Vapor pressure is stated as an *absolute* pressure and has the units of psia. On the absolute pressure scale, zero pressure represents a complete vacuum. On earth, the weight of the atmosphere exerts a pressure on any liquid surface open to the air. At sea level, the absolute pressure exerted by the atmosphere is approximately 14.7 psia.

Figure 5-6

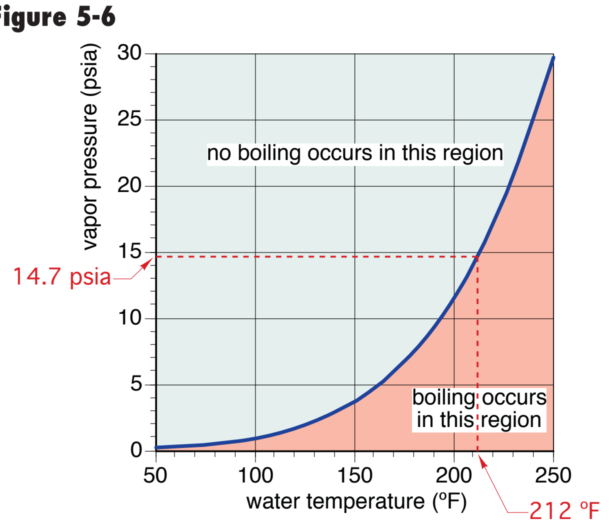


Figure 5-6 shows this relationship between the vapor pressure and temperature of water between 50ºF and 250ºF.

The vapor pressure of water at 212ºF is 14.7 psia. Because 14.7 psia is the absolute pressure the atmosphere exerts at sea level, and water at sea level boils at 212ºF.

If a pot of water is carried to an elevation of 5,000 feet above sea level and then heated, its vapor pressure will rise until it equals the reduced atmospheric pressure at this elevation, and boiling will begin at about 202ºF. Conversely, if a container of water is sealed from the atmosphere and pressurized to about 15.3 psi above atmospheric pressure (e.g., to an absolute pressure of 30 psia), Figure 5-6 indicates that the water must be heated to about 250ºF before it will boil.

Closed-loop hydronic systems should be designed to ensure that the absolute pressure of the water at all points in the system remains safely above the water's vapor pressure under all operating condition. This prevents problems such as cavitation in circulators and valves or "steam flash" in piping.

VISCOSITY:

The viscosity of a fluid is an indicator of its resistance to flow. The higher the viscosity of a fluid, the more drag it creates as it flows through pipes, fittings, valves or any other component in a hydronic circuit. Higher viscosity fluids also require more pumping power to maintain a given flow rate compared to fluids with lower viscosities.

The viscosity of water depends on its temperature. The viscosity of water and water-based antifreeze solutions decreases as their temperature increases.

Solutions of water and glycol-based antifreeze have significantly higher viscosities than water alone. The greater the concentration of antifreeze, the higher the viscosity of the solution. This increase in viscosity can result in significantly lower flow rates within a hydronic system if it is not accounted for during design.

DISSOLVED AIR IN WATER:

Water can absorb air much like a sponge can absorb a liquid. Molecules of the gases that make up air, including oxygen and nitrogen, can exist "in solution" with water molecules. Even when water appears perfectly clear, it often contains some dissolved air.

When hydronic systems are first filled, dissolved air is present in the water throughout the system. If allowed to remain in the system, this air can create numerous problems, including noise, cavitation, corrosion, inadequate flow or even complete flow blockage.

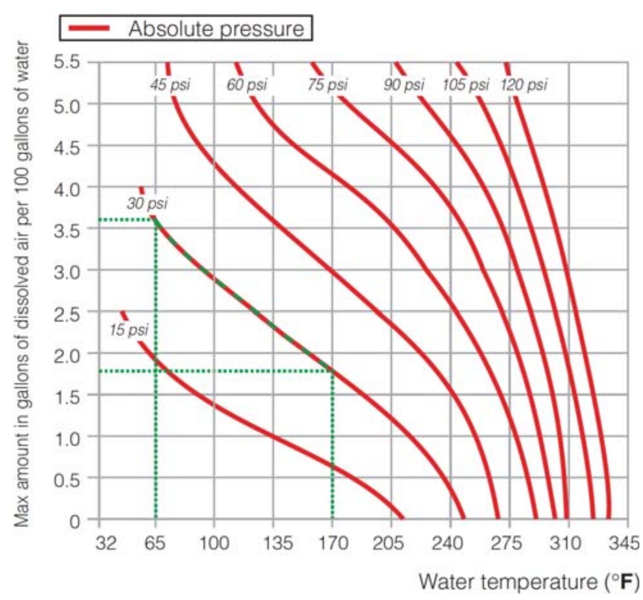
A well-designed hydronic system must be able to automatically capture dissolved air and expel it. A proper understanding of what affects the air content of water is crucial in designing a method to get rid of it.

The amount of air that can exist in solution with water is strongly dependent on the water's temperature. As water is heated, its ability to hold air in solution rapidly decreases. However, the opposite is also true. As water cools, it has a propensity to absorb air. Heating water to release dissolved air is like squeezing a sponge to expel a liquid. Allowing the water to cool is like letting the sponge expand to soak up more liquid. Together these properties can be exploited to help capture and eliminate air from the system.

The water's pressure also affects its ability to hold dissolved air. When the pressure of the water is lowered, its ability to contain dissolved air decreases, and vice versa.

Figure 5-7 shows the *maximum* amount of air that can be contained (dissolved) in water as a percentage of the water's volume. The effects of both temperature and pressure on dissolved air content can be determined from this graph.

Figure 5-7



To understand this graph, first consider a single curve such as the one labeled 30 (psi). This curve represents a constant pressure condition for the water. Notice that as the water temperature increases, the curve descends. This indicates a decrease in the ability of water to contain dissolved air as its temperature increases. This trend also holds true for all other constant pressure curves.

The vertical axis of the graph indicates the *maximum* possible amount of dissolved air in water. This condition represents water that is "saturated" with dissolved air, and thus unable to absorb any more air.

The water in a properly deaereated hydronic system will eventually hold much less dissolved air than the numbers on the vertical axis indicate. Under this condition, the water would be described as "unsaturated." This means it can absorb air it comes in contact with within the system. High-performance air separators, such as a Caleffi Discal, maintain system water in this unsaturated state, and thus enable it to absorb air pockets that might form within the system due to component maintenance or other reasons.

The change in the ability of water to retain air can be read by subtracting values on the vertical axis. For example, at an absolute pressure of 30 psia and temperature of 65ºF, the maximum amount of dissolved air is 3.6 gallons of air per 100 gallons of water. If that water is then heated to 170ºF and its pressure is maintained at 30 psia, its ability to hold air in solution has decreased to about 1.8 gallons of air per 100 gallons of water. Thus, the amount of air released from the water during this temperature increase is estimated at 3.6-1.8 = 1.8 gallons per 100 gallons of water.

The release of dissolved air can be observed in a kettle of water that is being heated on a stove. As the water heats up, bubbles will form at the bottom of the kettle (e.g., where the water is hottest). These bubbles are formed as gas molecules that were dissolved in the cool water are forced out of solution by increasing temperature. The molecules join together to create bubbles. This process is called coalescing. If the kettle of water is allowed to cool, these bubbles will disappear as the gas molecules go back into solution.

One can think of the dissolved air initially contained in a hydronic system as being "cooked" out of the solution as water temperature increases. This usually takes place inside the heat source when it is first operated after being filled with "fresh" water containing dissolved gases. Well-designed hydronic systems take advantage of this situation by capturing air bubbles with an air separator that is typically placed near the outlet of the heat source (e.g., where the water is hottest).

idronics #2 contains a detailed discussion of air removal from hydronic systems.

INCOMPRESSIBILITY:

When liquid water is put under pressure there is very little change in its volume. This change is so small that it can be ignored in the context of designing hydronic systems. Thus water, as well as most other liquids, can be treated as if they are *incompressible*.

In practical terms, this means that liquids cannot be squeezed together or "stretched." It also implies that if a liquid's flow rate is known at any one point in a closed series piping circuit, it must be the same at all other points, regardless of the pipe size. If the piping circuit contains parallel branch circuits, the flow can divide up through them, but the sum of all branch flow rates must equal the total system flow rate.

Incompressibility also implies that liquids can exert tremendous pressure in any type of closed container when they expand due to heating. For example, a closed hydronic piping system completely filled with water and not equipped with an expansion tank or pressure-relief valve would quickly burst at its weakest component as the water temperature increased.

STATIC PRESSURE:

Static pressure develops within a fluid due to its own weight. The static pressure that is present within a liquid depends on the depth of the liquid below (or above) some reference elevation.

Consider a pipe with a cross-sectional area of 1 square inch filled with water to a height of 10 feet, as shown in Figure 5-8.

Assuming the water's temperature is 60ºF, its density is 62.4 lb/ft^3. The weight of the water in the pipe could be calculated by multiplying its volume by its density.

$$\text{Weight} = \left(1\text{ in}^2\right)\left(10\text{ ft.}\right)\left(62.4\frac{\text{lb}}{\text{ft}^3}\right)\left(\frac{1\text{ ft}^2}{144\text{ in}^2}\right) = 4.33\text{ lb}$$

The water exerts a pressure on the bottom of the pipe due to its weight. This pressure is equal to the weight of the water column divided by the area of the pipe. In this case:

$$\text{Static pressure} = \frac{\text{weight of water}}{\text{area of pipe}} = \frac{4.33\text{ lb}}{1\text{ in}^2} = 4.33\frac{\text{lb}}{\text{in}^2} = 4.33\text{ psi}$$

If a very accurate pressure gauge were mounted into the bottom of the pipe, as shown in Figure 5-8, it would read exactly 4.33 psi.

The size or shape of the "container" holding the water has no effect on the static pressure at any given depth. Thus, an accurate pressure gauge placed 10 feet below the surface of a lake containing 60ºF water would also read exactly 4.33 psi.

The static pressure of a liquid increases proportionally from some value at the surface to larger values at greater depths below the surface. Formula 5-4 can be used to determine the static pressure below the surface of a liquid.

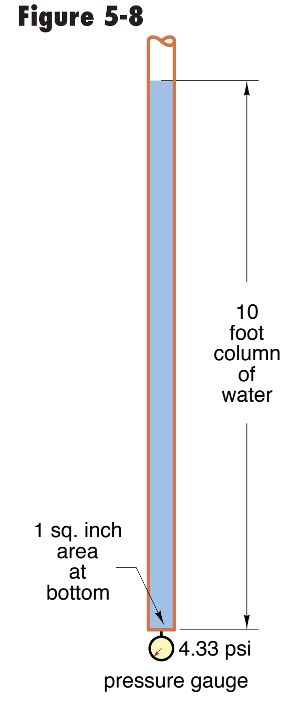
Formula 5-4

$$P_{static} = \left(\frac{D}{144}\right)h + P_{surface}$$

where:

P_{static} = static pressure at a given depth, h, below the surface of the water (psi)
h = depth below the fluid's surface (ft)
D = density of fluid in system (lb/ft^3)
$P_{surface}$ = any pressure applied at the surface of the water (psi)

Figure 5-8



If the fluid is open to the atmosphere at the top of the container, even through a pin-hole size opening, then $P_{surface}$ = 0. If the container is completely closed, then $P_{surface}$ is the pressure exerted at the top of the fluid.

Knowing the static pressure at a given location in a hydronic system lets the designer evaluate the potential for problems such as circulator cavitation. Static pressure can also help in determining if a piping loop having a known elevation is completely filled with water. Static pressure at a given location is also important when sizing and pressurizing a diaphragm-type expansion tank.

FLOW RATE:

Flow rate is a measurement of the *volume of fluid that passes a given location in a pipe in a given time*. In North America, the customary units for flow rate are *U.S. gallons per minute* (abbreviated as **gpm**). In this and other issues of idronics, flow rate is represented in formulas by the symbol f.

FLOW VELOCITY:

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

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

Formula 5-5 can be used to calculate the average flow velocity associated with a known flow rate in a round pipe.

Formula 5-5

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

where:

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

SELECTING A PIPE SIZE:

When selecting a pipe size to accommodate a given flow rate, the resulting average flow velocity should be kept between 2 and 4 feet per second. *The lower end of this flow velocity range is based on the ability of flowing water to move air bubbles along a vertical pipe.* Average flow velocities of 2 feet per second or higher can entrain air bubbles and move them in all directions, including straight down. The ability of flowing water to entrain air bubbles is important when a system is filled and purged, as well as when air has to be removed following system maintenance.

The upper end of this flow velocity range (4 feet per second) is based on minimizing noise generated by the flow. Average flow velocities above 4 feet per second can cause noticeable flow noise and should be avoided.

Figure 5-9 shows the results of applying Formula 5-5 to selected sizes of type M copper, PEX and PEX-AL-PEX tubing. Each formula in the second column can be used to calculate the flow velocity associated with a given flow rate. Also given are the flow rates corresponding to average flow velocities of 2 feet per second and 4 feet per second.

Figure 5-9

Tubing size/type	Flow velocity	Minimum flow rate (based on 2 ft/sec) (gpm)	Maximum flow rate (based on 4 ft/sec) (gpm)
3/8" copper	$v = 2.02\ f$	1.0	2.0
1/2" copper	$v = 1.26\ f$	1.6	3.2
3/4" copper	$v = 0.62\ f$	3.2	6.5
1" copper	$v = 0.367\ f$	5.5	10.9
1.25" copper	$v = 0.245\ f$	8.2	16.3
1.5" copper	$v = 0.175\ f$	11.4	22.9
2" copper	$v = 0.101\ f$	19.8	39.6
2.5" copper	$v = 0.0655\ f$	30.5	61.1
3" copper	$v = 0.0459\ f$	43.6	87.1
3/8" PEX	$v = 3.15\ f$	0.6	1.3
1/2" PEX	$v = 1.73\ f$	1.2	2.3
5/8" PEX	$v = 1.20\ f$	1.7	3.3
3/4" PEX	$v = 0.880\ f$	2.3	4.6
1" PEX	$v = 0.533\ f$	3.8	7.5
1.25" PEX	$v = 0.357\ f$	5.6	11.2
1.5" PEX	$v = 0.256\ f$	7.8	15.6
2" PEX	$v = 0.149\ f$	13.4	26.8
3/8" PEX-AL-PEX	$v = 3.41\ f$	0.6	1.2
1/2" PEX-AL-PEX	$v = 1.63\ f$	1.2	2.5
5/8" PEX-AL-PEX	$v = 1.00\ f$	2	4.0
3/4" PEX-AL-PEX	$v = 0.628\ f$	3.2	6.4
1" PEX-AL-PEX	$v = 0.383\ f$	5.2	10.4

HEAD ENERGY:

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

The mechanical energy contained in a fluid is called *head*. The units for head energy are (ft•lb/lb), shown in proper mathematical form below.

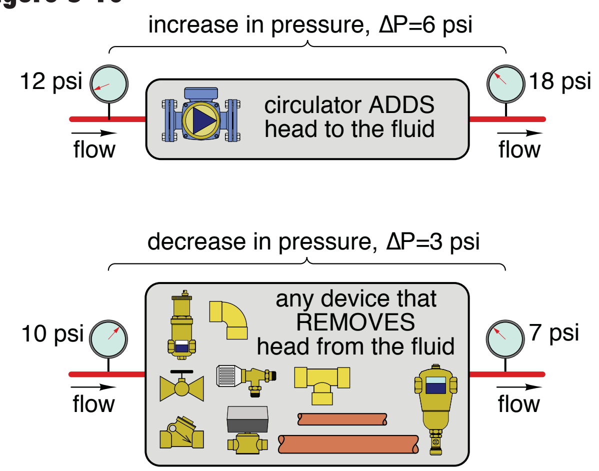
$$\frac{ft \cdot lb}{lb}$$

The unit of ft•lb (pronounced "foot pound") is a unit of *energy.* As such, it can be converted to any other unit of energy, such as a Btu.

Hydraulic engineers long ago chose to cancel the unit of pounds (lb.) in the numerator and denominator of this ratio, as shown below, and express head in the sole remaining unit of feet.

$$\frac{ft \cdot \cancel{lb}}{\cancel{lb}} = ft$$

Figure 5-10



To make a distinction between feet as a unit of distance and feet as a unit of mechanical energy in a fluid, the latter can be stated as feet *of head*.

When head energy is added to or removed from a liquid in a closed-loop piping system, there will always be an associated change in the pressure of that fluid.

Just as a change in temperature is “evidence” of a gain or loss of thermal energy, a change in pressure is evidence of a gain or loss in head energy. When head energy is lost, pressure decreases. When head energy is added, pressure increases. This principle is illustrated in Figure 5-10.

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

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

Formula 5-6 can be used to calculate the change in pressure associated with head energy being added or removed from a fluid.

Formula 5-6

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

where:

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

CALCULATING HEAD LOSS:

When designing a hydronic circuit, it is important to know how much head loss will occur, depending on the flow rate passing through that circuit. Formula 5-7 can be used to calculate this head loss *for piping circuits constructed of smooth tubing, such as copper, PEX, PEX-AL-PEX, PE-RT or PP.*

Formula 5-7

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

Where:

H_L = head loss of the circuit (feet of head)
a = fluid properties factor (see Figure 5-11)
c = pipe size coefficient (see Figure 5-12)
L = total equivalent length of the circuit (feet)
f = flow rate through the circuit (gpm)
1.75 = an exponent applied to flow rate (f)

Formula 5-7 should only be used for circuits constructed of smooth tubing. It will underestimate head loss for circuits constructed of rougher tubing, such as steel or iron pipe.

Figure 5-11

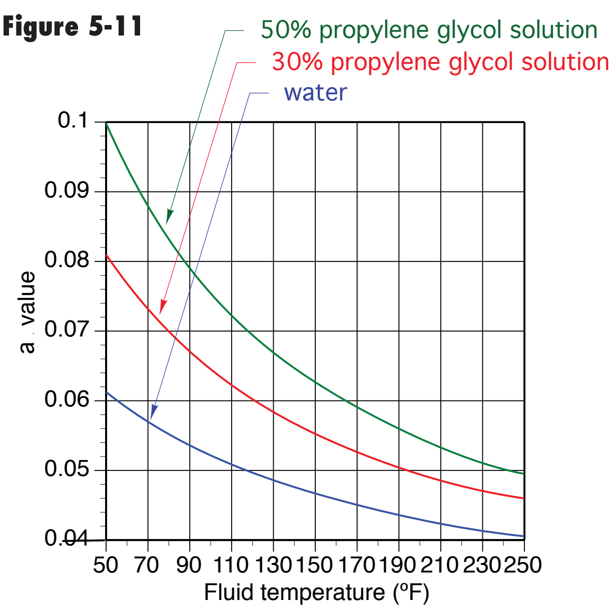


Figure 5-12

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

To calculate the head loss of a circuit using Formula 5-7, the designer must gather information from other graphs and tables. This includes values for the fluid properties factor (e.g., "a" value) and the pipe size coefficient (e.g., "c" value).

The fluid properties factor of a fluid, represents the combined effects of its density and viscosity. As such, it varies with the fluid's temperature. Figure 5-11 can be used to find the fluid properties factor (a) for water and two concentrations of antifreeze over a range of fluid temperatures.

The value of the pipe size coefficient (c) in Formula 5-7 is a constant for a given tubing type and size. It can be found for several sizes of copper, PEX and PEX-AL-PEX tubing from the table in Figure 5-12.

EQUIVALENT LENGTH:

To use Formula 5-7, the designer must also determine the total equivalent length of the piping circuit. *The total equivalent length is the sum of the equivalent lengths of all fittings, valves and other devices in the circuit, plus the total length of all tubing in the circuit.*

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

Figure 5-13 lists the equivalent lengths of common fittings and valves.

Figure 5-13

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

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

Figure 5-14

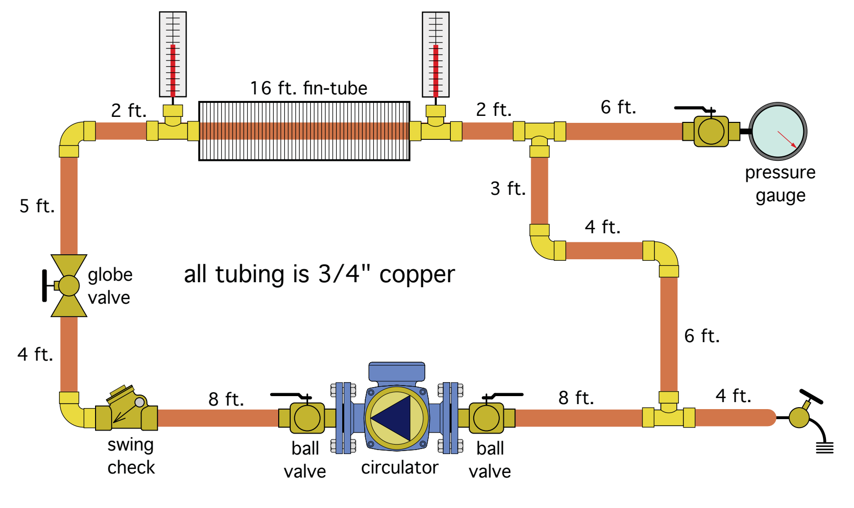
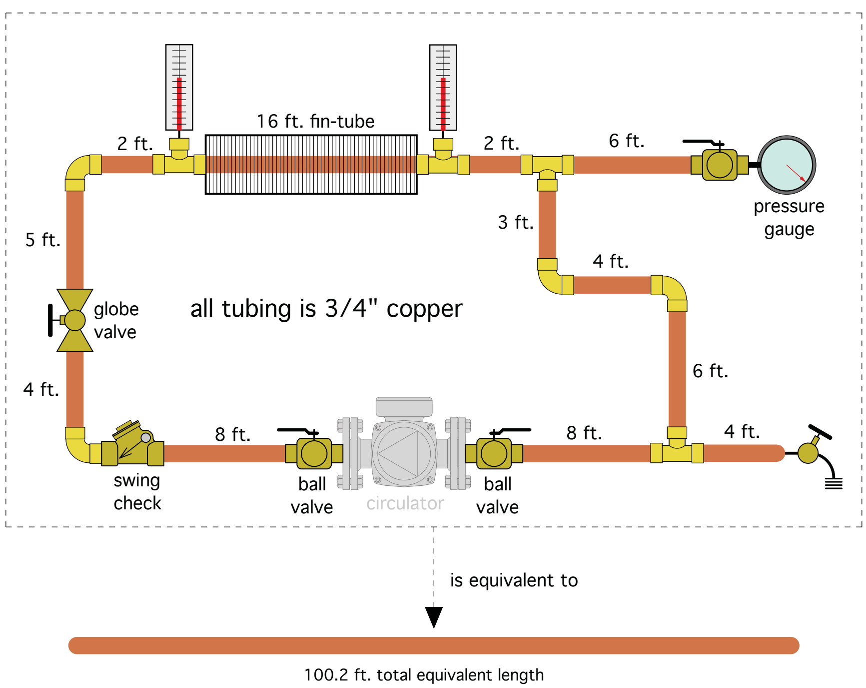


Figure 5-15

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

Figure 5-16



Example: Determine the total equivalent length of the piping circuit shown in Figure 5-14.

Solution: Only components that are in the flow path are counted when determining equivalent length. Thus, the pressure gauge seen in the upper right corner of the circuit, the 6 feet of piping and the ball valve leading up to it are not counted. Neither are the 4 feet of piping and drain valve in the lower right corner of the circuit.

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

Figure 5-15 shows the tally of equivalent lengths for all piping and components in the flow path of the circuit shown in Figure 5-14.

Thus, the total equivalent length of the circuit shown in Figure 5-14 is 100.2 feet of ¾" copper tubing. Figure 5-16 illustrates how the circuit, for purposes of determining head loss, can be thought of as a straight length of ¾" copper tubing with a length of 100.2 feet.

The head loss of this circuit can now be calculated using Formula 5-7, where the value of (L) is 100.2 feet.

Example: Determine the head loss of the circuit shown in Figure 5-14, assuming it has water flowing through it at an average temperature of 140°F and a flow rate of 5 gpm.

Solution: To use Formula 5-7, the values of (a) and (c) must first be determined.

The value of (a) for water at 140°F is found in Figure 5-11: a = 0.0475.

The value of (c) for ¾" copper is found in Figure 5-12: c = 0.061957

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

Putting these numbers into Formula 5-7 yields the head loss of the circuit under these operating conditions.

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

Keep in mind that the head loss of this circuit is only valid for the stated flow rate and operating conditions (e.g., water at an average temperature of 140ºF). Any change in the average temperature or type of fluid used in the circuit will affect the value of (a), and thus change the head loss calculated using Formula 5-7. This is also true for any change in pipe size, or components that affect the equivalent length of the circuit. Finally, if the circuit operates at a flow rate other than 5 gpm, the head loss will also change.

HEAD LOSS CURVE:

The flow rate that will develop in a hydronic circuit depends on the head loss characteristics of that circuit, as well as the head added by the selected circulator. To find that flow rate, designers need to construct a head loss curve for the piping circuit. The head loss curve is a "picture" (e.g., graph) of Formula 5-7, applied to a particular piping circuit and operating condition.

For a given circuit operating with a specific fluid at a specific average fluid temperature, Formula 5-7 can be viewed as follows:

Formula 5-7 repeated

$$H_L = (\alpha c L)(f)^{1.75} = (\text{number})(f)^{1.75}$$

Under these conditions, the head loss around the piping circuit depends only on flow rate. Formula 5-7 can be graphed by selecting several flow rates, calculating the head loss at each of them and then plotting the resulting points. Once the points are plotted, a smooth curve can be drawn through them.

Example: Use the piping circuit and operating conditions from the previous example to construct a system head loss curve.

Solution: For this circuit and these operating conditions, Formula 5-7 simplifies to the following:

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

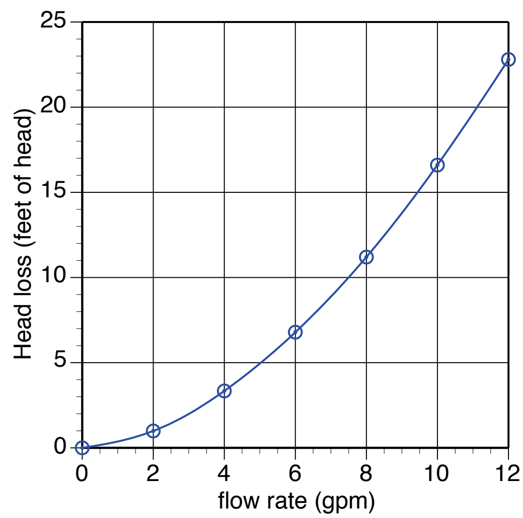
The next step is to select several random flow rates and use them in Formula 5-7 to determine the corresponding head losses. This is best done using a table as shown in Figure 5-17.

Figure 5-17

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

The next step is to plot these points and draw a smooth curve through them, as shown in Figure 5-18.

Figure 5-18



This graph is called a system head loss curve. It represents the relationship between flow rate and head loss for a given piping circuit using a specific fluid at a specific average temperature.

All piping circuits have a unique system head loss curve. It can be thought of as the analytical "fingerprint" of that piping circuit. Constructing a system head loss curve for a piping circuit is an essential step in properly selecting a circulator for that circuit.

Notice that the system head loss curve starts at zero head loss and zero flow rate. This will always be the case for *fluid-filled piping circuits*. Also notice that the system head loss curve gets progressively steeper at higher flow rates. This will also remain true for all piping circuits. It's the result of the exponent (1.75) in Formula 5-7.

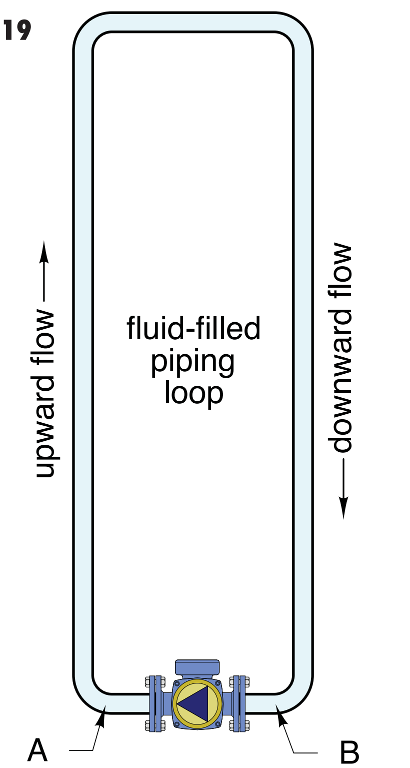
Remember that any changes to the piping components, tubing, fluid or average fluid temperature will change the value of (acL) in Formula 5-7, and thus will affect the system curve. If the value of (acL) increases, the system curve gets steeper. If the value of (acL) decreases, the system curve becomes shallower.

FLUID-FILLED PIPING CIRCUITS:

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

Consider the **fluid-filled piping loop** shown in Figure 5-19.

Figure 5-19



Assume this loop is filled with fluid that has the same temperature at all locations. A static pressure is present at point A due to the weight of the fluid column on the left side of the loop.

It might appear that a circulator placed as shown would have to overcome this pressure to lift the fluid and establish flow around the loop. This is *NOT* true. The reason is that the static pressure exerted by the fluid at point A will always be the same as the static pressure exerted at point B (which is at the same elevation as point A).

Now consider what happens when the circulator operates: The weight of fluid moving up the left side of this circuit over any given time is always the same as the weight of fluid moving down the right side during that time. This has to be true because the fluid has nowhere else to go within the circuit. If, for example, we assume that 100 pounds of fluid went up the left side of the circuit in one minute, and that only 99 pounds of fluid came down the right side over that minute, the question remains: Where did the difference (1 pound) of fluid go? In a closed circuit (with no leaks), the only possible answer is nowhere. Thus, the initial assumption of 100 pounds of fluid moving up per minute, and 99 pounds of fluid moving down during that minute, has to be false.

One can liken fluid moving around a closed, fluid-filled circuit to the operation of a ferris wheel with the same weight in each seat. The weight in the seats moving up exactly balances the weight in the seats moving down. If it were not for friction in the bearings and air resistance, this balanced ferris wheel would continue to rotate indefinitely once started. Similarly, since the flow rate is always the same throughout a fluid-filled piping loop, the weight of the fluid moving up the circuit in a given amount of time is always balanced by the weight of fluid moving down the circuit during that time. If it were not for the viscous friction of the fluid, it too would continue to circulate indefinitely within a piping loop.

This effect holds true regardless of the size, shape or height of the piping path in the upward flowing portion of the loop versus those in the downward flowing portion.

Another important principle has also been established by considering this circuit. Namely, that the circulator in a closed fluid-filled piping circuit is only responsible for replacing the head energy lost due to friction through the piping components. *The circulator is NOT responsible for "lifting" the fluid in the upward flowing portion of the circuit.*

This explains why a small circulator can establish and maintain flow in a filled piping loop, even if the top of the loop is several stories above the circulator and contains hundreds or even thousands of gallons of fluid. Unfortunately, many circulators in hydronic systems are needlessly oversized because this principle is not understood.

PUMP CURVES:

As previously stated, an operating circulator adds head energy to fluid as it passes through the circulator. The amount of head energy a given circulator adds depends on the flow rate passing through it. The greater the flow rate, the lower the amount of head energy added to each pound of fluid. This is a characteristic of all hydronic circulators and can be represented graphically as a "pump curve." An example of a pump curve for a small hydronic circulator is shown in Figure 5-20.

Figure 5-20

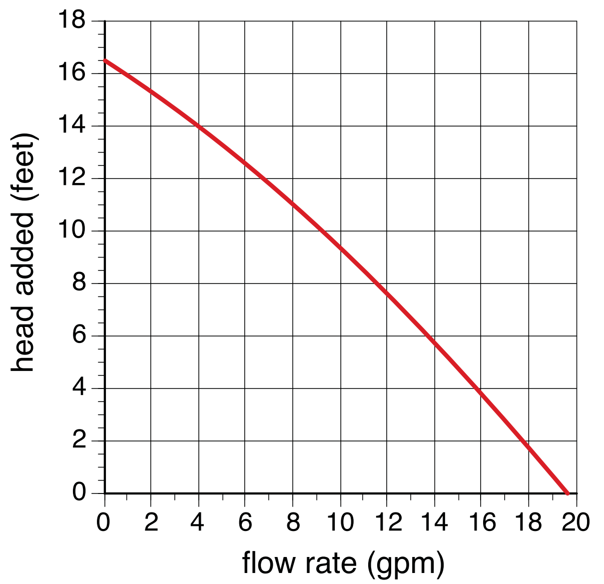


Figure 5-21

Courtesy of Grundfos

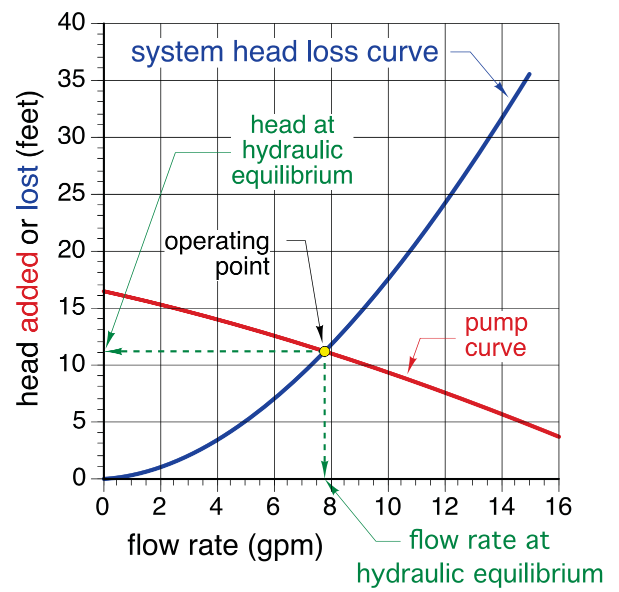
The pump curve of a circulator is developed from test data using water in the temperature range of 60ºF to 80ºF. For fluids with higher viscosities, such as glycol-based antifreeze solutions, there is a very small decrease in head and flow rate capacity of the circulator. However, for the fluids and temperature ranges commonly used in hydronic heating systems, this variation is so small that it can be safely ignored. Thus, for most residential and light commercial hydronic systems, *pump curves (but not system head loss curves) may be considered to be independent of the fluid being circulated.*

Pump curves are extremely important in matching the performance of a circulator to the flow requirements of a piping circuit. All circulator manufacturers publish these curves for the circulator models they offer. In many cases, the pump curves for multiple circulators are plotted on the same graph so that performance comparisons can be made. Figure 5-21 shows an example of a family of pump curves.

HYDRAULIC EQUILIBRIUM:

It is possible to predict the flow rate that will develop when a specific circulator is installed in a specific hydronic circuit. *That flow rate will be such that the head energy added by the circulator is exactly the same as the head energy dissipated by the piping circuit.* This condition is called hydraulic equilibrium. The flow rate at hydraulic equilibrium is found by plotting the head loss curve for the circuit on the same graph as the pump curve for the circulator, as shown in Figure 5-22.

Figure 5-22



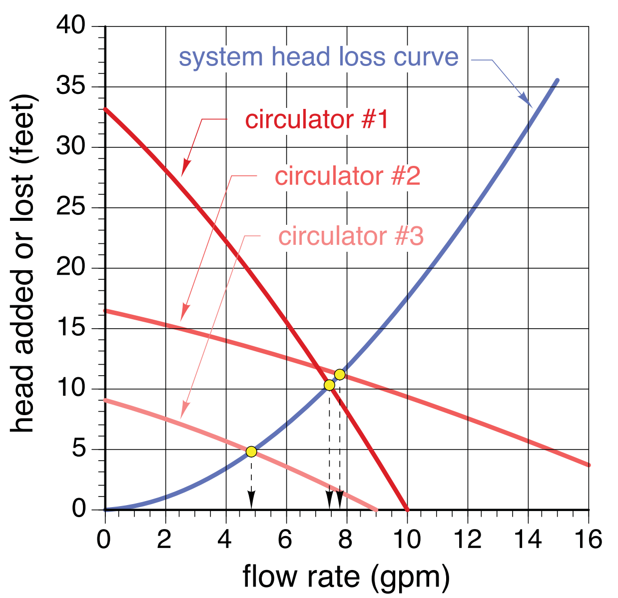
The point where the head loss curve of the piping circuit crosses the pump curve for the circulator is called the operating point. This is where hydraulic equilibrium occurs.

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

A performance comparison of several "candidate" circulators within a given piping circuit can be made by plotting their individual pump curves on the same graph as that circuit's head loss curve. The intersection of each circulator's pump curve with the circuit's head loss curve indicates the operating point for that circulator. By projecting vertical lines from the operating points down to the horizontal axis, the designer can determine the flow rate each circulator would produce within that circuit, as shown in Figure 5-23.

Notice that even though the curves for circulators 1 and 2 are markedly different, they intersect the system head loss curve at almost the same point. Therefore, these two circulators would produce very similar flow rates of about 7.5 gpm and 7.8 gpm in this piping system. The flow rate produced by circulator #3, about 4.9 gpm, is considerably lower.

Figure 5-23



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

Examples of what could change the head loss curve include:

• The type of fluid in the circuit changes
• The average temperature of the fluid changes
• Changes are made to the pipe type, pipe size or to the components in the piping circuit
• Valve settings are changed

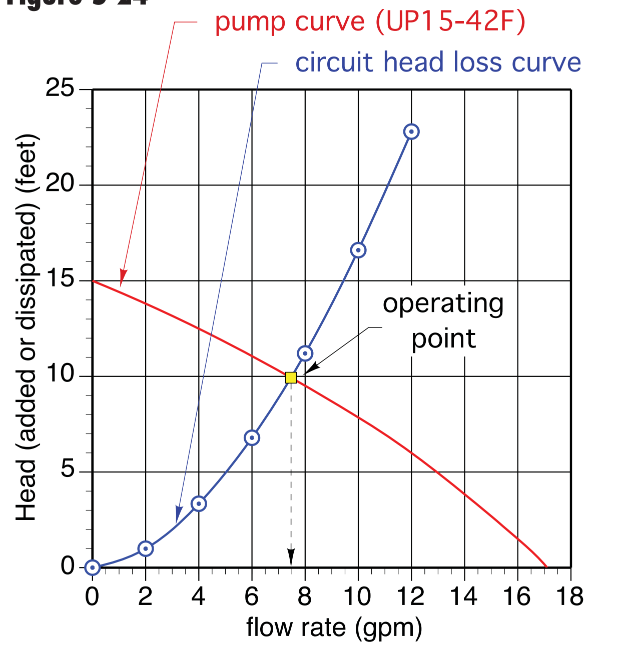
Examples of what could change the pump curve include:

• Change to a different circulator
• Change to a different motor speed setting
• Altering the circulator's impeller

Example: Determine the flow rate for the circuit shown in Figure 5-14, assuming a UP15-42F circulator is used (pump curve shown in Figure 5-21). The circuit operates with an average water temperature of 140ºF.

Solution: The system curve for this piping system operating with the stated conditions has already been determined and is shown in Figure 5-18. If this curve is plotted on the same graph with the pump curve of the UP15-42F circulator, the operating point is easily determined as shown in Figure 5-24.

Figure 5-24



Drawing a line from the operating point to the horizontal axis indicates that the circuit will operate at a flow rate of about 7.5 gpm.

HYDRAULIC SEPARATION:

Many hydronic systems contain multiple independently controlled circulators. These circulators can vary significantly in their flow and head characteristics. Some may operate at constant speed, while others will operate at variable speeds.

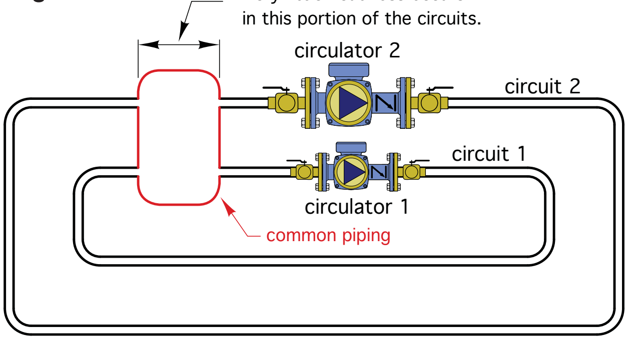
When two or more circulators operate simultaneously in the same system, they each attempt to establish differential pressures based on their own pump curves.

Ideally, each circulator in a system should establish a differential pressure and flow rates that is <u>*unaffected*</u> *by the presence of another operating circulator within the system.* When this desirable condition is established, the circulators are said to be hydraulically separated from each other.

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

The degree to which two or more operating circulators interact with each other depends on the head loss of the piping path they have in common. This piping path is called the "common piping," since it is in common with both circuits. *The lower the head loss of the common piping, the less the circulators will interfere with each other.* Figure 5-25 illustrates this principle for a system with two independently operated circuits.

Figure 5-25



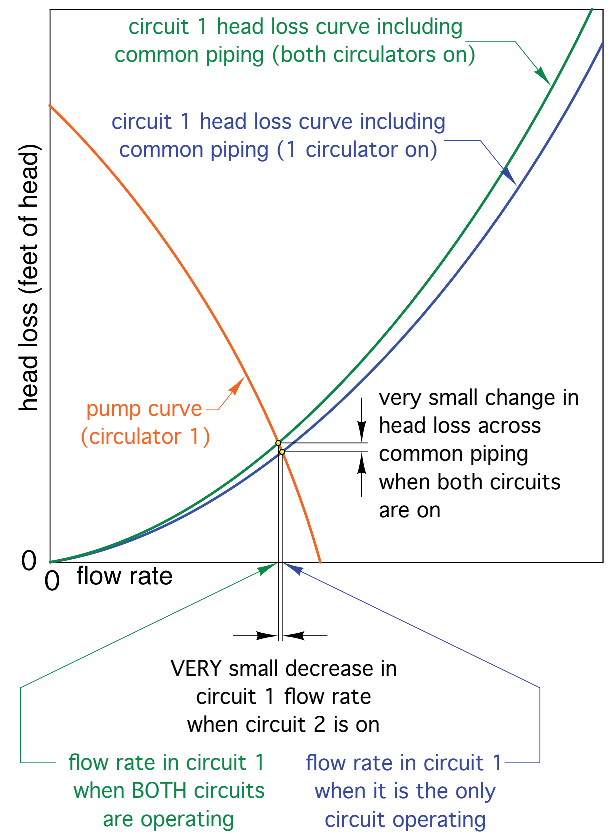
In this system, both circuits share common piping. The spacious geometry of this common piping creates very low flow velocity through it. As a result, very little head loss or pressure drop can occur across it.

Assume that circulator 1 is operating, but that circulator 2 is off. The lower (blue) system head loss curve in Figure 5-26 applies to this situation.

The point where this lower system head loss curve crosses the circulator's pump curve establishes the flow rate in circuit 1.

Next, assume circulator 2 is turned on and circulator 1 remains on. The flow rate through the common piping increases, and so does the head loss and pressure drop across it. However, because of its spacious geometry,

Figure 5-26

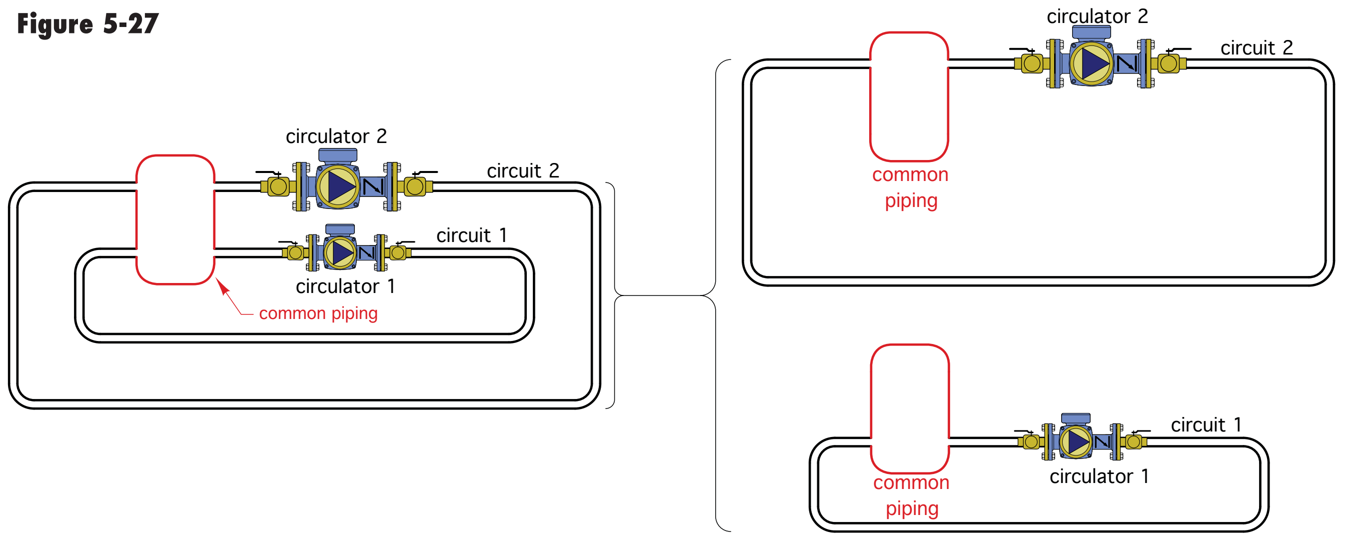


the increase in head loss and pressure drop is very slight. The system head loss curve that is now seen by circulator 1 has very slightly steepened. It is the upper (green) curve shown in Figure 5-26. The operating point of circuit 1 has moved very slightly to the left, and as a result, the flow rate through circuit 1 has decreased very slightly.

Such a small change in the flow rate through circuit 1 will have virtually no effect on its ability to deliver heat. Thus, the interference created when circulator 2 was turned on is of no consequence. We could say that this situation represents *almost* perfect hydraulic separation between the two circulators.

One could imagine a hypothetical situation in which the head loss across the common piping was zero, even with both circuits operating. Because no head loss occurs across the common piping, it would be impossible for either circulator to have any effect on the other circulator. Such a condition would represent "perfect" hydraulic separation and would be ideal. Fortunately, perfect hydraulic separation is not required to ensure that the flow rates through independently operated circuits, each with their own circulator and each sharing the same low

Figure 5-27



head loss common piping, remain reasonably stable, and thus are capable of delivering consistent heat transfer.

With good hydraulic separation, the simultaneously operating circulators can barely detect each other's presence within the system. Thus, each circuit operates as if it were an independent circuit. One can think of circuits that are hydraulically separated as if they were physically disconnected from each other, as illustrated in Figure 5-27.

Any component or combination of components that has very low head loss and is common to two or more hydronic circuits can provide hydraulic separation between those circuits.

One way to create low head loss is to keep the flow path through the common piping very short. Another way to create low head loss is to slow the flow velocity through the common piping.

Examples of devices that use these principles include:

- Closely spaced tees
- A tank (which might also serves other purposes in the system)
- A hydraulic separator

The common piping in Figure 5-28 consists of the closely spaced tees and generously sized headers. Because they are positioned as close to each other as possible, there is virtually no head loss between the tees. These tees form the common piping between the heat source circuit and the distribution circuits, and thus provide hydraulic separation between these circuits.

Figure 5-28

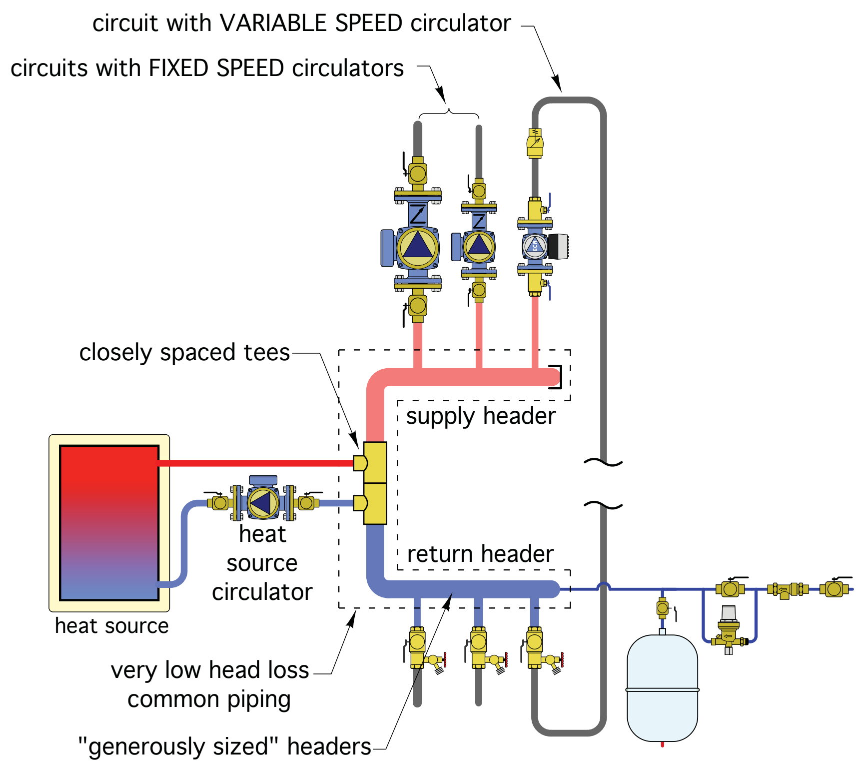


Figure 5-29

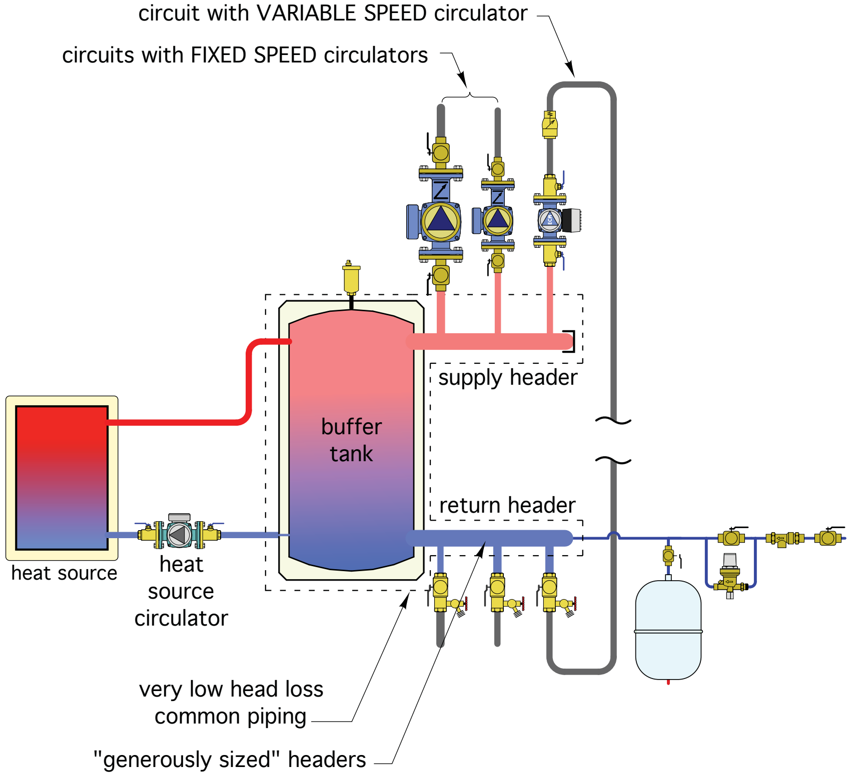
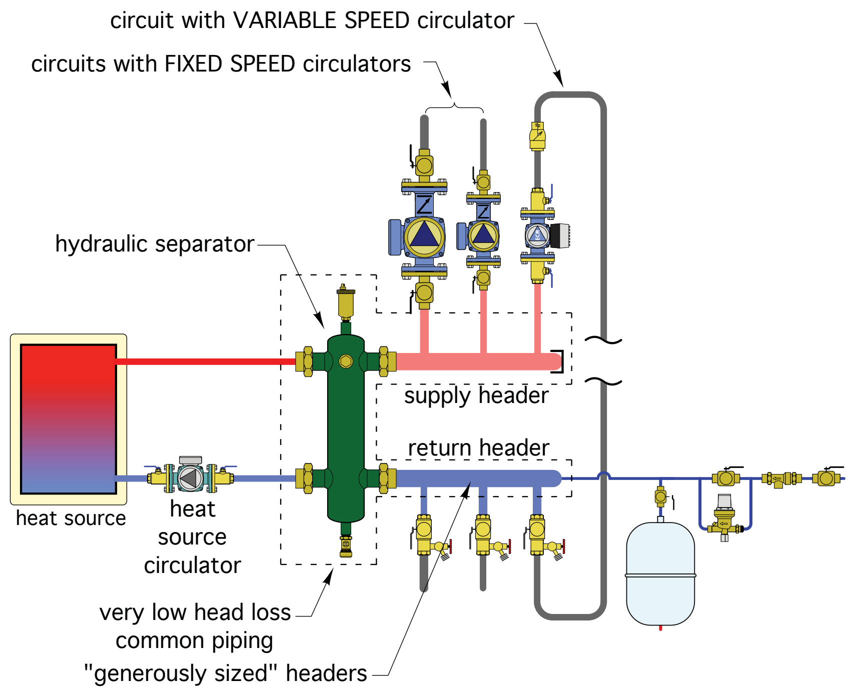


Figure 5-30



The generously sized headers create low flow velocity and low head loss, and thus provide hydraulic separation between the three distribution circulators. *These headers should be sized so that the maximum flow velocity when all circulators served by the header are on is no more than 2 feet per second.* Together, these details create hydraulic separation between all four circulators in the system.

The closely spaced tees allow the heat source circulator to only "see" the flow resistance of the heat source and piping between the boiler and closely spaced tees. The heat source circulator doesn't help move water through the distribution circuits. Likewise, the three distribution circulators are only responsible for circulation through their respective circuits and do not assist in moving flow through the heat source.

Notice also that constant speed and variable-speed circulators, perhaps of different sizes, can be combined onto the same generously sized header system. Interaction between these circulators will be very minimal because of the very low head loss of those headers.

Figure 5-29 shows a buffer tank combined with generously sized headers serving as the low head loss common piping that provides hydraulic separation between the heat source circulator and each of the distribution circulators. This demonstrates that hydraulic separation can sometimes be accomplished as an ancillary function to the main purpose of the device (e.g., hydraulic separation is not the main function of the buffer tank).

Still another method of providing hydraulic separation is through use of a device that is appropriately called a "hydraulic separator." Figure 5-30 shows a hydraulic separator installed in place of the buffer tank of Figure 5-29. Note the similarity of the piping connections between the buffer tank and hydraulic separator.

Hydraulic separators, which are sometimes also called low loss headers, create a zone of low flow velocity within their vertical body. The diameter of the body is typically three times the diameter of the connected piping. This causes the vertical flow velocity in the vertical body to be approximately $1/9^{th}$ that of the connecting piping. Such low velocity creates very little head loss and very little pressure drop between the upper and lower connections. Thus, a hydraulic separator provides hydraulic separation in a manner similar to a buffer tank, only smaller.

The reduced flow velocity within a hydraulic separator allows it to perform two additional functions. First, air bubbles can rise upward within the vertical body and be captured in the upper chamber. When sufficient air collects at the top of the unit, the float-type air vent allows it to be ejected from the system. Thus, a hydraulic separator can replace the need for a high-performance air separator.

Second, the reduced flow velocity allows dirt particles to drop into a collection chamber at the bottom of the vertical body. A valve at the bottom can be periodically opened to flush out the accumulated dirt. Thus, the hydraulic separator also serves as a dirt separator.

The ability of modern hydraulic separators to provide three functions—hydraulic separation, air removal and dirt removal—makes them well-suited for a variety of systems, especially systems in which an older distribution system, one that may have some accumulated sludge, is connected to a new heat source.

idronics #1 provides a more detailed description of hydraulic separators and shows ways to apply them.

6: HYDRONIC HEAT SOURCES

One of the benefits of hydronic systems is that they can operate using a wide variety of heat sources. Almost any device that can heat water is a *potential* hydronic heat source. This section briefly discusses several hydronic heat sources that are commonly used in residential and light commercial systems.

CONVENTIONAL BOILERS:

Boilers can be classified in many ways, such as by the fuel they use, heat output rating, construction material, heat exchanger geometry and methods for exhausting combustion gases. However, from the standpoint of system design, it is important to distinguish between "conventional" boilers and "condensing" boilers.

Conventional boilers are intended to operate so that the water vapor produced during combustion does not condense on a sustained basis within the boiler or its venting system.

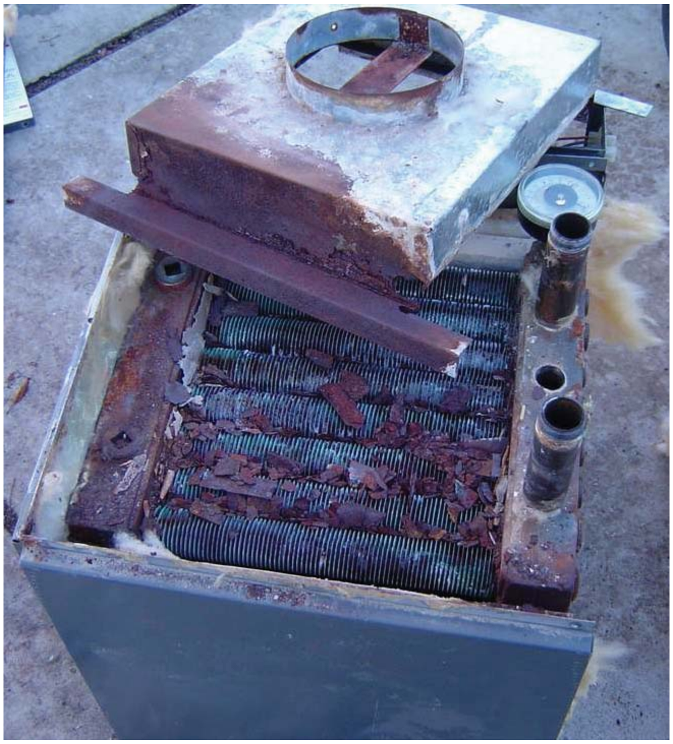
Figure 6-1

Courtesy of Weil Mclain

Nearly all boilers with cast iron, carbon steel or copper heat exchangers fall into this category. An example of a small gas-fired cast iron boiler is shown in Figure 6-1. Also shown is the schematic symbol used to represent any type of conventional boiler in this and other issues of *idronics*.

It's important to understand that given the right operating conditions, ANY boiler can be forced to operate with sustained flue gas condensation. The condensate that can form is not pure water. It contains several chemical compounds based on the fuel burned and the efficiency at which combustion occurs. These compounds make the condensate highly corrosive. Materials such as cast iron, steel, copper and galvanized steel vent connector piping can rapidly corrode if repeatedly exposed to this condensate. The boiler heat exchanger shown in Figure 6-2 is evidence of such damage.

Figure 6-2



Courtesy of Dave Stroman

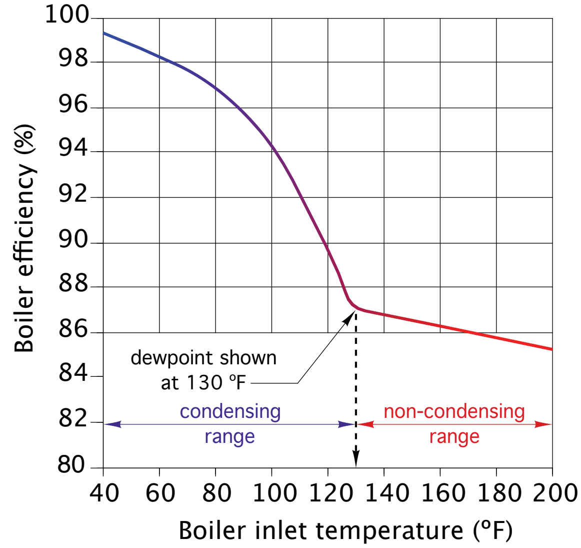
It is therefore *imperative that conventional boilers are applied and operated so that sustained flue gas condensation does not occur.*

It's also important to realize that *all* boilers experience *intermittent* flue gas condensation when starting from a cold condition. However, when a conventional boiler is properly applied, intermittent flue gas condensation quickly evaporates as the boiler warms. Intermittent flue gas condensation is not detrimental to the boiler.

The presence or absence of sustained flue gas condensation is determined by the temperature of the water entering the boiler, as well as the air/fuel ratio at which the combustion process occurs. Of these, only the boiler inlet water temperature is easily controllable by how the boiler is applied.

Figure 6-3 shows a typical relationship between the temperature of the water entering a boiler and that boiler's associated thermal efficiency. The latter is the percentage of the chemical energy content of the fuel that is converted to heat and transferred to the water in the boiler's heat exchanger.

Figure 6-3



If the temperature of the water entering the boiler is relatively high (i.e., 200ºF), the combustion-side surfaces of the boiler's heat exchanger are well above the dewpoint temperature of the water vapor in the exhaust gas stream. Under such conditions, no condensation occurs within the boiler.

However, as the temperature of the water entering the boiler decreases, so does the temperature of the boiler's heat exchanger surfaces. At some entering water temperature, these surfaces reach the dewpoint temperature of the water vapor in the exhaust stream. This temperature varies with the type of fuel burned, as well as the air/fuel ratio supplied to the combustion chamber. For a conventional boiler operating with natural gas, the inlet water temperature at which condensation begins to form on the boiler's heat exchanger is approximately 130ºF. The lower the entering water temperature, the more condensate that forms. In theory, if all the water vapor formed during combustion was condensed, approximately 1.15 gallons of liquid would be generated for each therm (e.g., 100,000 Btu) of natural gas burned.

idronics #7 provides a detailed discussion of how various mixing strategies can be used to protect conventional boilers for operating with sustained flue gas condensation.

CONDENSING BOILERS:

Figure 6-3 shows that the thermal efficiency of a boiler increases rapidly when that boiler is operating in "condensing mode," (e.g., with sustained flue gas condensation). The increased thermal efficiency comes from capturing the latent heat released as water vapor changes to liquid. Each pound of water vapor converted to liquid releases about 970 Btu of heat to the boiler's heat exchanger. In a conventional boiler, this latent heat is carried away with the exhaust stream.

When fuel was inexpensive, there was little motivation to capture this latent heat. However, steadily increasing fuel costs have provided the incentive for boiler manufacturers to create boilers that are *intended to operate with sustained flue gas condensation.* Appropriately, they are called "condensing boilers."

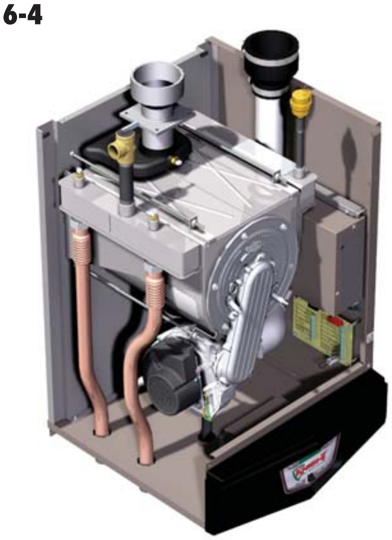
These boilers are constructed with large heat exchanger surfaces made of stainless steel or aluminum. These heat exchangers are capable of extracting more heat from the exhaust gases compared to the heat exchangers used in conventional boilers. When operated with suitably low inlet water temperatures, these boilers can easily cool the exhaust stream below the dewpoint temperature of the water vapor, and thus allow condensation to occur.

MODULATING/CONDENSING BOILERS:

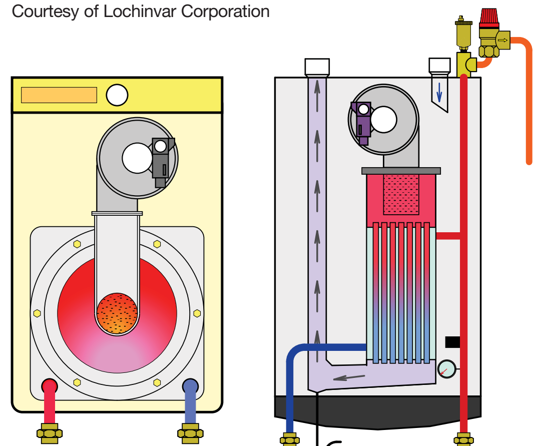
Early generation condensing boilers were built to operate at a fixed firing rate and thus a fixed rate of heat output. Today, nearly all condensing boilers can vary their heat output from some maximum rate down to approximately 20% of that maximum output. These boilers are said to be modulating. The term "mod/con" is often used to describe boilers that can vary their firing rates and are intended to operate with sustained flue gas condensation.

Figure 6-4 shows an example of a modern mod/con boiler, along with the schematic symbols used to represent mod/con boilers in this and other issues of *idronics*.

Figure 6-4



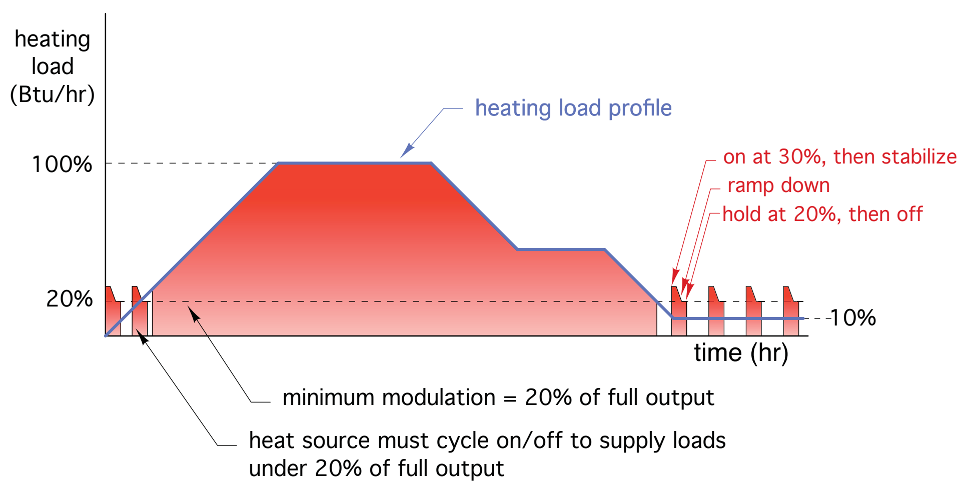
Courtesy of Lochinvar Corporation



mod/con boilers

Modulating the heat output of a boiler is similar to changing the throttle setting on an internal combustion engine. The rate of heat production is lowered by reducing the fuel and air sent to the burner. A variable-speed blower controls the air flow rate entering a sealed combustion chamber. Natural gas or propane is metered into this airstream in proportion to the air flow rate. The slightly pressurized mixture of gas and air is forced into a burner

Figure 6-5



head and ignited. Burner heads are typically made of stainless steel or porous ceramic materials.

The ability of a mod/con boiler to modulate is expressed by its “turndown ratio.” This is the reciprocal of the lowest possible percentage of full heat output rate the boiler can maintain. For example: If a boiler can maintain stable operation down to 20% of its maximum heat output rate, it would have a turndown ratio of 5:1 (e.g., 5 being the reciprocal of 20% expressed as a decimal number).

Figure 6-5 shows how the heat output of a modulating boiler with a turndown ratio of 5:1 matches its heat output to a hypothetical load.

The blue line that borders the upper edge of the large shaded area is called a heating load profile. It represents the rate at which heat is required by the load over a period of several hours. The load begins at zero, steadily increases to a maximum value and then reduces in various slopes and plateaus over time, eventually stabilizing at 10% of maximum load.

The red-shaded areas represent the heat output from a mod/con boiler. That boiler is assumed to have a turn down ratio of 5:1, and it is sized so that it exactly matches the maximum heating demand when it operates at 100% output.

The pulses seen near the beginning and end of the load profile result from the boiler turning on at approximately 30% output, then quickly reducing output to the lower modulation limit of 20% and finally turning off for a short time.

The large red-shaded area shows how the mod/con boiler accurately matches the load whenever it is 20% or more of the maximum load. This is ideal.

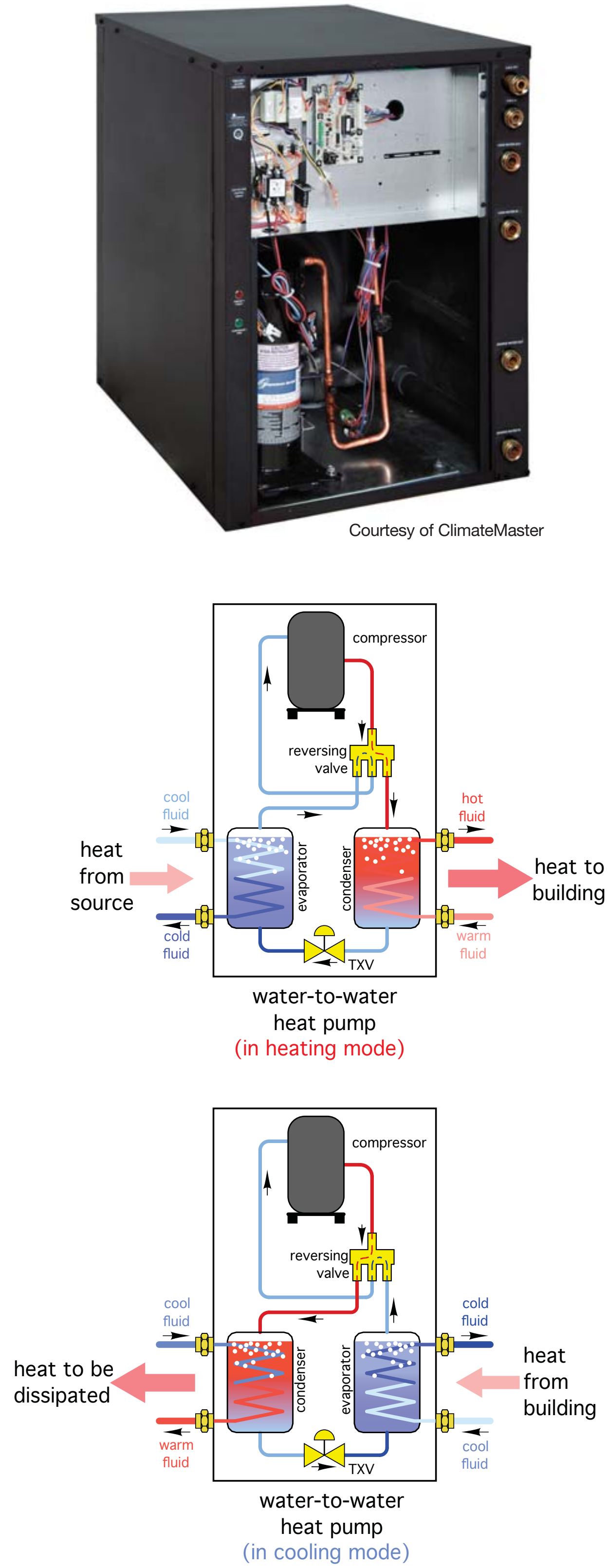
When loads drop below the minimum modulation rate (which in this case is 20%), the burner must cycle on and off to avoid oversupplying the load. When these on/off cycles occur frequently, they are known as “short cycling.” Such operation is always undesirable. It causes excessive wear on components such as the boiler’s ignition system. It also results in higher emissions and low thermal efficiency. Fortunately, short cycling can be avoided by proper system design and other components discussed later in this issue.

HEAT PUMPS:

The low water temperatures at which some hydronic distribution systems can operate allows the possibility to use a heat pump as the system’s heat source.

All heat pumps move heat from some material at a lower temperature (e.g., a temperature at which the material

Figure 6-6



Courtesy of ClimateMaster

cannot directly supply heat to the load) to another material at a higher temperature. This is done using a refrigeration cycle.

The two types of heat pumps most often used in hydronic systems are:

- Water-to-water heat pumps
- Air-to-water heat pumps

Water-to-water heat pumps typically absorb low temperature heat from tubing buried in the earth or directly from ground water. Such heat pumps are further classified as "geothermal" heat pumps. They then deliver higher temperature water to a hydronic distribution system.

Figure 6-6 shows an example of a water-to-water heat pump, along with the schematic symbols used to represent it.

Air-to-water heat pumps absorb low temperature heat directly from outside air. Then, as with water-to-water heat pumps, they deliver higher temperature water to a hydronic distribution system within the building.

Figure 6-7 shows an example of an air-to-water heat pump mounted on a concrete base outside a house, along with the schematic symbols used to represent it.

A unique feature of both water-to-water and air-to-water heat pumps is that they can reverse the direction of heat movement. Thus, the same heat pump that supplies warm water for space heating in winter can also supply chilled water for space cooling in summer. This ability opens up a wide variety of applications.

idronics #9 provides detailed descriptions of water-to-water and air-to-water heat pumps. It also describes the thermal performance of these heat pumps and shows how to integrate them into hydronic heating and cooling systems.

SOLAR THERMAL COLLECTORS:

Significant increases in the price of conventional fuel combined with increased interest in environmental stewardship have created a growing market for thermally based use of solar energy. Solar thermal collectors, such as the one shown in Figure 6-8, can be easily integrated as a hydronic heating source in systems that supply both space heating and domestic hot water. Such systems are called solar "combisystems."

Figure 6-7

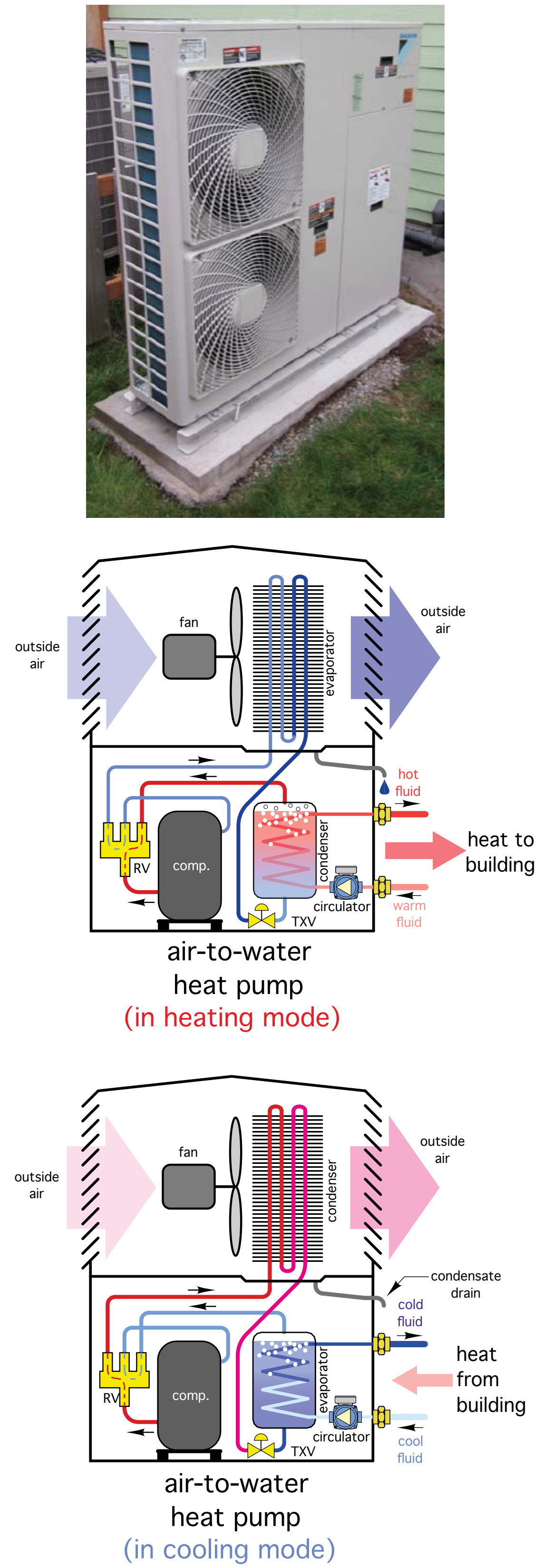


Figure 6-8

Nearly all solar combisystems are equipped with "auxiliary" heating sources such as a boiler. Most of these systems are designed so that the auxiliary heat source will automatically operate to supply the heating load if the temperature of the thermal storage tank is insufficient to do so.

idronics #3 and idronics #6 discuss both solar thermal collectors and systems in more detail. idronics #3 is an introduction to the subject, and idronics #6 is specifically focused on solar combisystems.

SOLID FUEL BOILERS:

Many areas of North America have abundant forests that could sustainably yield much more wood than is currently used. The wood can be harvested and processed in several ways. The most common form in which wood is supplied as a heating fuel is as cut, split and dried cordwood.

Figure 6-9

Several types of hydronic heat sources have been developed to burn cordwood. They include outdoor wood-fired furnaces and pressurized boilers. The latter can be further categorized as either atmospheric boilers or wood-gasification boilers. An example of the latter is shown in Figure 6-10.

Figure 6-10

Courtesy of Abasco, Inc.

When operated with properly dried (20% or lower moisture content) wood, gasification boilers can now produce combustion efficiencies approaching 90%. They also operate with much lower emissions compared to non-gasification wood-burning devices.

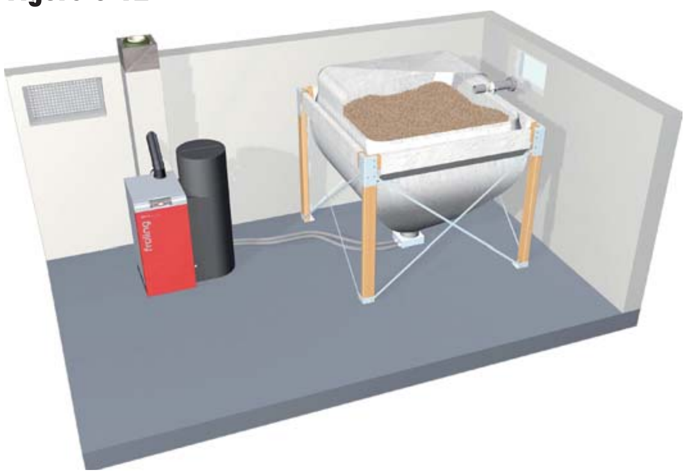
Pellets are another form of wood fuel. They are made by compressing dried wood fiber under very high pressure (60,000 psi). This forces the natural resins within the wood to bind the fibers together. The resulting ¼" diameter pellets are shown in Figure 6-11.

Figure 6-11

Given their small size and consistent form, pellets can be moved from storage bins to boilers by motorized augers or pneumatic transport systems. Thus, pellet-fired boilers can operate with very little manual intervention. These boilers can automatically ignite the pellets when heat is required, and even modulate heat production based on the rate at which pellets are burned.

Figure 6-12 shows an example of a pellet-fired boiler that is automatically supplied from an inside pellet storage hopper.

Figure 6-12



Courtesy of Tarm BioMass

Systems that supply both space heating and domestic hot water can be designed around wood-fired and pellet-fired boilers. These systems usually include a well-insulated thermal storage tank that can absorb any excess heat production from the boiler when its output exceeds the current heating load.

idronics #10 provides a complete discussion of how hydronics systems can be assembled around both wood-fired and pellet-fired boilers.

7: ESSENTIAL HYDRONICS HARDWARE

Although the heat source is a critical component in any hydronic heating system, it is useless without a variety of other “building block” components, such as pipe, fittings and valves. This section provides basic information on the latter. Later sections will provide basic information and several other building block components.

PIPING:

Early generation hydronic systems were piped with rigid steel and iron piping. Threaded fittings were used to join this piping in residential and light commercial systems. In large systems, the piping was usually joined by welding. These piping materials and joining systems were used largely because they were available at the time.

In North America, some steel and iron piping is still used. However, its use is typically limited to “near boiler” piping in the mechanical room. It is seldom used for distribution piping through the building.

After World War II, copper water tube gradually replaced iron and steel piping as the dominant piping material for smaller hydronic systems. In the United States, copper water tube is manufactured according to the ASTM B88 standard. In this category, pipe size refers to the nominal inside diameter of the tube. The word “nominal” means that the measured *inside* diameter is *approximately* equal to the stated pipe size. For example, the actual inside diameter of a 3/4” type M copper tube is 0.811 inches, not 0.750 inches as the stated pipe size might imply.

The outside diameter of copper water tube is always 1/8 inch larger than the nominal inside diameter. For example, the outside diameter (O.D.) of 3/4” type M copper tube is 0.875 inches. This is exactly 7/8 inches, or 1/8 inch larger than the nominal pipe size.

Copper water tube is available in three wall thicknesses designated as types K, L and M in order of decreasing wall thickness. The outside diameters of K, L and M copper tubing are identical. This allows all three types of tubing to be used with the same fittings and valves.

Because the operating pressures of residential and light commercial hydronic heating systems are relatively low, the thinnest-wall copper tubing (type M) is most often used. This wall thickness provides several times the pressure rating of other common hydronic system components. Some codes may require the use of type L copper water tube for conveying domestic water. Type L tubing is also preferred if the tubing will be mechanically bent to provide changes in direction.

Copper tubing has relatively good resistance to corrosion that can result from carrying water containing dissolved oxygen. However, copper tubing is not immune from all corrosion. The acids formed by glycol-based antifreeze that have chemically degraded due to extreme temperatures can be aggressive toward copper. This type of corrosion can occur in solar thermal systems where collectors are allowed to repeatedly "stagnate" with no flow during times of high solar radiation intensity and high air temperatures. Copper tubing can also be corroded by water with high concentrations of hydrogen sulfide.

Common methods of joining copper tubing in hydronic systems include soft soldering and press fitting. Figure 7-1 shows copper tubing joined to a tee fitting using soft soldering.

Figure 7-1

A newer method of joining copper tubing for hydronic system applications is called press fitting. This joining system uses special fittings containing elastomer (EPDM) O-rings that are mechanically compressed against the tube wall using a special tool. Figure 7-2 shows this pressing tool in use.

Advantages of press fit joints include:

- They can be made without cleaning or fluxing tube and fittings
- They can be made without flame from torches
- They can be made on fittings that is not completely dry
- They reduce installation time relative to soldered joints

The use of press fittings also has some limitations:
- Once made, press fit joints are permanent; they cannot be separated as soldered joints can.
- Press fitting requires special tools and fittings

Figure 7-2

Courtesy of Viega

- The O-rings used in press fittings have temperature and pressure limitations that are lower than those associated with soldered joints. However, some fittings are rated for operation at temperatures over 200ºF, and thus are well-suited for use in modern, lower temperature hydronic heating systems.

PEX TUBING:

Crosslinked polyethylene tubing, commonly referred to as **PEX**, is a product that is now used worldwide for a variety of hydronic heating applications. It is best known for its use in hydronic radiant panel heating systems.

PEX has significantly higher pressure/temperature ratings than standard high density polyethylene (HDPE) tubing. In North America, most PEX tubing used in hydronic systems conforms to the ASTM F876 standard. It is commonly available in continuous rolls up through 1,000 feet, and in nominal pipe sizes from 5/16" through 2". Tube sizes of 3/8" through 3/4" are commonly used in radiant panels. Large sizes are used for distribution piping. Figure 7-3 shows samples of PEX tubing in sizes of 3/8", 1/2", 5/8" and 3/4".

Figure 7-3

With all plastic tubing, there is a tradeoff between operating temperature and allowable pressure. The ASTM F876 standard establishes three simultaneous temperature/pressure ratings for PEX tubing. The higher ratings for continuous operation are 180ºF at 100 psi and 200ºF at 80 psi.

Specialized fittings are available from the tubing manufacturers to transition from PEX tubing to standard metal pipe fittings, both threaded and soldered.

All polymer tubes allow diffusion of oxygen molecules from the outside of the tube to the fluid within them. When PEX tubing is used in closed-loop hydronic systems that contain any iron or steel components, the tubing should be specified with an oxygen diffusion barrier. This barrier is a thin layer of a special chemical that is either laminated to the outer surface of the tubing or co-extruded as an embedded layer. Its purpose is to reduce the diffusion of oxygen molecules through the tube wall, and thus significantly reduce the potential for oxygen-based corrosion within the system. All oxygen diffusion barriers on polymer tubing used for hydronic system applications should meet or exceed the requirements of the DIN 4726 standard.

PEX-AL-PEX TUBING:

Another type of tubing well-suited for hydronic heating systems is called PEX-AL-PEX, (a.k.a. "composite" tubing). It consists of three concentric layers bonded together with special adhesives. The inner and outer layers are PEX. The middle layer is longitudinally welded aluminum. A close-up of the cut edge of a PEX-AL-PEX tube is shown in Figure 7-4. The aluminum layer is easy to see between the inner and outer layers of PEX.

Figure 7-4

The PEX-AL-PEX tubing commonly used in hydronic heating systems conforms to the ASTM F1281 standard. This tubing has slightly higher temperature pressure ratings compared to PEX. This is attributable to the added strength of the aluminum layer. At 180ºF, ASTM F1281 PEX-AL-PEX tubing is rated for pressures up to 125 psi. At 210ºF, it is rated for pressures up to 115 psi.

The aluminum layer in PEX-AL-PEX tubing also significantly reduces the expansion movement relative to other (all-polymer) tubes. This characteristic makes PEX-AL-PEX well-suited for use in radiant panel construction where aluminum heat transfer plates are used as shown in Figure 7-5.

Figure 7-5

Courtesy of Harvey Youker

A wide variety of fittings are available for connecting PEX-AL-PEX tubing to itself as well as copper tubing and other components.

OTHER POLYMER TUBING:

Two other types of polymer tubing are being increasingly used in modern hydronic systems. They are PE-RT and PP-R.

Figure 7-6

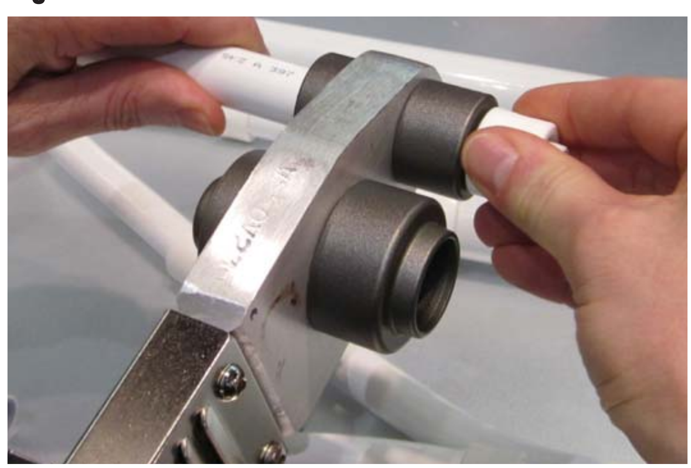


Figure 7-7

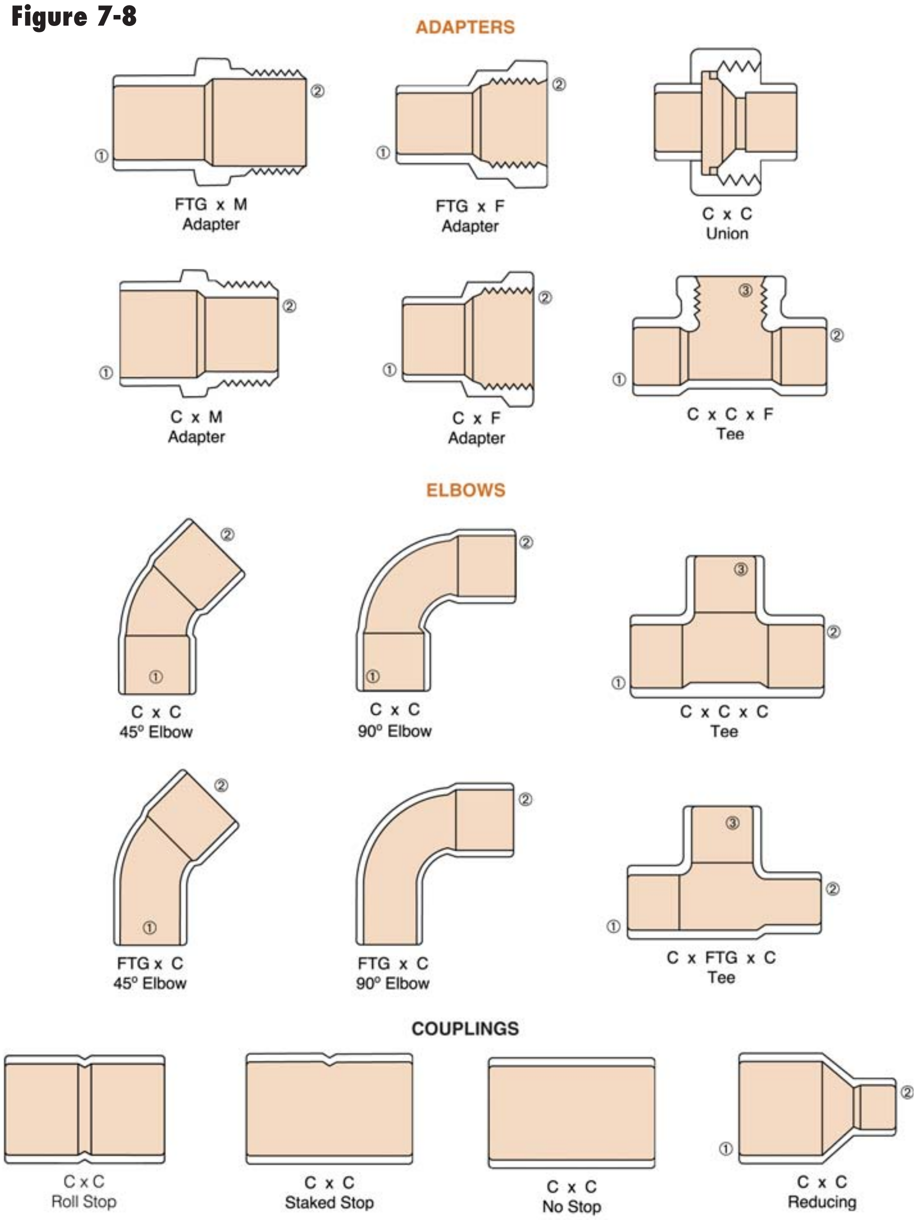
Courtesy of Aquatherm

PE-RT stands for PolyEthylene Raised Temperature. Although relatively new in North America, PE-RT tubing has been used for over 20 years in Europe. It provides pressure/temperature ratings of 200 psi at 73ºF and 100 psi at 180ºF. PE-RT tubing is available with an oxygen diffusion barrier for closed hydronic systems. It is also available with an aluminum core (e.g., PE-RT/AL/PE-RT). Like PEX, it can be joined with mechanical fittings. Because it is not crosslinked, it can also be permanently joined using socket fusion as shown in Figure 7-6.

Another polymer tubing that is relatively new to North American is Reinforced Polypropylene (PP-R). Available in sizes from 3/8” to 10”, PP-R tubing features a fiberglass-reinforced core that, in combination with the polypropylene inner and outer layers, limits thermal expansion and allows sustained operating temperatures up to 160ºF with a corresponding pressure of 70 psi and temporary operating temperatures up to 195ºF for 60 days.

PP-R tubing is joined by either butt fusion or socket fusion (depending on size). Figure 7-7 shows the latter, where the molten end of the tube is about to be pushed into the molten socket of the fitting. A heating fixture similar to that shown in Figure 7-6 is used to simultaneously heat these surfaces on the tube and fitting to approximately 500ºF before joining. The resulting joint is permanent and extremely strong. PP-R tubing is available in both coils and straight lengths. The latter is commonly used in lieu of rigid metal pipe within mechanical rooms and distribution systems for both heating and cooling.

Figure 7-8



GENERAL NOTES: (a) Fittings are designated by size in the order: 1x2x3 (b) Fitting designs and drawings are for illustration only.

Courtesy of the Copper Development Association

PIPE FITTINGS:

There are thousands of pipe fittings of various shapes, sizes and connections that can be used in hydronic systems. Figure 7-8 shows the more common types of fittings, in this case for copper tubing.

Learning to specify the type of fitting needed is an important part of hydronic design and installation. Some of the common terminology, abbreviations and protocols for specifying fittings are as follows:

• Male pipe thread (MPT)
• Female pipe thread (FPT)
• Copper socket (C)
• For reducer fittings (with two different pipe sizes), specify the larger connection first
• For tees, specify the larger run connection, then the other run connection, and finally the side connection

COMMON VALVES:

There are several common valves that are used in hydronic systems. They can be classified based on three basic purposes:

• Component isolation (the valve is typically fully open or fully closed)
• Flow regulation (the valve stem may be at any position as needed to limit flow rate)
• Preventing flow reversal

VALVES FOR COMPONENT ISOLATION:

The two most common valves used for component isolation are gate valves and ball valves. Of these, ball valves have now become more widely used, especially in smaller systems. Figure 7-9 shows a typical gate valve. Figure 7-10 shows a full-port ball valve.

Figure 7-9

Courtesy of NIBCO

Both gate valves and ball valves should be either fully open (when flow is allowed through them) or fully closed (to isolate a component from the remainder of the system). Operating either of these valves in a partially open mode can lead to chattering and eventual erosion of their internal trim or seals.

Figure 7-10

Courtesy of NIBCO

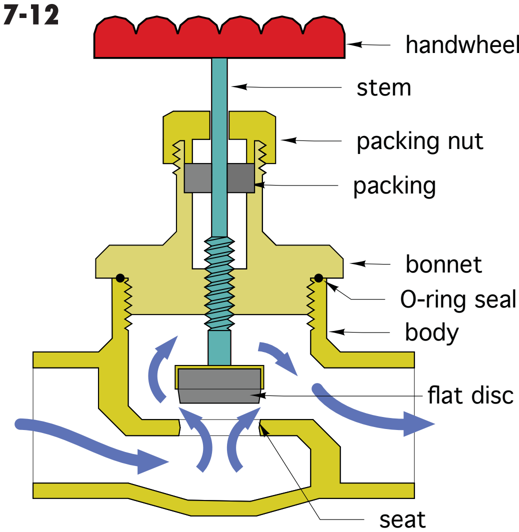
VALVES FOR FLOW REGULATION:

Several types of valves have been developed for flow regulation. The "classic" valve for this application is a globe valve, an example of which is shown in Figure 7-11.

Figure 7-11

Courtesy of NIBCO

Figure 7-12



Courtesy of NIBCO

Globe valves force fluid to pass through a path having several abrupt changes in direction as shown in Figure 7-12. The fluid enters the lower valve chamber, flows upward through the gap between the seat and disc and then exits sideways from the upper chamber. This enables the valve to dissipate head energy from the fluid, and thus create a pressure drop. The stem controls the position of the disc above the seat. The closer the disc is to the seat, the greater the pressure drop across the valve.

Always install globe valves so the fluid flows into the lower body chamber and upward toward the disc. Reverse flow through a globe valve can cause unstable flow regulation, noise and cavitation. All globe valves have an arrow on their body that indicates the proper flow direction through the valve.

Globe valves should never be used for component isolation. Even in their fully open position, globe valves remove considerably more head energy from the fluid compared to a fully open gate valve or fully open ball valve. Therefore, during the thousands of operating hours when component isolation is not needed, the globe valve unnecessarily wastes pumping energy. Using a globe valve for component isolation is like driving a car with the brakes partially applied all the time.

Several variations of the fundamental globe valve have been developed for hydronic systems. They are called balancing valves and are used to regulate how much flow passes through various branches of hydronic systems.

idronics #8 contains a detailed description of balancing valves and shows how they are installed and adjusted.

CHECK VALVES:

The term "check valve" applies to any valve intended to prevent flow reversal. Such valves are commonly used in many types of hydronic systems.

The simplest check valve is known as a *swing check*, an example of which is shown in Figure 7-13.

Swing check valves contain a disc hinged along its upper edge. When fluid moves through the valve in the allowed direction (as indicated by an arrow on the side of the valve's body), the disc swings up into a chamber and out of the fluid stream. The moving fluid holds it there. When flow stops or attempts to reverse itself, the disc swings down due to its weight and seals across the opening of the valve. The greater the back pressure, the tighter the seal.

Figure 7-13

Courtesy of NIBCO

Swing check valves should always be installed in *horizontal* piping with the bonnet of the valve in an upright position. Installation in other orientations can cause erratic operation, which can lead to dangerous water hammer effects.

It is also important to install swing check valves with a minimum of 12 pipe diameters of straight pipe upstream of the valve. This allows turbulence created by upstream components to partially dissipate before the flow enters the valve. Failure to do so can cause the valve's disk to rattle.

Another type of check valve is called a spring check valve. These valves rely on a small internal spring to close the valve's disc whenever fluid is not moving in the intended direction. This allows a spring check valve to be installed in any orientation. The force required to compress the spring does, however, create slightly more pressure drop compared to a swing check. An example of a spring check valve is shown in Figure 7-14.

Figure 7-14

Always install check valves with the arrow on the body pointing in the direction of intended flow. As with swing check valves, install at least 12 diameters of straight pipe upstream of spring-load check valves.

SPECIALTY VALVES:

Several types of valves have been developed for specific applications in hydronic heating and cooling systems. This section provides an overview of these valves. More detailed information on some of these valves can be found in other issues of *idronics*, as well as manufacturer's literature.

FEED WATER VALVE:

A feed water valve, which is also sometimes called a pressure-reducing valve, is used to lower the pressure of water from a domestic water distribution pipe before it enters a hydronic system. This valve is necessary because most buildings have domestic water pressure higher than the rated opening pressure of the relief valves used in hydronic systems. The feed water valve allows water to pass through whenever the pressure at its outlet side drops below its pressure setting. It allows small amounts of water to automatically enter the system as air is vented out. An example of a feed water valve is shown in Figure 7-15.

Figure 7-15

BACKFLOW PREVENTER:

These valves are designed to prevent any water that has entered a hydronic heating system from flowing back out toward the potable water plumbing. This prevents potentially toxic liquids such as certain types of antifreeze from entering potable water piping.

A backflow preventer consists of a pair of check valve assemblies in series, with an intermediate vent port that drains any backflow that migrates between the valve assemblies. An example of a small backflow preventer is shown in Figure 7-16.

A check valve, or even two check valves in series, are *not* acceptable substitutes for a backflow preventer. Such valves do not provide the vent port that would drain any minor amount of fluid passing through the downstream check assembly within a backflow preventer. Always be sure to install the backflow preventer with the arrow pointing into the hydronic system.

Figure 7-16

PRESSURE-RELIEF VALVE:

A pressure-relief valve is a code requirement on any closed-loop hydronic heating system. It functions by opening at a preset pressure rating, allowing fluid to be safely released from the system before higher pressures can develop. The pressure-relief valve is the final means of protection in a situation where all other controls fail to limit heat production.

Most mechanical codes require pressure-relief valves in *any* piping assembly that contains a heat source and is capable of being isolated by valves from the rest of the system. A pressure-relief valve is also required in any piping circuit supplied with heat through a heat exchanger.

Pressure-relief valves contain a disc that is held against its seat by a spring. The spring is calibrated so that when the pressure on the inlet side of the valve reaches a certain level, the spring will allow the disc to lift off its seat and thus discharge fluid from the system. This rated opening pressure, along with the maximum heating capacity of the equipment the valve is rated to protect, is stamped onto a permanent tag or plate attached to the valve, as seen in Figure 7-17.

Figure 7-17

All pressure-relief valves should be installed with their shaft in a vertical position. This minimizes the chance of sediment accumulation around the valve's disc. All pressure-relief valves should also have a waste pipe attached to their outlet port. This pipe routes any expelled fluid safely to a drain, or at least down near floor level. The waste pipe must be the same size as the valve's outlet port with a minimal number of turns and no valves or other means of shutoff.

THERMOSTATIC RADIATOR VALVES:

Thermostatic radiator valves (TRVs) can provide precise room-by-room temperature control in hydronic heating systems. They are installed in the supply pipe of a heat emitter, or in some cases integrated into the heat emitter. TRVs consist of two parts, the valve body and the thermostatic operator, as shown in Figure 7-18.

Figure 7-18

The nickel-plated brass valve body is available in either a straight pattern (as shown in Figure 7-18) or angle pattern. In the latter, the outlet port of the valve is rotated 90 degrees to the inlet port.

Inside the valve is a plug mounted on a spring-loaded shaft. The plug is held in its fully open position by the force of the spring. The fluid pathway through the valve is similar to that of a globe valve, thus making the valve well-suited for flow control. To close the valve, the shaft must be pushed inward against the spring force. No rotation is necessary.

Shaft movement is provided by the thermostatic operator. This operator contains a fluid in a sealed bellows chamber. As the air temperature surrounding the operator increases, the fluid expands inside the bellows, which forces the shaft of the valve inward towards its closed position.

When the valve is mounted in a pipe supplying a heat emitter, this action decreases water flow, and thus reduces heat output. As the room air temperature decreases, the fluid contracts, allowing the spring force to slowly reopen the valve plug and increase heat output from the heat emitter.

The combination of the valve and thermostatic operator represents a fully modulating temperature control system that continually adjusts flow through the heat emitter to maintain a constant (occupant determined) room temperature.

TRVs are often located close to the heat emitter. The preferred mounting position is with the valve stem in a horizontal position with the thermostatic operator facing away from the heat emitter. In no case should the valve be installed so the thermostatic operator is *above* the heat emitter. This will cause the valve to close prematurely and greatly limit heat output from the heat emitter.

Be sure the TRV is mounted with flow passing through it in the direction indicated by the arrow on the valve body.

MIXING VALVES:

Mixing valves are often used in hydronic heating systems to blend hot water from a heat source with cooler water returning from the distribution system. The goal of this blending is to achieve an appropriate supply water temperature to the distribution system, and thus produce the required heat output.

One of the most common types is called a 3-way mixing valve. It has three ports: one that allows hot water from the heat source to enter, another that allows cooler water from the return side of the distribution system to enter and a third port that supplies the mixture of the two entering flows to the distribution system.

Figure 7-19

3-way mixing valves can be controlled by thermostatic actuators or motorized actuators. Figure 7-19 shows a 3-way mixing *thermostatic* mixing valve. The proportions of flow entering the ports of this valve are controlled by a non-electric actuator that is built into the valve.

Figure 7-20 shows a *motorized* 3-way mixing valve. The actuator attached to this valve is operated by a separate electronic controller. The actuator can rotate the stem of the valve to the position needed to achieve the desired supply water temperature to the system.

Figure 7-20

idronics #7 provides a detailed discussion of both thermostatic and motorized mixing valves and shows how each are applied in a variety of hydronic heating applications.

ZONE VALVES:

One of the previously discussed benefits of hydronic heating is the ability to easily configure the system to independently supply heat to multiple zones within a building. Zone valves are often used to allow or prevent flow through piping serving each of these zones.

Figure 7-21

A typical electrically operated zone valve is shown in Figure 7-21. It consists of a valve body combined with an electrically powered actuator. The actuator is the device that produces movement of the valve shaft when supplied with an electrical signal.

Zone valves operate in either a fully open or a fully closed position. They do not modulate to regulate the flow rate in a piping circuit. Those with gear motor actuators, as shown in Figure 7-21, can move from fully closed to fully open in a few seconds.

idronics #5 provides more discussion of zone valves and how to apply them within systems.

8: HEAT EMITTER OPTIONS

There are many types of hydronic heat emitters designed to extract heat from water flowing through them and release that heat into a space. These devices use a combination of convection and thermal radiation to release heat. Some are manufactured devices, while others are site-built and an integral part of the building they heat. This section discusses the most common hydronic heat emitters that are used in modern hydronic systems.

THERMAL EQUILIBRIUM:

Before considering specific types of heat emitters, it's important to understand a general principal that applies to every hydronic systems, regardless of the type of heat emitters used. That principle is call thermal equilibrium.

Consider the hydronic heating system shown in Figure 8-1.

In this system, the heat source is adding heat to the water passing through it 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 given location in the circuit will remain steady.

Figure 8-1

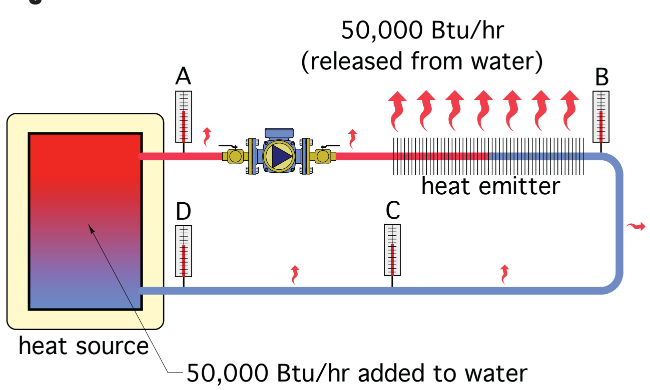
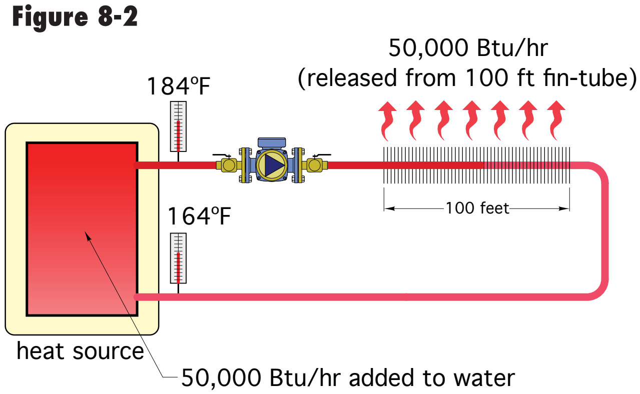


Figure 8-2



The exact value of the water temperature at any of these locations will depend on the heat dissipating characteristics of the distribution system versus the heat generating capability of the heat source. For a given rate of heat generation, 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.

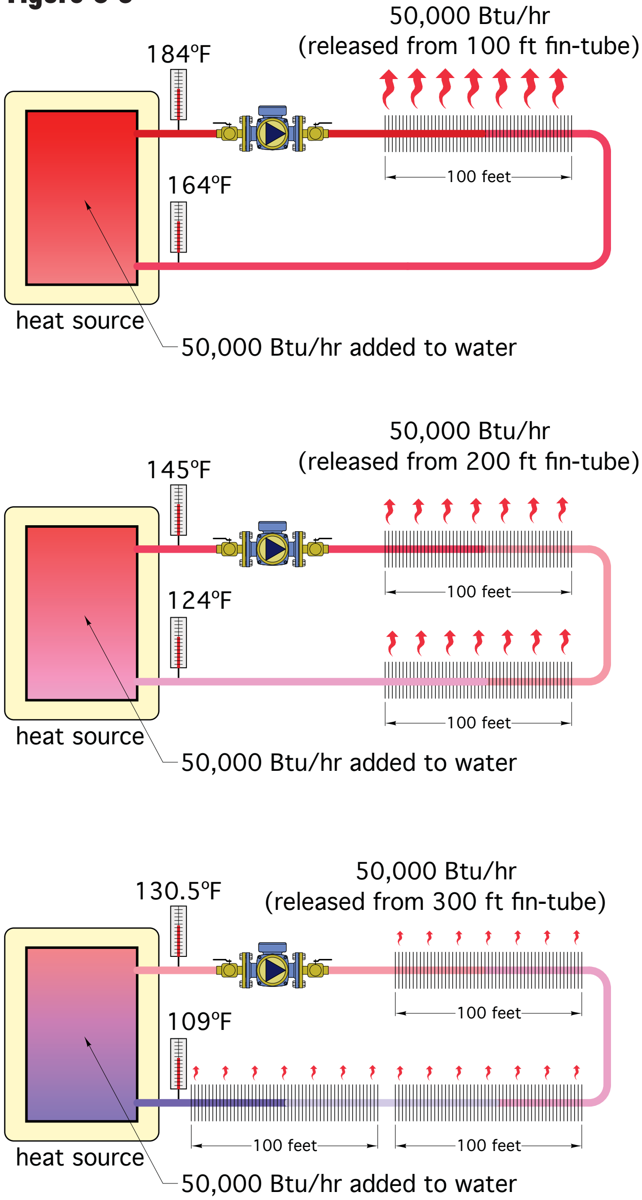
The system shown in Figure 8-2 contains 100 feet of typical residential-style fin-tube baseboard. For simplicity, assume all heat dissipation from the distribution system occurs at this baseboard. Assume the heat source is capable of generating water temperatures as high as 210ºF, and that the high-limit 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 then 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 this temperature reaches 184ºF, there is no further rise, even though the boiler remains in continuous operation.

The explanation for this behavior is that when water at 184ºF is supplied to 100 feet of fin-tube baseboard, it can dissipate 50,000 Btu/hr, exactly the same rate of heat transfer as is occurring at the heat source. There is no reason for the water temperature to climb any higher. The water temperature at all other locations around the circuit will also remain stable at this point (albeit, at lower 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 8-3. At that point, the modified system has again reached thermal equilibrium, where the 50,000 Btu/hr rate of heat input at the heat source is exactly matched by the 50,000 Btu/hr heat dissipation rate from 200 feet

Figure 8-3



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, which 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 capability of the distribution system lowers the supply water temperature at which thermal equilibrium occurs.

In this case, the heat dissipation capability was increased by adding more heat emitter surface area. 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 surface.

Lowering the water temperature at which thermal equilibrium occurs is generally acceptable, and even desirable, because it increases the thermal efficiency of most heat sources. However, "conventional" boilers have 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. This limitation was discussed in more detail in section 6.

idronics

idronics #7 also discusses sustained flue gas condensation within conventional boilers and how to avoid it.

If the heating capacity of the heat source is significantly higher than the heat dissipation ability of the distribution system, the temperature at which thermal equilibrium would occur is often much higher than the setting of the heat source's high-limit controller.

For example, if the baseboard length in the system shown in Figure 8-2 was decreased to 50 feet, the water temperature leaving the heat source when thermal equilibrium was finally 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 heat generation when the outlet temperature reached 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 would drop sufficiently to cause the heat source to turn back on, and the cycle would repeat itself. This is a very common operating mode in many systems during partial load conditions. It can even occur under design load conditions in systems having an oversized heat source.

In summary, every hydronic system "wants" to operate at thermal equilibrium. If not for the intervention of temperature-limiting controllers, the water temperature in any hydronic system will automatically adjust itself as necessary to make this occur. This water temperature may or may not provide the proper heat input to the building. Likewise, it may or may not be conducive to safe and efficient operation or long system life. Bluntly stated: The system “doesn’t care” if it’s delivering the proper amount of heat to the rooms or if it’s operating safely. It only “cares” about achieving a balance between heat input and heat output.

Good designers ensure that the heat source and heat emitters are properly matched so that thermal equilibrium occurs at water temperatures that are safe and allow for efficient operation of the heat source.

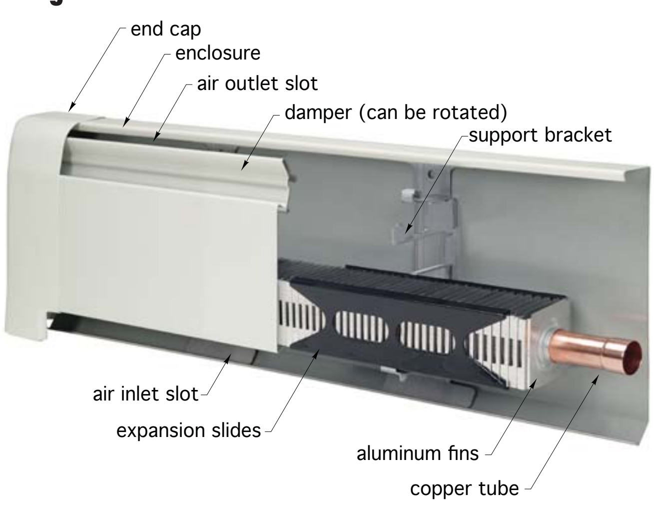
STANDARD FIN-TUBE BASEBOARD:

Perhaps the best known hydronic heat emitter in North America is standard fin-tube baseboard, an example of which is shown in Figure 8-4.

Figure 8-4

Standard fin-tube baseboard was originally developed as an alternative to cast iron radiators. It consists of a copper tube with attached aluminum fins. The purpose of the fins is to extend the surface area of the tube. The high conductivity of the aluminum fins “wicks” heat away from the copper tube, and transfers this heat to the air

Figure 8-5



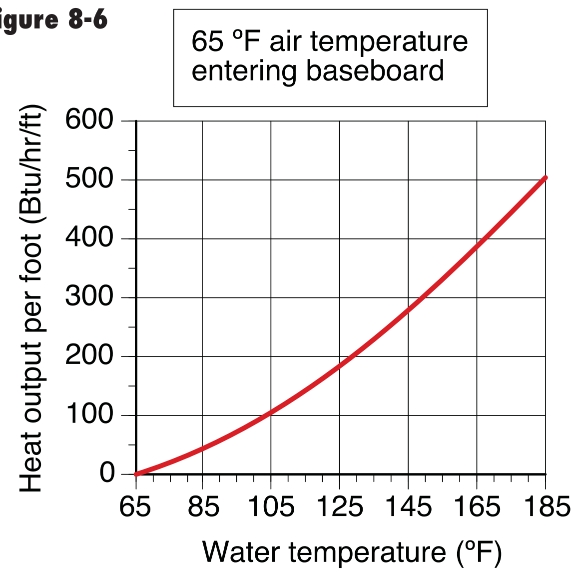
Courtesy of Cengage Learning

surrounding the fins. The warm air rises upward due to its lowered density. Cooler air near floor level flows inward and upward into the fin-tube element. This creates a gentle flow of warm air out of the slot at the top of the baseboard’s steel enclosure. A cut-away view of a typical fin-tube baseboard is shown in Figure 8-5.

Fin-tube baseboard is typically sold in straight lengths ranging from 2 to 8 feet in 1-foot increments. The finned tube element is a few inches shorter than the enclosure. Sheet metal accessories such as end caps, corners and straight enclosure splices are available to connect and trim the straight lengths.

Figure 8-6 shows a typical heat output versus water temperature relationship for residential grade fin-tube baseboard.

Figure 8-6



Designers should keep in mind that standard fin-tube baseboard was designed to operate at relatively high average water temperatures (160ºF to 200ºF). While such temperatures are available from conventional boilers, they are well above what most renewable energy heat sources such as solar thermal collectors and heat pumps can provide. These high water temperatures also lower the efficiency of mod/con boilers.

idronics #8 provides a more detailed discussion along with mathematical models for the performance of fin-tube baseboard over a range of water temperatures, entering air temperatures and flow rates.

LOW-TEMPERATURE FIN-TUBE BASEBOARD:

Low-temperature heat sources will be increasingly common in future hydronic systems. This has led to reconfigurations of traditional products in ways that allow them to operate at lower water temperatures. An example of one such product is the low-temperature fin-tube baseboard, shown in Figure 8-7.

Figure 8-7

Courtesy of Smith's Environmental Products

This baseboard uses two tubes within its element (versus one tube in a standard baseboard element). The fin area is also approximately three times larger than that in standard fin-tube baseboard. This larger element allows a given rate of heat output at a significantly lower average water temperature.

For example, at an *average* water temperature of 110ºF, this baseboard releases about 290 Btu/hr/ft when the two pipes are configured for parallel flow, and the total flow rate through the element is 1 gpm (0.5 gpm through each tube). This increases to about 345 Btu/hr/ft with a total flow rate of 4 gpm (e.g., 2 gpm per tube). Typical residential fin-tube would need an average water temperature of about 152ºF (at 4 gpm flow rate) to yield the same output.

Low-temperature baseboard is well-suited to systems that use renewable heat sources such as solar thermal collectors or heat pumps. It also allows mod/con boilers to operate a very high thermal efficiency.

PANEL RADIATORS:

Modern panel radiators come in a wide range of shapes and sizes. Most are constructed of steel, while some are constructed of aluminum. An example of a typical steel panel radiator is shown in Figure 8-8

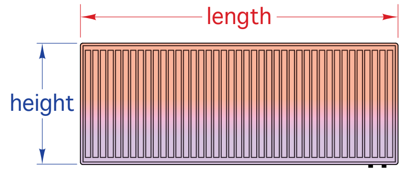
Figure 8-8

Courtesy of Harvey Youker

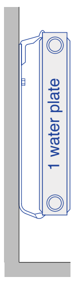
This panel radiator delivers radiant heating from the surfaces exposed to the room, as well as convective heating as room air is heated by fins concealed behind the front panel. Water is supplied to and carried away from the panel through the small-diameter tubes seen at the lower right of the panel. The knob at the upper right corner of the panel is a thermostatic operator that controls flow through a valve built into the radiator. This operator serves as the temperature regulator for the room. If the room's air temperature begins to drop, the operator responds by allowing the valve to open farther, and thus increases the flow of heated water through the radiator. If the room's temperature begins to rise above the desired setting, the operator causes the valve to reduce the flow of heated water through the panel. These actions are entirely controlled by the expansion and contraction of a fluid within the operator. These actuators requires no electrical power. A system that supplies several panel radiators, each equipped with thermostatic radiator valves, allows for room-by-room temperature control.

Panel radiators are available in different heights, widths and thicknesses. The larger the surface area of the panel radiator, the higher its heat output will be for a given average water temperature and room air temperature. Panel thickness is increased by adding water plates to the radiator. The radiator shown in Figure 8-8 has a single water plate (e.g., the front of the panel with the vertical water channels running between the upper and lower headers). A double water plate radiator has two water plates arranged back-to-back. A three-water plate panel adds one more water plate to the back of a double water plate panel. Multiple-water plate radiators are thicker, but this may be an acceptable tradeoff to keep the height of the panel low or reduce the required length of the panel. By using combinations of height, width and thickness, designers are able to specify radiators that can fit within locations with limited wall space, yet still provide adequate heat output.

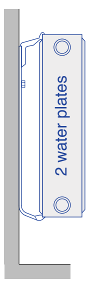
Figure 8-9



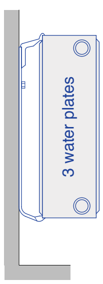
Heat output ratings (Btu/hr)
at reference conditions:
Average water temperature in panel = 180ºF
Room temperature = 68ºF
temperature drop across panel = 20ºF



	1 water plate panel thickness					
	16" long	24" long	36" long	48" long	64" long	72" long
24" high	1870	2817	4222	5630	7509	8447
20" high	1607	2421	3632	4842	6455	7260
16" high	1352	2032	3046	4060	5415	6091



	2 water plate panel thickness					
	16" long	24" long	36" long	48" long	64" long	72" long
24" high	3153	4750	7127	9500	12668	14254
20" high	2733	4123	6186	8245	10994	12368
16" high	2301	3455	5180	6907	9212	10363
10" high	1491	2247	3373	4498	5995	6745



	3 water plate panel thickness					
	16" long	24" long	36" long	48" long	64" long	72" long
24" high	4531	6830	10247	13664	18216	20494
20" high	3934	5937	9586	11870	15829	17807
16" high	3320	4978	7469	9957	13277	14938
10" high	2191	3304	4958	6609	8811	9913

Manufacturers provide heat output ratings for their panel radiators in either graphical or tabular form. In many cases, "reference" heat output ratings are stated along with corresponding average water temperature and room air temperatures.

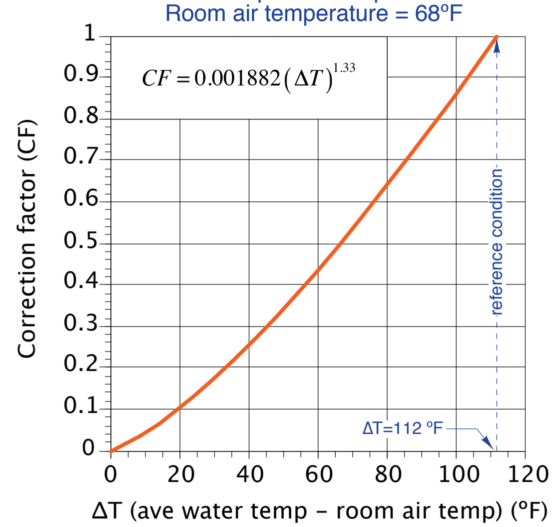
Figure 8-9 shows a heat output table for steel panel radiators of several different heights, widths and thicknesses.

The heat outputs shown in Figure 8-9 are referenced to an average water temperature of 180ºF and a room air temperature of 68ºF. If different operating conditions are expected or desired, it's necessary to multiply these reference heat output ratings by a correction factor. That factor can be determined from either the graph or formula shown in Figure 8-10.

Example: Find the output of a 24" high by 72" long single water plate panel if operated at an average water temperature of 110ºF in a room maintained at 68ºF.

Figure 8-10

Reference condition:
Ave water temperature in panel = 180ºF
Room air temperature = 68ºF



Solution: Figure 8-9 indicates that a panel with a single water plate measuring 24” high and 72” long has a heat output of 8,447 Btu/hr based on the reference conditions of 180ºF average water temperature and 68ºF room air temperature. Using the formula in Figure 8-10, the correction factor with an average panel water temperature of 110ºF and room temperature of 68ºF is:

$$CF = 0.001882(110-68)^{1.33} = 0.271$$

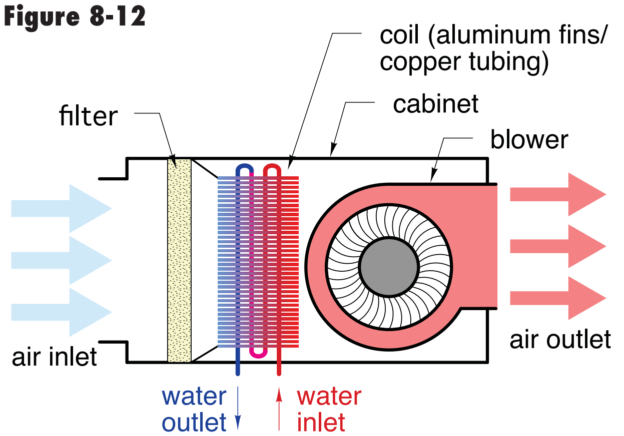
The estimated heat output at the lower water temperature is therefore:

$$Output = (0.271) \times 8447 = 2289\,Btu/hr$$

HYDRONIC FAN-COILS & AIR HANDLERS:

Another method of delivering heat to a building is through use of hydronic fan-coils and air handlers. These heat emitters receive heat from a hydronic distribution system, but deliver that heat to the room using forced air. A blower or fan within the unit creates the airflow. The forced air stream moves across a “coil” consisting of copper tubing with closely spaced aluminum fins. Figure 8-12 shows the typical construction of an air handler.

Figure 8-12



Although both fan-coils and air handlers share common characteristics, the term fan-coil usually describes a smaller heat emitter that is mounted to the surface of a wall or recessed into a wall cavity, as shown in Figure 8-13. Fan-coils typically deliver heat to a single room.

The term air handler describes a larger device that is usually designed for mounting in concealed spaces. Air handlers have larger coils and blowers contained within a vertical or horizontal sheet metal cabinet. They deliver an air stream to ducting that often distributes that air flow to more than one room. An example of a horizontal air handler is shown in Figure 8-14.

Figure 8-13

Courtesy of Myson, Inc.

Some air handlers are also capable of delivering cooling when their coil is supplied with chilled water. This type of air handler must be equipped with a condensate drip pan located under its coil. The drip pan collects water droplets that form on the coil as it dehumidifies the warm/moist air passing through it. The drip pan connects to a pipe that routes the condensate to a suitable drain.

Figure 8-14

Courtesy of Mestek

The thermal performance of hydronic fan coils is typically listed in a table for each specific model. The parameters that determine thermal performance include:

- Entering water temperature
- Entering air temperature
- Water flow rate through coil
- Air flow rate across coil

The following general guidelines hold true for more hydronic fan-coils and air handlers:

1. The heat output of a fan-coil or air handler is proportional to the temperature difference between the entering air and entering water.

Thus, if the heat output of a particular air handler was 10,000 Btu/hr when the entering air temperature was 65ºF and the entering water temperature was 140ºF, its output when supplied with 180ºF water could be estimated by the following proportion:

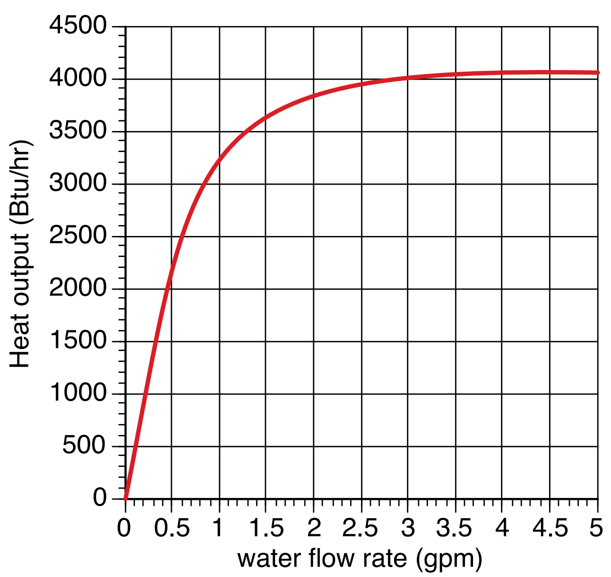
$$\text{estimated heat output} = 10{,}000 \times \left(\frac{180-65}{140-65}\right) = 15{,}333 \text{ Btu/hr}$$

2. Increasing either the water flow rate or air flow rate through the coil beyond the minimums stated by the manufacturer will marginally increase heat output.

The effect of changing the water flow through the coil of a small fan-coil unit above or below the heat output rating established at a flow rate of 1 gpm is shown in Figure 8-15. At flow rates above 1 gpm the increase in heat output is slight. However, at flow rates below the rated 1 gpm, heat output decreases quickly.

3. Air handlers or fan-coils with larger coils surfaces and/ or multiple tube passes through the coil are capable of delivering a rated heat output at lower entering water temperatures.

Figure 8-15



This relationship suggests that larger coils are preferred when the heat source is a mod/con boiler, solar thermal collector or heat pump. Fan coil units that operate at lower supply water temperatures should be located so that their discharge air stream does not directly impinge on room occupants. Although the temperature of the discharge air is warmer than the room air temperature, its velocity tends to make it feel cooler and may induce drafts of room air which cause discomfort. Registers or ceiling diffusers supplied by air handlers operating with lower temperature water should be selected and placed so that the air they discharge is well-mixed with room air.

RADIANT PANEL HEATING:

Radiant heating is the process of transferring thermal energy from one object to another by thermal radiation. Whenever two surfaces are "within sight of each other," and are at different temperatures, thermal radiation travels from the warmer surface to the cooler surface. All thermal radiation emitted by surfaces at temperatures lower than approximately 970ºF will be in the infrared portion of the electromagnetic spectrum, and as such cannot be seen by the human eye. All radiant panels discussed in this and other issues of *idronics* operate well below this temperature.

The thermal radiation emitted by a heated surface passes through room air with virtual no absorption of energy. Instead, the radiation is absorbed by the objects in the room. Most interior surfaces absorb the majority of any thermal radiation that strikes them. The small remaining portion is reflected, often to another surface where further absorption takes place.

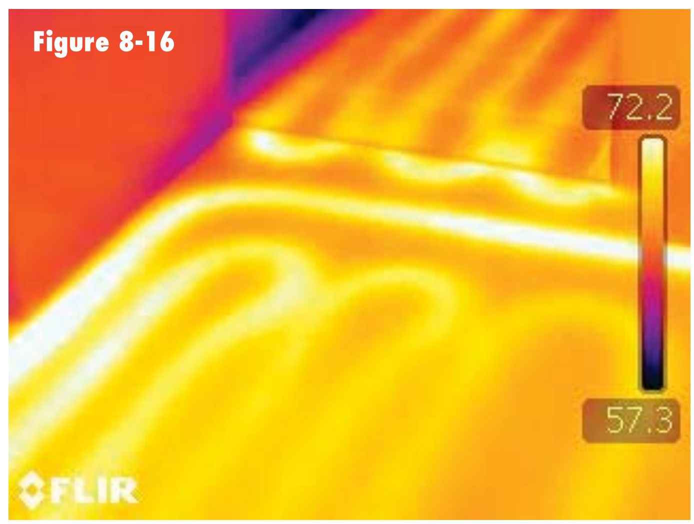


Figure 8-16

It is thus accurate to state that thermal radiation (more commonly referred to as "radiant heat") warms the objects in a room rather than directly heating the air. This difference is largely what separates radiant heating from convective heating. Because the absorbed thermal radiation raises the surface temperature of an object above the room air temperature, some heat will in turn be convected to room air.

Although human eyes cannot see thermal radiation, it can be detected by specialized thermographic cameras. These devices determine the wavelength of thermal radiation (and thus the surface temperature) being detected by each pixel of their imaging system. They then associate a color with this temperature and display a thermographic image along with the associated color/temperature scale. Figure 8-16 shows a thermographic image of a heated floor slab with tubing embedded about 2 inches below its surface.

It is important to understand that the thermal radiation emitter by a typical radiant panel is in no way unhealthy or harmful. Thermal radiation is constantly emitted from our skin and clothing surfaces to any cooler surfaces around us. This radiant heat emission is one of the ways our bodies dissipate the heat produced by metabolism.

The term hydronic radiant panel applies to any room surface (floor, wall or ceiling) that is heated or cooled by embedded tubing, and which has at least 50% of its heat output by thermal radiation. This section briefly discusses the more common types of hydronic radiant panels used for heating.

HEATED FLOOR SLABS:
Concrete slab floors with embedded tubing are by far the most common type of hydronic radiant panel. They are also one of the most economical, because the concrete, and often the underslab insulation, is already specified as part of the building.

Figure 8-18

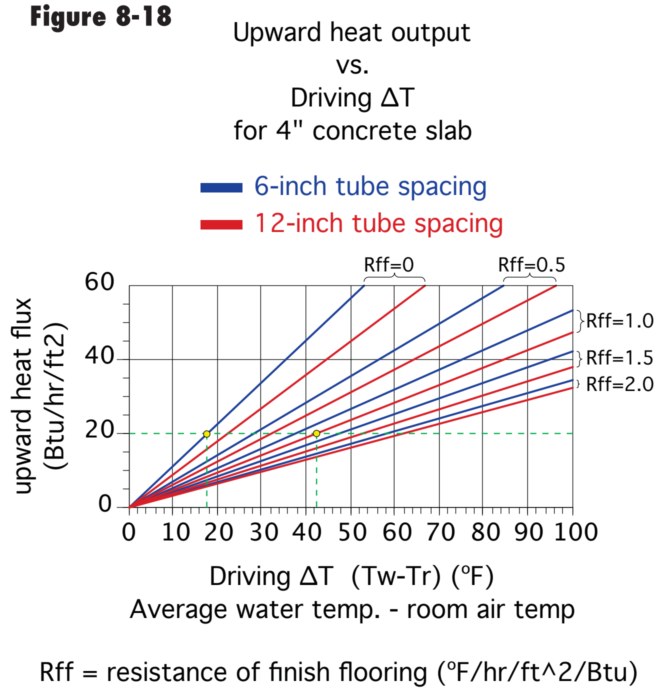


Figure 8-17

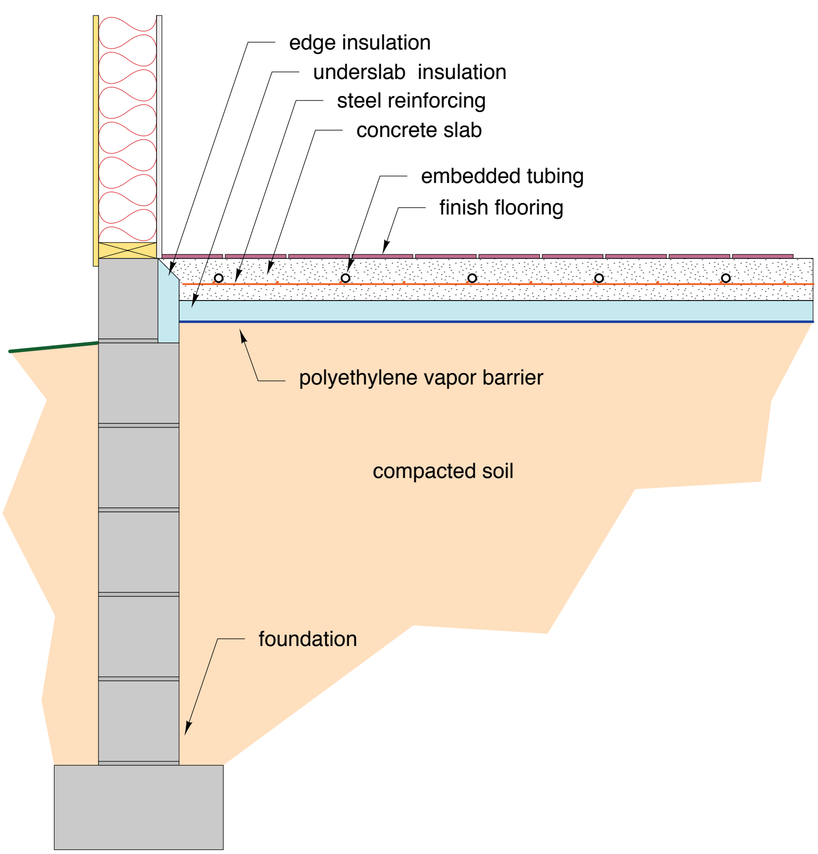


Figure 8-17 shows a cross-section for a heated floor slab. Notice that the tubing has been placed at approximately mid-depth within the slab, and that the underside and edge of the slab are well-insulated (R-10 °F•hr•ft^2/Btu minimum). Both of these details are imperative in achieving good low-temperature performance.

The graph in Figure 8-18 shows upward heat output from a heated slab based on tube spacing of 6”and 12”, and for finish floor resistances ranging from 0 to 2.0 (°F•hr•ft^2/Btu). The steeper the line, the lower the water temperature required for a specific rate of heat delivery.

For example, achieving an upward heat output of 20 Btu/hr/ft^2 from a slab with no covering (e.g., Rff = 0) and 6” tube spacing requires the “driving ΔT” (e.g., the difference between average water temperature in tubing and room air temperature) to be 17.5°F. Thus, in a room maintained at 70°F, the average water temperature in the circuit needs to be 87.5°F. The supply water temperature to the circuit would likely be in the range of 95 °F to 98°F. This is a relatively low supply water temperature and would allow heat sources such as mod/con boilers, solar thermal collectors and heat pumps to operate at high thermal efficiency.

For comparison, consider supplying the same 20 Btu/hr/ft^2 load using a heated floor slab with 12" tube spacing and a finish floor resistance of 1.0ºF•hr•ft^2/Btu. The driving ΔT must now be 42.5ºF. The average circuit water temperature required to maintain a room temperature of 70ºF would be 70 + 42.5 = 112.5ºF, and the supply temperature would be likely in the range of 120ºF to 123ºF. This higher temperature could be produced by conventional boilers but would lower the thermal efficiency of mod/con boilers, as well as solar collectors and heat pumps.

The following guidelines are suggested in applications where a heated floor slab will be used to deliver heat derived from a mod/con boiler or renewable energy heat source:

• Tube spacing within the slab should not exceed 12".

• Slab should have minimum of R-10 underside and edge insulation.

• Tubing should be placed at approximately half the slab depth below the surface, as shown in Figure 8-17. Doing so decreases the required water temperature required for a given rate of heat output. Lower water temperatures improve heat source efficiency.

• Bare, painted or stained slab surfaces are ideal because the finish floor resistance is essentially zero.

• Other floor finishes should have a total R-value of 1.0 or less.

Figure 8-19

Courtesy of Harvey Youker

HEATED THIN SLABS:

Another common method of installing floor heating uses a "thin slab" (1-1/2" to 2" thickness) poured over a wooden floor deck.

Figure 8-19 shows an example of such an installation awaiting placement of the concrete.

A concrete thin-slab installation begins by marking the location of all interior and exterior walls on the completed floor deck. The deck is then covered with 6-mill translucent polyethylene sheeting. This acts as a bond breaker layer between the wooden floor deck, and the concrete slab. It reduces shrinkage cracking of the concrete. The next step is to install 2x4 or 2x6 lumber where ever there is an interior partition or exterior wall. This lumber establishes the thickness of the thin slab. The tubing is then fastened in place using a specialized pneumatic stapler. Once all tubing circuits are installed and connected to their associated manifold, they are pressure-tested with compressed air to verify that there are no leaks. Control joint strips are added where required. These details have all been completed in the photo shown in Figure 8-19. The final step is to place the concrete and screed it level with the top of the 2x4 or 2x6 lumber. The concrete is then troweled as with any other slab. Framing can usually resume the following day.

Thin-slabs can also be constructed using poured gypsum underlayment rather than concrete. The installation sequence differs from that used for a thin concrete slab. First, there is no bond breaker layer required. Second, the slab is typically poured after all walls have been installed and finished. The poured gypsum underlayment is mixed outside the building and pumped in through a hose. This material has a fluid consistency similar to pancake batter. As such it provides a high degree of self-leveling as it is poured on the floor, as seen in Figure 8-20.

Figure 8-20

Poured gypsum underlayment is slightly lighter than concrete. When installed, it often brings hundreds of gallons of water into the building. Much of this water has to evaporate as the slab hardens. Buildings should be well-ventilated to avoid condensation on windows and other surfaces. Also keep in mind that poured gypsum underlayments are not waterproof, nor are they intended to serve as a finish floor surface. After curing, they must be protected against excessive water, and they must be covered with a finish floor surface.

Because thin-slabs are thinner than a typical slab-on-grade floor, they have slightly lower heat dispersal characteristics. This means that slightly higher average water temperatures are required for a given rate of heat output relative to that required for a slab-on-grade. This difference is slight. A 1-1/2”-thick concrete thin slab with 12” tube spacing and covered with a finish flooring resistance of 0.5ºF•hr•ft^2/Btu yields about 8% lower heat output than a 4”-thick slab with the same tube spacing and finish flooring. This can be easily compensated for by using 9” rather than 12” tube spacing.

The following guidelines are suggested for thin-slab installations:

- Tube spacing within the thin-slab should not exceed 9”.
- Slab should have minimum of R-19 underside insulation.
- Floor finishes should have a total R-value of 1.0 or less.
- Never use “lightweight” concrete for heated thin-slabs.

ABOVE FLOOR TUBE & PLATE RADIANT PANELS:

Although heated concrete floor slabs and thin-slabs make excellent radiant panels, they are not suitable for all installations. For example, in some cases, the floor structure cannot support the added dead loading of a thin-slab. In other cases, a nailed down hardwood finish floor cannot be properly fastened to a slab.

In such situations, designers have the option of using a tube & plate floor panel. These panels use aluminum plates as “wicks” to pull heat away from PEX or PEX-AL-PEX tubing and disperse it across much of the floor area. The thin layer of highly conductive aluminum becomes the substitute for a much thicker and heavier layer of concrete.

Figure 8-21 shows an example of an above floor tube & plate radiant panel during construction.

Figure 8-21

Courtesy of Harvey Youker

The installed tubing and aluminum plates will eventually be covered by a nailed-down hardwood finish floor. Adequate insulation must be installed under all above floor tube & plate radiant panels. If the space under the floor deck is heated or semi-heated, the insulation should have a minimum R-value of 19 ºF•hr•ft^2/Btu. If the space under the floor is unheated, the insulation should have a minimum R-value of 30 ºF•hr•ft^2/Btu.

Above floor tube & plate panels generally require higher water temperatures compared to slab systems for comparable heat output. Depending on the thermal resistance of the finish floor, average water temperatures of 120ºF to 140ºF are common under design load conditions.

BELOW FLOOR TUBE & PLATE RADIANT PANELS:

Tube & plate systems can also be installed on the *bottom* of a wood subfloor. Such a system is appropriately called a below floor tube & plate system. Figure 8-22 shows an example of such an installation, before installing the underfloor insulation.

Figure 8-22

Courtesy of Harvey Youker

Figure 8-23

Designers considering use of below floor tube & plate systems must consider possible interference with any existing or planned wiring, plumbing, ducting or structural details that may occupy the spaces between joists. In new installations, it is very important to have the tube and plates, and preferably the insulation, installed before other services that use the joist spaces are installed. Because of interference with wiring and plumbing, below floor tube & plate systems can be difficult to install in retrofit applications.

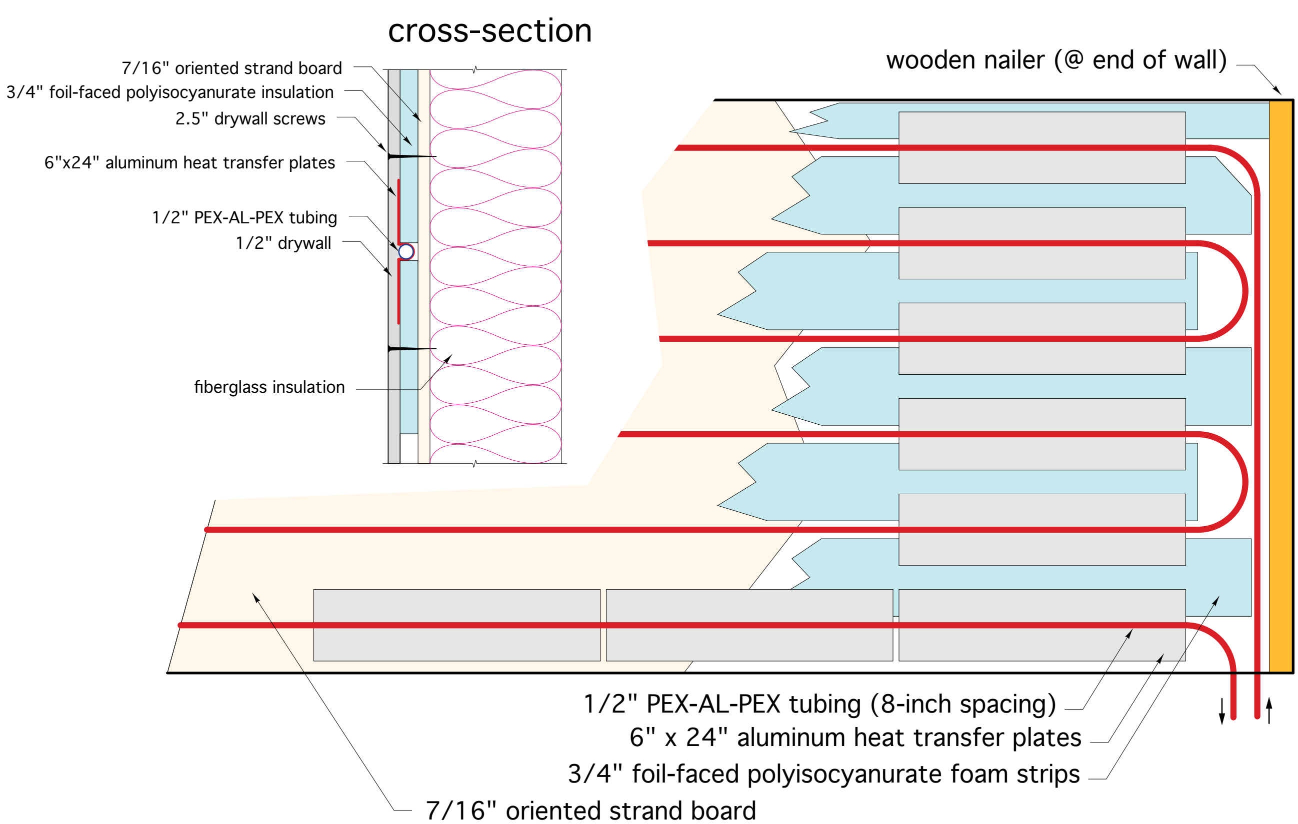
Below floor tube & plate systems must drive heat upward through the subfloor, as well as any finish flooring. This usually requires higher water temperatures relative to below floor tube & plates systems. Average water temperatures of 130ºF to 150ºF are common depending on the required rate of upward heat transfer and the thermal resistance of the finish flooring.

RADIANT WALL PANELS:

Radiant panels can also be integrated into walls and ceilings. In some cases, wall or ceiling panels are preferable over floor panels. Examples include situations where finish flooring with high thermal resistance will be used or when a large percentage of the floor area will be covered with objects that inhibit upward heat flow.

Figure 8-23 shows a radiant wall panel being constructed using ½" PEX-AL-PEX tubing and the same type of aluminum heat dispersion plates that are used for floor-based tube & plate systems. The construction details for this radiant wall panel are shown in Figure 8-24.

Figure 8-24



When finished, this radiant wall panel is indistinguishable from a standard interior wall. Its low thermal mass allows it to respond quickly to changing internal load conditions or zone setback schedules. The rate of heat emission to the room is approximately 0.8 Btu/hr/ft^2 for each degree Fahrenheit the average water temperature in the tubing exceeds room air temperature. Thus, if the wall operates with an average water temperature of 110ºF in a room with 70ºF air temperature, each square foot of wall would release about 0.8 x (110 - 70) = 32 Btu/hr/ft^2. This performance makes it well-suited for use with low-temperature hydronic heat sources.

Figure 8-25

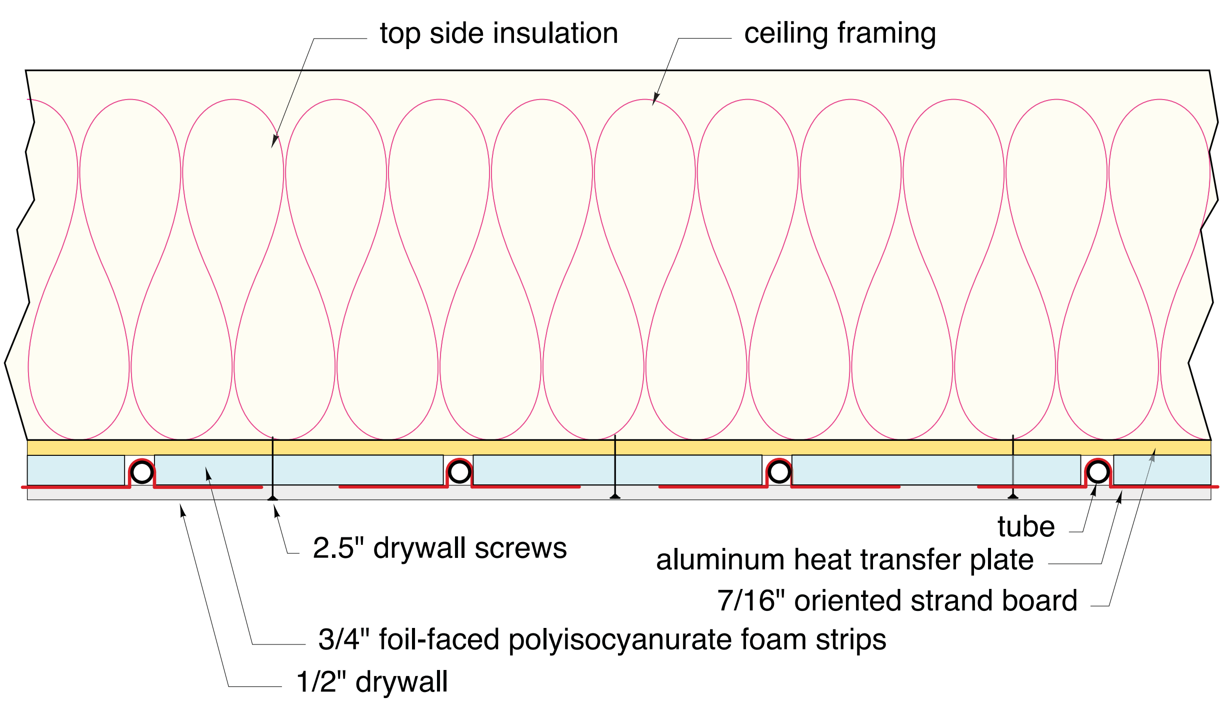
RADIANT CEILING PANELS:

Another possibility is a radiant ceiling using the same type of construction as the radiant wall. Figure 8-25 shows the radiant ceiling panel during installation. Figure 8-26 shows the construction details.

As with the radiant wall, this radiant ceiling has low thermal mass and responds quickly to interior temperature changes. Heated ceilings also have the advantage of not being covered or blocked by coverings or furniture, and thus are likely to retain good performance over the life of the building.

For the construction shown in Figure 8-26, the rate of heat emission is approximately 0.71 Btu/hr/ft^2 for each degree Fahrenheit the average water temperature in the tubing exceeds room air temperature. Thus, if the ceiling operated with an average water temperature of 110ºF in a room with 70ºF air temperature, each square foot of wall would release about 0.71 x (110 - 70) = 28.4 Btu/hr/ft^2. This performance makes the radiant ceiling well-suited for use with low-temperature heat sources.

Figure 8-26



9. HYDRONIC DISTRIBUTION SYSTEMS

A hydronic distribution system is an arrangement of pipes, valves and one or more circulators that conveys heat from the heat source and delivers it to the heat emitters. Distribution systems can be specifically tailored to the needs of each building, taking into account factors such as the type of heat sources used, the kind(s) of heat emitters used, how the building will be zoned and the ease (or lack thereof) of installing piping within a building.

TRADITIONAL HYDRONIC DISTRIBUTION SYSTEMS:

The majority of hydronic distribution systems used in North America over the last several decades were designed around rigid metal pipe. This section describes these traditional designs and discusses their individual strengths and limitations. It then moves on to discuss more contemporary distribution systems that make use of flexible tubing and modern methods of zoning.

SERIES LOOP SYSTEMS:

One of the simplest distribution systems is as a series loop. Flow passes from the heat source, through each heat emitter and back to the heat source. Figure 9-1 shows an example of a series loop system supplying several fin-tube baseboard heat emitters.

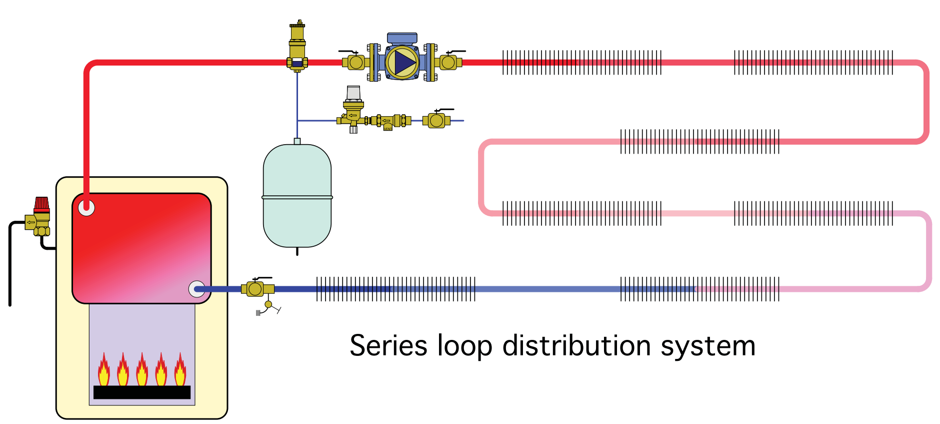
The following points summarize single series loop distribution systems:

• The temperature of the water decreases as flow passes from one heat emitter to the next. Proper design requires that each heat emitter be sized for the water temperature *at its location within the circuit.* If, instead, each heat emitter is sized for the average circuit water temperature, those near the beginning of the circuit will be oversized, and those near the end of the circuit will be undersized. In such cases, complaints of underheating from the heat emitters near the end of the circuit are likely. The longer the series circuit, the more pronounced this effect becomes.

• The total head loss of a series circuit is the sum of the head loss of each component in that circuit. The circulator must supply this total head loss as the design flow rate. Designers must avoid creating series circuits with excessively high head loss. Such a situation can occur if a component with relatively high flow resistance is included in the circuit, especially if combined with several heat emitters that also create significant head loss. If the circuit's head loss is high, it may become necessary to use a "high head" circulator. Such circulators often draw two to three times more wattage than standard circulators, which results in significantly higher operating cost. Insufficient flow through a high-head-loss series circuit can also result in reduced heat output.

• All heat emitters in the series loop provide simultaneous heat output. It is not possible to zone on a room-by-room basis using a series loop. Changing the flow rate or supply water temperature affects the heat output of all heat emitters on the circuit. For this reason, all rooms served by a series circuit should be considered as a single zone. If some rooms experience significant internal heat gains and others do not, a series circuit should not be used. In general, series circuits should be limited to relatively open floor plans where all areas are to be maintained at the same comfort level.

Figure 9-1



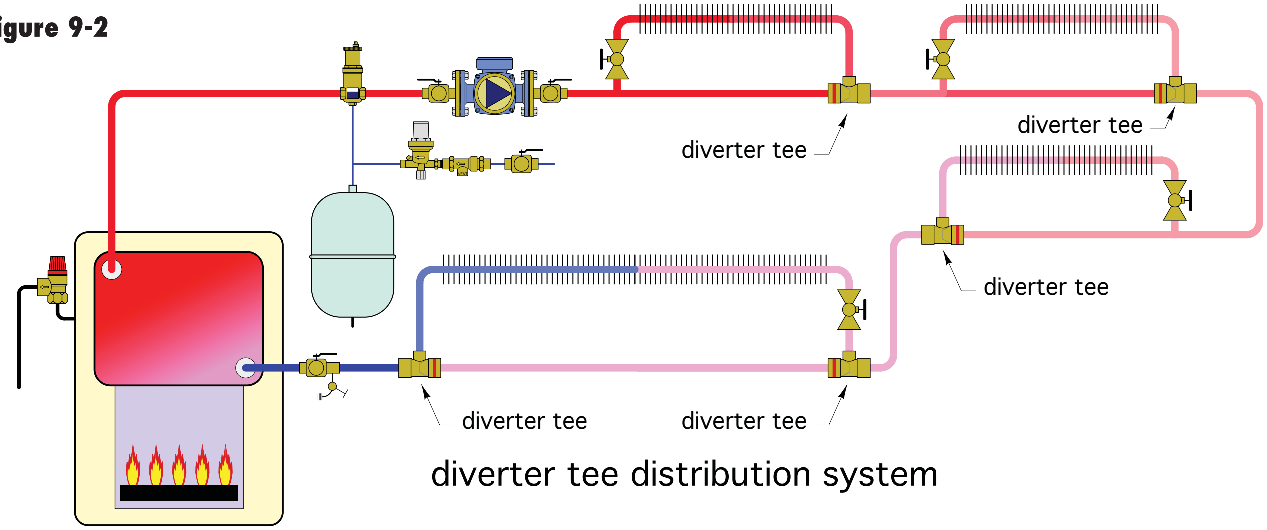
Series loop distribution system

DIVERTER TEE SYSTEMS:

Similar in some respects to series loops, diverter tee distribution systems use a special fitting called a diverter tee to route some of the flow in the main piping circuit through a branch path containing one or more heat emitters. Figure 9-2 shows an example of such a system.

A valve is often installed in each branch to regulate flow through the heat emitter(s) in that branch. This valve may be manually operated, electrically operated or a thermostatic radiator valve. By controlling the flow in each branch, it's possible to provide room-by-room comfort control (provided there is always flow in the main circuit whenever any branch requires heating).

Figure 9-2



diverter tee distribution system

One diverter tee, installed on the return side of the branch, usually provides sufficient flow when that branch is located *above* the main pipe. If the branch circuit is long, contains several heat emitters or is located *below* the main pipe, two diverter tees, one "pushing" and the other "pulling," are generally recommended. In either case, there should be a minimum of one foot of straight pipe between the upstream and downstream tees to minimize turbulence. Always be sure the diverter tees are installed in the proper direction.

Diverter tee distribution systems have the following characteristics:

• As with a series loop, water temperature decreases in the direction of flow along the main pipe. This circuit temperature drop depends on which branches are active at any given time. Designers should size heat emitters assuming all branches are operating simultaneously.

• Each diverter tee adds significant flow resistance to the main circuit. It is not uncommon for a diverter tee fitting to have an equivalent length of at least 25 feet of straight tubing. Adding several diverter tees to a circuit significantly increases the head loss of that circuit, and thus the head required from the circulator to achieve a given flow rate. Designers should carefully assess this effect when designing the circuit and selecting a circulator.

• The only significant advantage a one-pipe diverter tee distribution system has over a series loop system is when valves are used to control flow through each branch. This advantage comes at an associated higher installation and operating cost. Other types of distribution systems are better suited to control the flow through each heat emitter.

"2-PIPE" DIRECT RETURN SYSTEMS:

Hydronic heat emitters can also be piped in a parallel rather than series. One approach puts each heat emitter in its own "crossover" pipe, which connects to a common supply and return main, as shown in Figure 9-3. This configuration is called a "2-pipe" direct return distribution system.

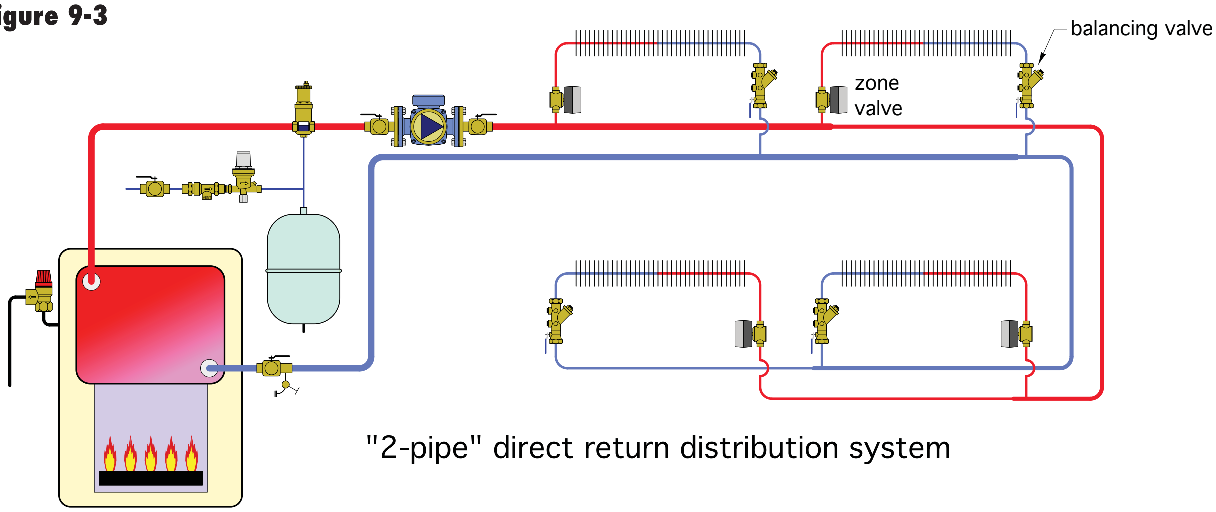
Two-pipe direct return systems have the following characteristics:

• Each heat emitter receives fluid at approximately the same temperature. This simplifies sizing, since the water temperature drop associated with series loops and diverter tee systems is eliminated.

• The heat emitter closest to the circulator on the supply main is also closest to the circulator on the return main. The next heat emitter connected to the supply main has a greater length of piping between it and the circulator. The heat emitter farthest away from the circulator has the longest overall piping path length. If each crossover in the system has the same flow resistance, the highest flow rate will be through the shortest piping path. If uncorrected, this situation reduces the flow rate through heat emitters located farther away from the circulator. Balancing valves installed in each crossover allow flow rates to be adjusted in proportion to the required heat output.

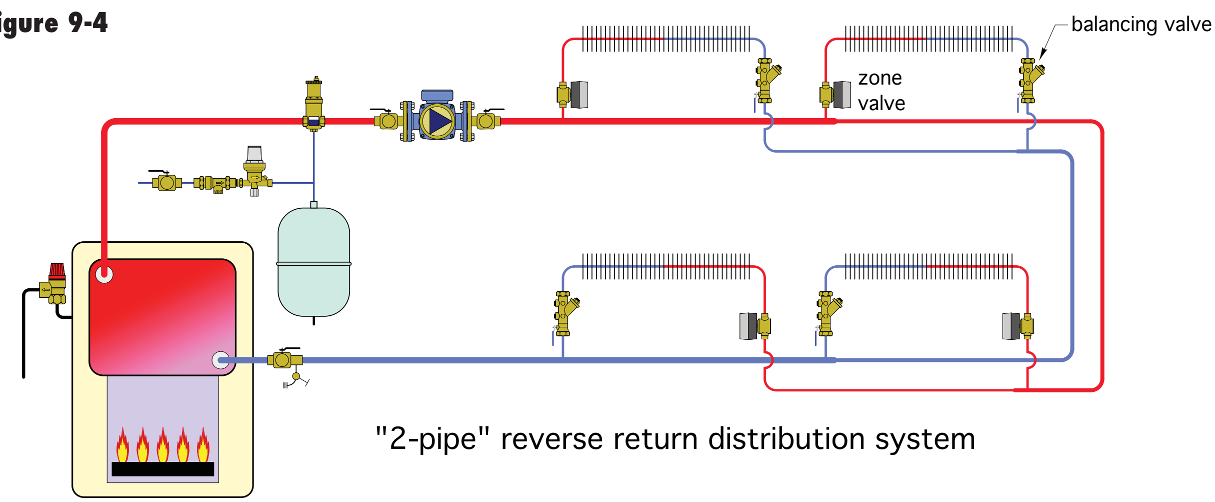
• Flow through each crossover can be controlled using either an on/off zone valve or a modulating flow control valve. This allows each heat emitter to be independently controlled, and thus room-by-room zoning is possible when the system is properly configured.

Figure 9-3



"2-pipe" direct return distribution system

Figure 9-4



"2-pipe" reverse return distribution system

• Different types of heat emitters can be used on the various crossovers provided they are all sized for the same supply water temperature.

"2-PIPE" REVERSE RETURN SYSTEMS:

If the piping design used for the direct return system of Figure 9-3 is modified as shown in Figure 9-4, the result is called a "2-pipe" reverse return system.

Notice that the heat emitter closest to the circulator along the supply main is now farthest from the circulator on the return main. This makes the length of the piping path to and from the circulator approximately the same for each crossover.

If each crossover has the same flow resistance, and if the supply/return mains are sized for approximately the same head loss per unit of length at all locations in the system, this configuration is largely self-balancing. However, balancing valves should still be installed , because the flow resistance through each heat emitter and its associated crossover will not always be the same.

Reverse return systems are well-suited to applications where the supply and return mains run side by side and make a loop from the mechanical room around the building and back to the mechanical room.

2-pipe reverse return systems have the following characteristics:

• Each heat emitter receives water at approximately the same temperature. This simplifies heat emitter sizing because the water temperature drop associated with series loops and diverter tee systems is eliminated.

• Flow through each crossover can be controlled using either an on/off zone valve or a modulating flow control valve. This allows each heat emitter to be independently controlled, and thus room-by-room comfort control is possible when the system is properly configured.

• Different types of heat emitters can be used on the various crossovers provided they are all sized for the same supply water temperature.

• 2-pipe reverse return systems are often favored over 2-pipe direct-return systems because they are closer to self-balancing.

ZONING USING CIRCULATORS:

A traditional North American approach for creating multi-zone hydronic systems uses a separate circulator for each zone circuit, as shown in Figure 9-5.

When a zone needs heat, the associated circulator and the boiler are turned on. When a zone thermostat is satisfied, its circulator is turned off. When all the thermostats are satisfied, the boiler is turned off. All zone circuits begin and end at headers. These headers should be sized for a maximum flow velocity of 2 feet per second to provide good hydraulic separation between the circulators.

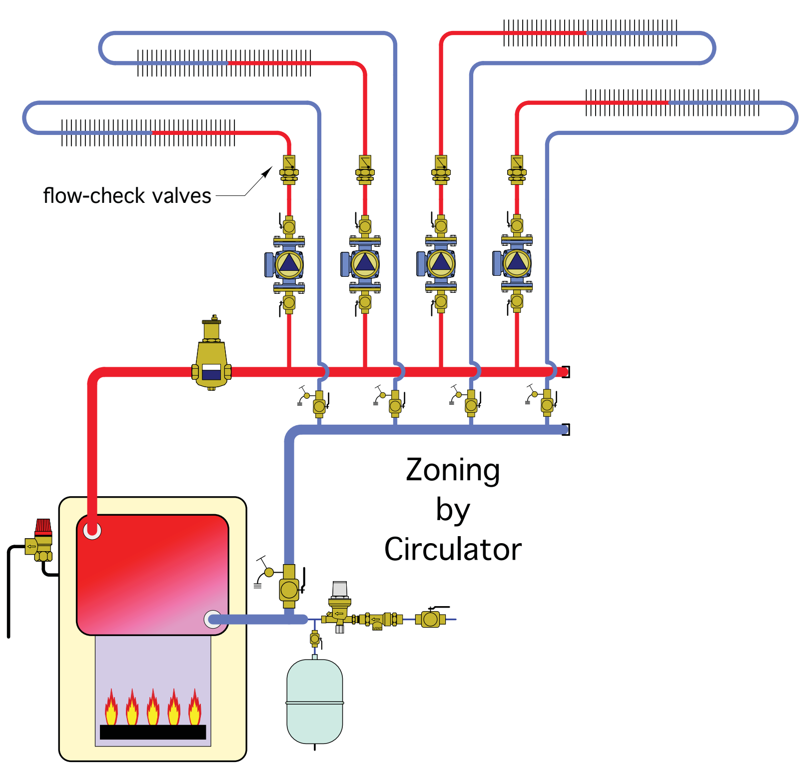
Notice that each zone circuit contains a flow-check valve. This valve can be a separate component mounted downstream of the circulator, as shown in Figure 9-5, or it can be integral to the circulator. The flow-check valve prevents reverse flow through inactive zone circuits while other zones are operating. It also prevents buoyancy-driven hot water migration through inactive zone circuits. Each zone circuit should also be equipped with its own purging valve to expedite air removal during start-up and servicing.

Zone circulator systems have the following characteristics:

• Each zone circuit receives the same supply water temperature.

• The combined flow rate of all operating zones must pass through whatever the headers connect to. To minimize interaction between zone circuits, headers, as well as the piping components connected to the headers, should provide low head loss. This is generally not a problem when cast iron boilers or other low-head-loss heat sources are used. However, heat sources with higher flow resistance need to be hydraulically separated from the headers using either closely spaced tees or a hydraulic separator.

Figure 9-5



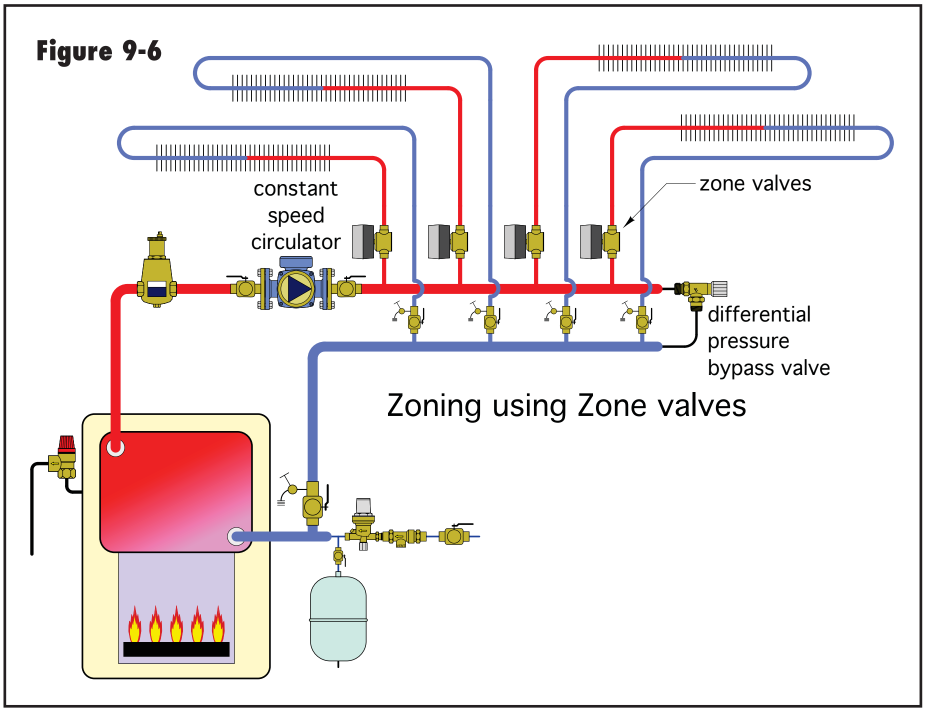
• The total electrical power demand of a zone circulator system, at design load, tends to be higher than with other methods of zoning. Designers should carefully scrutinize circulator selection and speed settings to minimize “over-pumping” and excessive electrical energy use.

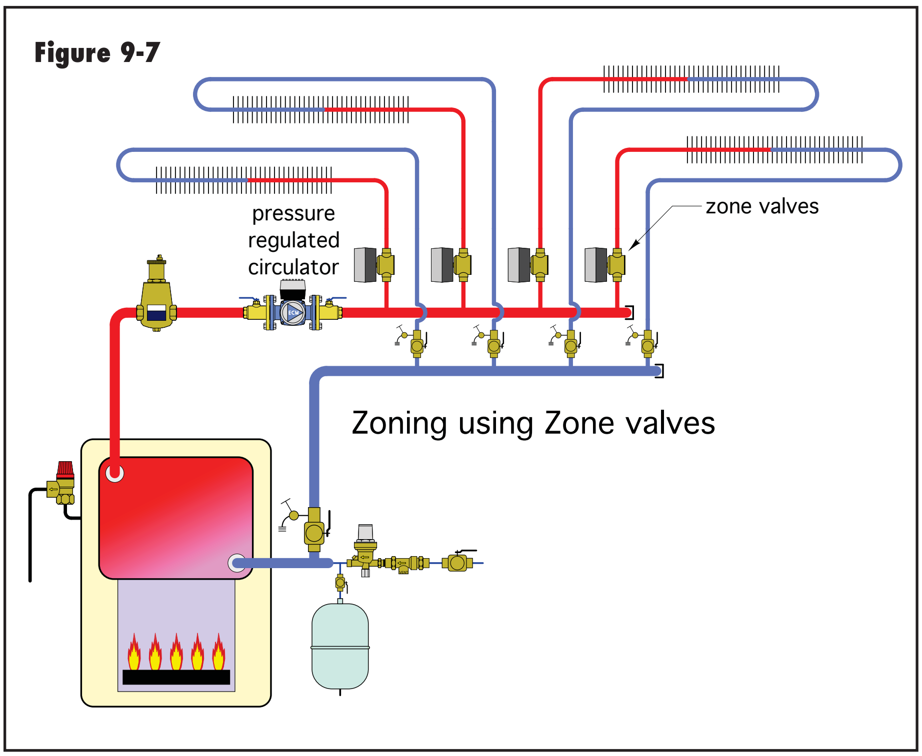
ZONE VALVE SYSTEMS:

Another method of constructing multi-zone systems uses a single circulator in combination with an electrically operated zone valve on each zone circuit, as shown in Figure 9-6.

When any zone thermostat calls for heat, its associated zone valve opens. An end switch within the zone valve closes once the valve reaches its fully open position. This switch signals the circulator and boiler to operate. All zone circuits begin and end at headers.

The differential pressure bypass valve installed at the right end of the headers limits the increase in differential pressure created by a constant-speed circulator as the number of active zones decreases. Without this valve, the increase in differential pressure created





by a constant-speed circulator may cause excessive flow velocity and noise in operating zone circuits.

Zone valve systems have the following characteristics:

• Each zone circuit receives the same supply water temperature.

• The combined flow rate of all operating zones must pass through whatever the headers connect to. To minimize interaction between zone circuits, headers, as well as the piping components connected to the headers, should provide low head loss. This is generally not a problem when cast iron boilers or other low-head-loss heat sources are used. However, heat sources with higher flow resistance need to be hydraulically separated from the headers using either closely spaced tees or a hydraulic separator.

• The total electrical power demand of a zone valve system at design load tends to be lower than an equivalent system using zone circulators.

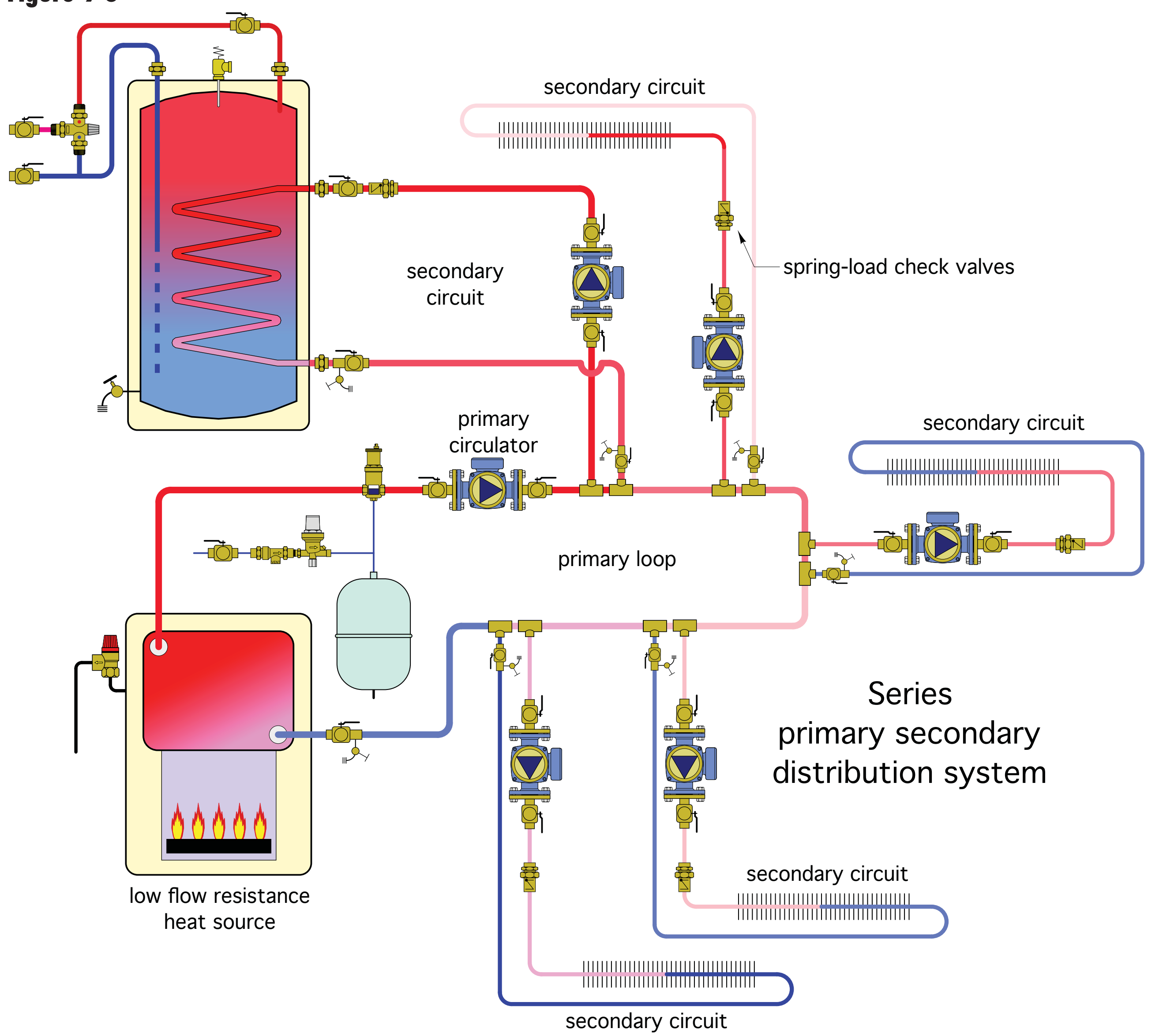
• Zone valve systems are well-suited to variable-speed pressure-regulated circulators that automatically adjust speed and electrical power input to maintain a constant differential pressure between the supply and return headers, regardless of which zone circuits are operating. When a pressure-regulated circulator is used, there is no need of a differential pressure bypass valve. Figure 9-7 shows the system of Figure 9-6 modified for use with a pressure-regulated circulator.

idronics #5 provides a complete description of zoning with zone valves

SERIES PRIMARY/SECONDARY SYSTEMS:

Series primary/secondary piping has been used for several decades in larger commercial hydronic systems. During the last two decades, this approach has also been adapted in multi-load/multi-temperature residential hydronic systems within North America.

Figure 9-8



A primary/secondary system usually consists of a single primary loop that delivers heated water to one or more secondary circuits. The latter are connected to the primary loop using pairs of closely spaced tees, as shown in Figures 9-8.

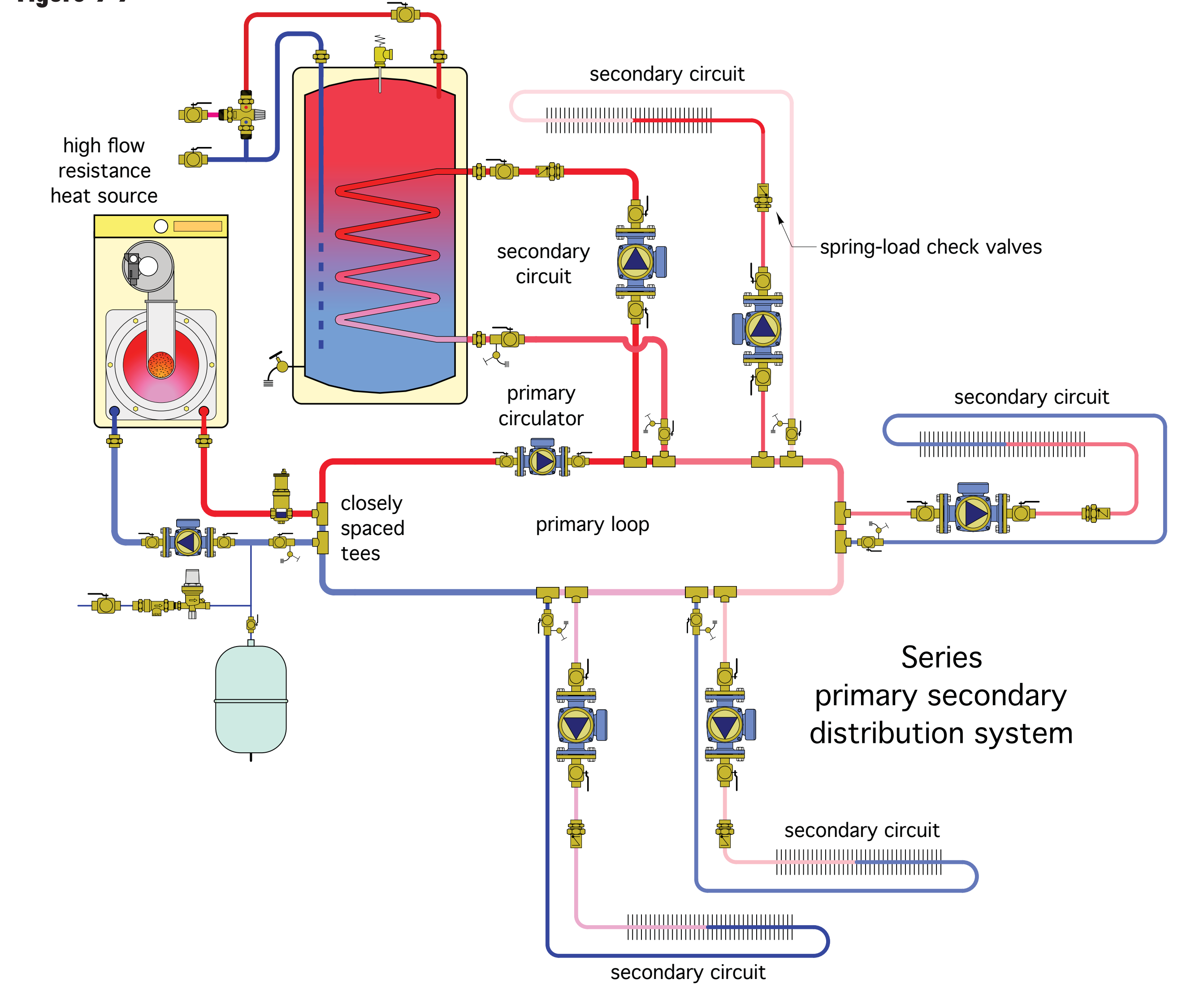
The pressure drop between a pair of closely spaced tees is almost zero. Thus, there is very little tendency for flow along the primary loop to induce flow within a secondary circuit. The closely spaced tees provide hydraulic separation between the primary circuit and each of the secondary circuits. Flow in a secondary circuit occurs when its own "secondary circulator" is turned on.

The word "series" describes how the sets of closely spaced tees are arranged, one set after another, along the primary loop.

If the heat source has low flow resistance, it is usually piped as part of the primary loop as shown in Figure 9-8. However, if the heat source has higher flow resistance, it is usually provided with its own circulator and connected to the primary loop using a set of closely spaced tees, as shown in Figure 9-9.

The following points summarize series primary/secondary systems:

Figure 9-9



• The sequential arrangement of secondary circuits around the primary loop creates a temperature drop wherever primary loop flow passes a pair of tees connected to an operating secondary circuit. As in series loops and diverter tee systems, this reduction in water temperature must be accounted for when sizing heat emitters.

• Each circuit in a primary/secondary system is hydraulically isolated from the other circuits. This prevents interference between circulators. Each circuit can be designed as if it's a "stand-alone" circuit unaffected by other circuits in the system.

• A flow-check valve or spring-loaded check valve must be installed on the supply side of each secondary circuit to prevent buoyancy-driven heat migration.

• The primary circulator must operate whenever any one or more of the secondary circulators is operating. This increases electrical energy consumption relative to systems that do not require a primary circulator.

PARALLEL PRIMARY/SECONDARY SYSTEMS:

In a parallel primary/secondary system, the primary loop is divided into multiple parallel crossovers, as shown in Figure 9-10. Water from the heat source splits up through all these crossovers whenever the primary circulator is operating. A pair of closely spaced tees within each crossover provides hydraulic separation of each secondary circuit.

The following points summarize parallel primary/secondary systems:

Figure 9-10

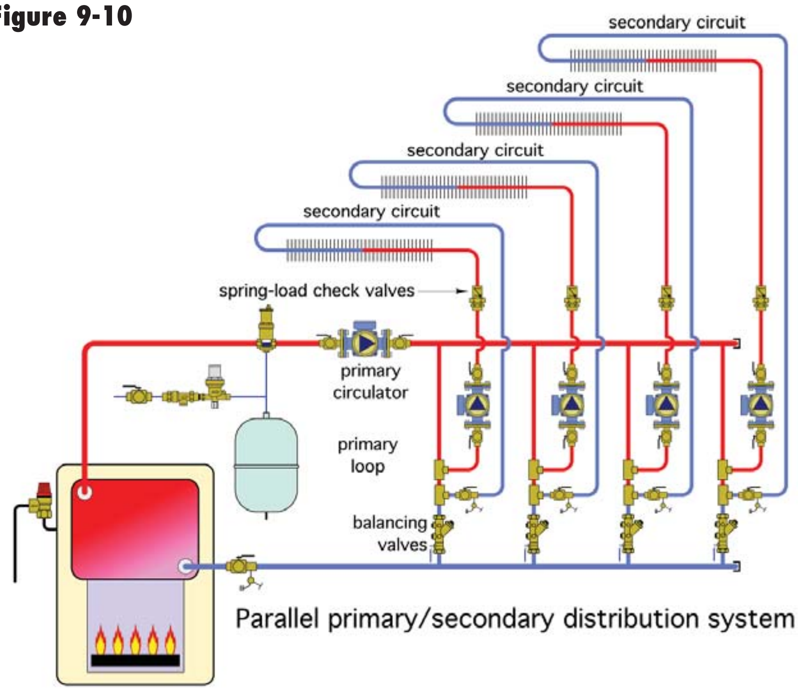
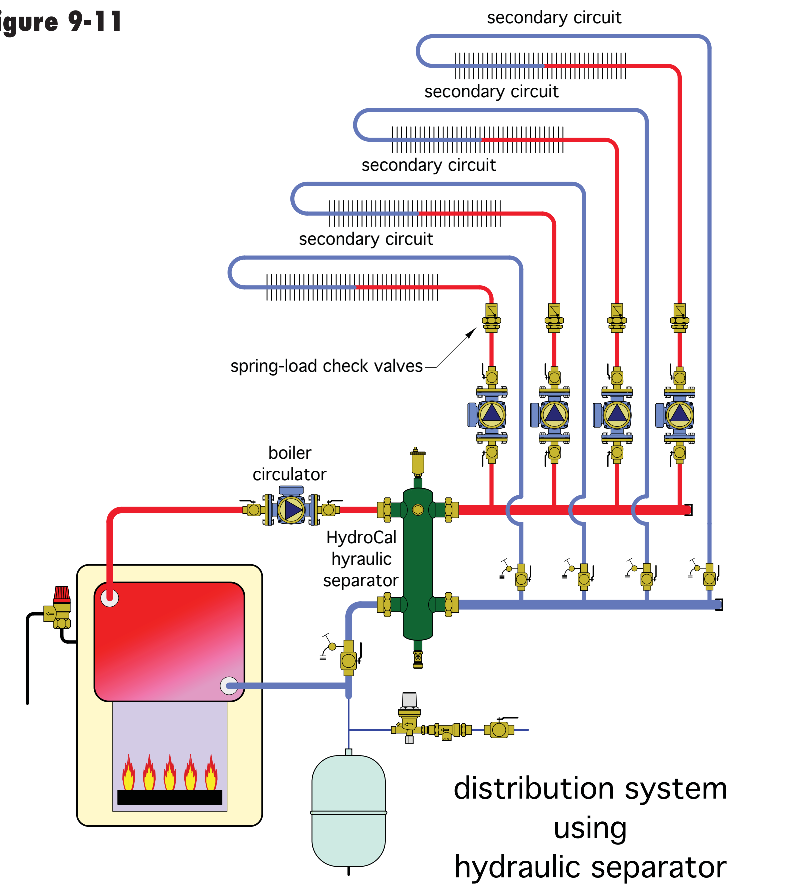


Figure 9-11



• The parallel arrangement of crossovers provides approximately the same water temperature to each secondary circuit. This simplifies heat emitter sizing.

• A balancing valve should be placed in each crossover so flow can be adjusted in proportion to the percentage of total load handled by the secondary circuit connected to each crossover. This adds to the installed cost and commissioning cost of the system.

• Each circuit in a parallel primary/secondary system is hydraulically isolated from the other circuits. This prevents interference between circulators.

• A flow-check valve or spring-loaded check valve must be installed on the supply side of each secondary circuit to prevent buoyancy-driven heat migration.

• The primary circulator must operate whenever any one or more of the secondary circulators is operating. This increases the electrical energy consumption of the system relative to systems that do not require a primary circulator.

Although a parallel primary/secondary system improves upon a series primary/secondary system by providing equal supply water temperatures to each secondary circuit, it does so with added complexity and expense.

The benefits offered by a parallel primary/secondary system can be provided with simpler piping by using a hydraulic separator as shown in Figure 9-11. This system provides equivalent thermal and hydraulic performance to the system shown in Figure 9-10. High-performance air and dirt separation is also provided by the Caleffi HydroCal hydraulic separator.

MANIFOLD-TYPE DISTRIBUTION SYSTEMS:

Polymer-based tubing, such as PEX, PEX-AL-PEX and PE-RT (polyethylene - raised temperature), has become increasingly popular worldwide as an alternative to rigid metal tubing. In North America, these flexible tubing materials are best known for their use in hydronic radiant panel heating systems. Beyond such applications, their temperature and pressure ratings, flexible characteristics and long continuous lengths make them well-suited to other types of hydronic distribution systems.

In North America, one or more of these flexible tubing materials is currently available in sizes from 5/16” to 2” nominal inside diameter. Of these, 3/8” to 3/4” are the most commonly used sizes for branch circuits in manifold-type distribution systems.

Special fittings are available for connecting both PEX and PEX-AL-PEX tubing to cast brass manifolds, as shown in Figure 9-12.

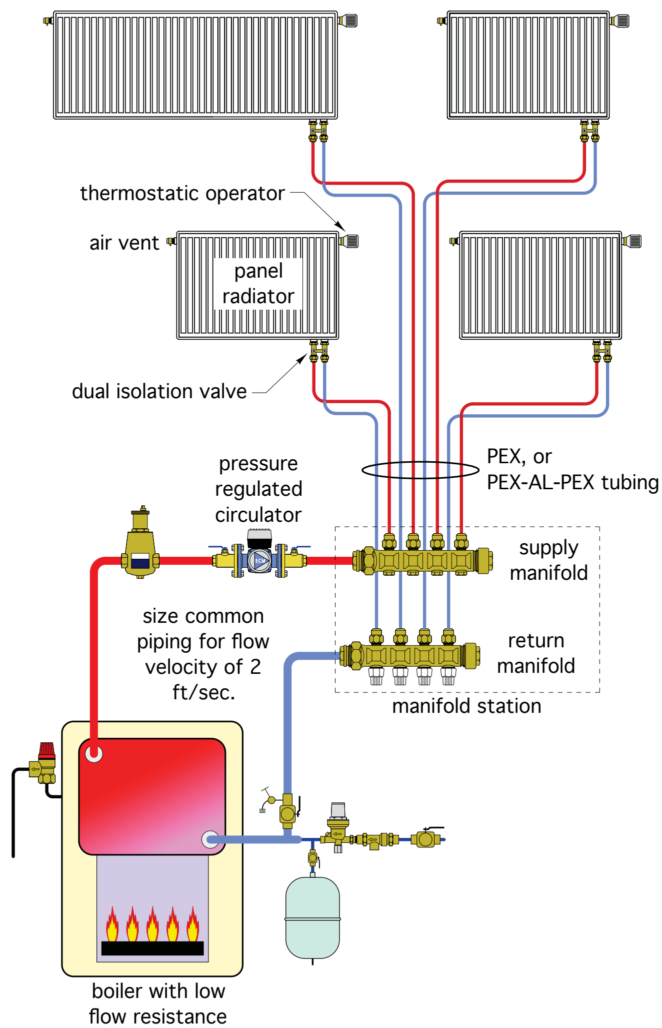
Figure 9-12

These fittings use a single, wrench-tightened mechanical compression nut and collar in combination with precision O-rings to ensure high-quality seals at both the tube and manifold connection. When selecting manifold connection fittings, be sure their temperature/pressure rating is compatible with the intended operating conditions.

A simple manifold-based hydronic distribution system is shown in Figure 9-13.

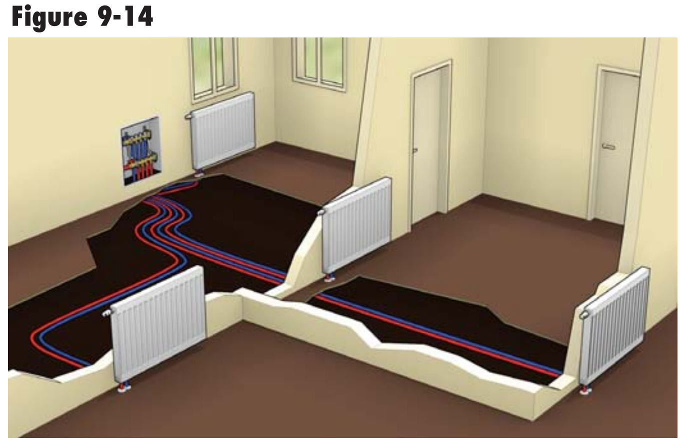
The low-flow-resistance boiler supplies heated water to a single manifold station. The heated water enters the supply manifold and splits up among the various branch circuits based on their current needs. These branch circuits are piped with PEX or PEX-AL-PEX tubing. Each branch serves a single heat emitter—in this case, panel radiators equipped with thermostatic radiator valves. The common piping that connects the manifold station and boiler should be sized for a maximum flow velocity of 2 feet per second to minimize any tendency for flow in one branch circuit to affect flow in other branch circuits.

Figure 9-13



The manifold station can be installed within a wall cavity, as shown in Figure 9-14. It can also be installed under a floor deck or in the basement below the floor level(s) where the heat emitters are located.

Figure 9-14



Manifold-based distribution systems offer several advantages compared to distribution systems using rigid metal pipe.

• ***The use of small-diameter flexible tubing allows for easy installation of individual circuits.*** This is especially important in retrofit installations where the installation of rigid piping often requires cutting through finished walls and ceilings. In many of these situations, it's possible to pull small-diameter flexible tubing through confined spaces, much like flexible electrical cable can be "snaked" from one point to another within a building (see Figure 9-15).

Figure 9-15

• ***They supply water at approximately the same temperature to each heat emitter.*** Thus, heat emitters can all be selected based on the same design load supply water temperature. Each heat emitter also receives equal water temperature reduction in systems using outdoor reset control.

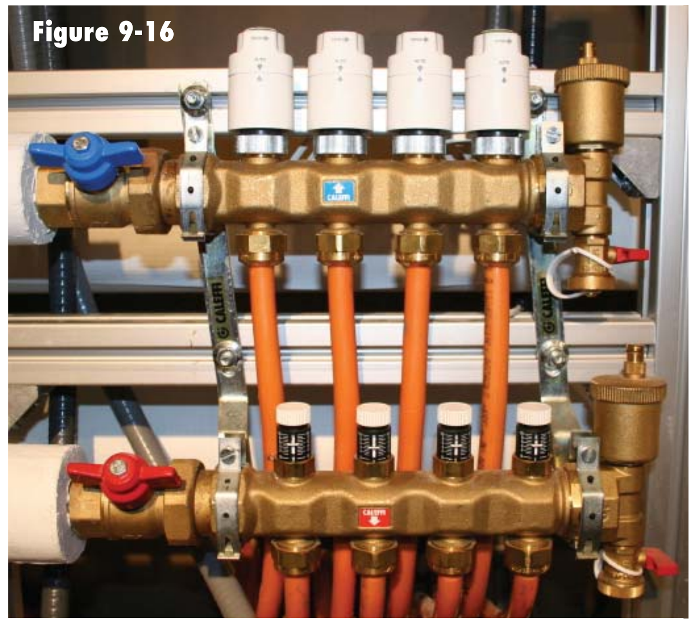
• ***Manifold systems, when combined with either thermostatic radiator valves or individual room thermostats, allow room-by-room temperature control.*** This allows each heat emitter to adjust its output to compensate for internal heat gains created by sunlight, people or equipment. It also allows each room to be adjusted for individual temperature preferences or unoccupied temperature setbacks.

• ***They allow a variety of heat emitters to be used in the same distribution system.*** Heat emitters such as fin-tube baseboard, panel radiators, fan-coils and towel warmers can all be combined into the same system, each on their own branch circuit, provided they are all sized to operate at the same supply water temperature at design load conditions.

• ***The flow rate in each circuit can be adjusted independently using balancing valves built into the manifold station or valves attached to or integrated into each heat emitter.*** Flow adjustments can compensate for "overflow" in shorter circuits and prevent flow starvation in longer circuits.

• ***The flow in each circuit can be turned on or off, or modulated.*** On/off flow control is done using manifold valve actuators. The upper (return) manifold shown in Figure 9-16 is equipped with 4 of these actuators.

Figure 9-16

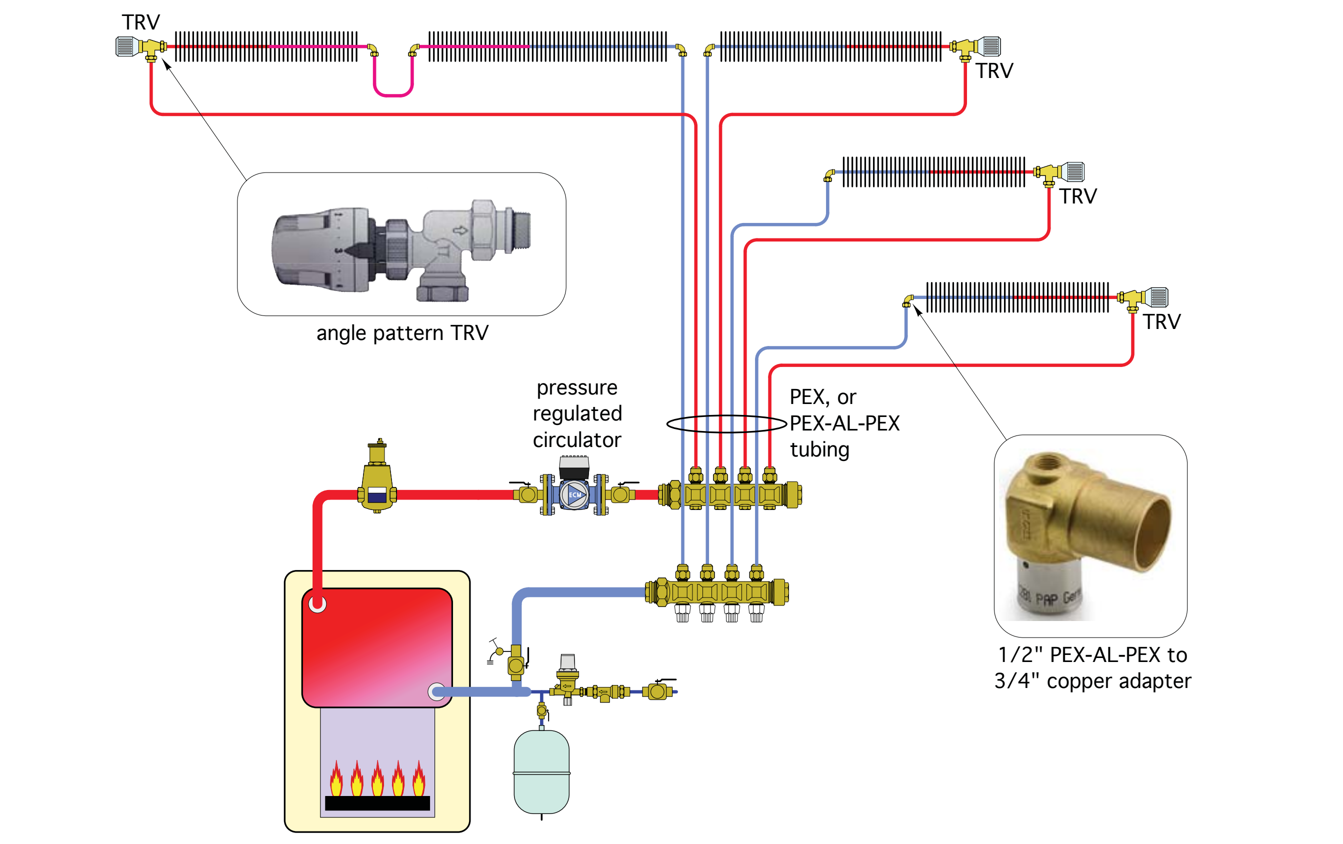


Manifold valve actuators are powered by a 24 VAC thermostat circuit. When screwed into place over a manifold valve, they force the valve's shaft to its closed position. When 24 VAC electrical power is supplied to the actuator, it retracts its stem and allows the spring-loaded manifold valve to open.

• ***Manifold distribution systems that use either on/off or modulating flow control through each circuit are ideally suited to ECM-based pressure-regulated circulators.*** Figure 9-17 shows how fin-tube baseboard can be supplied using a manifold distribution system in combination with thermostatic radiator valves. This type of installation is faster, easier and often less expensive than installing fin-tube baseboard using soldered copper tubing.

• ***Adaptability to Variable-Speed Pressure-Regulated Circulators***. All manifold systems that use valves to regulate flow through branch circuits need a means of differential pressure control. It can be provided by a differential pressure bypass valve or use of a pressure-regulated circulator. The latter scenario is becoming

Figure 9-17



increasingly popular as new ECM (Electronically Commutated Motor) circulators become available in North America. Such circulators are ideal for manifold-type distribution systems. They automatically control differential pressure and significantly reduce electrical power consumption under partial-load conditions.

• ***Lower operating cost: The parallel branches of a manifold distribution system often create less total flow resistance than series or diverter tee-type distribution systems.*** This reduces the electrical power required to maintain circulation, both at design load conditions when all circuits are active, as well as during part-load conditions when some circuits are off and a variable-speed pressure-regulated circulator is used.

The system shown in Figure 9-18 is a typical series loop consisting of five baseboards. Each is sized to deliver 10,000 Btu/hr when the circuit is supplied with 180ºF water. The loop is constructed of 3/4" copper tubing and fittings. It's assumed to follow the perimeter of a 30' x 50' house. With fittings factored in, this circuit has a total equivalent length of 280 feet. Delivering a total of 50,000 Btu/hr with an assumed temperature drop of 20ºF requires a flow rate of 5 gpm. Under these conditions, the pressure drop around the circuit is 5.51 psi. If this flow and differential pressure requirement is provided by a typical wet rotor circulator with a wire-to-water efficiency of 22%, the electrical wattage required to maintain circulation is 54.4 watts.

For comparison, Figure 9-19 shows a manifold distribution system designed to provide the same heating load. Each branch circuit consists of 1/2" PEX tubing and a 20-foot length of 3/4" fin-tube baseboard. The equivalent length of each branch circuit is assumed to be 120 feet of 1/2" PEX tubing. The manifold is supplied with 180ºF water through 1" copper tubing. At a flow rate of 5 gpm, this distribution system only develops a pressure drop of 1.68 psi. Assuming a circulator with the same 22% wire-to-water efficiency was used, the electrical wattage required to maintain circulation would be 16.6 watts, less than one third that required by the series loop system.

If each distribution system operated 3,000 hours per year in a location where electrical energy costs 12 cents per kwhr, the annual electrical savings associated with the manifold system would be approximately $13.60. If this cost inflated by 4% each year, the total savings over 20 years would be $405.

This comparison is based on the nominal 22% peak wire-to-water efficiency of a constant speed PSC-based wet rotor circulator. Beyond this scenario is the possibility of using a variable-speed pressure-regulated circulator for the manifold system. These circulators have demonstrated the ability to reduce electrical energy usage by 60% or more, relative to constant-speed, PSC-based wet rotor circulators.

idronics #5 provides additional information on zoning using manifold valve actuators.

Figure 9-18

$$w_e = \frac{0.4344 \times f \times \Delta P}{n_{w/w}} = \frac{0.4344 \times 5.0 \times 5.51}{0.22} = 54.4\,watts$$

We = electrical wattage
f = flow rate (gpm)
ΔP = differential pressure (psi)
nw/w = wire-to water efficiency of circulator

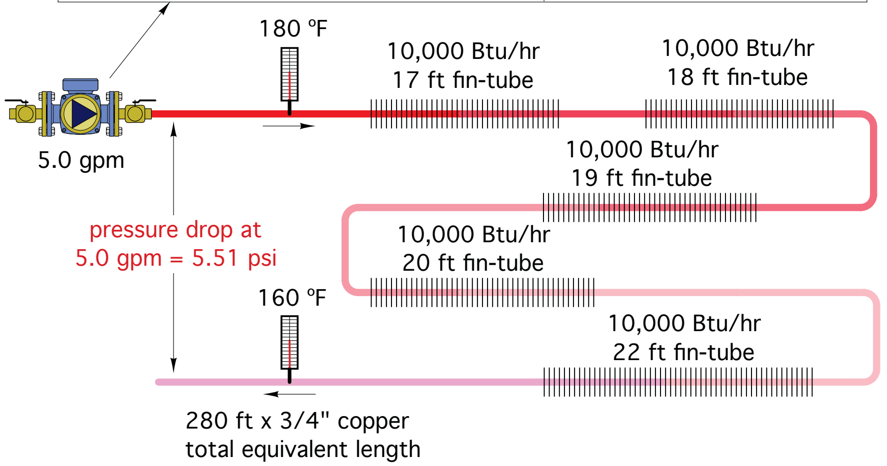
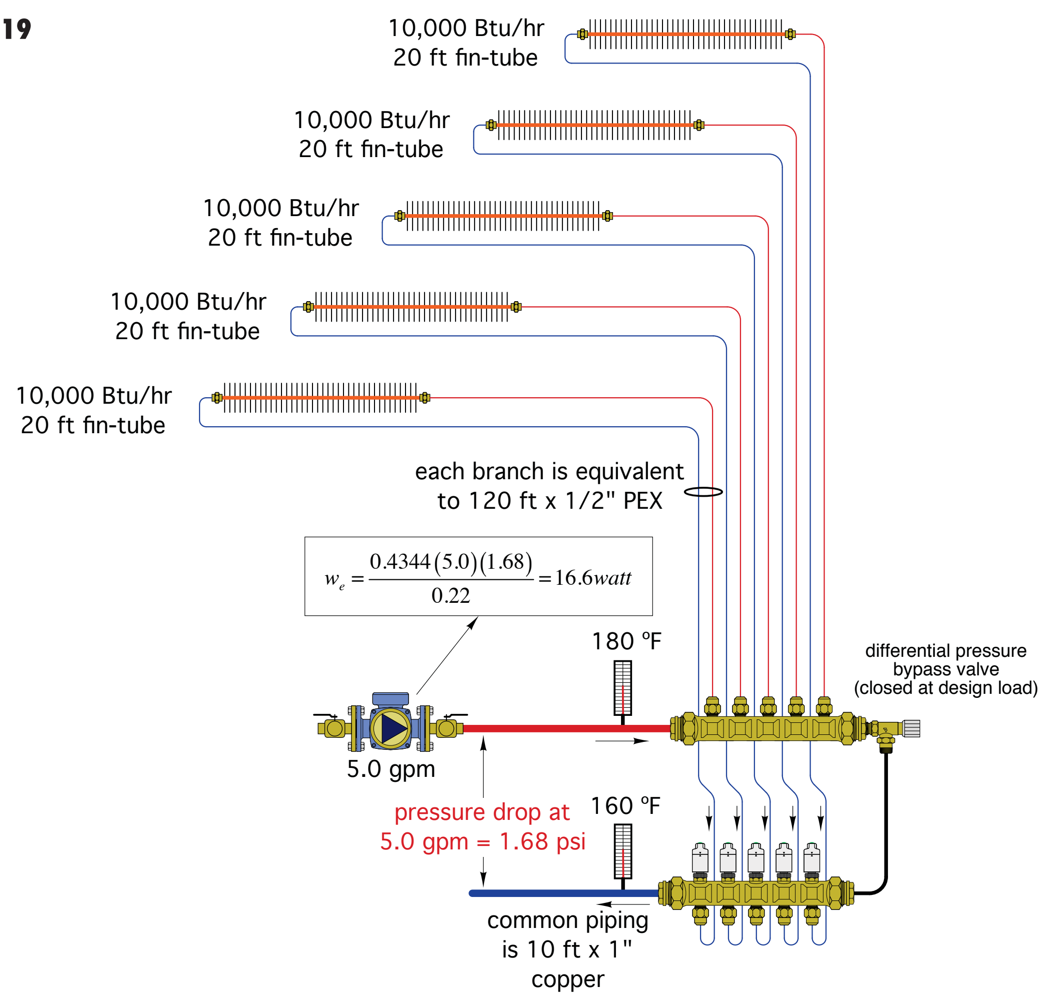


Figure 9-19



$$w_e = \frac{0.4344(5.0)(1.68)}{0.22} = 16.6\,watt$$

BASICS OF BALANCING:

Many of the distribution systems discussed in this section use valves to regulate flow through two or more branch circuits. In most cases, the flow rates through these branch circuits need to be different due to differing heat delivery requirements.

The term “balancing” describes the process by which the flow rate in each branch of such systems is adjusted so that it is at or close to the flow required for proper heat delivery.

Most hydronic heating professionals agree that balanced systems are desirable. However, opinions are widely varied on what constitutes a balanced system. The following are some of the common descriptions of a properly balanced system:

• The system is properly balanced if all simultaneously operating circuits have the same temperature drop.

• The system is properly balanced if the ratio of the flow rate through a branch circuit divided by the total system flow rate is the same as the ratio of the required heat output from that branch divided by the total system heat output.

• The system is properly balanced if all branch circuits are identically constructed (e.g., same type, size and length of tubing; same fittings and valves; same heat emitter).

• The system is properly balanced if constructed with a reverse return piping layout.

• The system is properly balanced if the installer doesn’t receive a complaint about some rooms being too hot while others are too cool.

While some of these definitions of proper balancing are related, none of them is totally correct or complete. It follows that any attempt at balancing a system is pointless without a proper definition and “end goal” for the balancing process. The following statement correctly identifies the end goal of balancing a hydronic system:

A properly balanced hydronic system is one that consistently delivers the proper rate of heat transfer to each space served by the system.

Balancing hydronic systems requires simultaneous changes in the hydraulic operating conditions (e.g., flow rates, head losses, pressure drops), as well as the thermal operating conditions (e.g., fluid temperatures, room air temperatures) of the system. These operating conditions will always interact as the system continually seeks both hydraulic equilibrium and thermal equilibrium. The operating conditions will also be determined, in part, by the characteristics of the heat emitters and circulator used in the system.

Considering that there are often hundreds, if not thousands, of piping and heat emitter components in a system, and that nearly all of them have some influence on flow rates and heat transfer rates, it is readily apparent that a theoretical approach to balancing can be complicated. However, in many cases, it is sufficient to have a clear understanding of how and why certain conditions exist or develop within a system, even without numbers to show the exact changes. Such an understanding can guide the balancing process in the field and help the balancing technician avoid mistakes or incorrect adjustments that delay or prevent a properly balanced condition from being attained.

idronics #8 provides an in-depth discussion of several methods of hydronic balancing along with the proper placement and adjustment of balancing valves.

Figure 9-20

The control subsystem is the “brain” of any hydronic heating or cooling system. It determines exactly when and for how long devices such as circulators, burners, compressors and mixing valves will operate. Even the best hydronic heat source, distribution system and heat emitters, cannot provide proper comfort without a well-planned control strategy.

This section gives an overview of the basic controllers used in modern hydronic heating systems. It also provides information on how the heat output of a system responds to changes in water temperature, as well as changes in flow rate, which is also essential when planning control strategies.

CONTROLLING HEAT OUTPUT:

There are several ways to regulate the rate of heat delivery from a hydronic system to the building it serves. Fundamentally, they can be categorized as follows:

- Turn the heat source and distribution circulator on and off
- Regulate the temperature of the water supplied to the heat emitters
- Regulate the flow rate of water passing through heat emitters.

ON/OFF CONTROL OF HEAT SOURCE:

Many of the heat sources (e.g., boilers, heat pumps, etc.) used in residential and light commercial hydronic systems have only two operating modes: on or off. Whenever the heat source is on, it produces heat at a fixed rate. When the heat source is off, it produces no heat output. Such heat sources require a simple on/off control signal from their controller. This is typically provided by the closing and opening of an electrical contact.

The total amount of heat added to the system over time is regulated by periodically turning the heat source on and off. This is illustrated in Figure 10-1.

Figure 10-1

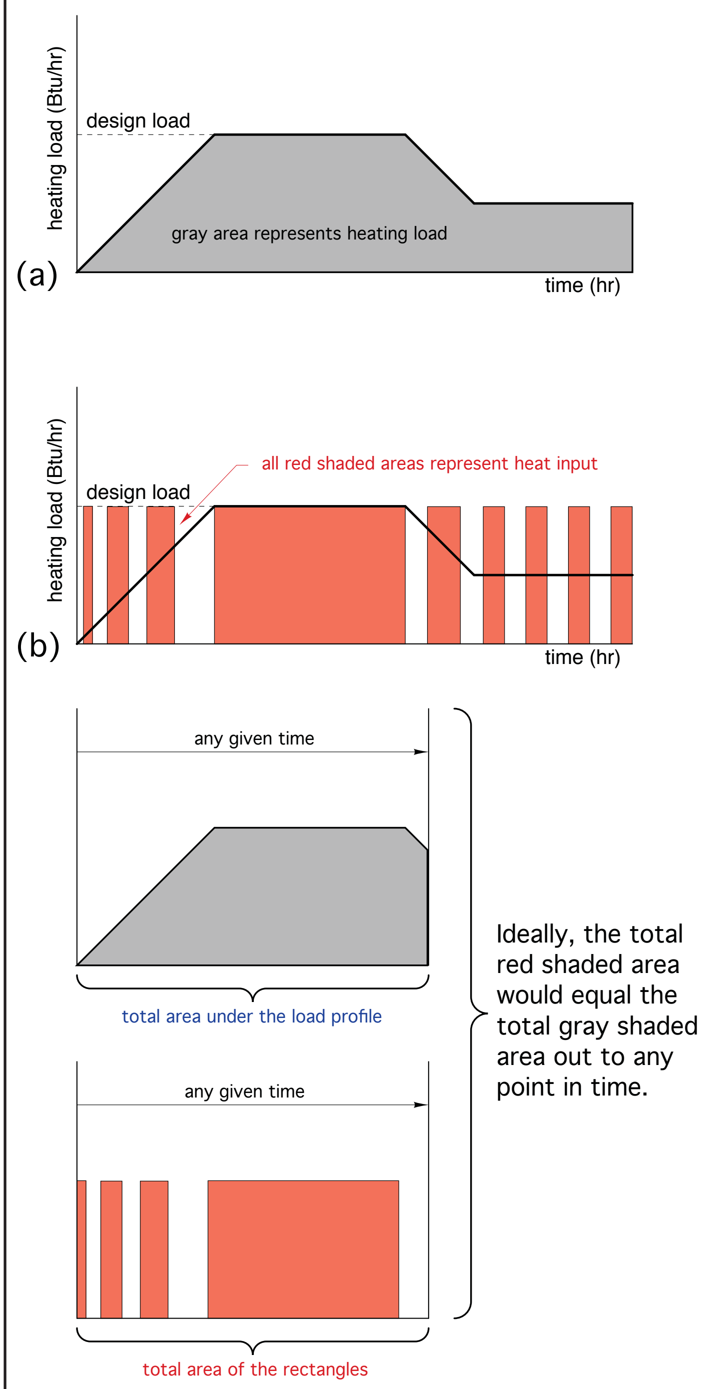


Figure 10-1a shows a hypothetical heating load profile (e.g., the rate of heat delivery required by the load over time). The load begins at zero, rises at a constant rate to its maximum design value and then decreases back to half its design value. The time over which this occurs can be assumed to be a few hours.

Figure 10-1b shows how a heat source operated by an on/off controller supplies heat to this load. The red-shaded rectangles represent heat input from the heat source. The height of the rectangles remains the same because the rate of heat delivery is constant whenever the heat source is on. In this case, that rate matches the design load (e.g., the maximum required load).

The width of the rectangles increases as the load increases. This means that the duration of the on-cycle is increasing. When design load occurs, the rectangle remains uninterrupted. Since the heat source is sized to the design load of the building, it must operate continuously whenever design load conditions exist. As the load decreases, the rectangles become narrower. When the load stabilizes at 50% of design load, the width of the rectangles remains constant. Under this condition, the heat source is on 50% of the time. When on, its rate of heat output is twice the rate at which the building requires heat.

The area of each red-shaded rectangle represents the quantity of heat delivery by the heat source during its on cycle. Ideally, the gray-shaded area under the load profile, from when the load begins and out to some point in time, exactly equals the total area of the red-shaded rectangles over that same time. This implies that the total heat production of the heat source exactly matches the total load over that period.

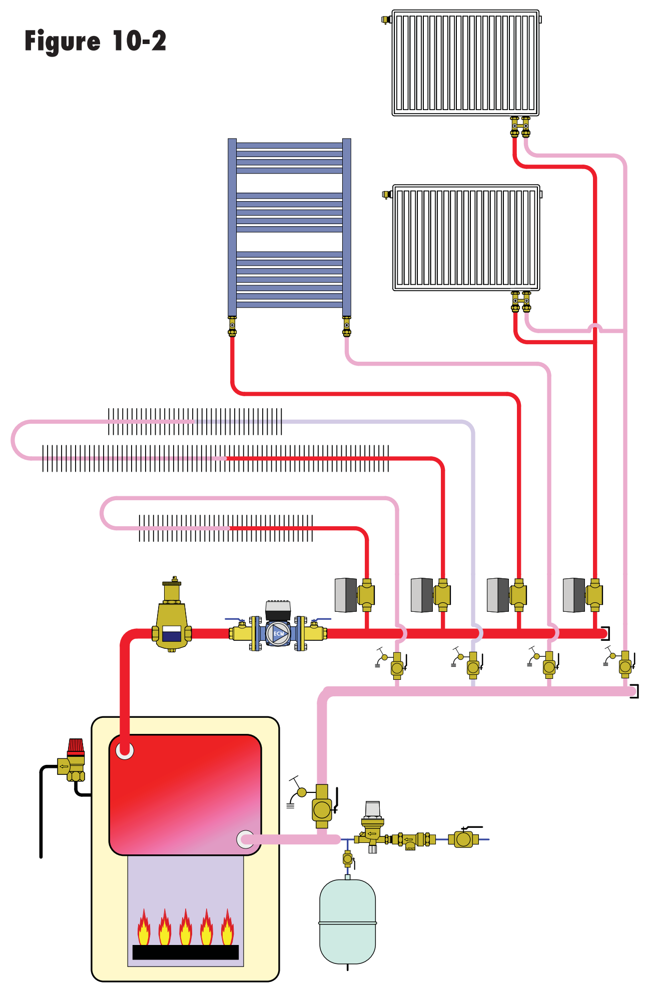
The match between heat output and heating load at low and medium load conditions is not ideal. For example, consider the time when the load is very small. Under such conditions, the heat source sends short pulses of heat to the load. The height of these pulses corresponds to the full heating capacity of the heat source, which under partial load conditions may be several times higher than the load.

In heating systems with low thermal mass, this on/off cycling of the heat source is often noticeable and generally undesirable. A forced-air furnace with a constant-speed blower and fixed firing rate is an example of such a system. Under partial load conditions, the furnace produces its full rate of heat output whenever it is on. Since the air in the building, along with the metal in the furnace's heat exchanger, provides very little thermal mass for absorbing this heat, the air temperature in the building increases quickly. When the thermostat reaches its setpoint temperature, its contacts open and the furnace is turned off. An operating cycle could be as short as 2 or 3 minutes under partial-load conditions.

During this time, the occupants sense the rapidly changing room air temperature and usually do not like it. These short operating cycles repeat themselves as long as the load persists at its present level.

Many hydronic systems, especially those having high thermal mass embodied by the heat source, buffer tank or high mass heat emitters, have an advantage in this respect. Their greater thermal mass can temporarily absorb some of the surplus heat output generated by the heat source and spread out its delivery over time. This helps stabilize room temperatures. The system shown in Figure 10-2 is an example.

Figure 10-2



The heat source is assumed to be a small cast iron boiler. The heat exchanger in a typical small cast-iron boiler contains approximately 400 pounds of cast iron and 10 gallons of water. The combined thermal mass of the metal and water allow the boiler to continue delivering heat to the zoned distribution system for several minutes after the burner has turned off. After the boiler's thermal mass has cooled by several degrees, the burner is turned back on and will likely operate for several minutes before the water temperature leaving the boiler can again climb high enough to turn the burner off. This prevents "short cycling" of the burner. Longer on-cycles also improve combustion efficiency and reduce emissions.

Systems with low-mass heat sources, such as certain types of mod/con boilers or water-to-water heat pumps, often require additional mass to prevent short cycling.

Figure 10-3

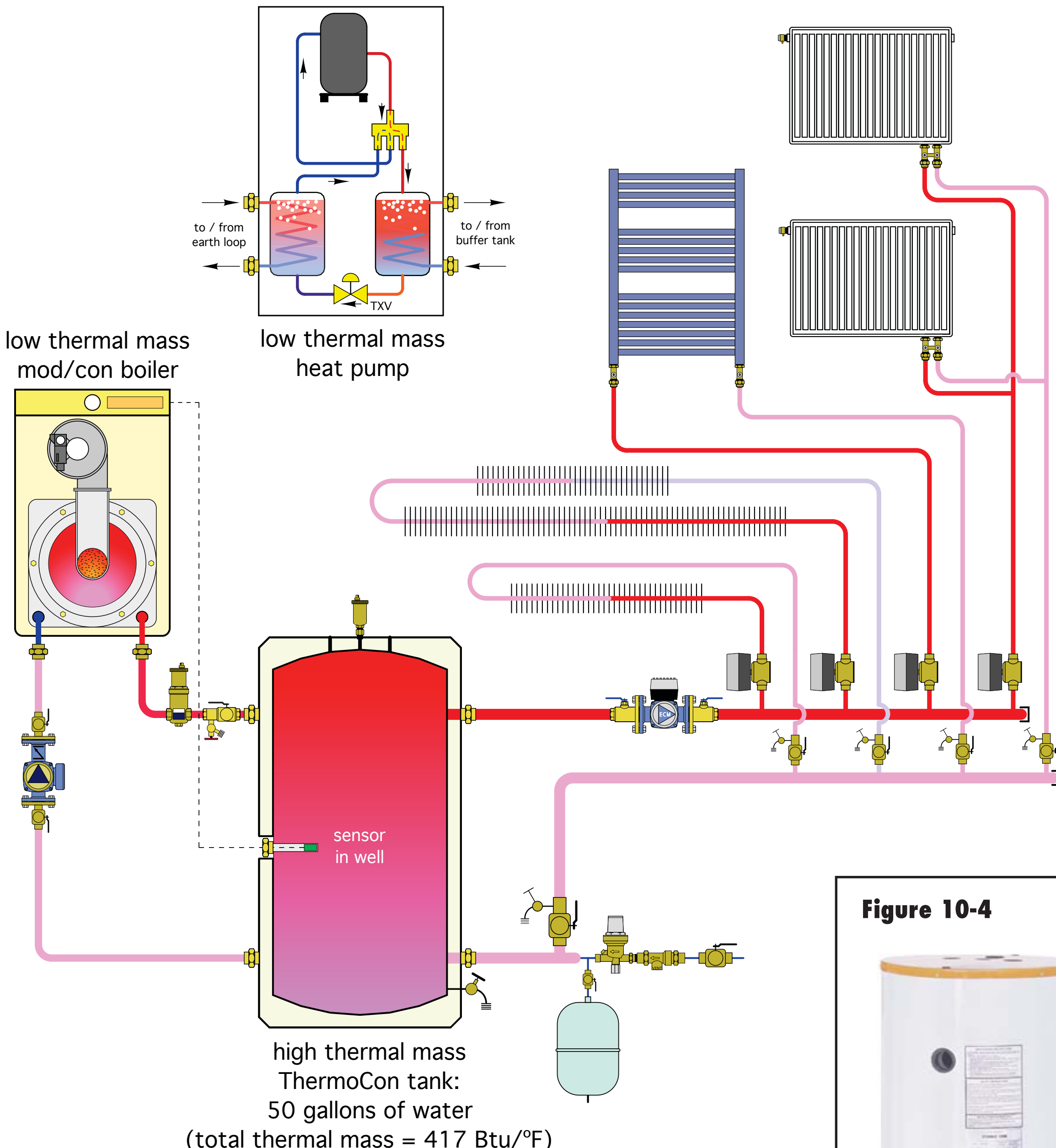


Figure 10-4

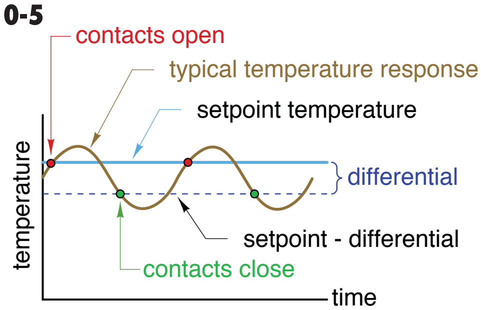
This added mass can be provided through use of a buffer tank, as shown in Figure 10-3.

Buffer tanks, such as the Caleffi ThermoCon tank shown in Figure 10-4, can be sized to provide a specific minimum on-cycle. A commonly accepted minimum on-cycle for boilers and heat pumps is 10 minutes. Appendix B provides relatively simple formulas for determining the required volume of the buffer tank based on the heat capacity of the heat source, and a specified minimum on-cycle.

ON/OFF SETPOINT TEMPERATURE CONTROL:

Many control actions in hydronic systems are initiated and terminated based on temperature. An example would be burner operation in an on/off gas-fired boiler. When there is a "call" for heat by some other controller in the system, control authority is given to the boiler's high-limit controller. If the water temperature in the boiler is at or below the current "setpoint" of the high-limit controller minus a specific "differential," the burner is turned on. When (and if) the water temperature in the boiler reaches the setpoint temperature of the high-limit controller, the burner is turned off. This control action is illustrated in Figure 10-5.

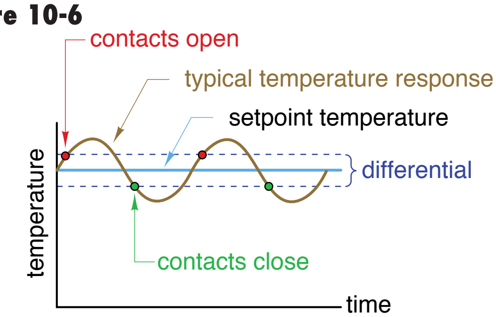
Figure 10-5



NOTE: differential below setpoint

The "differential" of a setpoint temperature controller is the difference in temperature between where the electrical contacts on the controller close and when they open. Some setpoint temperature controllers place the differential below the setpoint, as shown in Figure 10-5. For example, if the setpoint of the controller is 140ºF and the differential is 10ºF, the electrical contacts on the controller would close when the temperature being sensed drops to 130ºF (e.g., 140ºF - 10ºF differential). These contacts would reopen if and when the temperature rises to 140ºF.

Figure 10-6



NOTE: differential centered on setpoint

Other setpoint controllers center the differential on the setpoint temperature, as shown in Figure 10-6. In this case, if the setpoint was 140ºF and the differential was 10ºF, the contacts would close at 135ºF and reopen at 145ºF.

Both the setpoint and differential are adjustable on most temperature setpoint controllers used in hydronic heating and cooling systems. Setting the differential involves a compromise: The greater the differential, the longer the on and off cycles of the controlled device will be. This is generally considered beneficial to the life expectancy of devices such as burners or compressors. However, the wider the differential, the greater the variation in temperature between the desired setpoint and what the temperature may be at any given time. If the actual temperature varies too far from the setpoint, comfort and efficiency may be negatively affected.

A typical room thermostat is also a temperature setpoint controller. When the room air temperature drops slightly below the setpoint of the thermostat, it closes its electrical contacts to initiate a "call" for heat from the remainder of the system. When the room air temperature rises and approaches the setpoint, the thermostat's contacts open to stop further heat generation by the heating system.

FLOATING CONTROL:

Floating control was developed to operate motorized valves or dampers that need to be powered open as well as powered closed. In hydronic heating systems, floating control is commonly used to drive 3-way and 4-way motorized mixing valves.

Figure 10-7

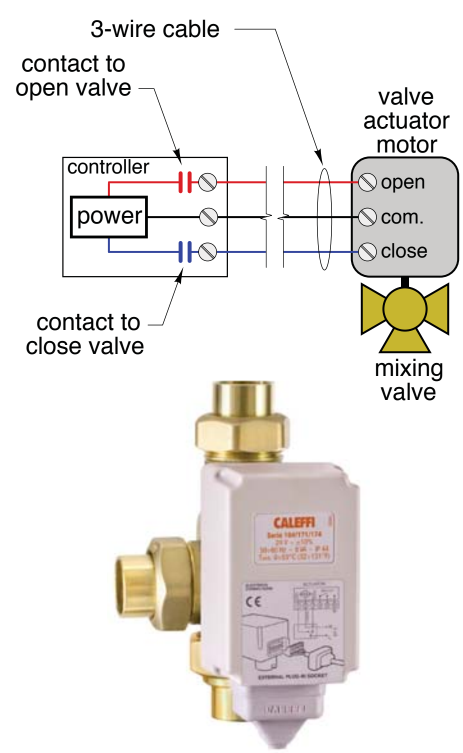


Figure 10-7 shows how a controller operating with floating control would be wired to a typical motorized mixing valve (an example of which is seen below the schematic).

With floating control, the controlled device is always in one of three possible states:

1. The controlled device is opening
2. The controlled device is closing
3. The controlled device is holding its current position

These states are created by the status of two normally open relay contacts within the controller, as seen in Figure 10-7. One electrical contact closes to drive the controlled device's actuating motor in a clockwise (CW) direction. The other electrical contact closes to drive the actuator motor counterclockwise (CCW). Only one contact can be closed at any time. The actuating motor driving the controlled device turns very slowly. Some actuating motors can take up to three minutes to rotate the shaft of a valve or damper over its full rotational range of 90 degrees. This slow operation is desirable, since it allows the sensor ample time to provide feedback to the controller for stable operation.

Floating control is sometimes also called "3-wire control" because three wires are required between the controller and the actuating motor of the controlled device.

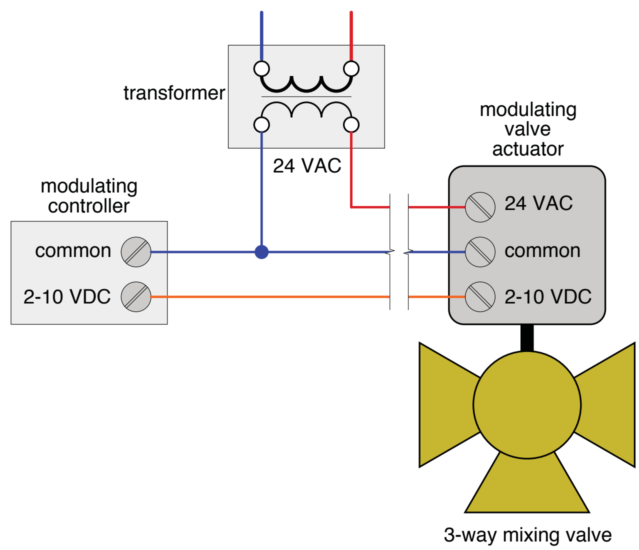
MODULATING TEMPERATURE CONTROL:

Some control actions used in hydronic heating systems require gradual changes within a predetermined range. For example, depending on the current heating load, the firing rate of a burner in a mod/con boiler may need to vary from full output down to about 20% of that output. Another example would be where the speed of a circulator has to vary from completely off to full speed. These control actions are provided by controllers with modulating output signals.

Modulating controllers output a continuous electrical signal, either 2 to 10 volts DC or 4 to 20 milliamp current. The value of this voltage or current within its overall range is what determines the response of the controlled device. For example, a controller output signal of 2 volts or less supplied to a motor is "instructing" the motor to remain off. A 10-volt DC signal to the same motor means that it should be operating at full speed. A signal of 6 volts DC, which is 50% of the overall range of 2 to 10 volts, means that the motor should be running at 50% of full speed.

Keep in mind that these variable DC outputs are for control only and do not supply the electrical power to drive the device. For example, the typical wiring of a 2 to 10 VDC modulating valve is shown in Figure 10-8. Notice that 24 volts AC power must also be supplied to the valve to power the motor and the actuator's circuitry. Likewise, a variable-speed pump that is controlled by a 2 to 10 VDC or 4 to 20 milliamp signal typically requires line voltage (120 VAC) to supply operating power.

Figure 10-8



The reason these control signals do not begin at zero voltage or current is to prevent electrical interference or "noise" from affecting the controlled device. This electrical noise can come from nearby motors, electrical wiring or certain types of lighting.

CONTROLLING HEAT OUTPUT FROM HEAT EMITTERS:

There are two fundamental methods of controlling the heat output of hydronic heat emitters:

1. Vary the water temperature supplied to the heat emitter while maintaining a constant flow rate through the heat emitter.

2. Vary the flow rate through the heat emitter while maintaining a constant supply water temperature to the heat emitter.

Both approaches have been successfully used in many types of hydronic heating applications over several decades. It is important for system designers to understand the differences, as well as the strengths and weaknesses, of each approach.

VARIABLE WATER TEMPERATURE CONTROL:

The heat output of any hydronic heat emitter is approximately proportional to the difference between supply water temperature and room air temperature. This can be represented mathematically as Formula 10-1:

Formula 10-1

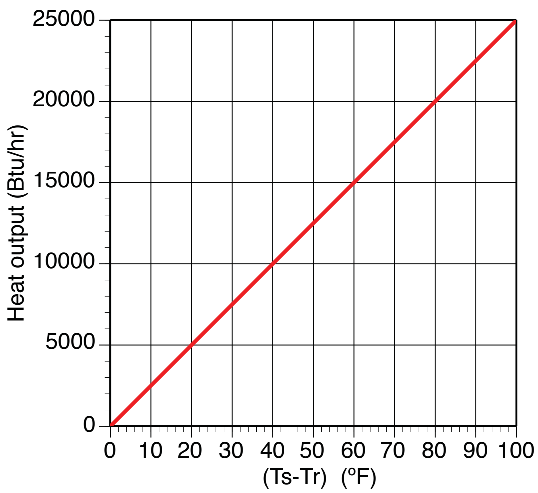
$$Q_o = k\left(T_s - T_r\right)$$

where:

Q_o = rate of heat output from heat emitter (Btu/hr)
k = a constant dependent on the heat emitter used
T_s = fluid temperature supplied to the heat emitter (ºF)
T_r = air temperature of room where heat emitter is located (ºF)

A graph of Formula 10-1 is a straight line, as shown in Figure 10-9. For this graph, the value of "k" in Formula 10-1 is 250.

Figure 10-9

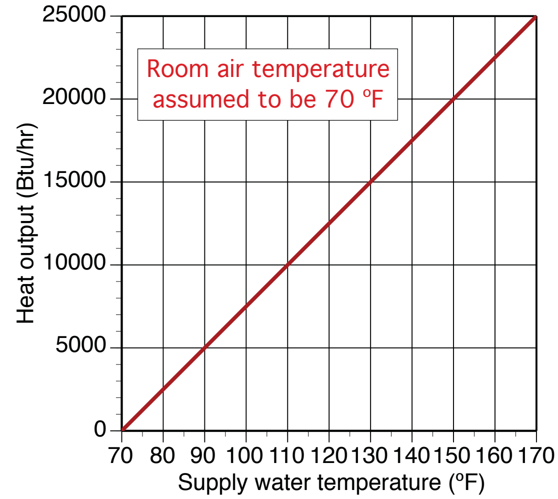


If water is supplied to the heat emitter at room temperature (whatever that temperature happens to be), there will be zero heat output from the heat emitter. As the temperature of the water supplied to the heat emitter climbs above room air temperature, its heat output also increases. This graph shows that when the supply water temperature is 50ºF above the room air temperature, the heat emitter will release 12,500 Btu/hr into the room. If the supply water temperature climbs to 100ºF above the room's air temperature, the heat emitter releases 25,000 Btu/hr into the room. Thus, doubling the temperature difference doubles the heat output.

The slope of the line in such a graph depends on the characteristics of the heat emitter. The greater the ability of the heat emitter to release heat at a given supply water temperature, the steeper the slope of the line.

If one assumes a desired room air temperature of 70ºF, the graph shown in Figure 10-9 can be modified to that shown in Figure 10-10.

Figure 10-10



The only difference is that the supply water temperature is now shown on the horizontal axis. This proportional relationship between supply water temperature and heat output holds true, as a reasonable approximation, for most hydronic heat emitters. It is an important characteristic in the context of another control technique called outdoor reset control.

OUTDOOR RESET CONTROL:

As outdoor temperatures change, so does the heating load of a building. Ideally, every heating system would continually adjust its rate of heat delivery to match its building's current rate of heat loss. This would allow inside air temperature to remain constant regardless of outside conditions. Outdoor reset control was developed to do this by changing the temperature of the water supplied to the system in response to outdoor temperature.

An outdoor reset controller continuously calculates the ideal "target" supply water temperature to a hydronic system. This temperature depends on the type of heat emitters in the system, as well as the current outdoor temperature. It therefore has the potential to change from one moment to the next.

Outdoor reset controllers use Formula 10-2 to calculate the target water temperature that (ideally) should be supplied to a given hydronic distribution system at any given time.

Formula 10-2

$$T_{target} = T_{indoor} + (RR) \times \left(T_{indoor} - T_{outdoor}\right)$$

Where:

T_{target} = the "ideal" target supply water temperature to the system
T_{indoor} = desired indoor air temperature
RR = reset ratio (slope of reset line)

The reset ratio (RR) is determined as follows:

Formula 10-3

$$RR = \frac{(T_{wd} - T_{wnl})}{(T_{anl} - T_{ad})}$$

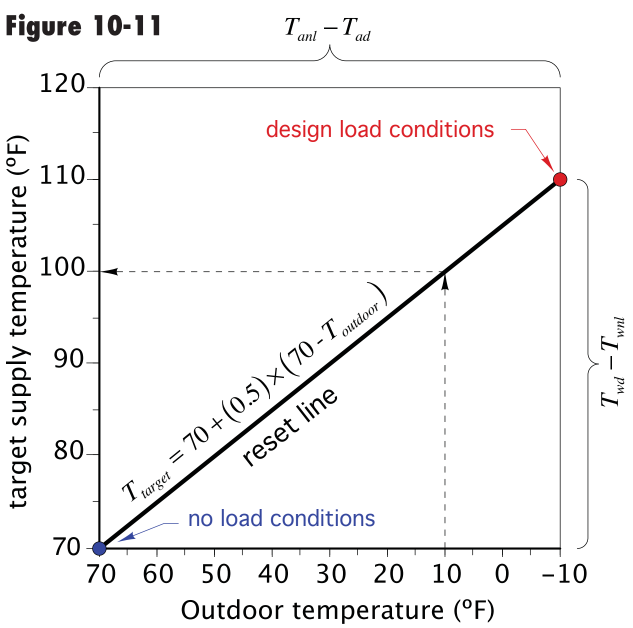
Where:

T_{wd} = required supply water temperature to distribution system at design load
T_{wnl} = water temperature supplied to distribution system at no load
T_{ad} = outdoor air temperature at design load
T_{anl} = outdoor air temperature at no load

The graph in Figure 10-11 is a good way to visualize these relationships.

The red dot in the upper right portion of the graph represents design load conditions (e.g., the coldest day of winter). The blue dot in the lower left corner represents no load conditions (e.g., where no heat output is needed from the heat emitters). The sloping line that connects these two dots is called the reset line.

Figure 10-11



$$RR = \frac{(T_{wd} - T_{wnl})}{(T_{anl} - T_{ad})} = \frac{(110 - 70)}{(70 - [-10])} = \frac{40}{80} = 0.5$$

$$T_{target} = 70 + (0.5) \times (70 - T_{outdoor})$$

The reset ratio (RR) is the slope of the reset line. It is found by dividing the vertical temperature difference between the red dot and blue dot by the horizontal temperature difference between the same dots. For the case represented in Figure 10-11, the reset ratio is:

$$RR = \frac{(T_{wd} - T_{wnl})}{(T_{anl} - T_{ad})} = \frac{(110 - 70)}{(70 - [-10])} = \frac{40}{80} = 0.5$$

The graph of a reset line can be used to find the target water temperature for any outdoor temperature. First, locate the outdoor temperature on the horizontal axis. Next, draw a vertical line up to intersect the reset line. Finally, draw a horizontal line from this intersection to the vertical axis and read the required target water temperature. For example, when the outdoor temperature is 10ºF, the reset line in Figure 10-11 indicates the target supply water temperature is 100ºF.

The target water temperature can also be determined by entering the outdoor temperature into the formula for the reset line. For the line shown in Figure 10-11, the target water temperature when the outdoor temperature is 10ºF would be:

$$T_{target} = 70 + (0.5) \times (70 - 10) = 70 + 30 = 100^{\circ} F$$

IMPLEMENTING OUTDOOR RESET CONTROL:

There are three ways to implement outdoor reset control in hydronic systems:

- Heat source reset (for on/off heat sources)
- Heat source reset (for modulating heat sources)
- Mixing reset

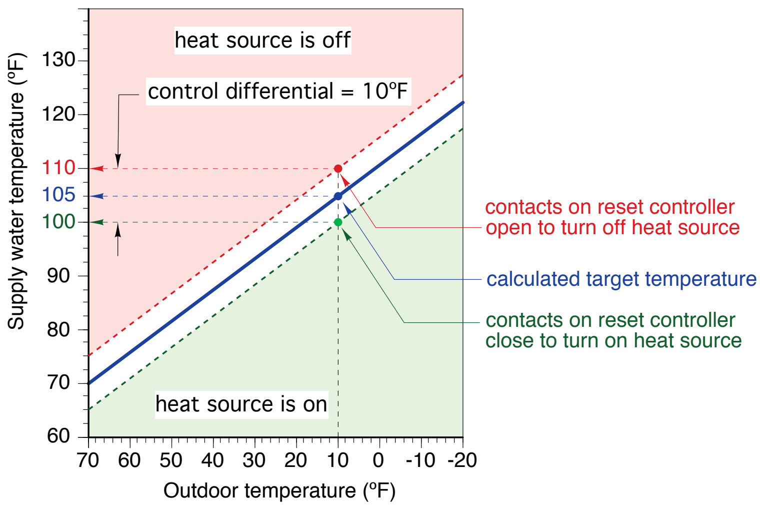
These techniques can be used individually or together.

HEAT SOURCE RESET (FOR ON/OFF HEAT SOURCES):

The temperature of water supplied to a distribution system can be controlled by turning a heat source such as a boiler or heat pump on and off.

The graph in Figure 10-12 illustrates the control logic used by a reset controller that controls an on/off heat source. The sloping blue line represents the "target temperature" (e.g., the ideal supply water temperature for the distribution system over a range of outdoor temperatures). For example, if the outdoor temperature

Figure 10-12



is 10ºF, the blue line in Figure 10-12 indicates a target temperature of 105ºF (as indicated by the blue dot).

When the reset controller is powered on, it measures both outdoor temperature and the current supply water temperature. It then uses the measured outdoor temperature, along with its settings, to calculate the target temperature. Then it compares the calculated target temperature to the measured supply water temperature.

If the measured supply water temperature is equal or close to the target temperature, no control action is taken. However, if there is sufficient deviation between these temperatures, the reset controller takes action.

The settings represented in Figure 10-12 would result in the following control actions when the outdoor temperature is 10ºF:

• If the water temperature supplied to the distribution system is above 100ºF (e.g., 105 less one half the differential of 10ºF), the contacts in the reset controller remain open and the heat source remains off.

• If the water temperature supplied to the distribution system is below 100ºF, the contacts in the reset controller close to turn on the heat source. Once turned on, the heat source would remain on until the water supplied to the distribution system reaches 110ºF (105 plus one half the control differential) or higher.

The 10ºF differential between the temperatures at which the heat source is turned on and off discourages short cycling. The value of the differential can be adjusted on most outdoor reset controllers. Smaller control differentials reduce the variation in supply water temperature both above and below the target temperature. However, if the control differential is too small, the heat source will cycle on and off excessively. This is not good for the life of components such as boiler ignition systems or heat pump compressors and contactors.

HEAT SOURCE RESET (FOR MODULATING BOILERS):

Many modern boilers, and even some heat pumps, can adjust their heat output over a relatively wide range. This is called modulation, and it allows the heat source to better match the heating load. Modulation reduces issues such as short cycling and temperature variations that are more common in systems using on/off heat sources.

Modulating heat sources usually have their own internal outdoor reset controllers. They continually measure outdoor temperature, calculate the target supply water temperature and compare it to the measured supply water temperature. Deviations between these temperatures cause the internal reset controller to regulate the speed of the combustion air blower on a modulating boiler or compressor speed on a modulating heat pump. The goal is to keep the measured supply temperature very close to the calculated target temperature.

MIXING RESET:

Outdoor reset control can also be implemented by a mixing assembly such as 3-way and 4-way mixing valves or a variable-speed injection mixing pump. The logic used for mixing reset is similar to that already described for modulating heat sources. The outdoor temperature is measured. The target temperature is then calculated. The measured supply temperature is then compared to the target temperature. The deviation between these temperatures determines the output signal from the reset controller to the mixing assembly. The goal is the same—to keep the water temperature supplied to the distribution system at, or very close to, the calculated target temperature.

idronics #7 provides a complete description of the benefits of outdoor reset control. It also shows several ways to implement outdoor reset control using various mixing hardware.

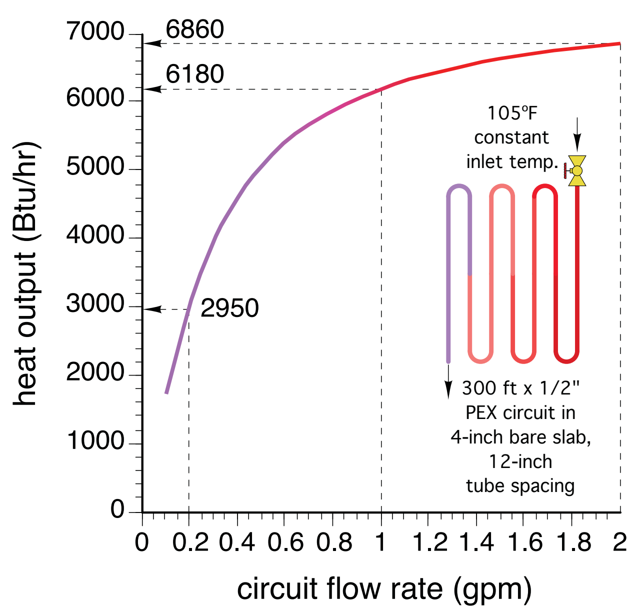
CONTROLLING HEAT OUTPUT USING FLOW RATE: The flow rate through any heat emitter affects its heat output. The following principle always applies:

The faster a heated fluid passes through a heat emitter, the greater the rate of heat transfer, all other conditions being equal.

It might seem intuitive to assume that heat transfer from a heat emitter increases in proportion to flow rate through it (i.e., doubling the flow rate through the heat emitter would double its heat output). Unfortunately, this is not true. The rate of change of heat output from any hydronic heat emitter is a strong function of flow rate. At low flow rates, heat output rises rapidly with increasing flow, but the greater the flow rate becomes, the slower the rate of increase in heat output.

To illustrate this, consider the situation shown in Figure 10-13, which shows the heat output of a typical radiant floor circuit versus the flow rate through it. This circuit is supplied with water at a constant temperature of 105ºF.

Figure 10-13

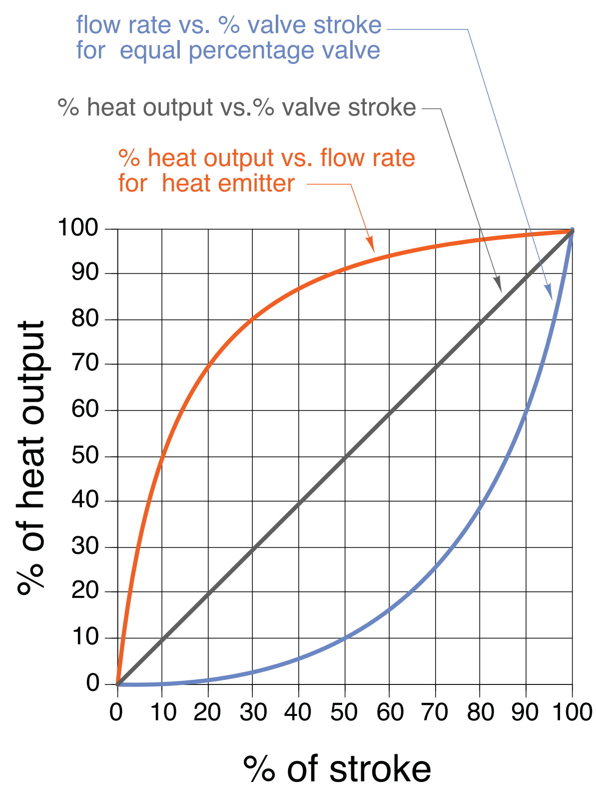


The circuit's maximum heat output of 6,860 Btu/hr occurs at the maximum flow rate shown (2.0 gpm). Decreasing the circuit's flow rate by 50% (to 1.0 gpm) decreases its heat output to 6,180 Btu/hr, a drop of only about 10%. Reducing the flow rate to 10% of the maximum value (to 0.2 gpm) still allows the circuit to release 2,950 Btu/hr, about 43% of its maximum output.

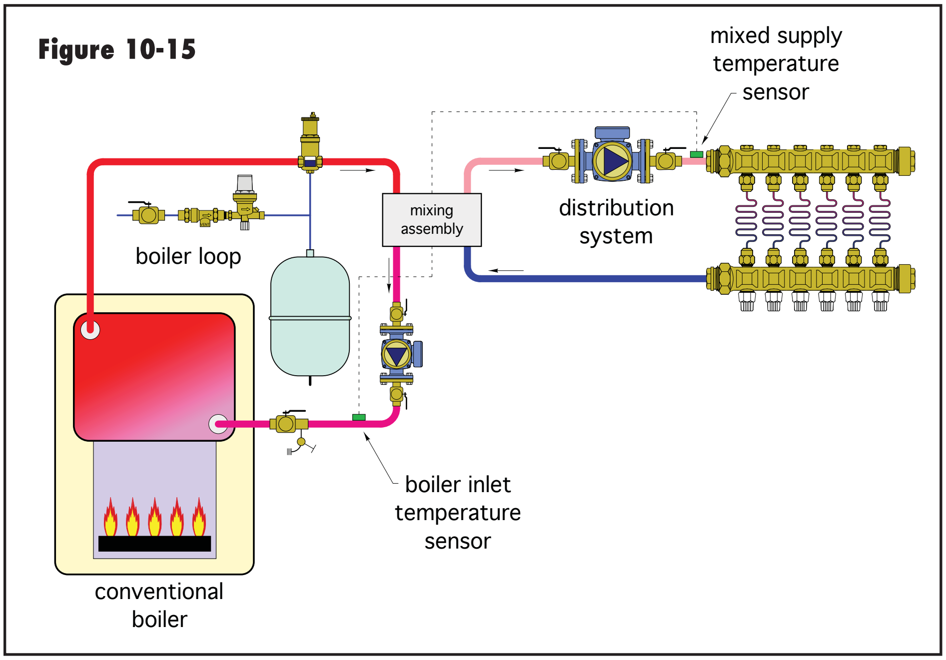
This "non-linear" relationship between heat output and flow is typical of all hydronic heat emitters. It tends to make adjusting heat output with a valve more complicated than what one might assume. For example, as a technician first begins closing a standard globe valve that is used for balancing a hydronic circuit, there is relatively little change in the heat output of the circuit. However, when that valve is only 10% to 25% open, small adjustments will yield relatively large changes in heat output.

Special types of valves with equal percentage characteristics have been developed to compensate for the rapid rise in heat output low flow rates. These valves allow flow rate to rise very slowly as they first start to open. The farther the valve stem is opened, the faster flow through the valve develops. The resulting combination provides a more proportional relationship between the position of the valve stem and the rate of heat output from the heat emitter being regulated by the valve, as shown in Figure 10-14.

Figure 10-14



idronics 8 provide a detailed discussion of how equal percentage valves are constructed and used to control heat output.



MIXING FUNDAMENTALS:

The water temperature supplied to a hydronic distribution system is often controlled by a device that blends hot water from the heat source with cooler water returning from the distribution system. An example would be water supplied at 140ºF from the storage tank of a solar thermal system being blended with water at 90ºF returning from a radiant floor distribution system in proportions that create a supply water temperature of 110ºF back to the distribution system.

In hydronic systems, mixing takes place within a "mixing assembly." This assembly can be composed of several different devices, such as valves, controllers or variable-speed circulators. However, the thermodynamics of mixing are independent of the specific hardware used within the mixing assembly. Thus, the mixing assembly can be considered as a "bridge" between the heat source loop and the distribution system, as shown in Figure 10-15. Notice that all heat passing from the heat source to the distribution system must pass through the mixing assembly.

PURPOSE(S) OF MIXING:

One purpose of all mixing assemblies is to provide the proper supply water temperature to the distribution system.

When a conventional boiler (e.g., a boiler that requires protection against sustained flue gas condensation) serves as the heat source, the mixing assembly must also provide that protection. It typically does so by sensing the water temperature entering the boiler. If that temperature is at or below some present minimum value (usually around 130ºF for gas-fired boilers), the mixing assembly restricts the rate at which heat is allowed to pass from the boiler loop to the distribution system. Providing this protection requires the mixing assembly to continually monitor boiler input temperature.

THERMODYNAMICS OF MIXING:

The first law of thermodynamics states that energy cannot be created or destroyed—only changed in form. This principle always holds true when flow streams are mixed together in hydronic heating systems. In such applications, it is convenient to express the first law as follows:

The rate of heat flow into a mixing assembly must equal the rate of heat flow out of the assembly.

It is also true, based on a principle called conservation of mass, that the flow rate of an incompressible fluid entering any fluid-filled component in a hydronic system must equal the flow rate leaving that component. Common hydronic system fluids like water or a mixture of water and antifreeze are incompressible, and thus this relationship must hold true. For a mixing assembly, it can be stated as:

The total flow rate of fluid entering a mixing device must equal the total flow rate of fluid leaving the mixing device.

When these two physical relationships are combined mathematically, the result is a simple but powerful formula for determining the temperature on the outlet side of any mixing device. It is stated in Formula 10-4:

Formula 10-4

$$T_{mix} = \frac{(T_1 \times f_1) + (T_2 \times f_2)}{f_1 + f_2}$$

Where:

T_{mix} = resulting temperature of the mixed fluid stream
T_1 = entering temperature of fluid stream #1
T_2 = entering temperature of fluid stream #2
f_1 = flow rate of fluid stream #1
f_2 = flow rate of fluid stream #2

Figure 10-16

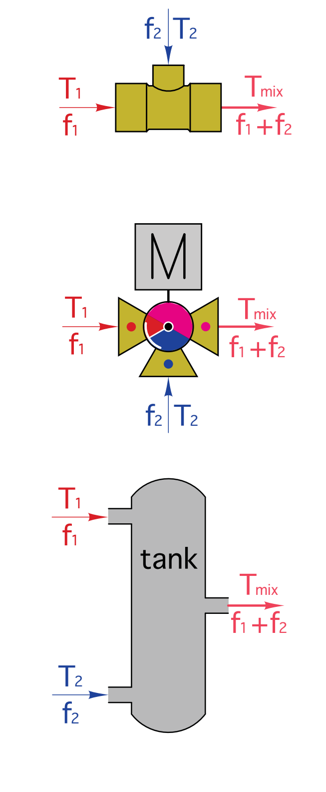
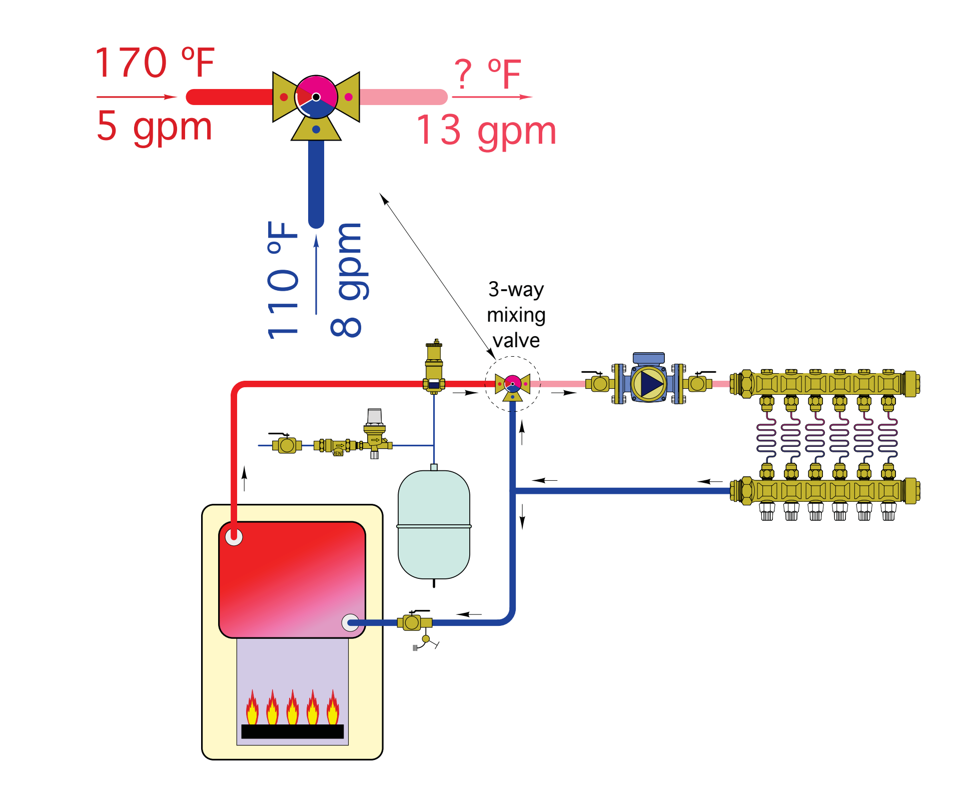


Figure 10-17



The temperatures and flow rates used in Formula 10-4 can be expressed in any consistent system of units. This formula applies to water and other fluids, such as antifreeze solutions, provided the same type of fluid enters both inlets to the mixing assembly.

This formula also holds true regardless of the physical device in which the fluid streams come together. It applies to a tee, a tank, or any type of mixing valve, as illustrated in Figure 10-16.

Example: Hot water is supplied from a conventional boiler at a temperature of 170ºF and flow rate of 5 gpm. It flows into the hot port of a 3-way mixing valve as shown in Figure 10-17. Cooler water returns from the distribution system at 110ºF and 8 gpm. Determine the mixed supply water temperature to the distribution system under these conditions.

Solution: Enter the values for temperature and flow rate of each entering fluid stream entering into Formula 10-4 and solve:

$$T_{mix} = \frac{(T_1 \times f_1) + (T_2 \times f_2)}{f_1 + f_2} = \frac{(170 \times 5) + (110 \times 8)}{5 + 8} = 133^\circ F$$

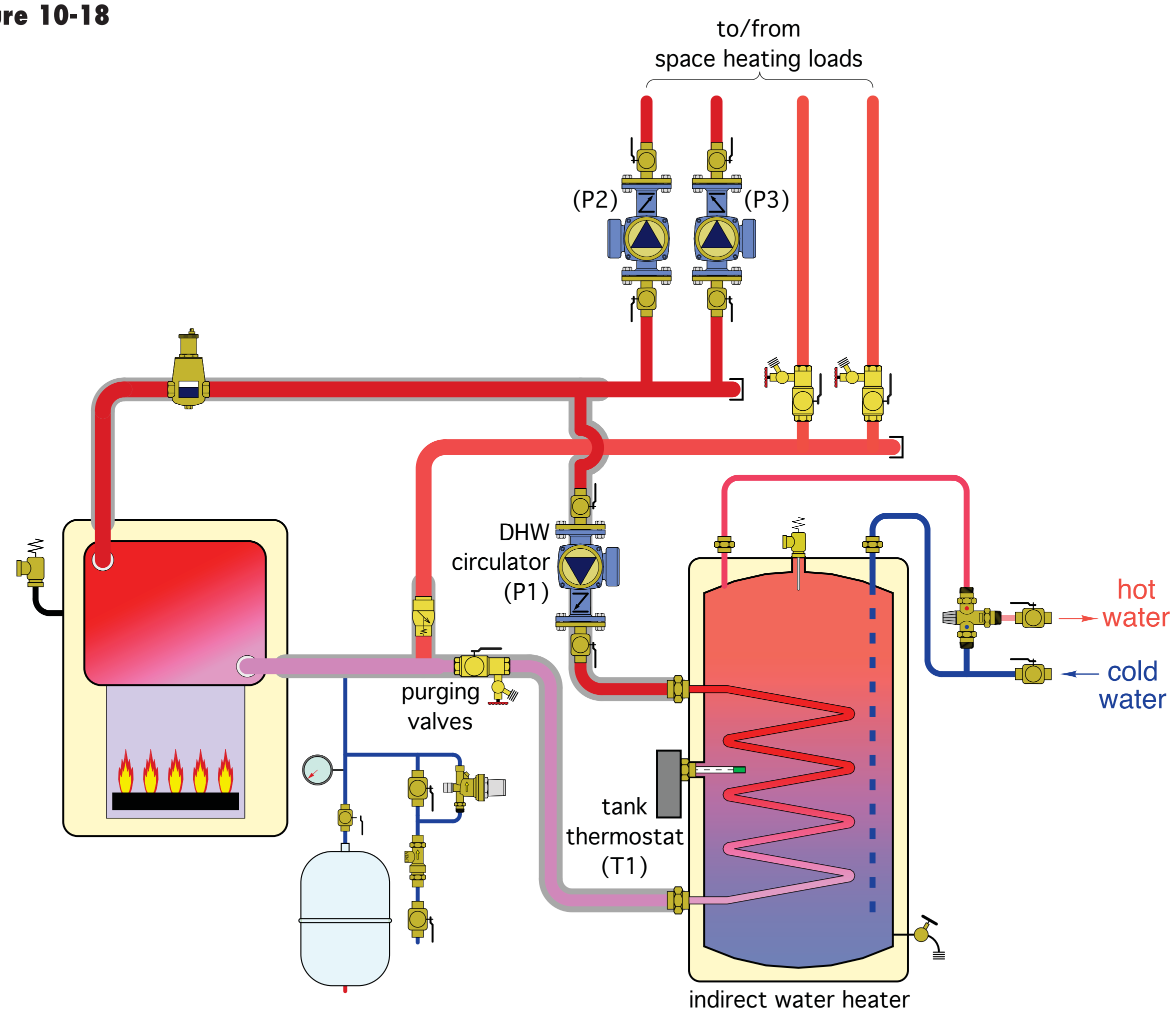
idronics #7 provides a more in-depth discussion of mixing, including a wide variety of hardware devices that can be used to build the mixing assembly.

PRIORITIZED LOADS:

There are many hydronic heating systems that supply more than just space heating. Some common “ancillary” loads include domestic water heating, snowmelting and pool heating. When designing the heat source for such systems, some designers add up all the heating capacity required by all of these loads, assuming they might operate at the same time, then size the heat source to supply the total.

Although this practice might seem intuitive, it is very seldom necessary. The diversity of when these loads require heat, as well as the timespan over which they can accept heat, often allows the heat source to be sized significantly smaller than the total of all connected loads. This, in combination with “prioritized load management,” allows for lower installed cost and higher operating efficiency.

Figure 10-18



Prioritized load management refers to assigning a specific "priority" to each of the connected loads. The higher the priority of the load, the sooner it will receive available heat from the heat source.

One of the most common scenarios in hydronic systems is assigning top priority to domestic water heating, along with subsequent priority to space heating. Loads such as pool heating are generally assigned lower priority because their high thermal mass allows for gradual temperature changes over several hours. Thus, they can accept heat from the heat source at almost any time of day without undergoing temperature changes of more than a few degrees.

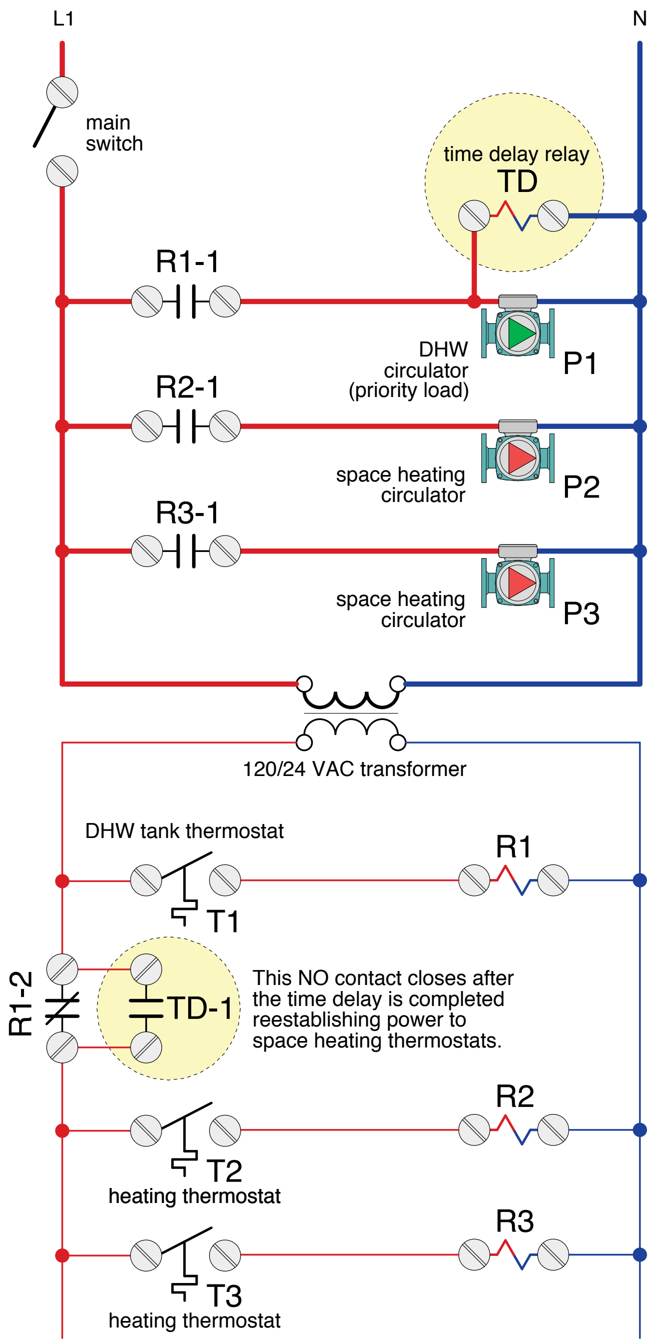
Priority control can be accomplished using hard-wired relay logic or with a prebuilt multi-zone relay center that includes priority logic. The latter is the fastest and easiest way to implement priority control.

Consider the system shown in Figure 10-18. It consists of two space heating zones, each operated by independent circulators (P2) and (P3). Domestic hot water is provided by an indirect water heater with internal heat exchanger. Circulator (P1) operates to move hot boiler water through this heat exchanger whenever the tank's thermostat calls for heat.

If priority is to be given to domestic water heating, circulators (P2) and (P3) must turn off when circulator (P1) is operating. The ladder diagram in Figure 10-19 shows the relay logic used to do this.

Upon a call for heat from the tank thermostat (T1), the coil of relay (R1) is powered on by the 24 VAC circuit. Relay contact (R1-1) closes to supply line voltage to the DHW circulator. This contact also supplies 120 VAC to power-on a time delay relay (TD-1). This time delay relay uses a "delay-on-make" configuration. As such, its normally

Figure 10-19



open contacts will only close *after a preset time has elapsed.* For priority DHW control, this time is typically set for 30 minutes.

Another relay contact associated with relay (R1) is located in the low voltage portion of the ladder diagram. The contacts (R1-2) are normally closed. As such, they open when relay (R1) is powered on. This interrupts 24 VAC to all space heating thermostats, which turns off any active space heating zones.

At this point, the domestic water heating load is active, all space heating zones are off and the timer in relay (TD-1) is counting towards 30 minutes. If the domestic water thermostat is satisfied within 30 minutes, relay (R1) is turned off and 24 VAC is resupplied to the space heating thermostats. If the domestic water thermostat is not satisfied within 30 minutes, the normally open contact (TD-1), wired in parallel with contact (R1-2), closes. This re-establishes 24 VAC to the space heating thermostats *and* allows the domestic water heater to continue operating. This latter function is called priority override. It prevents a long-term “lockout” of space heating in the event that the domestic water heating load cannot be satisfied.

idronics #11 provides a more in-depth discussion of prioritized loads in combination with high capacity, hydronic-based domestic water heating.

11: EXAMPLES OF MODERN HYDRONIC SYSTEMS

Previous sections have described the building blocks of modern hydronic systems. This section combines the principles and hardware discussed in previous sections into full systems.

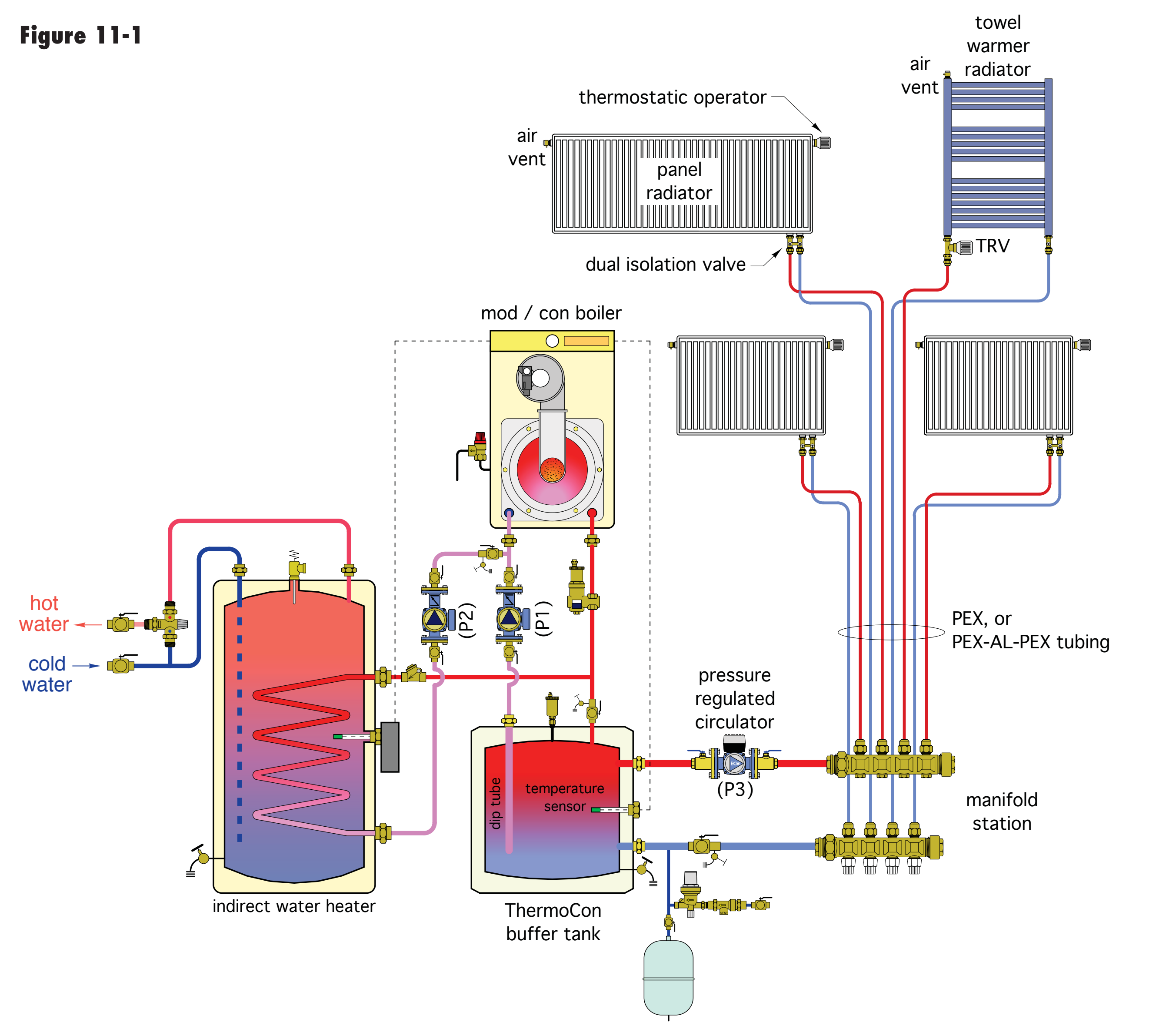
Although there are virtually unlimited ways to combine design concepts and hardware into working systems, the designs presented in this section are focused on the most modern and cost-effective concepts. These designs are also built around common residential and light commercial building requirements.

SYSTEM #1:

This system uses a mod/con boiler in combination with a manifold-based distribution system serving panel radiators and a towel warmer radiator. The heat output of each panel radiator and the towel warmer is regulated by a thermostatic radiator valve.

A 25-gallon Caleffi ThermoCon buffer tank interfaces the boiler to the distribution system. It provides hydraulic separation between the boiler circulator (P1) and the variable-speed pressure-regulated distribution circulator (P3). The thermal mass of this tank also helps prevent boiler short cycling when only one of the panel radiators is operating.

Figure 11-1



The boiler is operated by its internal outdoor reset controller. This controller monitors the temperature sensor installed in a well within the buffer tank whenever the system is "heating enabled." The boiler operates as necessary to maintain the buffer tank at an appropriate temperature based on outdoor temperature. Thus, whenever the system is heating enabled, warm water is available to flow to any panel radiator with a partially open (or fully open) thermostatic valve.

The distribution circulator (P3) is set for constant differential pressure control. As such, it maintains a consistent pressure difference between the supply and return manifolds. As more thermostatic radiator valves open, the differential pressure across the manifolds attempts to drop. However, the circulator senses this and compensates by immediately increasing its speed so that the set differential pressure is restored. The value of this pressure differential would be that required when all the thermostatic radiator valves are fully open.

Domestic water heating is handled as a prioritized load based on the logic within the boiler circuitry. When the thermostat monitoring the water temperature in the indirect water heater calls for heat, the control logic within the boiler automatically stops supplying heat to the buffer tank by turning off circulator (P1). The boiler targets a higher setpoint temperature that has been preset for domestic water heating mode. The boiler's internal control circuitry also turns on circulator (P2) to create flow through the heat exchanger of the indirect water heater. The system remains in this mode until either the water temperature in the indirect tank reaches the desired setpoint or the time limit on the domestic water heating mode expires, in which case space heating resumes.

There is no need to stop circulator (P3) from operating during the domestic water heating mode. It can draw upon the thermal mass of the buffer tank to supply heat to the radiators.

The panel radiators are sized so that they can supply design load heat output at a supply water temperature of 130ºF, and flow rates that correspond to a design load temperature drop of 20ºF. Thus, their average water temperature at design load is 120ºF. The relatively low water temperatures allow the mod/con boiler to operate with sustained flue gas condensation, and thus high thermal efficiency, under both design load and partial-load conditions. Selecting the radiators around a low design water temperature also "future-proofs" the distribution system. It enables other low-temperature heat sources such as solar collectors or heat pumps to potentially be used as a replacement for the original heat source.

The "heat-enabled" status of the system can be invoked several ways. One is to enable the system whenever the outdoor temperature drops below a preset value, such as 60ºF. Some boiler controllers provide this function. Another is to install a manually operated switch in the living space, and thus turn control over to the occupants. A third way is to enable heating using a "master thermostat" located within the living space. Whenever the indoor temperature drops slightly below the setting of this master thermostat, its contacts close to enable the system to begin heating operation.

In summary, this system takes advantage of modern concepts such as a high thermal efficiency mod/con boiler, low wattage pressure-regulated distribution circulator, outdoor reset control, small-diameter flexible distribution tubing, prioritized domestic water heating and "wireless" room-by-room zoning using thermostatic radiator valves.

idronics #5 provides a broad discussion of manifold-based distribution systems, as well as variable-speed pressure-regulated circulators.

SYSTEM #2:

When a building contains multiple types of heat emitters, it is often necessary to create a distribution system that can simultaneously deliver multiple water temperatures. The schematic in Figure 11-2 is an example of such a system.

Heat is supplied by a conventional gas-fired boiler. As discussed in section 6, such boilers require protection from sustained flue gas condensation. This protection is provided by a high flow capacity thermostatic mixing valve (Caleffi 280). When the boiler undergoes a cold start, this valve recirculates water leaving the boiler's outlet port, back to its inlet port. No hot water is allowed to pass farther into the distribution system. This allows the boiler to warm above the dewpoint temperature of its flue gases as soon as possible and remain above that dewpoint for the remainder of its operating cycle.

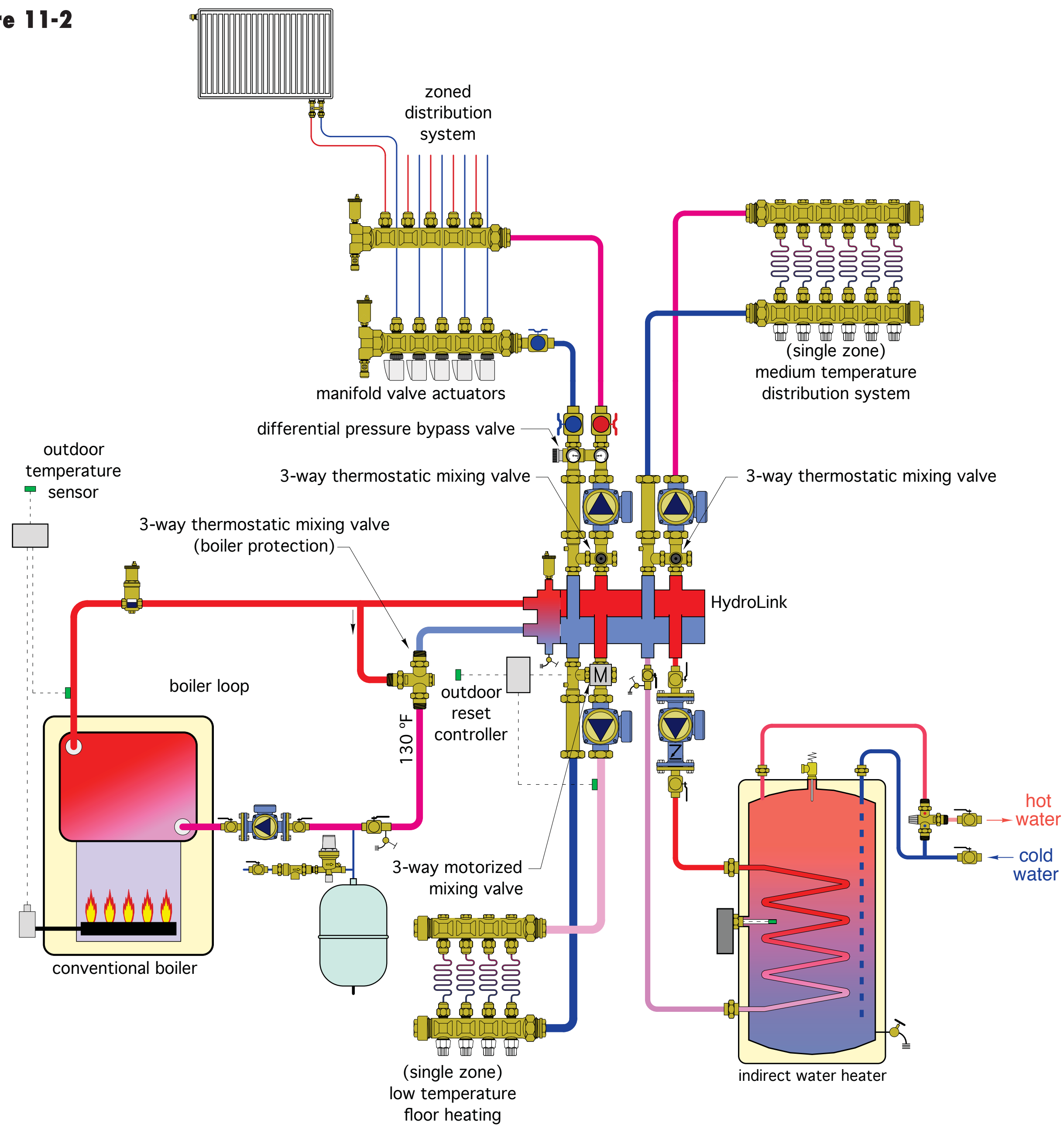
Once the boiler is operating at a temperature hot enough to prevent flue gas condensation, the thermostatic valve allows hot water to flow onward to the Caleffi HydroLink. This device combines a hydraulic separation chamber with two high flow capacity manifold chambers. All heating loads are supplied from and returned to the HydroLink. The hydraulic separation chamber on the left side of the HydroLink prevents the load circulators from interfering with each other.

The indirect domestic water heater is supplied directly from the HydroLink. Thus, it received water at or close to the temperature of the water leaving the boiler. Domestic water heating is treated as the priority load. The other three load circulators would not operate while domestic water heating is active. Also note that the circulator supplying the indirect water heater is equipped with an internal check valve to prevent reverse circulation when the water heating load is off, but other loads are on.

The three space heating subsystems each operate at different water temperatures. This schematic shows three possible mixing assemblies that could be used to create these temperatures.

The panel radiators are supplied from a manifold station that is equipped with electric valve operators on each circuit. These operators allow flow through their respective circuits when energized with a 24 VAC electrical signal from associated thermostats. Because different circuit valves

Figure 11-2



will be open at different times, it is necessary to regulate differential pressure across the manifold. This is handled by a differential pressure bypass valve that is integrated into the mixing assembly (Caleffi 163 HydroMixer) attached to the HydroLink. The 163 HydroMixer also reduces the water temperature supplied to the manifold using an integrated thermostatic mixing valve.

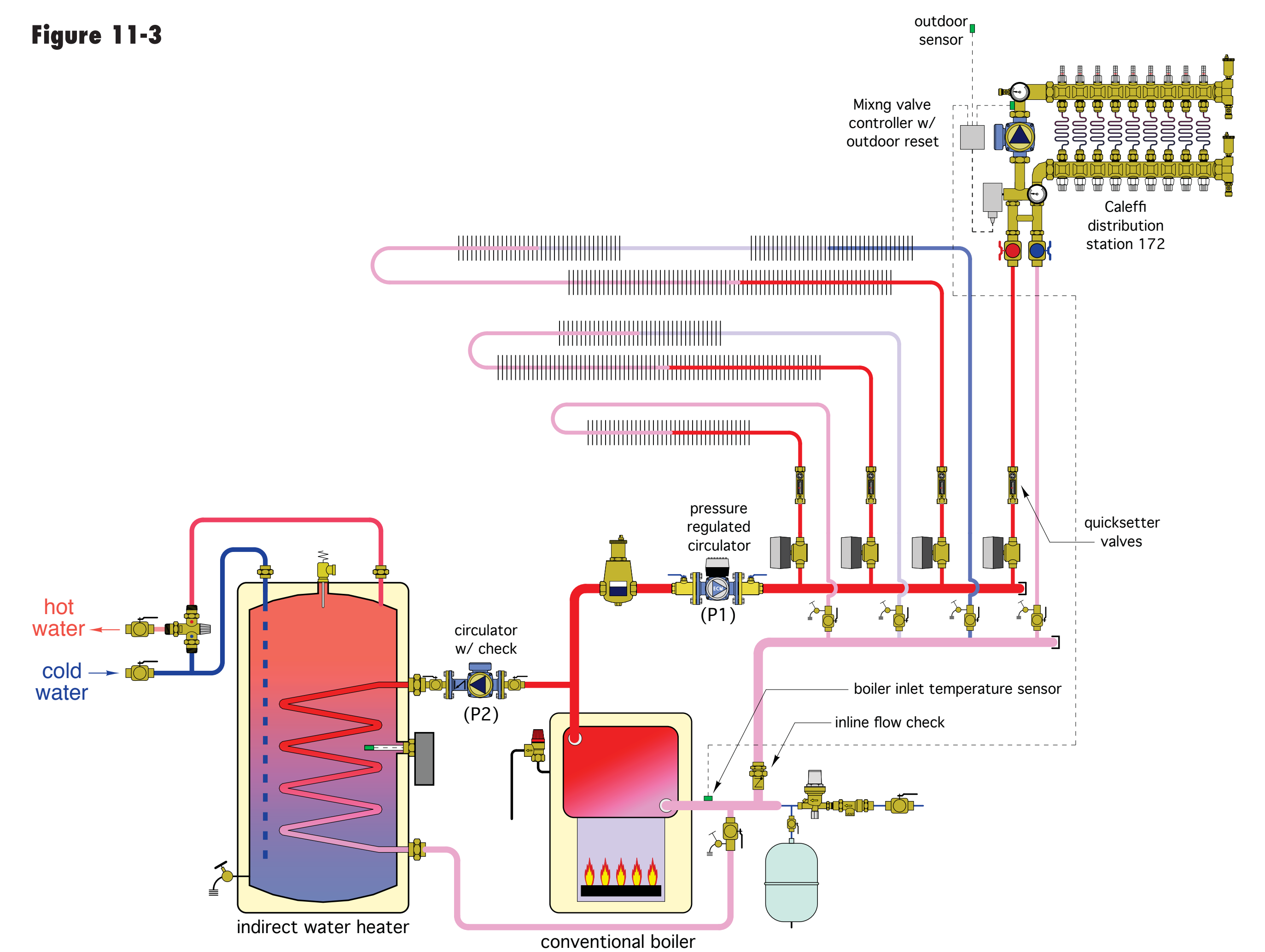
The medium-temperature radiant panel circuits in the upper right corner of the schematic operate as a single zone. As such, no differential pressure regulation is necessary. Mixing is again handled by a Caleffi 163 HydroMixer, but without the differential pressure bypass subassembly.

The lower temperature radiant panel circuits in the lower left of the schematic also operate as a single zone. The mixed water temperature supplied to these circuits is created by a Caleffi 167 HydroMixer, which uses a motorized mixing valve and floating control actuator rather than a thermostatic mixing valve. This allows full reset of the supply water temperature based on the current outdoor temperature. It also allows the possibility of constant circulation through the radiant panel circuits, which is preferred by some designers. An outdoor reset controller that supplies a 3-wire floating output signal is used to operate the motorized actuator on the 167 HydroMixer.

***idronics* #1 provides more information on the HydroLink.**

***idronics* #7 provide more information on mixing in hydronic systems.**

Figure 11-3



SYSTEM #3:
In residential applications, it's common to see a conventional boiler supplying low-temperature floor heating to a basement slab, while also supplying higher temperature water to heat emitters, such as fin-tube baseboard, on the first and second floors. The need for domestic hot water rounds out the requirement. The system shown in Figure 11-3 provides a state-of-the-art solution.

This system contains 3 zones of fin-tube baseboard, each controlled by zone valve. The zones have differing amounts of baseboard, and thus likely need to operate at different flow rates. Each circuit is therefore equipped with a Caleffi 132 QuickSetter, which combines a flow meter with a balancing valve. These valves can be adjusted as necessary to achieve the required "design" flow rates in each circuit.

The low-temperature floor heating circuits in the basement are supplied by a Caleffi 172 distribution station. This station combines a motorized 3-way mixing valve, circulator, supply and return manifolds and a flow bypass connection that provides hydraulic separation between the circuit supplying hot water to the station and the circulator within the station. An outdoor reset controller with a floating output signal is used to control the motorized 3-way mixing valve. This controller also measures the water temperature entering the boiler, and if necessary, reduces hot water flow into the 3-way mixing valve to ensure the boiler does not operate with sustained flue gas condensation. Note the position of the boiler inlet temperature sensor. In this location, the sensor detects how *all* operating loads affect the boiler inlet temperature.

Flow to all space heating circuits is provided by a variable-speed pressure-regulated circulator that is set to maintain a constant differential pressure between the supply and return manifolds. As zone valves open up, this circulator automatically increases speed to maintain the fixed differential pressure, and vice versa.

Flow to the indirect water heater is provided by a fixed-speed circulator. If domestic water heating is to be a priority load, the space heating circulator (P1) is turned off during the domestic water heating cycle. If no load priority is used, circulators (P1) and (P2) can operate simultaneously. Either scenario is possible depending on the preferences of the designer and occupants.

The inline flow check valve on the return header prevents heat migration into that header. This is especially beneficial in warm weather, when domestic water heating is likely to be the only active load.

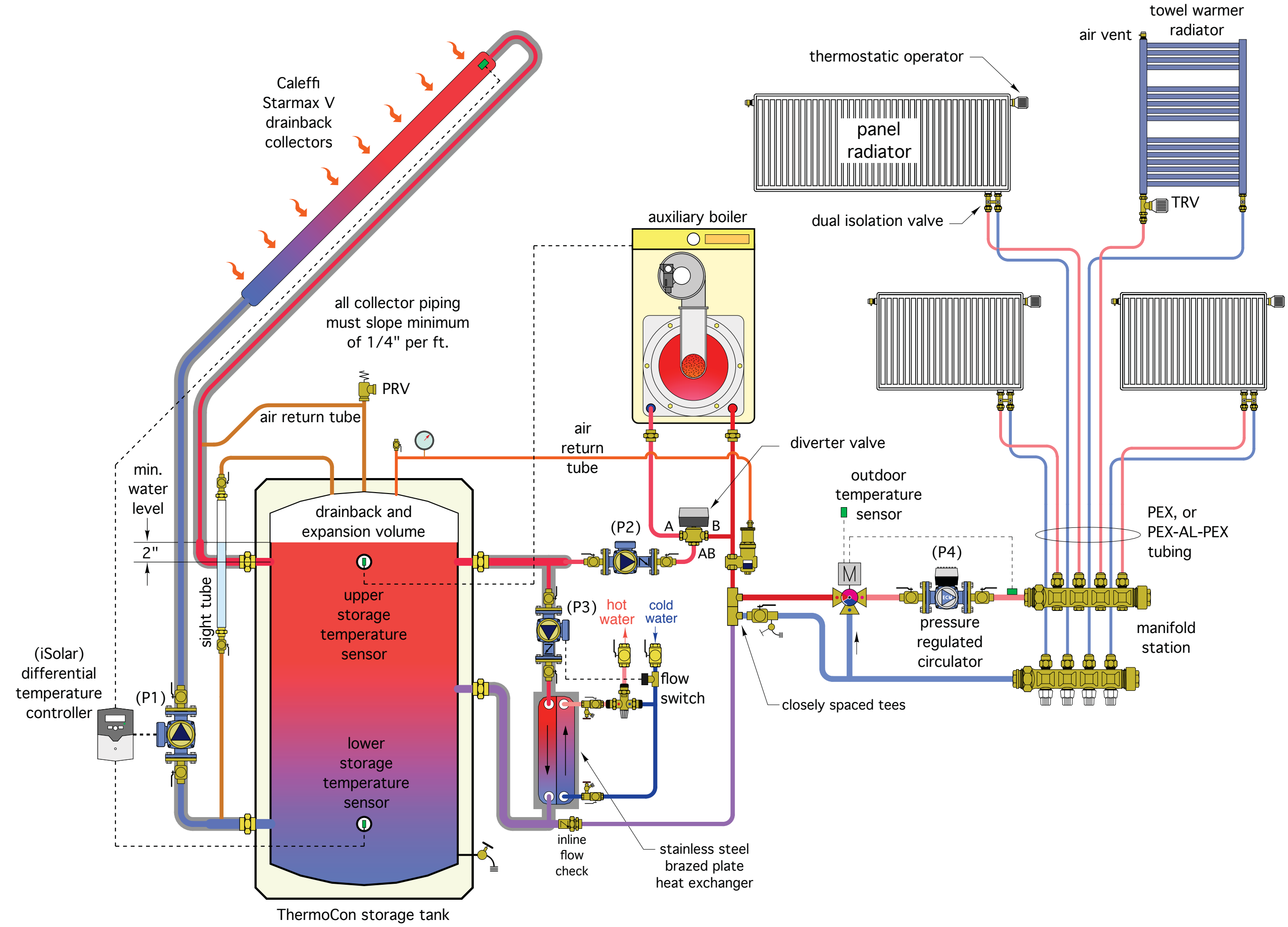
SYSTEM #4:
The benefits of hydronic systems can be merged with several types of renewable energy heat sources. One of the most common applications is to use an array of solar thermal collectors as the heat source for a system that provides both space heating and domestic hot water. Such systems are called "combisystems" and typically include an auxiliary heat source for times when the solar collector array cannot provide all the heat needed. The system in Figure 11-4 is an example of a solar thermal combisystem.

This system uses drainback freeze protection of the collector array. When the differential temperature controller senses that the collectors are a few degrees warmer than the water in the lower portion of the Caleffi ThermoCon storage tank, it turns on the collector circulator (P1). Water is driven up through the collector array, and air is pushed back down into the upper portion of the storage tank. Flow is sustained between the lower portion of the storage tank and the collector array, as long as the latter is a few degrees warmer than the lower storage tank. When this condition is no longer true, the collector circulator turns off and water in the collector array drains back to the storage tank. Air from the upper portion of the storage tank moves back into the collector array and associated piping. It is imperative that all piping between the collector array and storage tank be sloped a minimum of ¼ inch per foot to ensure complete drainage. All collector circuit piping should also be insulated to minimize heat loss, especially to unconditioned space.

The water temperature at the top of the storage tank is monitored by the auxiliary boiler. This boiler fires when necessary to maintain the upper portion of the storage tank at a temperature high enough to supply domestic water at the expected temperature. If solar input to the tank is sufficient to maintain this temperature, there is no need for the boiler to operate.

Domestic water is heated instantaneously whenever a hot water faucet is opened. When the domestic water flow rate reaches 0.5 gpm, the contacts of a flow detecting switch close. This completes an electrical circuit that allows the circulator (P3) to operate. Hot water from the top of the storage tank immediately flows through the left side of the stainless steel brazed plate heat exchanger, as cold domestic water flows into the right side of this heat exchanger. Heat is immediately transferred to the entering cold water, and it leaves the heat exchanger at a temperature high enough for delivery to the faucets. An ASSE 1017 thermostatic mixing valve with a maximum outlet temperature setting of 120ºF ensures that scalding hot water will not be delivered to the faucets in the event

Figure 11-4



the storage tank is at a very high temperature. When the faucet is closed, the flow switch opens its contacts and circulator (P3) turns off.

When there is a demand for space heating, circulator (P2) operates to move hot water from the top of the storage tank through the diverting valve (from port AB to port B) and on to a pair of closely spaced tees. These tees provide hydraulic separation between circulator (P2) and circulator (P4). Hot water is drawn into the 3-way motorized mixing valve, which operates based on outdoor reset control to provide the required supply water temperature to the manifold-based distribution system. (P4) is a variable-speed pressure-regulated circulator that maintains a constant differential pressure across the manifold station that supplies each panel radiator. Flow through each radiator is managed by a thermostatic radiator valve.

As the storage tank temperature drops, the boiler fires and the diverter valve reroutes flow through the boiler (from port AB to port A). Boiler operation can be simultaneous with either space heating or domestic water heating, or it could occur when neither load is active, simply to maintain the tank at a suitable minimum temperature for domestic water heating. Circulator (P2) operates whenever the boiler is firing.

The thermal mass of the heated water in the storage tank provides excellent buffering of the zoned distribution system, and thus prevents the boiler from short cycling. This thermal mass also provides reserve capacity for domestic water heating.

Any air captured by the vertical air separator is returned to the storage tank through the air return tube. This is an essential detail. It represents “air control” rather than “air elimination" and helps maintain the slight air pressurization within the system. If necessary, air can be added to the top of the storage tank to maintain this pressurization.

The captive air volume at the top of the tank also absorbs the expansion volume changes of the system's water as its temperature varies. There is no need of a separate expansion tank. Also note that there is no automatic makeup water subassembly within this system. Instead, water can be added or removed through a drain valve at the bottom of the storage tank. The water level within the tank is monitored by a transparent sight tube seen at the left of the storage tank.

idronics #6 provides a broad discussion of drainback-protected as well as other types of solar thermal combisystems.

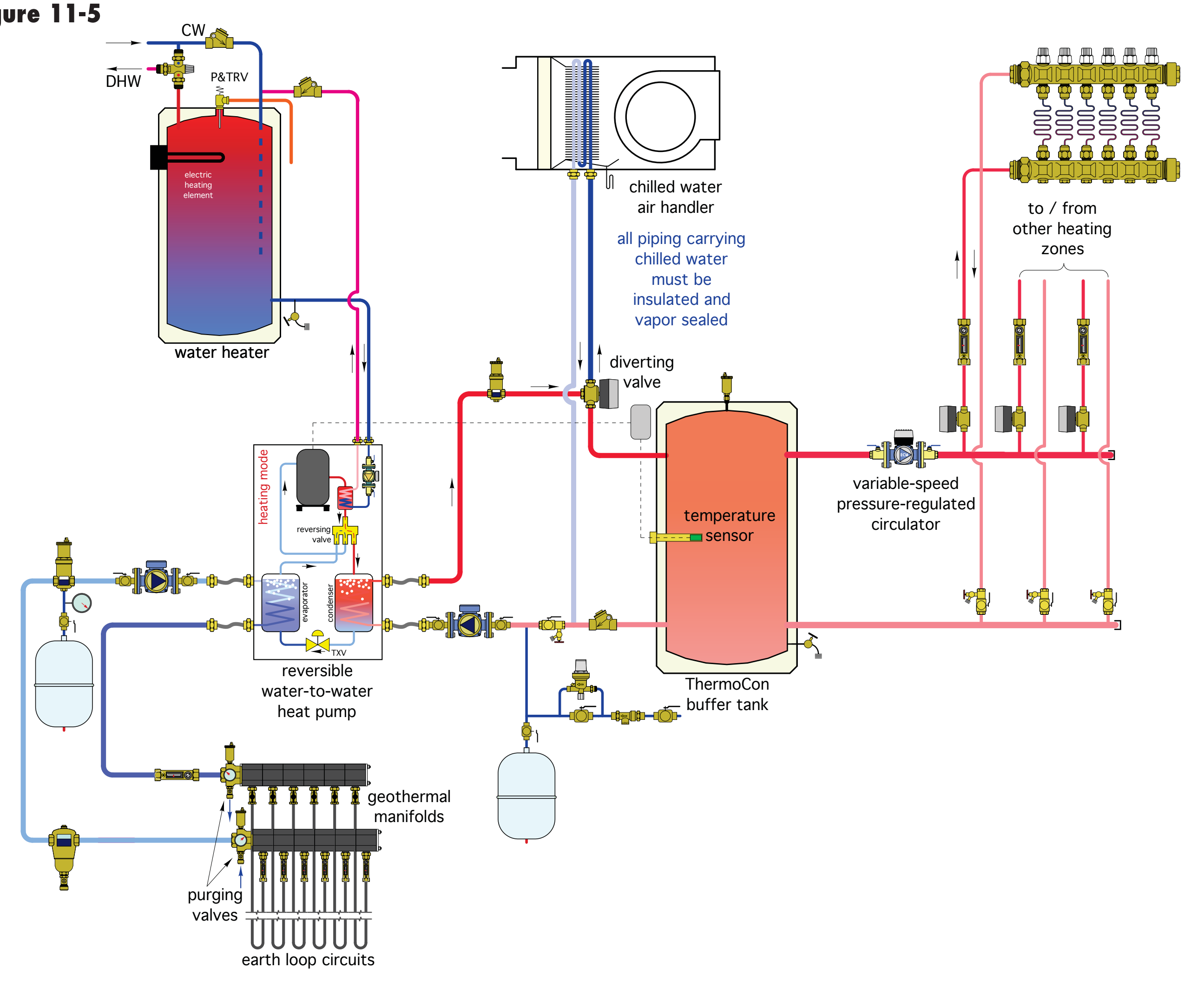
SYSTEM #5:

Geothermal heat pumps can provide both heating and cooling. A reversible water-to-water geothermal heat pump is well-suited as the heating and cooling source for a hydronic distribution system. The system shown in Figure 11-5 shows one example.

This system allows for multiple heating zones, along with a single zone of cooling. The latter is accomplished using an appropriately sized chilled water air handler equipped with a drip pan.

Whenever an on/off heat pump is used as the heat source for a zoned hydronic distribution system, a buffer tank must be used to prevent the heat pump from short cycling. The buffer tank stabilizes the system when the rate of heat generation by the heat pump is significantly different from the rate of dissipation by the zoned distribution system.

Figure 11-5



The buffer tank also provides hydraulic separation between the circulator on the load side of the heat pump and the variable-speed circulator in the distribution system.

The size of the buffer tank depends on the acceptable minimum run time of the heat pump and the allowed temperature differential of the tank between when the heat pump turns on and when it turns off. Appendix B gives the necessary formulas for determining the size of the buffer tank.

The heat pump is turned on and off by an outdoor reset controller that monitors the temperature of the buffer tank whenever the system is in heating mode. Whenever the temperature of the storage tank drops below a calculated lower limit, the heat pump and its two associated circulators are turned on. When the tank reaches a calculated upper temperature limit, the heat pump and circulators are turned off.

Because the buffer tank is not active during the cooling mode, the total cooling capacity of the air handler must be matched to the cooling capacity of the heat pump at a chilled water supply temperature not lower than 45ºF. This will allow the air handler to provide good moisture removal from the air and prevent the heat pump from short-cycling.

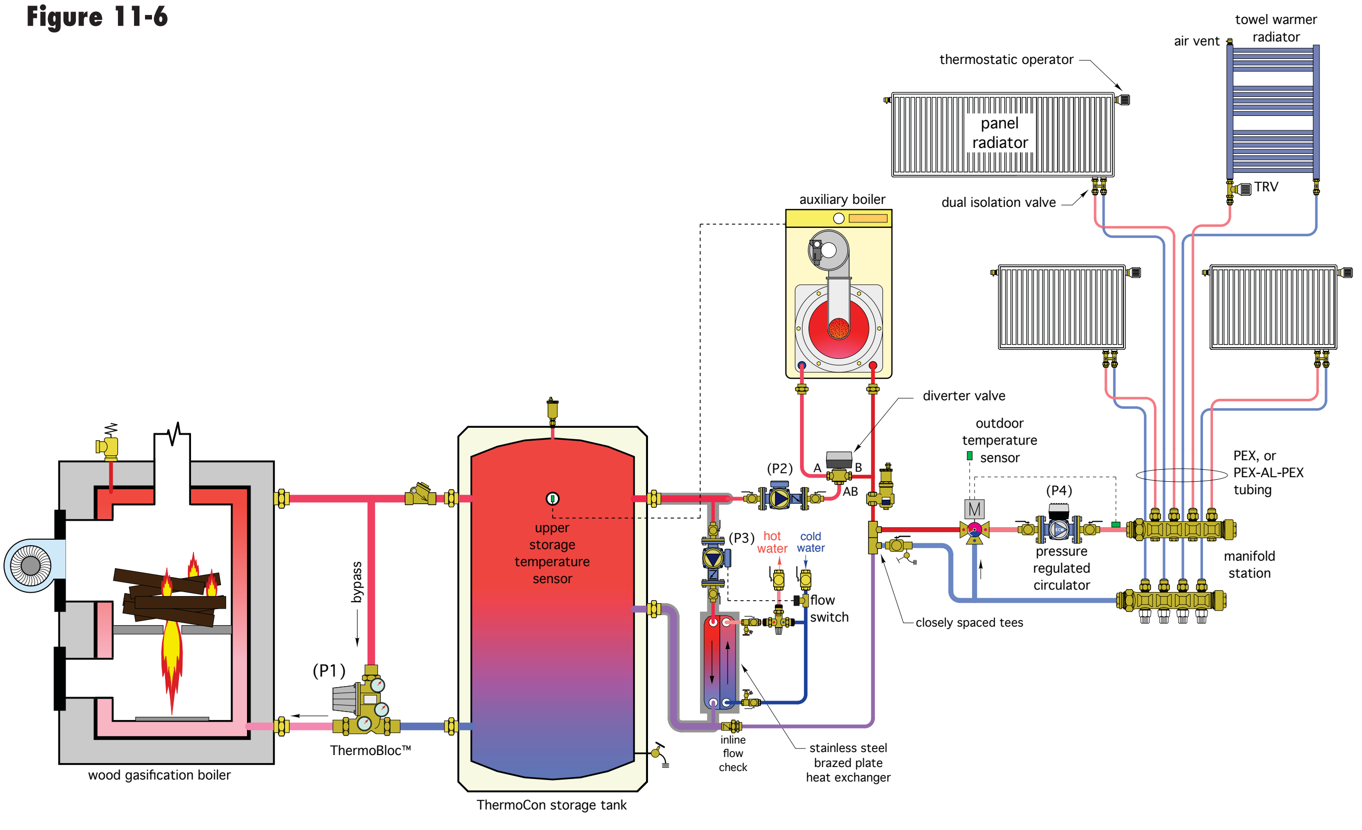
The heat pump is also shown equipped with a desuperheater heat exchanger for preheating domestic water. Desuperheaters are an option offered by several manufacturers of water-to-water heat pumps. They are refrigerant-to-water heat exchangers that receive the highest temperature refrigerant gas directly from the compressor and transfer heat from that gas to a stream of domestic water being circulated through the desuperheater.

idronics #9 provides a broad discussion of hydronic heating and cooling systems designed around geothermal heat pumps.

SYSTEM #6:

With the increasing cost of conventional fuels, boilers that burn firewood or wood pellets have become increasingly popular, especially in rural areas. Modern wood gasification boilers, when properly operated with dry wood, can yield thermal efficiencies approaching

Figure 11-6



90%. However, to attain such performance, they must operate at a high burn rate. In this mode, they often produce heat at rates significantly higher than the current space heating requirement of the building they serve. The solution is to install a thermal storage tank.

With the exception of some pellet-fired boilers, wood-fired boilers cannot automatically start themselves. Thus, there will likely be times when heat is needed by the building, or perhaps for domestic hot water, and the wood-fired boiler is not operating. To accommodate these situations, many systems incorporate an auxiliary boiler that can automatically turn on to provide space heating and domestic water until the output of the wood-fired boiler is sufficient to cover the load.

The system shown in Figure 11-6 provides a thermal storage tank, as well as an auxiliary boiler.

From the ThermoCon storage tank out through the distribution system, this schematic is identical to the schematic in Figure 11-4. The only difference is that a wood gasification boiler serves as the primary heat source rather than an array of solar collectors.

The piping leading into wood gasification boiler is equipped with a Caleffi ThermoBloc anti-condensation device. This device combines a circulator and thermostatic mixing valve. It operates whenever the wood gasification boiler is operating and prevents the boiler from operating with sustained condensation of the flue gases it produces. This reduces the possibility of creosote formation within the boiler.

During a power failure, the ThermoBloc also allows thermosyphon flow between the storage tank and boiler. This flow dissipates residual heat that continues to be generated (albeit at a much lower rate) within the boiler.

If the wood gasification boiler can maintain the thermal storage tank at a temperature sufficient to provide domestic hot water and space heating, the auxiliary boiler will not operate. If the tank temperature drops due to insufficient heat from the wood gasification boiler, the auxiliary boiler automatically fires to maintain a minimum accepted water temperature at the top of the storage tank.

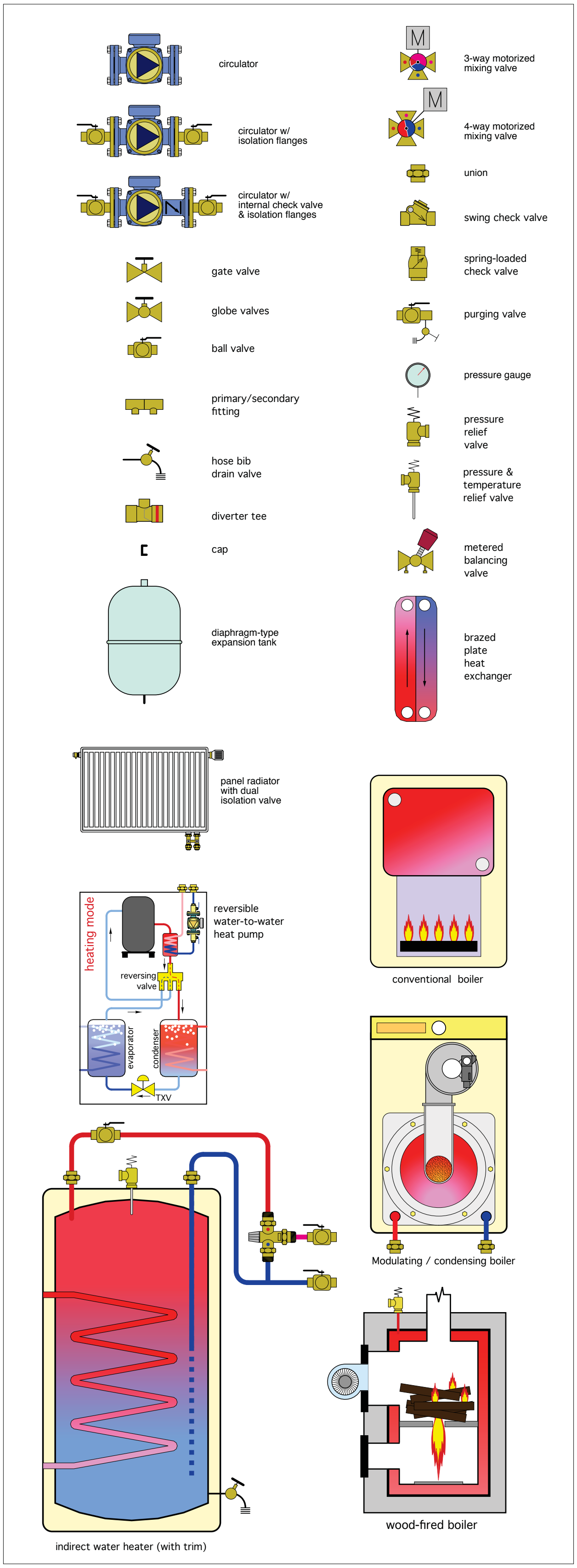
idronics #10 provides a broad discussion of hydronic heating and cooling systems designed around wood- and pellet-fired boilers.

SUMMARY:

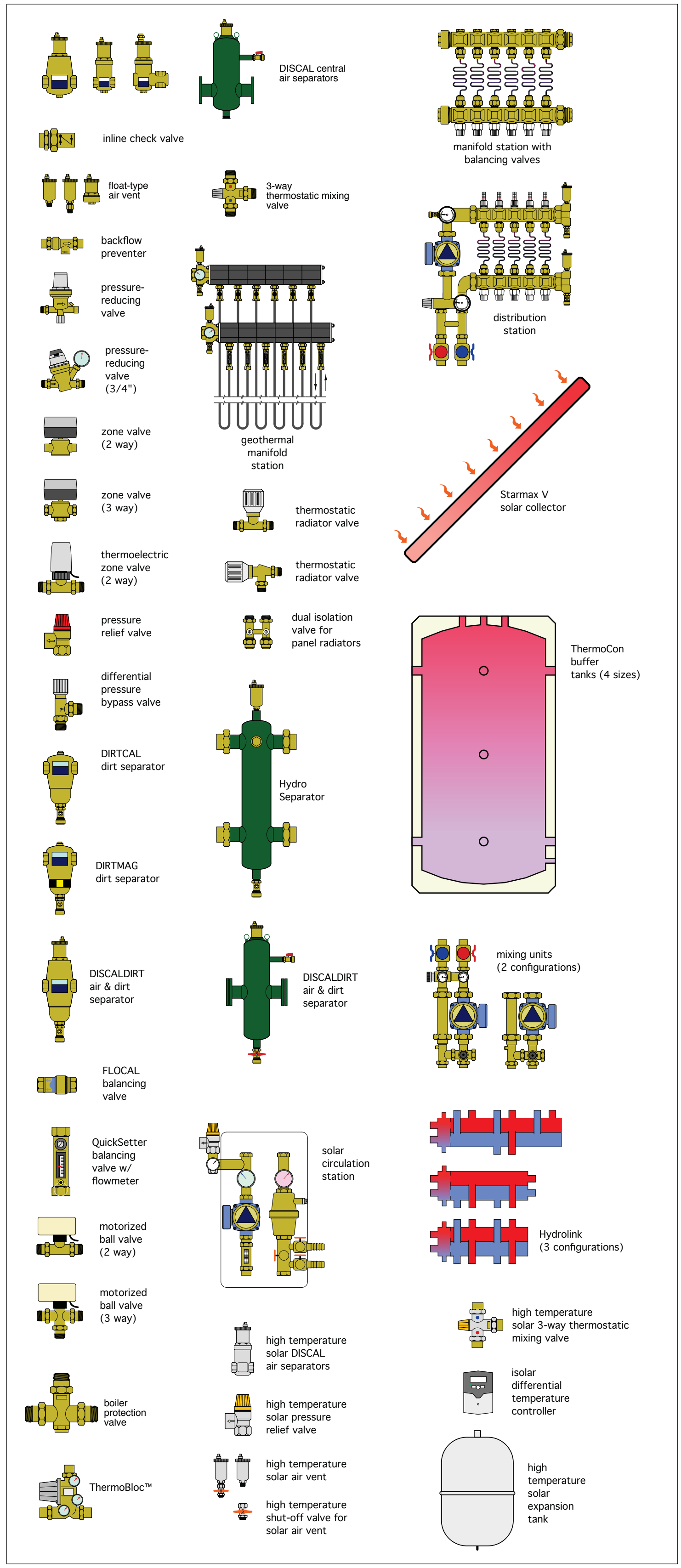
The systems shown in this section are but a sampling of thousands of possible variations involving different heat sources, heat emitters and distribution systems. They are meant to convey state-of-the-art concepts in combination with high quality hardware. The reader is strongly encouraged to reference previous issues of idronics for additional information on specific heat sources, as well as several other types of systems designed around those heat sources.

APPENDIX A: Schematic symbols

GENERIC COMPONENTS



CALEFFI COMPONENTS



APPENDIX B: Buffer tank sizing

The size of the buffer tank used in a system is determined by the desired "on-time" of the heat source and the acceptable temperature rise of the water in the buffer tank during this on-time. The on-time is that time between when the heat source turns on to begin warming the tank and when it turns off after lifting the tank temperature through the acceptable temperature rise. Formula B-1 can be used to determine tank size.

Formula B-1

$$V_{btank} = \frac{t\left(Q_{HP} - Q_L\right)}{500\left(\Delta T\right)}$$

Where:
V_{btank} = required volume of buffer tank (gallons)
t = desired on-time for heat source (minutes)
Q_{HP} = heating output of heat source (Btu/hr)
Q_L = any heating load served by buffer tank while charging (Btu/hr)
ΔT = allowed temperature rise of tank during heat source on-time (ºF)

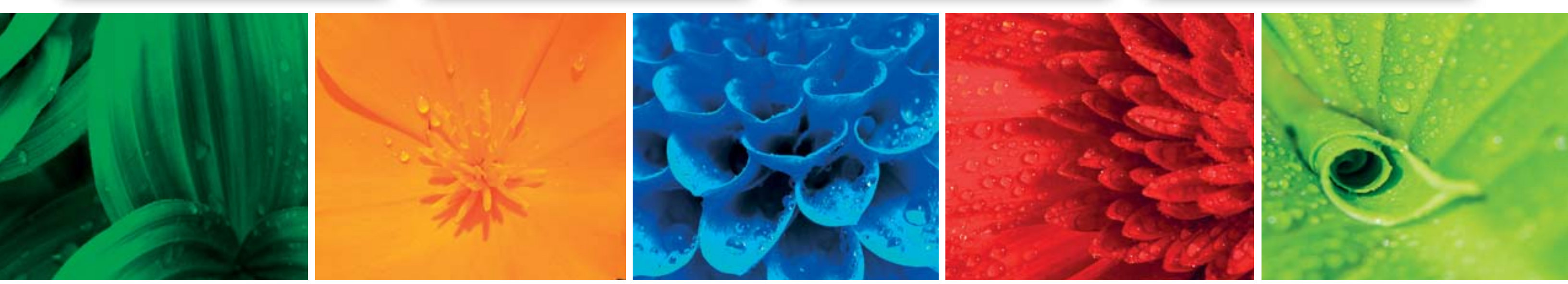
Example: Determine the size of a buffer tank that will absorb 48,000 Btu/hr while increasing in temperature from 90ºF to 110ºF during a heat source on-cycle of 10 minutes. There is no heating load on the tank during this charging.

Solution: The temperature rise (ΔT) is 110-90 = 20ºF. Putting this and the remaining data into Formula B-1 yields:

$$V_{btank} = \frac{t\left(Q_{HP} - Q_L\right)}{500\left(\Delta T\right)} = \frac{10\left(48000 - 0\right)}{500\left(20\right)} = 48\,gallons$$

If the allowed temperature rise was 10ºF, rather than 20ºF, the required tank volume would double to 96 gallons. If the desired on-time was only 5 minutes rather than 10 minutes, the volume would be cut in half. Anything that increases the desired on-time or decreases the allowed temperature rise during this on-time will increase the required tank volume, and vice versa.

The same formula can be used to determine the size of a buffer tank used for cooling. The value of Q_{HP} would be the cooling capacity of the heat pump. The value of ΔT would be the allowed temperature drop in the tank during this on-time.



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