july 2015
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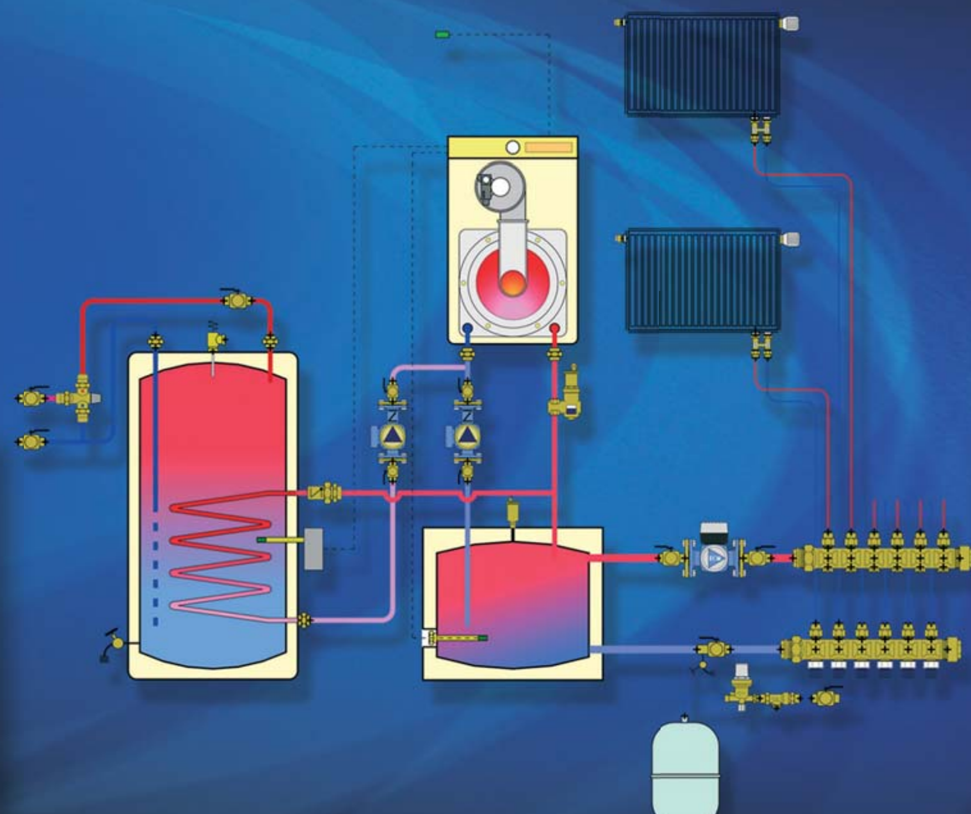
JOURNAL OF DESIGN INNOVATION FOR HYDRONIC PROFESSIONALS

Thermal Storage in Hydronic Systems



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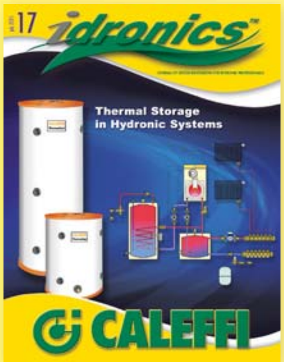

A Technical Journal
from
Caleffi Hydronic Solutions

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Printed: Milwaukee, Wisconsin USA

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Dear Hydronic and Plumbing Professional,

As a boy, I visited my grandfather's farm in northern Michigan. He had an old John Deere tractor with levers sticking out it from everywhere. He described it as "Older than the war." I often wondered what war he was referring to...

That tractor had a massive cast iron flywheel attached to its side. It would spin like crazy as the engine revved. "Smooth's out the ride Mark" he would say over thunderous exhaust noise. "Not so hard on the engine that way." As I watched the tractor's engine having to work only slightly harder as it climbed a small hill, I gained my first understanding of "the flywheel effect."

Years later, while on Caleffi customer visits, I've heard folks refer to a "thermal flywheel." I couldn't help from smiling knowing that they were describing a thermal storage tank, or what our industry often refers to as a buffer tank.

This issue of *idronics*™ addresses thermal storage in hydronic systems. Like the flywheel on that old John Deere engine, thermal storage can smoothen the load for many types of heat sources and chillers. And…. unlike that tractor, you don't have to worry about getting your sleeve caught in them!

Beyond the heat source or chiller, thermal storage can be used in many creative ways to stabilize system operation, store low cost or even free energy when it's available, and even prolong the life of system components

We hope you enjoy this 17th issue of *idronics*™ and encourage you to send us any feedback by e-mailing us at idronics@caleffi.com.

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Mark Olson

Mark Olson

General Manager & CEO

INDEX

Thermal Storage in Hydronic Systems

1. INTRODUCTION:

Water is one of the best substances on earth for storing heat. A given volume of water can absorb almost 3500 times more sensible heat than the same volume of air when both undergo the same temperature change.

The extraordinary ability of water to store heat makes it an ideal "mediator" for situations where the rate of heat production is different from the rate at which heat is needed by a thermal load.

A common example is a 30-gallon electric water heater. The heating element in most water heaters of this size is rated at 3.8 kW (12,969 Btu/hr) of heat production. Assuming cold water enters this heater at 45ºF, and that the desired hot-water delivery temperature is 110ºF, a 3.8 kW heat input rate could only maintain the desired supply water temperature at flow rates up to 0.4 gallons per minute (gpm). To put this in perspective, a low-flow shower head typically operates at 1.5 to 2.0 gpm. Thus, there is no way that the 3.8 kW heating element, by itself, could supply even one operating shower head. So why have millions of such heaters been successfully used in residential water-heating applications?

The answer lies in the 30 gallons of stored water. This water is heated to some upper temperature setting by the element, even when there is no concurrent demand for domestic hot water. This thermal storage allows the rate at which thermal energy is removed from the water heater to be very different from the rate at which the electric element adds heat to the tank. It also allows the on-time of the element to be different from the timing of hot-water demand.

Similar situations often occur in hydronic heating systems. The rate and timing of heat demand can be very different from the rate of heat production. The extent of this difference, along with the operating characteristics of the heat source and balance of the system, determine if the resulting operation of the heat source is acceptable.

A common *undesirable* effect that results from a mismatch between heat production and heat demand is "short cycling" of the heat source. This describes a situation in which the heat source only remains on for very short periods of time. The subsequent "off-cycle" of the heat source is also relatively short.

Short cycling reduces the thermal efficiency of most heat sources. It also increases emissions, increases maintenance and shortens equipment life.

The most common way to avoid short cycling is to provide adequate thermal mass within the heating system. Because of its excellent heat storage characteristics, adding water to the system is the best way to increase its thermal mass. The device that contains this water is often called a buffer tank, because it "buffers" (e.g., helps protect) the heat source against the undesirable effects of short cycling.

ANCILLARY BENEFITS OF THERMAL STORAGE:

Beyond the ability to eliminate short cycling and its undesirable consequences, thermal storage tanks can provide several other benefits when properly applied in specific hydronic systems. These benefits include:

• Reducing the size of the heat source relative to peak load.

• Providing thermal mass to buffer domestic water-heating loads.

• Allowing small and short duration heating loads to be supplied without necessarily operating the heat source.

• Providing hydraulic separation between circulators.

• Providing a temporary source of heat for the distribution system if the primary heat source is down for maintenance.

• The ability to store heat supplied using lower cost time-of-use electrical rates.

• The ability to store heat produced by renewable heat sources such as solar thermal collector or wood- and pellet-fired boilers.

• The ability to store surplus heat produced by a combined heat and power (CHP) system.

This issue of *idronics* explores many situations in which these benefits enhance the operation of hydronic heating and cooling systems. It describes how to properly size thermal storage tanks for several applications. It also shows specific piping methods that allow thermal storage tanks to perform to their full potential.

2. A HISTORICAL PERSPECTIVE ON THERMAL STORAGE IN HYDRONIC SYSTEMS

Many of the boilers used in North American hydronic systems through much of the 20th century, were constructed of steel and cast iron — *LOTS of steel and cast iron*.

Figure 2-1 shows a cast iron boiler block that was typical of technology used for small gas-fired and oil-fired boilers during this time, and in some cases is still used today. This is the core component of a cast iron boiler.

Even small cast iron and steel boilers often contain several hundred pounds of metal, and hold 10 to 30 gallons of water. These materials create what could be described as a "self-buffering" heat source. The thermal mass of these boilers is substantial, especially in comparison to modern wall-hung boilers.

For example, a cast iron boiler containing 400 pounds of cast iron and 15 gallons of water has approximately *30 times more* thermal mass than a compact mod/con boiler containing 10 pounds of stainless steel and one gallon of water. An example of the latter is shown in Figure 2-2.

Most hydronic distribution systems used in North American homes prior to the 1980's had one to perhaps three zones. This inherently limited how *small* the heating demand of a single zone could be, relative to the output capacity of the home's boiler.

The heat emitters used in older hydronic systems were usually cast iron radiators or cast iron baseboard. These heat emitters contain significantly more metal

Figure 2-1

Courtesy of Weil McLain

Figure 2-2

Courtesy of Harvey Youker

and water than more contemporary heat emitters, such as fin-tube baseboard, panel radiators or fan-coils.

Even the piping used in older hydronic systems often contained more metal and more water than the piping used in modern systems. This is especially true if the original piping was sized for a "gravity" type system (e.g., no circulator), or a system that was designed for steam heating and later converted to water heating.

The high thermal mass of older hydronic systems could inherently store the surplus heat production from the boiler that was not immediately needed by the load. This characteristic allowed most older hydronic systems to operate in an acceptably stable manner.

MASS EVOLUTION:

The availability of new materials, manufacturing processes, engineering methods and market demands have significantly changed how hydronic heating and cooling systems are currently built, compared to those used in the latter half of the 20th century.

From the generalized standpoints of thermal mass and zoning, the high thermal mass and minimally zoned systems of the past have evolved into much lower mass and more extensively zoned systems today. This transition has both good and bad consequences.

Low thermal mass has the advantage of fast thermal response. Low-mass heat sources combined with low-mass distribution systems and heat emitters can deliver heat to spaces relatively quickly following a call for heat from a thermostat. Modern low-mass systems also contain significantly less fluid, which reduces the size of their expansion tank, and decreases the amount of antifreeze required when freeze protection is needed.

However, low thermal mass systems lack the inherent thermal mass (e.g., heat storage ability) that stabilized older systems. Thus, when a heat source with a minimal heat production rate of 20,000 Btu/hr turns on to supply a small zone that only requires 3,000 Btu/hr, there is very little metal and water mass to absorb the 17,000 Btu/hr of surplus heat production. This allows the water temperature in the heat source to rise rapidly. It quickly reaches a setpoint temperature at which a controller turns off the heat source to prevent overheating. When not anticipated, or properly addressed during design, these operating characteristics guarantee undesirable short cycling.

Because short cycling was uncommon in older system, one might argue that the industry should return to the "heavy metal" approach for boiler construction. Perhaps we should also return to systems constructed of generously sized black iron pipe and cast iron radiators rather than PEX tubing and panel radiators? Maybe we should even consider building residential-scale heat pumps with multi-hundred pound steel condensers rather than coaxial heat exchangers that weigh 5 to 10 pounds and hold two quarts of water?

However, before drawing such conclusions, consider what a well-insulated thermal storage tank offers as an alternative to the "mass-by-metal" approach.

Water-based thermal storage provides these advantages:

• It allows heat to be stored in a container that is much better insulated than a boiler. Far less heat is lost through uncontrolled processes, such as from the boiler's jacket or by convective air currents up the chimney.

• It allows the boiler's heat exchanger to be small and light. This reduces cost and greatly improves response time. With proper controls, it also allows the boiler to quickly warm to temperatures that prevent sustained flue gas condensation.

• It allows other heat sources that may be present in the system or added in the future to interact with thermal storage.

• It allows for beneficial temperature stratification within the water.

• If necessary, it can store heated water during the heating season, as well as chilled water for summertime cooling.

• In many buildings, it's easier to move an empty tank into position and then fill it with water, compared to moving a boiler that weighs several hundred into the same space.

• A given *volume* of water can store approximately 16% more heat than the same volume of cast iron.

MOVING FORWARD:

Water-based thermal storage that is properly sized, placed and piped is an indispensable tool in modern hydronic heating and cooling design. It is a versatile design element that can be tailored to the unique operating characteristics of different heat sources, chillers and hydronic distribution systems. It is the "mediator" that enables the heat source and chillers to perform up to their full potential, while also allowing their associated distribution systems to deliver unsurpassed comfort.

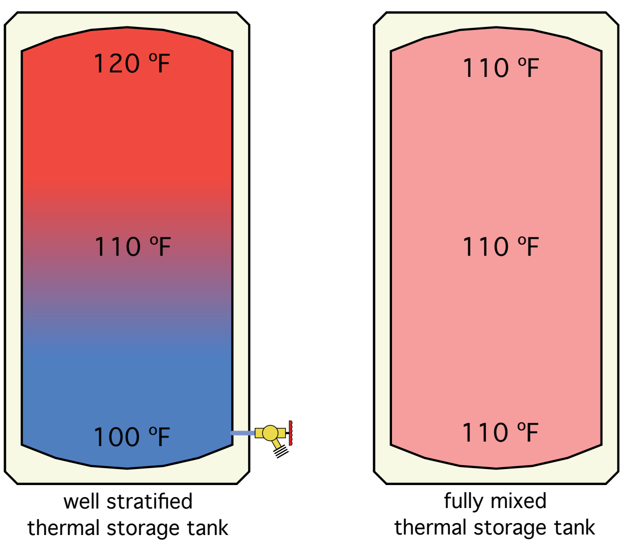
3. STRATIFICATION IN THERMAL STORAGE TANKS

The density of water changes significantly with temperature. When water is heated, it expands and its density decreases. When water is cooled, it contracts and its density increases.

When carefully introduced into the upper portion of a storage tank, hot water, with its lower density, tends to “float” above cooler water already in the tank. This phenomena is called temperature stratification, and it is a desirable effect in nearly all thermal storage applications.

Figure 3-1 shows two identical thermal storage tanks. Both are assumed to be used in a heating application. The tank on the left is stratified. The water temperature at the top is 120ºF. The water temperature at the bottom is 100ºF. Assuming a linear temperature gradient from top to bottom, the average water temperature in this tank is 110ºF.

Figure 3-1



The tank on the right is fully mixed. The water temperature at the top, middle and bottom of this tank is a uniform 110ºF.

Both tanks contain the same *amount* of heat. However, the temperature stratification present in the left tank gives it a “thermodynamic advantage.” To understand why, one needs to understand the thermodynamic principle of *exergy*, which is based on the 2nd law of thermodynamics.

EXERGY:

Exergy is a number that determines the maximum ability of the energy contained in a material to effect change in its surroundings. In practical terms, *exergy* can be thought of as a measure of the “usefulness” of the energy in a material, rather just the *amount* of energy in that material. *Energy that has higher exergy is always preferred because it is more useful.*

Although both tanks in Figure 3-1 contain the same *amount* of thermal energy, it can be shown that the *exergy* of the energy in the stratified tank is greater than the *exergy* of the fully mixed tank.

Here one simple rationale that supports that claim: The water in the upper half of the stratified tank has a temperature from just over 115ºF to 120ºF. This water could transfer heat to another material that has a temperature of 115ºF. However, *none* of the 110ºF water in the fully mixed tank could directly transfer heat to another material at 115ºF. Thus, the water in the stratified tank is more useful and provides greater potential for how the energy can be used to supply a load.

Whenever mixing occurs between warmer and cooler fluids, there is a loss of exergy.

One way to demonstrate this principle is to calculate the “equivalent temperature” (T_e) of a fully mixed tank that has the same *exergy* as a stratified tank. This can be done using Formula 3-1.

Formula 3-1:

$$T_e = 2\left(T_{mixed}\right) - e^{\left[\frac{1}{H}\int_{h=0}^{H} \ln[T(h)]dh\right]}$$

Where:

T_e = equivalent temperature of a fully mixed tank that would have the same exergy as a stratified tank (ºF)

T_{mixed}= Actual temperature of the fully mixed tank (ºF)

h = vertical position above bottom of tank (ft)

ln = natural logarithm

T(h) = a function that gives the temperature at some height (h) within the tank (ºF)

e = 2.718281828

To evaluate this formula, the temperature profile from the bottom to the top of the tank needs to be expressed as a function of tank height (h).

For example, consider a thermal storage tank that is 6 feet tall. Assume that the water temperature at the bottom of the tank is 100ºF, and the water temperature at the top is 120ºF. If a linear temperature profile is assumed from bottom to top, the water temperature as a function of tank height can be expressed as:

$$T(h) = 100 + \left[\frac{20}{6}\right]h$$

If this function is substituted into Formula 3-1 and evaluated, the equivalent temperature (T_e) can be calculated as 110.15ºF. This is slightly greater than the actual mixed tank temperature of 110ºF. Since the equivalent temperature (T_e) is higher than the mixed temperature, there would be a loss of *exergy* if mixing occurs.

It can be mathematically proven that this will always be the case for any possible temperature profile established by stratification within the tank. The bottom line: *Mixing should always be avoided (or minimized) to preserve the usefulness of the heat contained in a thermal storage tank.*

FACTORS INFLUENCING TEMPERATURE STRATIFICATION:

Many factors affect the degree to which temperature stratification exists in a storage tank. These include:

- The temperature of water being added to the tank
- The position and orientation of the piping inlets to the tank
- The presence of specific devices called inlet flow diffusers within the tank
- The thermal conductivity of the tank walls
- The insulation on the tank
- The timing and rate at which water is added and removed from the tank

Thermal storage tanks used in hydronic systems should be designed and operated to encourage thermal stratification. The following guidelines help in this regard:

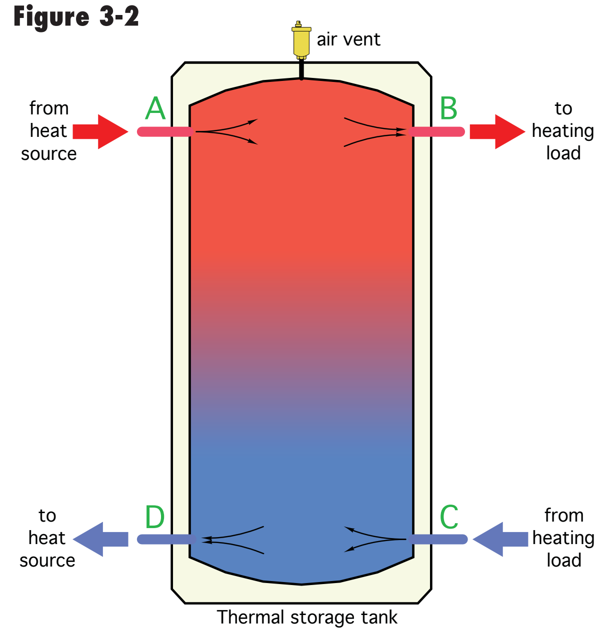
- Use vertically oriented rather than horizontally oriented tanks.
- Introduce heated water into the upper portion of a thermal storage tank.
- Extract cooler water from the lower portion of the tank.
- Provide piping connections, and possibly internal details, that allow incoming flow to enter at low velocities and in a horizontal direction.
- Avoid piping connections that create vertical flow jets within the tank.
- Use good insulation on the tank (R-12 ºF•hr•ft^2/Btu suggested minimum)
- When possible, use tanks built of materials with low thermal conductivity

4-PIPE VERSUS 2-PIPE TANK CONFIGURATIONS:

Figure 3-2 shows one tank piping arrangement that embodies many of the stratification-encouraging details just mentioned.

This is called a “4-pipe” configuration based on the 4 primary connections that connect the tank to the heat source (at points A and D) and to the load (at points B and C).

Figure 3-2

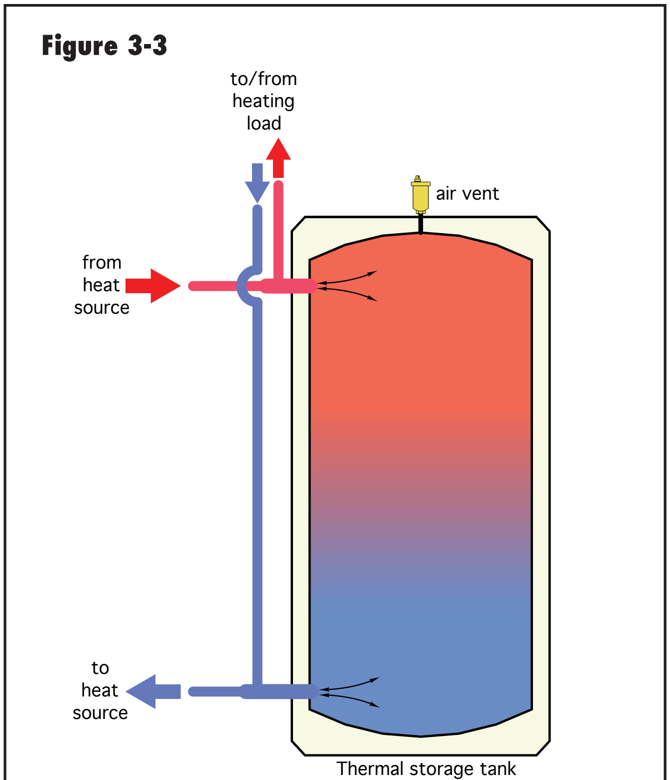


4-pipe configuration

This configuration allows heated water entering the tank at point (A) to remain in the upper portion of the tank due to its lower density in comparison to the density of cooler water lower in the tank.

Still, there is some mixing between the entering water and the water already in the upper portion of the tank. This will create a “delay” between the time that hot water enters the tank at point (A) and water *at that same temperature* leaves the tank for the load at point (B).

For systems that maintain a relatively consistent load, this is generally not a problem. However, for systems that need to transfer heat to the load quickly during a boiler startup, or following a recovery from a significant temperature setback, any interaction with the thermal



2-pipe configuration

mass of the tank is undesirable. In such situations, the "2-pipe" tank configuration shown in Figure 3-3 provides advantages.

In a 2-pipe configuration, some of the flow from the heat source *may* be diverted to the load before reaching the thermal storage tank. This situation, when it occurs, provides several benefits:

• When necessary, it allows flow from the heat source to pass directly to the load without first "interacting" with the thermal mass in the upper portion of the tank.

• When there is flow through the heat source and flow to the load, the flow rate entering the upper portion of the thermal storage tank is reduced. This is illustrated in Figure 3-4 for a heat source flow rate of 10 gpm and a corresponding load flow rate of 8 gpm.

With the 2-pipe configuration, the flow rate entering the upper portion of the tank is 2 gpm, as is the downward flow rate through the tank. However, if the same tank were set up with a 4-pipe configuration, the flow rate entering the tank is 10 gpm, which divides up between 8 gpm flowing across the tank toward the upper right outlet, and 2 gpm flowing vertically downward.

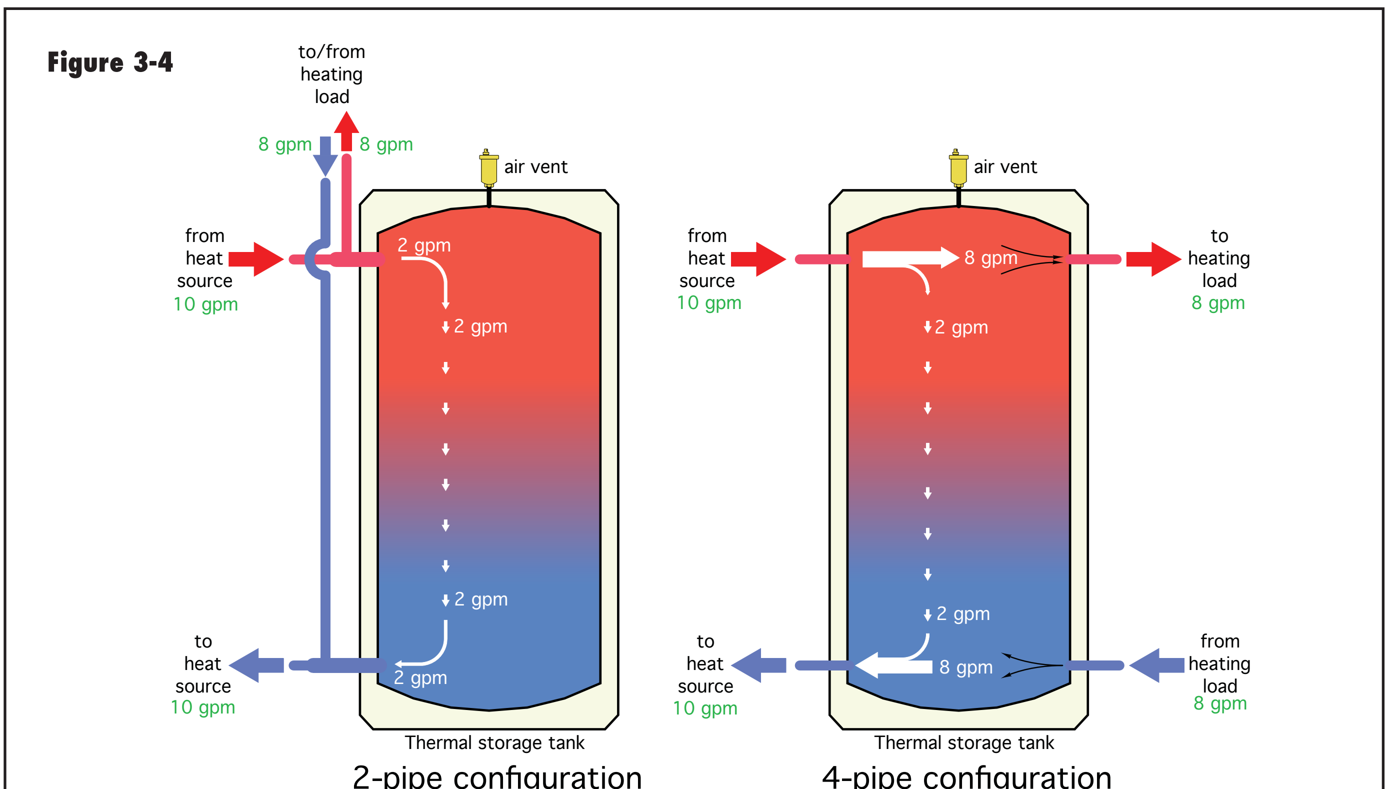


Figure 3-5

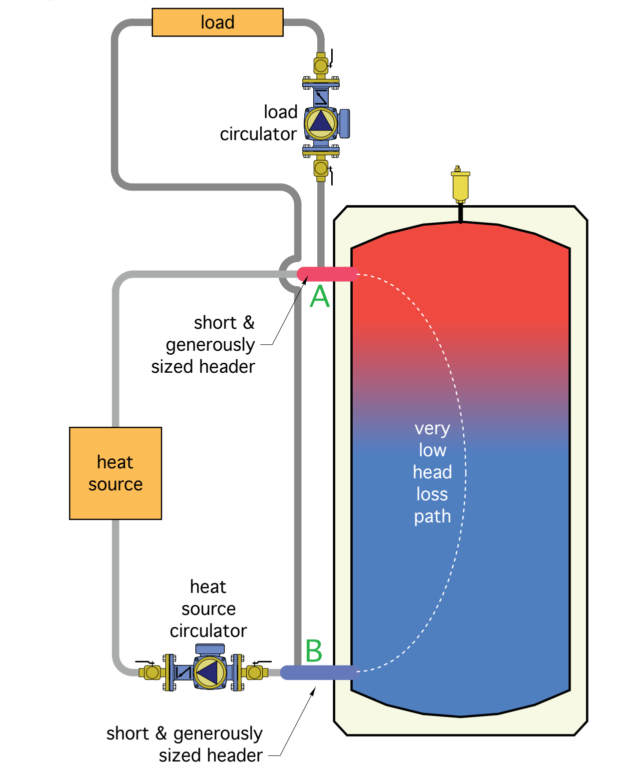
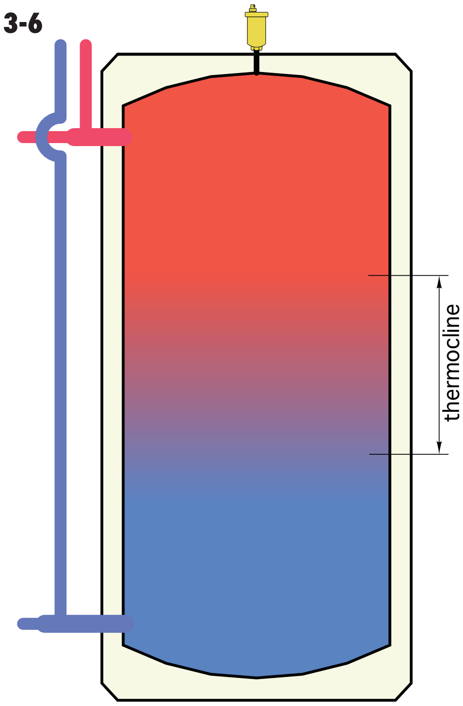


Figure 3-6



The higher the flow rate entering the tank, the higher the flow velocity, and thus the greater the mixing action within the tank. Mixing tends to degrade temperature stratification and should be avoided. Thus, a 2-pipe configuration under this flow scenario is preferred from the standpoint of preserving temperature stratification.

A 2-pipe tank configuration has the potential to slightly reduce installation cost due to having less piping connections. However, tanks with additional connections, such as would be used for a 4-pipe configuration, are still potentially useful for connecting other loads or measuring hardware such as temperature sensors, thermometers, pressure gauges and sight gauges. Thus, tanks with more connections are generally preferred based on the flexibility they provide to accommodate both piping and measuring hardware. If tanks with additional connections are used for a 2-pipe configuration, any unused connections can be easily plugged.

Thermal storage tanks set up as either 4-pipe or 2-pipe configurations can provide excellent hydraulic separation between circulators used for heat source(s) or loads.

When using a 2-pipe configuration, it is important to keep the tees connecting to the load piping as close to the tank as possible. The header piping between these tees and the tank should also be generously sized to reduce head loss to almost zero. Thus, the piping that the heat source circuit and the load circuit share in common has very low head loss, as illustrated in Figure 3-5. This ensures excellent hydraulic separation.

THERMOCLINE MOVEMENT:

Consider a well-stratified tank "at rest" (e.g., without flow entering or leaving). Under these conditions, a transition region called a thermocline forms between the hotter water at the top and the cooler fluid at the bottom, as illustrated in Figure 3-6.

The depth of the thermocline will vary based on how heat was previously added or removed from the tank, as well as the thermal conductivity of the tank walls and the insulation system used on the tank. Temperature gradients over 50ºF are possible under some circumstances.

When heated water having a temperature equal to or above the temperature in the upper portion of the tank enters the upper tank header, the thermocline moves downward. This occurs whenever the flow rate from the heat source is greater than the flow rate to the load. When the flow rate to the load exceeds the flow rate from the heat source, the thermocline moves upward. These two situations are illustrated in Figure 3-7.

Figure 3-7

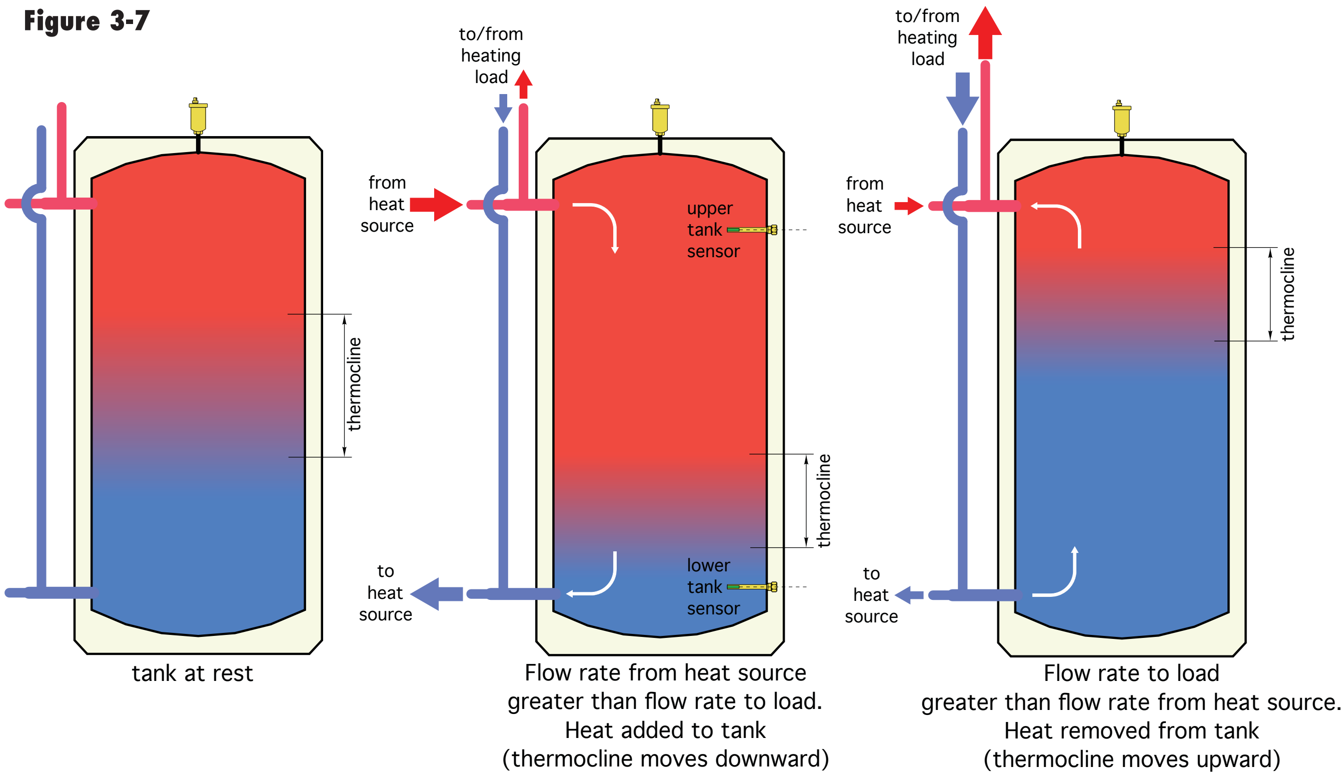
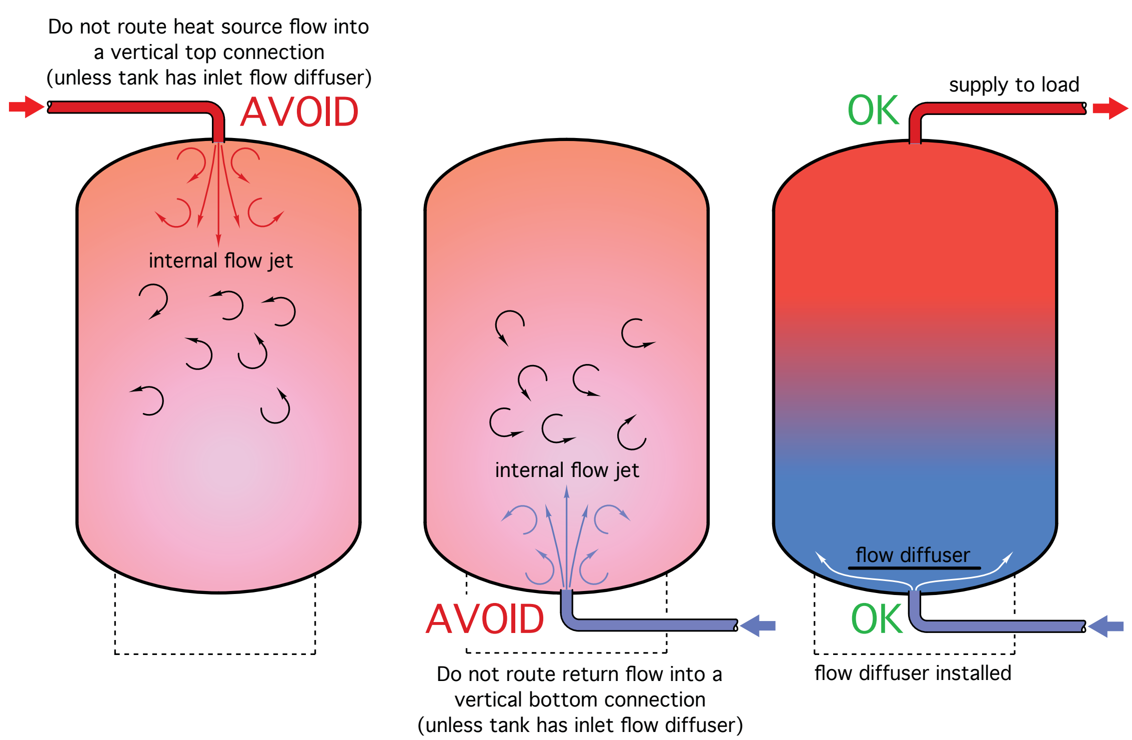


Figure 3-8



The vertical movement of the thermocline can be used to turn heat sources on or off. This requires coordination with sensor placement in the tank. This technique, which is discussed in more detail later, is useful in matching the response of the buffer tank to the warm up time required for slow-responding heat sources such as boilers fueled by pellets or wood chips.

A SITUATION TO AVOID (FLOW DISRUPTING STRATIFICATION):

Some pressurized tanks are made with semi-elliptical heads joined to cylindrical shells. When this type of tank is installed vertically, it is often equipped with a top and bottom center connection.

If flow from the return side of the distribution system is routed to the bottom center connection, a vertical flow jet will be established that causes mixing within the tank. Experience has shown that this can reduce, or even destroy temperature stratification.

This type of connection should be avoided *unless the tank is equipped with an internal flow diffuser* that can reorient the flow from vertical to horizontal and slow the velocity at which this flow mixes with water in the lower portion of the tank. Figure 3-8 illustrates piping details to avoid, and one concept for a flow diffuser that significantly reduces internal mixing.

USING BUFFER TANKS IN CHILLED-WATER COOLING SYSTEMS:

Just as hydronic heat sources can short cycle when supplying highly zoned distribution systems, so can chillers connected to highly zoned chilled-water distribution systems. Incorporating a chilled-water buffer tank in the system allows the chiller to operate for longer cycles, where it attains better efficiency.

Figure 3-9 shows an example of a chilled-water system with a thermal storage tank piped in a 2-pipe configuration.

Figure 3-9

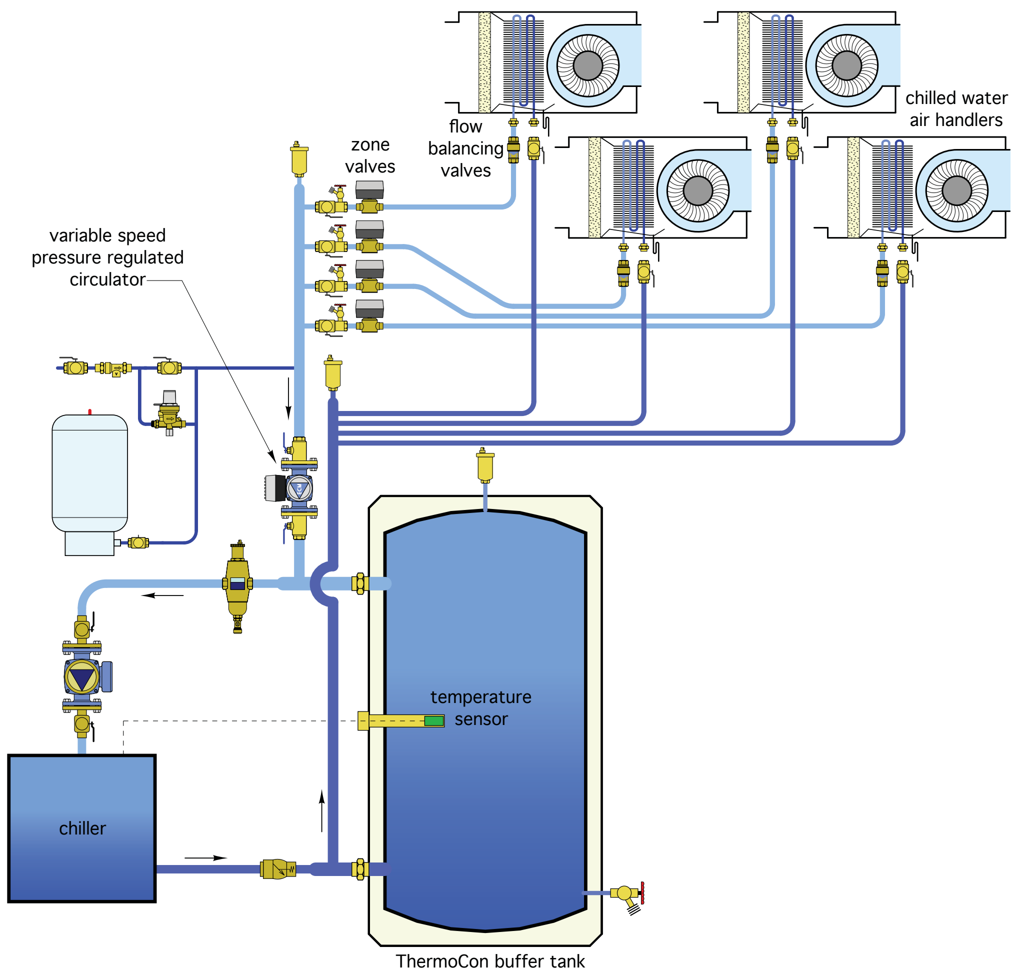
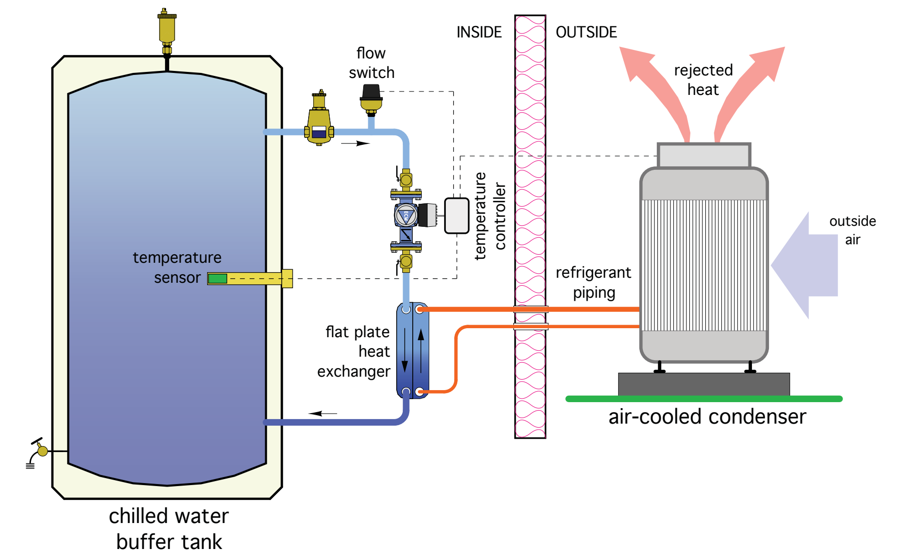


Figure 3-10



Cooling is supplied by a distribution system containing 4 independently controlled air handlers.

Flow from the chiller passes into the lower side connection of the thermal storage tank. This is the *opposite* of how piping from a heat source would connect to the tank. It allows the flow direction to "coordinate" with the temperature stratification that establishes itself in the tank. The coldest water will be at the bottom, and the warmest water at the top.

Thermal storage tanks that are used to store chilled water must be insulated and sealed against moisture diffusion through the insulation. Only tanks with monolithic (e.g., continuous) foam insulation are recommended for chilled-water storage. Furthermore, all piping connections to chilled-water storage tanks should be carefully detailed to ensure that both the insulation and any vapor barrier over the insulation are continuous from the piping to the tank. The vapor barrier on the piping should be carefully sealed to the vapor barrier on the tank. Any seams in the tank's jacket should be sealed with aluminum foil tape or other vapor-impermeable material.

Some chilled-water cooling systems use a refrigerant-to-water heat exchanger between the refrigeration line set of the chiller and the water. An example of such a system is shown in Figure 3-10.

In these systems, it's very import to include a flow switch in the water side piping of the heat exchanger. This switch detects when the water flow reaches and sustains a rate that is sufficient to transfer heat to the refrigerant in the heat exchanger without causing ice crystals to form. If the flow rate is not sufficient, the flow switch is wired to interrupt operation of the chiller to prevent freezing water within the heat exchanger.

BI-MODAL BUFFER TANKS VERSUS DUAL BUFFER TANKS:

Many hydronic systems supplied by heat pumps can provide heating and cooling.

When both the heating and cooling distribution systems are zoned, a buffer tank can be used to prevent the heat pump from short cycling in either heating or cooling mode.

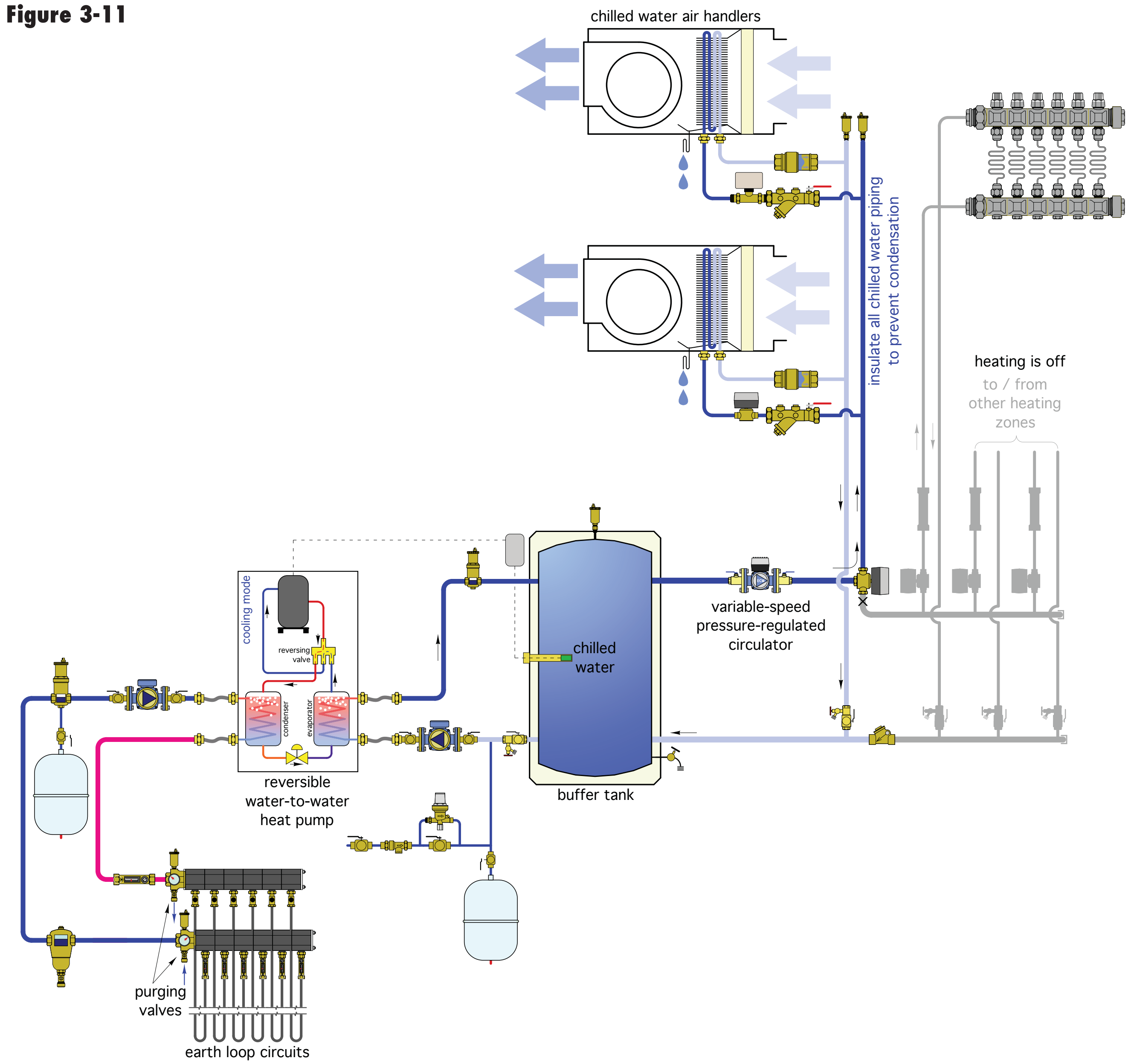
In climates where there is a "dead band" time of at least two weeks between the end of the heating season and the beginning of the cooling season, and vice versa, and for buildings that do not need simultaneous heating and cooling, it's possible to use the same thermal storage tank to store heated water during the heating season and chilled water for warm weather cooling. An example of a system using this type of "bi-modal" buffer tank, and operating in its cooling mode, is shown in Figure 3-11.

A water-to-water heat pump connected to a closed earth loop supplies heated water to the buffer tank for heating and chilled water for cooling. Both the heating and cooling distribution subsystems are zoned. A single variable-speed pressure-regulated circulator provides flow to both of these distribution subsystems. A motorized diverter valve determines the routing of water coming from this circulator.

Notice that the buffer tank is piped to enhance stratification during *heating* mode. When operating in cooling mode, the water in this tank will mix due to the coolest water from the heat pump entering at the top of the tank. The load side piping connections are also optimized for heating mode operation.

Such piping is often deemed acceptable in applications where heating season operation is highly dominant over cooling season operation. However, if the heating and cooling operating hours are more evenly divided, it is

Figure 3-11

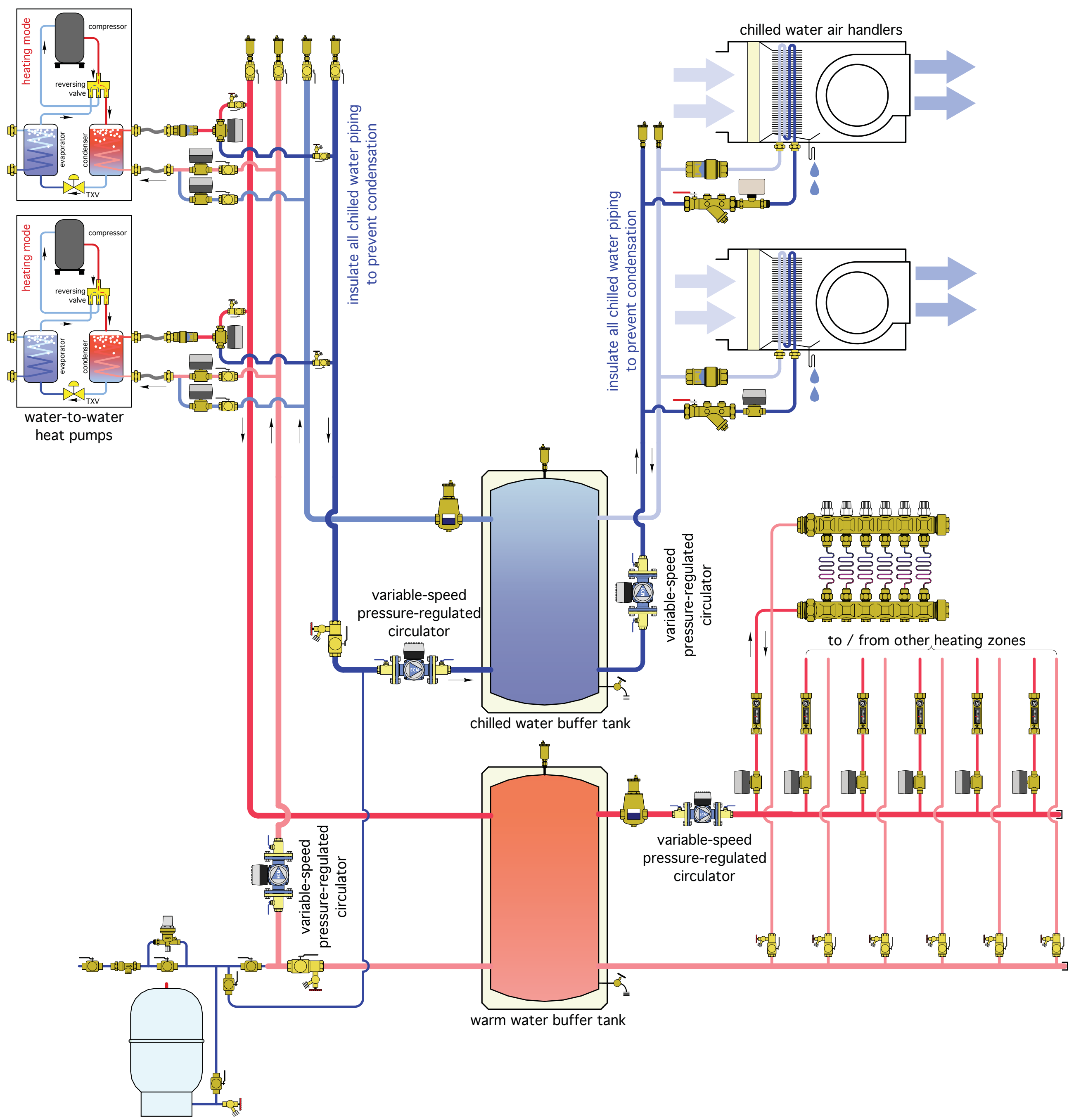


possible to reverse the flow directions through the tank to optimize stratification in both modes. Several details for doing so are presented in section 7.

Some hydronic systems need to provide *simultaneous* heating and cooling. This requirement is more common in larger buildings with core cooling loads that exist throughout the year. When the heating and cooling distribution systems are zoned, and capacity control is not low enough to ensure that short cycling will not occur, separate tanks for storing heated and chilled water are often specified. Figure 3-12 shows an example of such a system.

This system use two independently controlled water-to-water heat pumps for creating heated water and chilled water. Motorized valves are configured so that either heat pump can connect to the heated-water risers or the chilled-water risers. Both sets of risers lead back to their associated buffer tanks. Each buffer tank uses a 4-pipe configuration and provides excellent hydraulic separation

Figure 3-12



between the variable-speed circulators that drive flow through the heat pumps, and those that drive flow through the distribution systems.

The chilled-water buffer tank is piped so that the coldest water enters and leaves the lower connections. The warm-water buffer tank receives the hottest water at its upper left connection, and releases it to the load at its upper right connection. These piping details enhance stratification in both tanks.

The same type of piping could be used with multiple air-to-water heat pumps or a combination of water-to-water and air-to-water heat pumps.

4. AMOUNT OF HEAT IN A THERMAL STORAGE TANK

The amount of heat that any thermal storage tank can hold is determined by its volume, along with the temperature change of the fluid within the tank. Formula 4-1 can be used to calculate the heat added to or removed from a thermal storage tank filled with water:

Formula 4-1

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

Where:
Q = Amount of heat added or removed from tank (Btu)
v = volume of tank (gallons)
ΔT = temperature change of water in tank (ºF).
8.33 = a quasi-constant for water (Btu/gal/ºF)

****

For example, assume a fully mixed 500-gallon thermal storage tank is heated from a *uniform* initial water temperature of 60ºF to a *uniform* final temperature of 140ºF. How much heat was added to the tank in this process?

This is a very straightforward calculation:

$$Q = 8.33v(\Delta T) = 8.33(500)(140 - 60) = 333{,}200\,Btu$$

This calculation assumes that *all* water in the tank is 60ºF when heat input begins. It also assumes that *all* water in the tank is 140ºF when heat input stops. Although the starting assumption of a uniform 60ºF temperature is reasonable for cold water, a well-designed and properly piped thermal storage tank would develop temperature stratification as heat is added. Thus, it's likely that the water temperature within the tank will *not* be uniform from top to bottom when heat input stops.

If the upper portion of the tank was at 145ºF, and the lower portion at 135ºF, and a linear temperature profile from top to bottom was assumed, the calculation for heat added would be relatively accurate. However, if someone were to measure the water temperature at the top of the tank at 145ºF, and base the calculation of Formula 4-1 on that temperature, that calculation would overestimate the total heat added to the tank.

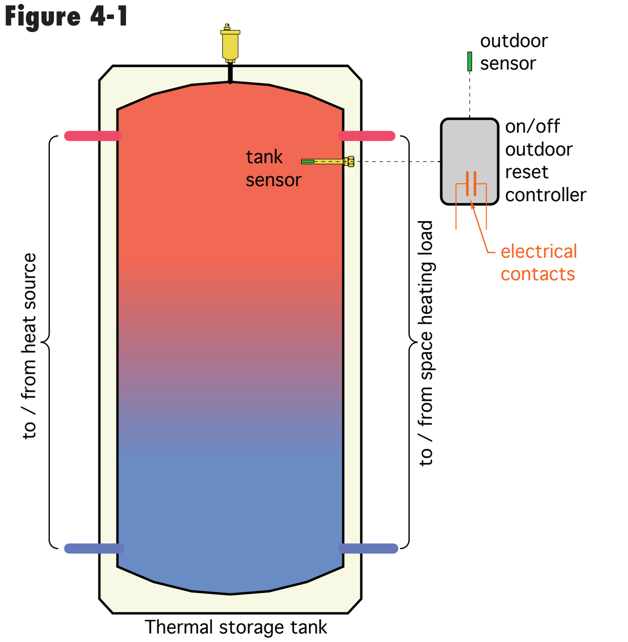
USABLE HEAT:

Although Formula 4-1 is thermodynamically correct, it implies that the tank is either fully mixed to a uniform temperature, or that reasonably accurate average temperatures are used for the calculation.

The amount of heat that is *usable* by a space-heating distribution system is determined by the average water temperature of the storage tank at the start of the heat extraction cycle *and the lowest temperature at which the tank can still supply heat to the load.*

The latter criteria is often determined by an outdoor reset controller that monitors the outdoor temperature and uses it, along with its settings, to calculate the minimum supply water temperature (often called the "target" temperature) that can satisfy the heating load. This controller also monitors the water temperature in the upper portion of the storage tank, as shown in Figure 4-1.

Figure 4-1



The outdoor reset controller compares the storage tank temperature to the calculated target supply water temperature. If the tank temperature is equal to or above the calculated target water temperature, the tank is deemed suitable to supply heat to the load. If the tank temperature is slightly below the calculated target temperature, a set of electrical contacts in the reset controller close. This closure is often used to signal other devices in the system that the tank can no longer meet the heating load, and thus heat from another source is required.

An on/off outdoor reset controller must operate with a differential (ΔT) between the tank sensor temperature at which its electrical contacts open and when they close.

This differential is usually adjustable and "centered" on the target temperature. The contacts open at the target temperature plus half the differential, and they close at the target temperature minus half the differential. This logic is shown below:

$$\text{If: } T_{\text{tank}} \geq T_{\text{target}} + \left(\frac{1}{2}\right)\Delta T \quad \text{Then: Tank is heat source}$$

$$\text{If: } T_{\text{tank}} \leq T_{\text{target}} - \left(\frac{1}{2}\right)\Delta T \quad \text{Then: Auxiliary heat is needed}$$

For example, if the calculated target temperature was 120ºF, and the controller's differential was set to 6ºF, the contacts would open at 123ºF and close at 117ºF.

Consider a hydronic distribution system that can deliver *design* heating load when supplied with 120ºF water, and operating with a temperature drop of 20ºF from supply to return. The design load is based on an outdoor temperature of 0ºF and an indoor temperature of 70ºF.

Figure 4-2 shows the theoretical supply and return water temperatures for this hydronic system as a function of outdoor temperature, *assuming the supply water temperature was maintained exactly at its calculated "target" value by an outdoor reset controller, and that the flow rate through the distribution system is constant.* This graph also assumes that the desired indoor air temperature is 70ºF, and that there are no internal heat gains in the building.

In this case, the temperature drop from supply to return is 20ºF under design load, but decreases proportionally as the outdoor temperature increases. Assuming the flow rate through the system remains constant, a decreasing temperature drop is the only possible way that heat output can decrease. When the outdoor temperature reaches 70ºF, both the supply and return water temperatures are also 70ºF. Thus, no heat would be released into the 70ºF building space.

An "ideal" on/off outdoor reset controller (e.g., one that could operate with 0 differential) would produce the target supply water temperature represented by the red line in Figure 4-2. However, any "real" on/off outdoor reset controllers must operate with a non-zero temperature differential between where its contacts close and open. If this type of controller is used to determine when the thermal storage tank temperature is adequate to supply the space-heating load, that differential should be kept relatively small.

Figure 4-3 shows a 5ºF differential superimposed on the target supply temperature line from Figure 4-2. The lower (**green**) dashed line shows the temperature at which the reset controller will turn on the auxiliary heat source.

Figure 4-2

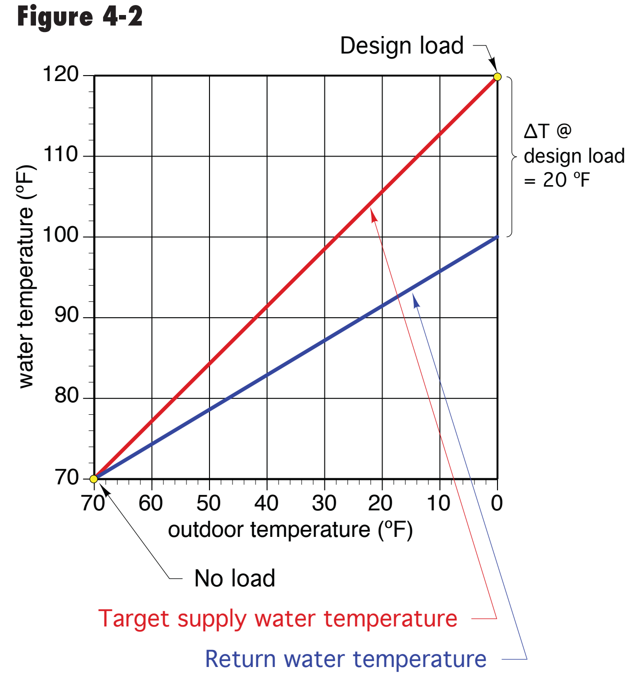
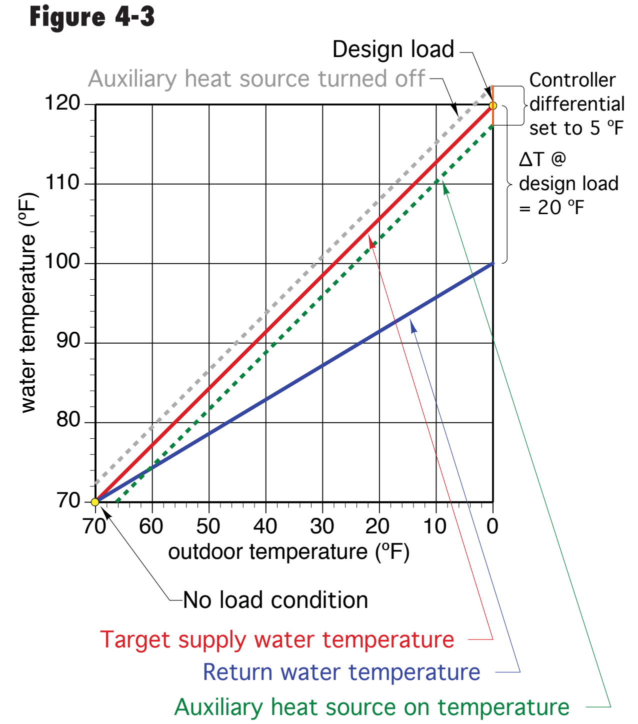


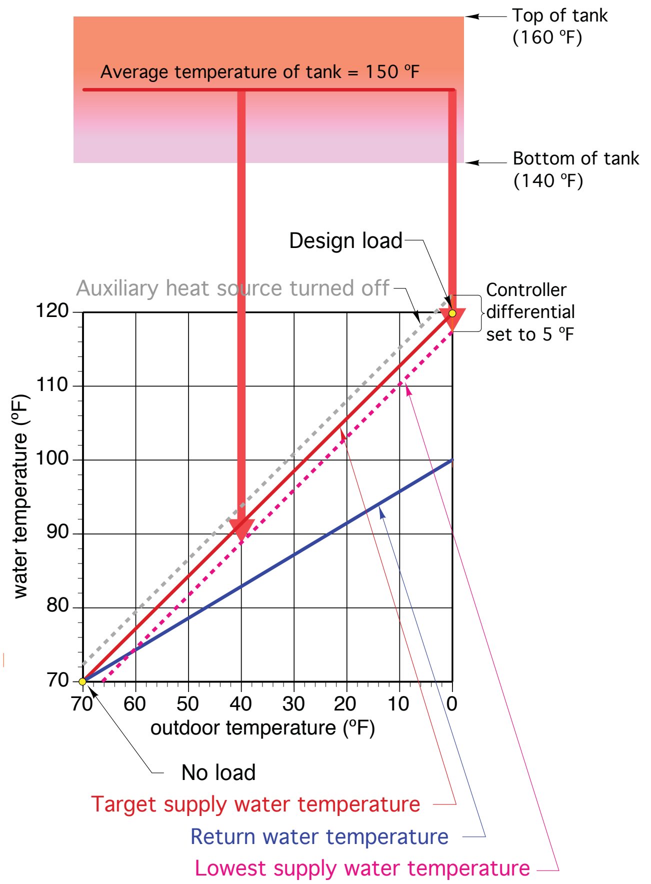
Figure 4-3



Under design load conditions, the reset controller allows the water temperature to drop to 117.5ºF (e.g., one half the 5ºF differential below the target temperature) before it "deems" the tank unable to supply the space-heating load. This lower supply water temperature will slightly decrease heat output from the distribution system. However, when the auxiliary heat source is turned on by the reset controller, it would remain on until the supply water temperature reached 122.5ºF (e.g., one half the 5ºF differential above the target temperature). This would slightly increase heat output from the distribution system. Over a period of operation, these two effects tend to compensate for each other and keep the total heat output from the distribution system the same as if it were supplied with a constant 120ºF water temperature.

These slight deviations above and below the target supply water temperature are also "smoothed out" by the thermal mass of the distribution system. It's very unlikely that building occupants will even notice these subtle changes in heat output, especially in comparison to standard on/off heat delivery systems.

Figure 4-4



In systems that use an on/off outdoor reset controller to determine if the thermal storage tank can supply the space-heating load, the **usable temperature drop** of the tank can be represented by Figure 4-4.

The heat extracted from the tank is represented by the large vertical arrows. The longer the arrow, the greater the amount of heat extracted from the tank.

This illustration is based on the assumption that the tank is at an *average* temperature of 150ºF at the start of the heat extraction period. Thus, if the temperature at the top of the tank was 160ºF, and the temperature at the bottom was 140ºF, with a linear temperature gradient from top to bottom, the initial amount of heat in the tank would be the same as if the entire tank was at 150ºF.

This graph also assumes that the distribution system requires a supply water temperature of 120ºF at design load conditions, and that the differential setting of the outdoor reset controller is 5ºF.

Heat is available from storage until the temperature at the upper tank sensor location drops 2.5ºF below the target temperature. At *design load conditions,* this will be 117.5ºF. However, at an outdoor temperature of 40ºF, the target temperature required by the distribution system is significantly lower, and heat can be delivered from storage to the distribution system until the temperature in the upper portion of the tank drops to about 88ºF. The greater the usable temperature drop of the thermal storage tank, the greater the amount of heat that can be extracted from it and used to satisfy the heating load.

This illustrates the advantage of using an outdoor reset controller to make the determination of when auxiliary heating is needed. It also demonstrates the advantage of designing the distribution system for low supply water temperatures. A suggested maximum value for the supply water temperature required by a hydronic distribution system under design load conditions is 120ºF. Even lower supply water temperatures are preferred when achievable through large surface area heat emitters or other means.

In some systems, the temperature cycling range of a thermal storage tank can be further extended based on comparing the temperature in the upper portion of the tank with the temperature *returning* from the distribution system. This will be discussed in section 7.

5. THERMAL STORAGE TANK OPTIONS

There are many types of thermal storage tanks available. They range in size from very small volumes of 10 gallons or less to several thousand gallons.

In addition to their volume, thermal storage tanks can be classified as:

- Unpressurized tanks
- Pressurized tanks

This section discusses both type of tanks, including their strengths and limitations. Later sections show how they can be applied.

UNPRESSURIZED THERMAL STORAGE TANKS:

An unpressurized tank is any tank that has a direct connection to the atmosphere, even through a small tube. This connection prevents the tank from sustaining any pressure other than the static pressure of water against the sides and bottom of the tank. It also prevents any negative pressure from developing within the tank as water is removed.

Because of their connection to the atmosphere, unpressurized tanks are also described as "open" from the standpoint of hydronic system design. As such, they are more limited in how they can be applied.

Only materials resistant to corrosion from oxygen should be used in any piping circuits connected to open tanks, or other open devices within the system. Materials such as copper, brass, bronze, stainless steel and PEX are generally acceptable. However, *components made of ferrous metals, such as cast iron or carbon steel, should not be used in piping circuits connected to open devices.* Doing so will lead to rapid corrosion due to the constant availability of oxygen to "fuel" the corrosion reaction.

Due to their final installed size, many of the larger unpressurized thermal storage tanks used in hydronic systems are shipped disassembled. This allows the tank's components to pass through standard passage doors, which is often a necessity, especially in existing buildings. Figure 5-1 shows one example of an unpressurized thermal storage tank that is shipped in panels and assembled onsite. Once assembled, this tank can hold several hundred gallons of water.

Most large unpressurized thermal storage tanks consist of a structural shell that also provides insulation, along with a flexible waterproof liner that fits inside the assembled shell. The liners are made of EPDM rubber, polypropylene,

Figure 5-1a

Figure 5-1b

Courtesy of American Solartechnics

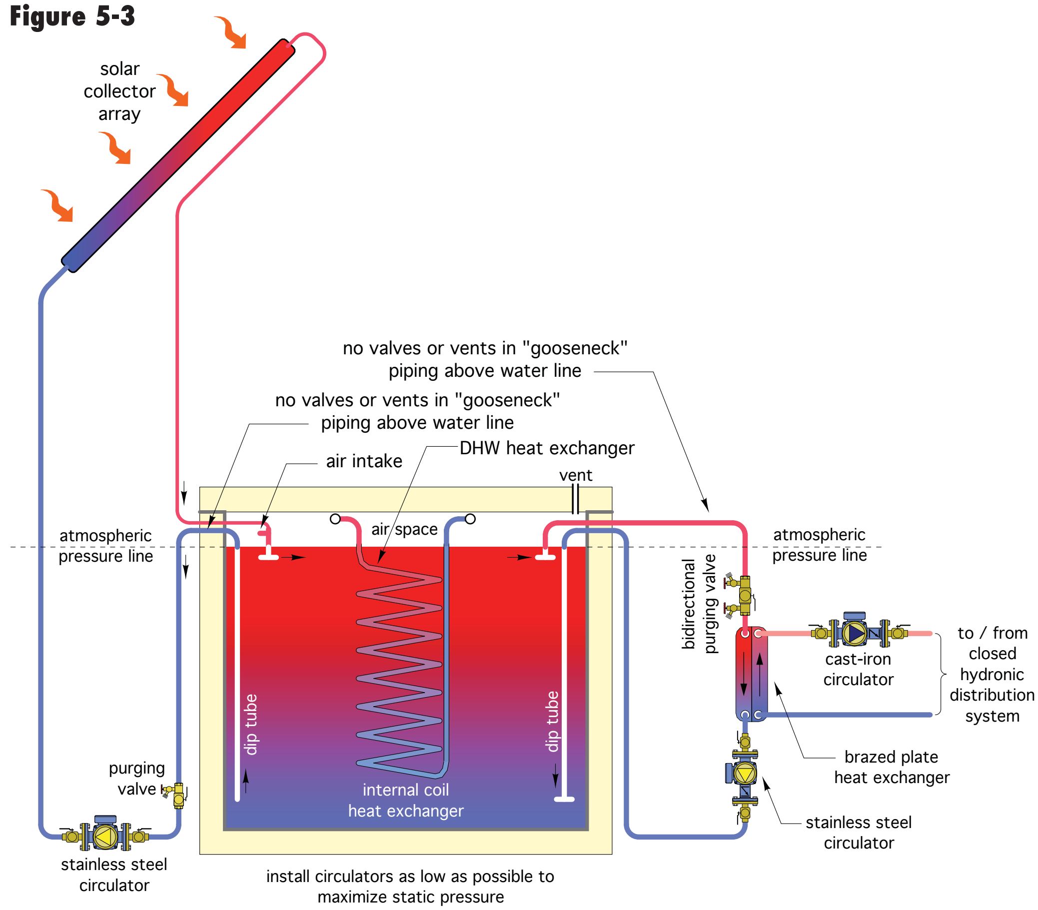
PVC or other proprietary materials. They are hung within the structural shell and conform to its shape as water is added. Figure 5-2 shows an example of a "fitted" liner inside a partially assembled polystyrene tank shell.

All unpressurized thermal storage tanks must also have an insulated cover that limits evaporation of water and

Figure 5-2

Courtesy of Cocoon Tanks

Figure 5-3



minimizes heat transfer. The cover of an unpressurized tank often has a small hole through which air pressure is allowed to equalize between the upper inside of the tank and the surrounding atmosphere.

CONNECTING PIPING TO UNPRESSURIZED TANKS:
In some applications, unpressurized tanks can connect directly to a hydronic circuit. In others, it is necessary to separate the water in the tank from the water in the piping circuit using a heat exchanger.

Figure 5-3 shows several possible piping details that can be used with unpressurized thermal storage tanks.

Piping connections to unpressurized tanks are usually made through the side of the shell and liner, *a few inches above the water line.* This limits the possibility of water seepage from the tank at the connections. The connections are made using bulkhead fittings with compressible gaskets.

The detail where the piping goes through the tank wall and then turns downward is called *gooseneck piping.* The pressure within the piping that is above the tank's water level will be slightly below atmospheric pressure. This is acceptable, provided that no air can leak into the piping. *No valves, vents or other piping devices with seals should be installed in the gooseneck piping.*

The air initially in the gooseneck piping must be blown downward and out the lower end of the dip tube when the system is commissioned. This is done with forced water flow through a purging valve seen near the inlet of the circulator. Once filled, the gooseneck piping will retain its water content.

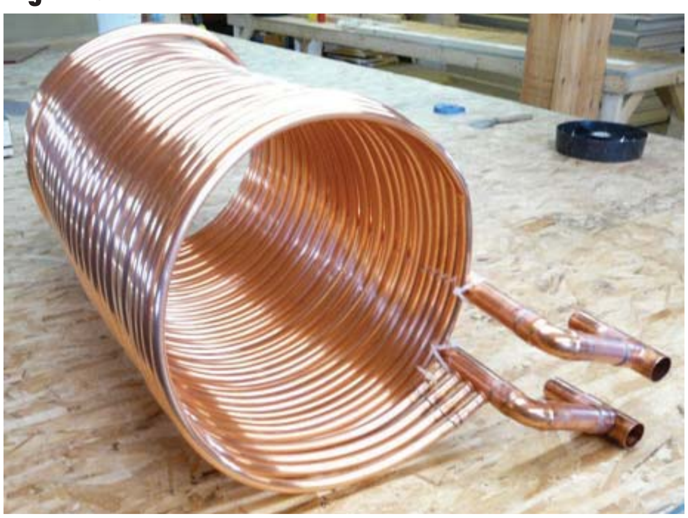
Unpressurized thermal storage tanks are not completely filled with water. This allows space for expansion of the water as it is heated.

It also allows space for the piping entering and leaving the tank to pass through the shell and liner above the water line.

Figure 5-3 shows a solar collector circuit that is directly connected to the water in the pressurized tank. Notice that water from the cooler lower portion of the tank is drawn upward through an internal dip tube, which then passes through the sidewall of the tank above the water line. The gooseneck piping then turns downward and eventually into the collector array circulator. Keeping the circulator as low as practical below the water level in the tank increases static pressure on the circulator's inlet. This reduces the potential for circulator cavitation.

The piping bringing heated water from the collector array to the tank also passes through the upper sidewall of the tank. It contains a tee with its sideport a few inches above the water line. This allows air to enter the piping when the collector circulator(s) shuts off, and thus initiates drainback freeze protection. Another tee is located downstream of this air inlet tee. Its purpose is to direct the entering flow in a horizontal direction, which minimizes vertical mixing within the tank and helps preserve temperature stratification.

Figure 5-4



Courtesy of American Solartechnics.

Another way of connecting piping circuits to unpressurized thermal storage tanks is through a heat exchanger. These heat exchangers may be internal and made of coiled copper tubing or corrugated stainless steel tubing, or they may be external heat exchangers made of stainless steel.

Figure 5-3 shows a coiled internal heat exchanger near the center of the tank. The right side of Figure 5-3 shows piping for an external stainless steel brazed-plate heat exchanger. Either type of heat exchanger provides complete isolation between the tank water and a closed piping circuit through the heat exchanger.

Copper coil heat exchangers have been traditionally used with unpressurized thermal storage tanks. They can be fabricated using a single helical coil, or multiple interlaced helical coils that are headered together, as shown in Figure 5-4. Multiple interlaced coils provide the required surface area, but create far lower pressure drop compared to a single helical coil of the same tube size and surface area.

Some tanks use multiple copper coil heat exchangers that are piped in parallel. This increases surface area while also reducing pressure drop.

For a given rate of heat transfer, the greater the surface area of the coil heat exchanger, the smaller the temperature difference needs to be between the fluid supplying heat and the fluid accepting heat. A common measurement of this temperature difference is called "approach temperature difference." It is the difference between the hottest water in the tank and the temperature of the fluid leaving the coil heat exchanger. An "ideal" heat exchanger would have an approach temperature difference of 0. Practical copper or stainless steel coil heat exchangers can be designed for approach temperature differences in the range of 5 to 10ºF.

Theoretical methods for estimating approach temperature and heat transfer rates through vertically oriented helical coil heat exchangers are very complex. Heat transfer is affected by many factors, such as flow velocity inside the tube, natural convection coefficients on the outer coil surfaces, diameter of the coil, height of the coil and overlapping versus staggering of vertically arranged coils. Companies that supply coil heat exchangers typically perform testing and use the results to generate tables that list the heat transfer rate of a specific coil as a function of the average tank temperature and the water temperature entering the coil.

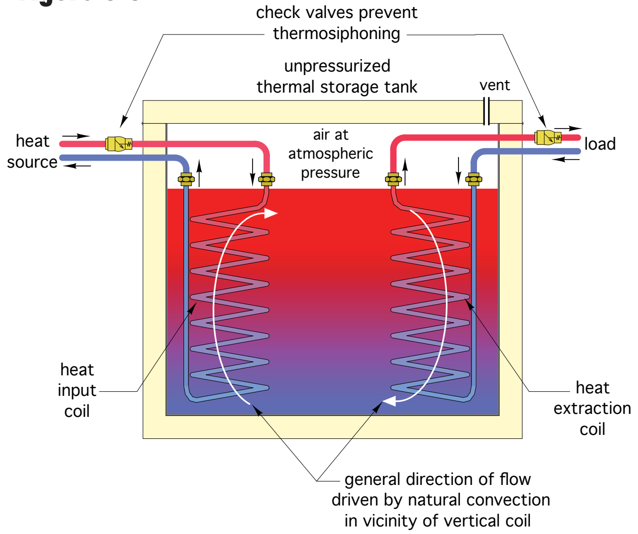
The flow direction through vertical coil heat exchangers is important. It should always be complementary to temperature stratification within the thermal storage tank.

Coils adding heat to the tank should have hot water entering the top of the coil and flowing downward.

Coils extracting heat from the tank should have cooler water entering at the bottom of the coil and flowing upward.

These flow directions, in combination with fluid movement due to natural convection within the tank,

Figure 5-5



create counterflow heat exchange. This maximizes the rate of heat transfer for a given set of operating conditions. Examples for both scenarios are shown in Figure 5-5.

STRENGTHS AND LIMITATIONS OF UNPRESSURIZED THERMAL STORAGE:

Like many devices used in hydronic systems, unpressurized thermal storage tanks have both strengths and limitations:

STRENGTHS:

• Most large unpressurized thermal storage tanks ship disassembled or partially collapsed, and as such, can fit through standard passage doors.

• Unpressurized tanks are usually less expensive than pressurized tanks on a dollar per gallon basis.

• Because they are vented, they are inherently protected from overpressure conditions. There is no need to install a pressure relief valve or vacuum breaker on an unpressurized tank.

• Because they are not completely filled, they can be designed to provide expansion space for the water they contain, as well as water in piping that is directly connected to the tank. Thus, there is no need to connect a separate expansion tank to an unpressurized thermal storage tank.

• Unpressurized tanks are not subject to mechanical codes that require ASME-certified welding on pressure vessels.

LIMITATIONS:

Because they are vented to the atmosphere:

1. Some water loss occurs due to evaporation. The water level must be periodically checked and new water added to make up for losses.

2. All piping circuits connected directly to the tank must use non-ferrous materials for piping, valves, circulators, heat emitters and other components.

3. There is the possibility of biological growth in the tank. This can be controlled with biocide additives.

4. Piping circuits connected directly to the tank and rising above the tank's water level will be below atmospheric pressure when circulators are off. Any component that allows air to enter this portion of the piping will cause water to drain back to the tank.

5. Circulators in piping circuits connected directly to the tank water operate under relatively low static pressure, and thus, are more prone to cavitation.

PRESSURIZED THERMAL STORAGE TANKS:

Pressurized storage tanks are available in a wide range of sizes and pressure ratings, with or without insulation, and equipped with one or more internal coil heat exchangers.

Pressurized thermal storage tanks allow easier interfacing with closed hydronic subsystems. Most hydronic piping circuits, other than those used for domestic water, can be connected directly to pressurized tank without need of heat exchangers.

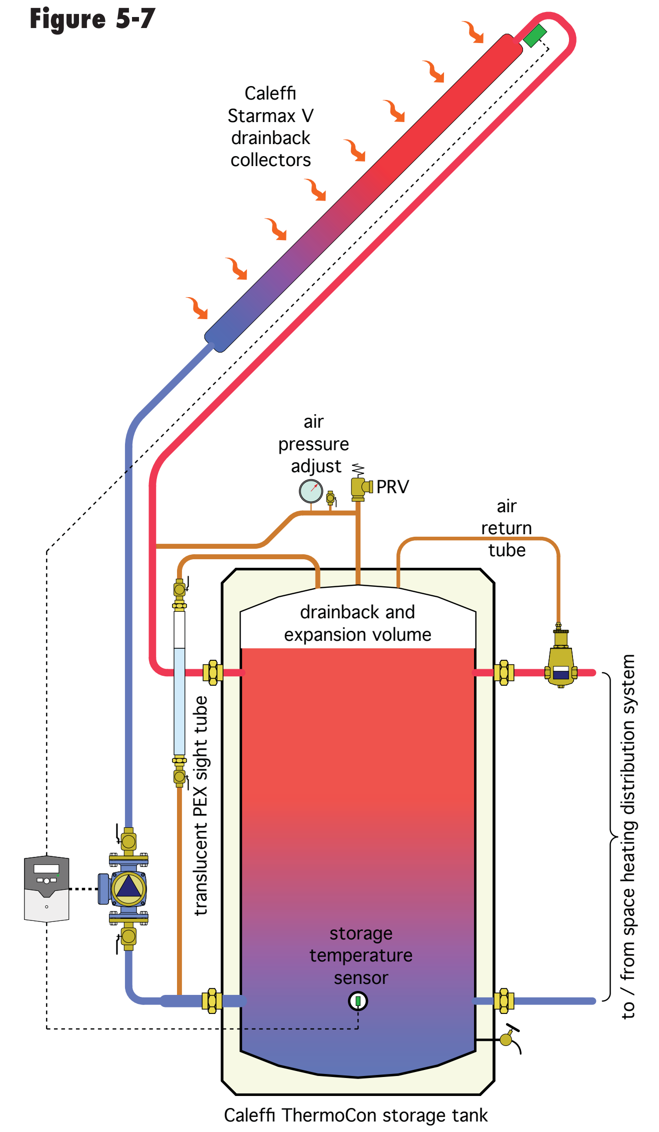
Figure 5-6

Figure 5-6 shows an example of a pre-insulated and pressure-rated thermal storage tank.

This tank is available in several sizes up to 119 gallons. It has several piping connections on its sides and top. These connections allow flexibility in how the tank is incorporated into a wide variety of systems. The larger pipe connections on the sides of this tank also minimize head loss, which in some piping configurations allows the tank to provide hydraulic separation of multiple circulators. A threaded brass or stainless steel bushing is used when smaller piping needs to be connected to the larger size tappings.

Figure 5-7 shows how the multiple connections on this tank allow it to be configured for a drainback solar thermal system.

Figure 5-7



Notice that the water flowing through the collectors also passes through the storage tank, as well as out into the space-heating distribution system. The absence of heat exchangers between these subsystems reduces cost and eliminates the thermal penalty associated with having a heat exchanger in the path of heat flow. This increases the efficiency of the solar thermal collectors, and ultimately allows more solar energy to be captured by the system.

LARGE PRESSURIZED THERMAL STORAGE TANKS:

Some hydronic systems supplied by intermediate heat sources, such as solar thermal collectors, wood-fired boilers or electric boilers operating on off-peak electricity, require tanks with volumes much greater than 119 gallons. In these situations, the designer may prefer to use a single pressurized tank, rather than an array of small tanks. Larger pressure-rated tanks are available both as standard products, as well as custom-welded products. Some are supplied with insulation and an outer jacket. Others are supplied as a bare pressure vessel that must be insulated onsite, usually after all piping connections are made.

Most of the larger pressure-rated tanks designed for use in closed hydronic systems are constructed of carbon steel.

Figure 5-8a

Some mechanical codes currently used in the United States require that pressurized storage vessels with volumes of 120 gallons or more be certified to the ASME Section VIII *"Rules For Construction Of Pressure Vessels"* code. This certification requires traceability of materials, as well as physical inspections and pressure testing as the tank is manufactured. Tanks

Figure 5-8b

Courtesy of Hydronic Specialty Supply

that pass the materials and testing requirements are identified with a placard and accompanying paperwork.

This ASME code does allow for certain exemptions where the tank would not have to be certified based on a temperature limit of 210ºF and a pressure limit of 15 psi. However, different mechanical codes in various jurisdictions may or may not allow these exemptions. It is always prudent to determine the exact requirements of a given code jurisdiction regarding ASME certification of thermal storage tanks and plan accordingly.

Figure 5-8a shows an example of an ASME certified 210-gallon uninsulated thermal storage tank with a flat top and bottom. Several internal stay rods (seen in Figure 5-8b) are welded between the top and bottom plates to prevent bulging under pressure.

Figure 5-9

Courtesy of Niles Steel Tanks

Figure 5-9 shows an example of another pressure-rated tank made in several sizes up to 860 gallons. This tank uses semi-elliptical heads to withstand internal pressure. It also uses a base ring that holds the tank in a vertical upright position and allows access to a connection at the low point of the tank.

The tanks shown in Figures 5-8 and 5-9 both require insulation. In most cases, this is applied onsite after the tank has been fully piped. Insulation products range from spray polyurethane foam, to form-fitting elastomeric foam and cellular glass materials.

It is imperative that the tank's insulation system can:

• Withstand the maximum temperature to which the tank may be heated
• Provide a minimum of R-12 (ºF•hr•ft^2/Btu) thermal resistance (more if possible)
• Meet all applicable fire and smoke spread ratings
• Provide reasonable resistance to aging, ultraviolet degradation or mechanical damage
• Provide a vapor barrier at the outer surface of the insulation if chilled water will be stored

Large steel storage tanks can weigh several hundred to several thousand pounds. Whenever such a tank will be used in a system, it's imperative to plan:

• How will the tank be delivered and moved into position within the building?
• How will all connections on the tank be accessed?
• How will the insulation system be installed on the tank?
• How much will the tank weigh when fully filled, and what will support this weight?
• If in an unheated location, how will water in the tank be protected from freezing?
• If ever necessary, how will the tank be removed from the building?

Imagine a situation in which a large thermal storage tank has been ordered and arrives on the jobsite. As the crew begins moving the tank into the building, they find that it is 1 inch wider than the door into the mechanical room. Or, assuming the tank has made it into the mechanical room, it lacks 1 inch of floor-to-ceiling clearance needed to rotate the tank into its vertical position. Or, assuming it has been rotated into position, there is insufficient ceiling clearance to install an air vent at the top connection. These are all possibilities that would undoubtedly lead to expensive corrections. Careful logistical planning for an entry, access and exit strategy is essential when dealing with large/heavy thermal storage tanks.

Many of the larger "stock" pressure-rated tanks currently available in North America are intended for storing domestic hot water. As such, they usually have an internal vitreous glass lining that prevents contact between domestic water and the steel pressure vessel. Although such a lining does not preclude the tank's use in a closed hydronic system, it adds cost compared to a standard carbon steel tank, which is generally sufficient in such systems.

Large tanks designed for storage of domestic hot water have piping connections that are not necessarily placed or sized to allow the tank to perform well in hydronic system applications. Although it may be possible to "shoe horn" such a tank into a hydronic thermal storage application, the system's performance can be compromised due to factors such as disruption of temperature stratification or less than optimal flow patterns caused by insufficient connections. It makes little sense to spend thousands of dollars on a tank that is not well suited for thermal storage in a hydronic system.

MULTIPLE THERMAL STORAGE TANK ARRAYS:

An alternative to a single large pressurized tank is a group of smaller pressurized tanks arranged side by side, as illustrated in Figure 5-10.

Figure 5-10

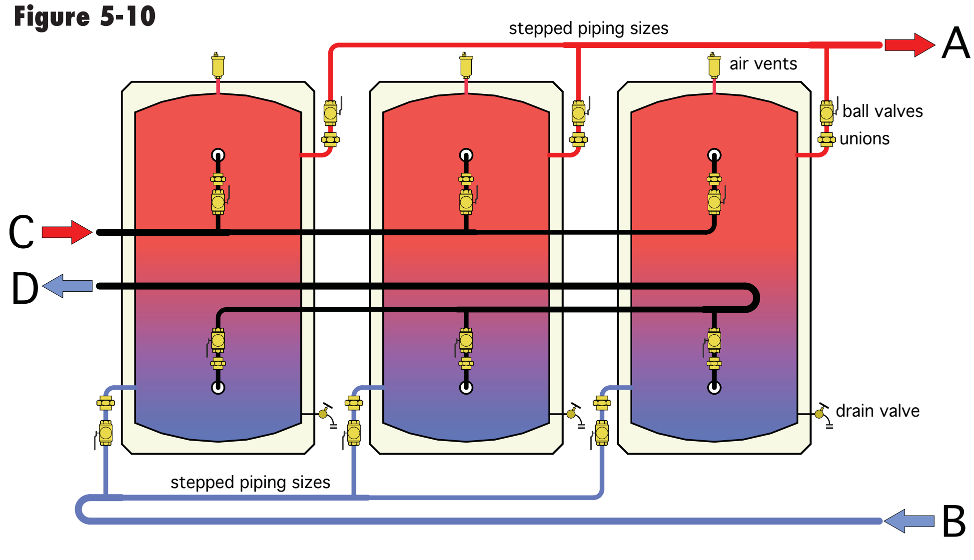
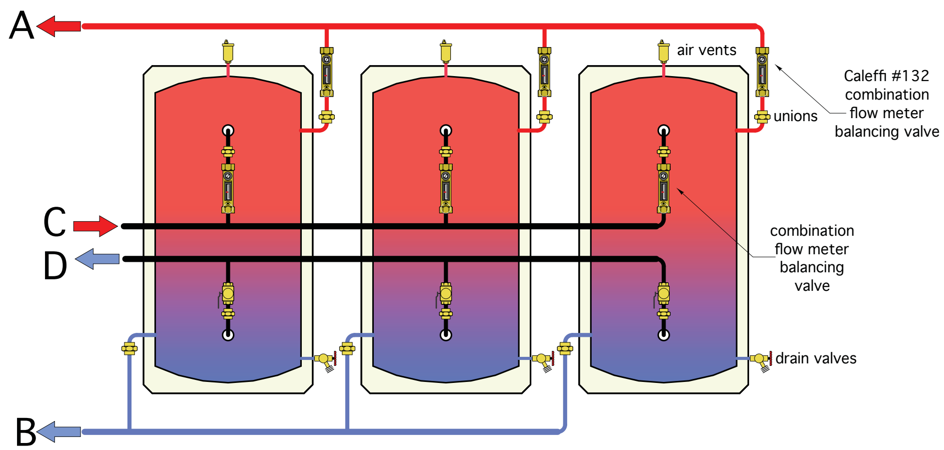


Figure 5-11



more piping, fittings and valves are needed to properly connect multiple tanks. These add to both hardware and installation cost.

Multiple tank arrays can be piped in parallel reverse return, as illustrated in Figure 5-10. This helps in achieving equal flow through each tank. The piping shown between points A and B is parallel reverse return, and so is the piping between points C and D. To reduce total piping heat loss, it's best to use the extra length of piping required for reverse return on the cooler piping supplying the lower connections on the tank.

In any type of reverse return piping, the pipe sizes should be stepped down in the downstream flow direction to approximate equal head loss per unit length of piping. This detailing, along with symmetrical piping between the headers and tanks, helps produce approximately equal flow rates through each tank.

It is also possible to pipe multiple tanks in parallel *direct* return, as shown in Figure 5-11.

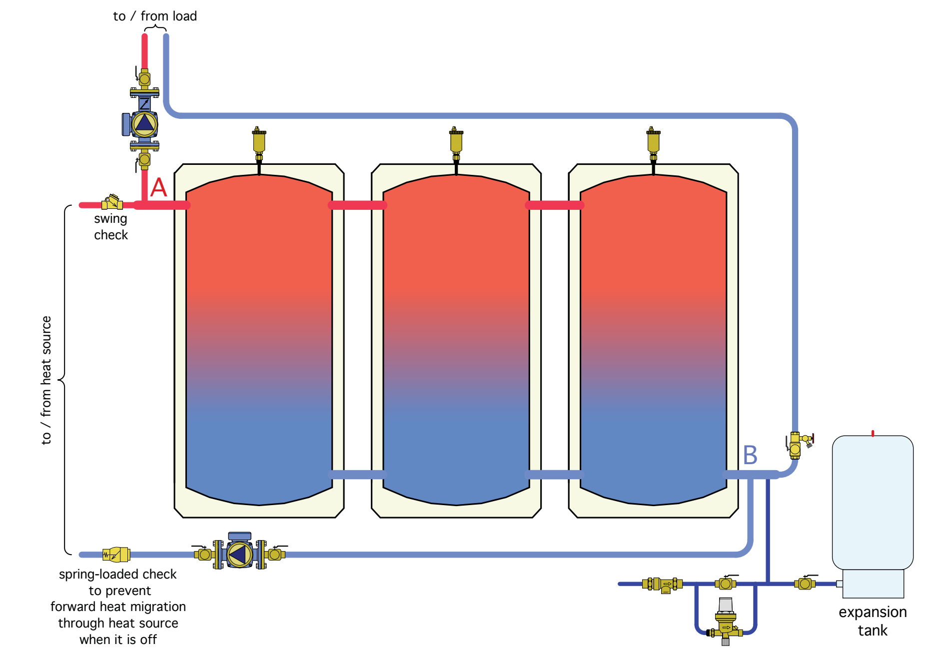
One advantage of a multiple tank array is that many of the tanks that would be used in such an array can pass through a 36-inch wide doorway. This is important during installation, as well as if a tank ever has to be replaced. Another advantage is that larger storage volumes can be attained without need of ASME-rated pressure vessels, provided that individual tanks do not exceed a volume of 119 gallons.

A disadvantage of multiple tank arrays is that they have significantly higher combined surface area than a single larger tank of equal volume. This creates greater standby heat loss, unless additional insulation is added to compensate for the greater surface area. Another disadvantage is that

Parallel direct return piping requires a balancing valve in the branch pipe to each tank connection. A combination flow meter/balancing valve, such as a Caleffi #132 Quicksetter, can be used instead of the combination of a standard ball valve and balancing valve. Each balancing valve would then be adjusted to provide equal flow rates through each tank.

The potential advantage of the piping arrangements shown in Figures 5-10 and 5-11 is that each tank can be isolated from the array. With proper space planning and placement of piping, it is also possible to remove any of the tanks without affecting the other tanks or disrupting system operation.

Figure 5-12



This connection requires a steel pipe nipple threaded into each tank. The nipple needs to be long enough to have its threads fully exposed outside the tank's insulation and jacket. A semi-flexible connector with FPT unions on each end connects between these nipples. When the pipe size of the connection is 2-inch, the shortest practical connector length is about 12 inches.

Figure 5-13b shows an example of a slightly different flexible stainless steel connector installed between two Caleffi ThermoCon thermal storage tanks.

If the thermal storage tanks have flange connections, it may be possible to use short expansion compensators with matching

Although this is a benefit, the likelihood of a having to replace a high-quality pressure-rated tank that has been properly applied and maintained in a closed loop system is very small. Given this low probability of failure, some designers would opt for potentially simpler and less costly ways to install multiple tanks, while foregoing the ability to individually isolate each tank. One simplified approach is shown in Figure 5-12.

This hybrid arrangement combines elements of both series and parallel piped tanks. It also represents an extended version of the 2-pipe arrangement discussed in section 3.

The goal is to keep the pressure drop between adjacent tanks as low as possible. This requires the shortest possible piping connections between tanks. These connections should also be generously sized to minimize head loss.

Another factor to consider is that "close coupling" tanks at two piping connections, as shown in Figure 5-12, requires some flexibility in the piping connections. This is necessary due to tolerances in the weldments at the piping connection points, as well as to compensate for floors under the tanks that may not be perfectly level.

When the tanks have FPT threaded connectors, it is possible to close couple them with a braid-reinforced corrugated stainless steel connector, as shown in Figure 5-13a.

Figure 5-13a

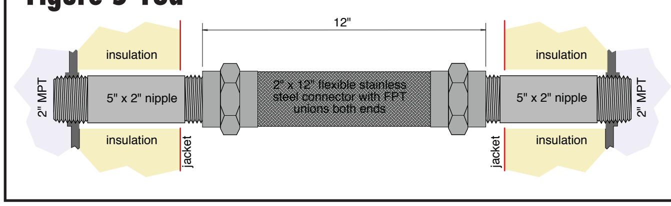
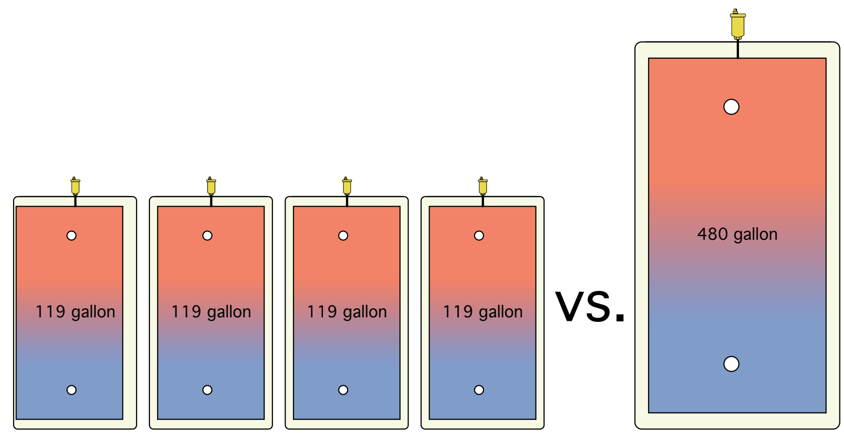


Figure 5-13b

Figure 5-14

Courtesy of Unaflex

Figure 5-15



flanges between the tanks. Figure 5-14 shows an example of a short-flanged expansion compensator.

SURFACE-TO-VOLUME RATIO:

Every thermal storage tank has both a volume and an exterior surface area. The greater the ratio of surface area to volume, the greater the heat loss (or gain) of the tank, assuming equal insulation on all surfaces. Thus, assuming all other factors are equal, a thermal storage tank with a lower surface-to-volume ratio would generally be preferable to another tank of the same volume, but with a higher surface-to-volume ratio.

If the only consideration in choosing a storage tank were to minimize the surface-to-volume ratio, all tanks would be specified as spheres. Although it's possible to build tanks of this shape, they are significantly more expensive due to the fabrication requirements. Spherical tanks are also much harder to insulate with any material other than a spray-applied foam. Given these complications and cost factors, spherical tanks are typically only justified in cryogenic storage containers that contain materials very close to absolute zero temperature.

The surface-to-volume ratio of candidate storage tanks can be compared using relatively quick calculations. For example: compare the surface-to-volume ratio of two candidate storage options, with one using four 119-gallon tanks and the other using a single 480-gallon tank. These options are shown in Figure 5-15.

Assume all tanks are flat-ended cylinders with a height-to-diameter ratio of 3:1 (e.g., the tank's height is always 3 times it's diameter).

For a 119-gallon tank, the required dimensions are: diameter = 22.7", height = 68"

For a 480-gallon tank, the required dimensions are: diameter = 36.1", height = 108.3"

The total surface area of any flat-ended cylinder can be found using Formula 5-1.

Formula 5-1:

$$A_s = \pi d(h + 0.5d)$$

Where:

A_S = total surface area (in^2)

d = diameter (inch)

h = height (inch)

For the 119-gallon tank, this yields:

$$A_s = \pi d(h+0.5d) = \pi[22.7](68+0.5[22.7]) = 5659 in^2 = 39.3 ft^2$$

For the 480-gallon tank, this yields:

$$A_s = \pi d(h+0.5d) = \pi[36.1](108.3+0.5[36.1]) = 14330 in^2 = 99.5 ft^2$$

The surface-to-volume ratio of each tank can now be calculated:

For the 119-gallon tank: S/V = 39.3ft^2/119 gallon = 0.33 ft^2/gallon.

For the 480-gallon tank: S/V = 99.5ft^2/480 gallon = 0.207 ft^2/gallon.

In this comparison, the use of four 119-gallon tanks yields a surface-to-volume ratio that is about 59% greater than the use of a single 480-gallon tank. This is a significant difference that will create higher heat loss (or gains) for the four-tank array compared to those for the single tank.

Assuming equal insulation and surrounding temperatures for all tanks, the heat loss will be approximately proportional to the difference in surface areas. In this case, that would be 4(39.3)/99.5 = 1.58, or 58% greater heat loss for the multiple tank system. Keep in mind that this doesn't include any additional heat loss from piping and valves needed to manifold the multiple tanks together.

A large thermal storage tank will usually have a lower surface-to-volume ratio than multiple small tanks of equal total volume (assume comparable shapes). This is an advantage that has to be weighed against the more complex handling and placement logistics of the large tank versus multiple smaller tanks.

Appendix C presents formulas for calculating the total surface areas and volumes of several tank shapes.

6. SIZING THERMAL STORAGE TANKS

One of the most common uses for a thermal storage tank is to buffer the heat output of an on/off heat source when it serves a highly zoned distribution system. This buffering protects the heat source against the undesirable effects of short cycling.

Although there is no formal definition of what constitutes a "short cycle," most heating professionals agree that it's better for heat sources to operate for a minimum of several minutes once they are turned on. A minimum on-time of 10 minutes is often suggested.

Field observations have shown that low thermal mass heat sources, such as wall-hung mod/con boilers and hydronic heat pumps, can experience on-cycle times of less than one minute! These short cycles are often caused by an extensively zoned hydronic distribution system containing many independently controlled heat emitters. *The heating needs of one small zone are often very low in comparison to the minimum heat production rate of the heat source.* Furthermore, there could be as little as 3 gallons of water in the active heating circuit between the heat source and one small heat emitter. Thus, there is very little thermal mass to absorb (or "buffer") the surplus heat production. The combination of these factors will inevitably lead to short cycling.

Over an entire heating season, short cycling can result in tens of thousands of starts and stops. This accelerates wear on components such as hot surface igniters in boilers, or compressor contactors in heat pumps. It inevitably leads to premature failure of these components, and is likely to shorten the useful life of the heat source.

Short cycling also reduces the seasonal efficiency of most combustion-type heat sources. A significant percentage of each overall operating cycle occurs under "transient" conditions, where heat exchange surfaces are cooler, compared to steady state conditions under which these surfaces attain temperatures that are conducive to better combustion and more effective heat transfer.

The most common way to avoid short cycling, while still using low mass heat sources and heat emitters, it to include a properly sized and placed buffer tank in the system.

The size of the buffer tank needed for a given system depends upon:

- The rated heat output of the heat source
- The minimum concurrent heat demand of the distribution system
- The minimum "on-time" for the heat source that does not constitute a short cycle
- The allowed temperature change of the buffer tank during the heat source on-time

SIZING A BUFFER TANK FOR AN ON/OFF HEAT SOURCE:

Formula 6-1 can be used to calculate the minimum required volume of a buffer tank for use with an on/off heat source.

Formula 6-1

$$V = \frac{t\left(Q_{hs} - Q_{loadmin}\right)}{500(\Delta T)}$$

Where:

V = minimum buffer tank volume (gallons)

t = minimum heat source on time (minutes)

Q_{hs} = rated heat output of heat source (Btu/hr)

$Q_{loadmin}$ = minimum concurrent heating load when heat source is on (Btu/hr)

ΔT = change in average tank temperature during minimum heat source on time (ºF)

For example: Assume a heat pump produces a maximum heat output of 48,000 Btu/hr. It supplies heat to a highly zoned distribution system where the smallest zone requires 2,500 Btu/hr when it's operating. The designer wants the heat pump to remain on for at least 10 minutes once it starts. The buffer tank is allowed to change its average temperature from 100 to 120ºF during this cycle. Determine the minimum size buffer tank for these conditions.

Solution: Just put the numbers in the formula and calculate:

$$V = \frac{t\left(Q_{hs} - Q_{loadmin}\right)}{500(\Delta T)} = \frac{10(48000 - 2500)}{500(20)} = 45.5\,gallons$$

The longer the desired minimum on-time for the heat source, and the smaller the allowed temperature swing of the thermal storage tank during this on-cycle, the larger the buffer tank needs to be. For example, if the desired on-cycle was 15 minutes, and the allowed temperature swing of the tank was only 10ºF, Formula 6-1 would show that a buffer tank with a minimum volume of 135.5 gallons would be required.

SIZING A BUFFER TANK FOR A MODULATING HEAT SOURCE:

The ability of a heat source to modulate (e.g., reduce) its heat output when necessary allows it to better track heating loads that are continually changing. This reduces the required size of the buffer tank. Formula 6-2 can be used to calculate the required size.

Formula 6-2:

$$V = \frac{t\left(Q_{hsmin} - Q_{loadmin}\right)}{500(\Delta T)}$$

Where:

V = minimum buffer tank volume (gallons)

t = minimum heat source on time (minutes)

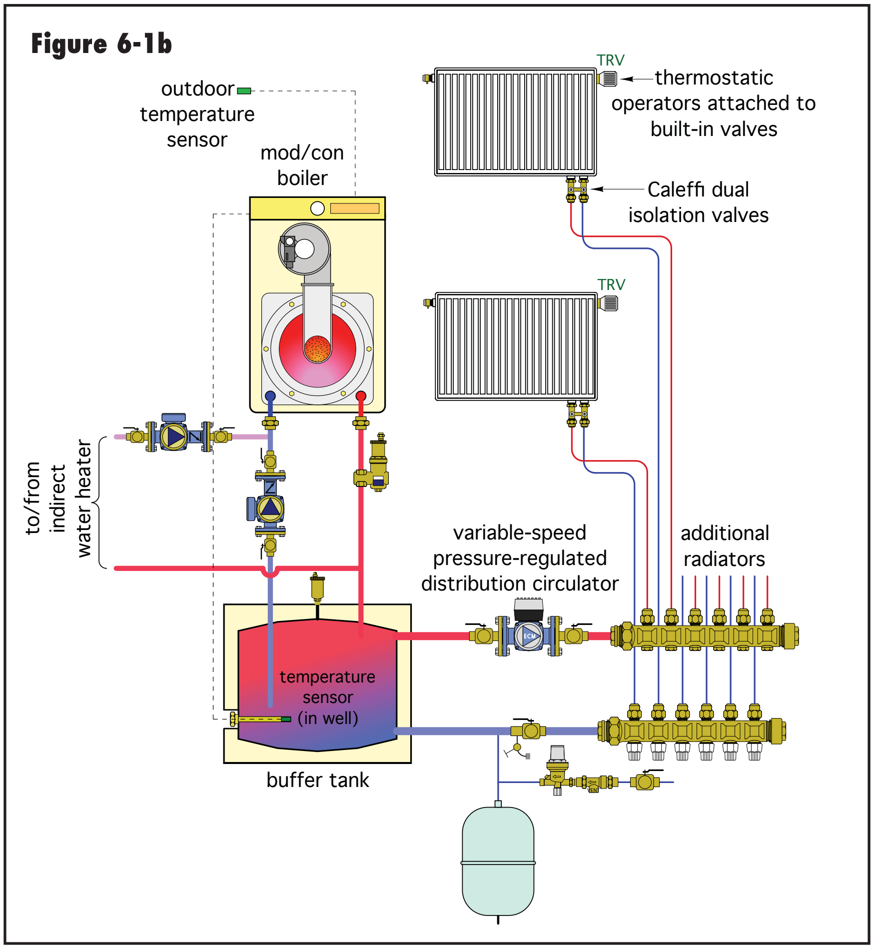
Q_{hsmin} = minimum stable heat output of heat source (Btu/hr)

$Q_{loadmin}$ = minimum concurrent heating load when heat source is on (Btu/hr)

ΔT = change in average tank temperature during minimum heat source on time (ºF)

A comparison of Formulas 6-1and 6-2 shows that the rated capacity of an on/off heat source (Q_{hs}) has been replaced by the minimum stable heat output of the modulating heat source (Q_{hsmin}).

Figure 6-1a



For example: Assume a mod/con boiler with a rated heat output of 50,000 Btu/hr can modulate its heat output down to 10,000 Btu/hr. Assume that the boiler should stay on for 20 minutes once it is started, and that the minimum concurrent load is only 1,500 Btu/hr. Also assume that the allowed change in buffer tank temperature is 20ºF. Determine the minimum required buffer tank volume.

Solution:

$$V=\frac{t\left(Q_{hsmin}-Q_{loadmin}\right)}{500(\Delta T)}=\frac{20\left(10000-1500\right)}{500(20)}=17\,gallons$$

This is significantly smaller than the buffer tank calculated for the on/off heat source. Small buffer tanks are available for such situations. Figure 6-1a shows a small buffer tank that is specifically designed to mount under a wall-hung mod/con boiler.

SIZING THERMAL STORAGE FOR SOLAR THERMAL SYSTEMS:

The thermal storage tank in a solar thermal combisystem may perform several functions including:

- Storing heat gathered from the collector array
- Providing heat to both space heating and domestic water heating
- Buffering the auxiliary boiler against short cycling
- Providing hydraulic separation between multiple circulators

Given its multiple roles, and the wide variability in solar energy resources from one location to another, it is best to determine the size of the thermal storage tank based on computer simulations of the system.

Still, guidelines have been developed over several decades of experience.

- The minimum suggested thermal storage for a solar domestic water-heating system is typically 1.0 gallon for each square foot of collector array.

- The maximum suggested thermal storage, based on both thermal performance and cost, is 3 gallons of water per square foot of collector array.

Many solar thermal systems have storage tanks closer to 1.5 to 2 gallons of water per square foot of collector array.

Thermal storage that is grossly oversized can actually reduce the performance of the solar thermal system, especially if it supplies space heating at moderate temperatures. Large thermal storage tanks will not necessary experience the temperature rise needed to create water that is hot enough to supply the space-heating load. Due to their greater surface area they may also have higher rates of standby heat loss, which reduces the controllability of the system.

SIZING THERMAL STORAGE FOR A WOOD-GASIFICATION BOILER:

Unlike most conventional boilers that can be quickly turned on and off to control heat generation, a wood-gasification boiler is designed to burn a full firebox of cordwood (e.g., often referred to as a “full charge”) as quickly as possible. This creates a 2-stage combustion process that significantly improves thermal efficiency and reduces emissions. It also creates a situation where heat output from the boiler is usually much higher than the corresponding heating load. Thus, it is imperative to provide thermal storage to optimize the operation of a wood-gasification boiler.

Various assumptions can be worked into the determination of ample storage capacity for a wood-gasification boiler. A conservative assumption is that the vast majority of the heat produced by burning a full charge of wood must be absorbed by the thermal storage tank with no concurrent heating load.

Formula 6-3 can be used to determine the required storage tank volume based on the assumption that 95% of the heat produced in burning a full charge of wood is transferred to thermal storage, with no concurrent heating load. The other 5% of the heat produced is assumed to be lost through the boiler's jacket or from piping connecting the boiler to the thermal storage tank.

Formula 6-3:

$$V = \frac{\left(\frac{0.95}{8.33}\right)\left[7950 - 90.34(m)\right](n)(D_{cc})(V_{cc})}{\Delta T}$$

Where:

V = thermal storage tank volume (gallons)

m = moisture content of wood (%)

n = full cycle thermal efficiency of wood-gasification boiler (decimal %) (typically 0.65 to 0.75)

D_{cc} = density of wood stacked in combustion chamber (lb/ft^3) (typically 10-20 lb/ft^3)

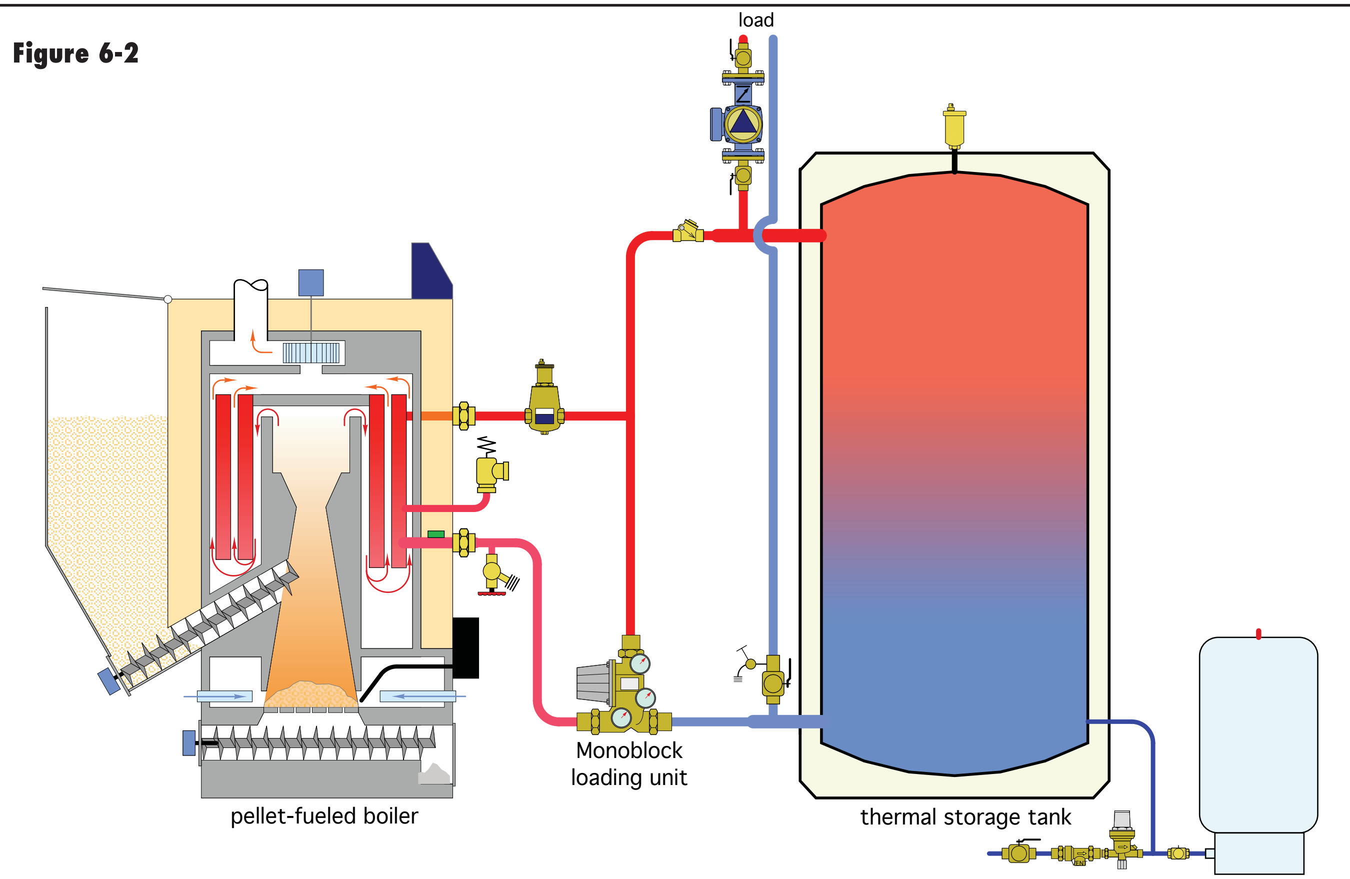
V_{cc} = volume of primary combustion chamber (ft^3)

ΔT = temperature swing of thermal storage while absorbing heat from full charge of wood (°F)

Here's an example. Assume that a wood-gasification boiler has a primary combustion chamber with a volume of 6 ft^3. When hardwood with an average moisture content of 18% is placed in the chamber, its stacked density is about 15 lb/ft^3. The overall cycle efficiency of the boiler is 70%. The temperature of the thermal storage tank is allowed to increase 65°F while absorbing the heat produced by burning a full charge (e.g., full firebox) of wood. Determine the required storage tank volume:

$$V = \frac{\left(\frac{0.95}{8.33}\right)\left[7950 - 90.34(m)\right](n)(D_{cc})(V_{cc})}{\Delta T} = \frac{\left(\frac{0.95}{8.33}\right)\left[7950 - 90.34(18)\right](0.70)(15)(5)}{65} = 583 gallons$$

Figure 6-2



This is a substantial thermal storage tank volume. Much larger than the tank volumes calculated for conventional on/off or modulating heat sources. This large volume is the result of having to absorb a large quantity of heat from burning a full charge of firewood, without any concurrent heat load that would otherwise reduce the amount of heat sent to storage.

SIZING THERMAL STORAGE FOR A PELLET-FIRED BOILER:

Modern pellet-fired boilers achieve their highest efficiency and lowest emissions when operating at or close to steady state conditions. They can also require 10 to 20 minutes to warm up to a condition where they can supply heat to a load following a cold start. Thus, once a pellet-fired boiler is turned on, it's best to keep it running for at least one hour, and in many cases, several hours.

These operating characteristics can be best coordinated with the comfort requirements of a zoned distribution system by including thermal storage into the system.

The generally recommended range of thermal storage for modern pellet-fired boilers is 1 to 2 gallons of water storage per 1,000 Btu/hr of rated boiler heat output.

The lower end of this range is appropriate for pellet-fired boilers that can modulate their heat output rate and supply minimally zoned distribution systems that have medium to high thermal mass. A good example would be a pellet-fired boiler supplying one or two zones of floor heating using a thin-slab radiant panel constructed with a 1.5-inch thick slab of concrete or poured gypsum underlayment. The thermal mass of the distribution system allows the water side thermal mass to be reduced. Pending further research on the interaction of pellet boilers with high thermal mass distribution systems, a minimum thermal storage volume of 1 gallon per 1,000 Btu/hr of rated boiler output is suggested.

Pellet boilers connected to highly zoned/low thermal mass distribution systems will perform better with higher amounts of thermal storage. *A minimum of 2 gallons per 1,000 Btu/hr of rated boiler output is suggested.*

Figure 6-2 shows one way to connect a pellet-fired boiler to a thermal storage tank. The tank uses a 2-pipe configuration, as discussed in section 3.

7. STRATEGIES FOR APPLYING THERMAL STORAGE

Previous sections have discussed how thermal storage tanks can be used to buffer the operation of heat sources to prevent short cycling. While this is one of the most common applications for thermal storage in hydronic systems, it is not the only manner in which storage can be applied. This section introduces several unique strategies for implementing and controlling thermal storage. These applications reinforce the value of having storage, and allow it to be used in ways that further enhance system performance.

ON-DEMAND DOMESTIC WATER-HEATING:

Many of the hydronic systems used in residential and light commercial buildings need to supply space heating and domestic hot water. The traditional approach to hydronic-based domestic water heating uses an indirect water heater supplied by the system's boiler, and often controlled as a priority load. If the system has an extensively zoned distribution system, it may also have a separate buffer tank to prevent boiler short cycling. Thus, these systems often contain two separate thermal storage tanks, one for buffering space-heating loads, and another for domestic water heating. An example of such a system is shown in Figure 7-1.

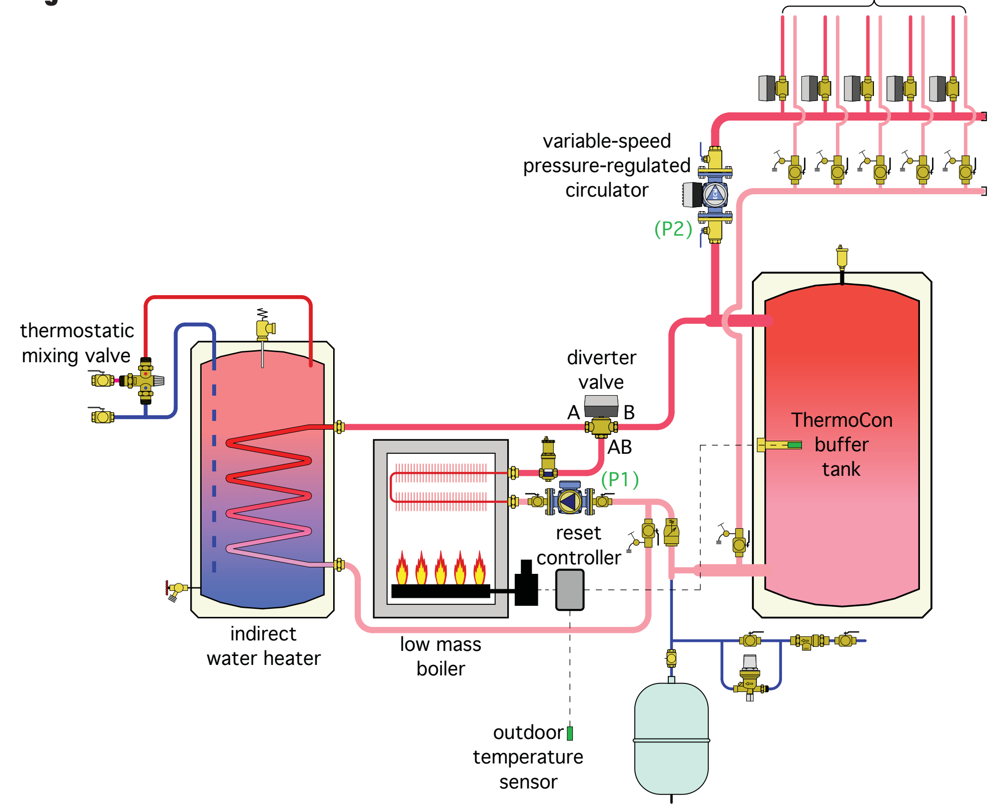
Although the "2-tank approach" has been used in many systems, there are some alternatives that may improve performance and lower installation cost. One of those alternatives is to use a single thermal storage tank to buffer both space-heating and domestic water-heating load. This is called a "single thermal mass" approach. It is based on the idea that heat can be readily transferred from "system water" (e.g., the water that passes through the space-heating portion of the system) to domestic water, or vice versa.

The system shown in Figure 7-2 shows one approach.

This system uses a special thermal storage tank that contains a stainless steel inner tank within a carbon steel outer tank. The inner tank contains domestic water, while the outer tank contains system water.

Heated water circulates from the boiler to the outer tank. This water is available to be drawn from the outer tank to the space-heating loads. Thus, the outer tank buffers the boiler against short cycling when the load is a highly zoned distribution system.

Figure 7-1

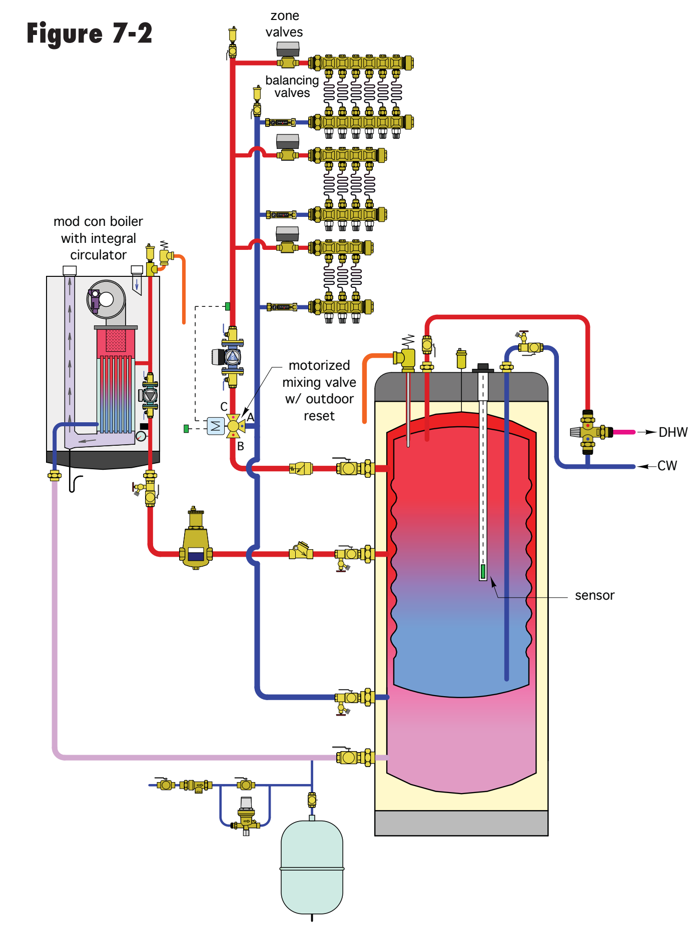


The inner tank readily accepts heat from the hot system water surrounding it in the outer tank. This is how domestic hot water is produced.

In most cases, a single tank system will reduce installation cost compared to a system using both a buffer tank for space heating, as well as an indirect water heater for domestic hot water.

The single tank system also requires less space in the mechanical room, and likely has lower standby heat loss than two separate tanks.

Figure 7-2



is always ready when needed. During sustained domestic hot water draws, the temperature of the water in the tank decreases, which initiates boiler operation.

The reverse indirect in Figure 7-3 is set up as a 2-pipe buffer tank. This arrangement coordinates well with several commercially available reverse indirect tanks. It also allows the tank to provide hydraulic separation between the boiler circulators and the distribution circulator.

Space heating is supplied from the piping just to the left of the tank. Hot water from either the boiler system (if operating), or the tank, or both, is supplied to the load through circulator (P1).

If a low-temperature space-heating distribution system is used, it is possible to reduce the water temperature to this load using either a 3-way mixing valve or a variable-speed injection pump. These options are shown in Figure 7-4.

A third approach for a single thermal mass combisystem is shown in Figure 7-5.

This system uses a Caleffi ThermoCon tank that is heated by a boiler, heat pump or other heat source.

Another "single thermal mass" storage device is known as a reverse indirect water heater. It consists of an insulated thermal storage tank with a large surface area internal coil heat exchanger. The later is used to heat domestic water. Figure 7-3 illustrates the concept.

The tank is usually made of carbon steel. As such, it should only be used in closed hydronic systems.

In a typical application, the boiler system maintains the water in the tank 10 to 20ºF higher than the required domestic water delivery temperature. Heat transfers from the water in the pressure vessel to the domestic water in the coil heat exchanger, which is typically made of copper or stainless steel tubing. Thus, domestic hot water

Figure 7-3

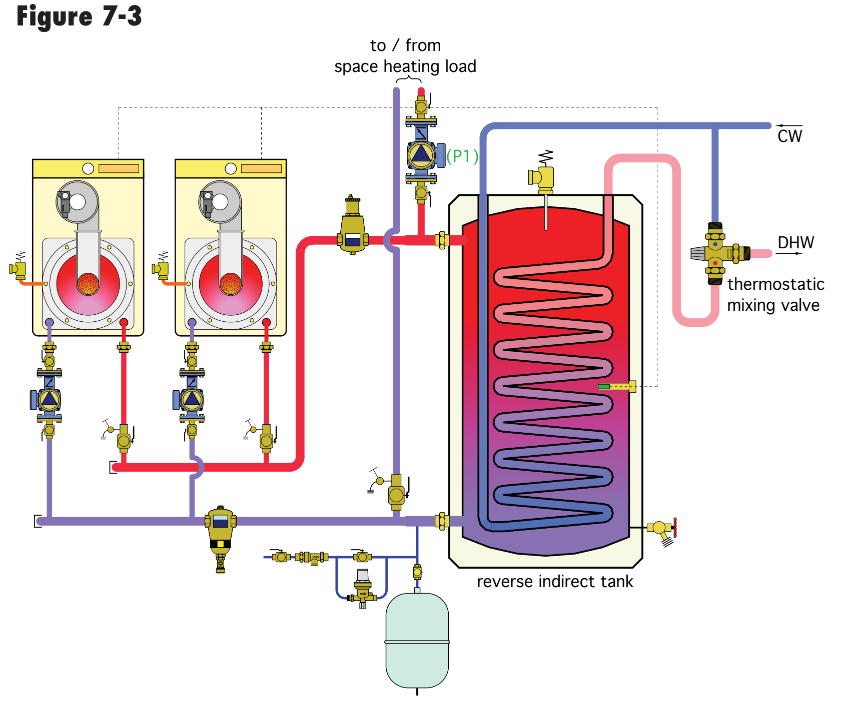
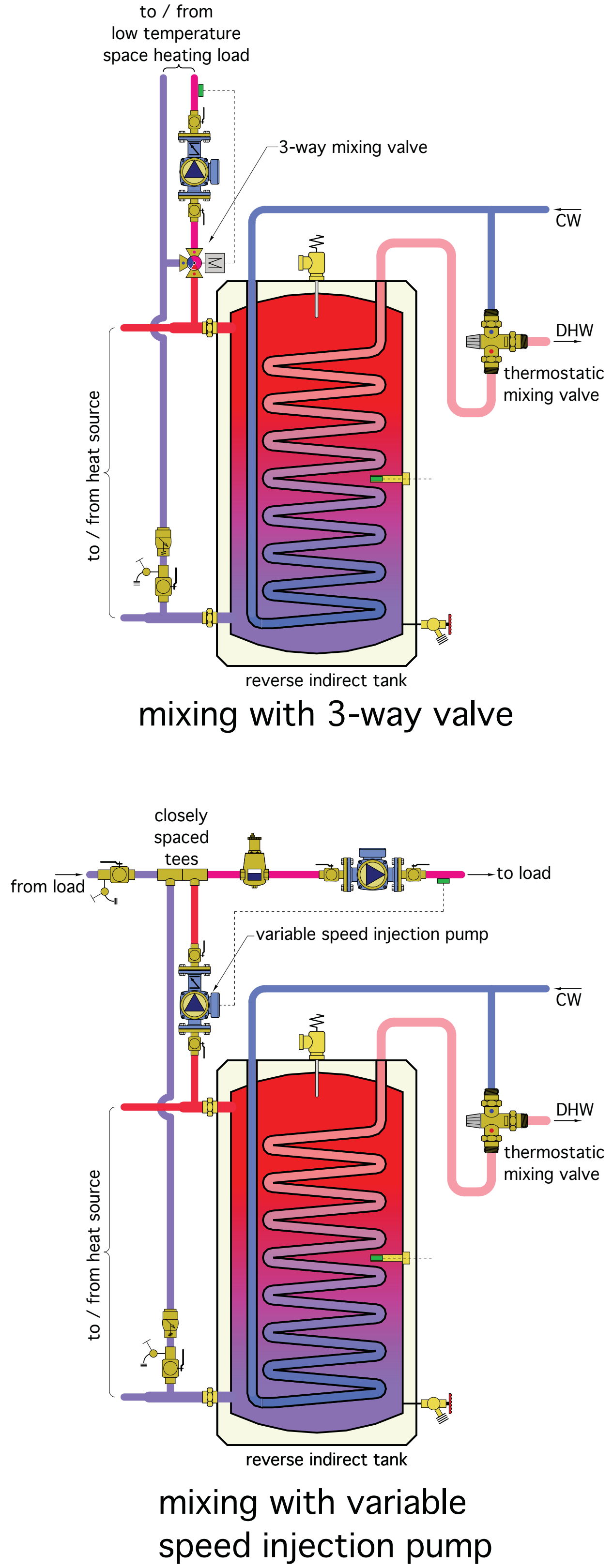


Figure 7-4



This tank provides buffering for the highly zoned space-heating distribution system.

The assembly seen on the right side of the tank is designed to provide domestic hot water by transferring heat from the system water in the tank to domestic water passing through the brazed-plate stainless steel heat exchanger. This heat transfer occurs whenever there is a "draw" for domestic hot water at a plumbing fixture.

A flow switch, located in the cold domestic water piping supplying this subassembly, detects when domestic water is being drawn at some threshold flow rate, which is typically in the range of 0.4 to 0.6 gpm. A set of contacts in the flow switch closes to energize the coil of a relay, which turns on a small circulator that moves heated water from the upper portion of the thermal storage tank through the primary side of a stainless steel heat exchanger. Cold domestic water absorbs heat as it passes through the other side of the heat exchanger.

Figure 7-6 shows an example of a small domestic water flow switch that is designed to thread into a 3/4-inch tee. The flow rate at which the contacts in the flow switch closes can be adjusted through use of an orifice within the sensing element. This device is designed to switch low-voltage circuits operating at low amperages. It can be used in combination with a relay to turn on a line voltage load operating at higher amperage.

The temperature of the heated domestic water leaving the heat exchanger depends on the temperature of the storage tank and the water flow rate. If the water in the thermal storage tank is at or above 130ºF, and the heat exchanger is properly sized, it's likely that the domestic water will be fully heated to, or above, the desired supply temperature. An ASSE 1070-listed thermostatic mixing valve is installed to protect against excessively high domestic hot water delivery temperatures.

If the thermal storage tank temperature is relatively low, perhaps in the range of 70 to 110ºF, the domestic water will only be preheated, and thus in need of auxiliary heating to achieve the desired delivery temperature. The auxiliary heating comes from a thermostatically controlled electric tankless water heater. The domestic water leaving the heat exchanger flows through this heater, which measures the incoming water temperature and determines the necessary electrical power input to its heating elements to provide any required temperature boost. If the water entering the tankless water heater is already at or above the heater's setpoint, the heating elements remain off.

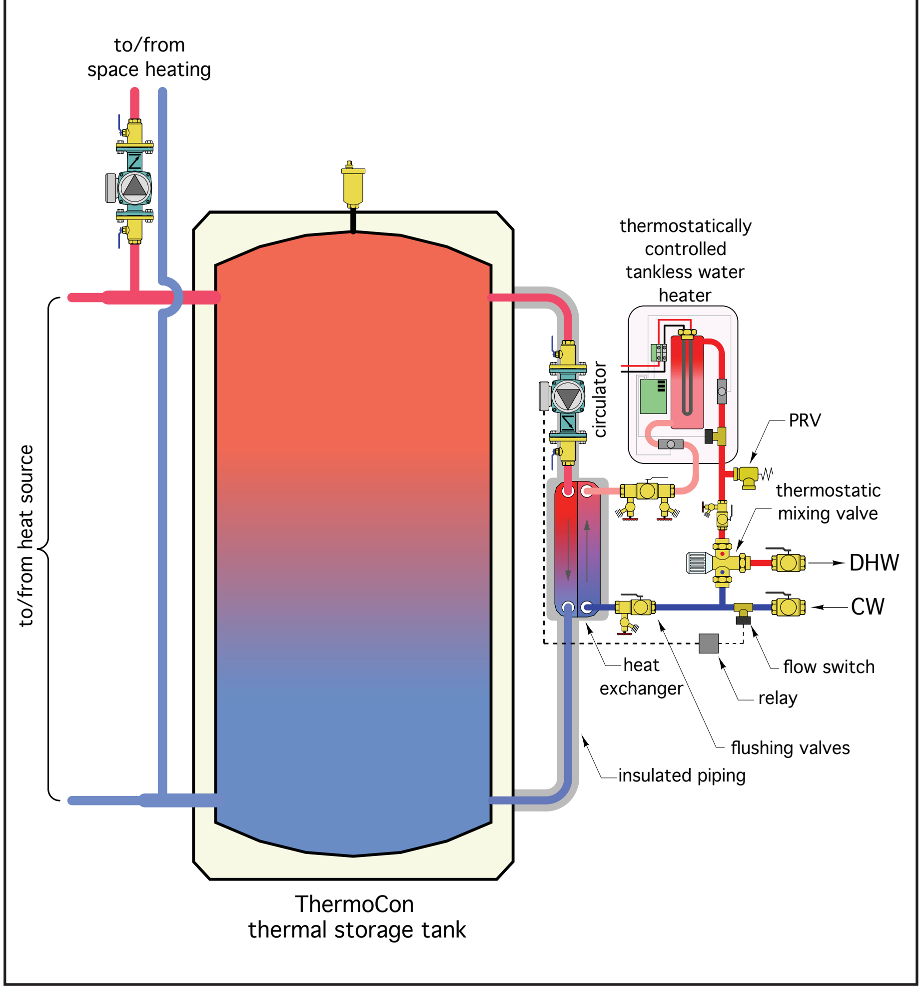


Figure 7-6a

Figure 7-6b

Courtesy of Harwil

For the fastest response, the piping between the thermal storage tank and heat exchanger should be short and fully insulated. A combination isolation/flushing valve should be installed on the domestic water inlet and outlet of the heat exchanger, as well as the inlet and outlet of the instantaneous heater. These valves allow for flushing to remove scale from the heat exchanger.

The stainless steel heat exchanger should be sized for a *maximum* approach temperature difference of 10ºF. This refers to the temperature difference between the water entering the primary side of the heat exchanger from storage and the water leaving the secondary side of the heat exchanger.

In applications where the temperature of the domestic water leaving the heat exchanger is likely to get above 140ºF, there is a possibility that safety devices in some electric tankless water heaters may automatically turn off the heat elements. A manual reset of the tankless water heater is usually required to re-enable its operation. Figure 7-7 shows one way to deal with this by installing a second thermostatic mixing valve on the inlet of the tankless water heater.

This valve should be set no higher than 5ºF *below* the temperature at which the safety device in the tankless water heater will operate. A setting of 125ºF is suggested for most applications. If the temperature of the domestic water leaving the heat exchanger is above this setting, the mixing valve blends in cold water to reduce the water temperature entering the tankless water heater. This prevents the safety device in the water heater from unnecessarily disabling its operation.

The benefits of this approach to domestic water heating include:

• No need for a separate domestic hot water storage tank. Thus, less space is required in the mechanical room.

Figure 7-7

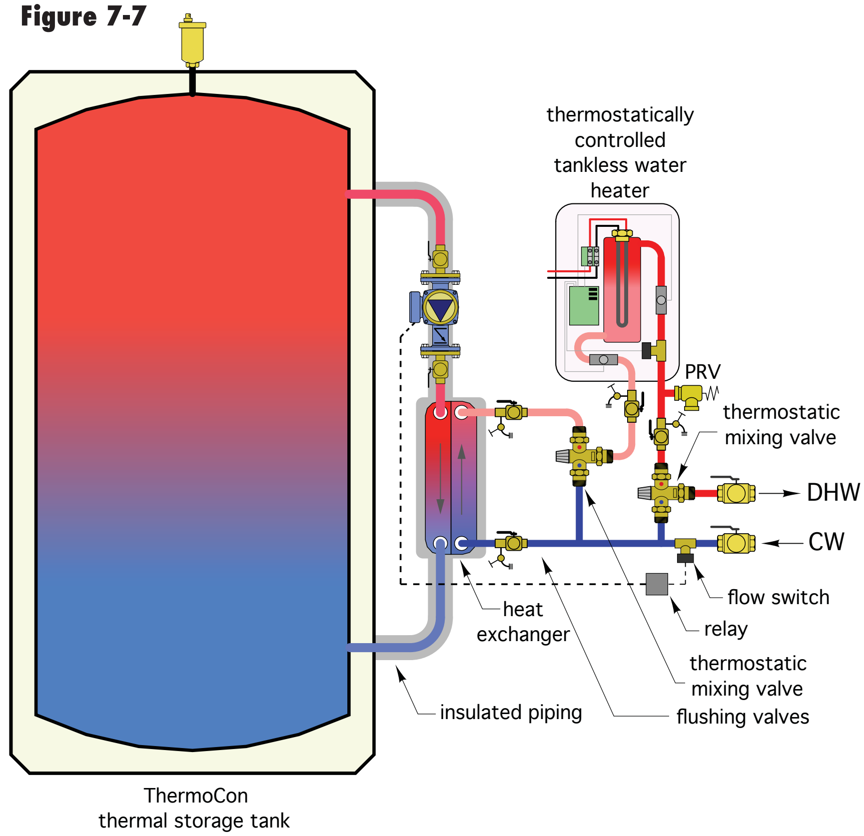
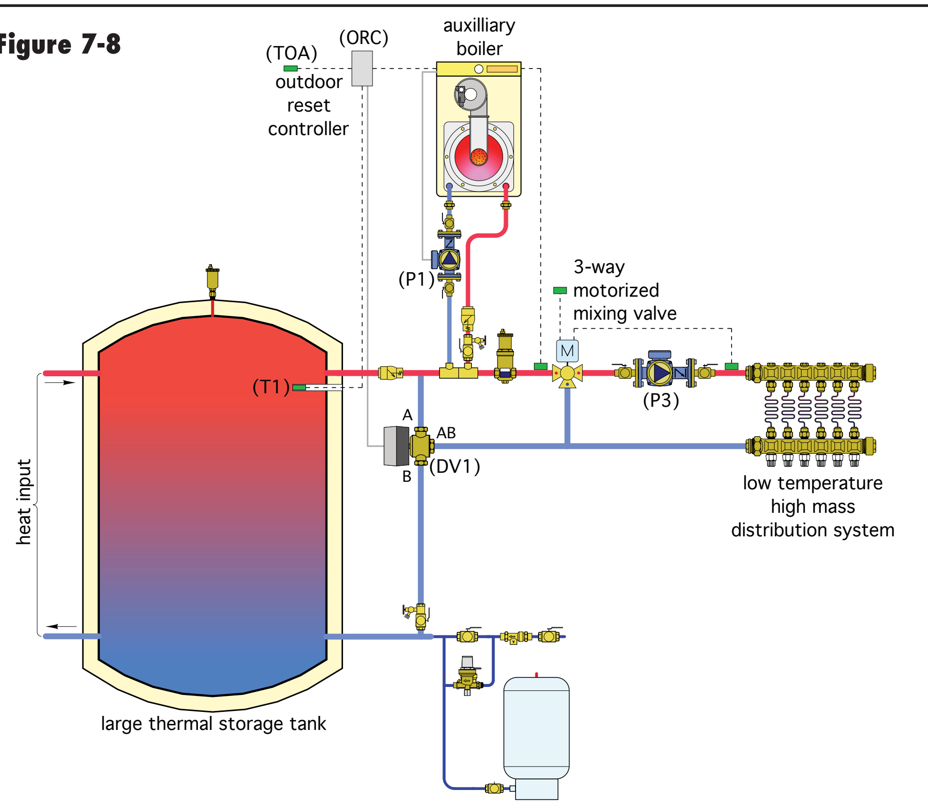


Figure 7-8



• The thermal mass of the storage tank is available to stabilize domestic hot water delivery temperature during lengthy draws.

• The standby heat loss associated with a separate DHW storage tank is eliminated.

• The stainless steel heat exchanger can be easily inspected, cleaned and replaced as necessary.

• The "warm up" time of this assembly is significantly shorter than that of a gas-fired tankless water heater because there is no need to initiate combustion.

• The possibility of Legionella is reduced since very little domestic hot water is "stored" in this assembly.

HEAT SCAVENGING FROM STORAGE:

In some applications, and under some circumstances, it is possible to simultaneously supply heat from a thermal storage tank and an auxiliary heat source. The goal is to extract heat from the thermal storage tank whenever possible, while avoiding any situation where heat from the auxiliary heat source is inadvertently added to thermal storage. This can be done by monitoring the *return* water temperature from the space-heating distribution system.

Consider the system shown in Figure 7-8. It's a space-heating system supplied from a thermal storage tank, and when necessary, an auxiliary boiler.

An outdoor reset controller (ORC) determines if the space-heating load is sourced from the thermal storage tank or the auxiliary boiler. The tank serves as the sole heat source

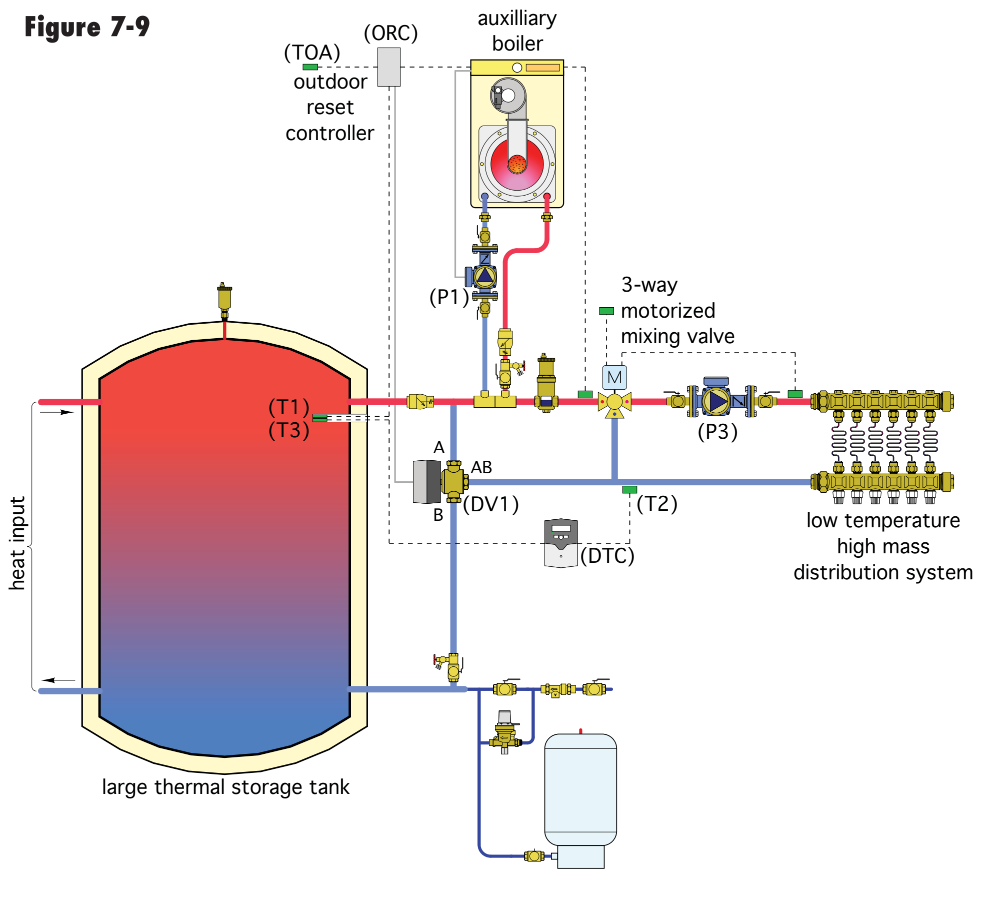


Figure 7-9

The differential temperature controller measures the temperature difference between the upper portion of the thermal storage tank at sensor (T3), and the return side of the distribution system at sensor (T2). As long as the temperature at the top of the tank is slightly warmer than the return side of the distribution system, the tank can contribute heat to the load, with the remaining heat added by the auxiliary (modulating) boiler. Under this condition, flow returning from the distribution system is routed through the thermal storage tank.

Once the temperature in the upper portion of the thermal storage tank drops to 2ºF above the return temperature from the distribution system, the differential temperature controller energizes the diverter valve (DV1) to prevent the flow returning from the distribution system from passing into storage, and instead routes it to the auxiliary boiler.

When the thermal storage tank is bypassed, there is no buffering of the auxiliary boiler. This is acceptable, provided that the boiler can modulate to meet the heating load variations of the distribution system.

Figure 7-10 shows a ladder diagram that contains the operating logic for a system piped as shown in Figure 7-9.

If the outdoor reset controller (ORC) determines that the tank is too cool to supply the space-heating load, its normally open contact closes to power up the differential temperature controller (DTC), as well as a relay (R1), which enables boiler operation.

The differential temperature controller (DTC) measures the temperature difference between water at the top of the storage tank and that returning from the distribution system. If the top tank temperature (T3) is at least 4ºF

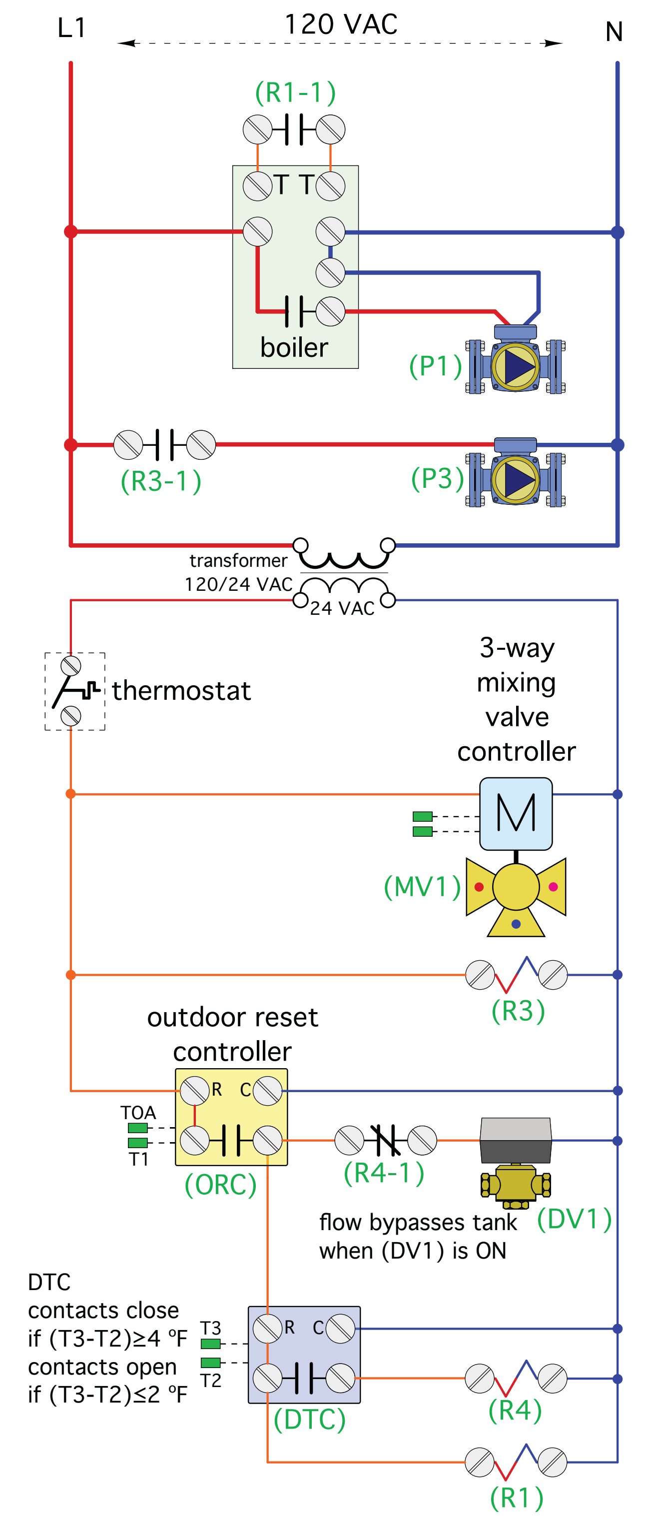
until its temperature drops below a value determined by the outdoor reset controller, and dependent on outdoor temperature. When this occurs, the outdoor reset controller turns on the diverter valve (DV1), which prevents the flow returning from the distribution system from entering the thermal storage tank. The outdoor reset controller also enables operation on the auxiliary boiler and its associated circulator (P1). With this control scenario, there is no mode where both the thermal storage tank *and* the auxiliary boiler could simultaneously supply heat to the load.

However, consider a situation where the temperature at the *top* of the storage tank is slightly lower than the lower limit established by the outdoor reset controller *but still warmer than the temperature of water returning from the distribution system.* Under such conditions, the tank could still contribute some heat to the space-heating load, with the remainder of the required heat supplied from the auxiliary boiler.

This additional heat can be extracted by adding a differential temperature controller, as shown in Figure 7-9.

above the distribution return temperature (T2), the normally open contact in the differential temperature controller (DTC) closes. This energizes the coil of relay (R4). The normally open contact (R4-1) opens, preventing the 24 VAC diverter valve from energizing (and thus routing return flow from the distribution system through the thermal storage tank).

Figure 7-10



The boiler and circulator (P1) are also operating at this time. The boiler is monitoring the temperature downstream of the closely spaced tees. It modulates to add just enough heat to maintain the supply water temperature close to the target temperature required by the distribution system based on the settings of the boiler's internal controller.

The mixing valve (MV1), operating under its own outdoor reset control logic, and with the same settings as the boiler's internal reset controller, should be at or very close to its fully open position, thus providing little if any mixing.

If the temperature difference between the top of the storage tank and the return side of the distribution system drops to 2ºF or less, there is very little useful heat remaining in the thermal storage tank. Under this condition, the differential temperature controller turns off relay (R4). The contact (R4-1) closes, allowing 24 VAC to energize the diverter valve (DV1). This stops the flow returning from the distribution system from passing into the thermal storage tank. All flow returning from the distribution system is routed into the auxiliary boiler.

When the storage tank temperature again rises to where the temperature differential between the top of the tank and the return side of the distribution system reaches 4ºF or higher, and there is a demand for space heating, the diverter valve is turned off, and flow returning from the distribution system again passes through the thermal storage tank.

This strategy is best applied in systems with relatively large storage volumes of 500 gallons or more. The ability to lower 500 gallons of water by an additional 10ºF implies a release of almost 42,000 additional Btus from the tank. Smaller tanks would contribute proportionally less heat, and thus provide less justification for the additional controls.

This strategy requires the differential temperature controller to detect differences in temperature as low as 2ºF. Because of this, it is very important that the temperature sensors for the differential temperature controller are identical. It is also important that both sensors are mounted in an *identical* manner. Ideally, both sensors would be mounted in identical sensor wells, immersed in the system water, and with ample coatings of thermal grease. If mounted directly to copper tubing, the sensors must make good contact with the surface of the tubes, be well secured and be fully wrapped with insulation.

TEMPERATURE STACKING IN THERMAL STORAGE USED WITH PELLET OR WOOD CHIP BOILERS:

Unlike most conventional boilers, which deliver heat almost immediately after being turned on, pellet-fired boilers and wood chip-fired boilers require 10 to 30 minutes from when they are called to operate to when they can deliver heat to loads at the desired water temperatures. Once this type of boiler is turned on, it's good to keep it operating, in some cases for several hours. Doing so improves combustion efficiency and reduces emissions.

It is possible to configure a thermal storage tank and controls to keep the boiler operating for longer periods, compared to controls that stop the boiler when the load is satisfied.

To maximize the energy content of the storage tank, it's temperature needs to be raised to relatively high values that are still considered safe and don't significantly lower the boiler's efficiency. This is true even in systems with low-temperature heat emitters. Such systems can use a 3-way motorized mixing valve or other mixing assembly to mediate between higher water temperature in the thermal storage tank and the lower supply water temperatures required by the distribution system.

Figure 7-11

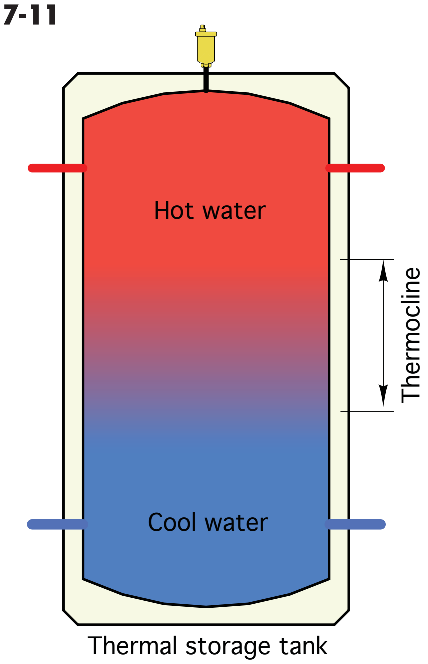


Figure 7-11 shows a thermal storage tank that has been previous charged with heat but is currently at rest (e.g., no flow in or our of the tank).

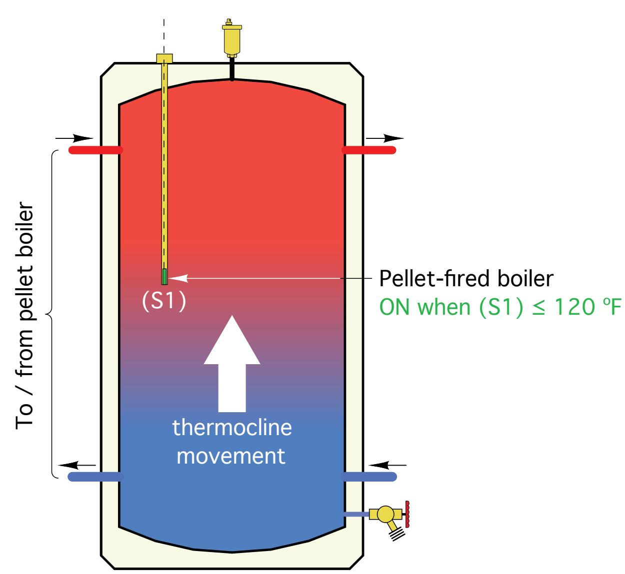
Under this condition, temperature stratification will be well established within the tank, and a thermocline will form between the cooler water in the lower portion of the tank and the hottest water in the upper portion.

When the next call for space heating occurs, hot water is drawn from the upper right side connection on the tank. Cool water returning from the distribution system enters the lower right connection. This causes the thermocline to move upward.

To provide continuity of heat delivery, the pellet-fired boiler (or wood chip boiler) should be turned on *before* the tank is depleted of hot water. This impending condition can be detected using a temperature sensor mounted in a well located *below* the outlet piping connection, as shown in Figure 7-12.

Assume that a low-temperature hydronic distribution system is being supplied by the thermal storage tank. Under design load conditions, this distribution system needs a supply water temperature of 120ºF. The boiler is turned on when the temperature several inches below the hot water outlet connection drops to or below 120ºF. This allows the tank to continue supplying heat at a reasonably stable temperature as the boiler is warming up.

Figure 7-12



Low-mass pellet-fired boilers may only require 10 minutes to reach suitable conditions where heat can be supplied to the thermal storage tank. Higher mass boilers burning pellets, and most boilers burning wood chips, can require 30 minutes or more to initiate combustion, raise their inlet water temperature to approximately 130ºF (to avoid sustained flue gas condensation), and only then begin delivering heat to the thermal storage tank. The vertical distance between the sensor well and outlet piping connection should factor in this warm up time. The longer the warm up, the greater this distance should be.

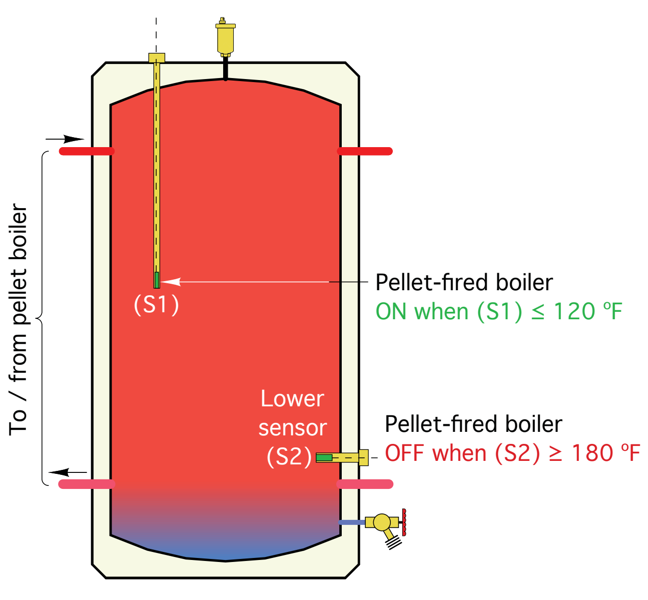
Once the boiler is operating, one of the following three scenarios will establish itself:

1. The rate of heat output from the boiler equals the rate of heat transfer to the load
2. The boiler is adding heat to the system faster than the load requires it
3. The load is removing heat from the system faster than the boiler is creating it

If scenario 1 occurs, the thermocline within the tank should remain relatively stationary. This is especially likely if "2-pipe" tank piping is used, as discussed in section 3.

If scenario 2 occurs, the thermocline will move downward within the tank due to hot water entering the upper portion of the tank.

Figure 7-13



If scenario 3 occurs, the thermocline will move upward in the tank due to heated water leaving from the top and cooler water from the return side of the distribution system entering at the bottom.

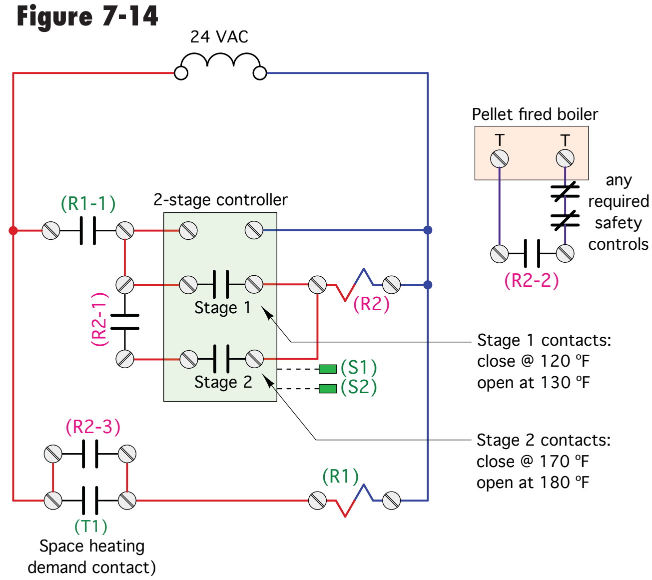
The upward movement of the thermocline can be detected by a temperature sensor (S1) in the upper portion of the tank. When this temperature drops to the point where the tank can no longer supply the heating load, the biomass boiler should be fired.

Once it is fired, the biomass boiler can continue operating *even after the heating load is satisfied. To achieve a long and efficient burn cycle, the objective is to continue heat delivery to the storage tank until it is nearly "full" of relatively hot (170 to 180ºF) water.* The latter condition is detected by a second temperature sensor mounted in the lower portion of the thermal storage tank, as shown in Figure 7-13.

When the lower temperature sensor is up to the assigned "turn off" temperature, the boiler is turned off.

This control technique produces longer *but fewer* boiler cycles over the course of a heating season, compared to the number of cycles that would otherwise occur if the boiler were turned off whenever the load is satisfied. This allows a higher percentage of each boiler cycle to be at, or close to, steady state combustion conditions, rather than at the lower efficiencies associated with warm up or fuel burnout. This strategy also assumes a well-insulated thermal storage tank with minimal heat loss at an elevated temperature.

Figure 7-14



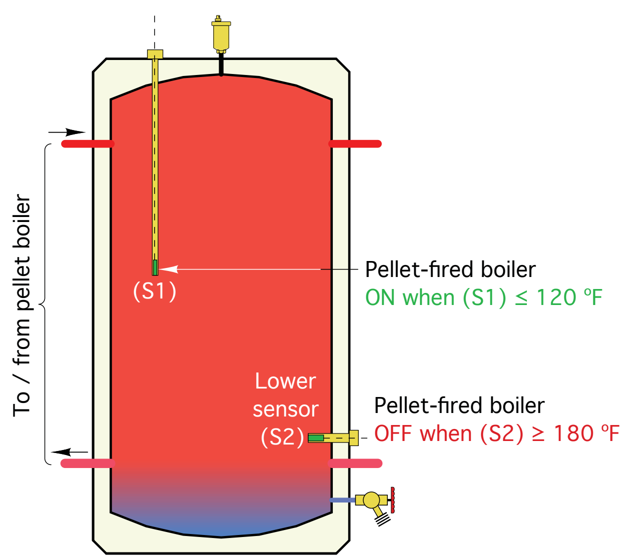
Some modern pellet- and wood chip-fired boilers come with internal controllers that provide this operating logic; others do not. The wiring schematic shown in Figure 7-14 allows this logic to be created using readily available 2-stage setpoint controllers and relays.

The demand for space heating comes from a thermostat or other dry contact (T1). This supplies 24 VAC to the coil of relay (R1). Contact (R1-1) closes to power up the 2-stage setpoint controller.

The 2-stage controller examines the temperatures at sensors (S1) and (S2). If the temperature at sensor (S1) in the upper portion of the tank is less than or equal to 120ºF, the stage 1 contact in the 2-stage controller closes.

When the stage 1 contacts of the setpoint control close, 24 VAC is passed to the coil of relay (R2). Contact (R2-2) closes to complete a circuit between the heat demand terminals (T T) on the pellet boiler, enabling it to turn on. Another contact (R2-1) also closes to pass 24 VAC to one side of the stage 2 contact in the setpoint controller. *This stage 2 contact will be closed because the temperature at the bottom of the tank is much lower than 170ºF (i.e., because the top of the tank is only 120ºF or less).* If the temperature at sensor (S1) increases to 130ºF or higher, the stage 1 contacts will open. *However, there is still a path for 24 VAC through contact (R2-1) and the stage 2 contact to keep the coil of relay (R2) energized, and thus keep the pellet-fired boiler operating.* When sensor (S2) reaches 180ºF, the stage 2 contact in the 2-stage controller opens, and the pellet-fired boiler turns off (see Figure 7-15).

Figure 7-15



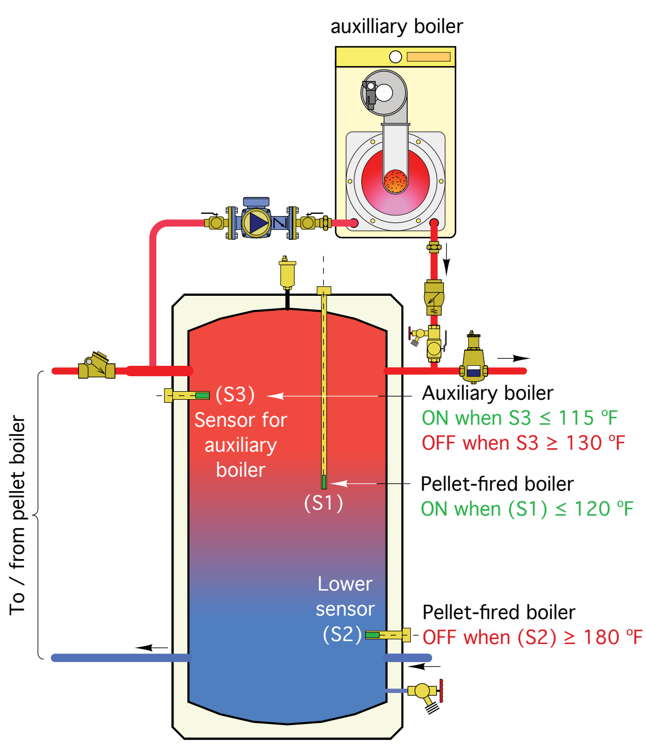
A third pole of relay (R2), contact (R2-3), closes in parallel with the heat demand contact (T1). This allows the pellet-fired boiler to remain on until the temperature stacking process is complete, even if the demand for space heating is no longer present.

It is also possible to combine the temperature-stacking control logic used with pellet or wood chip boilers with operating logic that allows an auxiliary boiler to be turned on when the biomass boiler cannot keep up with the heating load.

The method shown assumes that the auxiliary boiler is piped into the upper portion of the thermal storage tank, and thus only interacts with a small portion of the tank's thermal mass. This minimal interaction helps prevent the auxiliary boiler from short cycling when it supplies a highly zoned distribution system. At the same time, it prevents the auxiliary boiler from heating the majority of the water in the thermal storage tank.

This approach requires another temperature sensor in the upper portion of the thermal storage tank, as shown in Figure 7-16.

Figure 7-16



The objective is to turn on the pellet-fired boiler as the thermocline moves upward past sensor (S1). This allows sufficient time for the pellet-boiler to come online and deliver heat to the tank *before* all the hot water in the upper portion of the tank is sent to the distribution system.

If the load exceeds the rate of heat production from the pellet-fired boiler, the thermocline continues moving upward in the tank. The auxiliary boiler will be fired when the temperature a short distance below the hot water outlet drops to a specific setpoint. Given the relatively fast response of a low-mass auxiliary boiler, this should ensure that hot water at a suitable temperature remains available to the heating load.

Once the combined heat output of both boilers exceeds load, the temperature at the top of the tank will begin climbing, and the thermocline will move downward. The auxiliary boiler can be turned off when the temperature at the upper sensor has risen through some reasonable differential, such as 15 to 30ºF. This uses the thermal mass of the upper portion of the tank to prevent the auxiliary boiler from short cycling, but it doesn't allow the auxiliary boiler to heat the mid- and lower portions of the tank, or provide excessive heat the upper portion of the tank.

Figure 7-17

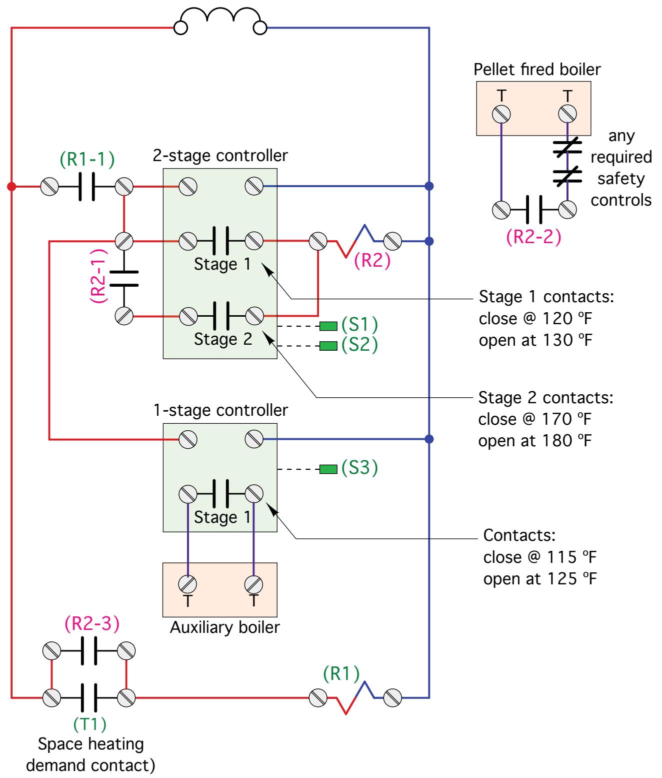


Figure 7-17 shows a modified electrical schematic that includes another temperature setpoint controller to operate the auxiliary boiler. Keep in mind that some boilers may already have the necessary logic for this control action.

STRATIFICATION INVERSION WITHIN THERMAL STORAGE TANKS:

Some thermal storage tanks need to hold heated water during the heating season and chilled water during the cooling season. This usually forces a compromise between the ideal piping and flow directions for heating mode versus those for cooling mode. However, there are ways to eliminate this compromise.

One approach uses two 3-way motorized diverter valves that coordinate to provide optimal flow directions through the tank in both heating and cooling modes. The placement of these valves is shown in Figure 7-18.

In heating mode, both diverter valves are off. Flow passes between the AB and B ports or vice versa. This allows

Figure 7-18

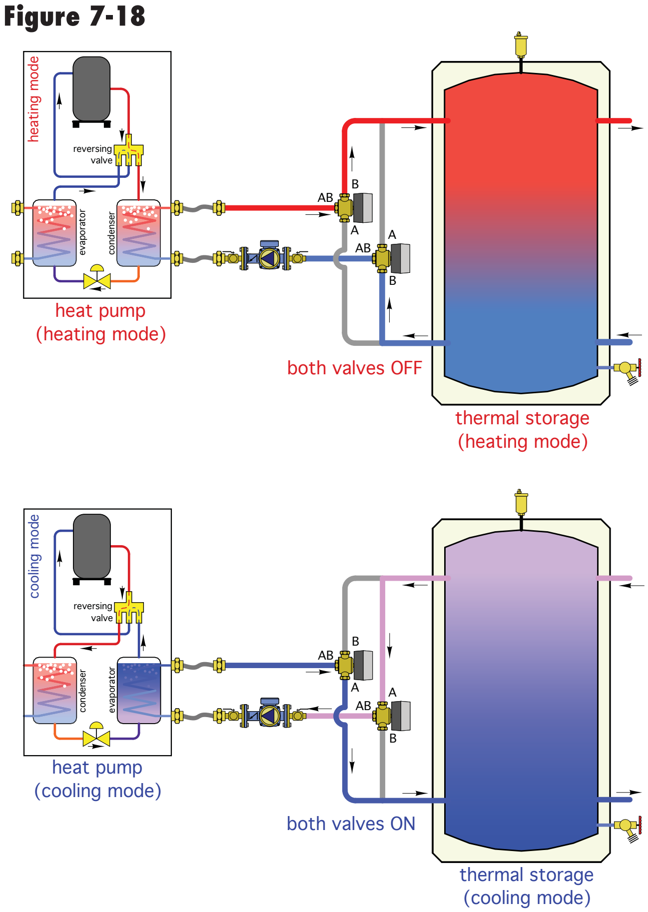
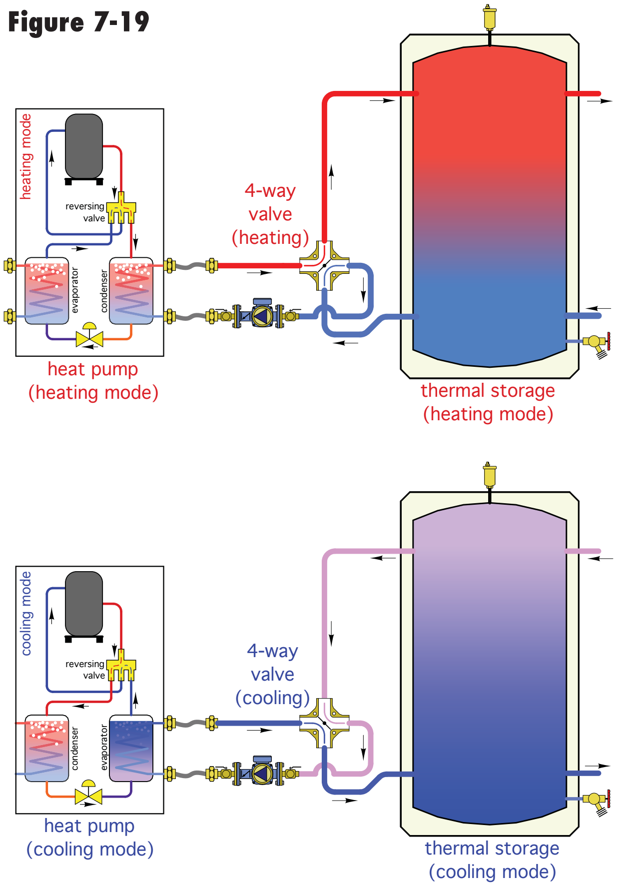


Figure 7-19



heated water to pass into the upper portion of the tank and cool water to be removed from the lower portion.

In cooling mode, both valves are turned on. Flow passes between the AB and A ports or vice versa. This allows chilled water from the heat pump to pass into the lower portion of the tank, while warmer water is returned to the heat pump from the upper portion of the tank.

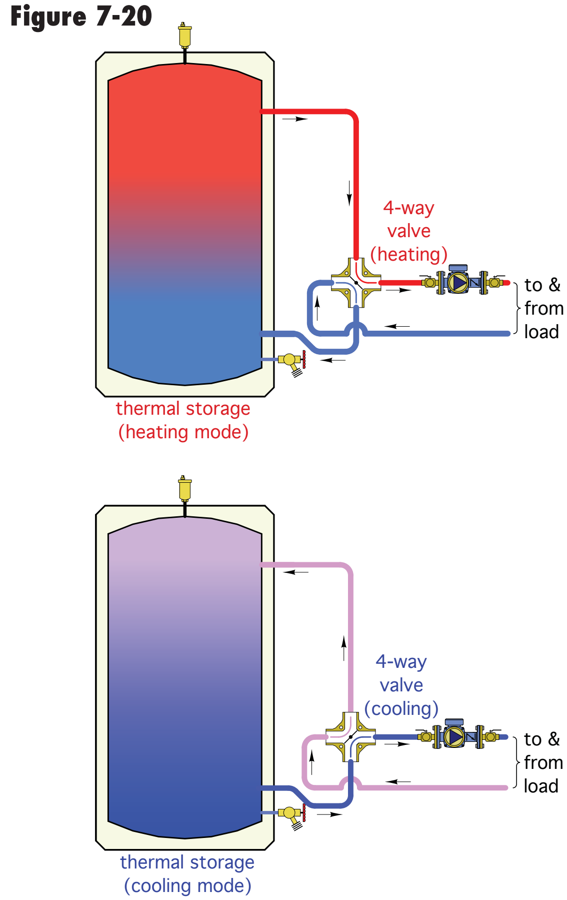
Note that the flow direction through the heat pump does not change.

It's is also possible to reverse the flow direction through a thermal storage tank using a single 4-way valve operated by a 2-position actuator, as shown in Figure 7-19.

The 2-position actuator needs to rotate the valve shaft 90° when changing from heating to cooling mode or vice versa. One option is to use a spring-return actuator that is powered on to move the shaft 90°, and then unpowered to allow the shaft to rotate in the opposite direction by 90°. Two-position spring return actuators are widely available, and can be configured for either 24 VAC or 120 VAC operation.

The combination of two 3-way diverting valves or a single 4-way motorized valve can also be used, when necessary, to reverse the flow direction through piping on the *load* side of the thermal storage tank. Figure 7-20 shows how this is done using a single 4-way valve with a 2-position 90° spring-return actuator.

Figure 7-20

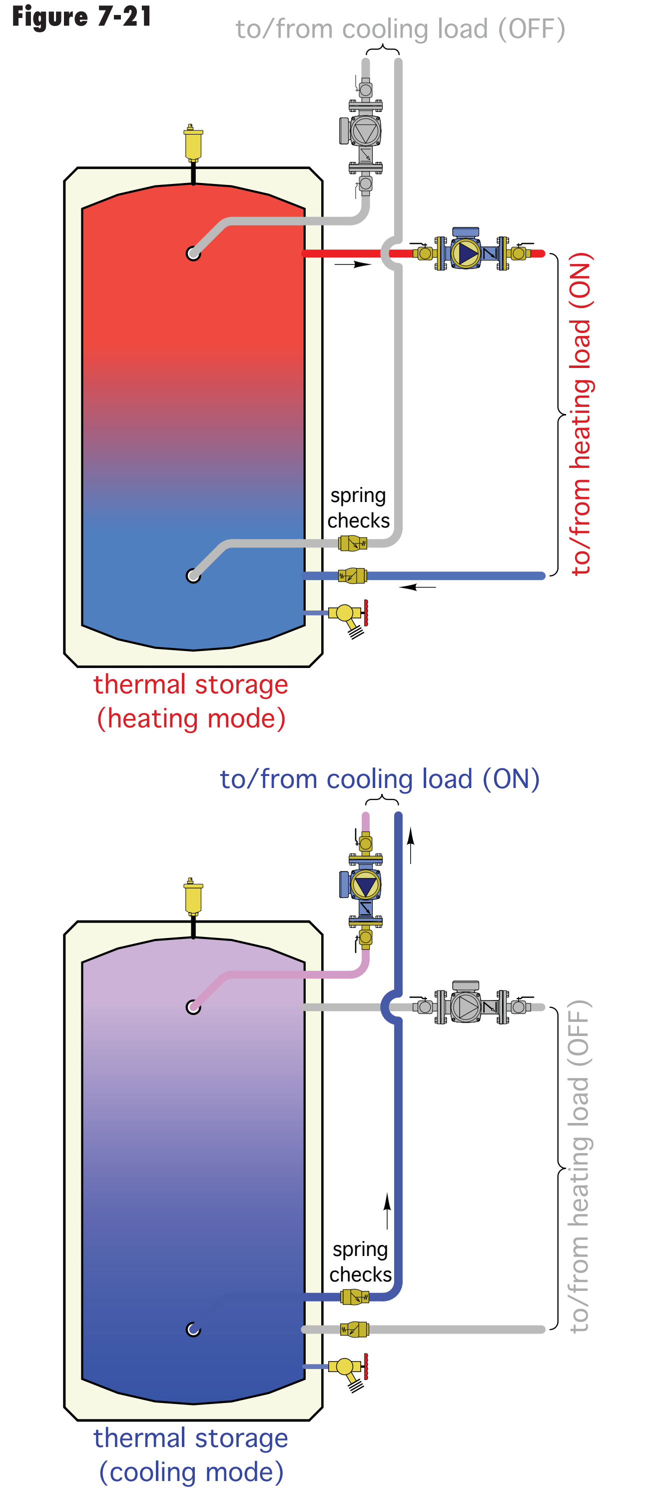


Be sure there are no check valves installed in any piping that will carry flow in both directions.

Flow reversal on the load side of a thermal storage tank maintains the same flow direction through terminal units that operate in heating as well as cooling mode, and in combination with a single distribution circulator that is used in both modes.

If separate terminal units are used for heating and cooling, and each subsystem has its own circulator, flow reversal on the load side of the thermal storage tank is not needed, as shown in Figure 7-21.

Figure 7-21



The use of ThermoCon tanks with multiple piping connections in the upper and lower portions of the tank makes it easy to create this piping scenario. Note that each circulator has an internal check valve, and that two spring-load check valves, with nominal 0.5 psi forward cracking pressure, are shown near the lower right side of the tanks. Their purpose is to minimize heat migration through piping that is inactive during one operating mode. Also be sure that all piping carrying chilled water is insulated and vapor-sealed to prevent condensation.

8. SYSTEM EXAMPLES

This section presents several examples of complete systems that use thermal storage in a variety of ways, and in combination with differential heat sources. Readers are encouraged to examine how the details presented in earlier sections are used in these systems.

EXAMPLE SYSTEM #1:

The system shown in Figure 8-1 uses a low-mass on/off (e.g., non-modulating) boiler to supply several zones of space heating as well as domestic hot water.

In space-heating mode, the boiler is operated by an outdoor reset controller that calculates the target supply water temperature for the distribution system. The minimum supply water temperature is 140ºF to prevent the boiler from operating with sustained flue gas condensation. The maximum supply water temperature is 180ºF.

Upon a call for space heating from any of the five zones, the associated zone valve opens, the system circulator (P2) is turned on and the outdoor reset controller is turned on through the 120/24 VAC transformer.

The outdoor reset controller measures the water temperature at the mid-height sensor in the buffer tank and compares it to the calculated target water temperature. If the temperature at the tank sensor is 10ºF or more below the target temperature, the boiler is fired. It continues to fire until the temperature at the tank sensor is 10ºF above the target water temperature. Thus, the temperature cycling range of the buffer tank is 20ºF.

Domestic water heating is provided through an indirect water heater that is operated as a priority load. Upon a call for domestic water heating, the diverter valve is turned on. An end switch in the diverter valve closes to call for boiler operation. The temperature in the boiler is controlled by the high limit controller, which is set for 180ºF. Flow passes from

Figure 8-1

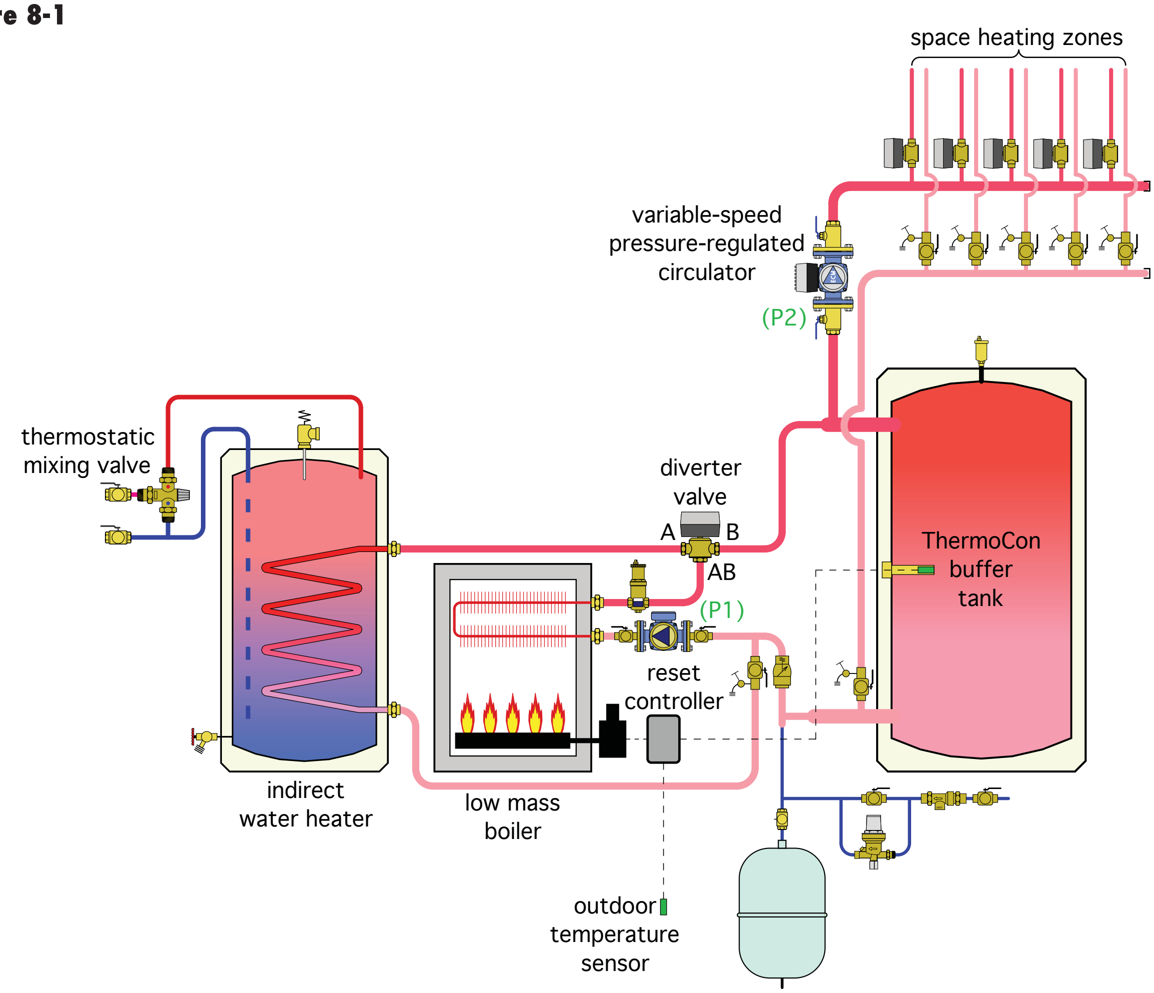
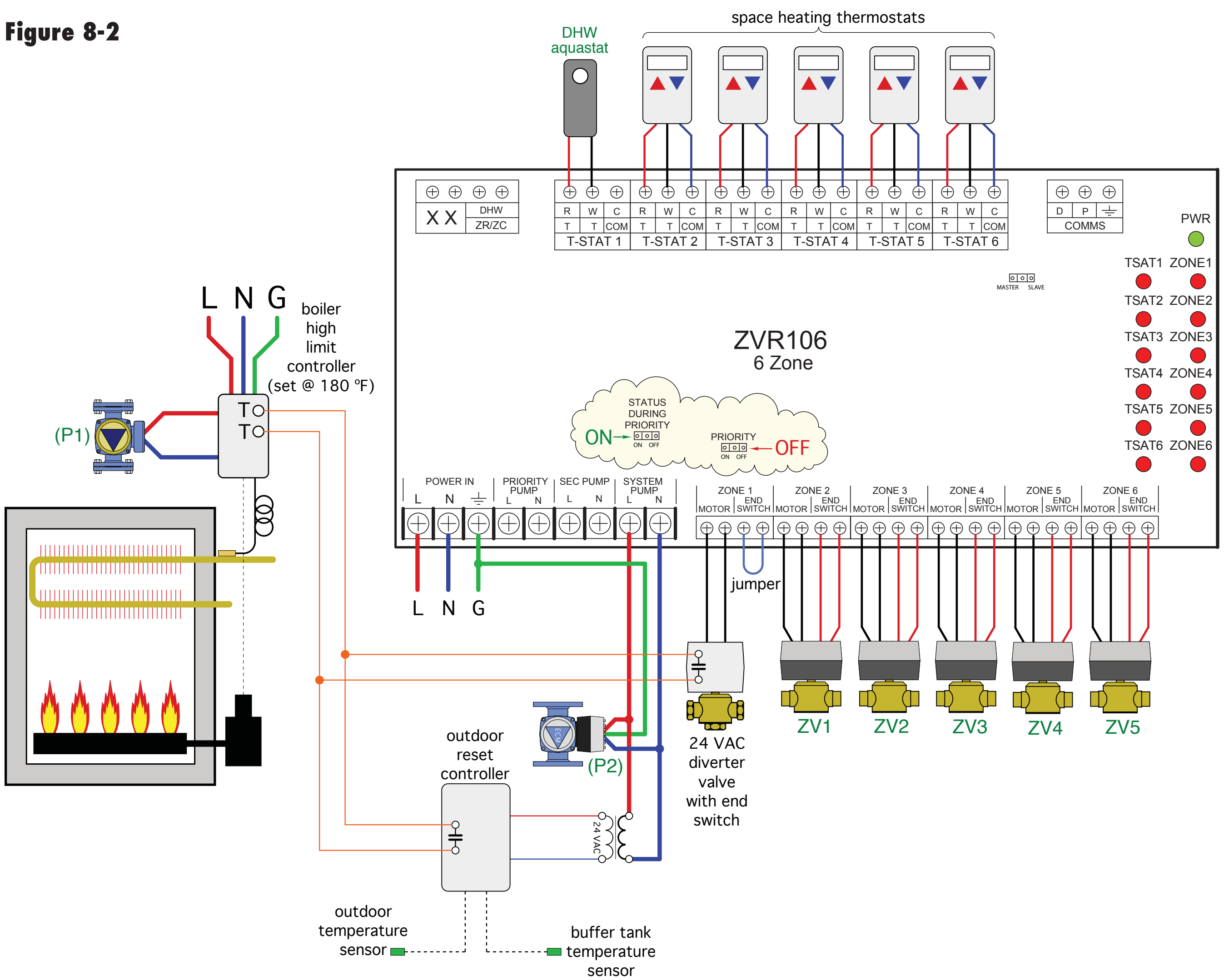


Figure 8-2



the (AB) to (A) port of this valve, through the heat exchanger in the indirect water heater, and back to the boiler.

When the domestic water mode is active, heat for any of the active zones can still be drawn from the buffer tank. This provides an advantage over standard piping methods used with priority control, which stops all heat flow to space heating during domestic water-heating mode.

The ThermoCon buffer tank uses a 2-pipe configuration and has been sized to allow the boiler to operate for a minimum of 10 minutes when the space-heating zone with the lowest heating load is operating.

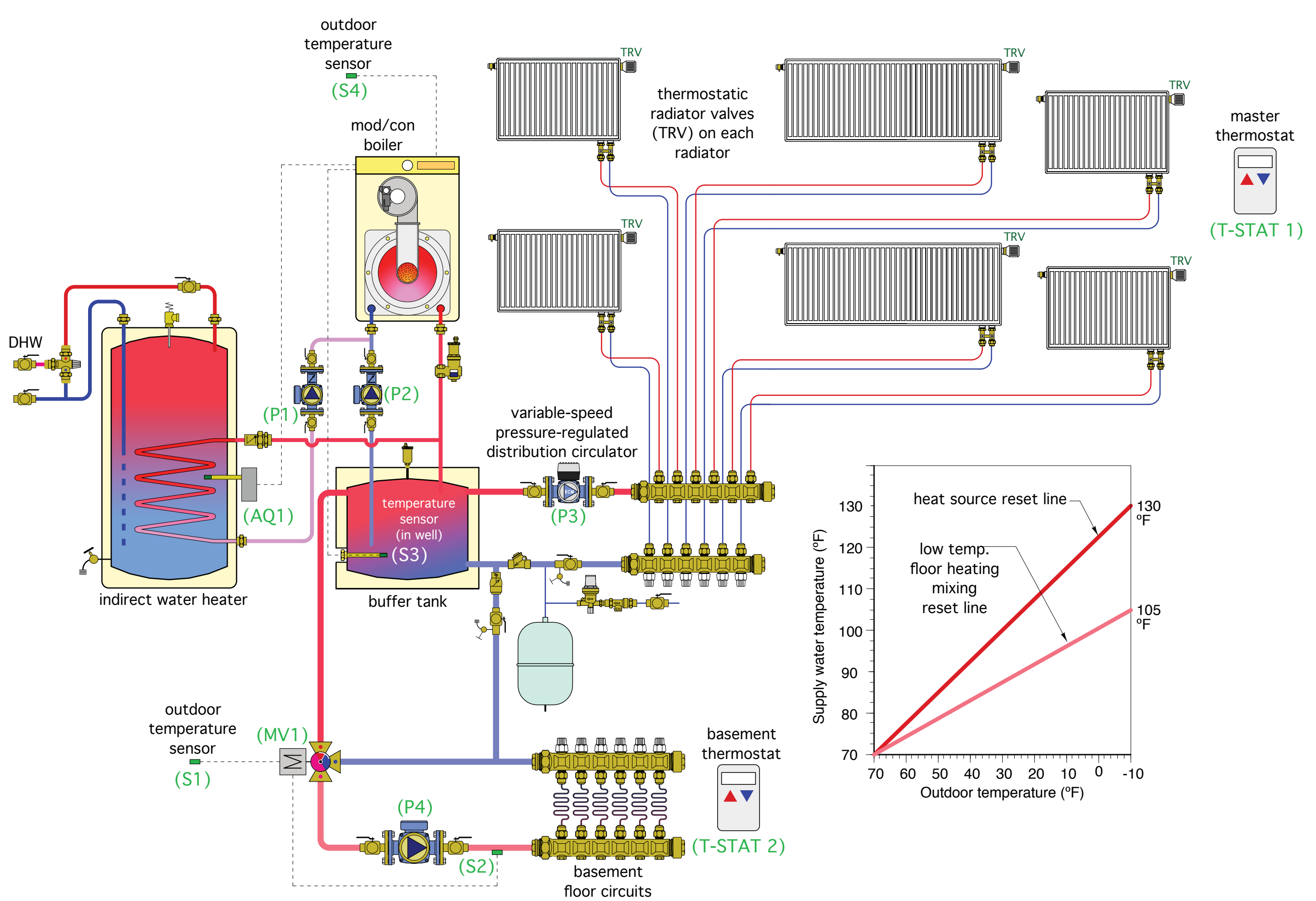
Figure 8-2 shows the wiring for the system shown in Figure 8-1.

EXAMPLE SYSTEM #2:

Figure 8-3 shows the piping for a system that supplies a single zone of radiant floor heating to a basement slab, along with several panel radiators on the main floor of a house.

Each panel radiator is equipped with a thermostatic radiator valve to allow room-by-room zoning. During space-heating mode, the mod/con boiler operates on its internal outdoor reset control, which has been set to produce a supply water temperature of 130ºF when the outdoor temperature is -10ºF. The low-temperature water for floor heating is produced by mixing down water from the buffer tank using a 3-way motorized mixing valve. This valve operates on 24 VAC and has its own outdoor reset logic. It is set to supply 105ºF water to the floor when the outdoor temperature is -10ºF. The reset lines for the radiant floor and panel radiators are shown on the piping schematic.

Figure 8-3



A small 25-gallon ThermoCon buffer tank provides thermal mass to stabilize the mod/con boiler against short cycling due to the small room-by-room heating loads represented by the independently controlled panel radiators.

Domestic hot water is produced by an indirect water heater, which is supplied as a priority load from the boiler. The boiler temporarily goes to a higher setpoint temperature during this mode of operation.

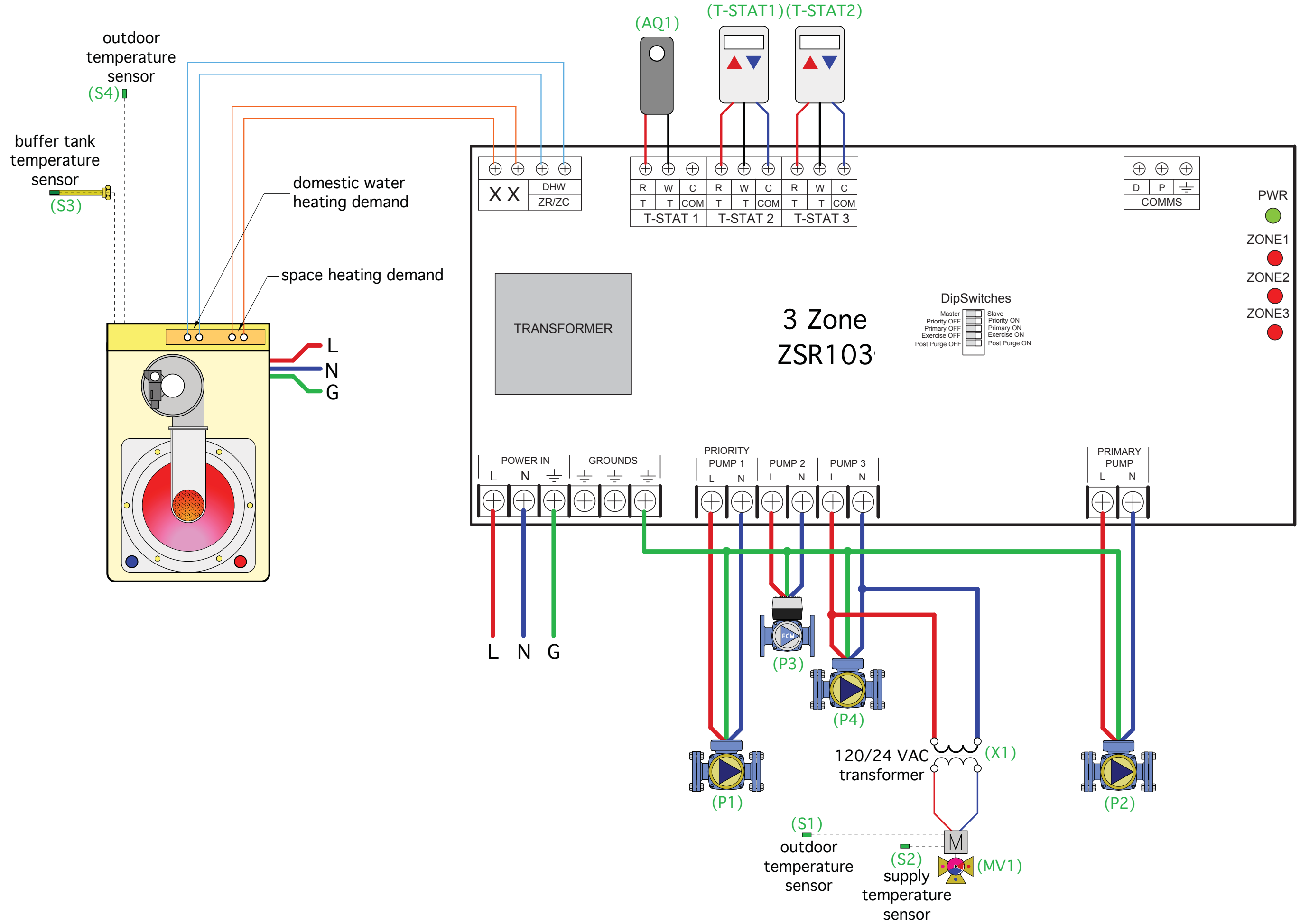
Figure 8-4 shows how the electrical controls for this system are implemented using a Caleffi ZSR103 multi-zone relay center as the primary control device.

The DIP switch on the ZSR103 has been set for priority operation of zone 1 (e.g., domestic water heating). Other DIP switches on the ZSR103 controller have also been set so the "primary pump" (P2) is temporarily turned off during priority mode operation. Circulator (P3) for the panel radiator circuits is turned on whenever there is a call from the main floor master thermostat (THM1). Circulator (P4) for the radiant floor circuits is turned on whenever there is a call from the basement thermostat (THM2). A 120/24 VAC transformer is wired in parallel with circulator (P4). It supplies 24 VAC power to operate the motorized mixing valve whenever the basement zone is active.

The following is a description of operation of example system #2:

1. Domestic Water Heating: *The temperature of the domestic hot water storage tank is monitored by aquastat (AQ1), which has been adjusted to close its contacts when the water temperature in the tank drops to 110ºF and open its contacts when the tank temperature reaches 125ºF. When aquastat (AQ1) calls for heat, it turns on the priority*

Figure 8-4

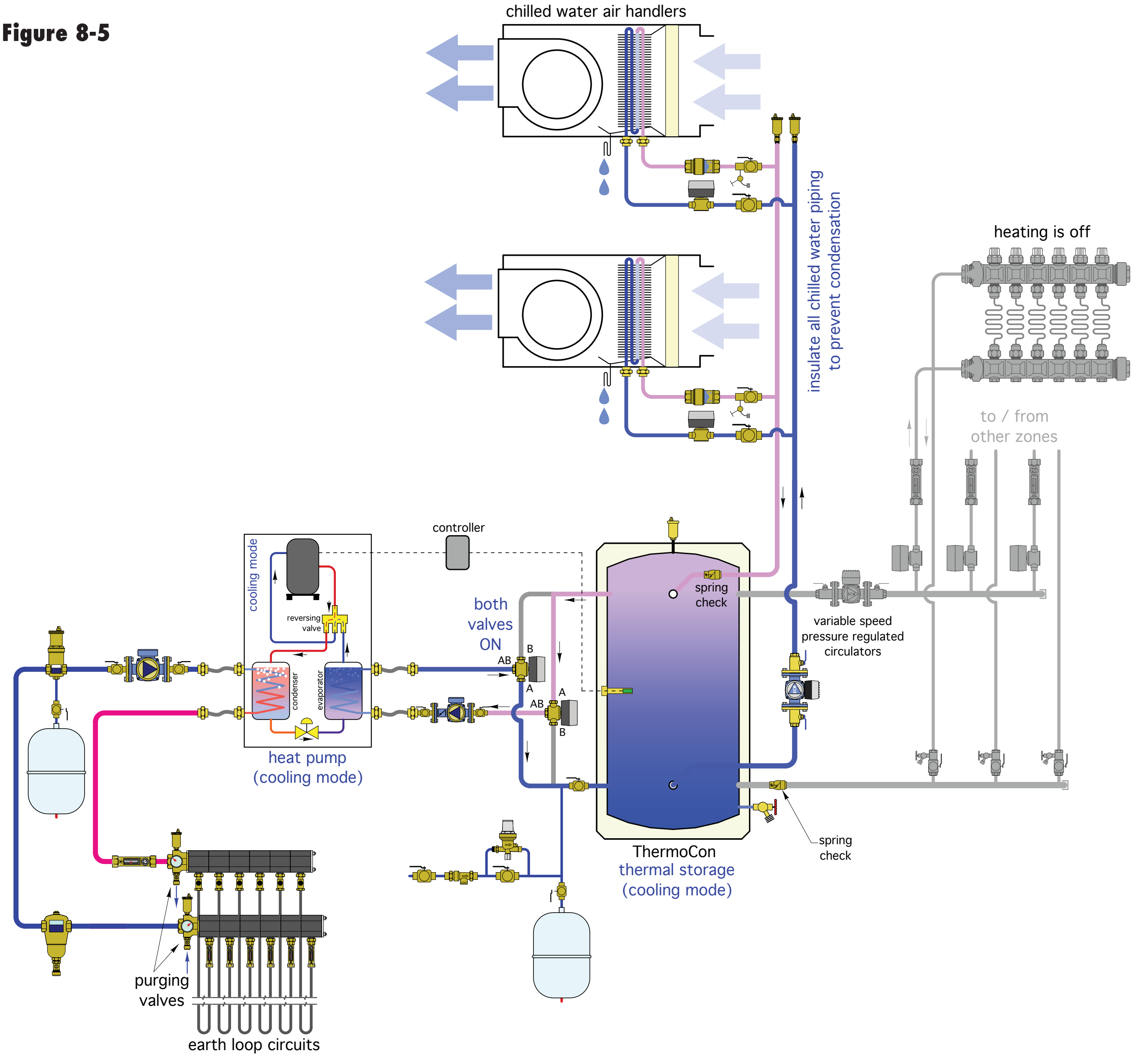


zone 1 of the multi-zone relay center (ZSR103). Circulator (P1) is turned on. A dry contact closes across the (ZR ZC) terminals in the (ZSR103) to provide a domestic water-heating demand to the mod/con boiler. The boiler fires and operates based on its own internal setpoint for the domestic water-heating mode. Any space-heating loads that were on are temporarily turned off during the priority domestic water-heating mode. When aquastat (AQ1) is satisfied, its contacts open. This turns off zone 1 circulator (P1) and opens the (ZR ZC) contact, which turns off the DHW demand to the boiler.

2. Basement Floor Heating: *Upon a call for heating from thermostat (T-STAT2), the multi-zone relay center (ZSR103) turns on circulator (P4) to provide flow through the basement floor circuits. A 120/24 VAC transformer (X1) is also turned on to supply 24 VAC to the motorized mixing valve controller (MV1). When powered on, the mixing valve controller (MV1) operates on outdoor reset control based on its internal settings and modulates to control the supply water temperature to the floor circuits. The (X X) terminals in the (ZSR103) close to provide a space-heating demand to the boiler. The boiler checks the temperature of sensor (S3) within the buffer tank. When necessary, it fires to maintain this temperature based on its own outdoor reset control settings. When thermostat (THM2) is satisfied, circulator (P4), motorized mixing valve (MV1) and the boiler are turned off.*

3. Panel Radiator Heating: *The master thermostat (T-STAT1) on the main floor is set to a temperature 2ºF above the normal comfort temperature. It is used to initiate a call for heating for all areas served by panel radiators. Upon a call for heating from (THM1), the multi-zone relay center (ZSR103) turns on circulator (P3) to provide flow through the manifold station supplying the circuits to each*

Figure 8-5



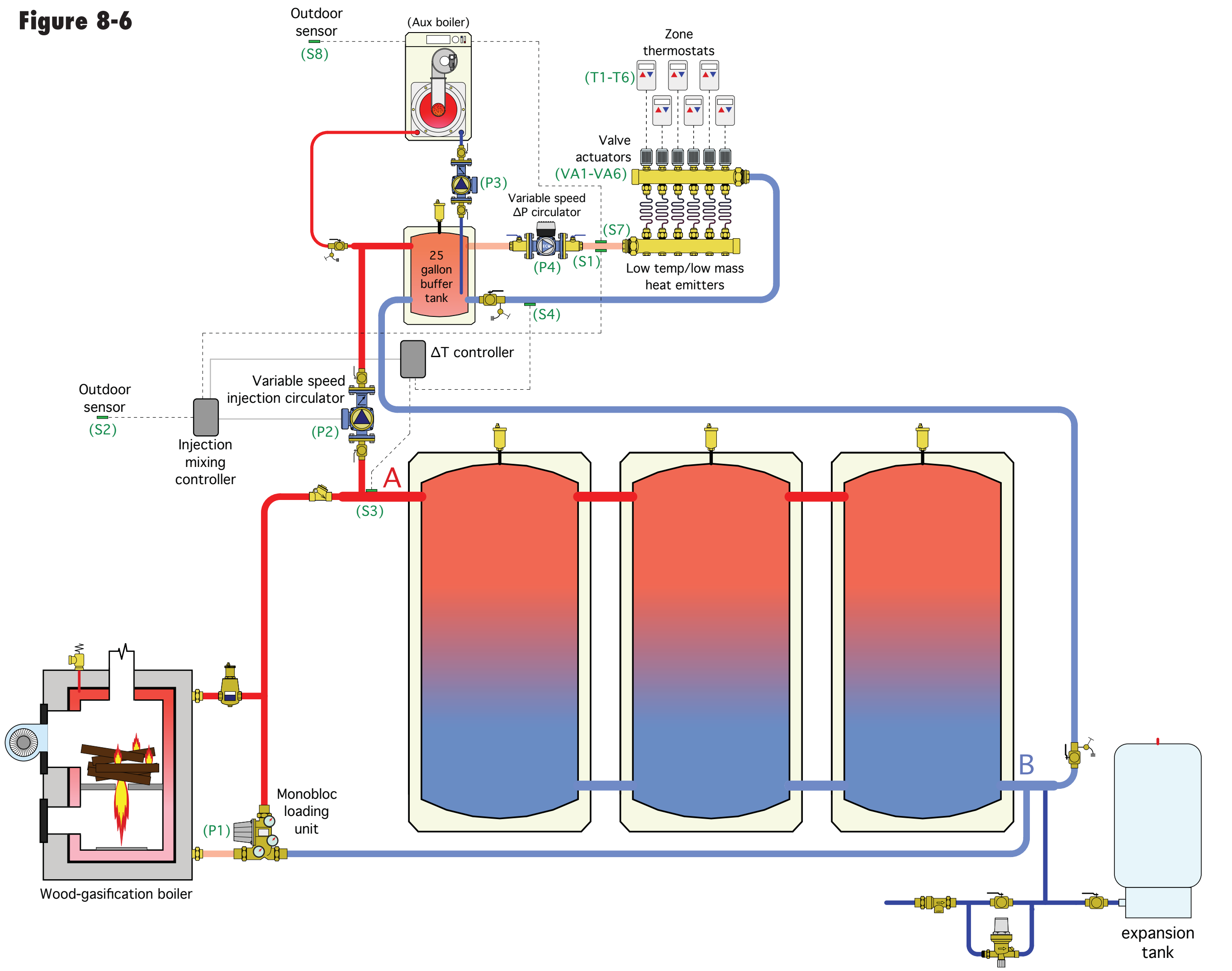
panel radiator. Flow to each radiator is determined by the setting of its thermostatic radiator valve. Circulator (P3) is a variable-speed pressure-regulated circulator set for constant differential pressure mode. The (X X) terminals in the (ZSR103) close to provide a space-heating demand to the boiler. The boiler monitors the temperature of sensor (S3) within the buffer tank. When necessary, it fires to maintain this temperature based on its own outdoor reset control settings. When master thermostat (T-STAT1) is satisfied, circulator (P3) and the boiler are turned off.

EXAMPLE SYSTEM #3:

Geothermal water-to-water heat pumps can supply warm water for heating and chilled water for cooling. Most of these heat pumps are on/off devices. As such, they cannot adjust their heating or cooling capacity to match the load requirements of zoned distribution systems. A buffer tank should be used between the heat pump and zoned distribution systems. Figure 8-5 shows one way to do this.

A pair of diverter valves between the heat pump and buffer tank control the flow direction through the buffer tank. Both of these valves are de-energized in heating

Figure 8-6



mode. Heat water from the heat pump passes into the upper portion of the buffer tank, while cooler water exits from the lower portion of the tank, and is routed back to the heat pump.

In cooling mode, both diverter valves are energized to reverse the flow direction through the buffer tank.

The heating and cooling distribution subsystems use zone valves for each zone circuit. Each subsystem uses a variable-speed pressure-regulated circulator that adjusts speed in response to opening or closing of zone valves. Each of these circulators has been sized to the specific flow and head requirements of its subsystem.

The multiple connections on the ThermoCon buffer tank allow simple connections for both subsystems. It also provides excellent hydraulic separation between the heat pump circulator and the variable-speed distribution circulators. Spring-loaded check valves are used to minimize heat migration into inactive portions of the system during each mode of operation.

Balancing valves (automatic and manually adjusted models) are used to set the flow rate through each of the zone circuits.

EXAMPLE SYSTEM #4:

Wood-gasification boilers attain their highest efficiency and lowest emissions while burning a full load of cordwood as fast and hot as possible. Heat production rates often exceed the rate at which the building requires heating. Thus, substantial thermal storage is necessary for optimal performance.

Figure 8-6 shows how this storage can be provided using multiple buffer tanks piped in a “close coupled” hybrid arrangement.

When the wood-gasification boiler is fired, the Caleffi Monobloc loading unit is turned on. This device contains a thermostatic mixing mechanism and circulator. It recirculates water through the boiler until its inlet temperature reaches approximately 140ºF. This allows the boiler to warm quickly and minimizes flue gas condensation. Once the desired boiler inlet temperature is achieved, the loading unit routes water from the boiler to the header connected to the tank array at point (A). If there is no concurrent heating load, all flow passes into the thermal storage tank array. Flow through the array is balanced, so that the thermocline moves downward through each tank as heat is added. Cooler water exits the tank array at point (B) and flows back to the Monobloc unit and eventually into the boiler.

When there is a heating demand from any of the 6 zones, the differential temperature controller is turned on to measure the difference between the temperature at sensor (S3) at point (A) and sensor (S4) on the return side of the distribution system. If the temperature at sensor (S3) is at least 5ºF higher than the temperature at sensor (S4), the differential temperature controller enables operation of an injection-mixing controller. This controller uses the outdoor temperature, along with its settings, to calculate the target supply water temperature for the distribution system. It then adjusts the speed of the injection circulator (P2), while measuring the supply temperature at sensor (S1). The controller’s objective is to keep the temperature at sensor (S1) as close to the calculated target temperature as possible.

If the injection-mixing controller cannot sustain the necessary supply water temperature, it closes a set of internal dry contacts to enable operation of the auxiliary boiler. This boiler injects heat into a small (25-gallon) ThermoCon buffer tank, from which heated water can supply the distribution system. The small buffer protects the auxiliary boiler from short cycling, and it provides hydraulic separation between the auxiliary boiler circulator (P3), the variable-speed distribution circulator (P4) and the injection circulator (P2).

While the auxiliary boiler is operating, the differential temperature controller continues to monitor the temperature difference between sensors (S3) and (S4). If the temperature at sensor (S3) drops to 3ºF or less *above* the temperature at sensor (S4), the storage tank array is deemed to be fully discharged, and the injection circulator (P2) is turned off. This prevents heat generated by the auxiliary boiler from being transferred into the storage tank array.

When the wood-gasification boiler is subsequently fired, an increase in temperature at sensor (S3) will eventually repeat the process for injecting heat into the distribution system. If the wood-gasification boiler is off for an extended time, the auxiliary boiler, in combination with the small buffer tank, and the distribution system provide heat to the building, operating essentially as an “isolated” system.

EXAMPLE SYSTEM #5:

For optimal performance, pellet-fired boilers require thermal storage. They also perform better when applied to “base loads,” which typically represent 80–95% of the total heating energy use over a full heating season. An auxiliary boiler can be used to supplement the pellet-fired boiler for peak load situations, while also providing some capacity to cover situations when the pellet-fired boiler may be down for maintenance.

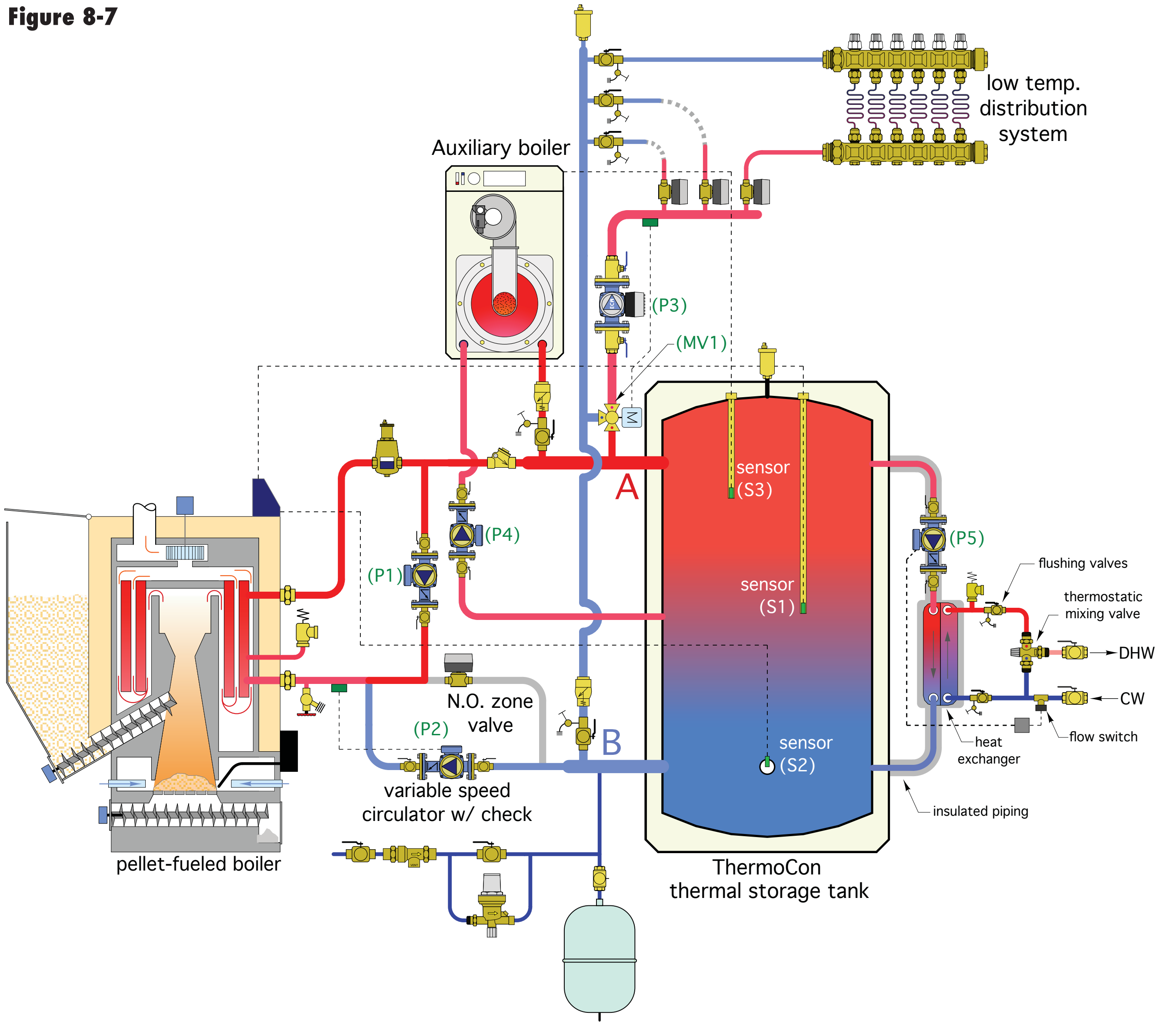
The schematic in Figure 8-7 shows a system that uses this approach.

Upon a call for space heating from any of the zones, circulator (P3) and mixing valve (MV1) are turned on. Water from the upper storage tank header (A) flows into the mixing valve, where it is mixed with return water to achieve the desired supply temperature. The pellet-fired boiler is also enabled and begins monitoring storage tank sensor (S1). If the temperature at sensor (S1) drops below a preset limit (such as 110ºF), the pellet-fired boiler initiates combustion, circulator (P1) turns on, and circulator (P2) is enabled to operate. Circulator (P2) monitors the boiler inlet temperature and only begins moving water from the lower tank header (B) into the boiler when the boiler’s inlet temperature climbs above 130ºF. This minimizes flue gas condensation within the boiler. The speed of circulator (P2) increases as the inlet temperature to the boiler increases.

Circulator (P2) is equipped with an internal spring-loaded check valve that provides sufficient forward cracking pressure to prevent any portion of the flow returning from the distribution system from passing through the pellet-fired boiler when it is not firing and circulator (P2) is off. If this flow were allowed to occur, it would create heat loss from the boiler’s jacket and induce convective airflow up its chimney.

Hot water from the pellet-fired boiler is routed to the upper tank header (A), where it can either be drawn into

Figure 8-7



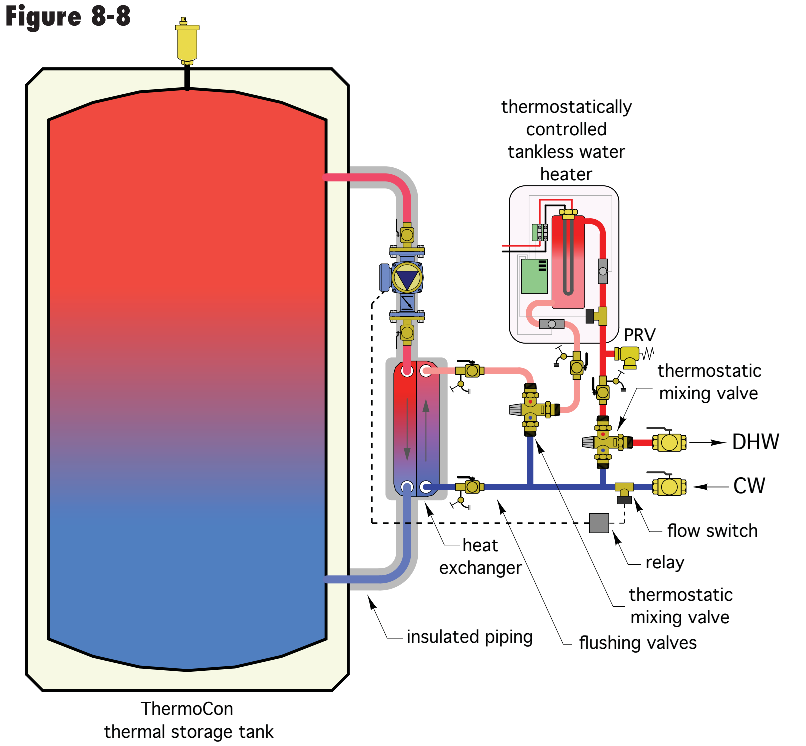
the distribution system by circulator (P3), or pass into the upper portion of the thermal storage tank.

The pellet-fired boiler continues to operate until the temperature at the lower tank sensor (S2) reaches 170ºF. This occurs even if all zone loads turn off before this condition is met. This “stacks” the tank nearly full of higher temperature water, which can be immediately used at the next call for heat. This action also lengthens the boiler cycle to improve efficiency and reduce emissions.

If a power failure occurs, the normally open zone valve above circulator (P2) opens to provide an unobstructed path around the spring-loaded check valve in circulator (P2). A thermosiphon flow can carry any residual heat from the boiler to the thermal storage tank. This zone valve remains closed at all other times.

The auxiliary boiler continuously monitors sensor (S3) in the upper portion of the storage tank. It fires when the temperature at this sensor drops to 120ºF and remains on until the temperature at sensor (S3) reaches 130ºF. *This is necessary to ensure that heat is always available for production of domestic hot water.* The height of sensor (S3) in the tank, in combination with temperature stratification, only allows the auxiliary boiler to interact with water in the upper portion of the tank, which provides sufficient thermal mass to prevent the auxiliary boiler from short

Figure 8-8



cycling. The thermal mass also stabilizes heat delivery to the domestic water-heating subassembly.

Domestic water is heated "on demand" by the subassembly connected to the right side of the thermal storage tank. A flow switch detects a domestic hot water demand of 0.6 gpm or higher and turns on circulator (P5) through a relay. Hot water from the upper portion of the storage tank passes through the primary side of the heat exchanger, and transfers heat to the domestic water passing through the other side. The temperature of the domestic water leaving the heat exchanger depends on the temperature of the thermal storage tank. At times, it may be heated well above the desired delivery temperature. At other times, it will only be heated to about 110ºF based on the temperature that the auxiliary boiler maintains in the upper portion of the thermal storage tank. The heated domestic water passes through a thermostatic mixing valve that reduces its temperature to no more than 110ºF.

Figure 8-8 shows how the domestic water-heating assembly can be modified by including a thermostatically controlled electric tankless water heater.

This heater is sized to boost the domestic water temperature to the desired setpoint based on the least amount of preheating from thermal storage. *If used, this heater allows the auxiliary boiler to remain off other than when required for space heating.* It also allows the auxiliary boiler to operate at the lowest possible water temperatures based on outdoor reset control, rather than maintaining a minimum temperature of 120 to 130ºF in the upper portion of the tank solely for domestic water heating.

EXAMPLE SYSTEM #6:

Some building owners prefer systems with two independent heat sources. A good example is when a geothermal heat pump is combined with a gas-fired boiler. Such a system can take advantage of favorable electrical energy pricing during certain times of the day. It provides the security that one heat source can cover some of the heating load if the other heat source is down for maintenance. This approach also allows the water-to-water heat pump to be sized for less than the building's design heating load. The latter may be necessary due to limited land area for installation of the earth loop or high installation cost.

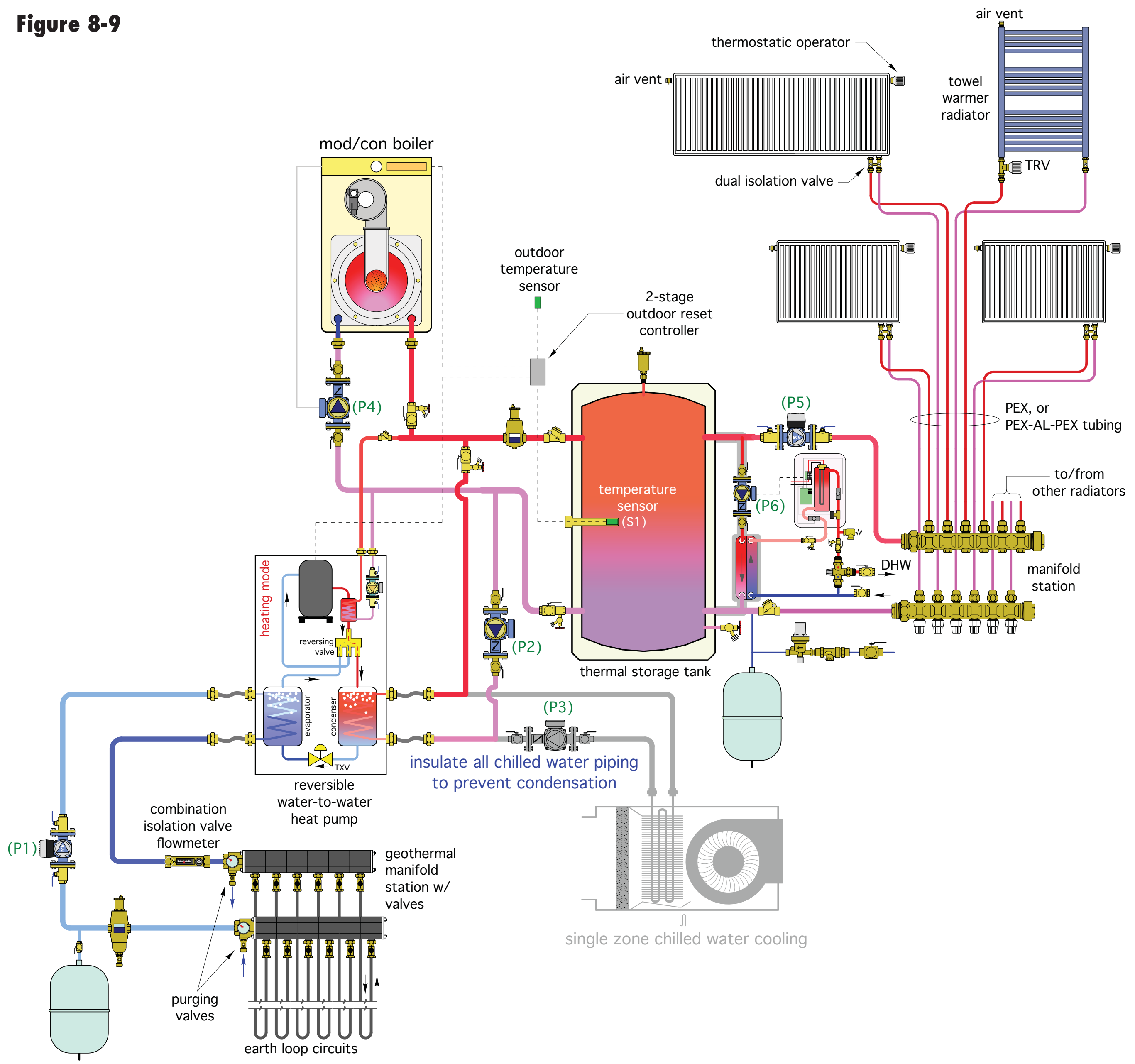
Thermal storage can be an integral part of such a system, as shown in Figure 8-9 (in heating-mode operation).

This system uses a closed, slightly pressurized earth loop, with a valved manifold station as the beginning and ending point for all parallel earth loop circuits. The earth loop is equipped with an expansion tank, as well as a combined air & dirt separator. Flow is provided by a single ECM-based high-efficiency circulator.

The thermal storage tank receives heat from either the heat pump or the gas-fired modulating/condensing boiler, protecting both from short cycling due to low loading from a highly zoned distribution system. It also allows both heat sources to operate simultaneously if needed under high-demand situations. Finally, this tank provides hydraulic separation between the various circulators used in the system.

The desuperheater within the geothermal heat pump adds heat to the thermal storage tank whenever the heat pump's compressor is operating. This heat can be used by the "on-demand" domestic water-heating subsystem. When the heat pump is operating in cooling mode, the heat transferred to the buffer tank from the desuperheater is truly "free heat" that would otherwise be dissipated to the earth loop. This allows the earth loop to operate at

Figure 8-9



slightly lower fluid temperatures, which slightly improves the EER of the heat pump. When the heat pump operates in heating mode, heat is transferred from the refrigerant to system water through the condenser as well as the desuperheater. This slightly increases the heat pump's COP, compared to its normal mode of operation in which only the condenser is removing heat from the refrigerant.

Space heating is provided by multiple panel radiators, sized to operate at supply water temperatures no higher than 120ºF at design load conditions. Each panel radiator is equipped with an integral valve that is automatically adjusted by a thermostatic operator, to allow room-by-room comfort control. All panel radiators are supplied through a homerun distribution system using 1/2-inch PEX or PEX-AL-PEX tubing from a common manifold station. Flow is created by a variable-speed pressure-regulated circulator set for constant differential pressure operation.

Operation of space-heating distribution circulator (P5) is controlled by the master thermostat (T1). If this thermostat is satisfied, circulator (P5) is turned off, and no heat flows to the distribution system. This allows the entire building to be put into a reduced temperature setback mode from a single thermostat. During the heating season, the setting of the master thermostat should be slightly above the normal desired air temperature. This maintains circulator (P5) in operation and allows the individual thermostatic radiator valves to "fine-tune" the comfort level in their respective spaces.

Cooling is provided by a *single* chilled-water air handler that has been matched to the cooling capacity of the heat pump. Thus, *the thermal storage tank is not used in the cooling mode.* The air handler is equipped with a drip pan and drain to dispose of condensate that forms on its coil. All portions of the piping system that handle chilled water are insulated and vapor-sealed to prevent condensation.

Both heat sources are operated by a 2-stage outdoor reset controller. Whenever there is a heating demand from the master thermostat, this controller calculates the target water temperature required by the space-heating distribution system. When heat input to the thermal storage tank is needed, the controller operates the heat pump as the first stage heat source. If the heat output from the heat pump is sufficient to keep the thermal storage tank temperature at or close to the target temperature, the boiler will not operate. However, if heat output from the heat pump cannot keep pace with the rate of heat removal from the buffer tank, the boiler is be turned on as the second stage heat source. In this scenario, both the heat pump and boiler can simultaneously add heat to the buffer tank.

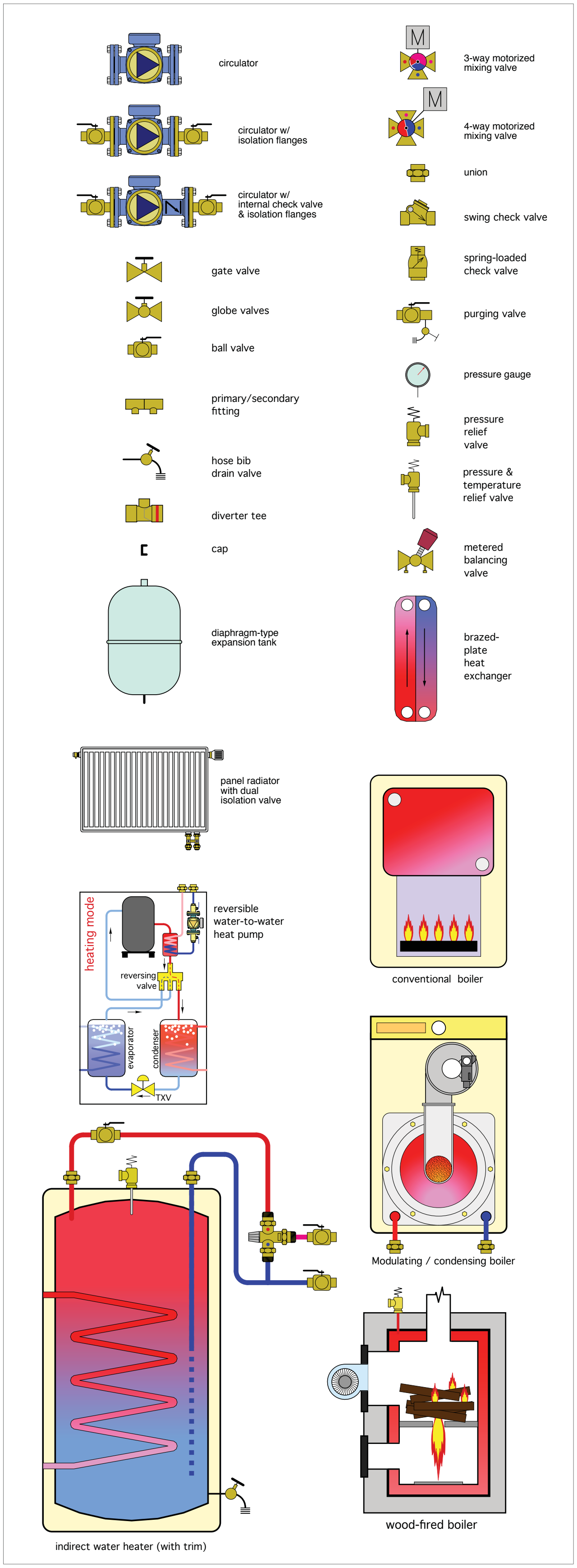
Domestic water is heated by the assembly previously shown in Figure 7-5. Whenever the demand for domestic hot water reaches 0.6 gallons per minute, circulator (P6) is turned on by a flow switch and associated relay. Heated water from the upper portion of the thermal storage tank flows through the primary side of heat exchanger (HX1), while cold domestic water flows into the secondary side of this heat exchanger. The temperature of the domestic water leaving the heat exchanger depends on the water temperature in the upper portion of the thermal storage tank. A thermostatically controlled electric tankless water heater provides any necessary temperature boost. A thermostatic 3-way mixing valve keeps the water temperature leaving this assembly no higher than 120ºF.

SUMMARY:

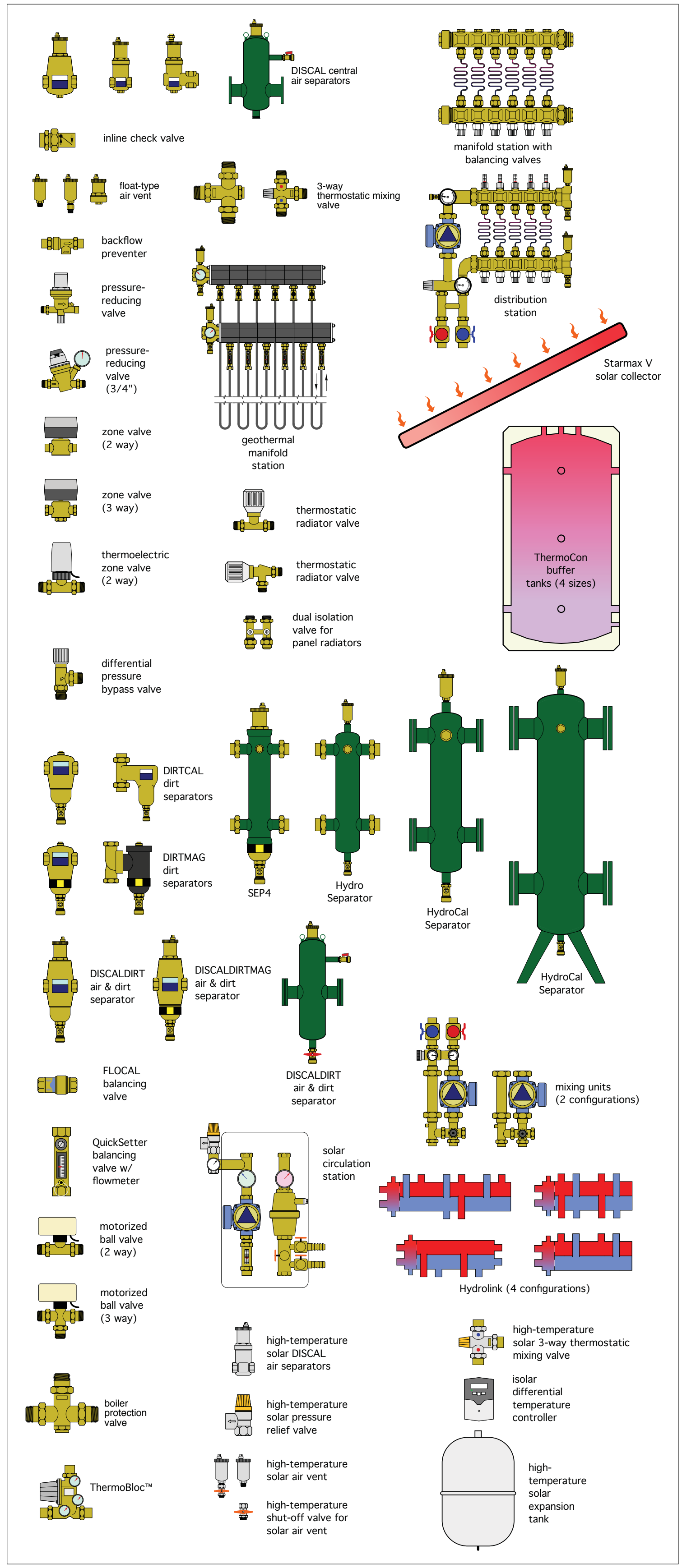
This issue of *idronics* has discussed several benefits of thermal storage in hydronic heating and cooling systems—from stabilizing boilers and heat pumps against short cycling, to taking advantage of off-peak electrical rates, to providing hydraulic separation between circulators. It has also given methods for sizing buffer tanks based on the operating characteristic of the heat source and the zoning of the distribution system. Further information related to volume and surface area calculations is given in Appendix C. Designers are encouraged to use the "best practices" presented, including generous tank insulation and design details that enhance temperature stratification.

APPENDIX A: PIPING SYMBOL LEGEND

GENERIC COMPONENTS



CALEFFI COMPONENTS

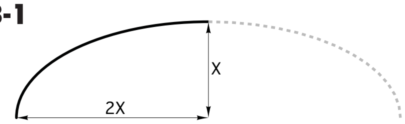


APPENDIX B: VOLUMES AND SURFACE AREAS OF VERTICAL CYLINDRICAL TANKS

The most common tank shape used in HVAC systems is a cylinder. For unpressurized tanks, this cylinder usually has a flat bottom and a flat top. The water level in unpressurized tanks usually does not reach the upper end of the cylinder-shaped shell.

For pressurized tanks, the most common shape is a cylinder with semi-elliptical top and bottom domes. The top and bottom domes are designed to withstand significant internal pressure. The most common semi-elliptical dome has an aspect ratio of 2:1, as illustrated in Figure B-1.

Figure B-1

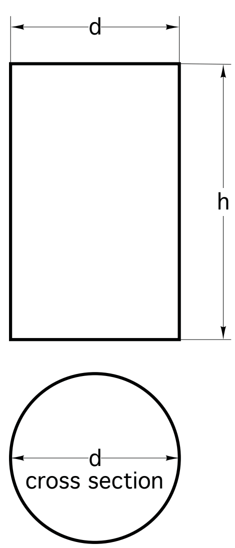


Some pressurized tanks are also made with flat tops and bottoms. They use a grid of internal steel stay rods running between and welded to the top and bottom plates to prevent excessive deflection under pressure.

VOLUME AND TOTAL SURFACE AREA OF A FLAT-ENDED VERTICAL CYLINDRICAL TANK

The following formulas can be used to calculate the volume and total surface area of a flat-ended cylindrical tank, as shown in Figure B-2.

Figure B-2



Volume:

Formula B-1:

$$V = \frac{\pi d^2 h}{924}$$

Where:
V = volume (gallons)
d = internal diameter (inches)
h = internal height (inches)

Total surface area (side, top, & bottom):

Formula B-2:

$$A_s = \pi d(h + 0.5d)$$

Where:
A_S = Total surface area (in^2)
d = outside diameter (inches)
h = outside height (inches)

The surface area calculated with Formula B-2 can be converted from square inches to square feet by dividing by 144.

Example: Estimate the volume of a flat-ended cylindrical tank with an inside diameter of 34 inches and an internal height of 60 inches.

Solution:

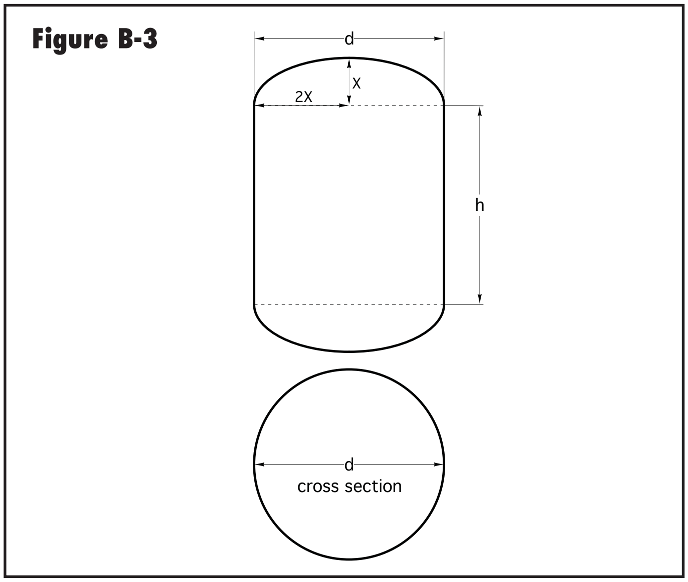
$$V = \frac{\pi d^2 h}{924} = \frac{\pi (34)^2 60}{924} = 235.8\,gallon$$

$$A_s = \pi d(h + 0.5d) = \pi[34](60 + 0.5[34]) = 8{,}225 in^2 = 57.1 ft^2$$

If the tank is only partially filled, the volume of fluid it contains can be calculated with Formula 8-1, with the value of h equal to the depth of fluid in the tank.

VOLUME AND TOTAL SURFACE AREA OF A VERTICAL CYLINDRICAL TANK WITH SEMI-ELLIPTICAL HEADS OF 2:1 ASPECT RATIO

The following formulas can be used to calculate the volume and total surface area of a cylindrical tank with semi-elliptical top and bottom domes, where the ellipse has an aspect ratio of 2:1, as shown in Figure B-3.



Volume:

Formula B-3:

$$V = \pi d^2 \left[\frac{h}{924} + 0.0003608(d) \right]$$

Where:

V = volume (gallons)

d = internal diameter (inches)

h = height of cylindrical portion of tank (inches)

Total surface area (side, top, & bottom):

Formula B-4:

$$A_s = 2.171(d^2) + \pi h d$$

Where:

A_s = Total surface area (in^2)

d = internal diameter (inches)

h = height of cylindrical portion of tank (inches)

The surface area calculated with Formula B-4 can be converted from square inches to square feet by dividing by 144.

Example: Estimate the volume and total surface area for a semi-elliptical headed tank (assuming 2:1 ratio ellipse) that has an *internal* diameter of 48 inches and a cylindrical shell height of 60 inches. Assume the tank shell is 0.25 inches thick.

Solution: The inside diameter of the tank is used for the volume calculation, whereas the outside diameter of the tank is used for the surface area:

$$V = \pi d^2 \left[\frac{h}{924} + 0.0003608(d) \right] = \pi (48)^2 \left[\frac{60}{924} + 0.0003608(48) \right] = 595\, gallons$$

$$A_s = 2.171(d^2) + \pi h d = 2.171(48.5^2) + \pi (60)(48.5) = 14{,}249 in^2 = 98.9 ft^2$$

APPENDIX C: ESTIMATING HEAT LOSS FROM CYLINDRICAL STORAGE TANKS

The instantaneous rate of heat loss from a vertical *cylindrical* tank with a *flat top and bottom* can be *approximated* using Formula C-1:

Formula C-1:

$$Q = \left[\left(\frac{2\pi k L}{\ln\left(\frac{d_o}{d_i}\right) + \frac{1.36k}{d_o}} \right) + \frac{\pi d_o^2}{2R_{tb}} \right] (T_w - T_a)$$

Where:

Q = instantaneous rate of heat loss from tank (Btu/hr)

d_o = outer diameter of tank insulation (ft)

d_i = inner diameter of tank insulation (ft)

L = height of cylindrical tank (ft)

k = thermal conductivity of tank sidewall insulation (Btu/°F•hr•ft)

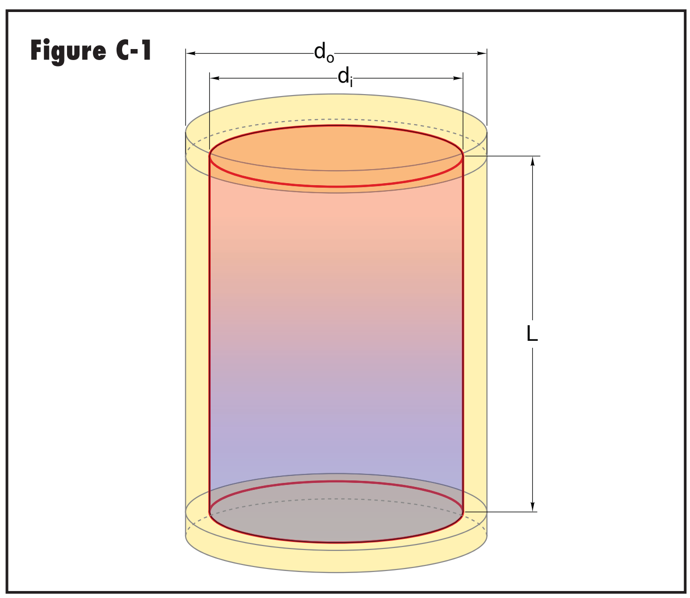
R_{tb} = R-value of insulation on top and bottom of tank (°F•hr•ft^2/Btu)

π = 3.141592654

T_w = *average* temperature of water in tank (°F)

T_a = temperature of air surrounding tank in tank (°F)

The relevant dimensions can be seen in Figure C-1.



Example: A flat-ended thermal storage tank measures 35 inches in diameter, and 80 inches tall. Assume that all surfaces of the tank have 3-inch thick polyurethane insulation (R = 6.0 per inch, k = 0.01389 Btu/hr•ft•°F). The average water temperature in the tank is 150°F. The air temperature surrounding the tank is 70°F, as is the temperature of the floor slab under the tank. Estimate the rate of heat loss from the tank under these conditions.

Solution: Before using Formula C-1, it's important to be sure that all the required data is expressed in the units noted under Formula C-1. The following unit conversions are necessary:

d_i = 35 inch = 2.9167 ft
d_o = (35+3+3) inch = 41 inch = 3.4167 ft
R_{tb} = 3 x (6.0/inch) = 18.0 (°F•hr•ft^2/Btu)

Putting the numbers into Formula C-1 and *carefully* processing them yields:

$$Q=\left[\left(\frac{2\pi kL}{\ln\left(\frac{d_o}{d_i}\right)+\frac{1.36k}{d_o}}\right)+\frac{\pi d_o^2}{2R_{tb}}\right](T_w-T_a)=\left[\left(\frac{2\pi(0.01389)(6.667)}{\ln\left(\frac{3.4167}{2.9167}\right)+\frac{1.36(0.01389)}{3.4167}}\right)+\frac{\pi(3.4167)^2}{2(18.0)}\right](150-70)==366\frac{Btu}{hr}$$

This result is an approximation. It assumes that the floor temperature under the tank is the same as the air temperature surrounding the tank. It does not deduct for any heat loss associated with heat transfer to piping connected to the tank. It also assumes a linear temperature gradient from the top to the bottom of the tank, and thus, is based on the *average* water temperature in the tank. It assumes no thermal resistance for the metal walls that form the tank. The calculated *rate* of heat loss is only valid at the stated conditions. As the tank cools, the water temperature decreases, and so does the rate of heat loss from the tank.

The time required for a vertical cylindrical storage tank to cool on it's own (e.g., without heat extraction by the loads it serves) can be estimated using Formula C-2.

Formula C-2:

$$T_w=T_a+(T_{wi}-T_a)e^{-\left(\frac{c}{8.33v}\right)t}$$

Where:

T_a = room air temperature (°F)
T_{wi} = initial average temperature of tank water (°F)
t = time (hours)
v = volume of water in tank (gallons)
e = 2.718281828

$$c=\left[\left(\frac{2\pi kL}{\ln\left(\frac{d_o}{d_i}\right)+\frac{1.36k}{d_o}}\right)+\frac{\pi d_o^2}{2R_{tb}}\right]$$

Notice that the expression for "c" is the same as the portion of Formula C-1 that appears in red, and before the expression (T_w-T_a). The value of "c" is entirely defined by the dimensions and insulation of the storage tank. It does not depend on the temperatures involved.

Example: Assuming the storage tank from the previous example starts at an initial average water temperature of 150°F, and that no heat is removed by the loads. The tank is surrounded by 70°F air. Estimate the tank's average temperature 24 hours later.

Solution: The value of "c" from the previous example is 4.57.

It is also necessary to determine the tank's water volume. This can be determined using Formula B-1:

$$v=\frac{\pi d_i^2 h}{924}=\frac{\pi(35)^2 80}{924}=333 gallons$$

This and the other values can now be substituted into Formula C-2:

$$T_w=T_a+(T_{wi}-T_a)e^{-\left(\frac{c}{8.33v}\right)t}=70+(150-70)e^{-\left(\frac{4.57}{8.33(333)}\right)24}=70+80e^{-0.03954}=146.9°F$$

This calculation estimates that the average water temperature in the tank drops by just over 3°F during the 24-hour period. This is a very low rate of heat loss. However, this calculation does not account for heat loss from piping—especially uninsulated piping—that may be connected to the tank.

Buffer storage tanks ThermoCon™

NAS200 series

Function

ThermoCon™ tanks are designed to be used for wood boilers, solar and geothermal storage, plus in heating systems with low-mass boilers, chilled water systems and low-mass radiation. ThermoCon tanks are used in systems operating below the design load condition, which is most of the time, or in systems having several low cooling or heating loads demands at different times. Boilers operating at low loads will short cycle, resulting in reduced operating efficiency and shorter equipment life. When piped correctly, the ThermoCon will serve as both a thermal buffer and a hydraulic separator. The solar, boiler or chiller system will be hydraulically separated from the distribution system.
Meets or exceeds ASHRAE 90.1b and CSA C309 requirements

Product Range

Code NAS20025 Storage tank..........25 gallon
Code NAS20050 Storage tank..........50 gallon
Code NAS20080 Storage tank..........80 gallon
Code NAS20120 Storage tank..........119 gallon

Technical characteristics

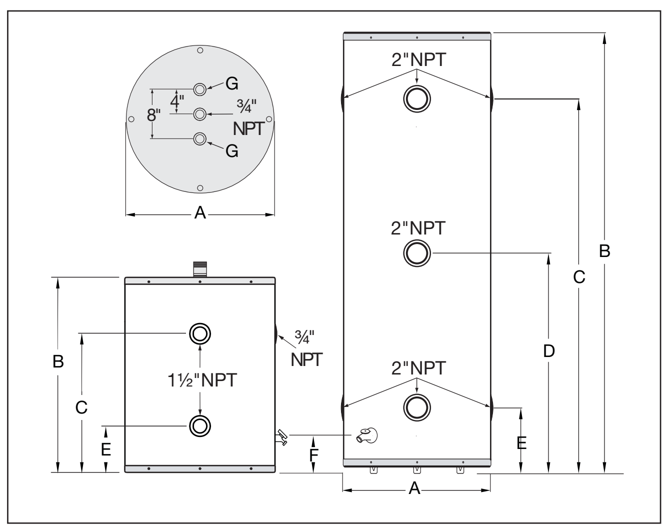
Tank materials:	porcelain coated steel
Tank insulation:	2" non-CFC foam
Tank external cover:	powder-coated steel (20-24 ga.)
Insulation thermal conductivity:	R16
Maximum working pressure:	150 psi
Testing pressure:	300 psi
Maximum tank temperature:	180°F
Recommended maximum delivery hot water temperature:	120°F

Construction details

The ThermoCon 25 gallon tank is engineered with six (6) 1½" NPT connections. Two top connections can be piped right below a wall hung modulating / condensing boiler. One of the top connections has a 1½" NPT male thread with a dip tube to draw cooler water from the bottom of tank. The other top 1½" NPT connection is female. The four side 1½" NPT female connections can be piping to the load.

The ThermoCon 50, 80 & 120 gallon tanks are engineered with seven (7) 2" NPT connections. Two connections can be piped to the solar, boiler or chiller side and two connections can be piped to the distribution system. Two additional connection are 90 degree from another which allows for positioning tank into a corner with the piping at a right angle. The tank has one 2" NPT connection for connecting an external heat exchange in the middle of the tank.

Dimensions



Code	Gal.	A	B	C	D	E	F	G	Wt. lbs.
NAS20025	25	22"	28½"	20½"	—	6¾"	4"	1½"	100
NAS20050	50	22"	48¼"	39½"	23½"	7¾"	4½"	¾"	200
NAS20080	80	24"	64"	53"	32"	11"	5"	¾"	250
NAS20120	120	28"	65"	53"	32"	11"	7"	¾"	350

Tank accessories

High discharge automatic air vent.
Brass body.
Max. working pressure: 150 psi.
Working temperature range: 32—250ºF.

Code 551004A...1/2" FNPT

Magnesium anode rod.

Code NA10229 Anode 36" fits 50 gal......................................3/4" NPT
Code NA10230 Anode 40" fits 80 & 120 gal.............................3/4" NPT

Pipe nipple for attaching air vent to top of storage tank with reducing bushing.

Code NA10160 Nipple 1/2" male x 3" ....1/2" NPT

Reducer bushing for installing into 2" NPT female connection in storage tank providing an ¾" NPT female thread. 1 ⅞" hex head. Low Lead

Code NA10234 Bushing....................................2" M NPT x 3/4" F NPT

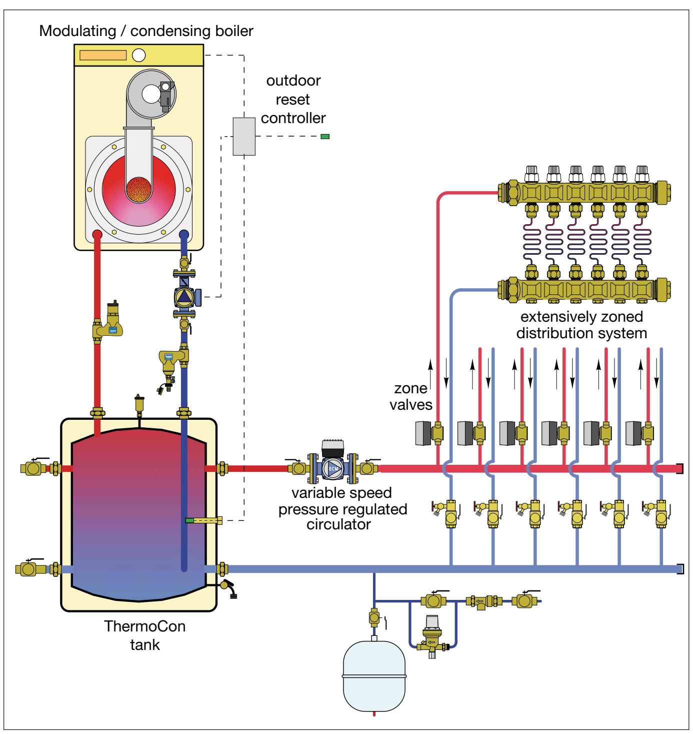
Reducer bushing for inserting into top of storage tank to attach pipe nipple to air vent. 1⅛" hex head.

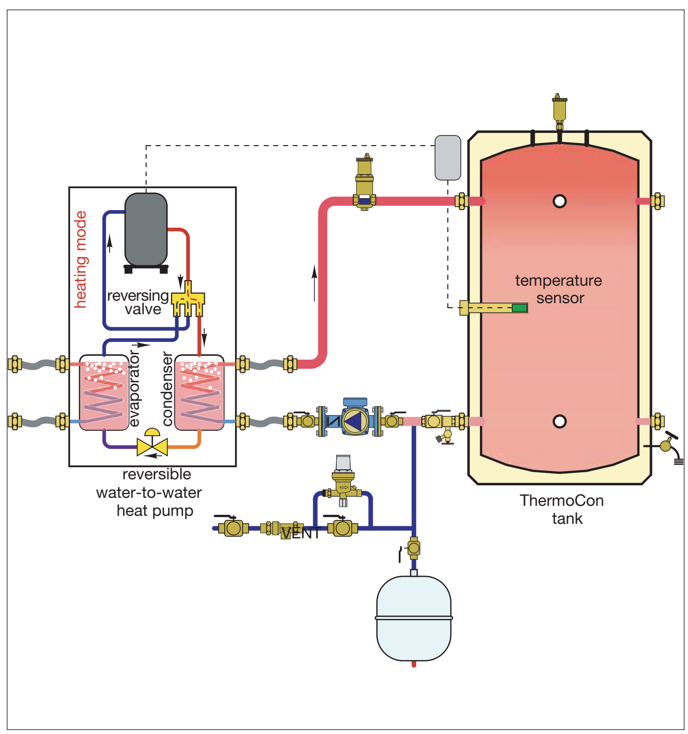
Code NA10082 Bushing.................................3/4" M NPT x 1/2" F NPT

Stainless steel male plug 1¼" square head.

Code NA10339 Plug ....2" M NPT

Application





Indirect water heater SolarCon™

NAS200 series

Function

Caleffi indirect water heaters have either one or two internal heat exchanger coils and a electric heating element in some models. The internal heat exchanger coils located inside the tank may be connected to a variety of different heating sources such as solar, biomass, geothermal and boilers. These heat sources can also be used separately or combined, resulting in greater application flexibility.

The one and two coil models have a 1-1⁄2" diameter porcelain coated steel heat exchanger located in the lower portion of the tank. Additionally, the two coil models have a 1-1⁄2" diameter porcelain coated steel heat exchanger located in the upper portion of the tank to provide enhanced heat transfer capability. The two coil indirect water heater is ideal for combination domestic hot water and space-heating applications or use both coils in parallel to boost 1st hour delivery up to 40% @ 180°F coil output.

Meets or exceeds ASHRAE 90.1b and CSA C309 requirements

Product Range

Code NAS20053	Indirect water heater with 1 heat exchanger and electric element	50 gallon
Code NAS20082	Indirect water heater with 2 heat exchangers	80 gallon
Code NAS20083	Indirect water heater with 1 heat exchanger and electric element	80 gallon
Code NAS20122	Indirect water heater with 2 heat exchangers	119 gallon
Code NAS20123	Indirect water heater with 1 heat exchanger and electric element	119 gallon
Code NAS20124	Indirect water heater with 2 heat exchangers and electric element	119 gallon

Technical Characteristics

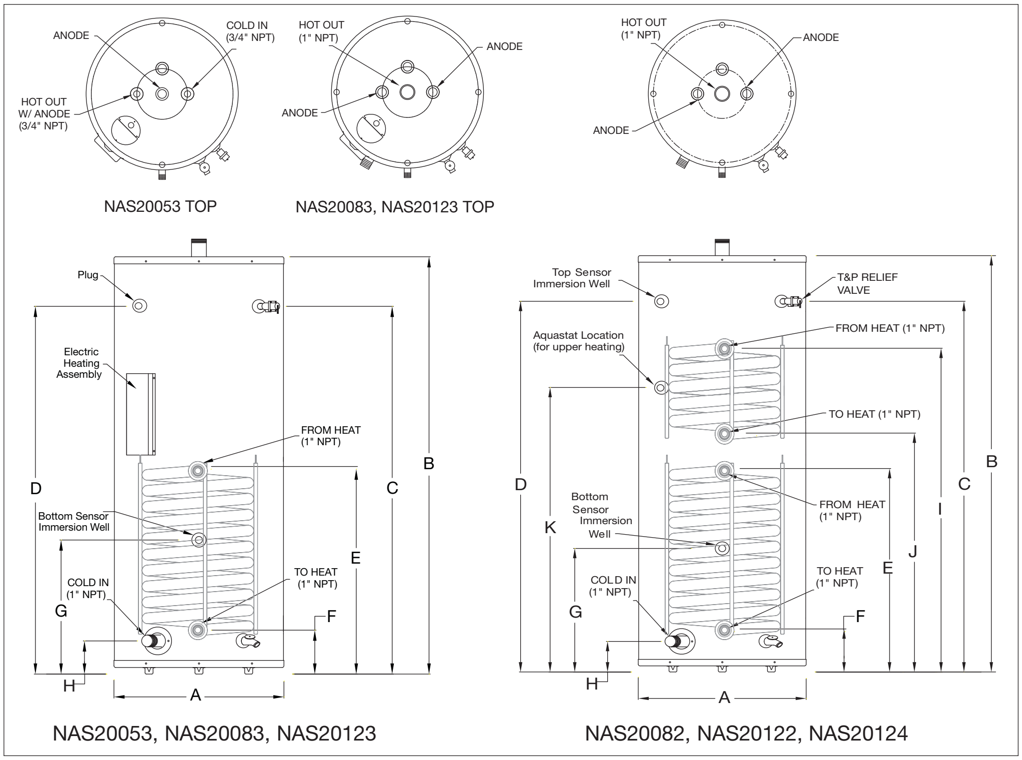
Tank materials:	porcelain coated steel
Tank insulation:	2" non-CFC foam
Tank external cover:	powder-coated steel (20-24 ga.)
Insulation thermal conductivity:	R16
Anode rods:	2 each magnesium
Internal heat exchanger coil (lower):	1-1/2" x 30' (50 gallon) 1-1/2" x 36' (80,119 gallon)
Internal heat exchanger coil (top):	1-1/2" x 24' (80,119 gallon)
Internal heat exchanger coil connections:	1" NPT
Potable connections:	3/4" NPT (50 gal.), 1" NPT (80, 119 gal.)
Maximum working pressure:	150 psi
Testing pressure:	300 psi
Temperature and pressure relief valve:	210°F/150 psi max
Maximum tank temperature:	180°F
Recommended maximum delivery hot water temperature:	120°F
Power requirements (electric element):	240 VAC
Power consumption (electric element):	4.5 kW
Agency approval:	UL listed

Capacity and performance

Model	Actual Tank Volume (gal)	Coil Volume Lower/Upper (gal)	Coil Surface Area Lower/Upper (ft^2)	Coil Friction Loss* Lower/Upper (ft. of head)	First Hour Delivery @135°F (gal)	Continous Draw Rating# @135°F (gal/hr)	Standby Loss Rating (°F/hr)
NAS20053	45	2.30/ -	11.78/ -	0.50/ -	205	185	1.1
NAS20083	75	2.76/ -	14.14/ -	0.60/ -	265	205	0.8
NAS20123	110	2.76/ -	14.14/ -	0.60/ -	300	205	1.2
NAS20082	73	2.76/1.84	14.14/9.42	0.60/0.40	370	285	0.8
NAS20122	108	2.76/1.84	14.14/9.42	0.60/0.40	420	305	1.2
NAS20124	108	2.76/1.84	14.14/9.42	0.60/0.40	420	305	1.2

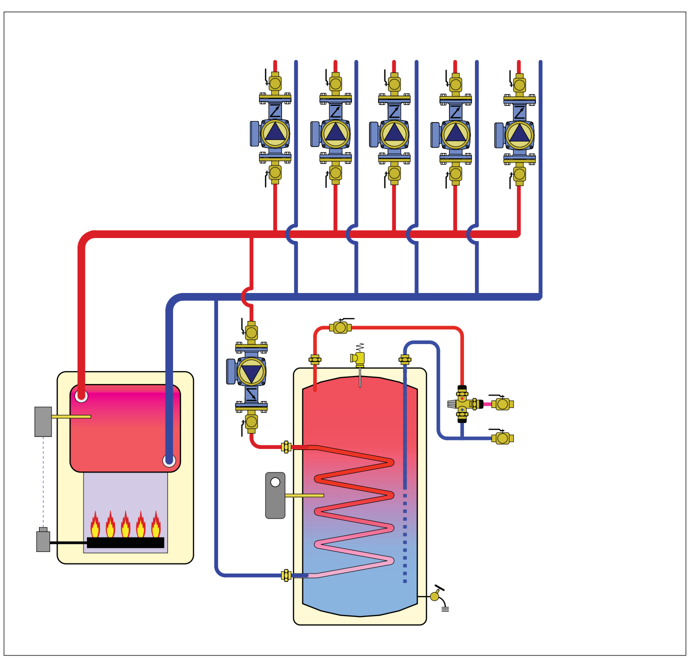
NOTES: * Based on 12 GPM flow rate.
Based on lower coil heat input of 180ºF @ 12 GPM. Depending on model, heat recovery is calculated with either a 4500W heating element or a upper coil heat with output of 180ºF @ 12 GPM. Potablewater temperature rise is 77ºF.

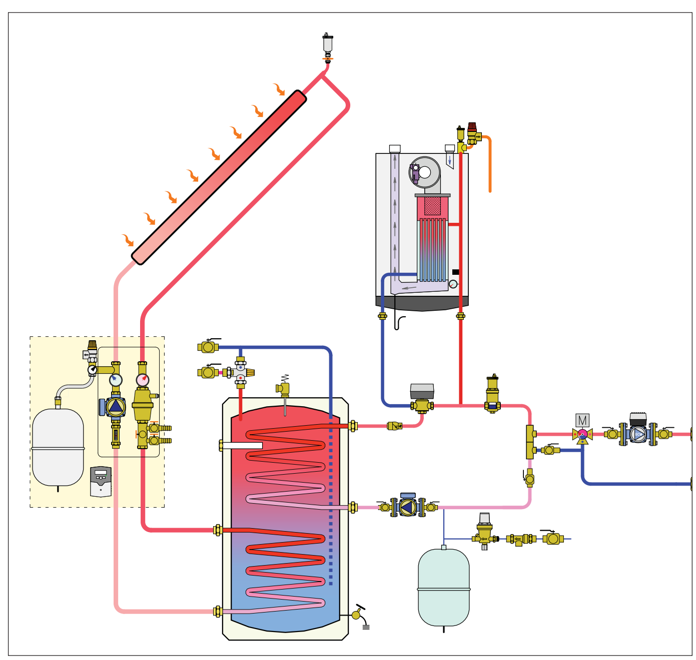
Dimensions



Model	A	B	C	D	E	F	G	H	I	J	K
NAS20053	22"	48¼"	39¾"	39¾"	31½"	6½"	16¾"	n/a	n/a	n/a	n/a
NAS20083	24"	64"	57⅛"	57⅛"	31½"	6½"	19½"	5"	n/a	n/a	n/a
NAS20123	28"	65"	57¾"	57¾"	33¾"	8¾"	16¼"	6½"	n/a	n/a	n/a
NAS20082	24"	64"	57⅛"	57⅛"	31½"	6½"	19¼"	5"	49½"	36½"	46⅞"
NAS20122	28"	65"	57¾"	57¾"	33¾"	8¾"	16¼"	6½"	51¾"	38¾"	49⅛"
NAS20124	28"	65"	57¾"	57¾"	33¾"	8¾"	16¼"	6½"	51¾"	38¾"	49⅛"

Application





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