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29
July 2021

Heat Exchangers in Hydronic and Plumbing Systems

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FROM THE GENERAL MANAGER & CEO

Dear Plumbing and Hydronic Professional,

While relaxing with some friends, someone posed this hypothetical question to the group, *"If you awakened tomorrow to find the nation's electrical grid had become disabled, what would you do?"* The discussion that followed made for interesting insight into the priorities people set in life, as well as their trust (or distrust) of human behavior. How long would food and water be available? How would people manage to survive, and who would survive? I suggest posing this question when you get together with friends. The conversation that's likely to follow is sure to be interesting and entertaining.

Now substitute "heat exchangers" for "electrical grid" in that hypothetical question. Would your friends arrive at similar conclusions? Those who work with thermal systems probably would. Why? Because without heat exchangers they would realize that many of the devices we take for granted simply wouldn't function. Examples include refrigerators, furnaces, boilers, chillers, air conditioners, dehumidifiers, computers, televisions, coffee makers, ovens, cell phones, TVs, automobiles, and more. Practically every device that manages thermal energy contains some type of heat exchanger. The same is true for electrical power plants — which takes us right back to the original question about the electrical grid!

In this issue of *idronics* we'll discuss how heat exchangers are used in heating, cooling and plumbing applications. Different categories of heat exchangers are described, as are common methods for assessing their performance. Installation details are provided that ensure lasting peak performance. The issue concludes with several examples of how heat exchangers can be applied within hydronic and plumbing systems. Some of these applications are common, while others are unique.

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

An interactive edition of this issue is available at idronics.caleffi.com. The content is fully searchable and is complimented with video and resource links. Prefer a hard copy? Visit us at www.caleffi.us and click on the idronics icon to register.

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General Manager and CEO

CALEFFI Hydronic Solutions

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1. INTRODUCTION

One of humankind's most significant achievements has been the ability to generate heat and guide its movement for useful purposes. This ability is essential to modern life, especially when it's used to establish and maintain thermal comfort in habitable spaces. Heat exchangers of all types are used for this purpose.

For centuries, heat was generated from fire, and methods were developed to guide that heat into occupied areas of buildings. An often-cited example of early heat exchange for improving human comfort was the Roman hypocaust. Wood-fueled fires would be maintained outside of occupied spaces. The hot combustion gases from these fires were channeled through spaces under stone floors supported on stone columns, as seen in Figure 1-1, as well as through hollow stone walls.

Figure 1-1

Because the combustion gases were significantly warmer than the building surfaces, heat would be absorbed into the stone. It would then transfer through the stone by conduction and be released into occupied spaces by thermal radiation and convection. This heat exchange process took place with minimal mixing of the combustion gases and the air in occupied spaces. As such, the hypocaust functioned as a heat exchanger.

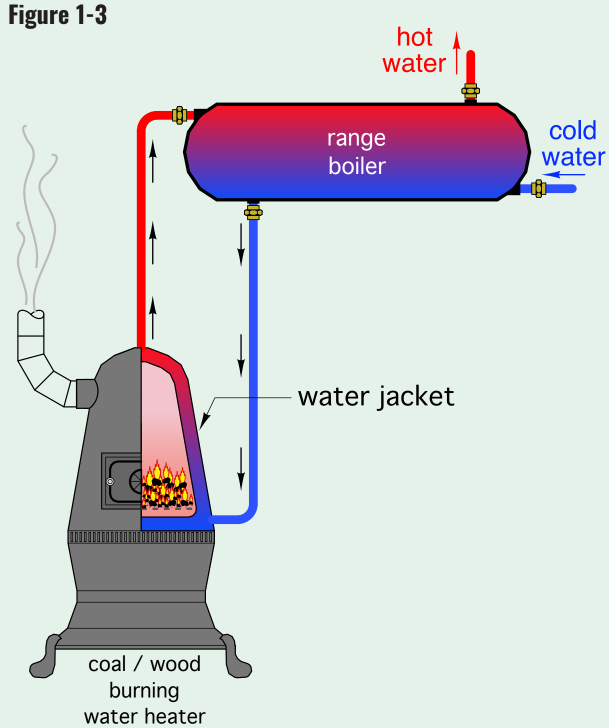
Today, modern heat exchangers allow heat transfer with *zero* mixing of the liquid or gas supplying the heat with the liquid or gas absorbing that heat.

Wood stoves are another example of centuries-old heat exchangers that allow heat from combustion gases to be transferred to occupied spaces with little, if any, release of those combustion gases into those spaces.

Some wood and coal burning stoves also included heat exchangers for heating domestic water. Figure 1-3 illustrates the concepts used in these early systems, which

Figure 1-2

Figure 1-3



were available long before electrically powered circulators. Flow between the "water jacket" of the stove and the "range boiler" tank was created by thermosiphoning due to the differences in density between hot and cold water.

The evolution of heat exchangers was also critical to development of internal combustion engines. Some of the earliest engines were cooled solely by surrounding air. As engine design improved and horsepower increased, it became impractical to rely solely on surrounding air to keep the engine temperature under control. Engineers of that era turned to the superior thermal properties of water as a means of conveying heat from inside engine blocks to a location where it could be dissipated to surrounding air. Figure 1-4 shows an example of a Ford Model T radiator.

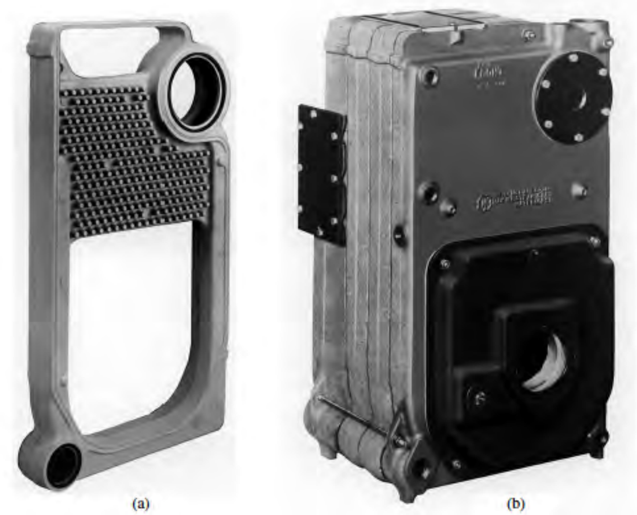
Figure 1-4

Courtesy of motormission.com

This radiator could be fundamentally described as a water-to-air heat exchanger. For its time — the early 1900s — it represented state-of-the-art-technology. Water from the upper portion of the engine block flowed into the upper portion of the radiator and divided up into multiple closed channels made of copper or brass. Air passed between these channels as a result of the car moving, as well as flow created by a simple fan connected to the engine's crankshaft by a leather belt. The higher thermal conductivity of the copper and brass channels provided minimal thermal resistance between water and the outer surfaces of the radiator. After giving up heat, the coolest water settled into a reservoir at the base of the radiator and flowed back to the lower portion of the engine. No water pump was used. All flow was driven by the changes in buoyancy of the water between the top of the engine and lower portion of the radiator. Unlike modern vehicle radiators, this radiator was not pressurized. Model T drivers learned to carry extra water with them to replace the water lost through evaporation, and in some cases, boiling inside the radiator.

All fuel-burning boilers used for heating buildings have a combustion chamber combined with a heat exchanger. Figure 1-5a shows an example of a cast iron section that is used to build the heat exchanger of a cast iron boiler. Figure 1-5b shows how this heat exchanger, which is also called a boiler "block," is made by joining several cast iron sections together.

Figure 1-5



Courtesy of Weil McLain

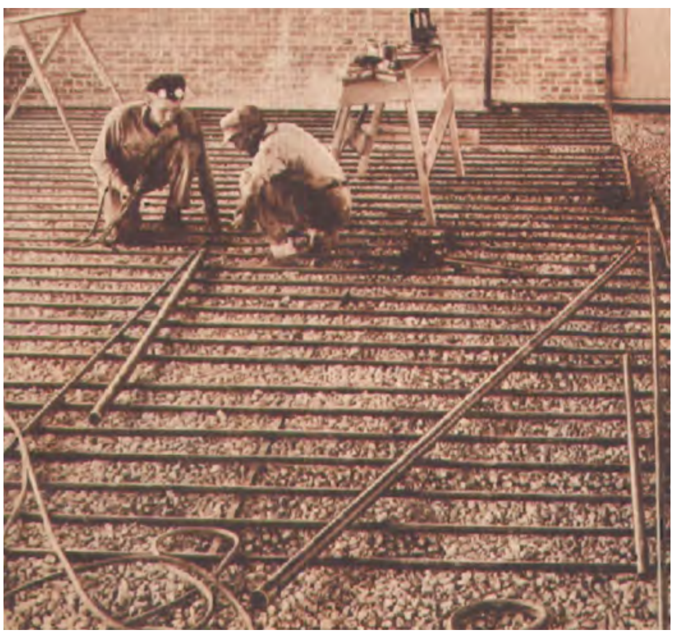
Hot gases pass upward from the combustion chamber and across the "pins" on the cast iron sections. The pins increase the heat transfer surface area of the section. Heat from the hot gases passes through the cast iron walls of each section and is absorbed by the water inside.

Heat exchanger technology continued to progress through the 20th century. Hundreds of heat exchanger designs were developed for use in boilers, radiators, chillers, fan-coils and convectors. Wrought iron pipe and copper tubing were embedded into concrete slabs to create "radiant panel heat exchangers," as seen in Figure 1-7. These panels transfer heat from heated water into occupied spaces using thermal radiation and convection.

Today, heat exchangers are precisely engineered for use in all types of stationary energy-processing equipment, as well as virtually all land-based vehicles, marine vessels, aircraft and spacecraft. These devices range from huge, multi-ton cooling towers used to dissipate heat from high-rise buildings (Figure 1-8), to tiny liquid cooling systems for microprocessors (Figure 1-9).

Figure 1-6

Figure 1-7

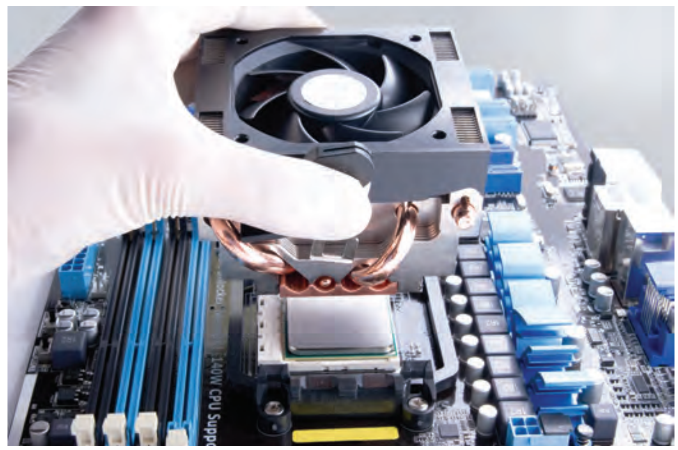


Every hydronic heating and cooling system involves the movement of thermal energy between water, or a water-based fluid, and one or more surrounding materials. This heat exchange begins at a heat source and ends at one or more heat emitters. With proper design, the *rate* of heat exchange at all locations within the system allows for safe operation and unsurpassed comfort.

Within hydronic heating and cooling systems, there are many situations where it's necessary to move heat from one fluid to another without letting those fluids contact each other. Examples include:

Figure 1-8

Figure 1-9



- Transferring heat from the "system" water in a boiler to domestic water.
- Transferring heat from system water to an antifreeze solution circulated through tubing for melting snow on pavements.
- Transferring heat from an antifreeze solution circulating through piping buried in the earth to the refrigerant within a heat pump.
- Transferring heat from a district heating system to a building located several miles away.
- Transferring heat from water circulating through a cooling system to the refrigerant within a chiller.

This issue of *idronics* focuses on heat exchangers used within hydronic heating and cooling systems, as well as in some plumbing applications. The discussion ranges from fundamental heat transfer concepts to performance evaluation and novel applications within modern hydronic systems.

2. HEAT EXCHANGER TYPES

A wide variety of heat exchangers are used in modern hydronic heating and cooling systems, as well as in some plumbing applications. This section illustrates the most common hardware configurations and describes common applications for these heat exchangers.

INTERNAL vs. EXTERNAL HEAT EXCHANGERS

In the context of hydronic systems, *internal* heat exchangers are typically helical coils made of a metal with high thermal conductivity and suspended within liquid-filled containers such as a thermal storage tank. Figure 2-1 shows an example of an internal coil heat exchanger permanently mounted in the lower portion of a pressure-rated stainless steel tank.

A heated fluid is circulated between the heat source and the inside of the coil heat exchanger. The coil is surrounded by domestic water within the tank's shell. Heat transfers from the outer coil surface to the domestic water by natural convection.

There are several variations of tanks with internal coil heat exchangers.

Figure 2-1

Courtesy of Heat-Flo, Inc.

Figure 2-2

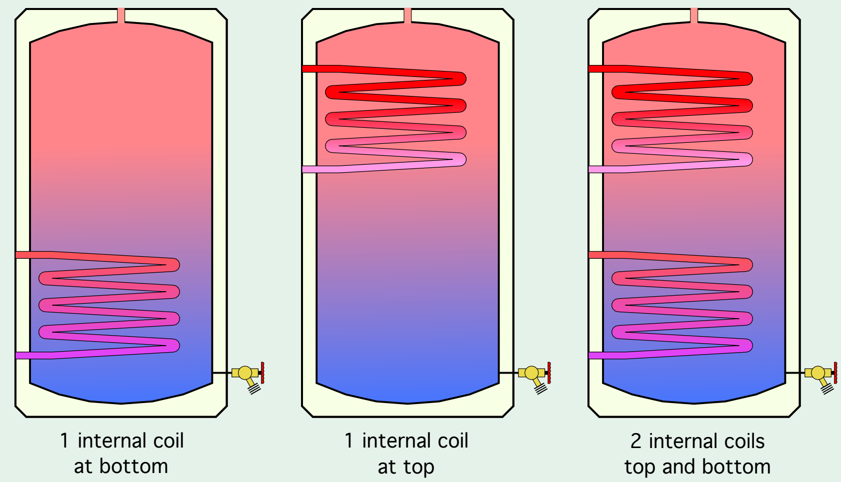


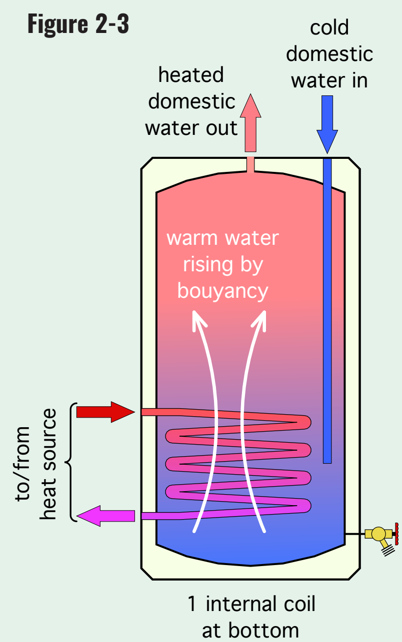
Figure 2-2 shows some common configurations.

Tanks with a single lower coil are commonly used for domestic water heating and are referred to as indirect water heaters. Figure 2-3 shows further details for this application.

The single lower coil releases heat into the coolest water, which, due to its density, collects at the bottom of the tank. When hot domestic water is drawn from the top of the tank, cold domestic water enters through a dip tube that carries it to the lower portion of the tank. This helps to preserve beneficial temperature stratification within the tank.

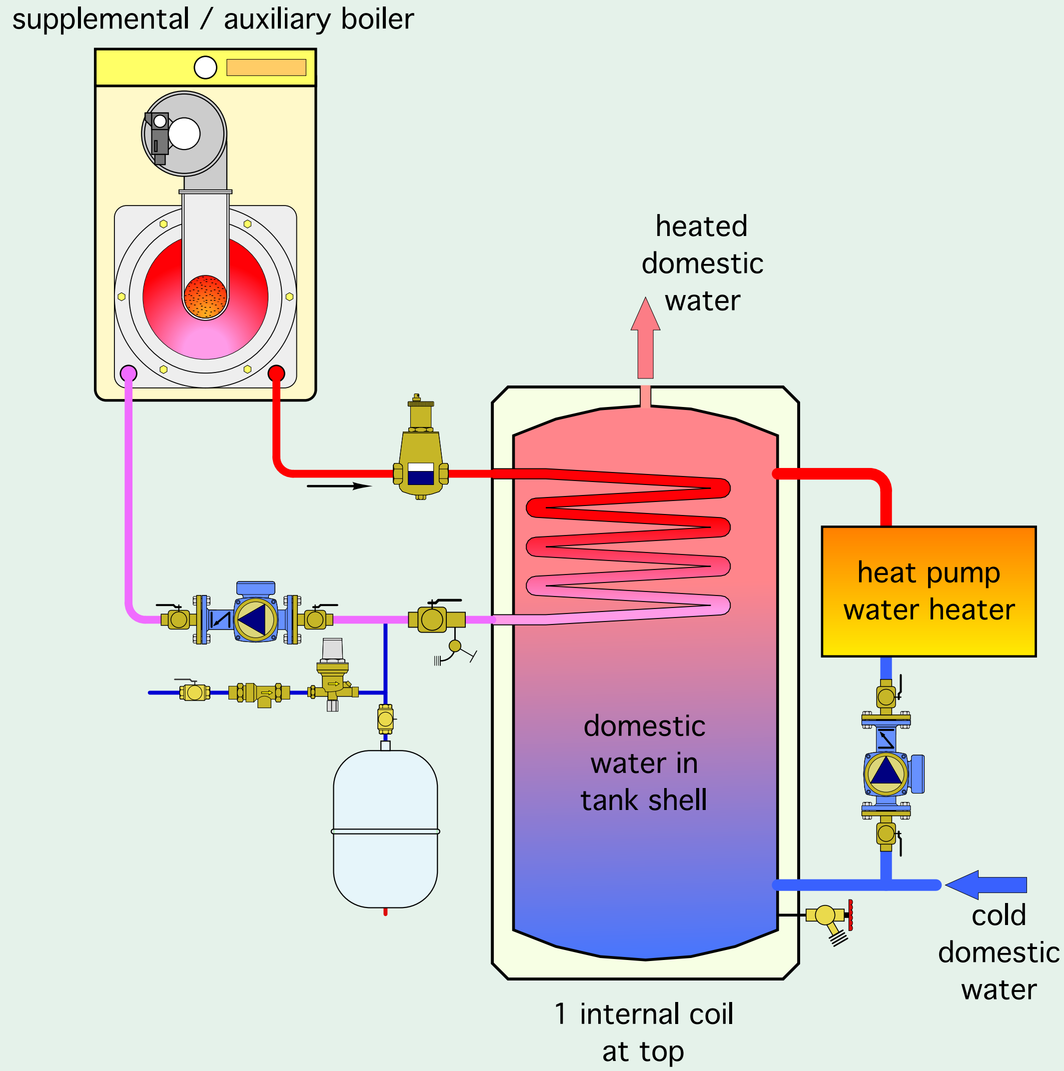
The heat source could be a boiler, heat pump or solar thermal collectors. Hot fluid from the heat source enters the upper coil connection and flows downward, exiting at the lower connection. This flow direction is opposite the direction that water within the tank flows after absorbing heat from the coil. Because the two fluids move in opposite directions, this situation is called "counterflow" heat exchange. Later sections will show why counterflow is important to maximize heat transfer rates.

Figure 2-3



Some systems using heat pumps are designed to supply the heat pump with the coolest water drawn for the lower portion of the tank. The lower the water temperature at which the heat pump operates, the higher its efficiency. Figure 2-4 shows such a system, but with a boiler supplying supplemental (or backup) heat through a coil heat exchanger mounted in the upper portion of the tank.

Figure 2-4



The upper coil heat exchanger allows the supplemental heat to be added to ensure that water supplied from the tank meets the desired setpoint, but without disturbing the coolest water in the lower portion of the tank.

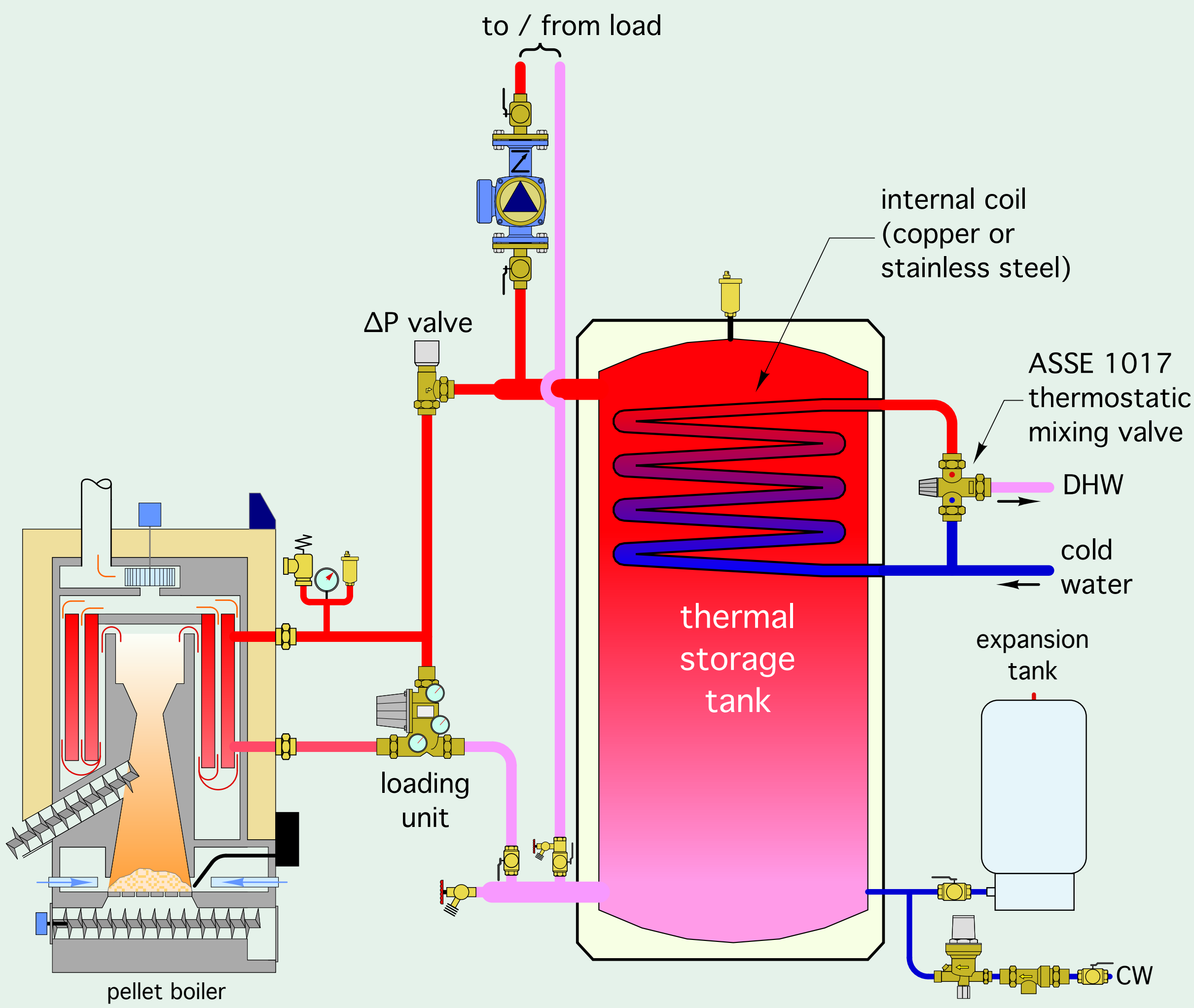
Another possibility is to route domestic water through a coil heat exchanger in the upper portion of the tank, as shown in Figure 2-5.

This requires the coil to be made from a material such as copper or stainless steel that is compatible with domestic water. The coil is surrounded by "system water" that is heated by a heat source and is also used for space heating. The tank in Figure 2-5 can also provide buffering to a zoned distribution system.

Tanks having upper and lower coil heat exchangers can be used for several applications. One is where the lower coil provides heat input from a solar thermal collector array, while the upper coil provides domestic water heating. The water in the tank shell is heated by a boiler and provides buffering for a highly zoned space-heating system. Figure 2-6 shows an example of such a system.

In some applications, the two internal coil heat exchangers are connected in series to create a single heat exchanger with increased surface area.

Figure 2-5



DOUBLE WALL COIL HEAT EXCHANGERS

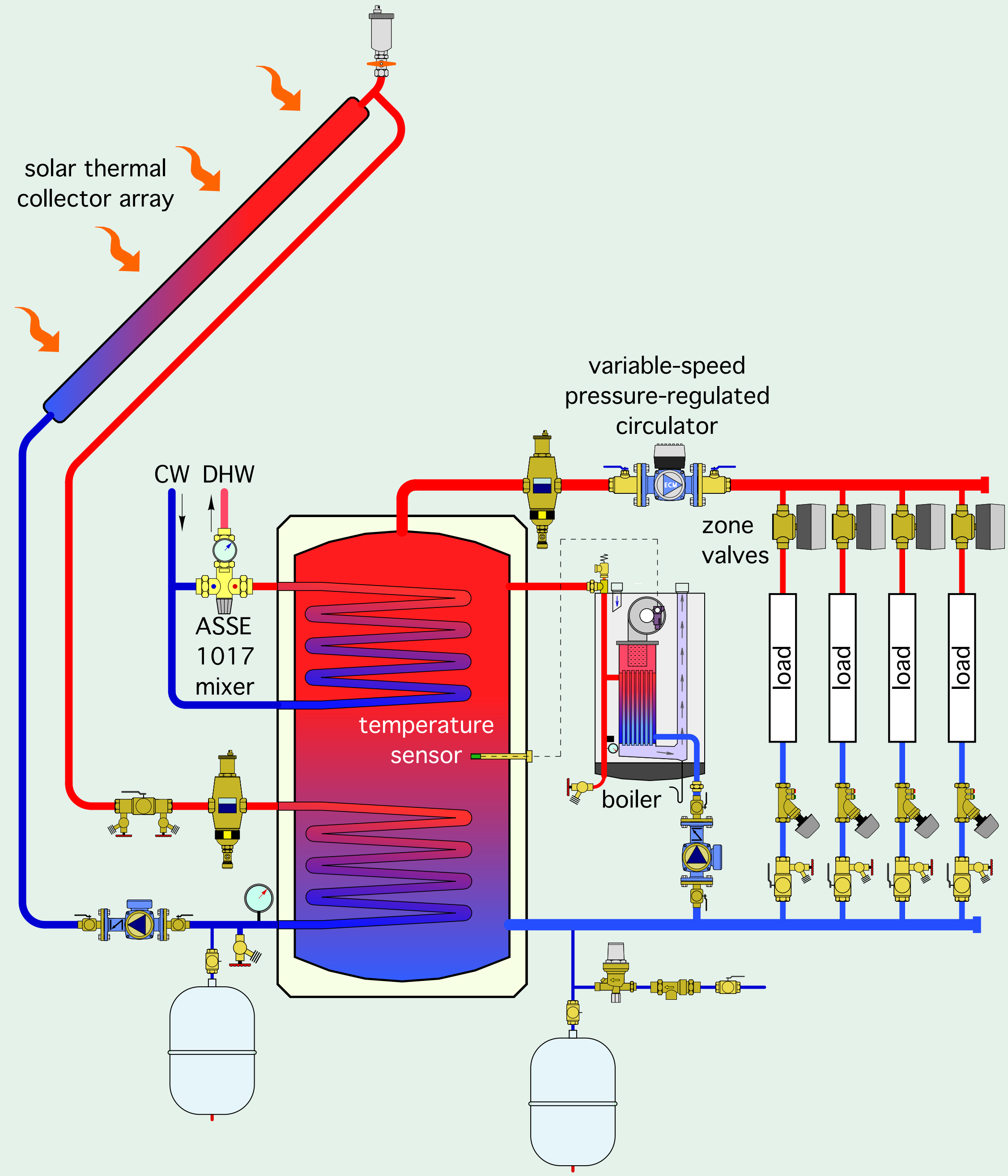
Some mechanical codes require a special type of coil heat exchanger when the application requires transferring heat from an antifreeze solution to potable water. That coil must be "double walled." There must be a partial air gap between the metal wall that receives heat from the antifreeze solution and the metal wall that transfers heat to potable water. This air gap creates a "leakage path" that would allow a leak in either metal wall to exit the heat exchanger outside of the tank, and thus provide visible evidence of the leak. Figure 2-7 illustrates the concept.

Double wall heat exchangers create lower heat transfer rates compared to single wall coils of equivalent surface area. They also tend to have higher pressure drops at a given flow rate relative to single wall heat exchangers. Opinions vary on their use, as do mechanical code requirements on when and where they must be used.

Internal coil heat exchangers, typically made of copper tubing, are also used for heat input and heat extraction from large non-pressurized thermal storage tanks often used in systems supplied by cordwood gasification or pellet-fired boilers. Figure 2-8 shows a typical configuration.

In some systems, one suspended coil is used for heat input while another is used for heat extraction. This is the scenario shown in Figure 2-8. Notice that the flow direction in each coil is opposite the direction in which

Figure 2-6



tank water moves by natural convection. Flow *inside* the heat input coil is from top to bottom. Flow *inside* the heat extraction coil is from bottom to top. These counterflow directions improve the rate of heat transfer through each coil.

Piping from each coil passes through the tank's sidewall above the highest water level. The piping penetrations are sealed to limit evaporation water loss but are typically not rated for submerged operation.

The use of two coil heat exchangers introduces two "thermal penalties" (e.g., undesirable but necessary temperature drops) into the heat transfer process between the heat source and the load. One is the temperature drop between the water entering the heat input coil from the heat source, and the average water temperature in the tank. The other is the temperature drop between the average tank temperature and the outlet temperature from the heat extraction coil. The extent of these temperature drops depends on the surface area of the coil and the

Figure 2-7

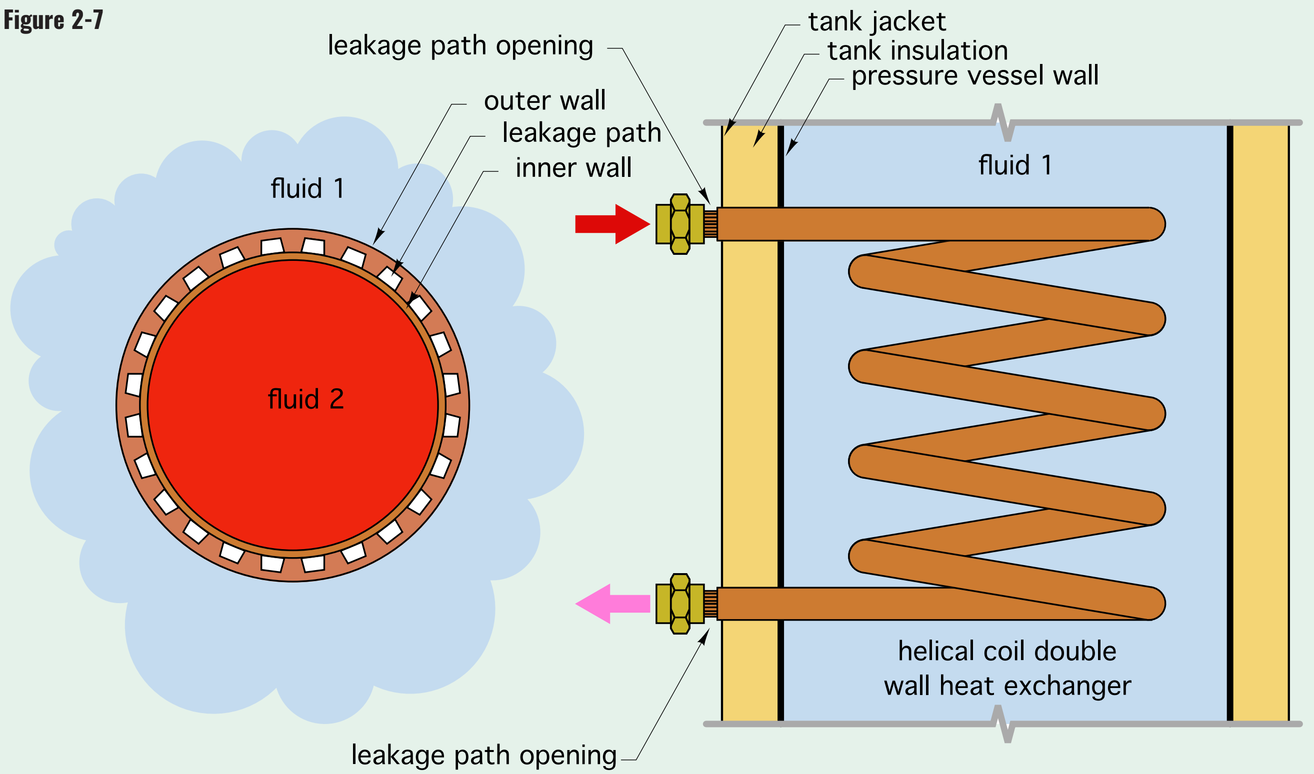
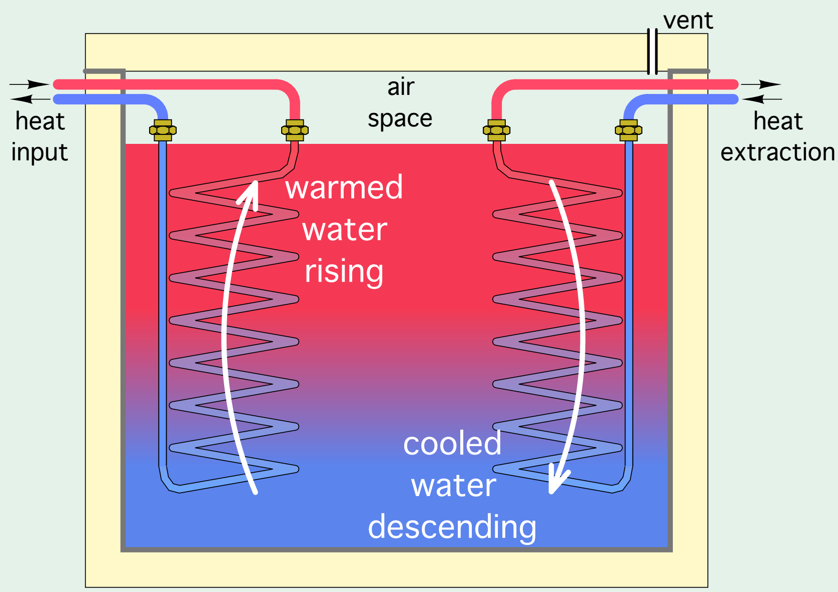


Figure 2-8



Courtesy of HydroFlex Systems

convection conditions between the outer surface of each coil and the tank water. In general, it's desirable to reduce or eliminate any thermal penalty between the heat source and the load. This is especially true for heat sources that require relative low operating temperatures to attain high efficiency.

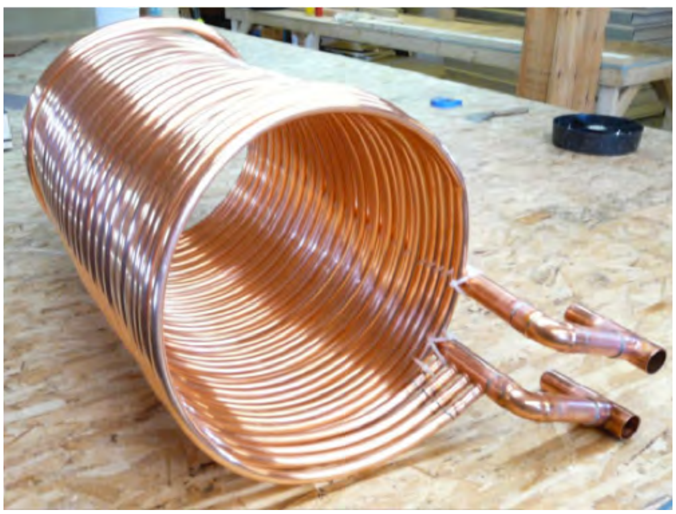
In some applications, multiple submerged coil heat exchangers are used to increase the total heat transfer area. Figure 2-9 shows an example of multiple copper coil heat exchangers connected in parallel to header piping.

Another approach is to "co-wind" multiple copper tubes into a single helical coil, as shown in Figure 2-10. This provides additional surface area while greatly reducing pressure drop relative to a single coil of the same total surface area. The individual coils

Figure 2-9

Courtesy of Hydroflex Systems, Inc.

Figure 2-10



Courtesy of American Solartechnics

are manifolded together at each end of the heat exchanger, allowing for a single supply and return connection.

EXTERNAL HEAT EXCHANGERS

The term "external" heat exchanger refers to any heat exchanger that is not immersed in a fluid inside another component such as a tank. Unless the term "internal" is used to describe the heat exchanger, assume that it is external, and thus just referred to as a heat exchanger.

The most commonly used heat exchangers in hydronic systems can be categorized as follows:

- liquid-to-liquid
- liquid-to-air

The first word in these descriptions refers to the fluid supplying heat. The final word refers to the fluid absorbing heat. For example, in some snowmelting systems, boiler water would be the source liquid, and an antifreeze solution would be the destination liquid. In a cooling application, the first word in the description would be the source of the cooling effect, and the last word would be the destination of the cooling effect.

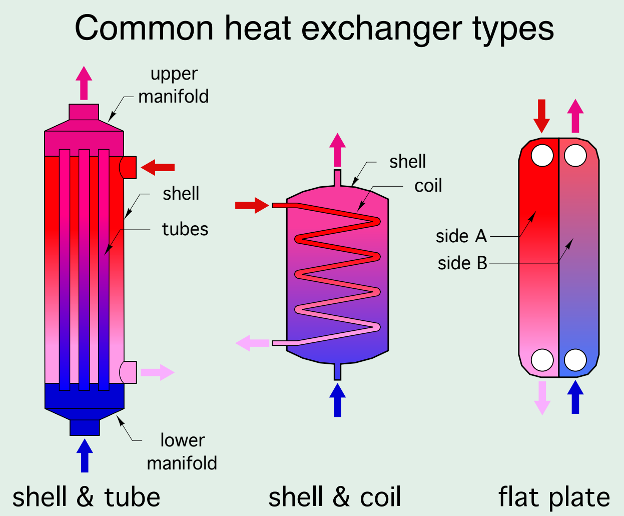
LIQUID-TO-LIQUID HEAT EXCHANGERS

The three common types of liquid-to-liquid heat exchangers used in hydronic systems are:

- Shell & tube heat exchangers
- Shell & coil heat exchangers
- Flat plate heat exchangers

Figure 2-11 illustrates the fundamental construction of these heat exchangers.

Figure 2-11



SHELL & TUBE HEAT EXCHANGERS

This heat exchanger design stems from the fundamental concept of a pipe that separates two liquids exchanging heat. One liquid surrounds the outer surface of the pipe and the other flows inside the pipe. The fluid surrounding the pipe is contained within another cylindrical vessel called the shell, as shown in Figure 2-12.

In most shell & tube heat exchangers, the cooler fluid passes through the annular space between the outer surface of the tube and the shell. This reduces heat loss from the outer surface of the shell.

Although it's possible to create a shell & tube heat exchanger with a single tube, as shown in Figure 2-12, the length of the heat exchanger required to create sufficient

Figure 2-12

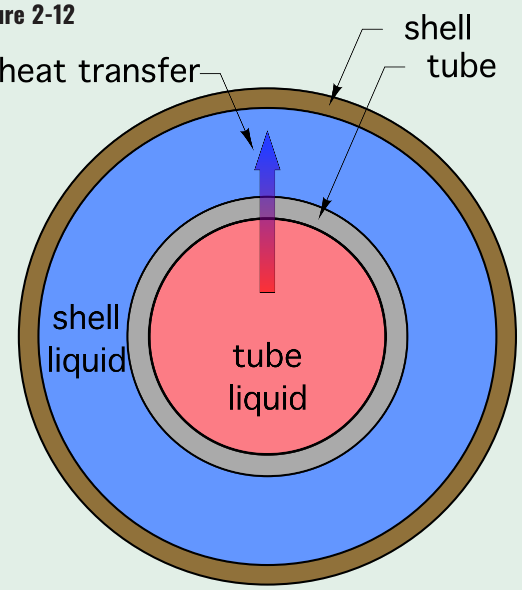


Figure 2-13

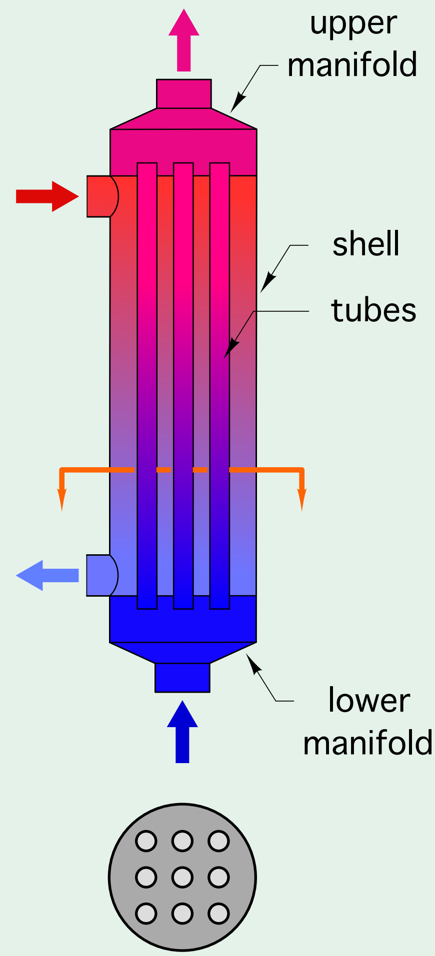
Courtesy of Packless Industries

heat transfer area for many applications is excessive. In some situations, this length issue is solved by coiling the overall heat exchanger, as shown in Figure 2-13.

This component is called a *coaxial tube-in-tube heat exchanger*. It is commonly used for water-to-refrigerant heat transfer in heat pumps.

When coiling is not possible, multiple straight tubes are used, as shown in Figure 2-14.

Figure 2-14



The tubes are welded or brazed to bulkhead plates near each end of the heat exchanger. The two ends of the exchanger serve as manifolds to distribute flow through the multiple tubes. The second liquid passes through the shell of the heat exchanger and around the outer surfaces of all the tubes. These designs can be scaled up to products capable of transferring several millions of Btu/hr.

Figure 2-15 shows an example of a large "tube bundle" that would be inserted into an appropriately sized shell. The baffle plates through which the tubes pass increase turbulence within the shell, which increases the convective heat transfer rate.

The tubes and shell are often made of different materials, depending on the chemical nature of the fluids exchanging heat, as well as the temperature and pressure at which the heat exchanger is rated to operate.

Figure 2-15

Figure 2-16

Shell & tube heat exchangers are also available with multiple tube passes. One liquid passes through half of the tube bundle in one direction, reverses through U-bends at the other end of the shell, and passes through the remaining tubes, eventually returning to the same end of the heat exchanger. Figure 2-16 shows an example of a 2-pass shell & tube heat exchanger having a welded steel shell and copper tubes. This heat exchanger has a removable cast iron end cap that allows the tube bundle to be removed for cleaning or replacement.

The concept of multiple tube passes can be increased to 3-pass and 4-pass designs. Most shell & tube heat exchangers also have internal baffle plates that increase turbulence on the shell side for improved convective heat transfer.

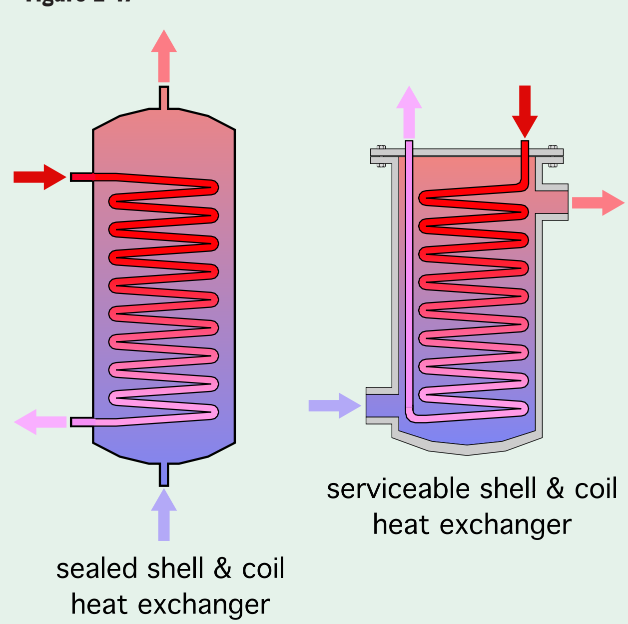
Shell & tube heat exchangers are commonly used for industrial applications, often at high temperatures and pressures. Many of these heat exchangers can be opened for servicing or replacement of the tube bundle. They are seldom used in smaller residential and light commercial hydronic systems. Their design requires significantly more metal volume relative to other heat exchanger designs of comparable capacity. The physical size of a shell & tube heat exchanger required to provide for a given internal area is also larger than that of other heat exchanger designs. Shell & tube heat exchangers also tend to have greater external surface area relative to other designs and thus experience higher parasitic heat loss for a given set of operating conditions. They are also more prone to fouling deposits due to lower internal flow velocities in the shell.

SHELL & COIL HEAT EXCHANGERS

A shell & coil heat exchanger is made by enclosing a helically shaped coil of tubing within a metal or composite shell. One fluid passes through the coil while the other passes through the shell and over the outer surface of the coil. Figure 2-17 shows two variations of shell & coil heat exchangers.

The geometry of the shell varies. Some shells are designed for vertical installation, with connection at the top and bottom. Others are designed for installation in a variety of orientations, with side connections on the shell and coil connections at one end. Some shell & coil heat exchangers have the coil permanently welded or brazed to the shell.

Figure 2-17



Others allow the coil to be removed by unbolting a sealed end plate at one end of the shell.

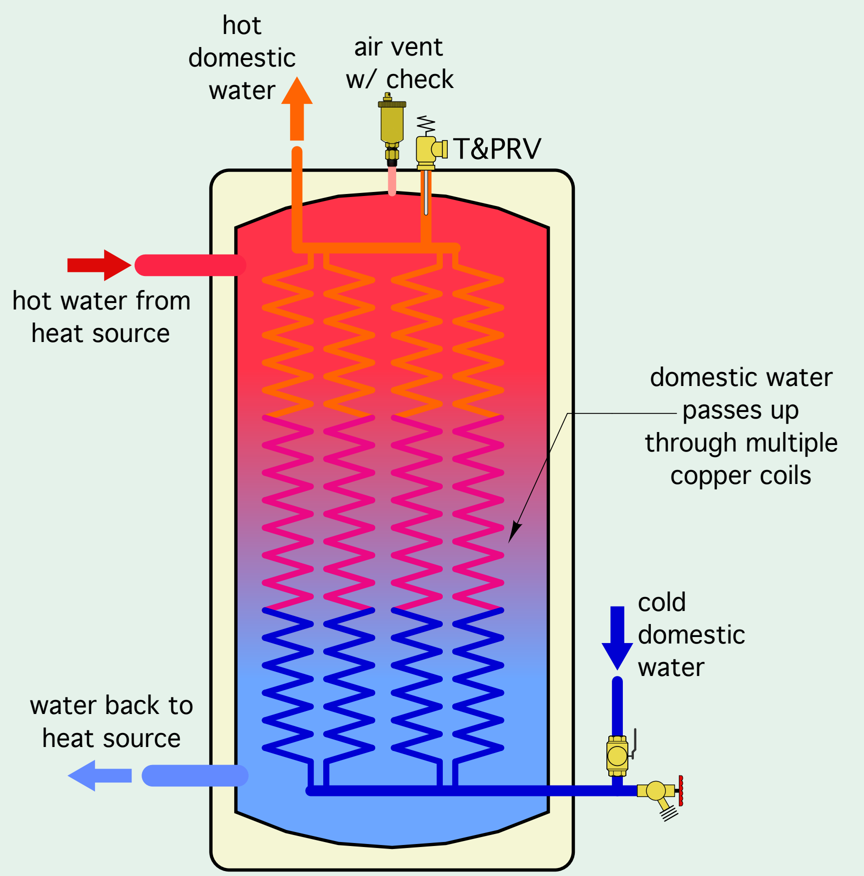
Shell & coil heat exchangers have been used for heat transfer between two liquids as well as between a liquid and a refrigerant. In the latter case, the refrigerant typically passes through a welded steel shell, while the liquid passes through a copper coil. In some systems, the volume of the shell also serves as a liquid accumulator within a refrigeration circuit.

Shell & coil heat exchangers are not commonly used for liquid-to-liquid heat transfer in hydronic systems. One limitation is the amount of coil surface area versus the overall size of the heat exchanger. Another is the higher fluid volume in the shell, which increases thermal mass and decreases the heat exchanger's response time to temperature changes relative to that of other heat exchanger designs.

One application where the increased thermal mass of a shell & coil design is *desirable* is when the heat exchanger also serves as a thermal storage device. A "reverse" indirect water heater, as shown in Figure 2-18, is a good example.

One can think of this reverse indirect water heater as a high surface area shell & coil heat exchanger with added thermal mass and insulation. Domestic water is fully heated on a single upward pass through multiple copper coils that are manifolded together at the top and bottom of the tank. Hot water from a boiler or other heat source passes through the steel shell of the tank, transferring heat to the copper coils.

Figure 2-18



FLAT PLATE HEAT EXCHANGERS

One of the most contemporary devices for fluid-to-fluid heat exchange is called a flat plate heat exchanger. This type of heat exchanger is now used in many types of hydronic heating and cooling systems, as well as for the evaporator and condenser in some refrigeration systems.

Fundamentally, a flat plate heat exchanger is created by stacking several pre-formed metal plates and sealing the perimeter of those plates together. The plates are shaped to create narrow flow channels between them. One fluid passes from one end of the heat exchanger to the other through the odd-numbered channels (1, 3, 5, 6, etc.). The other fluid passes from one end of the heat exchanger to the other through the even-numbered channels (2, 4, 6, 8, etc.). This concept is illustrated in Figure 2-19.

Figure 2-20 shows examples of the preformed stainless steel plates and a partially disassembled flat plate heat exchanger.

Figure 2-19

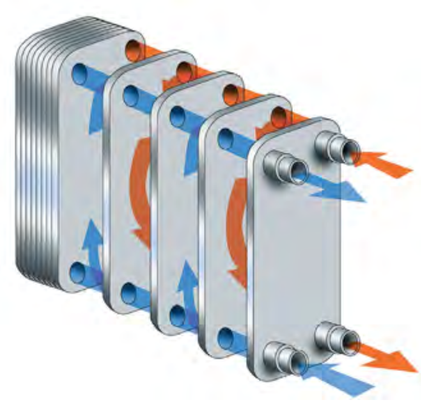


Figure 2-20a

Figure 2-20b

Courtesy of (a) Alfa Laval, and (b) GEA PHE Systems

The placement of the holes within the plates creates four internal manifold cavities. These cavities ensure even distribution of both fluids into their respective channels. The cavities transition to piping connections at one end of the heat exchanger, as seen in Figure 2-20b.

The patterned plates are designed to encourage turbulent flow, and thus high convection coefficients. Some manufacturers offer different plate patterns to create internal flows that are well-matched to specific fluid properties and a desired flow rate and pressure drop characteristics.

For a given set of entering temperatures, flow rates and fluid characteristics, the heat transfer capacity of a flat plate heat exchanger depends primarily on the size and number of plates used.

Flat plate heat exchangers can be further classified as:

- Brazed plate heat exchangers
- Plate & frame heat exchangers

BRAZED PLATE HEAT EXCHANGERS

As the name implies, all the plates on a brazed plate heat exchanger are brazed together at their perimeter, as well as where internal surfaces come in contact. The brazing is typically done with a copper alloy. This metallurgically seals the flow channels and creates two separate pressure-rated compartments within the heat exchanger. Once it is brazed, the heat exchanger cannot be modified or opened.

Most manufacturers have standardized plates sizes for their brazed plate heat exchangers. Typical plate dimensions are 3 x 8 inches, 5 x 12 inches and 10 x 20 inches. For a given plate size, the heat transfer capacity is increased by adding plates to the "stack" that becomes the overall heat exchanger. For example, a 5 x 12 x 40 flat plate heat exchanger is built using 40 plates of nominal dimension 5 inches by 12 inches. One unique plate forms the back of the heat exchanger (e.g., it has no holes through it). Another unique plate forms the front of the heat exchanger and transitions to the four piping connections.

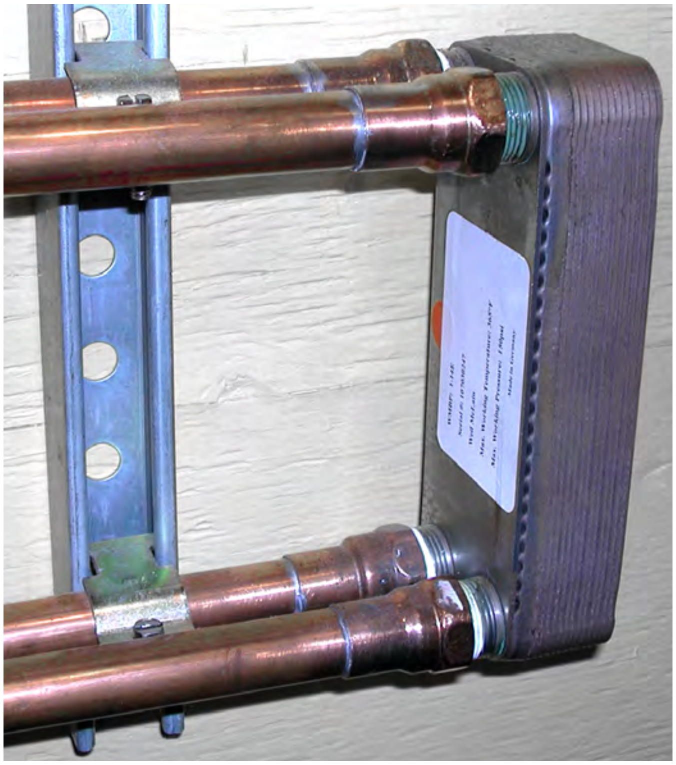
Brazed plate heat exchangers are typically used in small- to medium-scale hydronic systems, where heat transfer rates up to several hundred thousand Btu/hr are required. A small brazed plate heat exchanger might have 10 plates measuring a nominal 3 inches by 8 inches. A large brazed plate exchanger might have 100 plates measuring a nominal 10 inches by 20 inches.

Figure 2-21 shows three examples of brazed plate heat exchangers: A small 3 x 8-inch x 10 plate, a 5 x 12-inch x 30 plate, and a 5 x12-inch x 100 plate. The largest heat exchanger has 1-1/4-inch MPT connections. The medium-sized exchanger has 1 inch MPT connections. The small heat exchanger has 3/4-inch FPT connections.

Most of the brazed plate heat exchangers used in hydronic heating and cooling systems are constructed of 316 stainless steel. This allows them to operate with potable water having some chloride content, as well as a range of system fluids, including most antifreeze solutions. Some manufacturers also offer heat exchangers made of less-expensive 304 stainless steel for applications using fluids with low chloride content. Brazed plate heat exchangers typically have pressure and temperature ratings well above those needed in hydronic systems.

Figure 2-21

Figure 2-22



Small brazed plate heat exchangers are relatively light. A 3 x 8-inch heat exchanger can be supported by the 4 pipes connected to it. However, those pipes should be supported within a few inches of the heat exchanger, as shown in Figure 2-22.

Heat exchangers with 5-inch x 12-inch (or larger) plates should be supported by some type of bracket to reduce stress on the connected piping. Figure 2-23a shows an example of a fabricated bracket supporting an insulated 5 x 12-inch x 100 plate heat exchanger. Figure 2-23b shows a 5 x 12-inch x 40 plate heat exchanger supported on a small steel angle bracket screwed to a plywood wall. Some brazed plate heat exchangers are supplied with threaded studs that can be fastened to a metal bracket, as shown in Figure 2-23c.

Brazed plate heat exchangers are also available with double wall construction for situations where codes require them. These units provide a leakage path between adjacent plates that would route any leaked fluid outside the heat exchanger.

PLATE & FRAME HEAT EXCHANGERS

Plate & frame heat exchangers could be considered the "big brother" to brazed plate heat exchangers. They use the same concept of a stack of preformed plates to separate the two fluids in alternating channels. However, instead of brazing, plate & frame heat exchangers use gaskets to seal the fluid channels. The stack of plates is assembled on a frame between two thick steel pressure plates. When the required number of plates have been loaded onto the frame, several threaded steel rods are used to pull the pressure plates together and compress the stack into a pressure-tight assembly. The plate stack, pressure plates and tension rods can be seen on the heat exchanger in Figure 2-24.

Figure 2-23a

Figure 2-23b

Figure 2-23c

Figure 2-24

Courtesy of Alfa Laval

Large plate & frame heat exchangers are typically assembled close to, or exactly at, their final position within a mechanical room. Each plate is "hung" from an upper rail and slid into initial position to form the plate stack. The threaded steel draw rods are carefully tensioned using torque wrenches to ensure even compression of the stack. This assembly concept is shown in Figure 2-25.

Figure 2-26 shows an exploded view of a plate & frame heat exchanger. The alternating fluid channels, thick steel pressure plates and perimeter gaskets are visible. Some plate & frame heat exchangers also include a "shroud" around the completed plate stack.

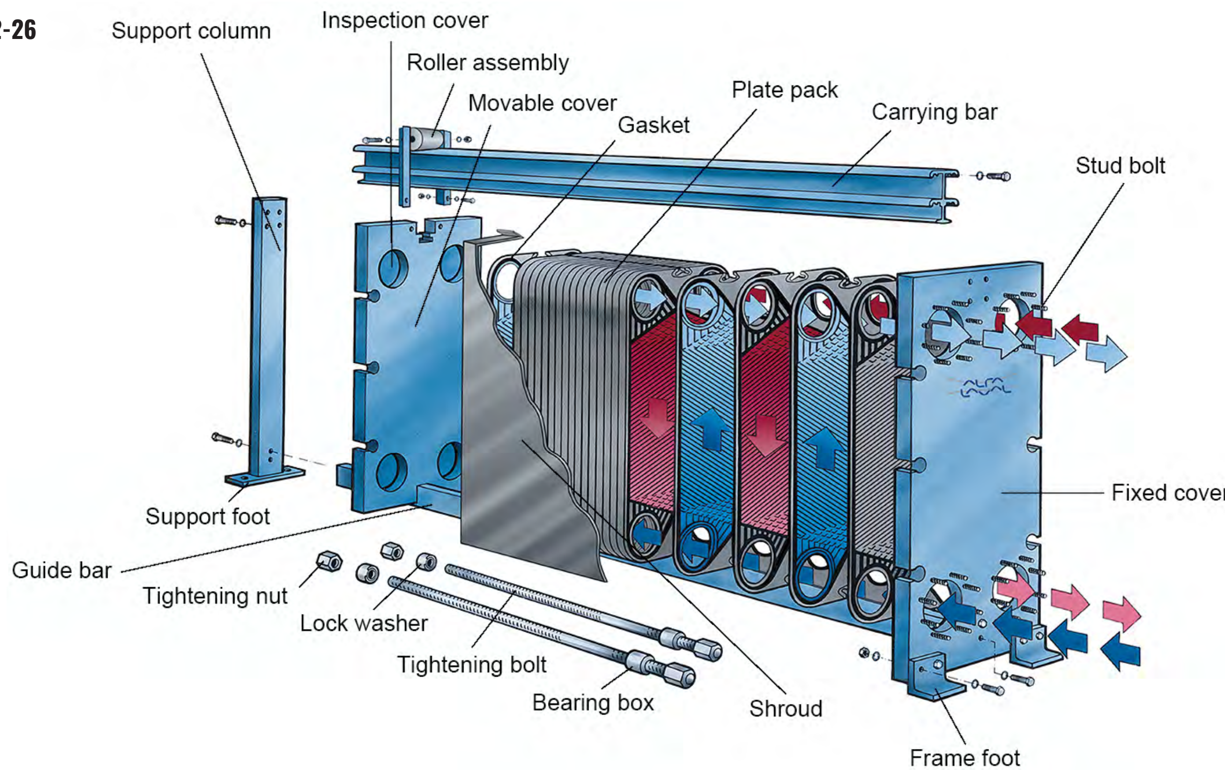
Large plate & frame heat exchangers are heavy. They are usually anchored to concrete pads within mechanical rooms, as seen in Figure 2-27.

The large plate & frame heat exchanger in Figure 2-27 has been fully insulated to reduce heat loss into

Figure 2-25

Courtesy of Alfa Laval

Figure 2-26



Courtesy of Alfa Laval

Figure 2-27

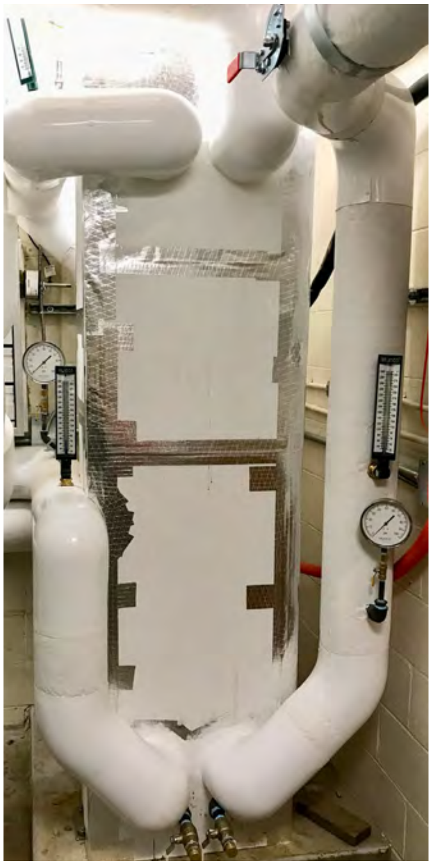


Figure 2-28

the mechanical room. The welded steel piping connected to this heat exchanger has also been insulated and jacketed. Note the installation of thermometers near all four ports of the heat exchanger. Drain valves that tee into the two lower pipes can also be seen.

The frames used for plate & frame heat exchangers are usually large enough to accommodate more plates than the initial design requires. More plates can be added if the capacity of the original system is increased. Figure 2-28 shows an example of the additional "rack space" available on a large plate & frame heat exchanger.

The extended ends of the tension rods are typically covered with plastic tubes for safety.

Although plate & frame heat exchangers can be disassembled for cleaning or plate replacement, this is a laborious task. *It is always better to use proper dirt separation details in the system to prevent debris from entering the heat exchanger.*

As is true with all heat exchangers, it is very important to pipe flat plate heat exchangers so that the two liquids pass through it in opposite directions (e.g., counterflow). In most cases, each fluid passage through a flat plate heat exchanger leads back to two connections, one above the other, at one end of the heat exchanger, as shown in Figure 2-19.

Flat plate heat exchangers have several benefits relative to other heat exchanger designs. They include:

• A very high ratio of surface area to internal volume. This allows flat plate heat exchangers to be significantly smaller than a shell & tube or shell & coil heat exchanger of equivalent heat transfer capacity. It also allows for a rapid response to temperature changes in either fluid. Small brazed plate heat exchangers can achieve steady state conditions within a few seconds after two stable fluid streams (e.g., constant flow rate and stable entering temperature) begin flowing through them.

• The metal plates can be thinner than the tubing used in other types of heat exchangers. This reduces the thermal resistance between the two fluids. Later Sections will show

how to account for plate thickness when determining heat exchanger performance.

• The patterns used on the plates allow for relatively high turbulence, which increases convective heat transfer and potentially reduces the required internal surface area of the heat exchanger. Higher turbulence also decreases the potential for dirt or other fouling materials to stick to the plates.

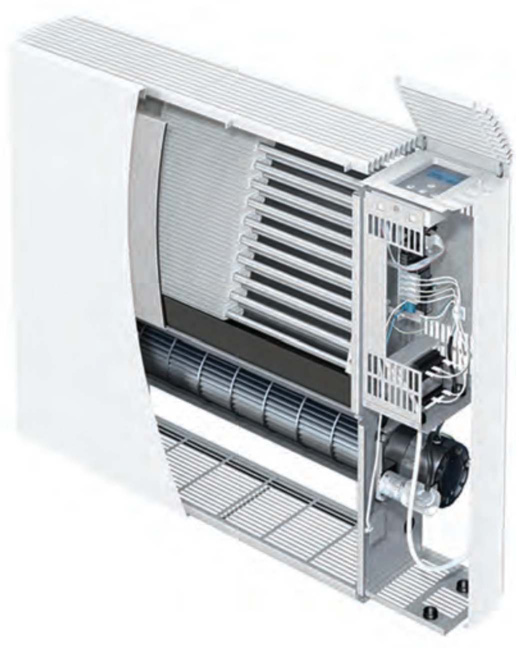
FAN-FORCED LIQUID-TO-AIR HEAT EXCHANGERS

One could think of *any* hydronic heat emitter, be it a cast iron radiator, heated floor slab or a finned-tube baseboard, as a liquid-to-air heat exchanger. All of these heat emitters — to some degree — rely on natural convection heat transfer to move heat from a stream of water to room air. These heat emitters have been discussed in several past issues of *idronics*. In this issue, the discussion focuses on devices that use a fan or blower to create airflow through the heat exchanger.

There are many situations in which the heat in a stream of water needs to be transferred directly to air within an interior space. Likewise, nearly all hydronic chilled-water cooling systems need a means of extracting heat and moisture from inside air. Many products have been developed for these applications using forced convection heat transfer on both the water and air side of the heat exchanger. They can be categorized as follows:

• Fan-coils
• Air handlers

Figure 2-29



Courtesy of Myson

FAN-COILS

A fan-coil is fundamentally a combination of a water-to-air heat exchanger, known as a "coil," with a fan or blower that creates forced convection on the air side of that coil. In general, fan-coils are designed to heat or cool individual spaces within a building. Multiple fans coils can be used to create heating and cooling zones within a building.

All fan-coils can be used for heating when supplied with heated water. Only fan-coils equipped with internal condensate drip pans can be used for cooling when supplied with chilled water.

Figure 2-29 shows the internal components of a modern "console" fan coil. The coil consists of copper tubing routed through closely spaced aluminum fins. It's visible near the top of the unit. A tangential blower wheel located below the coil and rotated by a high-efficiency, electronically commutated motor draws room air into the cabinet from a few inches above floor level. This air is blown across the coil and discharged through the upper grill.

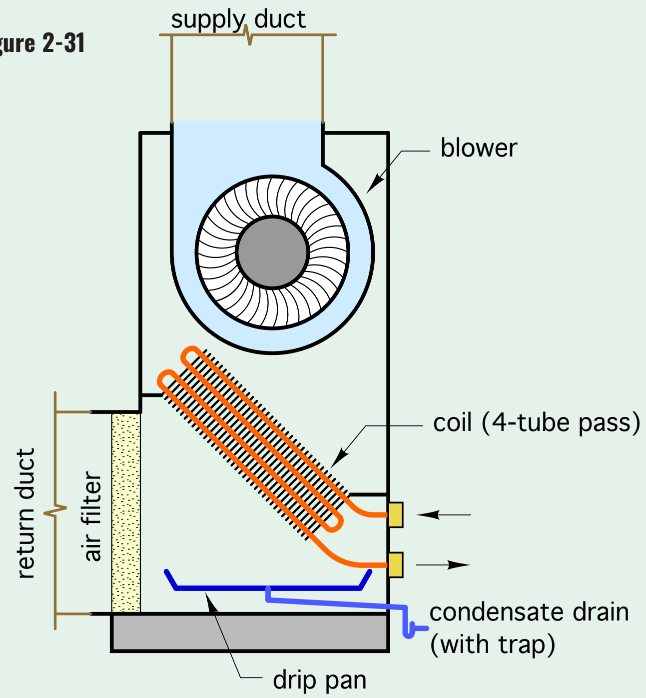
Beyond this basic heat transfer and air movement functionality are controls that vary from one manufacturer to another. In modern fan-coils, these controls can regulate blower speed and operate the fan-coil based on specific setpoint temperatures for heating and cooling and time of day.

Another form factor that is currently used for fan-coils is called a "high wall cassette," an example of which is shown in Figure 2-30.

High wall cassettes are equipped with condensate drain pans and can be used for hydronic heating or cooling. They draw room air into the top of the cabinet and discharge air through a lower opening equipped with oscillating vanes to dispense air in different directions. As shown, the unit in Figure 2-30 is off. The air discharge slot at the bottom of the unit is covered by a motorized damper.

Figure 2-30

Figure 2-31



High wall cassettes are typically turned on and off using a handheld remote. The speed of the fan can be adjusted from the remote or automatically for specific modes of operation. For example, a low fan speed is used when the unit is set for dehumidification mode, allowing the coil to extract more moisture from the air stream.

Some fan-coils are available for recessed mounting in wall cavities. Some have air filters, while others do not. They are available in a wide range of heating and cooling capacities depending on size and source water temperature.

AIR HANDLERS

The combination of a water-to-air heat exchanger and blower is also the basis for a category of devices known as air handlers. Figure 2-31 shows a typical vertically oriented hydronic air handler. Air handlers are also available with horizontally oriented cabinets.

Air handlers, as well as some fan-coils, are available in either “2-pipe” or “4-pipe” configurations. These configurations are shown in Figure 2-32.

Figure 2-32

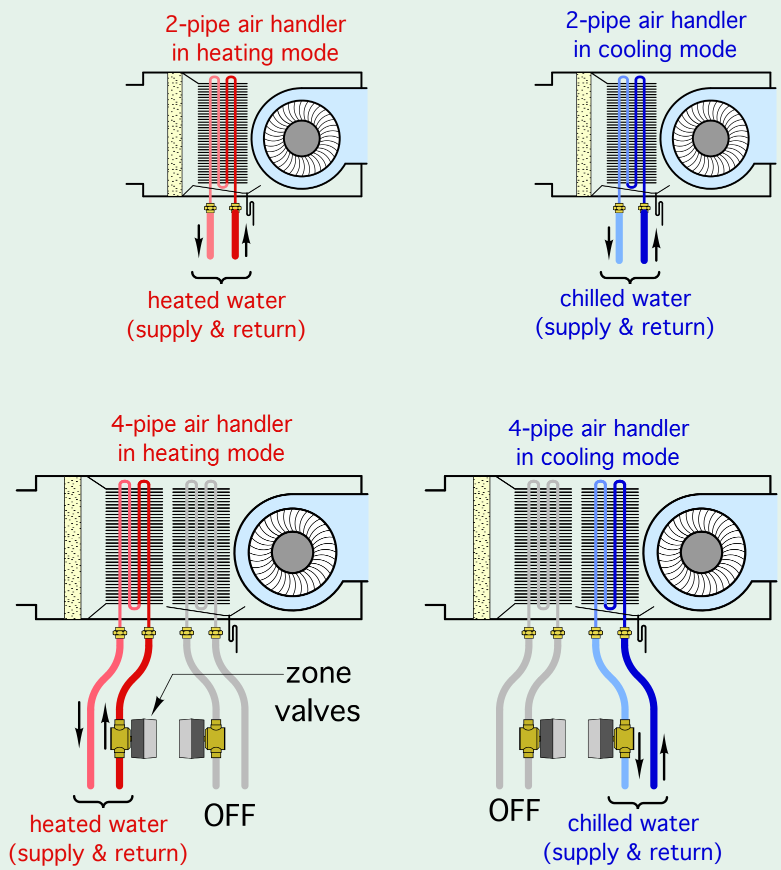


Figure 2-33

A "2-pipe" air handler has a single coil heat exchanger. That coil can provide heating or cooling depending on the temperature of the water stream supplied to it. Two-pipe air handlers are supplied from a distribution system that has a single supply main and single return main. This distribution system can operate with heated water or chilled water, but not both at the same time. Two-pipe air handlers are used in applications where the entire distribution system operates in a single mode (e.g., heating mode or cooling mode) at any given time.

Four-pipe air handlers have *two* water-to-air coil heat exchangers. One coil is piped to a distribution system for chilled water. The other coil is piped to a separate distribution system for heated water. Any 4-pipe air handler connected to these two independent distribution systems can operate in either heating mode or cooling mode. The operating mode is usually determined by a thermostat in the space served by the air handler. When heating is needed, a zone valve opens to allow hot water to flow through the heating coil in the air handler. The blower is also turned on. When the thermostat calls for cooling, a separate zone valve opens to allow chilled water to flow through the chilled water coil, along with blower operation. Four-pipe distribution systems connected to multiple 4-pipe air handlers allow for simultaneous heating and cooling in different areas of a building.

Notice that the direction of water flow through the coils in both 2-pipe and 4-pipe air handlers is opposite the direction of airflow. This is important in achieving counterflow heat exchange, and thus the highest possible rate of heat transfer for a given set of operating temperatures and flow rates.

Figure 2-33 shows a partially installed 4-pipe air handler that's suspended from a concrete slab and above what will eventually be the room's ceiling. This is a common air handler configuration for buildings such as hotels using a 4-pipe distribution system.

The four copper tubes seen at the left of the air handler cabinet provide supply and return water flow to the heating and cooling coils. They are fitted with adapters to connect to PEX distribution tubing. Each coil circuit is controlled by a *Z-one*™ zone valve within the cabinet. Each coil circuit is also equipped with a *QuickSetter*™ flow-balancing valve within the cabinet. The condensate drip pan, which is connected to a 3/4" PVC drainpipe, is visible at the base of the unit.

Figure 2-34

One fundamental distinction between fan coils and air handlers is that the latter is typically designed to connect to a ducted air delivery system. This allows the air stream created by the blower to be simultaneously delivered to several locations in a building.

Another distinction is that most air handlers are intended to be installed in non-occupied locations such as mechanical rooms, interior soffits, basements or attics. They are not designed with aesthetic details to make them acceptable in finished occupied spaces.

Figure 2-34 shows a small, horizontally oriented, 2-pipe air handler connected to a ducted forced-air delivery system.

This unit is mounted in an accessible area within a conditioned attic. A removable panel will eventually be installed to conceal the unit. The coil in this air handler is supplied by insulated piping carrying chilled water for cooling. Because it is installed above a finished space, this air handler is mounted over a secondary drain pan that would capture any condensate that might leak from the unit's primary condensate drip pan. The condensate formed on the coil during cooling mode is carried to a drain through the small PVC pipe seen at the lower right of the unit.

Small air handlers that are commonly used in homes or small commercial buildings may have heating and cooling capacities ranging from about 12,000 Btu/hr (1 ton) to about 60,000 Btu/hr (5 tons). However, many air handlers are available with heating and cooling capacities much higher than those of fan-coils.

Large air handlers can have heating and cooling capacities of several million Btu/hr. These large units are typically custom-built for specific applications. They typically have separate water-to-air coils for heating and cooling, separate blowers for the supply and return air streams, energy recovery devices, and more elaborate air-filtering systems compared to small air handlers. Figure 2-35 illustrates some of these internal details.

The thermal performance of fan-coils and air handlers will be discussed in Section 4.

GREYWATER HEAT RECOVERY HEAT EXCHANGERS

Heat exchangers are also used in plumbing applications. One example is recovering heat from domestic hot water that has passed through a fixture such as a lavatory or shower. This water is commonly referred to as "greywater."

Much of the heat in domestic hot water remains in that water as it goes down the drain. In most buildings, this heat is simply carried into the sewer. However, it is possible to recover up to 40% of this otherwise wasted heat using a greywater heat exchanger, an example of which is shown in Figure 2-36.

Figure 2-35

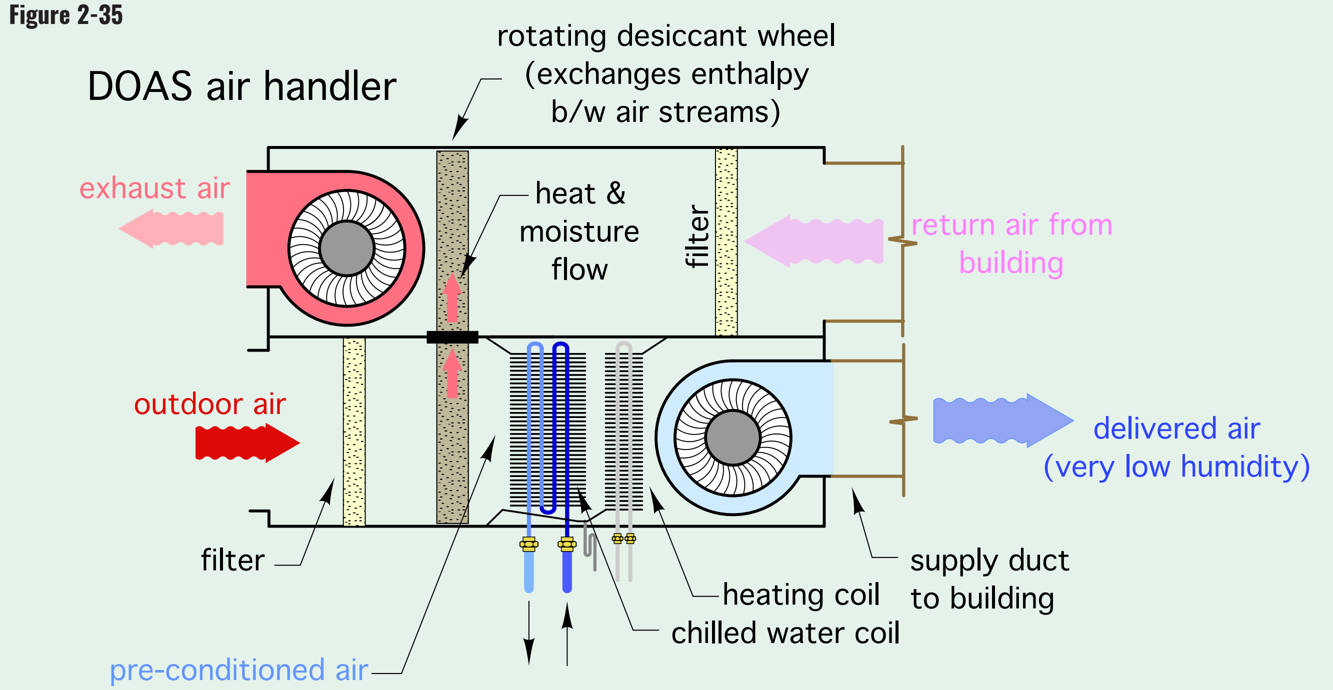


Figure 2-36

Source: Ecoinnovation.ca

Greywater heat exchangers are simple devices. They are fitted into a vertical section of greywater drainage piping. That pipe should be configured to carry drainage water from lavatories, showers, tubs and clothes washers, but not drainage from garbage disposals, toilets or bidets.

These heat exchangers consist of an inner copper pipe that's wrapped with one or more coils of tightly fitting, partially flattened copper tubing that is bonded to the outer surface of the inner pipe. The construction of a typical greywater heat exchanger is shown in Figure 2-37. A typical installation configuration is shown in Figure 2-38.

Greywater heat exchangers leverage the simultaneous flows of cold water and hot water to fixtures such as lavatories and showers. As hot domestic water is being used, cold domestic water enters the bottom connection of the heat exchanger and passes upward in a counterflow direction to the greywater. Heat from the greywater transfers through the copper tube walls and preheats cold domestic water. The two streams of water are always separated by *two*

Figure 2-37

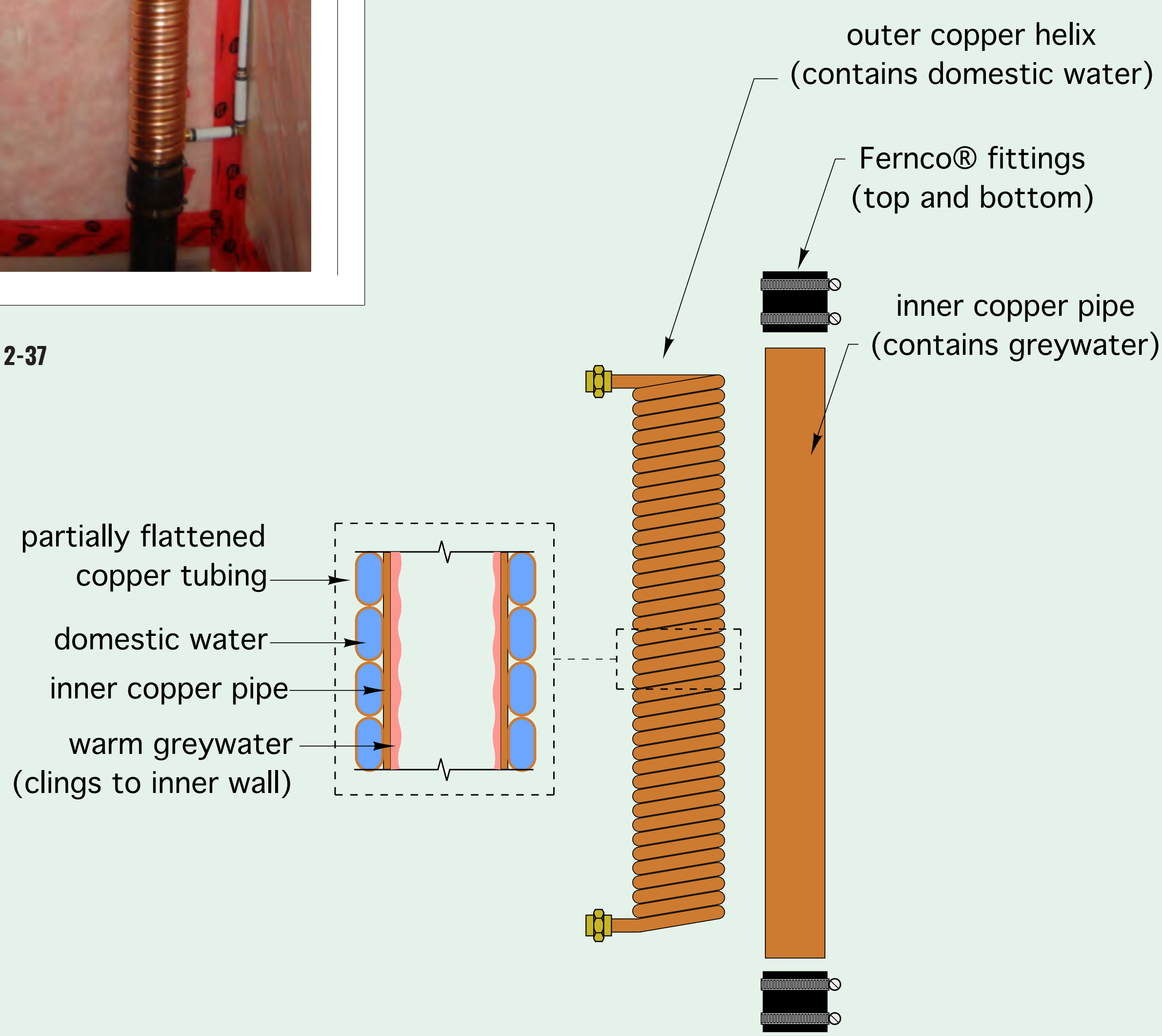
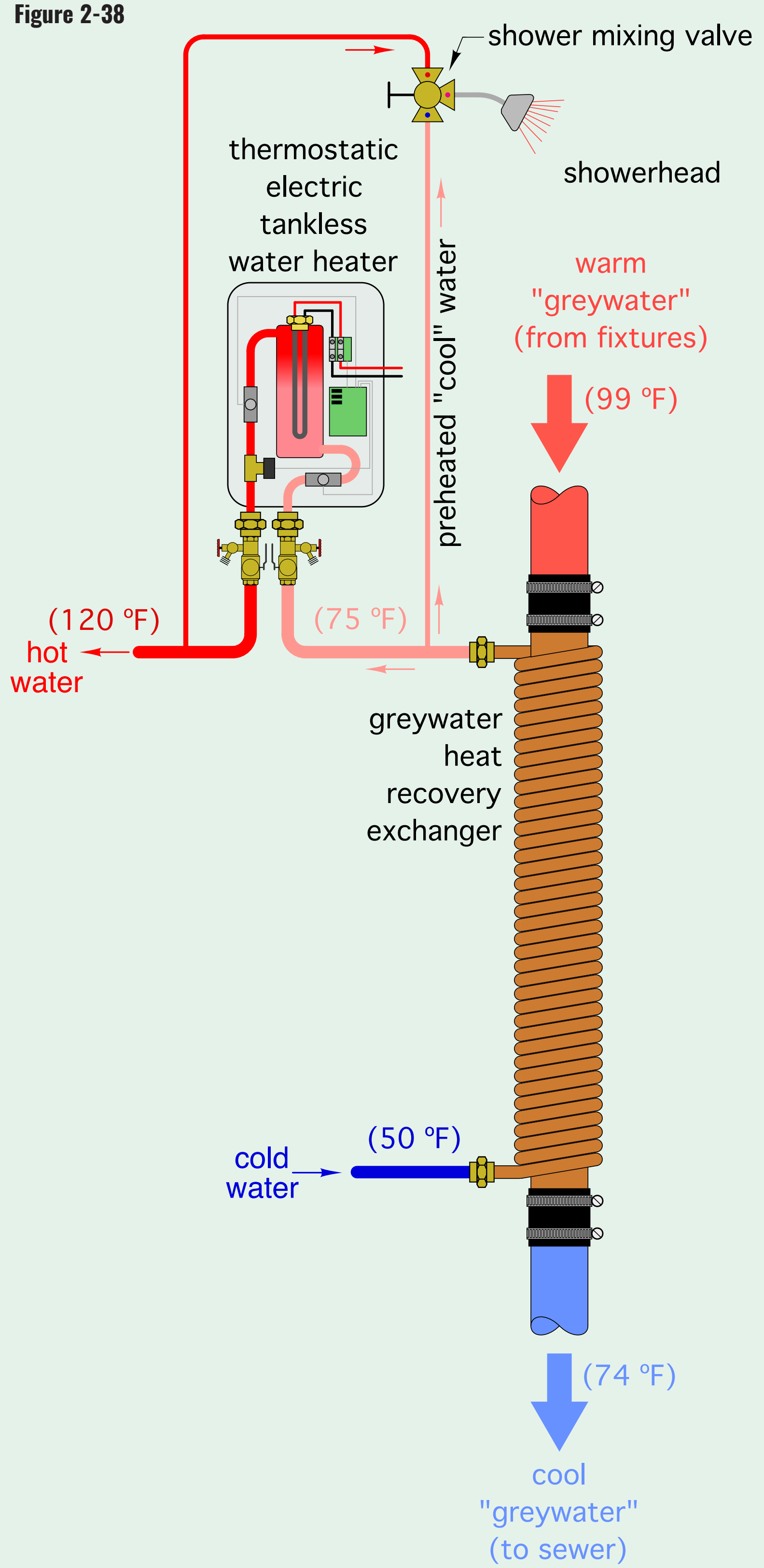


Figure 2-38



copper tube walls. A potential leak in either the inner pipe or outer coil would not cause contamination of domestic water.

In Figure 2-37, some of the preheated water leaving the greywater heat exchanger is piped directly to the "cold" water port of the shower valve. This reduces the required flow of fully heated domestic hot water to the shower valve.

Under typical operating conditions, the entering cold domestic water will be warmed 20° to 25ºF before it exits the heat exchanger. Thus, water entering the building at 50ºF, would be preheated to a temperature of 70° to 75ºF before entering the water heater. This reduces the water heating load by 29% to 36%, assuming a final desired delivery temperature of 120ºF. This is a significant contribution for a simple, unpowered and relatively inexpensive device.

Greywater heat exchangers should only be installed in vertical drainage pipes. They rely on the "film effect" of greywater passing through a *vertical* drainage pipe. Most of this water clings to the pipe wall, rather than falling through the air space inside the pipe. This is ideal from the standpoint of extracting heat from the inner copper pipe.

3. HEAT TRANSFER FUNDAMENTALS

Two fundamental concepts apply to all situations involving heat transfer:

1. Heat always moves from a material at some temperature to another material at a lower temperature.

2. The rate of heat transfer depends on the temperature difference between the two materials. The greater this difference, the higher the rate of heat transfer.

To understand the processes at work in different types of heat exchangers, it's essential to understand the three modes of heat transfer. A well-established scientific principal is that all heat movement occurs by one or more of the following modes:

- Conduction
- Convection
- Thermal radiation

CONDUCTION

Conduction heat transfer takes place within solid materials. It's the result of atomic vibrations within those materials. The atoms in all materials having temperatures above absolute zero (-458°F) vibrate to some extent. The higher the material's temperature, the more vigorous the atomic vibrations.

When heat is added to a solid material, the atomic vibrations become more energetic. These vibrations spread out across trillions of atoms that are bonded together, moving towards material at a lower temperature.

Figure 3-1

The physical property that determines how well a material transfers heat by conduction is called its *thermal conductivity*. The higher a material's thermal conductivity value, the faster heat can pass through it, with all other conditions being equal. The thermal conductivity of a material is usually determined by testing. The thermal conductivity of many materials is listed in references such as the *ASHRAE Handbook of Fundamentals*.

Heat moving through cast iron that separates the combustion gases from the water inside a boiler is an example of conduction heat transfer. Heat passing from tubing embedded in a concrete slab to the surface of that slab is another example. Heat passing from a person's hand to the cold metal handle of a snow shovel is another.

The *rate* of heat transfer by conduction is directly proportional to both the temperature difference across the material and its thermal conductivity. It is inversely proportional to the thickness of a material. Thus, if one were to double the thickness of a material while maintaining the same temperature difference between its sides, the rate of heat transfer through the material would be cut in half.

Materials such as copper, aluminum and steel have relatively high thermal conductivities. Copper and aluminum in particular are frequently used in applications where high rates of conduction heat transfer are necessary.

The *rate* of conduction heat transfer depends on the thermal conductivity of the material, its thickness, the area across which heat is passing, and the temperature difference across it. The relationship between these quantities is given as Formula 3-1.

Formula 3-1:

$$q = A\left(\frac{k}{\Delta x}\right)(\Delta T)$$

Where:
q = rate of heat transfer through the material (Btu/hr)
k = thermal conductivity of the material (Btu/°F•hr•ft)
Δx = thickness of the material in the direction of heat flow (ft)
ΔT = temperature difference across the material (°F)
A = area heat flows across (ft^2)

The mathematical term (k/Δx) in Formula 3.1 is sometimes called a heat transfer coefficient and is represented by the symbol (U). The reciprocal of the heat transfer coefficient U (e.g., 1/U) is called "thermal resistance," and is represented by (R).

Formula 3-1 can therefore be modified into Formula 3-2:

Formula 3-2:

$$q = \frac{A}{R}(\Delta T)$$

Where:
q = rate of heat transfer through the material (Btu/hr)
R = thermal resistance (or "R-value") of a material (ºF•hr•ft^2/Btu)
ΔT = temperature difference across the material (°F)
A = area heat flows across (ft^2)

The R-value of a material can be determined based on its thermal conductivity (k), and its thickness (ΔX). The relationship is given as Formula 3-3.

Formula 3-3:

$$R = \frac{\Delta x}{k}$$

Where:
k = thermal conductivity of the material (Btu/°F•hr•ft)
Δx = thickness of the material in the direction of heat flow (ft)

The U-values and R-values of materials used to build heat exchangers are critically important. In general, materials and thicknesses that provide high U-value, and thus low R-value, allow for higher thermal performance when heat needs to pass through a solid material separating the two fluids.

CONVECTION

Convection heat transfer occurs as the result of fluid movement. The fluid can be a liquid or a gas.

When heated, fluids expand. This lowers their density relative to surrounding cooler fluid. Lowered density increases buoyancy, which causes the warmer fluid to rise. Examples of the latter include warm air rising toward the ceiling in a room and heated water rising to the upper portion of tank-type water heater. Both processes occur without circulators or blowers. As such, they are examples of "natural" convection.

Convection heat transfer is also responsible for moving heat between a fluid and a solid.

For example, consider water at 100ºF flowing along a solid surface that has a temperature of 120ºF. The cooler water molecules contacting the warmer surface absorb heat from that surface. These molecules are churned about as the water moves along. Molecules that have absorbed heat from the surface are constantly being swept away from that surface into the bulk of the water stream and replaced by cooler molecules. One can envision this form of convective heat transfer as heat being "scrubbed off" the surface by the flowing water.

The *speed* of the fluid moving over the surface greatly affects the *rate* of convective heat transfer.

Most people have experienced "wind chill" as cold outside air blows past them. They feel "colder" compared to how they would feel if standing in still air at the same temperature. The faster the air blows past them, the greater the *rate* of convective heat transfer between their skin or clothing surfaces and the air stream. Although the person may *feel* as if the moving air is colder, it isn't. Instead, they're experiencing an increased *rate* of heat loss due to enhanced convective heat transfer. Achieving the same cooling sensation from calm air would require a much lower air temperature.

Convective heat transfer increases with increasing fluid speed. This happens because a layer of fluid called the "boundary layer," which clings to surfaces, gets thinner as the fluid's velocity increases (see Figure 3-2).

The thinner the boundary layer, the lower the thermal resistance between the core of the fluid stream and the surface. Less thermal resistance allows for higher rates of heat transfer between the fluid molecules in the core of the stream and the tubing wall.

Figure 3-2

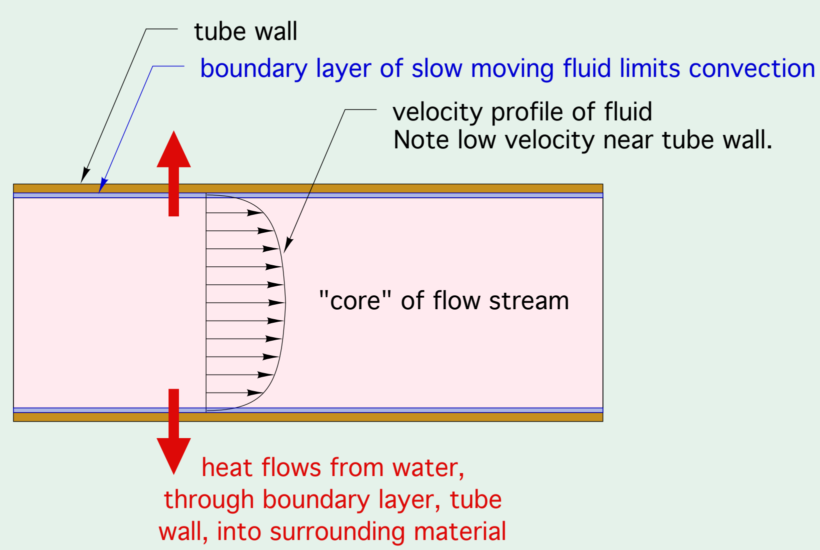
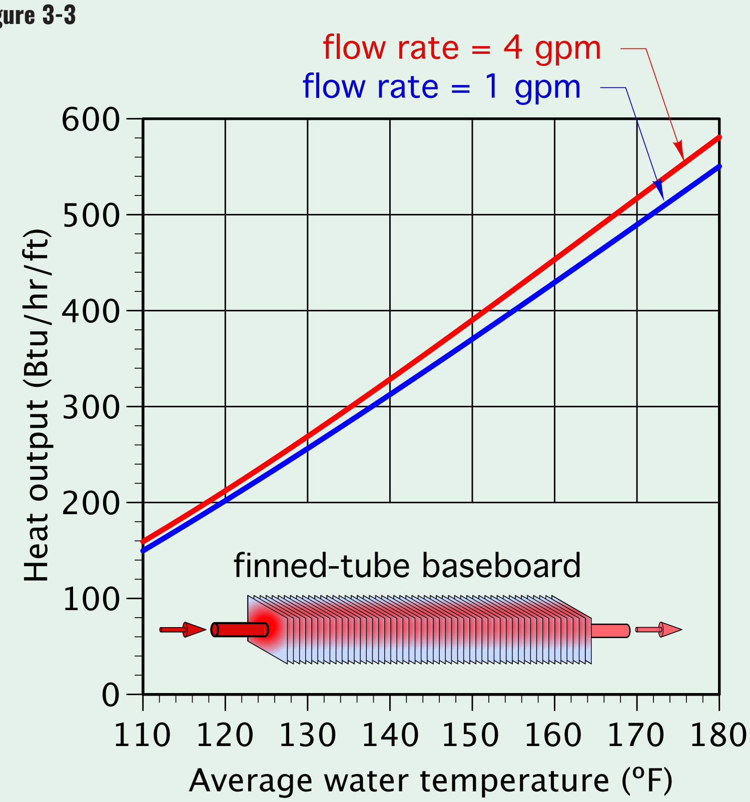


Figure 3-3



The relationship between convective heat transfer and fluid speed across a surface can be seen in many hydronic systems. For example, the faster water flows through a finned-tube baseboard, the higher it's heat output, assuming all other conditions remain the same. This effect is evident in the heat output rating data listed by manufacturers of finned-tube baseboard. Figure 3-3 shows a plot of this data for a typical residential finned-tube baseboard.

At any given water temperature, the heat output occurring at a flow rate of 4 gallons per minute (gpm) is slightly higher than at a flow rate of 1 gpm. This increase is due to a thinner boundary layer between the bulk of the fluid stream and the inner tube wall of the finned-tube element.

In general, the faster one or both fluids flow through any type of heat exchanger, the higher the rate of heat transfer between those fluids. _From the standpoint of heat transfer only, there is no such thing as the water moving "too fast" through any type of heat exchanger._ However, other considerations, such as pumping power, noise or long-term erosion of materials, impose practical limits on flow velocity through heat exchangers.

One example of increasing convective heat transfer with increasing flow rate can be found in heat output ratings for the liquid-to-air heat exchanger in a small fan-coil unit. Figure 3-4 shows how heat output increases with increasing water flow rate through the coil heat exchanger, while the inlet water temperature, the incoming air temperature and the airflow rate all remain constant.

The heat output rate increases rapidly at low flow rates. It continues to increase as flow rate increases, but the *rate of increase* is much slower

Figure 3-4

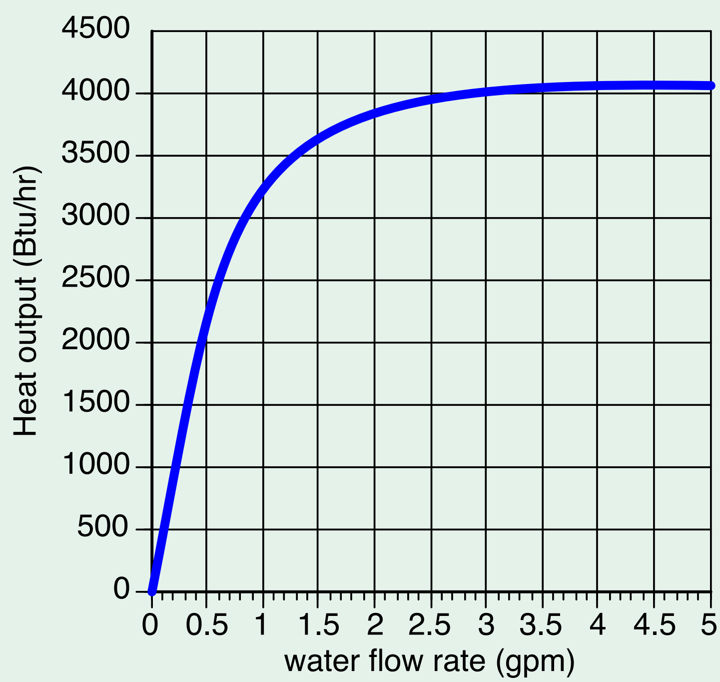
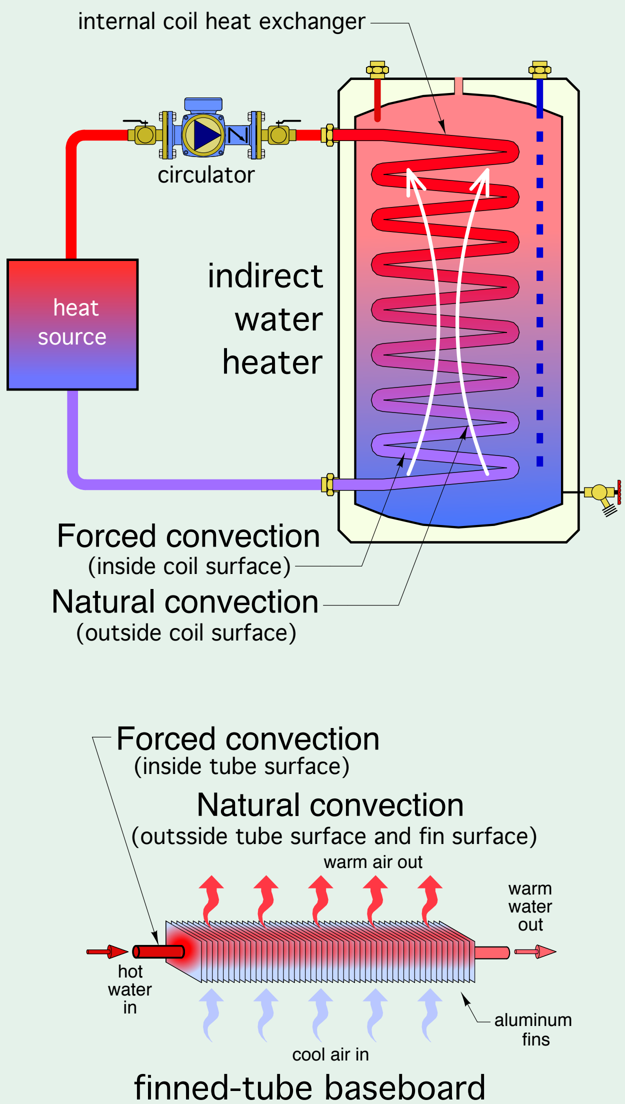


Figure 3-5



at higher flows. _This demonstrates that there are practical limits on how much the heat output of a hydronic heat emitter can be increased based on higher flow rates._

NATURAL vs. FORCED CONVECTION

When fluid motion is caused by a circulator, a blower, a fan or any other powered device, the convective heat transfer is called "forced convection." When the fluid motion is strictly the result of buoyancy differences within the fluid, the convective heat transfer is called "natural convection."

There are many types of heat exchangers that operate with natural convection on one surface and forced convection

Figure 3-6

situation	range of convection coefficient (h) (Btu/hr•ft²•°F)
natural convection involving air	1-5
forced convection involving air	3-100
convection involving water	20-3000

on the other side of that surface. One example is at the external surface of an internal coil heat exchanger within an indirect water heater. Another is the external surface of finned-tube baseboard, as shown in Figure 3-5.

Natural convection is typically a much "weaker" form of heat transfer compared to forced convection. This is the result of much slower fluid motion created by buoyancy differences in the fluid versus faster fluid motion created by a circulator or blower. Slower fluid motion increases boundary layer thickness, which creates greater resistance to heat transfer between the bulk of the fluid and the surface.

The difference between forced convection and natural convection explains why a small wall-mounted fan-coil that's only 18 inches wide can provide the equivalent heat output of 10+ feet of finned-tube baseboard when both are operating at the same water supply temperature and flow rate. The rate of heat transfer from the finned-tube element in the baseboard is limited by natural convection heat transfer between its outer surfaces and the surrounding air.

The rate of convective heat transfer can be estimated using Formula 3-4.

Formula 3-4:

$$q = hA(\Delta T)$$

Where:
q = rate of heat transfer by convection (Btu/hr)
h = convection coefficient (Btu/hr•ft^2•°F)
A = area over which fluid contacts a surface with which it exchanges heat (ft^2)
ΔT = temperature difference between bulk fluid stream and surface (°F)

Although Formula 3-4 is relatively simple, determining the value of the convection coefficient (h) is often a complex process. The value of (h) varies widely depending on the "geometry" of the surface relative to the fluid, the speed of the fluid and the physical properties of the fluid (e.g., thermal conductivity, specific heat, density and viscosity). In many cases, the value of (h) needs to be determined experimentally. Heat transfer textbooks sometimes list values of (h) or methods for determining (h) for very specific and simplified situations. The table in Figure 3-6 shows how wide the range of (h) values can be for specific situations.

THERMAL RADIATION

Heat transfer by thermal radiation is less intuitive compared to conduction and convection. To many, the word "radiation" denotes an undesirable or harmful effect associated with *nuclear* radiation. However thermal radiation and nuclear radiation are very different. Human skin as well as clothing surfaces emit thermal radiation to any surrounding surface that's at a lower temperature. A lightly clothed human body releases over half of its metabolic heat production as thermal radiation.

It's helpful to think of thermal radiation as *light*. More specifically *infrared* light. *Human eyes cannot see infrared light.* However, just like visible light, infrared light (a.k.a. thermal radiation) travels outward from its source in straight lines at the speed of light (186,000 miles per second). Although it cannot "bend" around corners, thermal radiation can be reflected by some surfaces. *It also travels equally well in any direction*, from some surface that emits it to another surface that absorbs it. For example, just as a ceiling-mounted lighting fixture can shine visible light onto objects below, thermal radiation can shine down from a heated ceiling and be absorbed by objects below.

The instant thermal radiation is absorbed by the surface of an object, it becomes heat that warms that object.

Unlike conduction or convection, thermal radiation needs no material (e.g., a fluid or solid) to transfer heat from one location to another. Thermal radiation cannot pass through a solid (as radiation). It can, to differing extents, pass through gases with minimal warming effect on those gases. Intense sunlight passing through very cold air in the upper layers of the earth's atmosphere is an example of the latter.

Consider a person kneeling a few feet away from a campfire on a cold winter day. If pointed toward the fire, their face likely feels warm, even though the air surrounding them is cold. This sensation is the result of thermal radiation emitted by the fire traveling through the cold air and being absorbed by their exposed skin. The air between the fire and the person absorbs very little of the energy being transferred from the flames to the person's face. Likewise, thermal radiation emitted from the warm surface of a heat emitter such as a panel radiator can pass through the air in a room without first heating that air. When the thermal radiation strikes another surface in the room, most of it is absorbed. At that instant, the energy carried by the thermal radiation becomes heat.

Figure 3-7

Every surface continually emits thermal radiation to any cooler surface within sight of it. The surface of a heat emitter that is warmer than our skin or clothing surfaces transfers heat to us by thermal radiation. Likewise, our skin and clothing continually give off thermal radiation to any surrounding surfaces at lower temperatures. A lightly clothed person standing next to a large cold window surface emits significant thermal radiation to that cold surface. This eventually leads to discomfort, even when the air temperature surrounding the person is in the normal comfort range of 68-72ºF.

When thermal radiation strikes an opaque surface, part of it is absorbed as heat and part is reflected away from the surface. The percentage of incoming radiation that is absorbed or reflected is determined by the optical characteristics of the surface and the wavelength of the radiation. Most interior building surfaces absorb the majority of thermal radiation that strikes them. The small percentage that is reflected typically strikes another surface within the room where most of it will be absorbed, and so on. Very little, if any, thermal radiation emitted by warm surfaces in a room escapes from the room.

The rate at which thermal radiation transfers heat between two surfaces depends upon their temperatures, an optical property of each surface called emissivity, as well as the angle and distance between the surfaces.

The rate of heat exchange by thermal radiation between two parallel flat surfaces having the same area can be estimated using Formula 3-5.

Formula 3-5:

$$q = \frac{sAF_{12}\left(T_1^4 - T_2^4\right)}{\left[\frac{1}{e_1} + \frac{1}{e_2} - 1\right]}$$

Where:

q = rate of heat transfer from hotter to cooler surface by thermal radiation (Btu/hr)

s = Stefan Boltzmann constant = 0.1714×10^{-8} Btu/hr•ft^2•°R^4

F_{12} = shape factor between the two surfaces (unitless)

A = area of either surface (ft^2)

T_1 = *absolute* temperature of the hotter surface (°R)

T_2 = *absolute* temperature of the cooler surface (°R)

e_1 = emissivity of the hotter surface (unitless)

e_2 = emissivity of the cooler surface (unitless)

This formula is more complex that those used for estimating conduction and convection heat transfer.

The value of the "shape factor" (F_{12}) is a number between 0 and 1.0. It's determined based on the relative angle and distance between the two surfaces exchanging radiant heat. Heat transfer textbooks give specific methods for finding values of the shape factor (F_{12}) for different surfaces and orientations. For two parallel planes having infinite width and depth, the value of the shape factor (F_{12}) is 1.0.

Factors e_1 and e_2 are the *emissivities* of surfaces 1 and 2. Emissivity is a surface property determined experimentally based on how well the surface emits thermal radiation. It must be a number between 0 and 1. A high value indicates that the surface is a good emitter, and vice versa. Emissivity values for various surfaces can be found in references such as heat transfer handbooks. Interestingly, the emissivity of a surface is not necessarily correlated with its color. A rough metal surface coated with *white* enamel paint has an emissivity of 0.91, and a flat black painted surface has an emissivity of 0.97. Freshly fallen snow can have an emissivity over 0.90. The emissivity of a polished copper surface is 0.023, while a heavily oxidized copper surface has an emissivity of 0.78. Most highly polished metal surfaces have low emissivities, and thus would not be good choices for the surface of a hydronic heat emitter that's expected to radiate heat into a room.

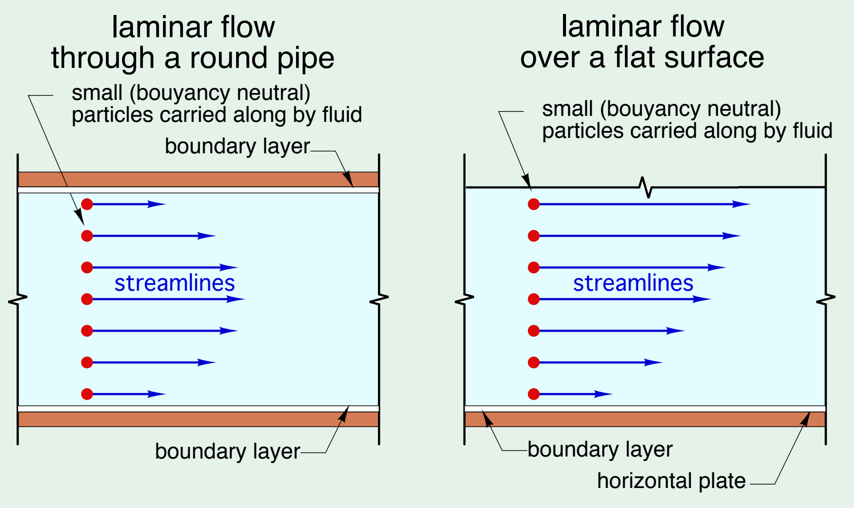
It's also important to understand that the temperatures (T1) and (T2) in Formula 3-5 must be *absolute temperatures*. Temperatures in °F can be converted to absolute temperatures in degrees Rankine (°R) by adding 458 degrees. Thus, 32°F becomes 32 + 458 = 490°R.

The mathematical result of the calculation (T_1^4-T_2^4) changes much more than the simple ΔT term used in the formulas for conduction and convection. For example, consider two surfaces exchanging radiant heat with temperatures of 100°F and 80°F. These temperatures would convert to 558°R and 538°R. The *difference* between these temperatures would be only 20°R, the same as the difference between 100°F and 80°F. However, when these Rankine temperatures are used in Formula 3-5 the resulting number for the term (T_1^4-T_2^4) is 131,700,000°R^4.

Due to mathematical complexities, as well as variability or uncertainty in properties such as surface emissivities, theoretical calculations of radiant heat transfer are often limited to relatively simple situations.

4. THERMAL CHARACTERISTICS OF HEAT EXCHANGERS

Figure 4-1



The thermal performance of heat exchangers is very dependent on the physical properties of the two fluids passing through them. It's also dependent on the "dynamic condition" of those fluids. The latter refers to the flow direction, velocity and degree of turbulence present in the fluids exchanging heat.

LAMINAR vs. TURBULENT FLOW

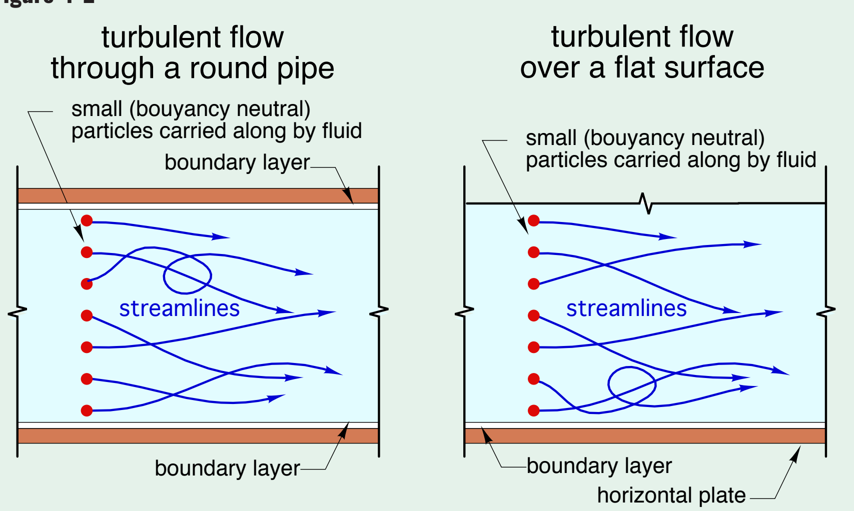
It's helpful to imagine liquids flowing through pipes or along the surface of a plate as a stream containing millions of tiny buoyancy-neutral "particles." The path that any one of these particles takes as it moves is called a *streamline*.

In some situations, the streamlines of all the fluid particles are parallel to each other, like multiple lanes of traffic on a major highway. The particles near the center of the pipe, or those farther away from a flat surface, move faster than those near the pipe wall or the flat surface, but they don't "change lanes." This type of fluid movement, represented by Figure 4-1, is called laminar flow.

Laminar flow can be desirable or undesirable depending on where it occurs in hydronic systems, as well as the intended purpose of those systems. If the goal is to *move fluid long distances with minimal pumping power,* laminar flow is desirable because it reduces pumping power. *However, if the goal is to maximize heat transfer from a flowing fluid, laminar flow is very undesirable.*

Because the fluid particles move in parallel streamlines during laminar flow, there is very little mixing between them. This increases the thickness of the boundary layer between the bulk of the fluid stream and the wall of the pipe or surface of a plate. Thicker boundary layers decrease convective heat transfer.

Figure 4-2



Turbulent flow causes the streamlines of the imaginary fluid particles to bend and twist as the fluid moves down the pipe or along a surface, as shown in Figure 4-2.

The erratically shaped streamlines present during turbulent flow create good mixing. A small quantity of fluid that's close to a solid surface one instant, can be swept into the bulk of the fluid stream the next instant. This decreases the boundary layer thickness and significantly improves convective heat transfer.

The convective heat transfer associated with a relatively viscous liquid at low temperatures, such as a 25% solution of propylene glycol antifreeze at 30ºF, under laminar flow is only about 7% of the convective heat transfer of the same fluid under turbulent flow. This implies that laminar flow will result in a very significant drop in heat transfer between a fluid and the inner wall of a tube or the surface of a plate. Such a change could have a major negative effect on the ability of a hydronic circuit, such as a ground loop heat exchanger supplying a geothermal heat pump, to transfer heat at the required rate.

Although turbulent flow improves convective heat transfer, it also increases head loss, and thus requires more pumping power to maintain a given flow rate relative to the power required for laminar flow. However, the increased heat transfer capabilities of turbulent flow often far outweigh the penalty associated with the increased pumping power. Thus, when possible, *all heat exchangers should be operated with turbulent flow of both fluids.*

REYNOLDS NUMBER

It's possible to predict if flow through a pipe or across a smooth plate will be laminar or turbulent. It's based on calculating a dimensionless quantity called the Reynolds number of the fluid (abbreviated as Re#).

The Reynolds number is a ratio of the inertial forces existing in a flow stream compared to the viscous forces in that flow stream. It takes several physical characteristics of the flowing fluid into account, including the fluid's density and viscosity, both of which are dependent on the fluid's temperature. It also accounts for the speed of the fluid and the geometry of surfaces in contact with the fluid.

When the Reynolds number is low, the inertial forces are relatively weak compared to the viscous forces. Any disturbance to the flow stream that might otherwise induce turbulence is quickly dampened out by viscous forces, and thus sustained turbulence cannot exist. However, when the inertial forces become dominant over the viscous forces, which would be characterized by higher Reynolds numbers, the dampening effects of viscosity are not able to prevent turbulent flow from being established and maintained.

For flow through a round pipe, the Re# is calculated using Formula 4-1.

Formula 4-1:

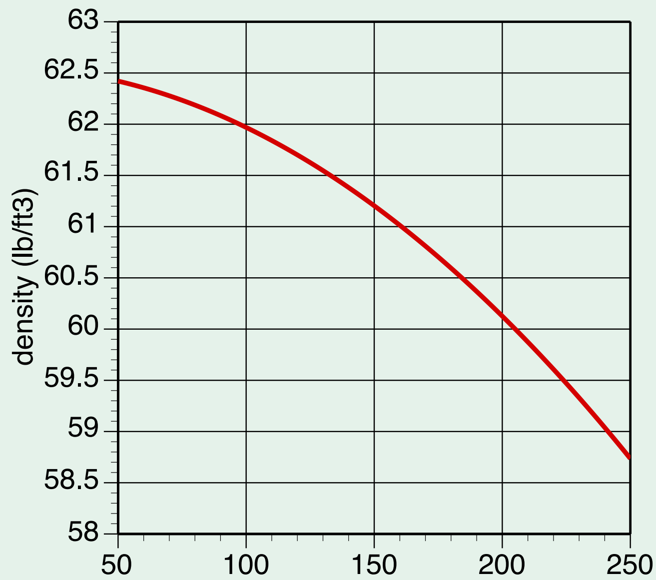
$$Re\# = \frac{vdD}{\mu}$$

where:

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

If the Reynolds number of flow through a round pipe is over 4,000, the flow will be turbulent. If the Re# is below 2,300, the flow through the pipe will be laminar. If the Re# is between 2300 and 4000, the flow could be either laminar or turbulent, and it could switch back and forth.

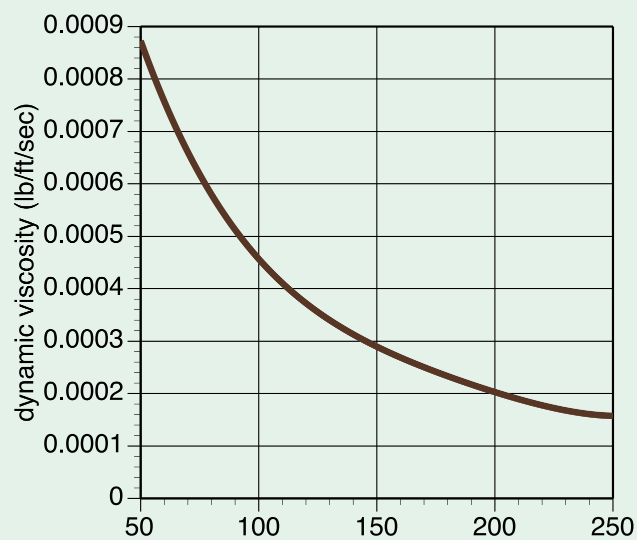
Figure 4-3



Here's an example: Determine the Reynolds number of water at 140°F flowing at 5 gpm through a 3/4-inch type M copper tube. Is this flow laminar or turbulent?

Solution: To calculate the Re#, the density and dynamic viscosity of the water must be determined. The density of water can be found from Figure 4-3. The dynamic viscosity of water can be found from Figure 4-4.

Figure 4-4



At 140ºF, the density of water is read from Figure 4-3 as 61.35 lb/ft^3. From Figure 4-4, the dynamic viscosity of water at 140ºF is 0.00032 lb/ft/sec.

The inside diameter of a 3/4-inch type M copper tube is 0.811 inches. This must be converted to feet to match the stated units for Formula 4-1:

$$d = (0.811\ in)\left(\frac{1\ ft}{12\ in}\right) = 0.06758\ ft$$

The average flow velocity corresponding to a flow rate of 5 gpm can be found using Formula 4-2.

Formula 4-2:

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

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

For 3/4-inch Type M copper tubing operating at a flow rate of 5 gpm, the average flow velocity is:

$$v = \left(\frac{0.408}{d^2}\right) f = \left(\frac{0.408}{(0.811)^2}\right) 5 = 3.1\frac{ft}{\sec}$$

The Reynolds number can now be calculated using Formula 4-1.

$$Re\# = \frac{\left(3.1\frac{ft}{\sec}\right)(0.06758\ ft)\left(61.35\frac{lb}{ft^3}\right)}{0.00032\frac{lb}{ft \cdot \sec}} = 40{,}165$$

This value is well above the 4,000 threshold and therefore the flow is turbulent.

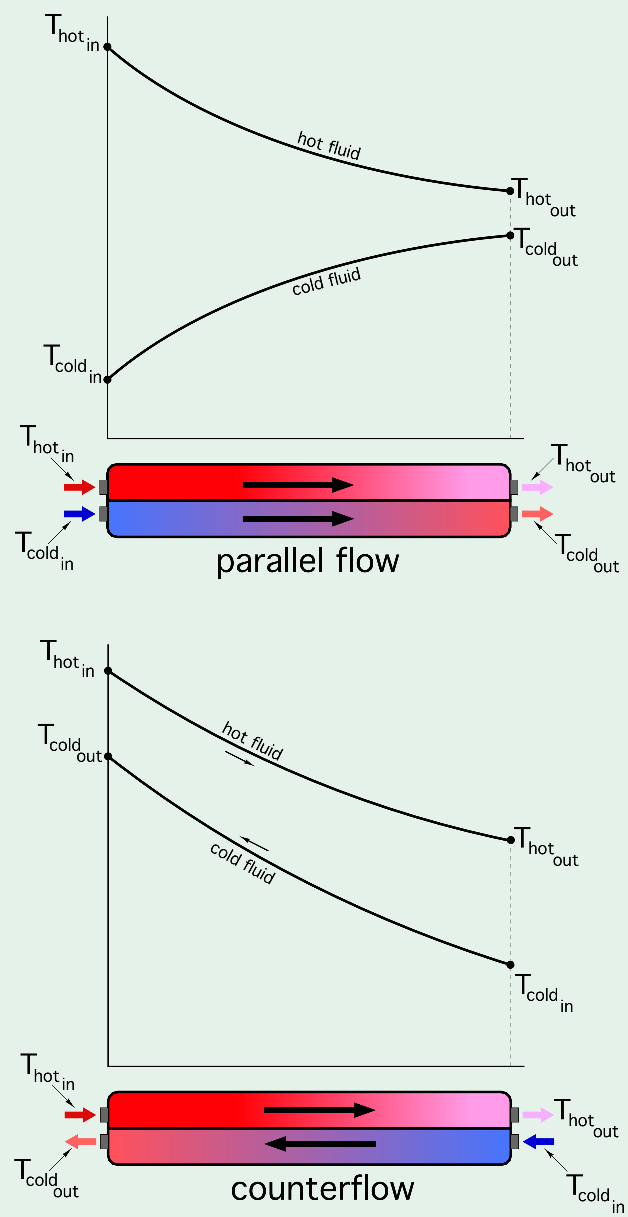
Notice that the units on the quantities used in Formula 4-1 cancel out completely. A valid Reynold number must always be a number — with no units, regardless of the use of the IP or SI unit system. When calculating a Reynolds number, it is good practice to enter the units on all quantities and be sure they all cancel out.

From a practical standpoint, most flows in hydronic heating and cooling systems will have Re# well above 4,000, and thus be turbulent. The higher the Re#, the more intense the turbulence. *Flows through any device that adds or removes heat from a fluid (e.g., any heat exchanger, hydronic heat emitter or cooling coil) should always be turbulent.*

FLOW DIRECTION THROUGH HEAT EXCHANGER

The liquid-to-liquid heat exchangers used in hydronic systems are configured so that the two fluid streams flow in mostly parallel directions to each other. The two parallel streams could be flowing in the same direction or in opposite directions. When the two streams flow in the same direction, the configuration is called "parallel flow," or sometimes "*cocurrent*" flow. When the two flow streams flow in opposite directions, the configuration is called "*counterflow*." These two flow configurations, along with representative temperature changes of both fluid streams, are shown in Figure 4-5.

Figure 4-5



One of the fundamental characteristics of heat exchangers is that the *rate* of heat transfer depends on the temperature difference between the fluid providing the heat and the fluid absorbing that heat. The greater this temperature difference, the higher the rate of heat transfer.

It's apparent that the temperature difference between the hot fluid and cool fluid in the heat exchangers represented in Figure 4-5 varies considerably depending on the location within the heat exchanger where this difference is measured. For the parallel flow heat exchanger, the temperature difference at the left side, where both fluids enter, is very large. But this temperature difference decreases rapidly as the fluids exchange heat and move toward the outlet ports. There's also a variation in the temperature difference between the fluids as they move through the counterflow heat exchanger.

To evaluate the rate of heat transfer across the entire heat exchanger, it is necessary to establish an "average" temperature difference. This average temperature difference could be defined as a value that if present at all locations that separate the fluids within the heat exchanger would result in the same overall heat exchange rate as will occur given the actual and highly variable temperature differences.

Heat transfer theory can be used to derive this "average" ΔT. It's called the *log mean temperature difference (LMTD)* and can be calculated using Formula 4-3.

Formula 4-3:

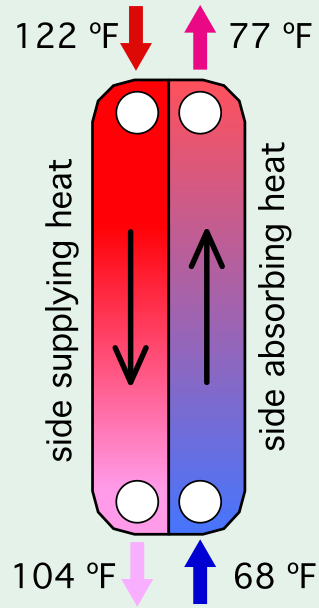
$$LMTD = \frac{(\Delta T)_1 - (\Delta T)_2}{\ln\left[\frac{(\Delta T)_1}{(\Delta T)_2}\right]}$$

Where:
LMTD = log mean temperature difference (ºF)
$(\Delta T)_1$ = temperature difference between the two fluids at one end of the heat exchanger (ºF)
$(\Delta T)_2$ = temperature difference between the two fluids at the other end of the heat exchanger (ºF)
ln [ ] = the natural logarithm of the quantity in the square brackets.

Formula 4-3 applies to both parallel flow and counter flow heat exchangers. It's also important to understand that it makes no difference which end of the heat exchanger is assigned to be $(\Delta T)_1$ as long as the other end is assigned to be $(\Delta T)_2$.

Here's an example: Calculate the LMTD for the heat exchanger shown in Figure 4-6.

Figure 4-6



To calculate the LMTD, assume that $(\Delta T)_1$ is for the top end of the heat exchanger. Thus, $(\Delta T)_1$ = 122-77 = 45 ºF. This means that $(\Delta T)_2$ is at bottom end of the heat exchanger. Thus, $(\Delta T)_2$ = 104-68 = 36ºF. Putting these values for $(\Delta T)_1$ and $(\Delta T)_2$ into Formula 4-3 yields:

$$LMTD = \frac{(\Delta T)_1 - (\Delta T)_2}{\ln\left[\frac{(\Delta T)_1}{(\Delta T)_2}\right]} = \frac{45-36}{\ln\left[\frac{45}{36}\right]} = \frac{9}{0.223} = 40.3^{\circ} F$$

Now suppose that the ends of the heat exchanger representing $(\Delta T)_1$ and $(\Delta T)_2$ were reversed. Thus $(\Delta T)_1$ = 36ºF, and $(\Delta T)_2$ = 45ºF. Putting these values into Formula 4-3 yields:

$$LMTD = \frac{(\Delta T)_1 - (\Delta T)_2}{\ln\left[\frac{(\Delta T)_1}{(\Delta T)_2}\right]} = \frac{36-45}{\ln\left[\frac{36}{45}\right]} = \frac{-9}{-0.223} = 40.3^{\circ} F$$

Heat transfer theory can also be used to prove a very important concept in the application of heat exchangers. Namely that *the LMTD of a heat exchanger configured for counterflow will always be higher than that of the same heat exchanger configured for parallel flow and having the same entering conditions for both flow streams.*

The higher the LMTD, the higher the rate of heat transfer. This implies that *to attain the highest possible rate of heat transfer, heat exchangers should always be configured* *for counterflow rather than parallel flow.* This concept should always be observed when planning and documenting the design of any hydronic system using heat exchangers.

DETERMINING OVERALL RATE OF HEAT TRANSFER (LMTD METHOD)

The heat transfer rate through a heat exchanger is directly proportional to the LMTD. It is also directly proportional to the surface area that separates the two fluids. Thus, the heat transfer rate is proportional to the multiplication of internal surface area (A) times the log mean temperature difference (LMTD). This can be expressed as the proportionality:

Formula 4-4:

$$q \propto A(LMTD)$$

Where:
q = rate of heat transfer across the heat exchanger (Btu/hr)
A = internal surface area separating the two fluids within the heat exchanger (ft^2)
LMTD = log mean temperature difference established by the current conditions (ºF)

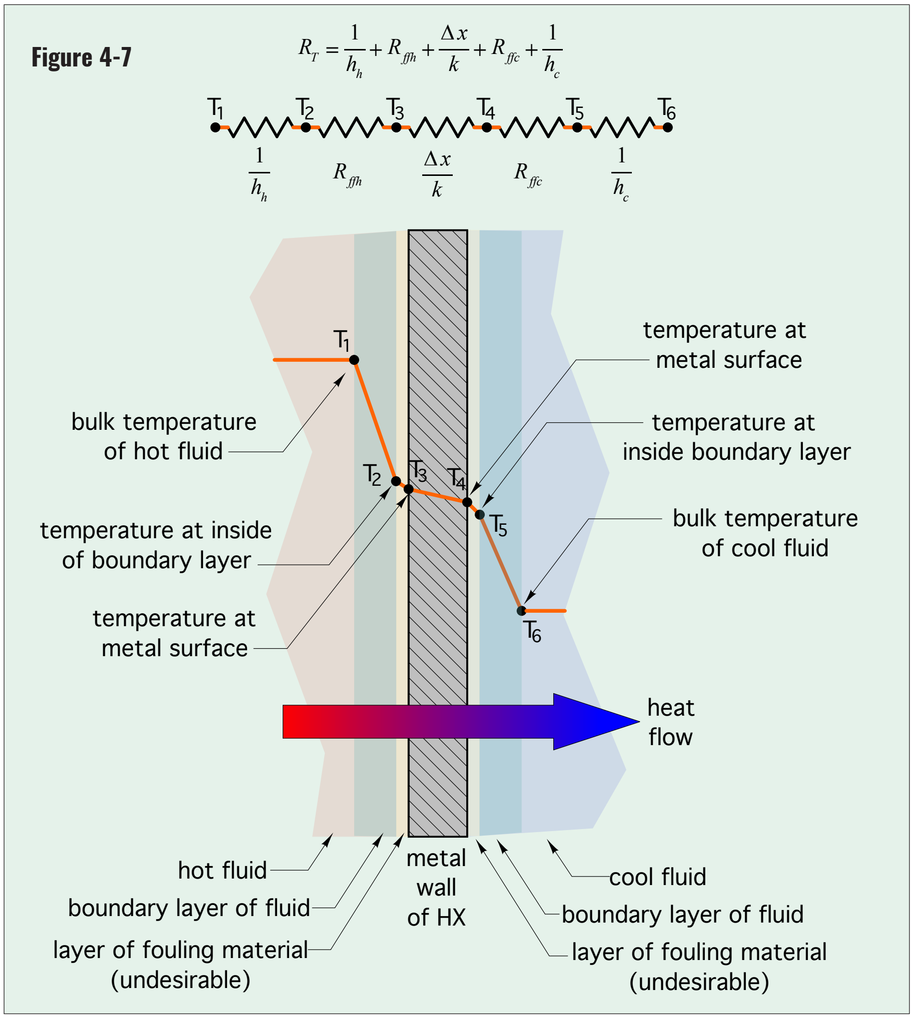
This proportionality can be changed to a formula (e.g., an equality with an = sign, rather than a proportionality) by introducing a constant called U.

Formula 4-5:

$$q = UA(LMTD)$$

The constant U is called the *overall heat transfer coefficient* of the heat exchanger. To make the units in Formula 4-5 consistent, U must have units of Btu/hr/ft^2/ºF. The value of U is the rate of heat transfer through the heat exchanger (in Btu/hr), per square foot of internal surface area, per ºF of log mean temperature difference. The U-value of a heat exchanger involves

Figure 4-7



knowing the convection coefficient for both fluid-to-surface interfaces, as well as the thermal conductivity of the wall separating the fluids. In some cases, additional thermal resistance effects called "fouling factors" are also included in the mathematical definition of the U-value.

Figure 4-7 shows how the U-value for a heat exchanger where the two fluids are separated by a flat plate can be viewed as a sum of several thermal resistances.

The orange lines in Figure 4-7 represent relative temperature changes. The bulk temperature of the hot fluid is represented by the black dot labelled (T1). There is a temperature drop from (T1) to (T2) across the convective boundary layer. The extent of this temperature drop depends on the characteristics of the boundary layer. Laminar flow would result in a relatively thick boundary layer and a larger temperature drop from (T1) to (T2). Highly turbulent flow would decrease the thickness of this boundary layer and reduce the temperature drop across it.

Another temperature drop, (T2) to (T3), occurs across the resistance associated with a fouling layer. If the heat exchanger is new, or completely clean of any dirt or accumulated films, the thermal resistance of the fouling film is zero, and thus the temperature drop from (T2) to (T3) is also zero. If operating conditions allow a fouling film to develop, the thermal resistance called “fouling factor” increases, and so does the temperature drop from (T2) to (T3).

Another temperature drop occurs across the metal wall of the heat exchanger that separates the two fluids. Because of the high thermal conductivity of the wall, the temperature drop from (T3) to (T4) will be quite small, perhaps only a fraction of one degree F. In some situations, this temperature drop is so small relative to the resistance of the fluid boundary layers and fouling films that it’s deemed insignificant and not factored into the calculation of the overall heat transfer coefficient (U).

Similar temperature drops occur on the right side of the metal wall, across any fouling film and the convective boundary layer of the cool fluid.

Each of the temperature drops occurs because of a resistance to heat flow. The diagram at the top of Figure 4-7 represents these five thermal resistances, with the temperatures (T1 through T6) present between these resistances. The numerical value of each thermal resistance can be estimated based on several factors, such as fluid properties, flow velocity, temperature, Reynolds numbers and more. Some of these calculations can be complex and require iteration.

The overall rate of heat transfer depends on these five thermal resistances. Formula 4-5 can be expanded to show how these resistances influence the overall rate of heat transfer. That expansion is shown as Formula 4-6.

Formula 4-6:

$$q = UA(LMTD) = \frac{A(LMTD)}{R_T} = \frac{A(LMTD)}{\left[\frac{1}{h_h} + R_{ffh} + \frac{\Delta x}{k} + R_{ffc} + \frac{1}{h_c}\right]}$$

Where:
q = rate of heat transfer across the heat exchanger (Btu/hr)
A = internal surface area separating the two fluids within the heat exchanger (ft^2)
LMTD = log mean temperature difference established by the current conditions (ºF)
h_h = convection coefficient on the hot side of the heat exchanger (Btu/hr/ft^2/ºF)
R_{ffh} = thermal resistance of fouling factor on hot side (ºF•hr•ft^2/Btu)
ΔX = thickness of metal wall of heat exchanger (ft)
k = thermal conductivity of metal wall of heat exchanger (Btu/hr/ft/ºF)
R_{ffc} = thermal resistance of fouling factor on cool side (ºF•hr•ft^2/Btu)
h_c = convection coefficient on the cool side of the heat exchanger (Btu/hr/ft^2/ºF)

From Formula 4-5 it follows that the overall heat transfer coefficient (U) can be written as Formula 4-7.

Formula 4-7:

$$U = \frac{1}{\left[\frac{1}{h_h} + R_{ffh} + \frac{\Delta x}{k} + R_{ffc} + \frac{1}{h_c}\right]}$$

The values of U are highly dependent on the fluids exchanging heat, the flow conditions and the characteristics of the material of which the heat exchanger is made. For water-to-water heat exchangers a guideline range for values of U is 150-300 Btu/hr/ft^2/ºF.

Numerical values for fouling factors are given in heat transfer reference books or on websites. The units for fouling factors are the same as for insulation R-values or other thermal resistances. In the IP units system, those units are (ºF•hr•ft^2/Btu). Figure 4-8 lists some fouling factors relevant to hydronic heating and cooling applications and their associated fluids.

Figure 4-8

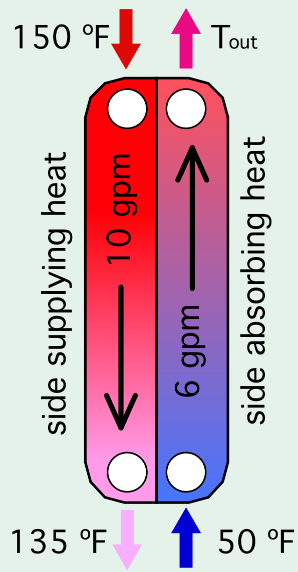
fluid	fouling factor (ºF•hr•ft²/Btu)
river water (untreated)	0.003
cooling tower water (untreated)	0.003
cooling tower water (treated)	0.00102
demineralized water	0.00051

These values show a range of thermal resistances associated with different water conditions. Notice the fouling factor associated with demineralized water is only about 17 percent of that associated with untreated river water or untreated cooling tower water. The presence of minerals in flow streams passing through heat exchangers, especially when those fluids are at elevated temperatures, can significantly increase fouling factors and thus decrease rates of heat transfer. This again points to the importance of using high-efficiency dirt separators as well as demineralized water for optimal hydronic system performance.

The following example illustrates how Formula 4-7 can be used.

A flat plate heat exchanger is used to heat domestic water. Boiler water enters the primary side of the heat exchanger at 150ºF and flow rate of 10 gpm. Cold domestic water

Figure 4-9



enters the secondary side of the heat exchanger at 50ºF and 6 gpm. The plates in the heat exchanger are 0.02 inches thick and made of 316 stainless steel having a thermal conductivity of 29 Btu/hr/ft/ºF. The heat exchanger is new, and therefore there are no fouling films on either side of the plates. The heat exchanger is configured for counterflow, as shown in Figure 4-9. Prior calculations by the designer established that the convection coefficient on the hot side of the heat exchanger is 250 Btu/hr/ft²/ºF, and on the cool side is 150 Btu/hr/ft²/ºF. Assume that the heat exchanger is operating under steady state conditions and is insulated so that external heat loss is negligible. Determine the rate of energy flow through the heat exchanger and the required internal area of the plates.

The rate of heat transfer into the heat exchanger can be determined based on the flow rate, temperature drop and physical properties of water flowing through the left side of the heat exchanger at an average temperature of (150+135)/2 = 142.5 ºF.

The density of water at 142.5ºF is 61.3 lb/ft³. The specific heat of water at this temperature is 1.00 Btu/lb/ºF.

The sensible heat rate equation (Formula 4-8) can be used to calculate the rate of heat transfer.

Formula 4-8:

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

Where:
q = rate of heat transfer (Btu/hr)
D = density of the fluid (lb/ft³)
c = Specific heat of the fluid (Btu/lb/ºF)
f = flow rate (gpm)
ΔT = temperature change of the fluid (ºF)
8.01 = constant to balance the units on each side of formula

Putting the values into Formula 4-8 yields:

$$q=(8.01Dc)f(\Delta T)=(8.01\times 61.3\times 1.00)(10)(150-135)=73{,}650\frac{Btu}{hr}$$

Since the heat exchanger is operating under steady state conditions, and with insignificant external heat loss, the right side must be releasing heat at the same rate as heat is entering the left side. This allows the sensible heat rate equation to be used to determine the outlet temperature on the secondary side.

$$T_{out}=\frac{73650}{(8.01\times 62.4\times 1.00)(6)}+50=74.6^{\circ}F$$

Since both inlet and outlet temperatures are now known, the log mean temperature difference can be calculated:

$$LMTD=\frac{(\Delta T)_1-(\Delta T)_2}{\ln\left[\frac{(\Delta T)_1}{(\Delta T)_2}\right]}=\frac{75.4-85}{\ln\left[\frac{75.4}{85}\right]}=\frac{-9.6}{-0.1198}=80.1^{\circ}F$$

The U value for the heat exchanger can also be determined based on Formula 4-7:

$$U=\frac{1}{\left[\frac{1}{h_h}+R_{ffh}+\frac{\Delta x}{k}+R_{ffc}+\frac{1}{h_c}\right]}=\frac{1}{\left[\frac{1}{250\frac{Btu}{hr\bullet ft^2\bullet^{\circ}F}}+0+\frac{0.00167\,ft}{29\frac{Btu}{hr\bullet ft\ \bullet^{\circ}F}}+0+\frac{1}{100\frac{Btu}{hr\bullet ft^2\bullet^{\circ}F}}\right]}=71.1\frac{Btu}{hr\bullet ft^2\bullet^{\circ}F}$$

Notice that the unit for the thickness of the heat exchanger plates was changed from 0.02 inches to 0.00167 ft. This is necessary to maintain consistent units in all terms, yielding the units of Btu/hr/ft²/ºF for the result.

Formula 4-5 can now be set up and solved for the required internal surface area of the heat exchanger.

$$q=UA(LMTD)=73{,}650\frac{Btu}{hr}=(71.1\frac{Btu}{hr\bullet ft^2\bullet^{\circ}F})A(80.1^{\circ}F)$$

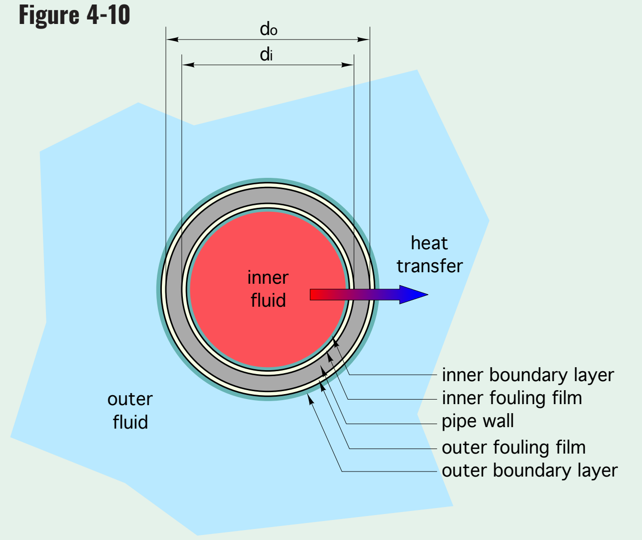
$$A=\frac{73{,}650\frac{Btu}{hr}}{(71.1\frac{Btu}{hr\bullet ft^2\bullet^{\circ}F})(80.1^{\circ}F)}=12.9\,ft^2$$

The designer would now consult manufacturer's specifications to locate candidate heat exchangers with internal surface areas of approximately 12.9 ft^2.

OVERALL RATE OF HEAT TRANSFER – PIPE HEAT EXCHANGER

Another common "geometry" for heat exchangers involves one or more cylindrical pipes separating the two fluids exchanging heat. This is illustrated in Figure 4-10.

Figure 4-10



Formula 4-9, which is similar in mathematical form to Formula 4-6, can be used to calculate the rate of heat transfer for a pipe wall separating a hot and cool fluid.

Formula 4-9:

$$q = \frac{(A)(LMTD)}{\left[\frac{1}{A_i h_i} + \frac{R_{ffi}}{A_i} + \frac{\ln\left(\frac{d_o}{d_i}\right)}{2\pi k L} + \frac{R_{ffo}}{A_o} + \frac{1}{A_o h_o}\right]}$$

Where:

q = rate of heat transfer across the heat exchanger (Btu/hr)

A_i = internal surface area of the pipe (ft^2)

A_o = outside surface area of the pipe (ft^2)

LMTD = log mean temperature difference established by the current conditions (ºF)

h_i = convection coefficient on the inside of the heat exchanger (Btu/hr/ft^2/ºF)

R_{ffi} = thermal resistance of fouling factor on inside of the pipe (ºF•hr•ft^2/Btu)

d_o = outer diameter of the pipe (ft)

d_i = inner diameter of the pipe (ft)

L = length of the pipe (ft)

k = thermal conductivity of metal wall of pipe (Btu/hr/ft/ºF)

R_{ffo} = thermal resistance of fouling factor on cool side (ºF•hr•ft^2/Btu)

h_o = convection coefficient on the cool side of the heat exchanger (Btu/hr/ft^2/ºF)

πi = 3.141592654

ln( ) = natural logarithm function of quantity inside ( )

Much of the effort in using a formula such as this involves gathering information and making estimates for quantities such as the two convection coefficients (h_1 and h_2) and the two fouling factors (R_{ffi} and R_{ffo}).

An important point regarding the overall heat transfer coefficient is that it depends on the convection coefficients on <u>both sides</u> of the heat exchanger. *If one of these convection coefficients is small relative to the other, it will be the factor that limits the overall performance of the heat exchanger.*

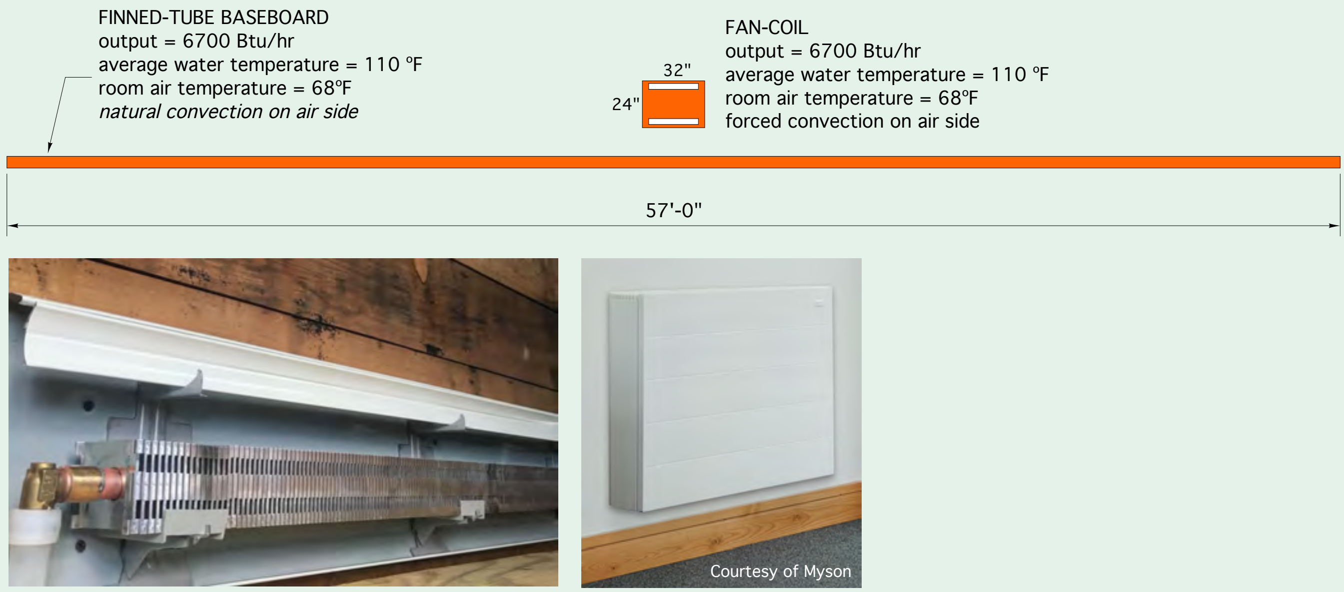
A good example of this is a coil heat exchanger inside a thermal storage tank. Based on some assumptions for typical operating conditions, the convection coefficient governing natural convection on the outer surface of the coil is only about 6% of the convection coefficient associated with forced flow inside the coil. This relatively low value will severely limit the overall heat transfer coefficient and thus require a substantially larger surface area (A) to create a given rate of heat transfer, relative to a situation where forced convection was present in both flow streams.

The difference between the relatively low convection coefficients associated with natural convection versus the higher values associated with forced convection can also be seen in a comparison between the length of finned-tube baseboard required for a certain heat output, versus the surface area of a coil in a wall-mounted fan-coil for equivalent heat output.

One commercially available fan-coil measuring 32 inches wide and 24 inches tall, when operating at 1.5 gpm, high-speed fan, and 110ºF average water temperature, can release approximately 6,700 Btu/hr. Standard residential finned-tube baseboard, operating at the same average water temperature and flow rate, can releases about 118 Btu/hr per foot of finned-tube length. To match the output of the fan-coil, the baseboard's finned-tube length would need to be 6700/118 = 57 feet. Figure 4-11 shows a *scaled* comparison of these two heat emitters.

Even though the forced flow occurring on the inside of the copper tube in the baseboard may have a high convection coefficient, heat transfer is severely limited by the natural convection occurring between the outer surface of the finned-tube element and the room air.

Figure 4-11



Courtesy of Nick Feriola @nickdaplumber Instagram

DETERMINING OVERALL RATE OF HEAT TRANSFER (EFFECTIVENESS METHOD)

The method of analyzing heat exchanger performance based on LMTD is useful when the entering and leaving temperatures of both flow streams are known, or they can be calculated based on an energy balance across the heat exchanger. However, when there is insufficient information to calculate the LMTD, the solution must resort to iterative calculations. These involve making “guesstimates” for the unknown temperatures, calculating heat transfer rates based on these temperatures, finding the “error” between the calculated rates of heat transfer on the two sides of the heat exchanger, correcting the guesstimated temperatures, and repeating the procedure. This is possible but very tedious. In these situations, another method of heat exchanger analysis is usually preferred. That method is based on the concept of “effectiveness” of the heat exchanger.

The effectiveness of a heat exchanger is defined by Formula 4-10.

Formula 4-10:

$$\varepsilon = \frac{q_{actual}}{q_{\max}}$$

Where:
ε = effectiveness of the heat exchanger (decimal number between 0 and 1)
q_{actual} = Actual rate of heat transfer across the heat exchanger (Btu/hr)
q_{max} = Maximum possible rate of heat transfer across the heat exchanger given the operating conditions (Btu/hr)

Effectiveness is based on the concept that, with a hypothetical heat exchanger having an *infinite* internal area, one of the two fluid streams could undergo the maximum possible temperature change, from the inlet temperature of the cold fluid up to the inlet temperature of the hot fluid. Which fluid could undergo this temperature change depends on which has the smaller capacitance rate. The latter term, capacitance rate, is the product of specific heat times mass flow rate, expressed in units of Btu/hr/ºF.

For example: Consider a situation where one side of a heat exchanger operates with water and a flow rate of 5 gpm. The other side operates with a 50% solution of propylene glycol and a flow rate of 6 gpm. The average temperature of the fluids is 120ºF.

The capacitance rate for the water side is:

$$C_{water} = \left(5\frac{gallon}{minute}\right)\left(1\frac{Btu}{lb\bullet^{o}F}\right)\left(\frac{8.33lb}{gallon}\right)\left(\frac{60minute}{hour}\right) = 2499\frac{Btu}{hr\cdot^{o}F}$$

The capacitance rate for the side operating with propylene glycol is:

$$C_{50\%PG} = \left(6\frac{gallon}{minute}\right)\left(0.88\frac{Btu}{lb\bullet^{o}F}\right)\left(\frac{8.54lb}{gallon}\right)\left(\frac{60minute}{hour}\right) = 2705\frac{Btu}{hr\cdot^{o}F}$$

Of these two capacitance rates, the one associated with the water side is smaller and will be designated as C_{min}. The larger capacitance rate will be designated C_{max}.

It's important to note that the physical properties of the two fluids were referenced to an *estimated* average temperature in the heat exchanger. Also notice that the units for capacitance rate are Btu/hr/ºF. One can interpret the value of capacitance rate as the rate of heat transfer (in Btu/hr) that each fluid stream could carry into or out of the heat exchanger per degree F change in temperature of that stream.

The fluid having the lower of the two capacitance rates is one of the factors that limit the maximum possible rate of heat transfer. That maximum possible rate for a hypothetical infinitely large heat exchanger would be the lower of the two capacitance rates (e.g., C_{min}), multiplied by the inlet temperature of the hot fluid minus the inlet temperature of the cold fluid. This allows the rate of heat transfer across the heat exchanger to be written as Formula 4-11.

Formula 4-11:

$$q = \varepsilon\left(C_{min}\right)\left(T_{hotin} - T_{coldin}\right)$$

Where:
q = actual rate of heat exchange across the heat exchanger (Btu/hr)
ε = effectiveness of the heat exchanger (unitless value between 0 and 1)
C_{min} = the smaller of the two fluid capacitance rates (Btu/hr/ºF)
T_{hotin} = inlet temperature of the hot fluid (ºF)
T_{coldin} = inlet temperature of the cold fluid (ºF)

Although Formula 4-11 appears relatively simple, there is additional work involved to determine the value of effectiveness (ε).
Two additional quantities need to be defined in order to determine the value of effectiveness (ε).

One is called the capacitance rate ratio. It's simply the minimum fluid capacitance rate of the two fluid streams divided by the larger fluid capacitance rate, stated as Formula 4-12:

Formula 4-12:

$$C_{ratio} = \frac{C_{min}}{C_{\max}}$$

Where:
C_{ratio} = capacitance rate ratio (unitless)
C_{min} = *smaller* of the two capacitance rates for the two flow streams involved (Btu/hr/ºF)
C_{max} = *larger* of the two capacitance rates for the two flow streams involved (Btu/hr/ºF)

The other quantity that's needed is called NTU, which stands for *number of transfer units*, and is defined as Formula 4-13.

Formula 4-13:

$$NTU = \frac{UA}{C_{min}}$$

Where:
NTU = number of transfer units (unitless number)
U = overall heat transfer coefficient for the heat exchanger (Btu/hr/ft^2/ºF)
A = internal area of the heat exchanger (ft^2)
C_{min} = *smaller* of the two capacitance rates of the two flow streams (Btu/hr/ºF)

One can think of the NTU as the ratio of the physical ability of a heat exchanger to transfer heat (based on its internal area and the overall heat transfer coefficient) divided by the ability of the "least able" flow stream to convey heat into or out of the heat exchanger. For example, a heat exchanger with a large internal area, and operating under favorable convection conditions, but also operating with a low capacitance rate on one side, would have high NTU. Because of its large internal area and high convection coefficients. it would be very "effective" in transferring heat to (or from) the fluid with the lower capacitance rate, possibly even approaching the performance of a hypothetical infinite heat exchanger, which would have an effectiveness of 1.0.

Once the capacitance rate ratio (C_{ratio}) and number of transfer units (NTU) are calculated, the effectiveness for a specific type of heat exchanger can be calculated based on either formulas or graphs in heat transfer reference books.

For example, for any counterflow heat exchanger, the effectiveness can be calculated using Formula 4-14a or 4-14b. Formula 4-14a is used in the typical case where $C_{ratio} < 1$. Formula 4-14b is used in the less typical but possible case where $C_{ratio} = 1$ (e.g., both fluid streams have the same capacitance rate ratio.

Formula 4-14a: (use when Cratio <1)

$$\varepsilon = \frac{1 - e^{(-NTU)(1-C_{ratio})}}{1-(C_{ratio})e^{(-NTU)(1-C_{ratio})}}$$

Formula 4-14b: (use when Cratio = 1)

$$\varepsilon = \frac{NTU}{1+NTU}$$

The terms shown in red in Formula 4-14a are identical and thus only have to be calculated once.

The effectiveness of a counterflow heat exchanger as a function of the number of transfer units (NTU) and the capacity rate ratio (C_{min}/C_{max}) can also be found using Figure 4-12.

Here's an example: Consider a counterflow heat exchanger with an internal area of 20 ft^2. It is operating with the two previously described flow streams of water at 5 gpm and 50% propylene glycol at 6 gpm. The water stream enters at 150ºF, and the propylene glycol solution enters at 60ºF. An analysis of the convection coefficients and fouling factors on both sides of the heat exchanger has been used to establish the overall heat transfer coefficient (U) as 150 Btu/hr/ft^2/ºF. Determine the rate of heat transfer across the heat exchanger.

The two fluid capacitance rates have already been determined as:

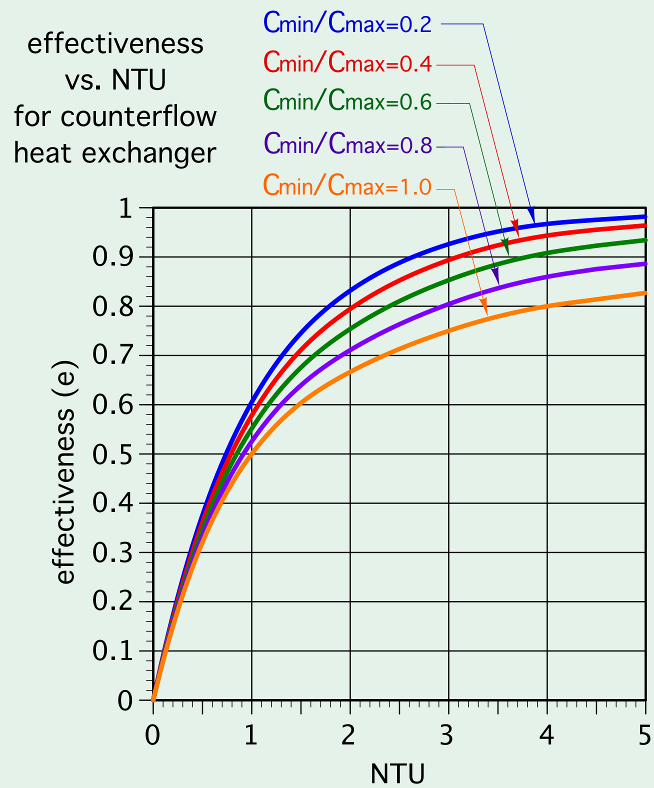
$$C_{water} = \left(5\frac{gallon}{minute}\right)\left(1\frac{Btu}{lb\cdot^{o}F}\right)\left(\frac{8.33lb}{gallon}\right)\left(\frac{60minute}{hour}\right) = 2499\frac{Btu}{hr\cdot^{o}F}$$

$$C_{50\%PG} = \left(6\frac{gallon}{minute}\right)\left(0.88\frac{Btu}{lb\cdot^{o}F}\right)\left(\frac{8.54lb}{gallon}\right)\left(\frac{60minute}{hour}\right) = 2705\frac{Btu}{hr\cdot^{o}F}$$

Of these two capacitance rates, the one for the water side is smaller. Thus, the capacitance rate ratio is:

$$C_{ratio} = \frac{C_{min}}{C_{\max}} = \frac{2499}{2705} = 0.924$$

Figure 4-12



The number of transfer units is:

$$NTU = \frac{UA}{C_{min}} = \frac{(150)20}{2499} = 1.2$$

These values can now be entered into Formula 4-14a:

$$\varepsilon = \frac{1 - e^{(-NTU)(1-C_{ratio})}}{1-(C_{ratio})e^{(-NTU)(1-C_{ratio})}} = \frac{1 - e^{(-1.2)(1-0.924)}}{1-(0.924)e^{(-1.2)(1-0.924)}} = \frac{0.0872}{0.1565} = 0.557$$

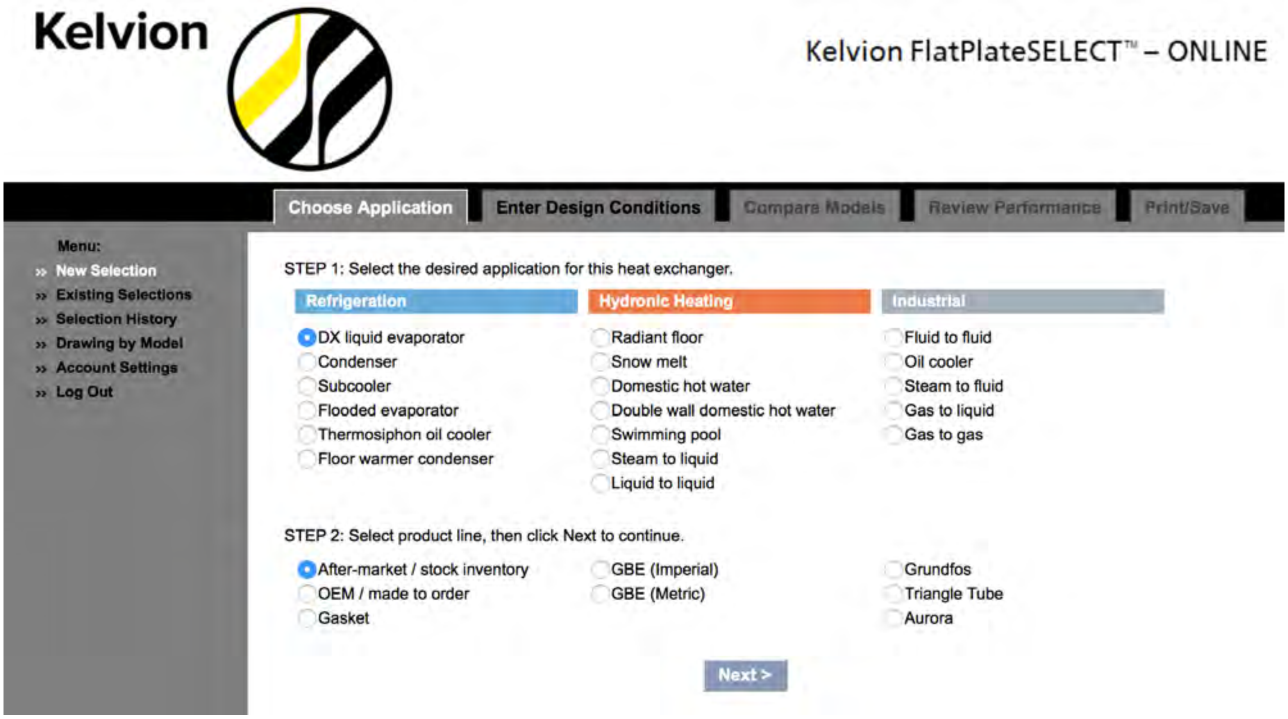
The final step is to apply Formula 4-11:

$$q = \varepsilon(C_{min})(T_{hotin} - T_{coldin}) = 0.557(2499)(150-60) = 125,280\frac{Btu}{hr}$$

SOFTWARE-BASED HEAT EXCHANGER SELECTION

The procedures introduced thus far — the LMTD method, and the effective (ε) method —can be used for manual calculations of heat exchanger performance. Both of these approaches demonstrate that the mathematics necessary for sizing heat exchangers can be complex. One of the more complex tasks —that was not documented in

Figure 4-13a



Source: Kelvion, www.flatplateselect.com

Figure 4-13b

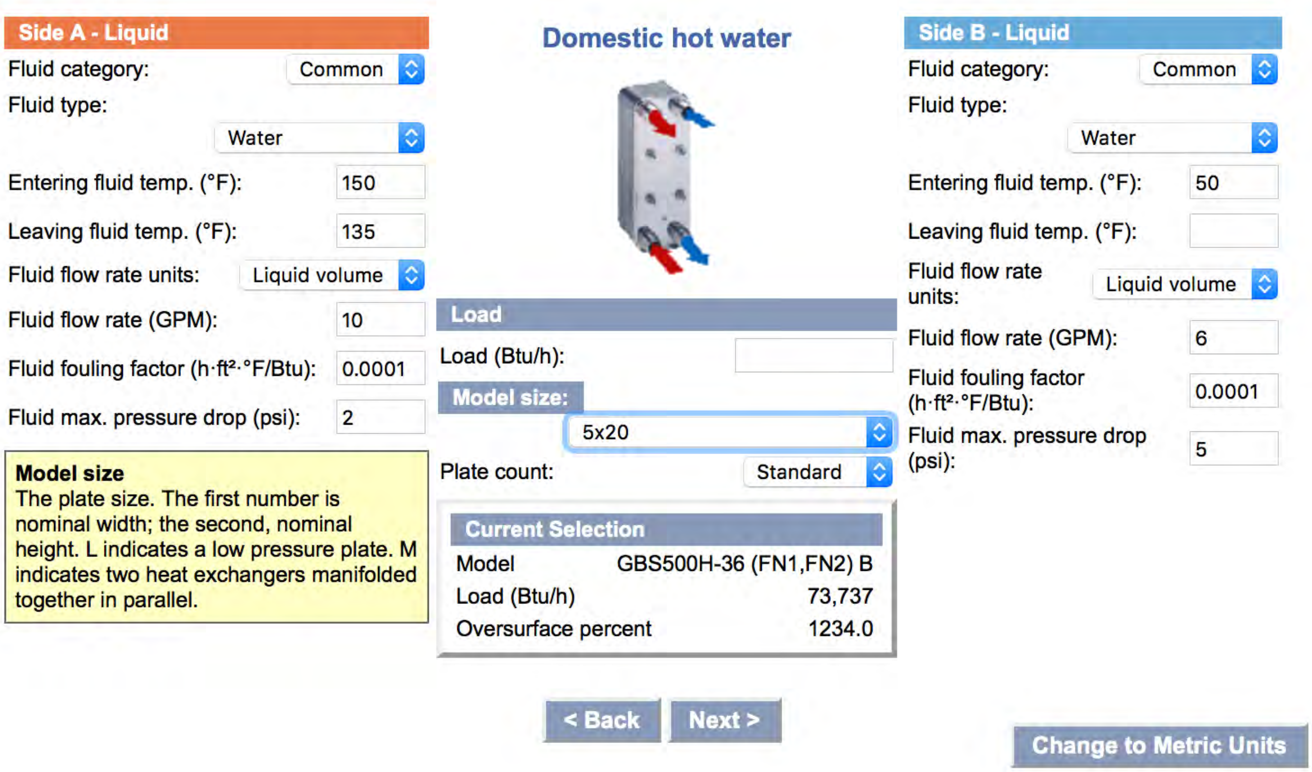


Figure 4-13c

Note: To download a PDF file version of this report, navigate to the Print/Save page.

Model: FP5X12L-12 (1" MPT)			
Selection ID	NUP2H5A7Z	Model size	5x12L
Application	Domestic hot water	Nominal surface (ft²)	3.8
Load (Btu/h)	73,737	Dimensions	4.9W x 12.2H x 1.4D
Log mean temp. diff. (°F)	80.1	Plate construction	Single wall
Overall HTC (Btu/h·ft²·°F)	687	Net weight (lb)	6.3
Oversurface percent	186.3		

Design Conditions	Side A - Liquid	Side B - Liquid
Fluid type	Water	Water
Fluid mass flow rate (lb/min)	82	50
Entering fluid temp. (°F)	150.0	50.0
Leaving fluid temp. (°F)	135.0	74.6
Fluid flow rate (GPM)	10.0	6.0
Fluid fouling factor (h·ft²·°F/Btu)	0.00010	0.00010
Model Parameters		
Number of channels	5	6
Velocity (ft/s)	1.90	0.95
Pressure drop (psi)	2.0	0.6
Heat transfer coef. (Btu/h·ft²·°F)	2,331	1,211
Internal volume (ft³)	0.015	0.018

Ratings at Varying Conditions					
Percent difference	**-15%**	**-7½%**	**0%**	**7½%**	**15%**
Pressure drop (psi) (Side A)	2.0	2.0	2.0	2.0	2.0
Pressure drop (psi) (Side B)	0.5	0.5	0.6	0.7	0.8
Load (Btu/h)	73,737	73,737	73,737	73,737	73,737
Fluid flow rate (GPM) (Side A)	10.0	10.0	10.0	10.0	10.0
Fluid mass flow rate (lb/min) (Side A)	82	82	82	82	82
Fluid flow rate (GPM) (Side B)	**5.1**	**5.6**	**6.0**	**6.4**	**6.9**
Fluid mass flow rate (lb/min) (Side B)	43	46	50	54	58
Entering fluid temp. (°F) (Side A)	150.0	150.0	150.0	150.0	150.0
Entering fluid temp. (°F) (Side B)	50.0	50.0	50.0	50.0	50.0
Leaving fluid temp. (°F) (Side A)	135.0	135.0	135.0	135.0	135.0
Leaving fluid temp. (°F) (Side B)	78.9	76.6	74.6	72.9	71.4
Oversurface percent	162.8	175.0	186.3	196.7	206.5

Disclaimer

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the previous examples — is estimating the values for the convection coefficients (h_h and h_c) in Formulas 4-7 and 4-9. Those calculations are often iterative and time consuming. As such, they don't lend themselves to repeated "what if" scenarios when a designer is trying to select an appropriate heat exchanger.

Many manufacturers now offer software that can rapidly evaluate the necessary mathematics for sizing and selection of their heat exchangers. In some cases, this software is accessed online. In other cases, it's necessary to download and install the software. Some suppliers also require submittal of design parameters and operate the software in-house.

One example of online-based sizing and selection software is FlatPlateSELECT, available at **http://flatplateselect.com/site/hx/chooseapp.aspx**. Figures 4-13a, b and c show some examples of this software.

Some heat exchanger sizing and selection software produces a list of several specific models that can meet the requirements based on input data. They also show an "oversurface" percent, which is the percent by which the surface area in the suggested heat exchanger exceeds the minimum surface area required for the specified heat exchange rate. In some cases, the extra surface area is necessary to limit the pressure drop of one or both sides of the heat exchanger. The software provides performance data such as the overall heat transfer coefficient, flow velocity and rate of heat transfer. Physical data on the candidate heat exchangers, including volume and dimensions, are also listed. The ability of this software to evaluate several "what if" scenarios in a fraction of the time required for manual calculations makes it an essential design tool.

HEAT EXCHANGER PERFORMANCE INDICES

Two indices are commonly used to describe the actual performance, or performance target, of heat exchangers. They are:

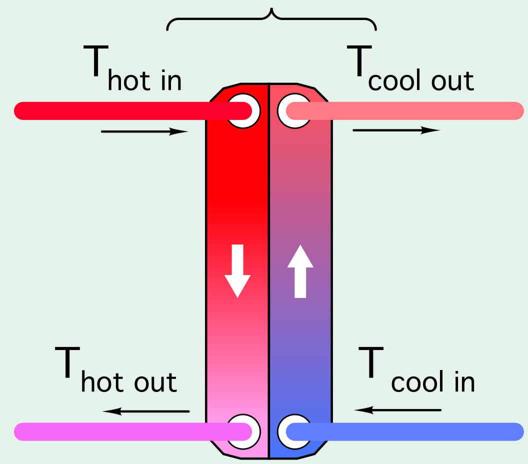
- Approach temperature difference
- Thermal length

Approach temperature difference is the temperature of the fluid entering the hot side of the heat exchanger, minus the temperature of the "cooler" fluid leaving the heat exchanger. Figure 4-14 illustrates this difference.

A theoretically infinite heat exchanger would have an approach temperature difference of 0. Any real heat exchanger will have an approach temperature difference greater than zero. Some brazed plate and plate & frame heat exchangers can operate at approach temperature differences as low as approximately 2ºF, although this

Figure 4-14

$$\text{Approach temperature difference} = \left(T_{\text{hot in}} - T_{\text{cool out}}\right)$$



requires a relatively large heat transfer area, and thus a large heat exchanger. The expense of such a large heat exchanger may not be justified based solely on a very small approach temperature difference. More typical approach temperature differences, with the heat exchanger operating at design load conditions, range from 4 to 10ºF.

Approach temperature differences are often selected based on the characteristics of the heat source. For example, the coefficient of performance (COP) of water-to-water and air-to-water heat pumps is very dependent on the load water temperature (e.g., the water temperature leaving the heat pump's compressor). If the heat produced by a heat pump needs to pass through a heat exchanger, the approach temperature difference should be as low as practically and economically justifiable. A suggested maximum approach temperature difference for heat pump applications is 5ºF.

In another scenario, where a boiler operating at 180ºF is transferring heat to a load operating at 110ºF, the approach temperature difference is able to be much higher without significantly affecting the boiler's efficiency. This allows use of a much smaller and less expensive heat exchanger.

Keep in mind that approach temperature difference is based on a specific steady-state load. The measured approach temperature difference of a heat exchanger can vary widely when the system is operating in transient conditions.

THERMAL LENGTH OF A HEAT EXCHANGER

Another concept that helps describe the expected duty of a heat exchanger is called *thermal length*. It's defined by Formula 4-15.

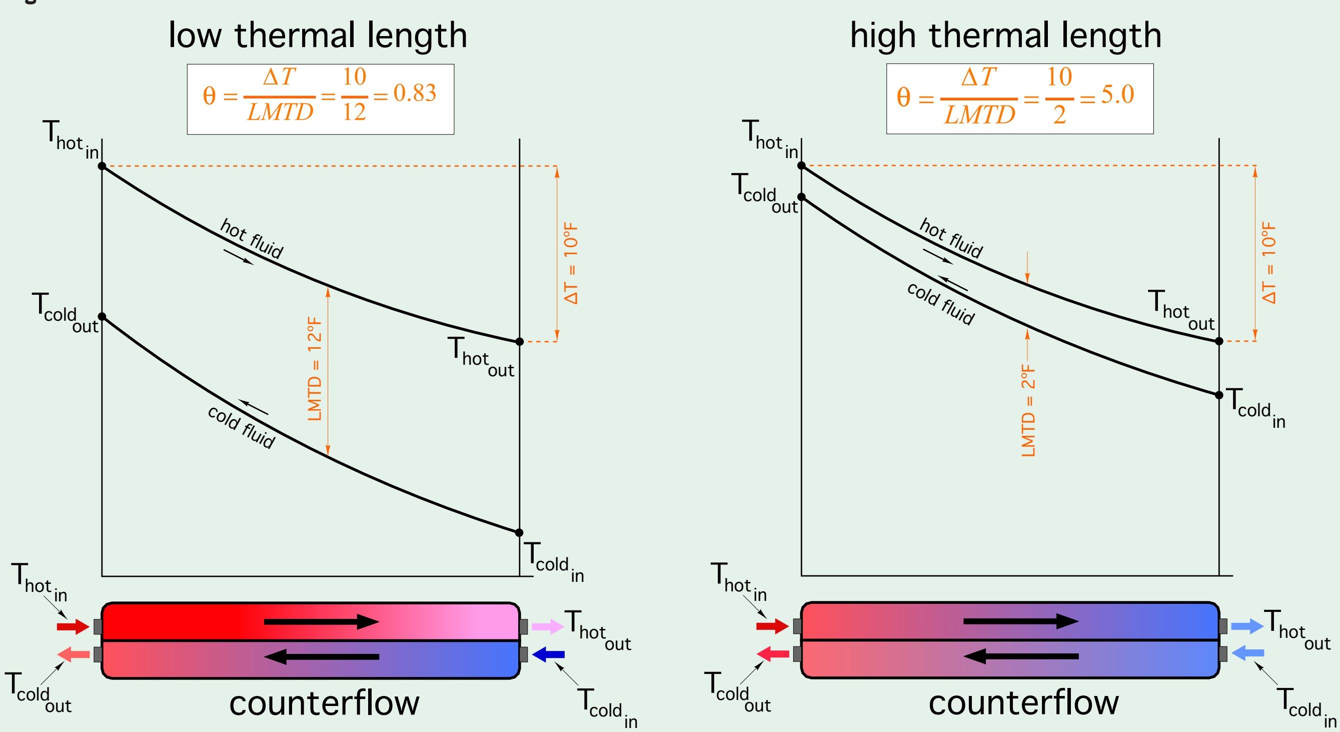
Formula 4-15:

$$\theta = \frac{\Delta T}{LMTD}$$

Where:
θ = (Greek letter theta) = thermal length (unitless)
ΔT = temperature change of fluid (inlet to outlet) on one side of the heat exchanger (ºF)
LMTD = log mean temperature drop at which the heat exchanger is operating (ºF)

Low values of (θ) indicate heat transfer conditions that are relatively easy to accomplish. High values of (θ) represent situations where there is very little difference between the temperature of the fluid providing heat and the fluid

Figure 4-15



absorbing heat. The latter is a much more challenging condition and generally requires flat plate heat exchangers with longer plates. Figure 4-15 illustrates the difference between heat transfer requirements having low and high thermal lengths (θ).

Heat transfer theory limits the thermal length (θ) of a single shell & tube heat exchanger to 1.0. Higher values of (θ) are possible if multiple shell & tube heat exchangers are piped in series, which is uncommon. Flat plate heat exchangers can operate with thermal lengths up to approximately 10.0. This makes flat plate heat exchangers better suited to situations where the temperature differences between the fluids is minimal. The thermal length of flat plate heat exchangers can also be affected by different pressing patterns on the plate. Manufacturers can advise on optimal plate patterns based on the intended duty of the heat exchanger.

THERMAL PERFORMANCE OF WATER-TO-AIR HEAT EXCHANGERS

There are several types of heat emitters that transfer heat from a stream of water to a fan-forced stream of air. Section 2 described several of these devices, categorizing them as fan-coils or air handlers. This section will describe thermal performance concepts for these devices.

The heat output of a fan-coil or air handler is dependent on several factors. Some are fixed by the design of the product and cannot be changed. These include the:

- Surface area of the coil
- Size, spacing and thickness of the fins
- Number of tube passes through the fins
- Air-moving ability of the blower

Other performance factors depend on the system into which the fan-coil is installed and the conditions under which that system operates. These include the:

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

Most manufactures publish tables or graphs showing the heat output of a fan-coil or air handler over a range of selected operating conditions, such as entering water temperature, entering room air temperature and the flow rates of both the water and air streams. However, because heat output varies continuously with changes in temperatures or flow rates, there is no guarantee that the unit will operate at one of the conditions listed by the manufacturer.

Figure 4-16

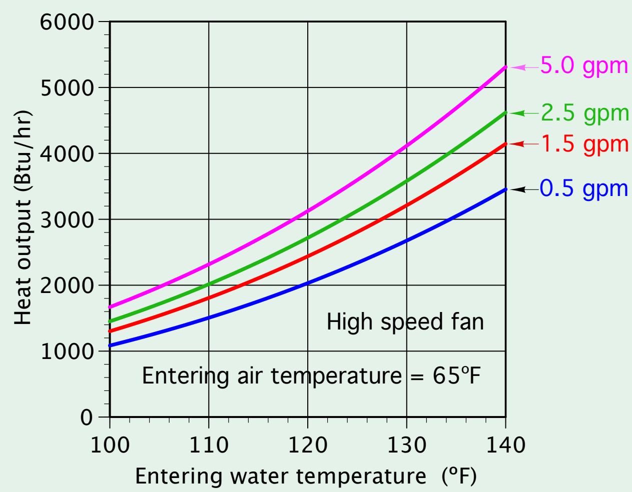


Figure 4-16 shows representative heat output from a small fan-coil unit as a function of entering water temperature and different water flow rates. The entering room air temperature is assumed constant at 65ºF, and the fan is set to its highest speed.

The manner in which the heat output of a fan-coil or air handler varies with operating conditions or coil configuration can be summarized in four principles:

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

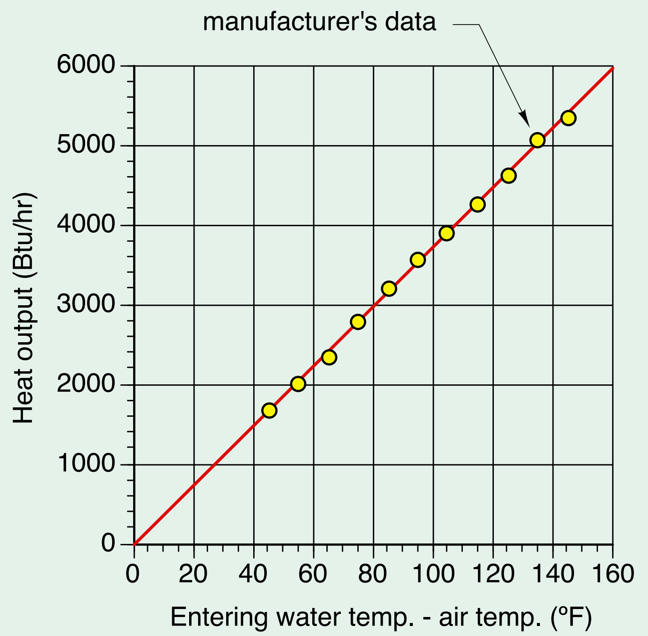
This principle can be expressed mathematically as:

Formula 4-16:

$$q_2 = q_1\left[\frac{\Delta T_2}{\Delta T_1}\right]$$

Where:
q_2 = heat output at an assumed water-to-air temperature difference (Btu/hr)
q_1 = heat output at a *known* water-to-air temperature difference (Btu/hr)
ΔT_1 = difference between entering water and entering air temperatures at a known operating condition (ºF)
ΔT_2 = difference between entering water and entering air temperatures at the assumed operating condition (ºF)

Figure 4-17



For example, assume a fan-coil has a known output of 5,000 Btu/hr when supplied with 180ºF water and operating with room air entering at 65ºF. The designer wants to estimate the fan-coil's output when supplied with 140ºF water, with all other conditions (e.g., room air temperature, water flow rate and air flow rate) remaining the same. Formula 4-16 yields the estimated output:

$$q_2 = q_1\left[\frac{\Delta T_2}{\Delta T_1}\right] = 5{,}000\left[\frac{140-65}{180-65}\right] = 3{,}261\frac{Btu}{hr}$$

A proportional relationship between any two quantities will result in a straight line that passes through or very close to the origin when the data is plotted on a graph. Figure 4-17 shows this to be true when the heat output data for a small fan-coil is plotted against the difference between the entering water and entering air temperatures.

This relationship can also be used to estimate the heat output from a fan-coil or air handler when the entering *air* temperature is above or below normal comfort temperatures. For example, if a fan-coil can deliver 3,261 Btu/hr with an entering water temperature of 140°F and entering air temperature of 65°F, its output while heating a garage maintained at 50°F could be estimated using the same formula:

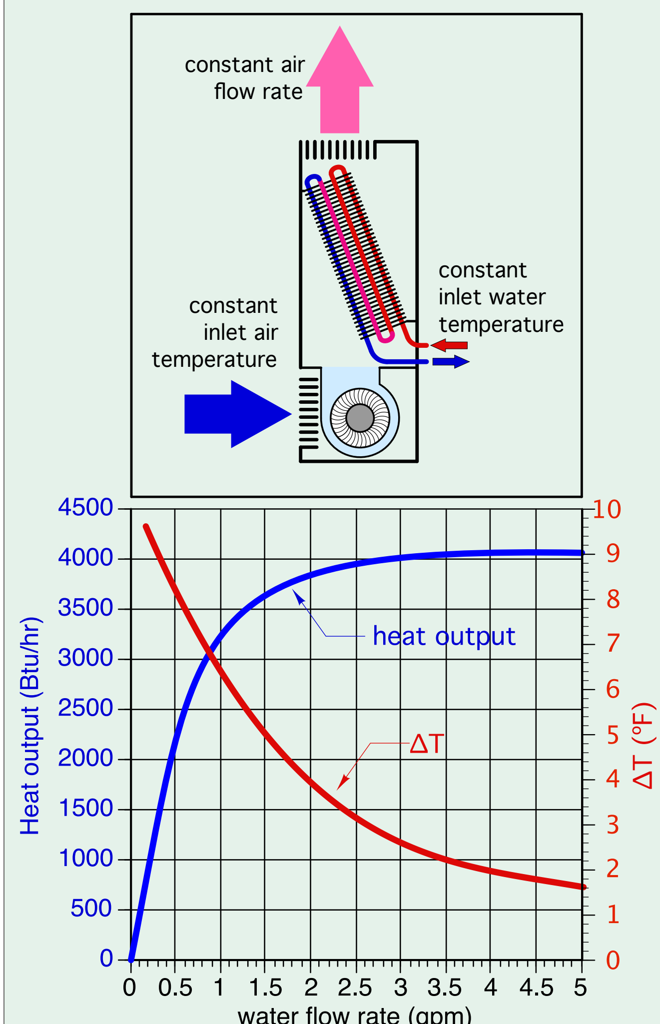
$$q_2 = q_1\left[\frac{\Delta T_2}{\Delta T_1}\right] = 3{,}261\left[\frac{140-50}{140-65}\right] = 3{,}913\frac{Btu}{hr}$$

Although principle #1 is helpful for quick performance estimates, *it should not replace the use of thermal rating data supplied by the manufacturer when it is available.*

Principle #2: Increasing the fluid flow rate through the coil will *marginally* increase the heating capacity of a fan-coil or air handler.

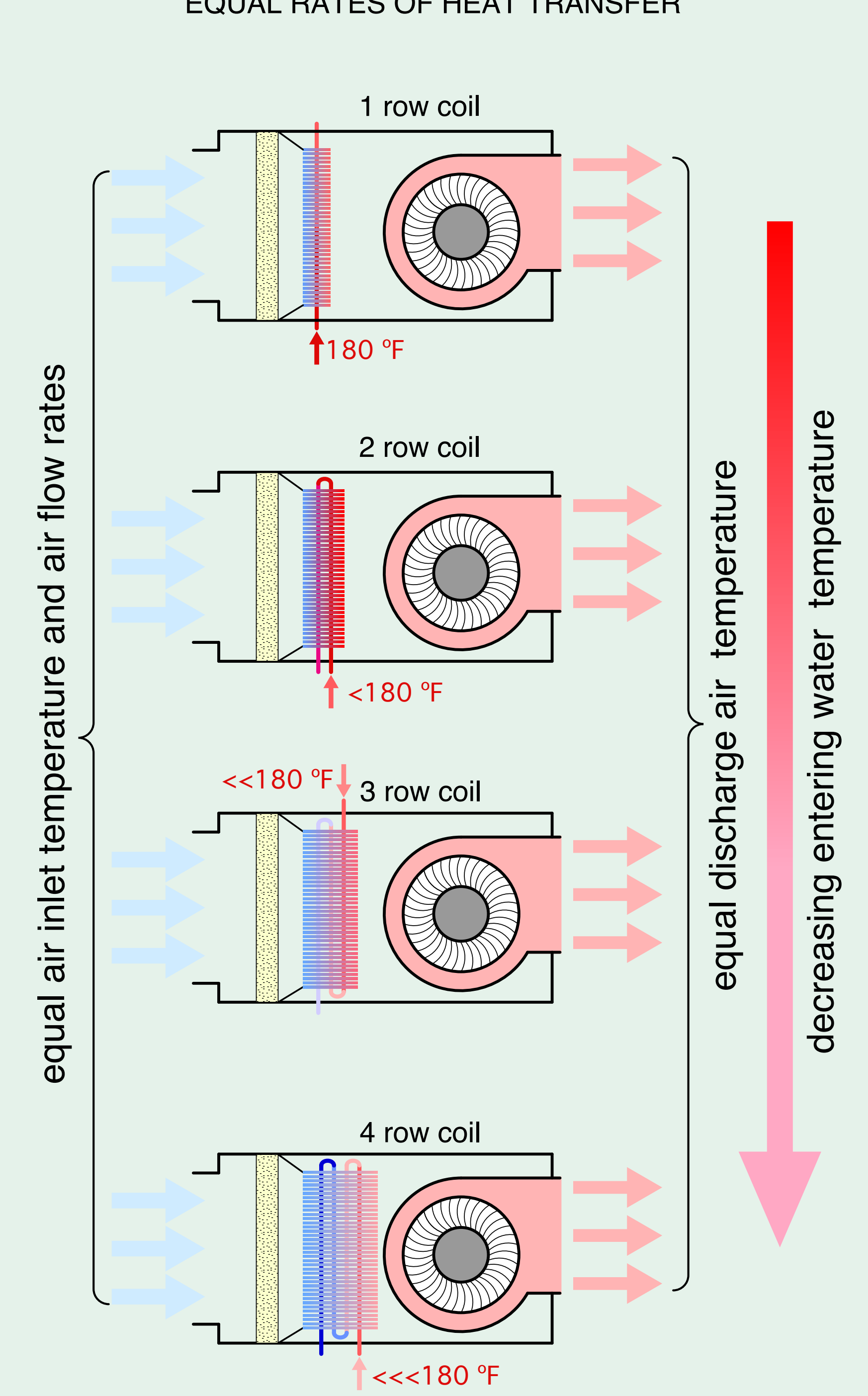
Some heating professionals "instinctively" disagree with this statement. They reason that because the water moves through the coil at a faster speed, it has less time during

Figure 4-18



which to release its heat. However, the time a given amount of water stays inside the fan-coil is irrelevant in a system with continuous flow. As previously discussed, the faster a fluid moves along a surface, the higher the rate of convective heat transfer. Higher rates of convective heat transfer between the water and inside surface of the tubes making up the coil result in higher heat outputs.

Figure 4-19



Another way of justifying this principle is to consider the *average* water temperature in the coil at different flow rates. As the flow rate through the coil heat exchanger increases, the temperature drop from inlet to outlet decreases. This implies that the *average* water temperature within the coil increases, and so does its heat output. Those who claim that a smaller temperature difference between the ingoing and outgoing fluid implies less heat is being released are overlooking the fact that the rate of heat transfer depends on *both* the temperature difference *and* flow rate. This relationship is described by Formula 4-8 discussed earlier in this section. This non-linear relationship between flow rate and heat output also applies to other heat emitters such as radiant panel circuits, panel radiators and finned-tube baseboard. It can be seen whenever heat output ratings are plotted against flow rate through the heat emitter. Figure 4-18 shows this relationship for a small fan-coil operated at a constant inlet water temperature, constant air inlet temperature and constant air-side flow rate.

The rate of heat output increases very quickly at low flow rates, but less quickly as flow through the coil increases. The fan-coil represented in Figure 4-18 released about 50 percent of its maximum heat output capacity at about 10 percent of its maximum flow rate. The strong curvature of this relationship implies that attempting to control the heat output of a fan-coil or air handler by adjusting flow rate can be tricky. Very small valve adjustments can create large changes in heat output at low flow rates. However, the same amount of valve adjustment will create almost no change in heat output at higher flow rates. This is also true for other types of hydronic heat emitters.

Special types of balancing valves known as "equal percentage" valves have been developed specifically to compensate for this characteristic. An equal percentage valve allows very small increases in flow rate as the valve begins to open from a closed position. The farther the valve opens. the faster the rate at which flow-through increases. The combination of the heat output versus flow rate characteristic of the coil heat exchanger, and the flow versus stem position characteristic of an equal percentage valve, attempts to make the change in heat output approximately proportional to the percentage of stem travel in the valve.

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More information on equal percentage valves can be found in idronics 8 (Hydronic Balancing).

The nonlinear relationship also implies that attempting to boost heat output from a fan-coil or air handler, and other hydronic heat emitters, by operating them at unusually high flow rates will yield very minor gains. The argument against doing this is further supported by large increases in pumping power with increasing flow rate.

Principle #3: Increasing the air flow rate across the coil heat exchanger in a fan-coil or air handler will *marginally* increase the heat output of the fan-coil.

This principle is also based on forced-convection heat transfer between the exterior coil surfaces and the air stream. Faster-moving air reduces boundary layer thickness and "scrubs" heat off the coil surface at a higher rate. As with water flow rates, the gain in heat output is very minor above the nominal air flow rate the unit is designed for.

Principle #4: Fan-coils with large coil surfaces and/or multiple tube passes through the fins can yield a given rate of heat output while operating at lower entering water temperatures.

Conductive heat transfer from the tubing carrying the fluid to the aluminum fins of the coil heat exchanger increases when more tubes are used to make the coil. Convective heat transfer is proportional to the contact area between the surface and the fluid. Both of these effects allow larger liquid-to-air heat exchangers to transfer a given rate of heat at lower average water temperatures. This principle is illustrated in Figure 4-19.

This principle is very important for hydronic systems that operate with low temperature heat sources such as heat pumps. In these applications, the designer may need to use fan-coils with larger coils and/or more tube passes through the fins of the coil to drive heat from the coil at the required rate. On larger air handlers, manufacturers may offer higher-performance coils with up to eight tube rows.

Care must be taken that comfort is not compromised when operating fan-coils or air handlers at reduced water temperatures. It is critically important to introduce the air stream from such units into the space so that it mixes with room air before passing by occupants. Failure to do so will usually lead to complaints of "cool air" blowing from the fan-coils or registers, even though the room is being maintained at the desired temperature.

5. INSTALLATION DETAILS FOR HEAT EXCHANGERS

This Section discusses specific details that allow heat exchangers to perform as expected in several types of applications.

HEAT EXCHANGER FOULING

To maintain peak performance, it's important to minimize the potential for fouling films to form on any heat transfer surfaces. Fouling can be the result of unintentional chemical interactions within the system. It can also occur due to dirt or other debris unintentionally present in the system.

CHEMICAL FOULING

Fouling films can form on heat exchanger surfaces due to fluid chemistry issues. For example, the solubility of minerals such as calcium and magnesium in water decreases as water temperature increases. If water containing these minerals passing along the internal surfaces of a heat exchanger becomes hot enough to reach the maximum solubility limit of the dissolved minerals, they will precipitate out of solution and form scale on those surfaces. The higher the surface temperatures, the greater the potential for scale formation. Figure 5-1 shows an example of the heat exchanger from a modulating/condensing boiler that has been severely scaled from minerals in the system water.

The best approach for reducing scaling of heat exchanger surfaces is to demineralize the system water. Demineralization is done by passing water through a column containing thousands of small porous polymer beads that are chemically formulated to capture positive and negative ions in the system water. These undesirable ions include positively charged *cations* of calcium (Ca^{++}), magnesia (Mg^{++}) and sodium (Na^{+}). They also include negatively charged anions of chlorine (Cl^{-}) and carbonate ($CO^{- -}$). When the resin beads capture these cations and anions, they release hydrogen cations (H+) and hydroxide anions (OH-). The released ions instantly bond to form pure H_2O.

Water in hydronic systems should be deionized to a condition where its total dissolved solids (TDS) content is between 10 and 30 parts per million (PPM). The small amount of remaining ions gives the water sufficient electrical conductivity to allow proper operation of low-water cut-off devices. This level of TDS also helps stabilize the water to prevent it from scavenging ions from metal surfaces.

idronics 18 provides more details on demineralizing water in hydronic systems

Figure 5-1

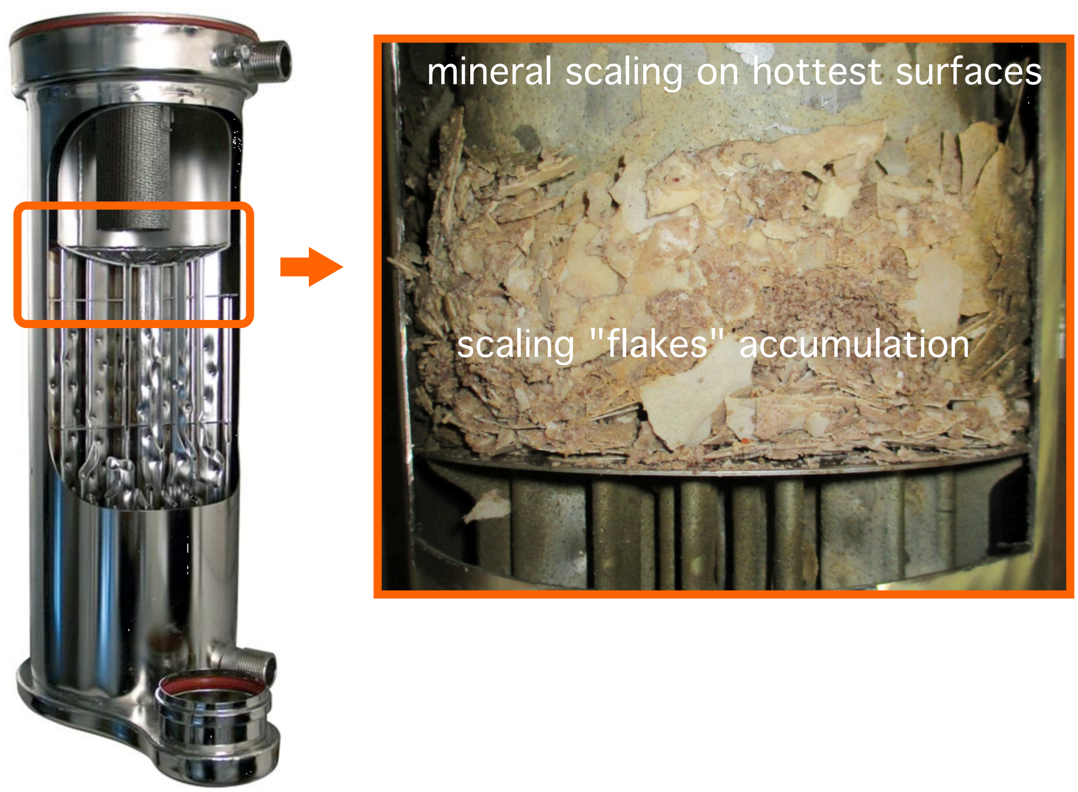


Figure 5-2

DEBRIS FOULING

Most hydronic systems are assembled from components that have been stored, transported and handled multiple times between manufacturing and installation. During these times, debris such as road dust, insects, pollens, drywall dust, solder balls, metal chips and sawdust can enter these components. These materials can accumulate on the internal surfaces of heat exchangers and significantly reduce heat transfer rates. These accumulations can also increase the pressure drop through components, resulting in lower flow rates, which further reduce heat transfer. Figure 5-2 shows a heavily fouled plate from a plate & frame heat exchanger.

Although some of the heat exchangers discussed earlier in this issue can be disassembled for cleaning, that process is difficult, time consuming and expensive. The heat exchanger must be isolated, disconnected from piping, disassembled, cleaned, reassembled and put back in service. In large systems, this could take several days and will prevent heating or cooling operation in any portion of the system involving the heat exchanger during that time.

Severely fouled brazed plate heat exchangers, or sealed shell & coil heat exchangers, may need to be replaced. Again, a costly and time-consuming process.

The best way to avoid these situations is to minimize the possibility of debris entering heat exchangers. This can be done by installing high-performance separators that capture dirt and magnetic particles upstream of the heat exchanger.

Some designers, by default, specify wye strainers as a means of capturing dirt particles in the system. While this approach is partially effective, it does have limitations. One is that the pressure drop through a wye strainer increases significantly as debris collects on the internal strainer basket, as shown in Figure 5-3. The increased pressure drop decreases flow in the circuit, which negatively impacts the heat transfer performance of any heat exchanger.

Another limitation of wye strainers is that they must be isolated and disassembled for cleaning. This requires temporary shutdown of the circuit and some minor drainage of system fluid.

Finally, wye strainers do not capture magnetic particles such as magnetite that form when dissolved oxygen in the water fluid reacts with steel or cast iron components in the system. Magnetite will be attracted to magnetic fields generated by circulator motors, especially the strong fields generated by permanent magnets in wet rotor circulators with electronically commutated motors.

Figure 5-3

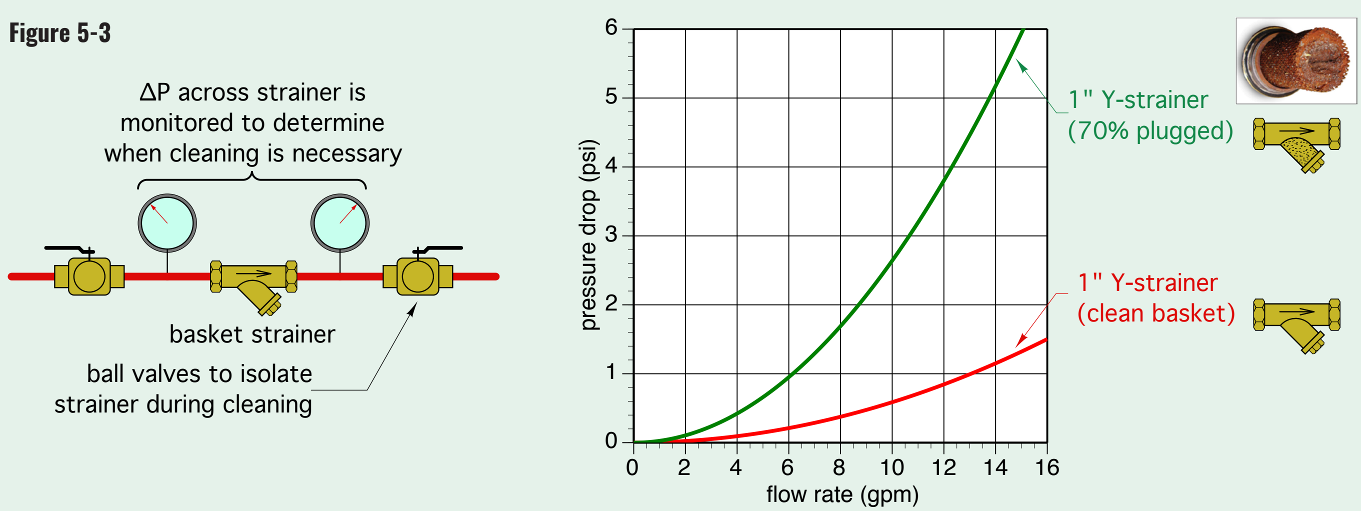


Figure 5-4a

Figure 5-4b

The modern alternatives to wye strainers are low-velocity zone separators equipped with removable magnets. Figure 5-4 shows examples of Caleffi *DIRTMAG*™ separators for both small and larger system applications.

Low-velocity zone dirt separators do not allow the dirt they capture to accumulate in the flow path, and thus create far less pressure drop than wye strainers. They can also be flushed without shutting down the circuit.

Figure 5-5

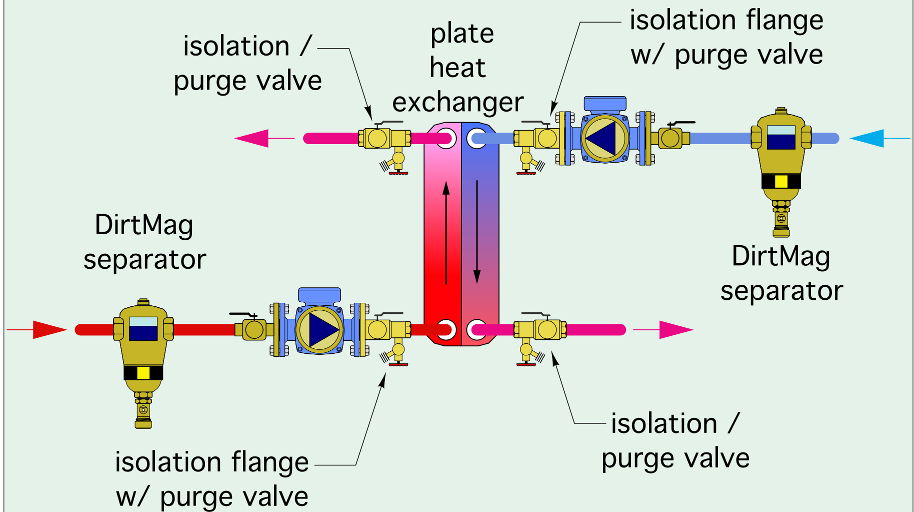


Figure 5-6

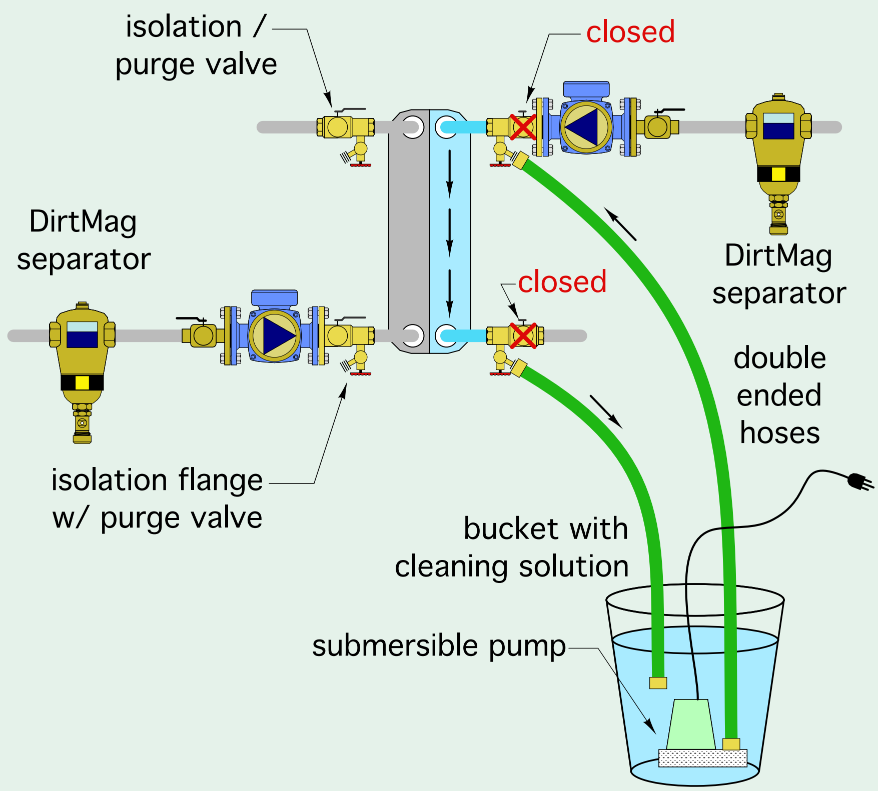


Figure 5-7a

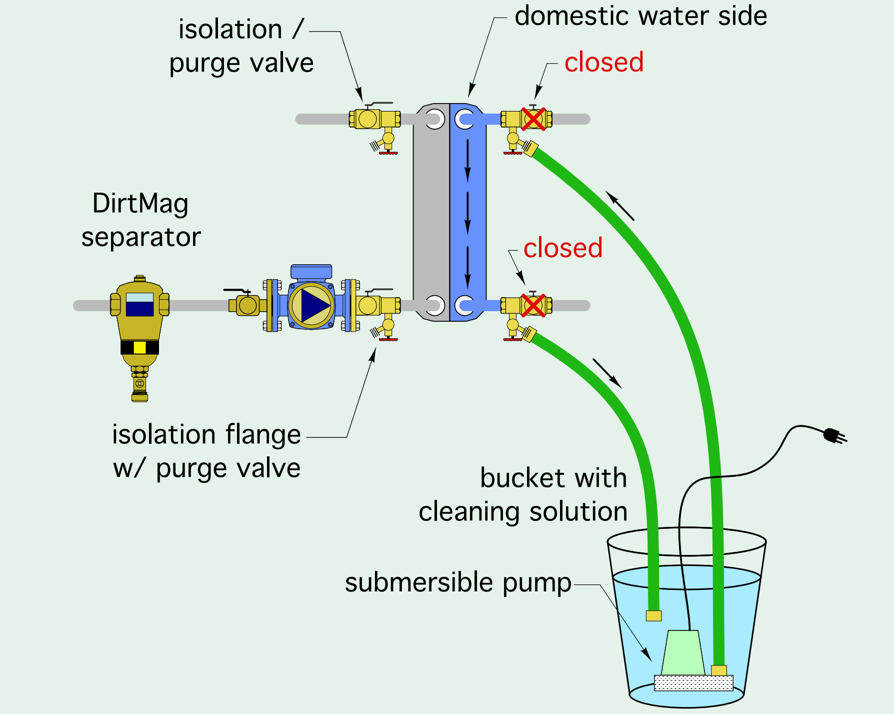


Figure 5-7b

Some dirt separators are available with strong permanent magnets that can capture very fine iron oxide particles formed from ferrous metals in the system.

These *DIRTMAG* separators should be installed upstream of heat exchangers, as shown in Figure 5-5.

The component arrangement shown in Figure 5-5 has the *DirtMag* separators upstream of both the heat exchanger and the circulators supplying it. A combination isolation flange/purge valve is shown on the outlet of each circulator. These flanges allow the circulator to be isolated and removed if necessary. In combination with isolation/purging valves on the outlet ports of the heat exchanger, they also allow each side of the heat exchanger to be isolated and flushed with a cleaning solution if necessary. Figure 5-6 shows how the heat exchanger could be flushed by circulating a cleaning fluid using a small submersible pump.

Figure 5-7 shows a slight variation on valving, which can be used when one side of the heat exchanger operates with domestic water.

A combination isolation/purging valve, rated for use with potable water, is installed at each port on the domestic water side of the heat exchanger. The inline balls of these valves are closed to isolate the heat exchanger from the other domestic water piping. The cleaning solution is circulated through the domestic water side of the heat exchanger to dissolve and remove mineral scale. The two hoses can also be reversed to change the direction of the cleaning solution through the exchanger. After the cleaning solution has been sufficiently circulated and drained, the heat exchanger should be flushed with domestic water to remove any residual cleaning solution.

HEAT EXCHANGERS SUPPLYING ANTIFREEZE-PROTECTED CIRCUITS

When a heat exchanger is installed as the “heat source” for a portion of the overall system, that portion is fully isolated from the remainder of the system. As such, it must include components used in all closed-loop hydronic systems. These include a pressure-relief valve, expansion tank, pressure gauge, provisions for filling and purging, and one or more separators to remove air, dirt and magnetic particles. Optional components include a flow meter and thermometers or temperature sensor wells to monitor the inlet and outlet temperatures of the heat exchanger. In some applications, specifically district heating systems, heat meters are also installed on the closed circuit served through the heat exchanger.

Figure 5-8 shows a configuration of these components for a situation in which the heat exchanger supplies a garage floor heating system that operates with an antifreeze solution. The heat source is assumed to be a convention boiler.

Whenever a conventional boiler is supplying a low temperature heat emitter system, that boiler should

Figure 5-8

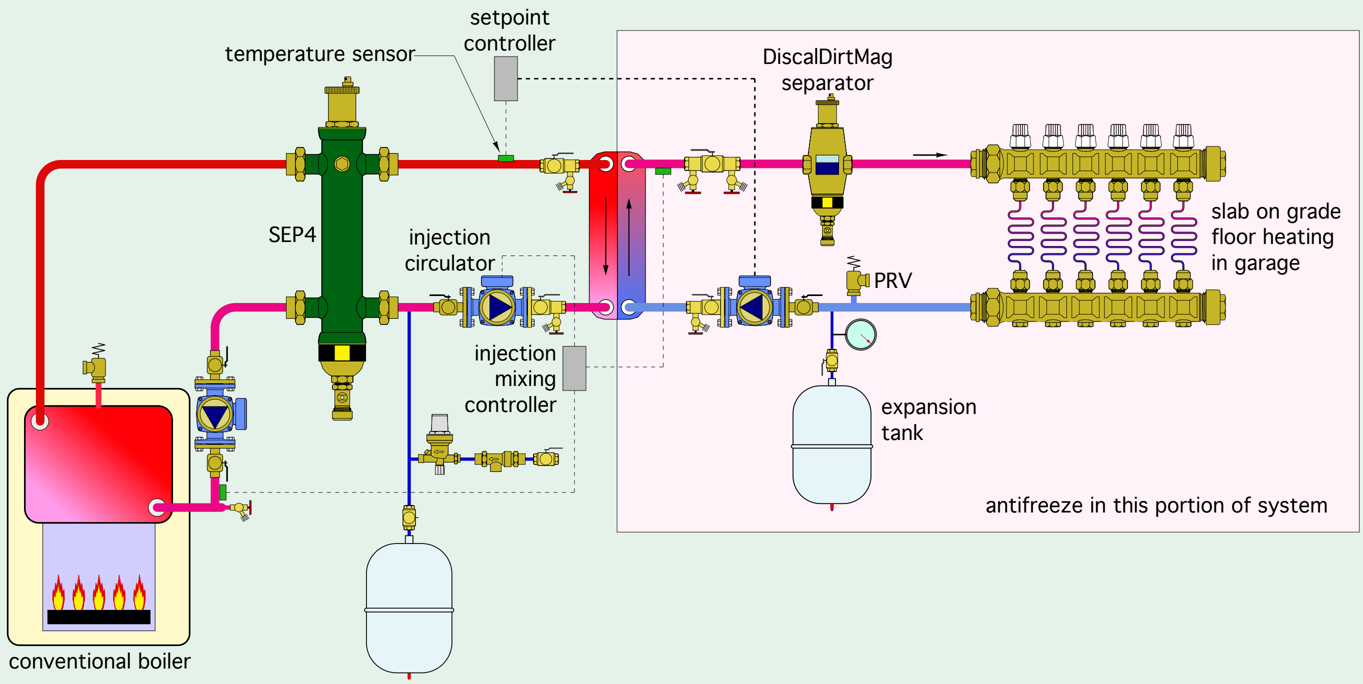
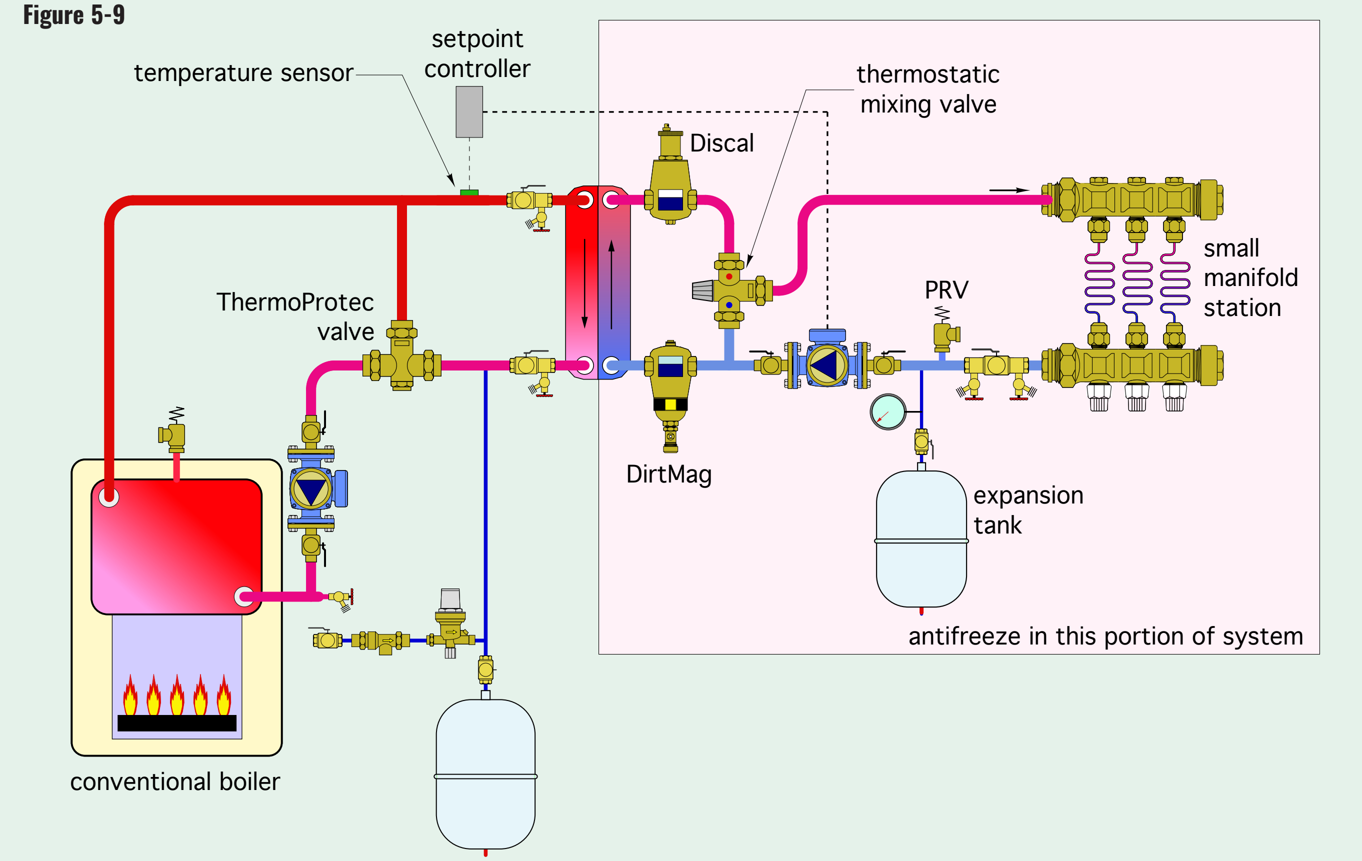
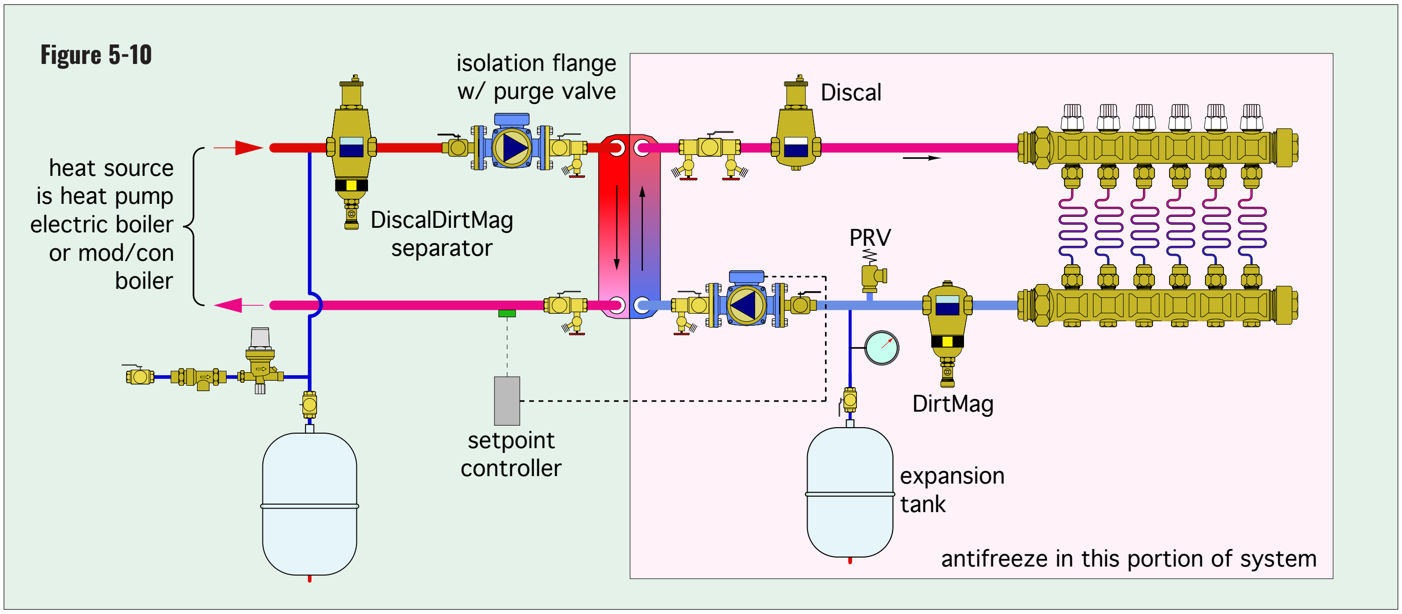


Figure 5-9





be protected against sustained flue gas condensation. Consider that a garage floor slab that has not operated for several days during the winter could be very cold — even below 32ºF — when this portion of the system starts operating. The very cold antifreeze solution returning from the floor circuits poses two "threats" to the system. One is that it could freeze the water on the primary side of the heat exchanger. Given the rapid response of a brazed plate heat exchanger, and its low water content, ice crystal can form quickly — in some cases, in less than one minute. Any delay in getting heat to the primary side of the heat exchanger increases the possibility of freezing. A failure of the circulator responsible for circulating heated water through the primary side of the heat exchanger could also lead to freezing.

One way to protect against this situation is to install a temperature setpoint controller that monitors the temperature of water *leaving* the primary side of the heat exchanger. When this temperature rises to some setpoint deemed sufficient to verify that heat is being transferred to the heat exchanger, this controller turns on the circulator for the antifreeze side of the heat exchanger.

The other "threat" posed by a very cold slab is sustained flue gas condensation in a conventional boiler. Keep in mind that the rate of heat transfer to a very cold slab could be significantly higher than under normal operation. Heat exchangers will "transmit" this high rate of heat flow back to the heat source.

One approach to protecting the boiler is to use an injection mixing controller to regulate the temperature of the antifreeze solution supplied to the floor circuits. This controller also monitors the inlet temperature to the boiler and reduces the flow of hot water to the primary side of the heat exchanger, when necessary, to prevent sustained flue gas condensation. A motorized 3-way mixing valve and suitable controller is another option that can provide both supply temperature control and boiler protection.

Another approach to this scenario uses two thermostatic mixing valves, as shown in Figure 5-9.

The 3-way mixing valve on the right side of the heat exchanger regulates the supply temperature to the floor circuits. It is important to select this mixing valve with a Cv rating close to the total flow rate required by the manifold station. This will limit the pressure drop across the mixing valve to approximately 1 psi. A Caleffi ThermoProtec valve on the left side of the heat exchanger limits hot water flow through the primary side of the heat exchanger, as necessary, to keep the boiler inlet temperature high enough to prevent sustained flue gas condensation.

When the heat source is a heat pump, electric boiler or mod/con boiler, it is not necessary to using mixing devices in the system. Figure 5-10 shows a representative configuration.

The heat exchanger should be selected so that it can supply the design heating load to the floor circuits while operating at an approach temperature drop of no more than 5ºF. This allows the heat source to operate just a few degrees F above the supply temperature of the floor circuits. Low operating temperature improves the efficiency of mod/con boilers and heat pumps.

6. HEAT EXCHANGER APPLICATIONS

This section describes several common applications, as well as some "novel" applications for heat exchangers in hydronic heating and cooling systems. They include:

- Snow & ice melting
- Pool heating
- Domestic water heating
- Heat input and extraction from non-pressurized thermal storage tanks
- District heating
- Heat interface units (HIUs)
- Lake water cooling

Conceptual piping diagrams are presented for these applications. Designers need to determine the suitability of the concepts, as well as the required specifications of the heat exchangers, piping, circulators, etc., for specific applications. References to other issues of *idronics* are given to provide further design assistance.

SNOW & ICE MELTING (SIM) SYSTEMS

The use of hydronics technology for snow and ice melting dates back to the mid-1900s, when copper tubing was embedded in exterior pavements. Today, PEX, PERT and PEX-AL-PEX tubing provides a reliable way to distribute heat from a boiler to exterior pavements such as sidewalks, steps, driveways, emergency room arrival areas and even complete parking lots.

SIM systems are categorized based on how they are expected to perform relative to snow fall rates and subsequent melting times.

Class 1 SIM systems are commonly used for residential walkway and driveway areas. They deliver heat to pavement at rates between 80 and 125 Btu/hr/ft^2. These rates are not necessarily high enough to melt snow as it falls, especially during strong storms. Several hours may be needed to completely melt the snow, especially when the system starts from a cold condition.

Class 2 SIM systems are generally sufficient for pavements at retail and commercial sites. They deliver heat at rates between 125 and 250 Btu/hr/ft^2. This class of SIM can usually melt snow as it falls — *once the pavement is up to normal operating temperature*.

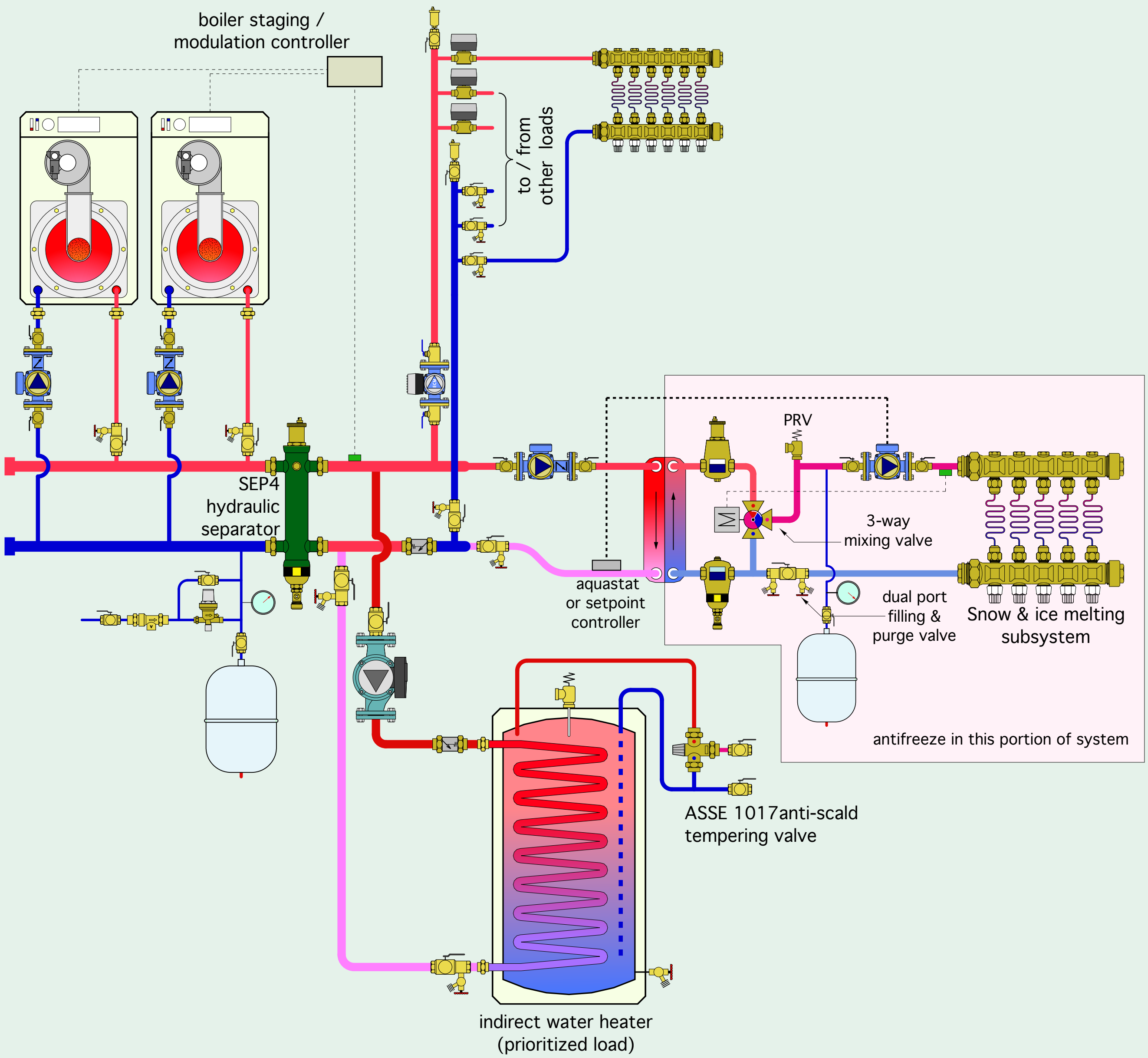
Class 3 SIM systems are for the most demanding applications where pavements must remain free of snow and ice at all times. Examples include steeply sloped exterior ramps at parking garages and hospital emergency room entrances. Typical heat delivery rates range from 250 to 450 Btu/hr/ft^2.

Figure 6-1

Courtesy of Gary Todd & REHAU

Courtesy of REHAU

Figure 6-2



Some SIM systems are single purpose. They are designed solely for snow and ice melting. These systems are usually supplied by one or more boilers. The entire system operates with an antifreeze solution, and no heat exchangers are needed between the boiler(s) and distribution system.

In other systems, snow and ice melting is one of several loads served by a common heat source. The other loads are typically space heating and domestic water heating. In these systems, a heat exchanger is often used to separate the SIM subsystem from the remainder of the system. This allows the majority of the system to operate with water rather than antifreeze. Figure 6-2 shows a representative configuration.

Figure 6-3 shows portions of snowmelting systems supplied through plate & frame heat exchangers.

Figure 6-3a

Figure 6-3b

Courtesy of West Coast Hydronics & Plumbing, Sacramento CA

POOL HEATING

Installations in which a boiler or hydronic heat pump supplies space heating in winter and year-round domestic hot water often have plenty of available heating capacity between the end of one heating season and the beginning of the next. If the property has a pool, that available heating capacity can be used to maintain it at comfortable conditions. Figure 6-4 shows how a heat exchanger specifically designed for pool heating can be installed.

This system is very similar to the one in Figure 6-2. The SIM subsystem from Figure 6-2 has been replaced by a pool heating subsystem.

Most heat exchangers intended for pool heating use a shell & tube design, which is better suited to higher flow rates needed for pool filter systems. Water from the heat source passes through the shell of the heat exchanger, which can be made of steel, cast iron or cast brass, and is part of the closed-loop system. Pool water passes through the tube bundle, which is typically made of titanium or titanium alloyed with stainless steel to resist the corrosive effects of highly chlorinated pool water.

The subsystem shown in Figure 6-4 uses the pool filter pump to move pool water through the tube bundle in the heat exchanger. Bypass piping is shown along with a manually operated bypass valve. The shell side of the heat exchanger is also piped to CPVC unions and ball valves. This detailing allows the heat exchanger to be removed from service if necessary, while still maintaining the ability to circulate pool water through the filter.

Swimming pool water temperature is typically maintained between 75 and 90ºF. This relatively low temperature allows heat sources such as mod/con boilers and hydronic heat pumps to operate at high efficiencies — assuming that the heat exchanger is sized for a "tight" approach temperature difference of perhaps 5ºF.

If one or more conventional boilers are used as the heat source, it is important to protect the boiler(s) from sustained flue gas condensation. This can be done using Caleffi *ThermoProtec*™ valves on the boiler(s), as shown in Figure 6-5.

Swimming pools have very high thermal mass. Without proper detailing, this mass can "dominate" the water temperature in the system, possibly preventing other subsystems that may be operating at the same time from meeting their respective loads. *This condition must be avoided.*

One approach is to use controls that make pool heating a lower priority load relative to domestic water heating. Thus, if the domestic water-heating load is active, flow to the primary side of the pool heat exchanger is temporarily suspended. This is essential if the boilers are programmed

Figure 6-4

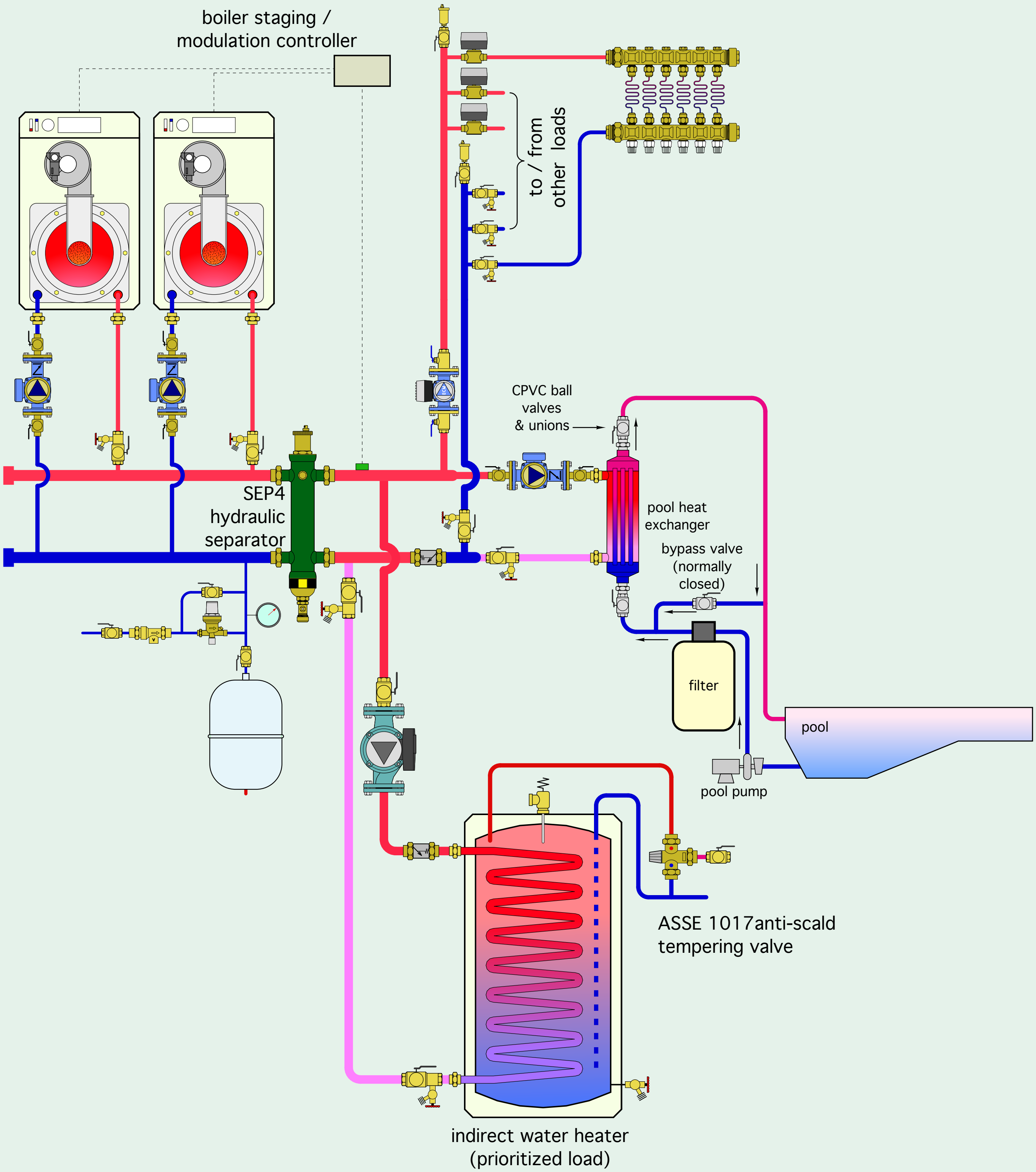
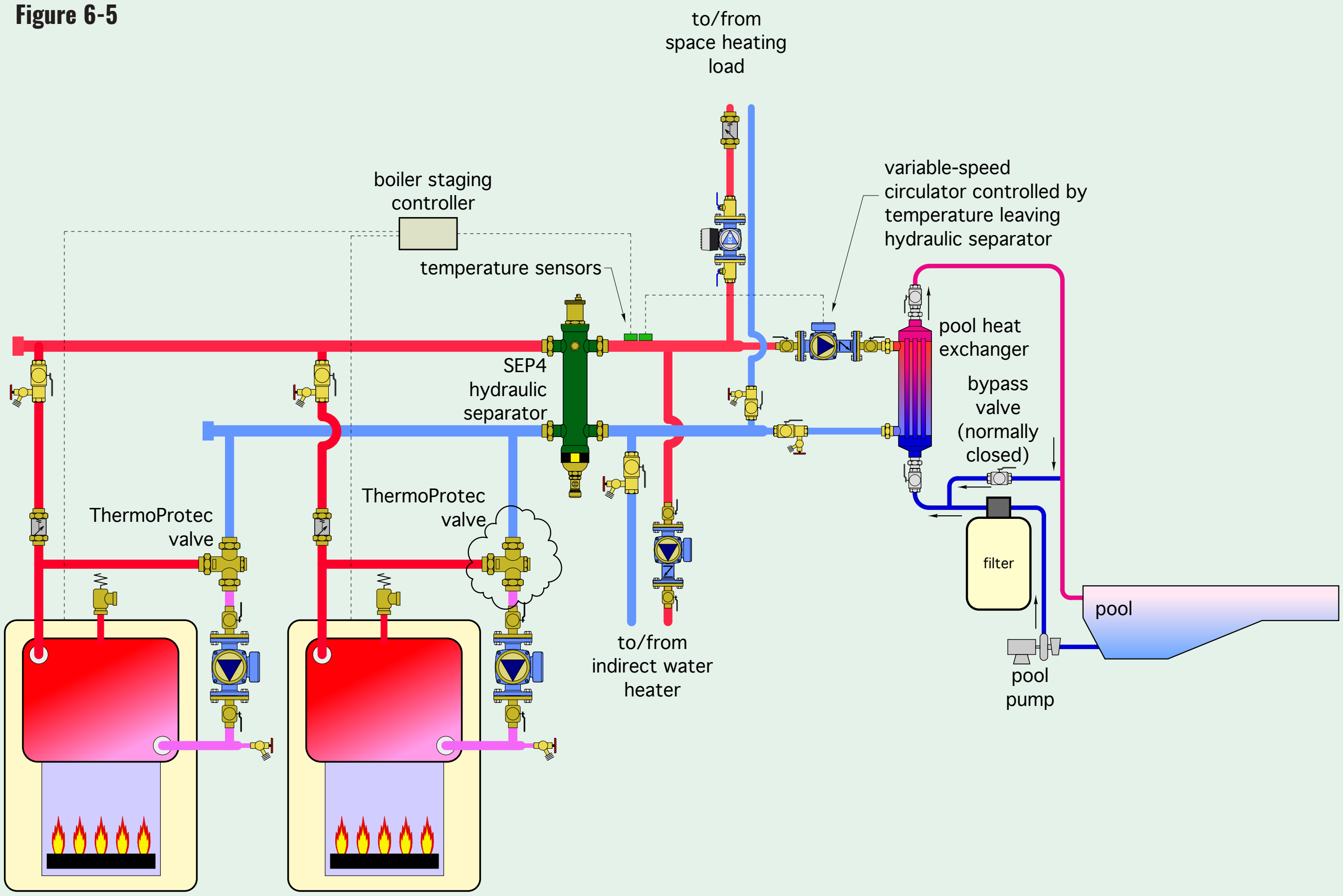

boiler staging /
modulation controller
to / from
other loads
CPVC ball
valves
& unions
SEP4
hydraulic
separator
pool heat
exchanger
bypass valve
(normally
closed)
filter
pool
pool pump
ASSE 1017anti-scald
tempering valve
indirect water heater
(prioritized load)

Figure 6-5



to operate at higher water temperatures during the domestic water-heating mode.

Another method is to use a "temperature-controlled" variable-speed circulator to supply water to the primary side of the pool heat exchanger. The speed of this circulator is controlled based on the water temperature leaving the hydraulic separator. If that temperature is below some predetermined value — deemed necessary to allow proper operation of other loads — the speed of the circulator decreases. This reduces the rate of heat transfer to the primary side of the pool heat exchanger. If the temperature is above this setpoint, the circulator operates at full speed. This detail is also shown in Figure 6-5.

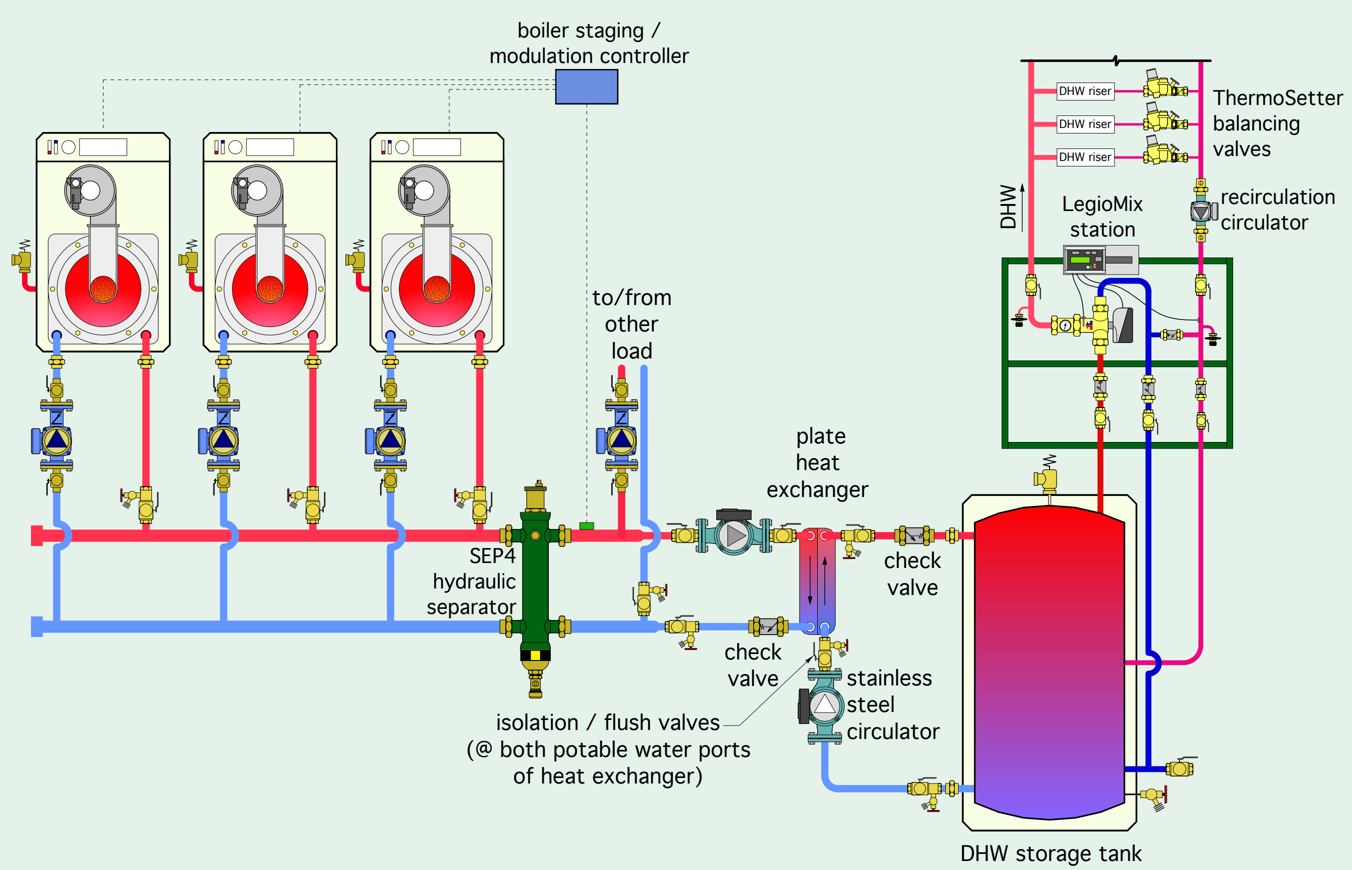
DOMESTIC WATER HEATING

The most common method of heating domestic water in hydronic systems that also supply space heating is by using an indirect water heater. Figures 6-2 and 6-4 show how such a water heater would be installed as a subsystem.

Indirect water heaters use internal coil heat exchangers. The rate at which they can transfer heat to the domestic water is typically limited by natural convection on the outer surface of the coil heat exchanger. In applications requiring very high rates of heat transfer, it may not be possible to find an indirect water heat with sufficient heat transfer capacity. In these situations, the use of a properly sized brazed plate heat exchanger can meet the heat transfer requirements. Figure 6-6 shows a possible piping configuration for this approach.

Assume that the water in the upper portion of the tank needs to be maintained at 140ºF. The stainless steel brazed plate heat exchanger would be sized to transfer the full output of the multiple boiler system with the water temperature supplied from the boilers at 145ºF. Assuming a nominal 20ºF temperature drop through the primary side of the heat exchanger, the 5ºF approach temperature difference keeps the boilers shown in Figure 6-5 operating at inlet water temperatures that produce some flue gas condensation, and thus an incremental increase in efficiency.

Figure 6-6



Combination isolation/flushing valves are installed on each domestic water port of the heat exchanger. These valves allow that side of the heat exchanger to be periodically isolated and flushed with a cleaning solution to remove scaling.

A stainless steel circulator is used between the heat exchanger and storage tank. The circulator supplying boiler water to the heat exchanger can be cast iron, since it is part of the closed-loop system. Designers must ensure that these circulators, as well as the associated piping, are sized for flow rates that can transfer the full output of the boiler system to the domestic water tank when necessary.

For example, if the maximum continuous domestic water demand was 15 gpm, with a temperature rise from 50 to 140ºF, the required heat transfer rate is:

Assuming a 20ºF temperature change across the primary side of the heat exchanger (e.g., 145ºF entering down to 125ºF leaving), the flow rate into the primary side must be 67.5 gpm. This will require a 2.5" copper tube to maintain a reasonable flow velocity.

A commercially available sizing software suggests that the minimum heat exchanger for this heat transfer requirement is a 10" x 20" x 50 plate brazed plate unit. At this flow rate, the pressure drop through that heat exchanger is about 1.7 psi (about 4 ft. of head). Assuming that the other components and piping between the hydraulic separator and heat exchanger add 5 feet of head, the selected circulator needs to provide a minimum of 67.5 gpm at 9 feet of head. This will likely require a light commercial class circulator, not a small zone circulator. If the wire-to-water efficiency of that circulator is estimated at 30%, the estimated input power to the circulator would be about 380 watts.

Unfortunately, there have been systems where an undersized circulator, along with undersized piping, have created flow "bottlenecks" in this approach to high-capacity domestic water heating. Flow bottlenecks cause

Figure 6-7

Courtesy of Diversified Heat Transfer, Inc.

heat transfer bottlenecks. The latter can severely limit the heat transfer rate between the boilers and domestic hot water. Under these conditions, the boilers will quickly reach their high limit settings and short cycle. The boiler plant may have sufficient heating capacity for this high demand DHW load, and the heat exchanger may have sufficient heat transfer capacity, but without adequate flow, the overall process can be severely limited.

Pre-engineered and pre-assembled products that use external brazed plate heat exchangers in combination with domestic hot water storage tanks are available for high-volume commercial applications. Figure 6-7 shows one example.

ON-DEMAND DOMESTIC WATER HEATING

Another approach to domestic water heating uses a brazed plate heat exchanger sourced from a buffer tank, or a boiler system, to create domestic hot water literally seconds before it is required at a fixture.

The assembly shown in Figure 6-8 can be used in residential and light commercial systems that have a heated buffer tank. That tank may be associated with renewable heat sources, such as solar thermal collectors, biomass boilers or heat pumps. It could also be heated by a fossil fuel boiler.

When the domestic hot water drawn by the plumbing distribution system

Figure 6-8

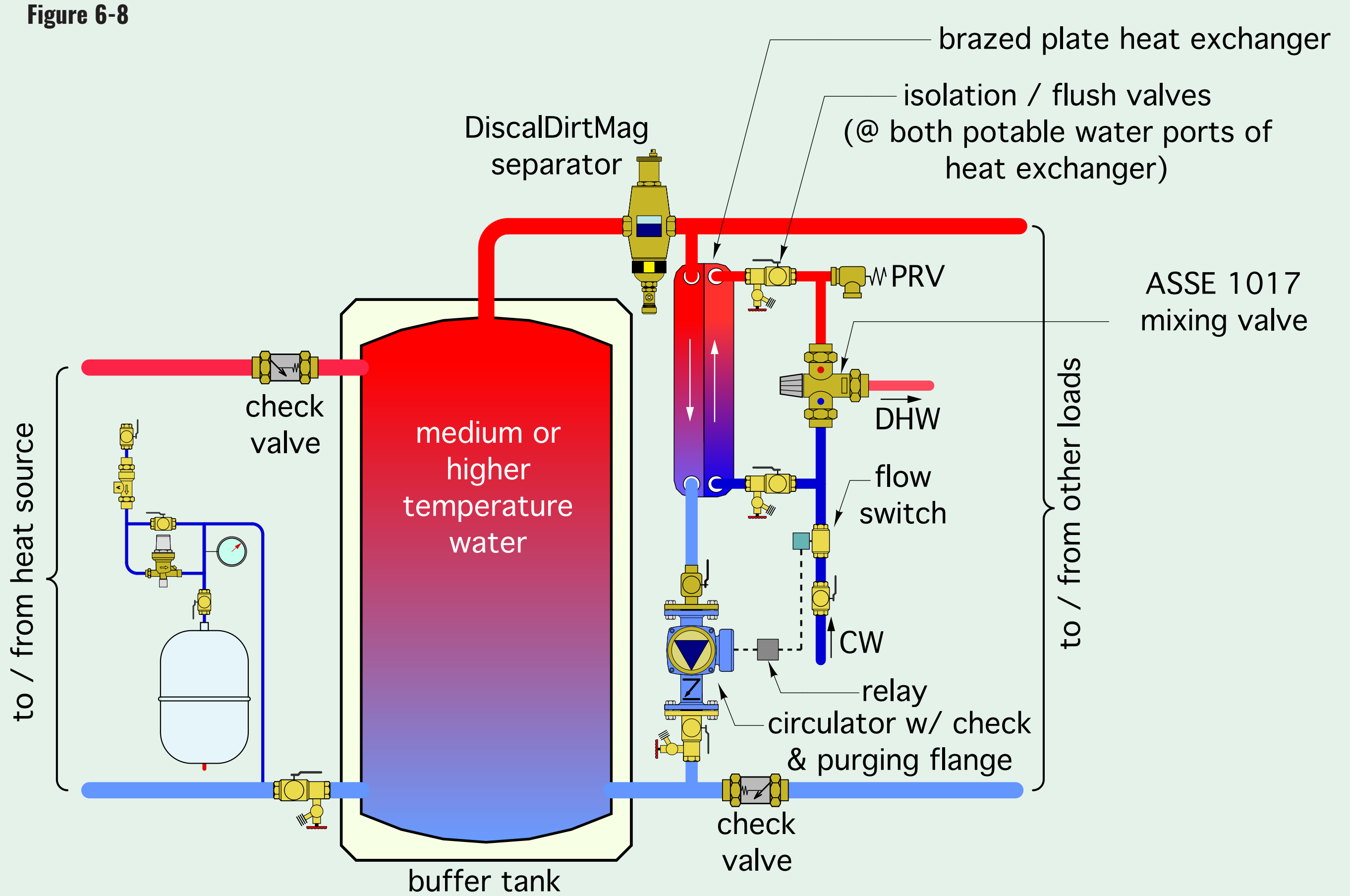


Figure 6-9a

Courtesy of Harwill

Figure 6-9b

exceeds 0.7 gpm, a flow switch, rated for use with domestic water, closes it contacts. This energizes the coil of a relay, which turns on a circulator to draw hot water from the top of the buffer tank through the primary side of the heat exchanger. After passing through the heat exchanger, this flow returns to the lower portion of the buffer tank. Cold domestic water flows through the secondary side of the heat exchanger in a counterflow direction, absorbing heat from the primary side.

Figure 6-9a shows an example of a flow switch suitable for use with domestic water. The switch closes its contacts at flow rates of 0.7 gpm or higher and opens its contacts at flow rates of 0.4 gpm or less. This switch is designed to thread into a 3/4" stainless steel tee, as shown in Figure 6-9b.

The high surface area-to-volume ratio of the brazed plate heat exchanger, combined with a minimal amount of piping connecting it to the buffer tank, allows heated domestic water to be available within 3 to 5 seconds of the flow switch contact closure. This is significantly faster than the response time required by a gas-fired tankless water heater. This approach also has the advantage of not storing domestic hot water when it is not operational, which reduces the potential for Legionella growth.

The system shown in Figure 6-8 assumes that the water in the buffer tank is maintained hot enough to fully heat the domestic water on a single pass. This may or may not be the case, depending on the heat source, the space heating emitters, the required domestic hot water delivery temperature and the sizing of the heat exchanger. When the thermal storage tank is relatively hot, and the heat exchanger is sized for a close-approach temperature difference, there is no need for a temperature boost after the domestic water leaves the heat exchanger. An ASSE 1017 point-of-distribution mixing valve should always be installed to protect the fixtures from hot water delivery temperatures over 120ºF.

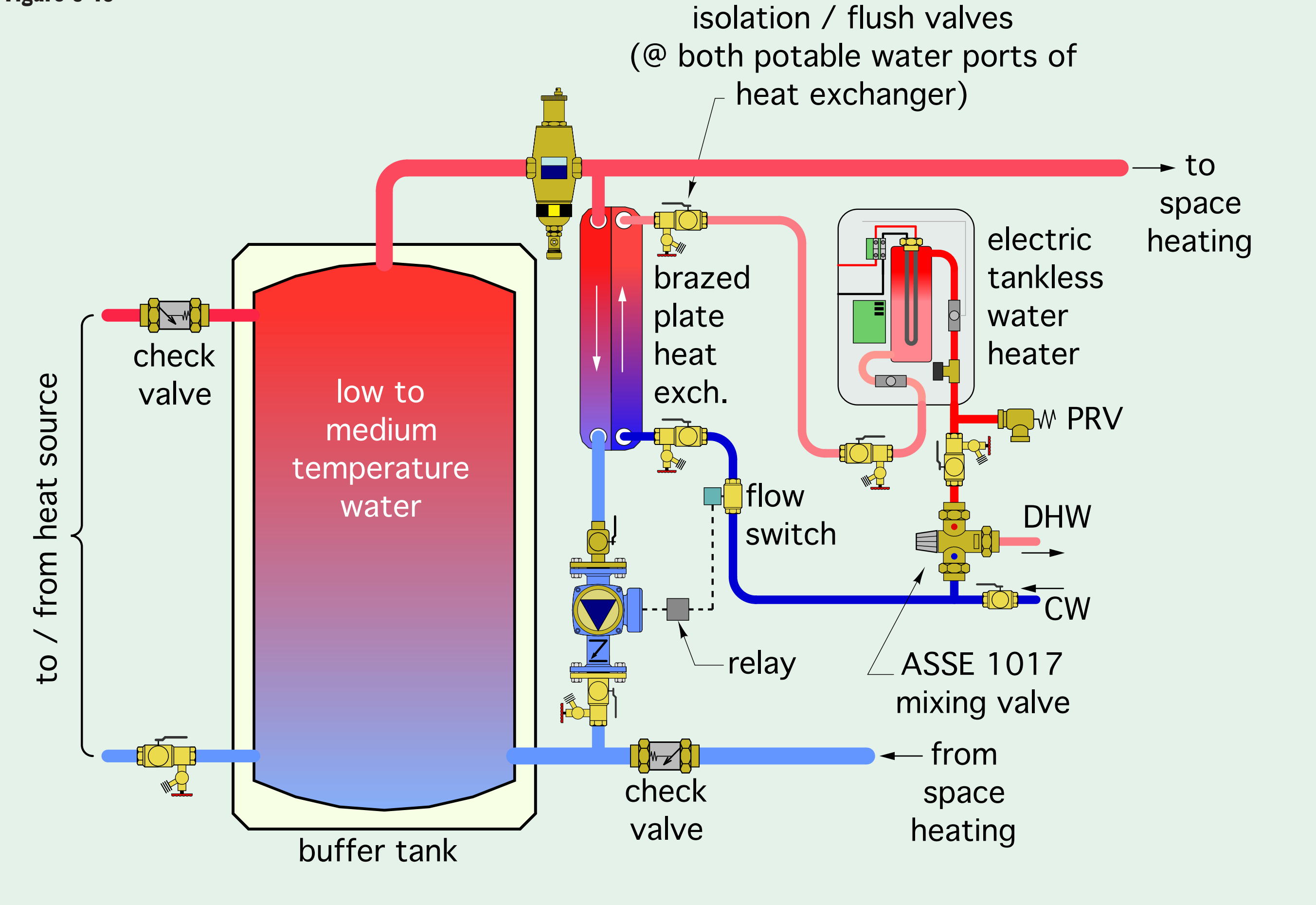
This application is a good example of when a heat exchanger with a high thermal length, as discussed in Section 4, is appropriate. The domestic water needs to undergo a large temperature change, but the LMTD between the two fluids will be relatively small. This type of application will favor a brazed plate heat exchanger with relatively long plate length.

This configuration includes two isolation/flushing valves installed at both domestic water ports of the heat exchanger. These valves allow the heat exchanger to be isolated from the rest of the domestic water system and flushed with a cleaning solution, as described in Section 5.

Another detail is the use of two spring-check valves adjacent to the buffer tank. The check valve near the upper left connection prevents reverse thermosiphoning between the buffer tank and heat source. The valve in the lower right limits heat migration into the return side of the space-heating circuits at times when space heating is not required.

The brazed plate heat exchanger and its associated piping should be kept as close to the buffer tank as possible.

Figure 6-10



This minimizes the amount of system water that needs to be displaced at the start of a DHW draw and helps keep the response time to only a few seconds. The heat exchanger and its associated piping should also be insulated to reduce heat loss to the mechanical room.

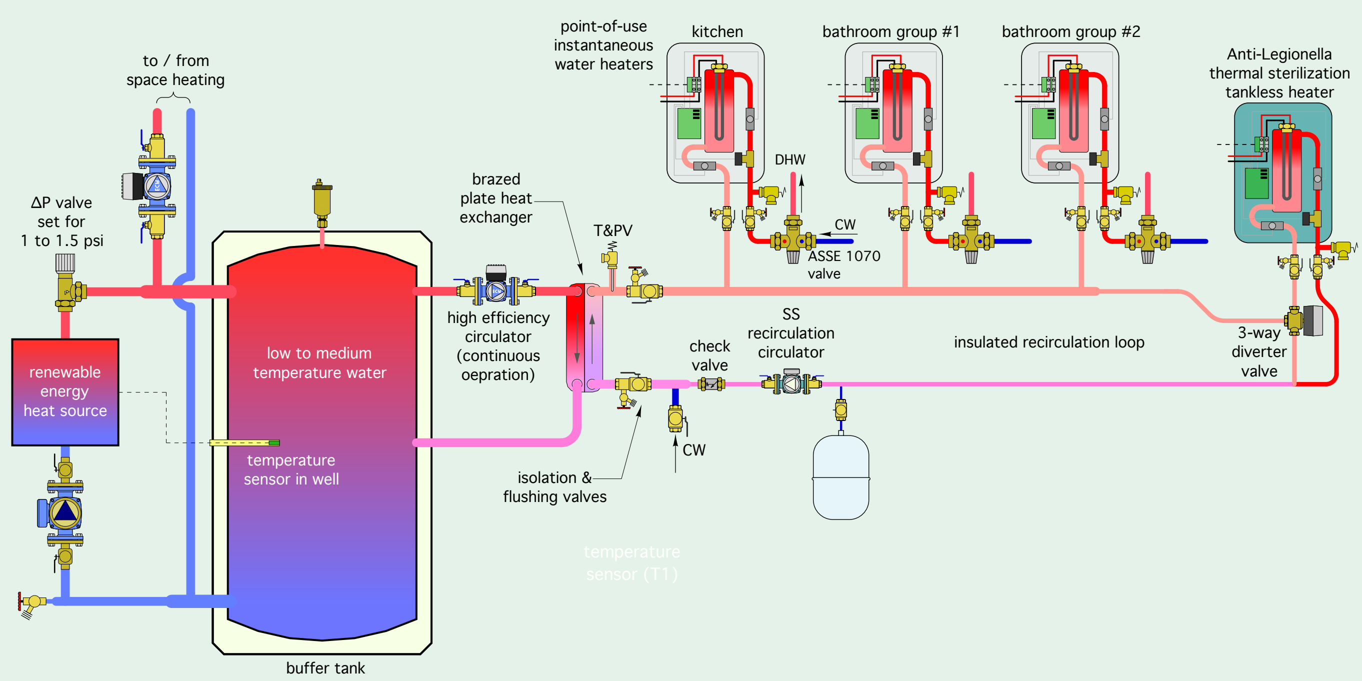
Figure 6-10 shows an extension of this approach. An electric tankless water heater has been added to boost the final DHW delivery temperature when necessary. This configuration would be useful when the buffer tank operates at temperatures below the required delivery temperature for DHW. In this case, the heat exchanger can still provide significant preheating, leveraging the low starting temperature of cold domestic water. Combination isolation/flushing valves are also installed on the piping connected to the tankless water heater to allow isolation and cleaning. It would also be possible to use a tank-type water heater to provide the final temperature boost.

Another variation on this approach to domestic water heating is shown in Figure 6-11.

This system uses a brazed plate heat exchanger to transfer heat from a buffer tank to a recirculation loop serving several DHW loads. The system shown in Figure 6-11 is based on use of a renewable energy heat source, such as a heat pump, to maintain the buffer tank in a low- to medium-temperature range — perhaps 90 to 110ºF. At these temperatures, the heat pump would operate at high COPs. If a space-heating load is also supplied by the system, as shown in Figure 6-11, the water temperature must be compatible with the heat emitters.

Several DHW load points are connected to the recirculation loop. Each is equipped with a point-of-use tankless electric water heater, which would provide the temperature boost required by each DHW load. Each tankless heater can be individually operated, and each can have a different delivery temperature setpoint.

Figure 6-11



If the water in the recirculation loop reaches a temperature equal to or above the DHW delivery temperature of one or more of the tankless heaters, the heating elements in those heaters do not operate.

The tankless heater and 3-way diverter valve at the far end of the recirculation loop is solely for thermal disinfection. Once each day, the diverter valve is powered on to direct flow through this tankless heater, which has a setpoint of 140ºF. This raises the temperature of the recirculation loop and holds it there for at least 30 minutes to kill any Legionella bacteria in the recirculation loop. Hot water drawn from this loop during the disinfection cycle passes through the ASSE 1070 point-of-use anti-scald mixing valves, which limit temperature to the fixtures to 120ºF. After the thermal disinfection cycle is complete, the dedicated tankless heater is powered off, and the diverter valve directs flow around it. The tankless heater used for thermal disinfection only needs to be sized to the heat loss of the recirculation system when operating at peak disinfection temperature. This may only be 3 or 4 kilowatts, depending on the amount of piping used and how it is insulated.

It is important to insulate all portions of the recirculation system, including the heat exchanger, to limit heat loss into the building.

The brazed plate heat exchanger would be sized for a tight approach temperature difference. It should be capable of transferring the full heat output rate of the renewable energy heat source when the tank is at its minimum temperature.

The buffer tank is configured in a “2-pipe” configuration. The piping supplying warm water for low-temperature space-heating tees into the tank headers just upstream of the tank. A differential pressure valve is used in the heat source circuit to provide minimum forward opening pressure of 1 to 1.5 psi. This prevents water returning from the space-heating distribution system from passing through the heat source when it is not operating. It also replaces the need for a check valve to prevent reverse thermosiphoning between the buffer tank and heat source.

If the renewable energy heat source supplying the buffer tank is turned on and off automatically, as would be the case with a heat pump or pellet boiler, that on/off operation must be based on the temperature of the water in the tank.

The concept of on-demand domestic water heating using an external heat exchanger can also be scaled up to high-volume commercial applications. Figure 6-12 shows one example of a pre-engineered, pre-assembled system that uses high-temperature boiler water in combination with a plate & frame heat exchanger for high-volume domestic water heating.

Figure 6-12

Courtesy of Diversified Heat Transfer, Inc.

The product shown in Figure 6-12 includes a modulating control valve to control boiler water flow through the primary side of the heat exchanger, and thus regulate domestic water delivery temperature. It also has controls, a stainless steel recirculation circulator, strainers and pressure-relief valves.

A final thought on the use of brazed plate or plate & frame heat exchangers for domestic water heating. Some practitioners argue against this practice. The common retort is that hard water can scale such heat exchangers. While this is a possibility, it's also a possibility with internal coil heat exchangers, especially when the latter are operated at high internal temperatures to force a given rate of heat exchange through a relatively small surface area.

Most indirect water heaters have permanently installed internal coil heat exchangers. When the exterior surface of that coil is scaled, there is no practical way to clean it. This is not the case with a brazed plate heat exchanger equipped with the combination isolation/flushing valves shown in several figures involving domestic water within heat exchangers. If the scaling is beyond the point where chemical cleaning can correct it, the brazed plate heat exchanger can be replaced. When the domestic water side of the internal coil heat exchanger is heavily scaled, the entire indirect tank needs to be replaced, which is typically much more costly.

Hard domestic water will scale *any* heat exchanger operating at temperatures high enough to cause dissolved minerals to precipitate out of that water. The hardness needs to be reduced through softening, or the scaling problems will persist, regardless of the type of heat exchanger used.

HEAT EXCHANGERS FOR NON-PRESSURIZED THERMAL STORAGE TANKS

Some renewable energy heating systems, such as those supplied by pellet-fired boilers or cordwood gasification boilers, require several hundred gallons of thermal storage. In some systems, that storage is provided by pressure-rated, ASME-listed steel tanks. These tanks are expensive, and due to their size, the logistics of moving them into place within a mechanical room can be challenging, and sometimes even impossible. One solution is the use of a "panelized," non-pressurized tank that can be moved into the mechanical space in pieces and then assembled.

Non-pressurized thermal storage tanks are vented to the atmosphere. As such, they are "open" to absorption of oxygen from the atmosphere. Although it is possible to design a hydronic system that can connect directly with a non-pressurized tank, there are several limitations to this approach that are not present in a "closed" hydronic system.

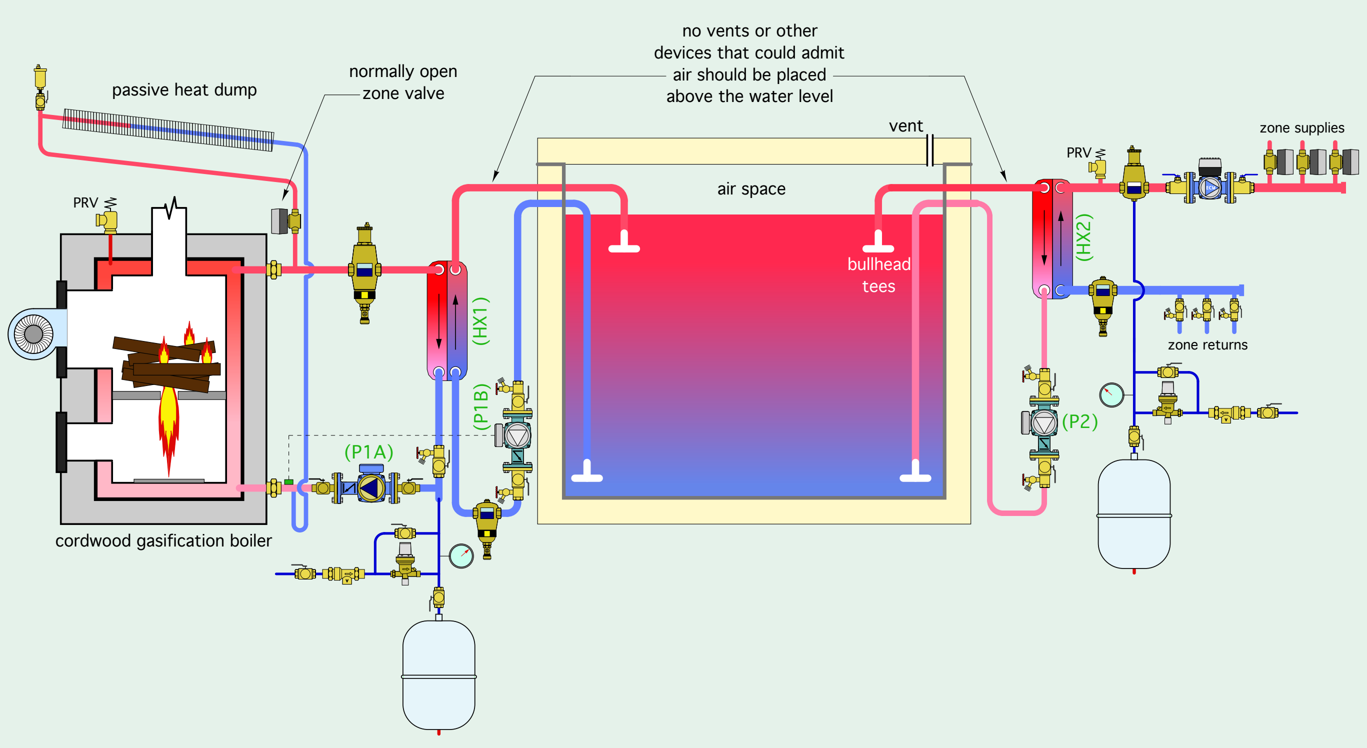
One issue is that any steel or cast iron components connected to the water in the non-pressurized tank would be subject to severe corrosion due to the higher dissolved oxygen content of the water. There are chemicals called oxygen scavengers that can partially compensate for this corrosion, but the concentration of these chemicals must be monitored and maintained, which unfortunately doesn't always happen. This can cause premature failure of expensive hardware, including boilers, circulators, radiators and other components made of carbon steel or cast iron.

A better solution is to separate the water in a non-pressurized thermal storage tank from both the steel boiler and any ferrous metal in the distribution system. This can be done using internal coil heat exchangers mounted within the storage tank, as discussed in Section 2. It can also be done with external brazed plate heat exchangers. Of these choices, the latter provides several advantages including:

• Forced convection on both sides of the heat exchanger for much higher convection coefficients, and the ability to achieve the necessary heat transfer rates using smaller heat exchanger surface area.

• The ability to accurately estimate the performance of the heat exchanger. It is difficult to find detailed performance models for internal coil heat exchangers operating over a wide range of conditions.

Figure 6-13



• The ability to service or replace the heat exchanger if ever necessary without opening the thermal storage tank.

• The possibility of using a single heat exchanger for both heat input and heat extraction from the tank.

The system shown in Figure 6-13 uses two brazed plate heat exchangers: One between the cordwood gasification boiler and the thermal storage tank, and the other between the tank and the distribution system.

The boiler-to-tank heat exchanger (HX1) is sized to transfer the boiler's maximum rate of heat output to the tank water with only 5ºF approach temperature difference between the water entering the heat exchanger from the boiler, and the water leaving it for the tank.

Circulator (P1A) runs at a constant speed whenever the boiler is enabled to operate. A variable-speed stainless steel circulator (P1B) provides flow between the heat exchanger and tank. The speed of this circulator varies in response to the boiler inlet temperature. At boiler inlet temperatures below 130ºF, circulator (P1B) is off. Its speed ramps up as boiler inlet temperature climbs above 130ºF, reaching full speed at an inlet temperature of 140ºF. This control action prevents the boiler from operating with sustained flue gas condensation.

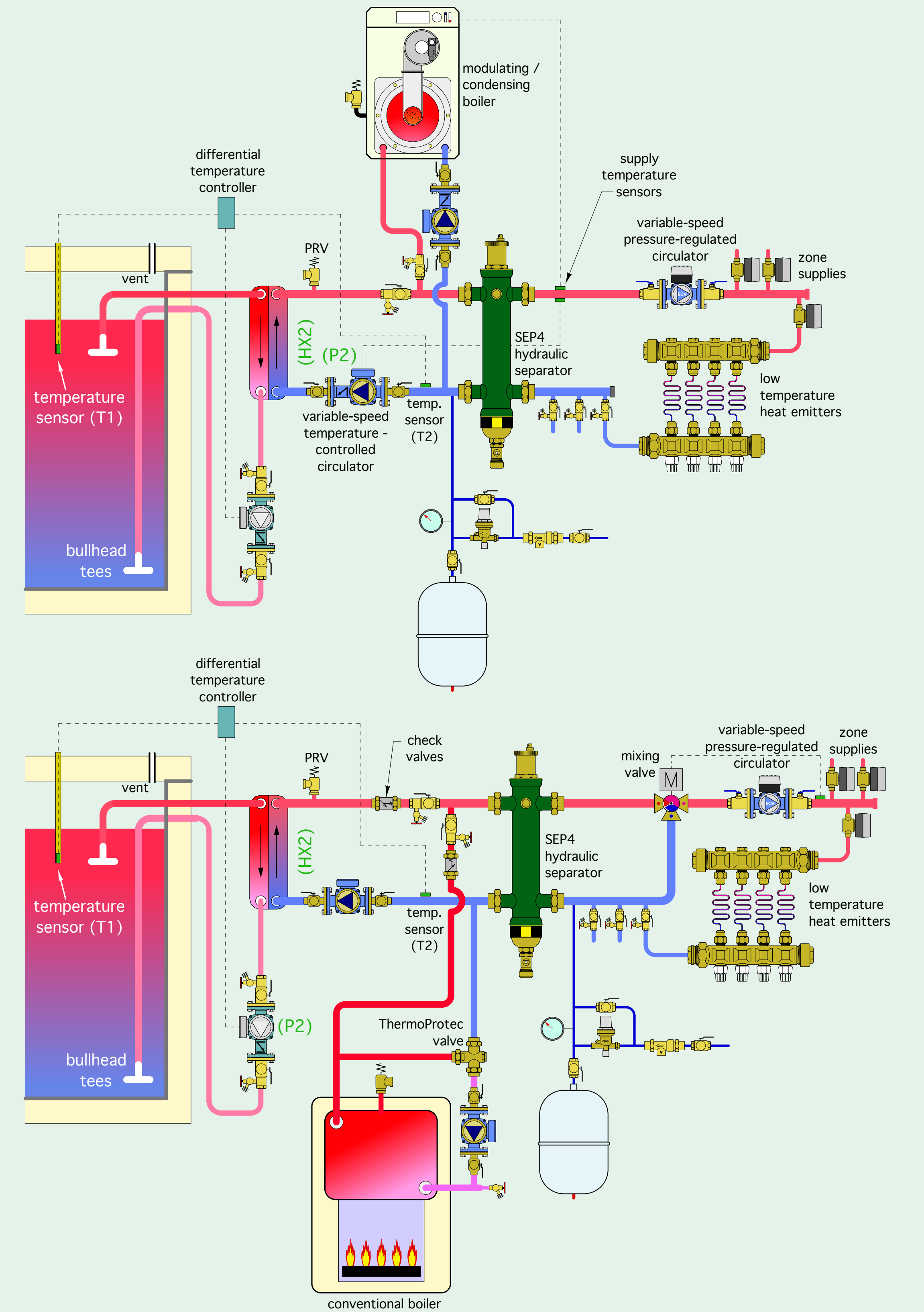
The pipe carrying hot water from heat exchanger (HX1) passes through the tank wall above the water line and terminates in a bullhead tee. This releases water horizontally and helps preserve beneficial temperature stratification within the tank.

Circulator (P1B) is mounted low relative to the water level inside the tank. This increases the net positive suction head available to the circulator and helps prevent cavitation. This is important because the water at the top of the tank is at zero gauge pressure and can be very hot. Both of these conditions *decrease* the margin against cavitation.

The piping that's slightly above the water level in the tank will be under slight sub-atmospheric pressure. There should not be any air vents, valves or other devices in this piping that could potentially allow air to enter the piping.

Purging flanges are installed on both ports of circulator (P1B). These allow water to be forced into the piping to displace air when the system is commissioned.

Figure 6-14



modulating / condensing boiler
differential temperature controller
supply temperature sensors
variable-speed pressure-regulated circulator
zone supplies
PRV
vent
(HX2)
(P2)
SEP4 hydraulic separator
low temperature heat emitters
temperature sensor (T1)
variable-speed temperature - controlled circulator
temp. sensor (T2)
bullhead tees
differential temperature controller
check valves
variable-speed pressure-regulated circulator
zone supplies
mixing valve
M
PRV
vent
(HX2)
SEP4 hydraulic separator
low temperature heat emitters
temperature sensor (T1)
temp. sensor (T2)
(P2)
ThermoProtec valve
bullhead tees
conventional boiler

The details between the tank and load side heat exchanger are similar to those on the heat input side. Circulator (P2) is also a stainless steel circulator.

A variable-speed pressure-regulated circulator provides flow to the zoned distribution system.

Because there are two completely isolated closed-loop portions within the overall system, two pressure relief valves, two expansion tanks and two make-up water assemblies are required. The expansion tanks can be relatively small because they only service their respective closed-loop subsystems. The expansion and contraction of water within the tank simply changes the water level slightly.

This system requires two heat exchangers between the heat source and the heat emitters. There must be some temperature drop across each of these heat exchangers to create the necessary rates of heat transfer. These temperature drops should be kept small (suggested values of not more than 5ºF). The lower the temperature drop across heat exchanger (HX2), the lower the temperature of the thermal storage tank can get while still being able to satisfy the heat transfer requirements of the heat emitters. This can prolong the discharge cycle of the tank and reduce the number of boiler firings required.

Figure 6-14 shows two possibilities for adding an auxiliary boiler or re-pipe an existing boiler to the system shown in Figure 6-13.

The distribution system shown in Figure 6-14 serves low-temperature heat emitters. If a conventional boiler is used, it should be protected against sustained flue gas condensation by a Caleffi *ThermoProtec* valve. This high flow-capacity mixing valve automatically blends hot water leaving the boiler with cool water returning from the distribution system so that the water entering the boiler is at or above some minimum temperature. A typical minimum inlet temperature for a gas-fired boiler is 130ºF.

The speed of circulator (P2) is regulated to maintain a desired supply water temperature to the heat emitters. This detail also limits the supply water temperature to the heat emitters if and when the thermal storage tank is at a relatively high temperature.

When the thermal storage tank cannot contribute heat to the system, circulator (P2) would be off. The boiler circulator would be turned on whenever the boiler is operating. Check valves are used to prevent flow through the heat exchanger when circulator (P2) is off, or through the boiler when it is off.

Figure 6-14 also shows a differential temperature controller in both systems. This controller is energized whenever one or more zones are operating. It monitors the temperature difference between the water near the top of the thermal storage tank at sensor (T1), and the water returning from the distribution system at sensor (T2). Circulator (P2) is allowed to operate when one or more zones is operating, and the temperature at sensor (T1) is 8ºF or more above the temperature at sensor (T2). When this temperature difference drops to 5ºF or less, circulator (P2) is turned off. This control action prevents heat created by the auxiliary boiler from being inadvertently transferred into the thermal storage tank. The on and off settings for the differential temperature controller can be adjusted based on the operating characteristics of the distribution system and the approach temperature difference across heat exchanger (HX2).

DISTRICT HEATING

There are situations where it's desirable to heat several buildings from a central heat plant, rather than use individual heat systems within each building. Examples include college campuses, multiple apartment buildings in close proximity or groups of retail stores forming a shopping center. This can be done using a district heating system.

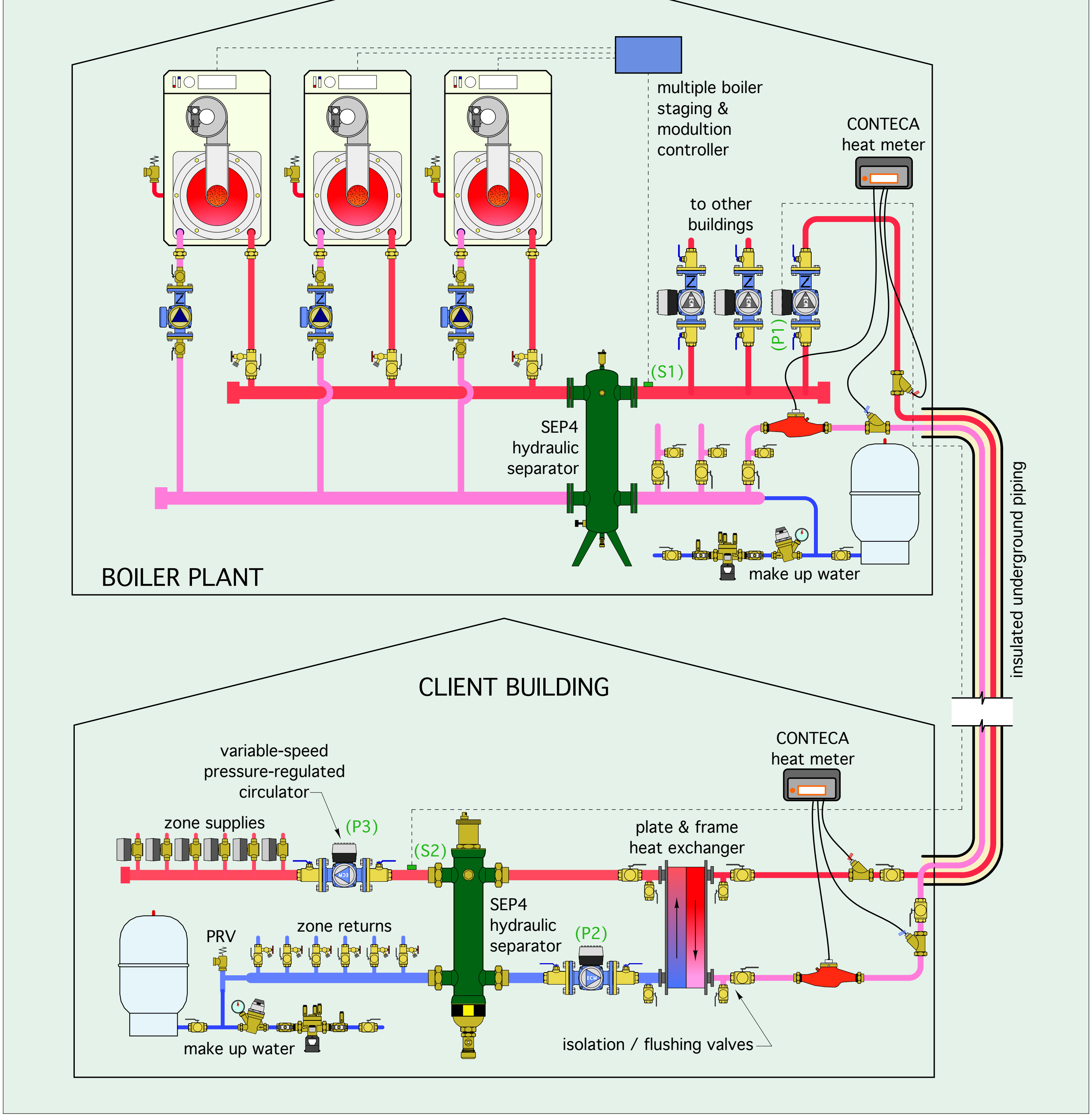
Most district heating systems have a central boiler plant where all heat is generated. That heat could come from fossil fuel boilers, renewable heat sources such as biomass boilers or a combination of these heat sources. The boiler plant typically is configured as a multiple boiler system to provide capacity control and provide redundancy if one boiler is down for service.

Heat from the boiler plant is distributed to multiple "client" buildings through insulated underground piping. The layout of this distribution piping depends on the locations of the client buildings. When the buildings are located along a street, a 2-pipe direct return piping system is typically used and installed parallel with the street. When the buildings form a cluster around the boiler plant, a homerun piping distribution system is appropriate. Figure 6-15 shows an example of the latter, with one of three client buildings represented.

In this system, heat is generated by three modulating/condensing boilers. Assuming that each boiler has a 10:1 turndown ratio, this provides a wide range of heat output capability, specifically a 30:1 system turndown ratio. This helps maintain good efficiency under partial load conditions.

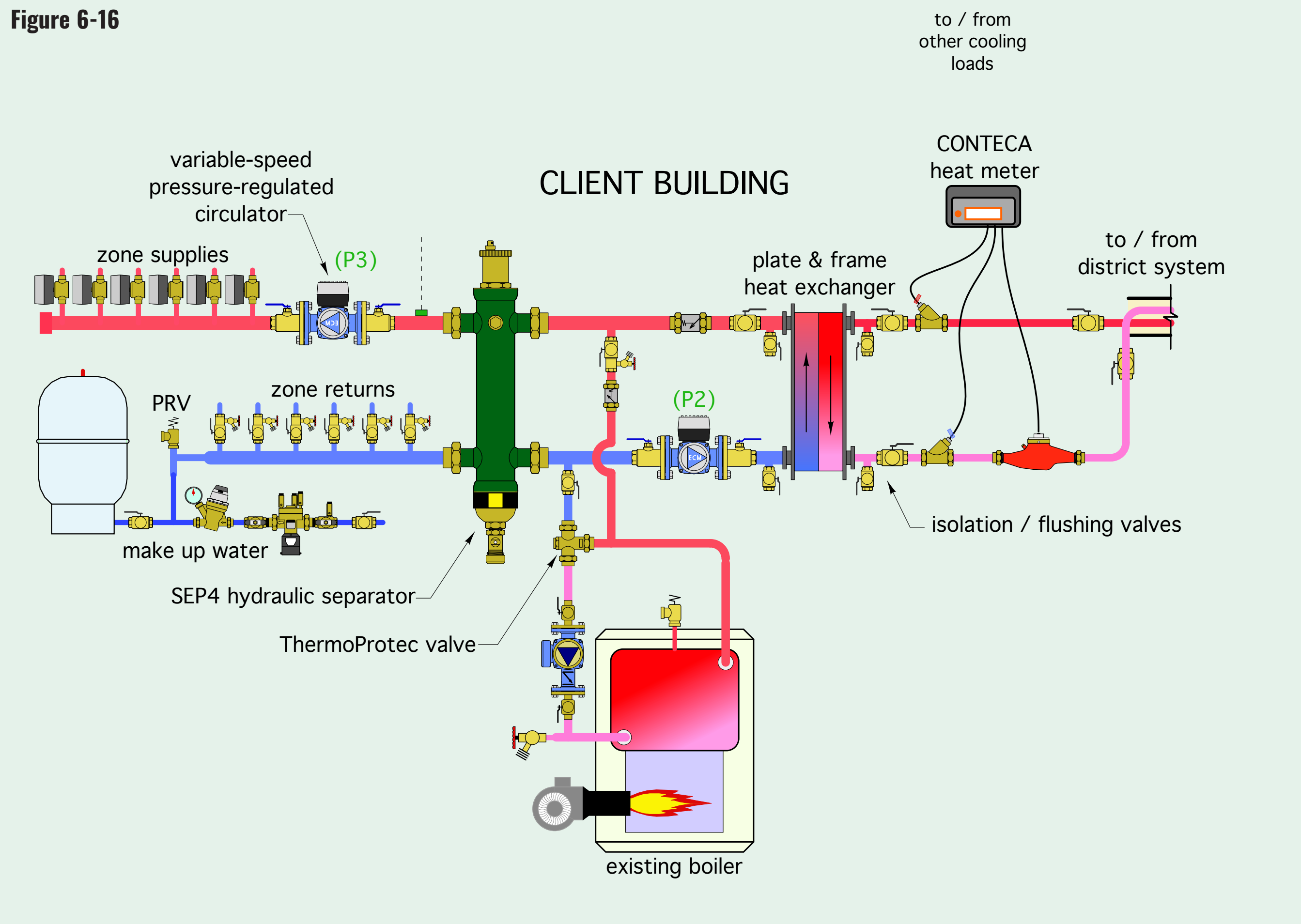
The boilers supply hot water to the SEP4™ hydraulic separator, which provides air, dirt and magnetic particle separation. It also isolates the pressure dynamics of the boiler circulators from the variable-speed circulators that provide flow to the client buildings.

Figure 6-15



Circulator (P1) is one of the distribution circulators. Its speed is based on temperature feedback from sensor (S2) on the supply side of the client building's distribution system. The controls for the system strive to maintain a target temperature at sensor (S2). That temperature could be a setpoint or based on outdoor reset control. If the measured temperature at sensor (S2) is below the target value, the speed of circulator (P1) is increased, and vice versa. This control action reduces pumping energy to the minimum needed for the current load in the client building.

Figure 6-16



A plate & frame heat exchanger is used to isolate the boiler plant and underground piping from the distribution system in the client building. This heat exchanger is fully insulated to minimize heat loss. This prevents a potential leak or servicing requirement on the client side of the heat exchanger from disrupting service to the other client buildings. This heat exchanger is provided with valves for isolation and flushing if necessary.

The left side of the heat exchanger, and all components connected to that side, form a closed-loop system that is completely separated from the boiler plant and underground piping. As such, this portion of the system requires a pressure relief valve, expansion tank, make-up water assembly, circulator and provisions for air, dirt and magnetic particle separation. These functions are provided by Caleffi SEP4 hydraulic separators.

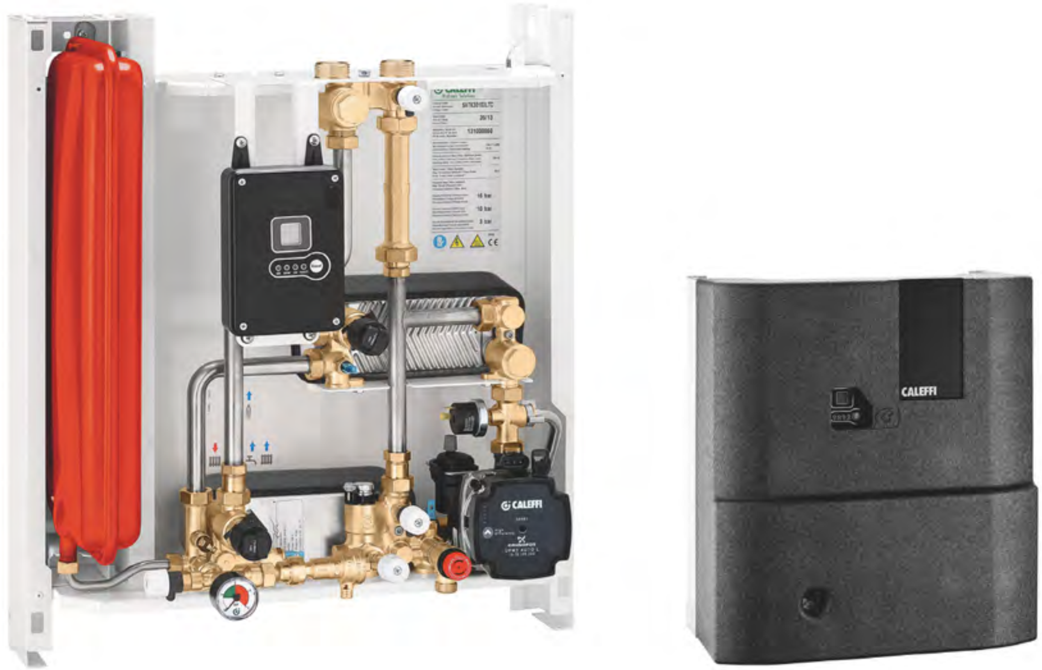
Two CONTECA™ heat-metering systems are shown in this system. The one in the boiler plant monitors all heat leaving the plant to the client building. The heat- metering system in the client building monitors all heat arriving at the building from the district system. The difference in readings between these two heat-metering systems would give the thermal loss in the underground piping. Some district systems may only need one of these heat-metering systems, depending on how the thermal losses of the underground piping will factor into the energy charges for the client buildings.

For more information on heat metering, see *idronics* 24.

Flow in the client building's distribution system is provided by a variable-speed pressure-regulated circulator (P2). The speed of this circulator automatically adjusts as the zone valves in the client building open and close.

There are many possible variations on this system. For example, one or more of the client buildings may have

Figure 6-17



an existing boiler that could provide heat to the building if the district system was down. One piping design that could accommodate this existing boiler is shown in Figure 6-16.

Zoning within the client building might use multiple circulators rather than zone valves. Some district systems can also be designed to provide chilled water cooling in summer as well has heating in winter.

HEAT INTERFACE UNITS

Another application that relies on brazed plate heat exchangers uses centralized heat production, or district heating, to supply space heating and domestic hot water for a multi-unit building containing apartments, condominiums, office spaces or retail shops. Heat from a central "plant" or district system is transported throughout the building using a 2-pipe distribution system. A third pipe carrying cold domestic water is routed along with these two pipes. At each building unit, the three pipes connect to a compact assembly called a *heat interface unit* (HIU). The heat passing from the 2-pipe building distribution system into each HUI is accurately metered. Each unit within the building is then charged for the thermal energy they use. The HIU also meters the total domestic water used.

Heat exchangers are a critical component within HIUs. They allow the primary water supplied from the building's central heat plant to be separated from the water used for space heating or domestic hot water within each unit. The separation of heating plant water from domestic water is an obvious requirement. The separation of heating plant water from the water used for space heating within each unit ensures that a potential leak in the space-heating subsystem within that unit can be isolated and repaired without disrupting service to other units served by the central heating plant.

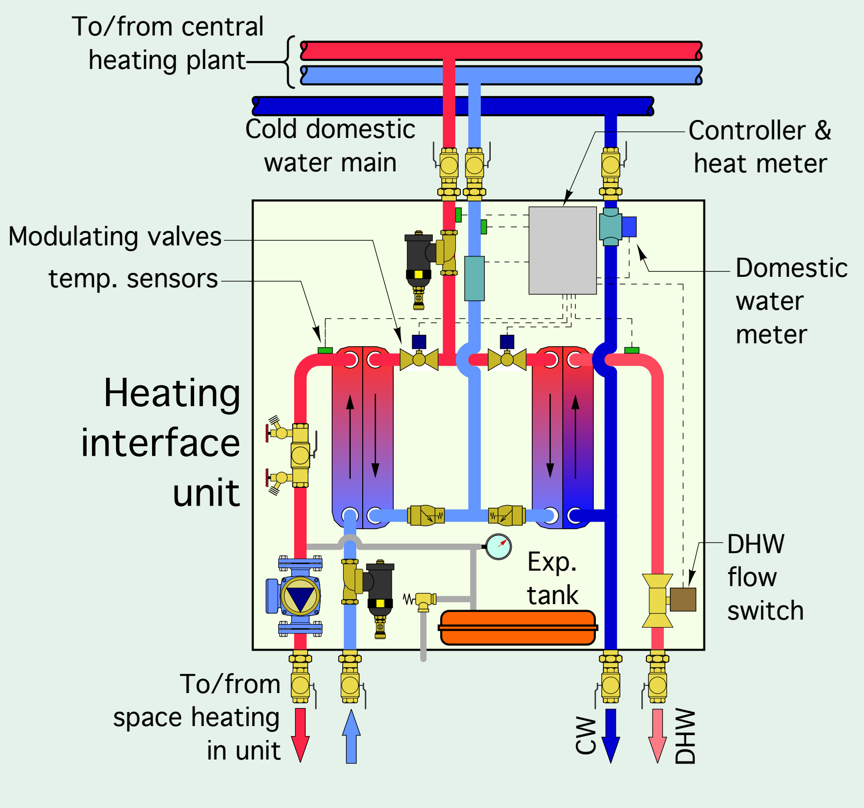
Figure 6-17 shows an example of a modern HIU that uses two brazed plate heat exchangers — one for space heating and the other for heating domestic water.

The HIU is mounted within a wall cavity in the building unit, close to the distribution pipes from the central plant. It has connections for the heating supply and return pipes, as well as one connection for cold domestic water.

When space heating is required, a modulating valve within the HIU regulates the flow of hot water from the central plant through the primary side of one brazed plate heat exchanger. This flow regulation is based on maintaining a desired supply water temperature leaving the secondary side of the heat exchanger and supplying the unit's space-heating distribution system.

A flow switch detects when there is a demand for domestic hot water. A second modulating valve then regulates flow from the central plant through the primary side of a second brazed plate heat exchanger. The flow regulation is based

Figure 6-18



on providing a desired domestic hot water temperature leaving the secondary side of the heat exchanger.

The total space heating and domestic hot water energy used is based on the flow rate and temperature drop of the central plant water from where it enters and leaves the HIU. The controller within the HIU records the total thermal energy used.

For more information on heat metering, see *idronics* 24.

The HIU shown in Figure 6-17 also includes a circulator and expansion tank for the space-heating distribution system within the building unit.

Figure 6-18 shows a *conceptual* piping arrangement for an HIU that provides the functions just described. It does not show exact hardware devices or their placement.

HIUs allow individually metered thermal energy for space heating and domestic hot water without the need for combustion heat sources within each unit. This reduces cost and increases safety by eliminating the need for gas piping and venting of combustion appliances dispersed throughout a building.

LAKE WATER COOLING

At depths of approximately 40 feet or more, the water temperature in lakes, located in climates with cold winters, remains relatively constant at about 39ºF. Water at this temperature is ideal for chilled-water cooling, if it can be accessed from shore.

The potential to use direct lake source cooling for smaller buildings depends on several factors. First, the building must be located close to a lake, and such that the intake pipe can be located at a sufficient depth to extract cool water. Ideally, the intake pipe would be at least 30 feet beneath the lake surface. However, lesser depths may be possible depending on the water temperatures required by the cooling system.

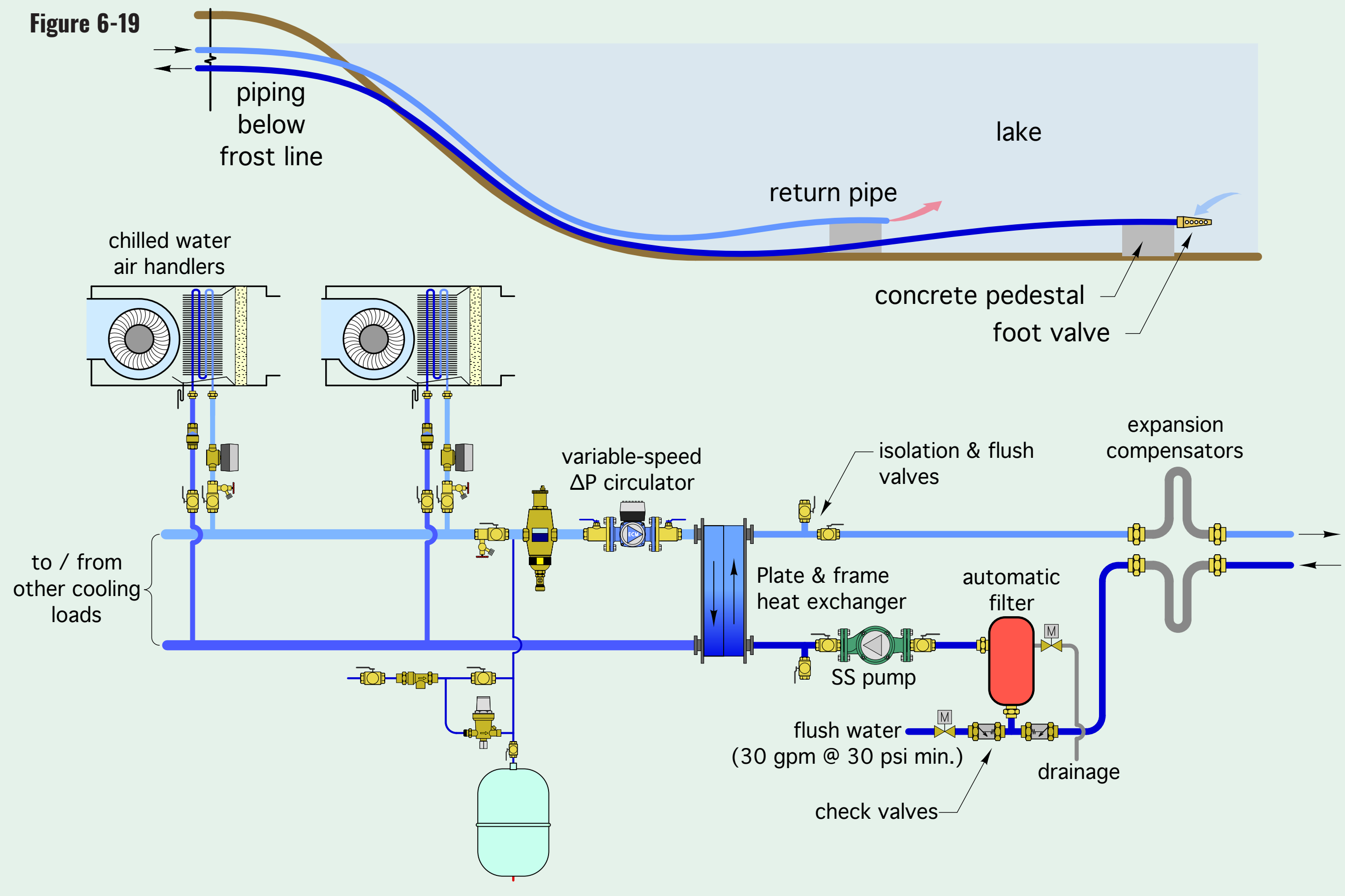
Second, the project must conform to any regulations regarding use of lake water. Different requirements may apply to “navigable waters” versus lakes in which boating is not allowed. Compliance checks should be done at the onset of a feasibility study to rule out possible restrictions that would prevent further consideration of this approach.

If regulations allow further pursuit of the project, water temperature measurements should be taken at several depths below the lake surface during late summer to assess available water temperatures during months when cooling is needed. This will help in determining the required depth of the water intake pipe.

Assuming the project reaches the design phase, Figure 6-19 shows one possible method for “harvesting” cool lake water for building cooling. This system uses readily available components.

A shallow well pump provides flow for the lake piping. This pump might be within the building being cooled or a separate building closer to the lake. If water remains in the system during winter, the pump and piping must be protected against freezing. Shallow well pumps are limited in their ability to lift water above the surface of the lake. If the lift requirement is not more than approximately 20 feet above the lake surface, and the piping to and

Figure 6-19



from the lake is sized for low head loss, a shallow well pump should suffice. The objective is to keep the pump as low as possible, relative to the lake surface.

Lake water is drawn into a foot valve at the end of a high-density polyethylene (HDPE) pipe. The end of this pipe and foot valve are secured to a concrete ballast block that prevents the pipe from floating. This ballast block also suspends the foot valve above silt on the lakebed. Depending on the length of pipe required, it may be necessary to use additional concrete ballast blocks to ensure the piping in the lake stays on, or close to, the lakebed. The piping must make the transition to shore at depths that prevent the water it contains from freezing in winter.

The incoming lake water passes into an automatic filter. This device monitors the differential pressure across its inlet and outlet ports. When the pressure differential reaches a set value, the filter initiates an automatic cleaning cycle. In this mode, a typical automatic filter needs a minimum flow of 30 gpm at 30 psi pressure to provide proper cleaning. This flow is provided from a separate water source routed through a pilot-operated solenoid valve that opens simultaneously with the waste valve on the filter. The accumulated sediment is washed off the internal stainless steel screen and blown out through a waste pipe. A filtering criterion of 100 microns is suggested to minimize any accumulation of sediment within the heat exchanger. The frequency of filter operation depends on the characteristics of the lake. If the intake pipe is located deep within a lake that has minimal surface water flow, the intake water should remain relatively clean. However, if the lake water experiences significant weather-related disturbances, the water may be turbid at times, and thus will require more filtering.

Water passes from the automatic filter to the pump. Ideally, this pump should be constructed with a stainless steel volute to provide maximum corrosion resistance. Full port ball valves should be installed to isolate the pump if maintenance is required. A priming line may also be required to add water to the intake pipe when the system is being put into operation.

From the pump, lake water passes into one side of a stainless steel plate & frame heat exchanger. The heat exchanger should be sized so that the approach temperature difference between the incoming lake water and the chilled water leaving the other side of the heat exchanger is not more than 5ºF under design cooling load conditions. Lower temperature rises across the heat exchanger may allow for smaller coils in air handlers, lower flow rates and lower circulator operating costs.

After passing through the heat exchanger, the lake water returns to the lake through another HDPE pipe. Like the supply pipe, the return pipe should be kept on or near the lakebed using concrete ballast blocks. It should discharge into the lake several feet away from the foot valve, and preferably in a direction that carries the water away from the foot valve.

The load side of the heat exchanger connects to a 2-pipe chilled-water distribution system. The ECM-based circulator operates on differential pressure control and changes speed in response to the opening and closing of the zone valves controlling flow to each air handler.

The electrical energy used by the lake water pump could be further reduced by using a variable-speed circulator that responds to either the temperature rise across the lake side of the heat exchanger or to the temperature of the lake water leaving the heat exchanger. Either would allow the flow rate of lake water to be adjusted based on the current cooling load on the other side of the heat exchanger.

The lake water circuit could also be used to supply low-temperature heat to a water-source heat pump for space heating or for domestic water heating.

Figure 6-20

Courtesy of AWEB Supply.

Figure 6-21

Courtesy of AWEB Supply.

LAKE PLATE HEAT EXCHANGERS

Another method for harvesting the cooling potential of lakes uses a closed plate-type heat exchanger, an example of which is shown in Figures 6-20 and 6-21.

The lake heat exchanger consists of multiple stainless steel plate assemblies. Each assembly has two stainless steel plates that are specially patterned to create flow channels. These plates are welded together along their perimeter. Each

Figure 6-22

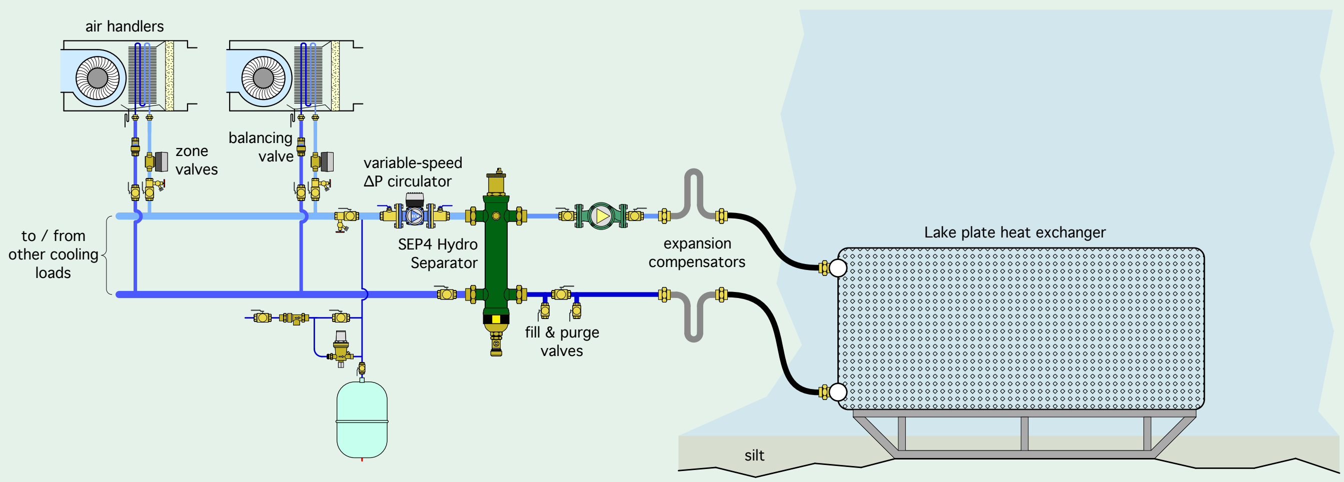


plate assembly is connected to a supply and return header. The overall assembly is welded to a stainless steel base that supports the plates several inches above the surface they rest on. HDPE tubing is routed from the headers to the shore.

This type of heat exchanger is typically used with water source heat pumps. However, when properly sized and used in a lake where the water temperatures at the lakebed are stable and low, it could provide direct heat exchange to a chilled-water cooling system, as shown in Figure 6-22.

This type of heat exchanger allows for a completely closed hydronic cooling system. This eliminates the need for the automatic blowdown filter, as well as the flat plate heat exchanger. However, a dirt-separating device designed for a closed hydronic circuit should still be installed to establish and maintain a very low level of suspended solids in the recirculating water. In Figure 6-22, a SEP4 hydraulic separator provides air, dirt and magnetic particle separation. It also allows for different flow rates between the lake heat exchanger and the chilled-water distribution system.

Manufacturers of lake water heat exchangers provide design assistance software that can be used to select a specific heat exchanger based on total cooling load, lake water temperature, flow rate through the heat exchanger and temperature change across the heat exchanger.

As with direct lake water cooling, designers should verify if any regulations prevent the placement of a lake plate heat exchanger, especially in navigable waters, as an initial step in the feasibility of this type of cooling system.

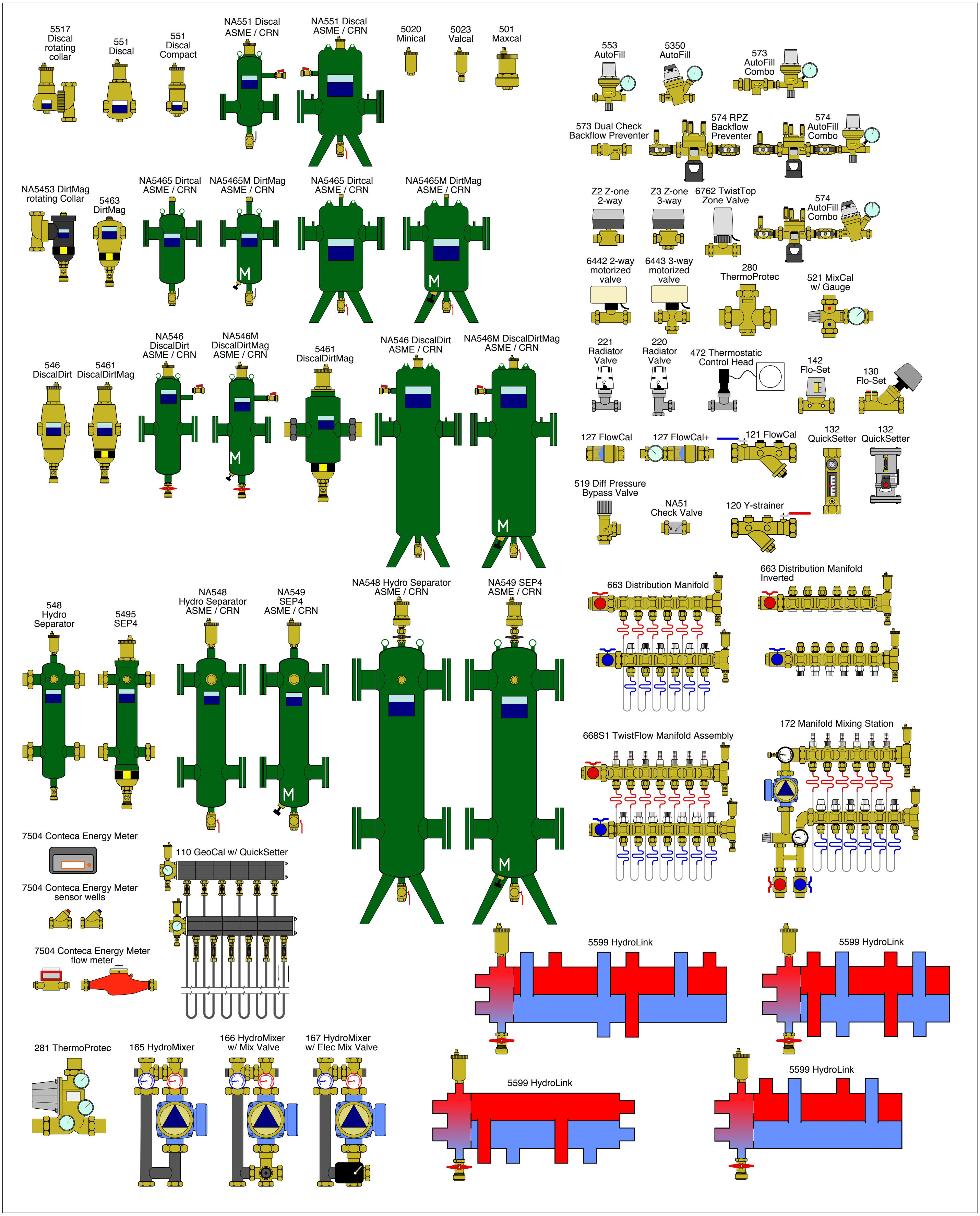
SUMMARY:

Heat exchangers add exceptional versatility to modern hydronic systems. They allow physical separation of fluids, but passage of heat, enabling applications such as domestic water heating, snowmelting and pool heating.

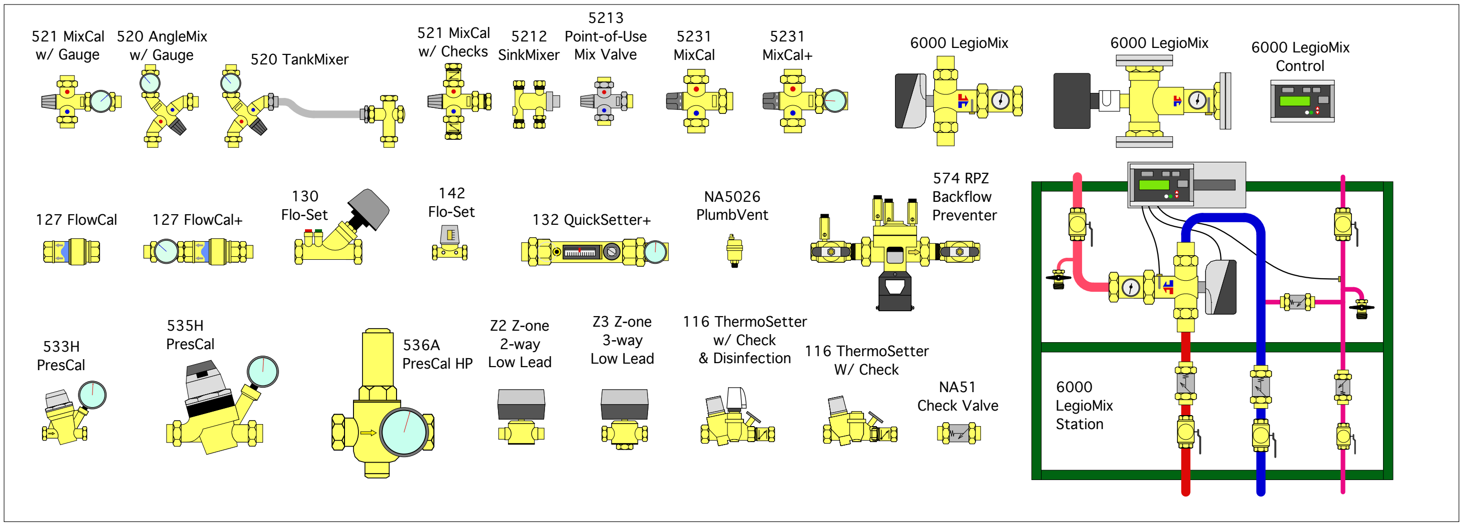
Section 4 gives the basis for theoretical performance assessment of heat exchangers. The associated mathematics can be challenging. Most practicing designers now use software available from several heat exchanger manufacturers to narrow the design down to an appropriate heat exchanger model.

To ensure reliable performance, heat exchangers need to be maintained with little if any fouling. The use of demineralized water as well as high-efficiency dirt separators is important.

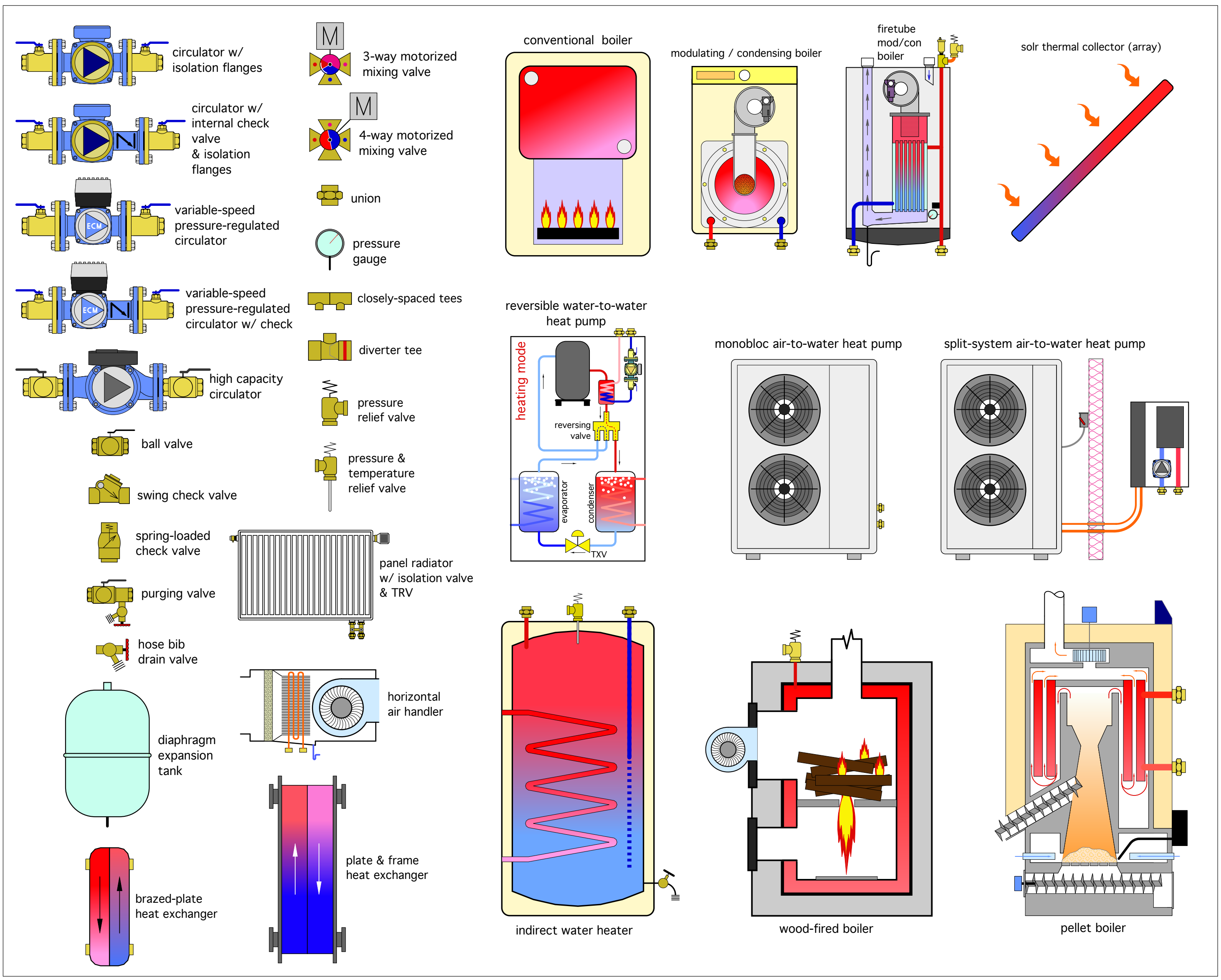
For more information on demineralized water, see *idronics* 18.


5517 Discal rotating collar
551 Discal
551 Discal Compact
NA551 Discal ASME / CRN
NA551 Discal ASME / CRN
5020 Minical
5023 Valcal
501 Maxcal
553 AutoFill
5350 AutoFill
573 AutoFill Combo
573 Dual Check Backflow Preventer
574 RPZ Backflow Preventer
574 AutoFill Combo
NA5453 DirtMag rotating Collar
5463 DirtMag
NA5465 Dirtcal ASME / CRN
NA5465M DirtMag ASME / CRN
NA5465 Dirtcal ASME / CRN
NA5465M DirtMag ASME / CRN
Z2 Z-one 2-way
Z3 Z-one 3-way
6762 TwistTop Zone Valve
574 AutoFill Combo
6442 2-way motorized valve
6443 3-way motorized valve
280 ThermoProtec
521 MixCal w/ Gauge
546 DiscalDirt
5461 DiscalDirtMag
NA546 DiscalDirt ASME / CRN
NA546M DiscalDirtMag ASME / CRN
5461 DiscalDirtMag
NA546 DiscalDirt ASME / CRN
NA546M DiscalDirtMag ASME / CRN
221 Radiator Valve
220 Radiator Valve
472 Thermostatic Control Head
142 Flo-Set
130 Flo-Set
127 FlowCal
127 FlowCal+
121 FlowCal
132 QuickSetter
132 QuickSetter
519 Diff Pressure Bypass Valve
NA51 Check Valve
120 Y-strainer
548 Hydro Separator
5495 SEP4
NA548 Hydro Separator ASME / CRN
NA549 SEP4 ASME / CRN
NA548 Hydro Separator ASME / CRN
NA549 SEP4 ASME / CRN
663 Distribution Manifold
663 Distribution Manifold Inverted
668S1 TwistFlow Manifold Assembly
172 Manifold Mixing Station
7504 Conteca Energy Meter
110 GeoCal w/ QuickSetter
7504 Conteca Energy Meter sensor wells
7504 Conteca Energy Meter flow meter
5599 HydroLink
5599 HydroLink
281 ThermoProtec
165 HydroMixer
166 HydroMixer w/ Mix Valve
167 HydroMixer w/ Elec Mix Valve
5599 HydroLink
5599 HydroLink

APPENDIX B: CALEFFI PLUMBING COMPONENTS



APPENDIX C: GENERIC COMPONENTS



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