

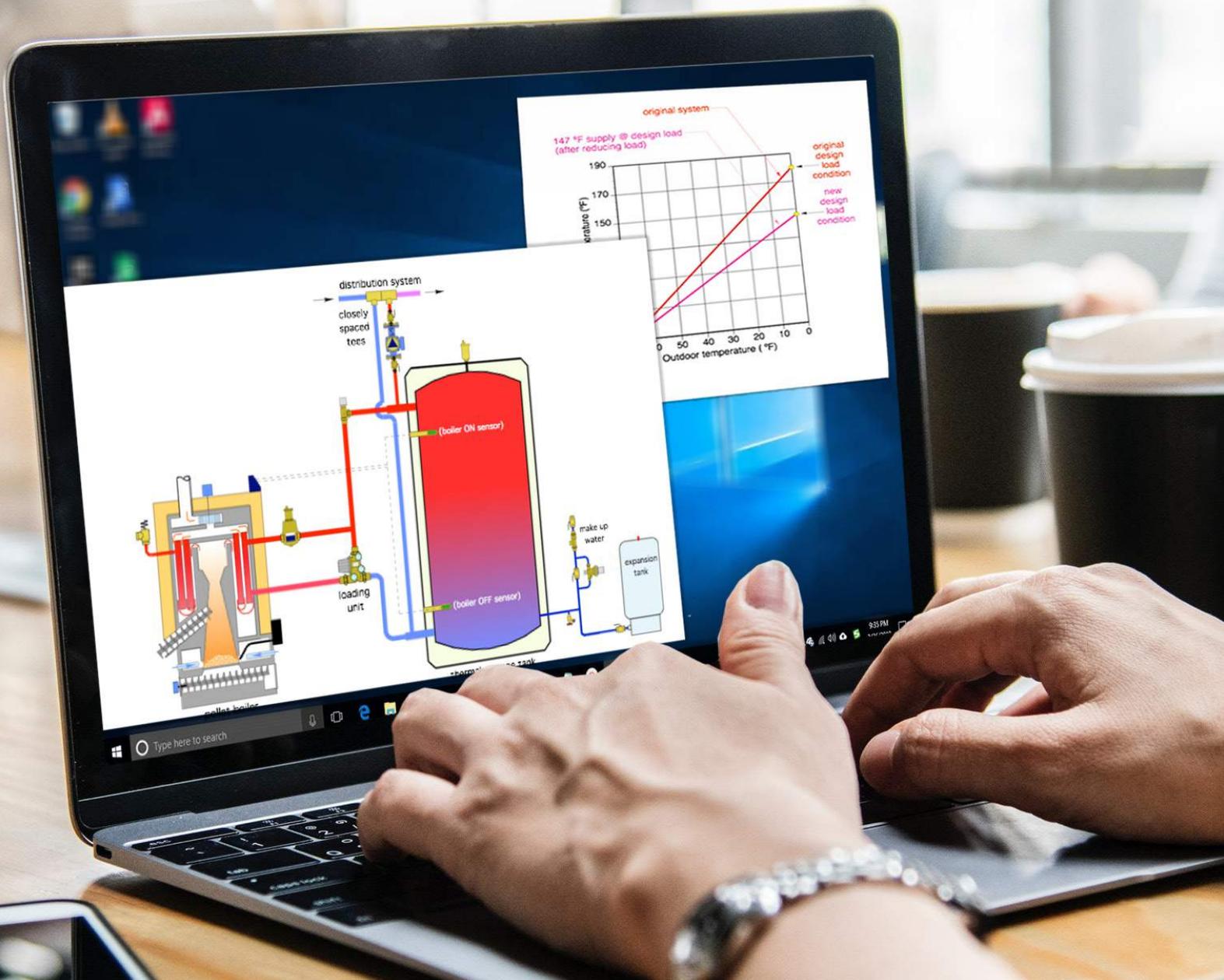
idronics™

JOURNAL OF DESIGN INNOVATION FOR HYDRONIC AND PLUMBING PROFESSIONALS

CALEFFI
Hydronic Solutions



Lowering Water Temperature in Existing Hydronic Heating Systems





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FROM THE GENERAL MANAGER & CEO

Dear Plumbing and Hydronic Professional,

Perhaps you are old enough to remember the oil embargoes of the 1970s. I sure do. Suddenly my parents adapted new favorite sayings such as “which of you kids left these lights on” and “who turned this thermostat up?” For good reason. Like the rest of the country my parents’ utility bills had escalated quickly and in dramatic fashion.



These energy shortages did more than cause households to become conservation minded. They helped spark a trend towards more efficient heating systems. Condensing boilers were developed that had the potential to significantly boost fuel efficiency. PEX tubing reinvigorated the radiant panel industry. Renewable energy heat sources such as solar thermal collectors and hydronic heat pumps, that operate best at low water temperatures, provided alternatives to fossil fuel boilers.

Today, energy efficient, low temperature hydronic systems are easily applied in new construction. However, modern heat sources that perform best at low water temperatures are not as easily adapted to existing buildings with older “high temperature” distribution systems. This mismatch is a significant impediment to upgrading older buildings to contemporary hydronic heat sources.

This issue of *idronics* discusses techniques for lowering the water temperature in existing systems to make them compatible with modern high efficiency heat sources. It shows how reductions to heating loads impact supply water temperature. It exams practical methods for modifying existing distribution systems, and adding supplemental heat emitters. It presents analytical methods for evaluating options. Lastly, it presents examples that demonstrate practical solutions for lowering water temperature without compromising comfort.

We hope you enjoy this 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

A handwritten signature in black ink that reads "Mark Olson". The signature is written in a cursive, flowing style.

General Manager & CEO

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LOWERING WATER TEMPERATURE IN EXISTING HYDRONIC HEATING SYSTEMS

The rate of heat output from any hydronic heat emitter, such as a fin-tube baseboard, a cast iron radiator or a fan-coil, depends on the temperature of water entering that emitter. The higher that temperature is relative to the surrounding air temperature, the higher the rate of heat output.

The size of the heat emitter required to maintain a space at a selected “comfort” temperature under design load conditions also depends on the temperature of the water supplied to it. For any given design heating load, the higher the water temperature, the smaller the required heat emitter.

Smaller heat emitters reduce installation cost. When energy was inexpensive and seemingly abundant, efforts to conserve it usually took low priority compared to efforts to reduce installation cost. This was the case in the North American hydronics industry through much of the twentieth century. Most hydronic systems installed during that time were designed around high water temperatures. In some cases, the water supplied to the heat emitters exceeded 200°F.

Figure 1-1



Another reason underlying past use of high water temperatures in hydronic heating was conversion of steam heating systems to operate with hot water. The heat emitters installed as part of the steam heating system were typically cast iron radiators sized for low-pressure steam supplied at temperatures in the range of 215 to 220°F. To maintain reasonably equivalent heat output, it was necessary to operate those radiators at relatively high water temperatures.

Almost all North American hydronic heating systems installed prior to the 1980s were supplied by boilers burning fuels such as coal, fuel oil and natural gas. With combustion temperatures in the range of 2,000°F, it was easy to produce water temperatures of 200°F or higher.

Figure 1-2



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Some twentieth century boilers also maintained minimum water temperatures in the range of 140°F, even during summer months, to produce on-demand domestic hot water from an inserted copper heat exchanger called a “tankless coil.” Figure 1-3 shows an example of such a coil.

In most areas of North America, it is still legal to design hydronic heating systems around relatively high supply water temperature. Several manufacturers of fin-tube baseboard currently publish heat output rating

Figure 1-4

NET AHRI WATER RATINGS BTUH Per linear Foot At Average Water Temperatures Indicated								
150 °F	160 °F	170 °F	180 °F	190 °F	200 °F	210 °F	220 °F	230 °F
340	390	450	510	560	610	670	720	770
320	370	430	480	530	580	630	680	730

tables for water temperatures of up to 220°F. Fan-coil heat output is often listed up to 200°F water temperature. Some heat output ratings for cast iron baseboard and radiators still reference water

temperatures as high as 230°F, as shown in figure 1-4.

Maintaining water as a liquid at temperatures over 212°F (at sea level) requires system pressurization. A sudden loss of this pressurization allows the superheated water to flash into steam. This can be extremely dangerous.

CHANGING PRIORITIES:

The common use of high water temperatures in hydronic heating systems started to be scrutinized in the late 1970s, in the wake of energy shortages and price volatility. Boiler manufacturers began looking for ways to push the thermal efficiencies of gas-fired boilers from the mid-80 percent range into the 90+ percent range. This necessitated recovery of latent heat from the water vapor produced during combustion. Sustained flue gas condensation was only possible when the water temperature entering the boiler was lower than about 130°F. Heat emitter sized around water temperatures of 180°F or higher could not consistently provide such low boiler entering water temperatures, and thus represented an impediment to significant gains in boiler efficiency.

The hydronic in-floor heating market in North America also redeveloped during the 1980s. This was largely due to the availability of cross-linked polyethylene tubing (PEX), which solved previous reliability issues associated with metal tubing embedded in concrete slabs. Most radiant panel systems require much

Figure 1-3

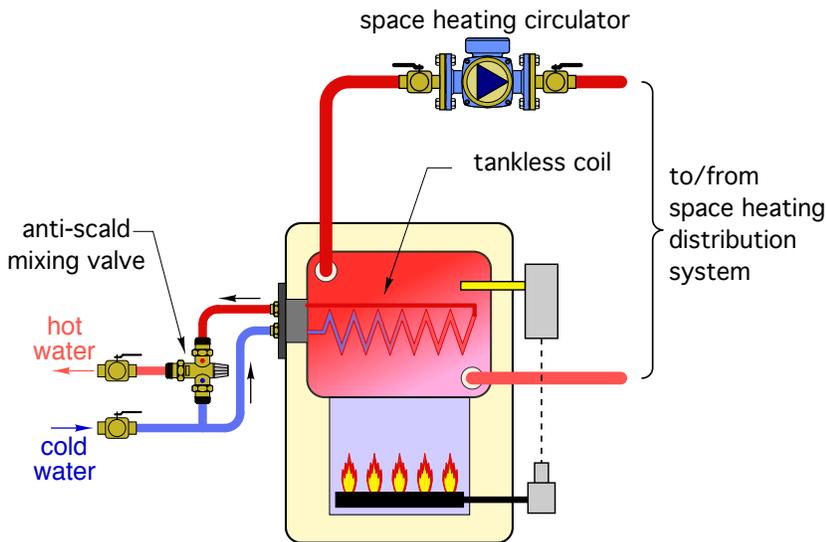
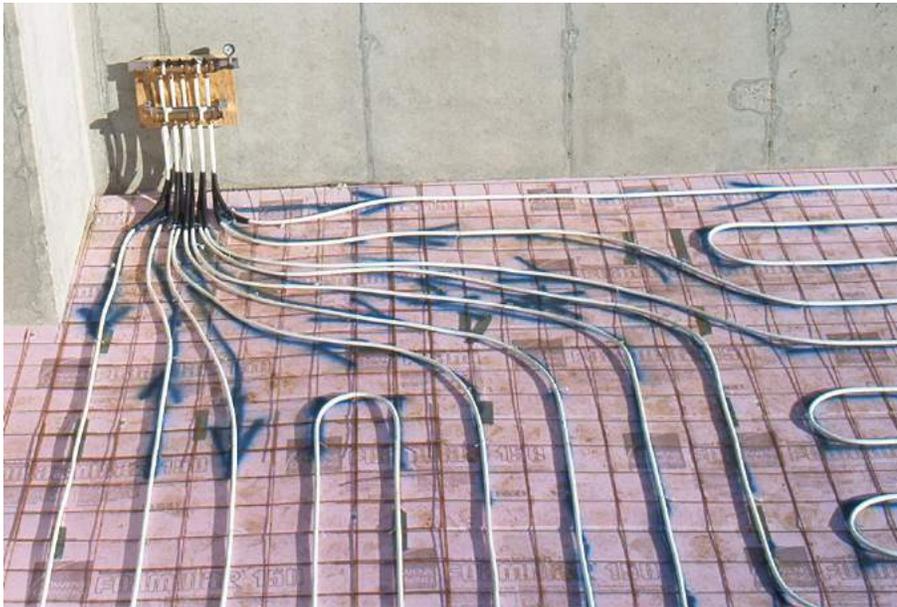


Figure 1-5



lower water temperatures compared to traditional heat emitters such as fin-tube baseboard or cast iron radiators. During the latter part of the twentieth century, most radiant systems were supplied by conventional boilers. A mixing assembly was installed between these boilers and the radiant panels. These assemblies mixed a portion of the cooler water returning from the radiant panels with hot boiler water to achieve the lower supply water temperature.

When condensing boilers became more readily available, the low water temperatures associated with most in-floor heating systems was well-suited to producing sustained flue gas condensation, and thus ensured high boiler efficiency. Condensing boilers now represent over half of all boilers sold in North America.

The last two decades of the twentieth century was also a time when a strong interest developed in using solar energy for building heating and producing domestic hot water. The efficiency of solar thermal collectors is very dependent on the

fluid temperature at which they operate. Lowering the collector inlet temperature from 170°F to 110°F on a sunny but cold (20°F ambient air temperature) winter day can nearly double the amount of useful energy captured by a flat plate solar thermal collector, such as shown in figure 1-6. Thus, when flat plate solar collectors are used for space heating, it is imperative to combine them with low-temperature heating emitters and compatible distribution systems.

Figure 1-6



Another market segment – hydronic heat pumps – also has roots in the later part of the twentieth century. Hydronic heat pumps include water-to-water units, which are commonly used with geothermal heat sources, and air-to-water heat pumps. Figure 1-7 shows an example of a water-to-water geothermal heat pump. Figure 1-8 shows an example of a “monobloc” type air-to-water heat pump.

Figure 1-7



Courtesy of Harvey Youker

Figure 1-8



Most hydronic heat pumps use vapor compression refrigeration cycles. With few exceptions, these heat pumps can only produce water temperatures up to about 125°F. So again, for good performance, it is imperative to combine hydronic heat pumps with low-temperature heat emitters and compatible distribution systems.

The future of hydronics will increasingly rely on these renewable energy heat sources, as well as high-efficiency, condensing-capable boilers. One characteristic shared by all these heat sources is that they deliver their highest efficiency when operating in low water temperature systems.

CHANGES ARE NEEDED:

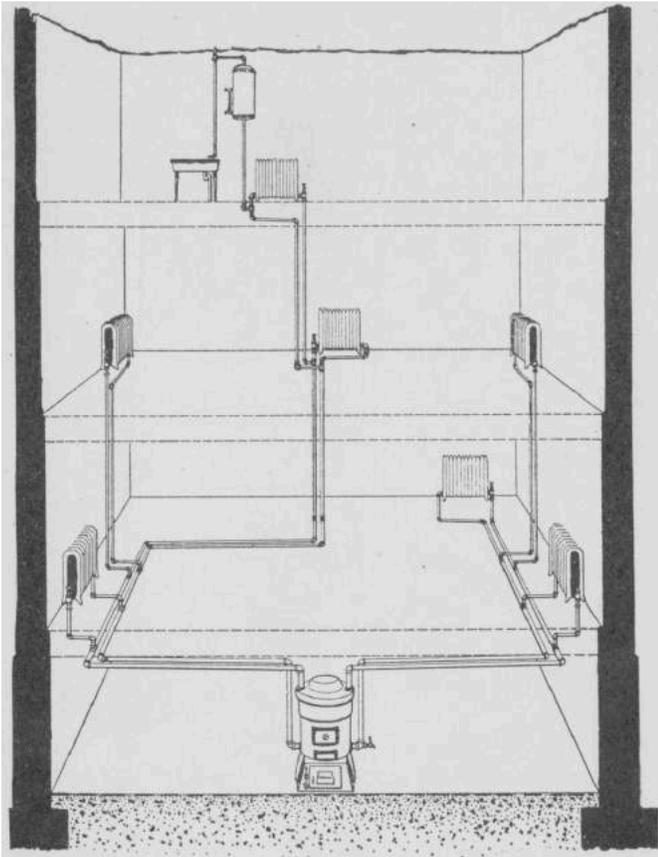
A major challenge for the North American hydronics industry will be modifying existing heating systems for compatibility with these evolving heat sources. Most hydronically heated homes and commercial buildings constructed in North America prior to 1980 have heat emitters and distribution systems designed around high water temperatures. The process of retrofitting a renewable or other low-temperature/high-efficiency heat source into such buildings should always include an evaluation of the existing heat emitters and distribution system. Based on this evaluation, designers can develop a plan for lowering water temperature without sacrificing comfort or compromising heat source performance.

This issue of idronics to explore several techniques for reducing water temperatures in existing hydronic heating systems. It discusses how reductions to heating loads through building envelope improvements impact supply water temperature requirements. It examines practical methods for modifying existing distribution systems and adding supplemental heat emitters. It presents analytical methods for evaluating options. This issue concludes with examples of how common high-temperature systems can be modified to operate at lower water temperatures.

WHY LOWER WATER TEMPERATURES ARE IMPORTANT

The roots of water-based (vs. steam-based) hydronic heating date back into the 1800s. These early systems did not have the luxury of circulators. Instead, they relied on the density difference between high-temperature water in the boiler and vertical supply piping, compared to the cooler water in the return piping as the “driver” for water circulation. The higher the water temperature in the boiler and supply piping, relative to the water in the return piping, the greater the differential pressure available to circulate water through the piping. Figure 2-1 shows an example of such a system. Notice that there is no circulator.

Figure 2-1



The development of electrically driven circulators in the 1930s greatly expanded the potential of hot water heating. However, boilers burning fuels such as coal and fuel oil remained the standard heat source for nearly all these systems. That continued to be the case until the late 1970s, when interest in “alternative energy” led to the development of several very different types of hydronic heat sources, including solar thermal collectors, heat pumps and biomass boilers.

This section examines the relationship between the thermal performance and operating temperature of

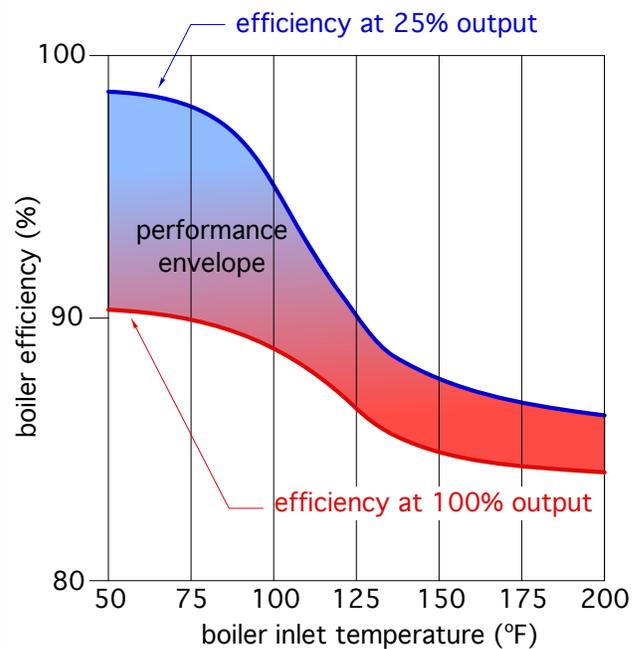
boilers, as well as several renewable energy heat sources. It demonstrates the consistent “inverse” relationship between system operation temperature and the thermal efficiency of these heat sources. As water temperature decreases, thermal efficiency always increases.

MODULATING/CONDENSING BOILERS:

Energy shortages and price volatility during the 1970s spurred interest in improving boiler efficiency. This led to development of boilers that could operate with sustained flue gas condensation. Such operation, when possible, allows the latent heat of the water vapor produced during combustion to be captured within the boiler and transferred to the water passing through it. This condition has the potential to boost boiler efficiency from the mid-80 percent range to the mid-90 percent range. However, maintaining sustained flue gas condensation requires the boiler’s heat exchanger to operate at relative low temperatures.

Figure 2-2 shows the relationship between a thermal efficiency and entering water temperature for a modulating boiler.

Figure 2-2



The boiler's efficiency is a function of both inlet water temperature and firing rate.

At full firing rate, the efficiency varies from about 85 percent when the inlet water temperature is 200°F, to just over 90 percent when the inlet water temperature is 50°F. The latter temperature is a temporary condition during a cold boiler startup. Low boiler inlet temperatures can also persist during warm-up of heated floor slabs or snow-melted pavements.

At a firing rate of 25 percent, the efficiency increases. This is due to reduced combustion gas flow spread across the same heat exchanger surface within the boiler. At an inlet water temperature of 200°F, the efficiency is about 87 percent but increases to about 98 percent at an inlet temperature of 50°F.

Notice the pronounced rate of increase when the entering water temperature drops below approximately 130°F. This "dewpoint" temperature is where water vapor in the flue gas stream begins to condense. Each pound of water vapor that condenses back to liquid releases about 970 Btus of heat. The lower the boiler's inlet water temperature, the greater the rate of flue gas condensation, and thus the higher the boiler's efficiency.

To maintain consistently high boiler efficiency, the inlet water temperature should be as low as practical.

Some heating systems using large surface area radiant panels can yield boiler entering water temperatures in the range of 90°F, even under design load conditions. Even lower boiler entering water temperatures are possible under partial load conditions. Such systems are ideal for modulating/condensing boilers.

This relationship between inlet water temperature and efficiency also implies that using a modulating/condensing capable boiler to supply loads that consistently yield boiler inlet temperatures above 130°F provides minimal performance benefits compared to using a conventional boiler (e.g., one not intended to operate with sustained flue gas condensation).

The seasonal thermal efficiency of a modulating/condensing boiler increases as the cumulative operating hours at inlet temperatures at or below the dewpoint of the flue gases increases. Modifying an existing high-temperature hydronic distribution system so that it can operate at lower water temperature can significantly improve the seasonal efficiency of a modulating/condensing boiler.

SOLAR THERMAL COLLECTORS:

The efficiency of a solar thermal collector is defined as the ratio of the useful thermal energy imparted to the fluid passing through the collector, divided by the intensity of the solar radiation incident on the collector. This efficiency depends on the temperature of the fluid entering the collector and the air temperature surrounding the collector. These simultaneous variables can be combined into a single value called the inlet fluid parameter, which is defined by formula 2-1.

Formula 2-1:

$$IFP = \frac{T_i - T_a}{I}$$

Where:

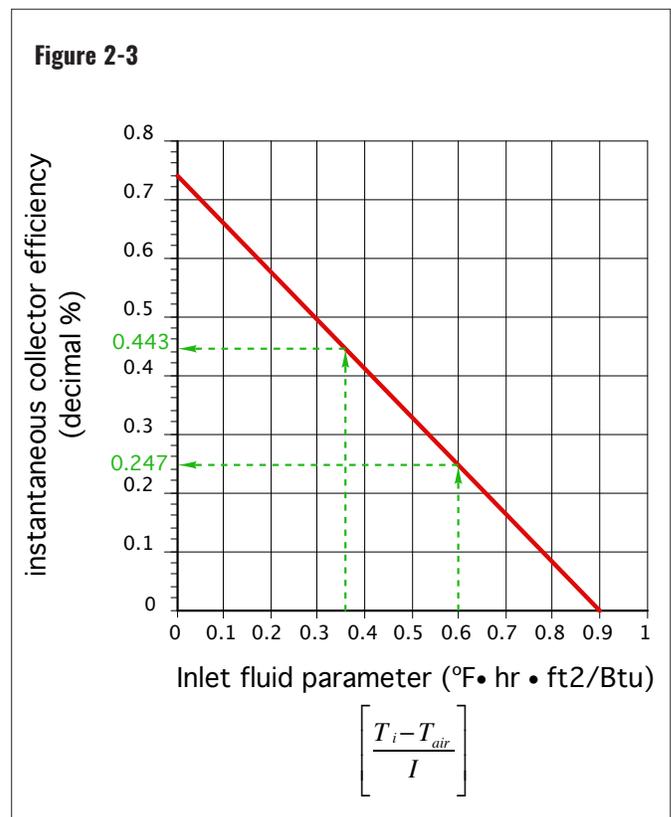
IFP = inlet fluid parameter (°F•ft²•hr/Btu)

T_i = fluid temperature entering collector (°F)

T_a = air temperature surrounding collector (°F)

I = intensity of solar radiation incident on the collector (Btu/hr/ft²)

Figure 2-3 shows how the thermal efficiency of a typical flat solar collector varies with the inlet fluid parameter.



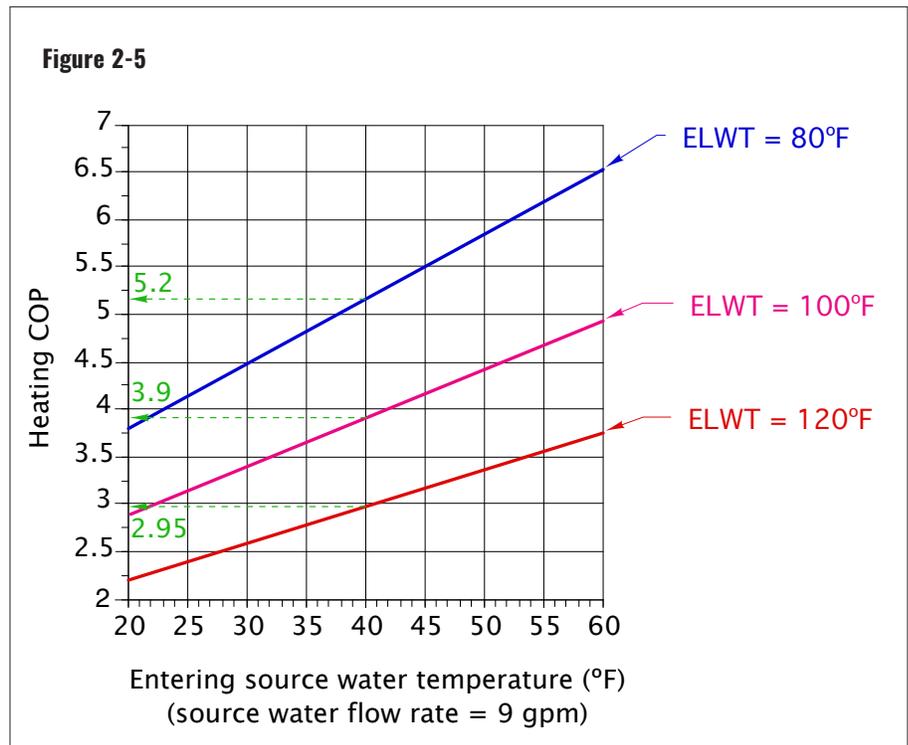
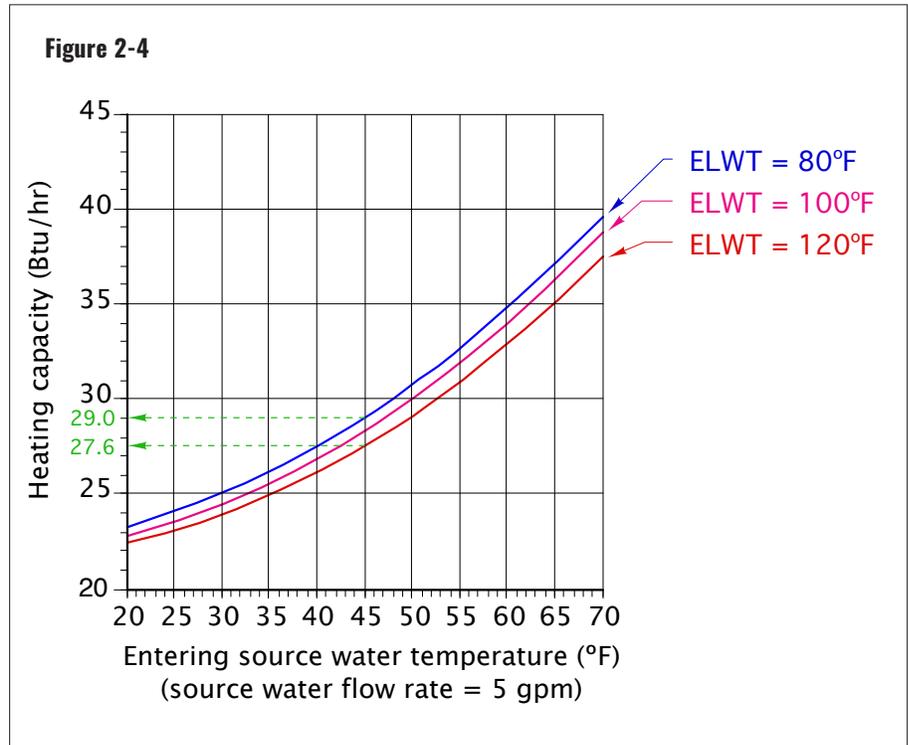
If a space-heating system required this collector to operate with an inlet fluid temperature of 170°F, on a cold but sunny winter day ($T_a = 20^\circ\text{F}$, $I = 250 \text{ Btu/hr/ft}^2$), the inlet fluid parameter would be $(170-20)/250 = 0.6 \text{ (}^\circ\text{F}\cdot\text{ft}^2\cdot\text{hr/Btu)}$. Under this condition the collector's efficiency can be read on figure 2-3 as 0.247 or about 25%. However, if the collector was instead supplying a lower temperature heating load, and operating at an inlet temperature of 110°F, the inlet fluid parameter would be $(110-20)/250 = 0.36$. The collector's efficiency under this condition would be 0.443 or 44.3%. That's a 79% gain in the amount of solar energy captured by the collector, solely due to the lower operating temperature. Thus, for good collector performance, it's imperative to design (or modify) systems so that heat emitters can operate at relatively low water temperatures.

GEOTHERMAL WATER-TO-WATER HEAT PUMPS:

The heat output and thermal efficiency of any hydronic heat pump is highly dependent on the fluid temperature entering the heat pump's condenser. Figure 2-4 shows how the heating capacity of a typical water-to-water heat pump varies with both the inlet fluid temperature from the source water – typically a geothermal earth loop – and the water temperature entering the condenser (ELWT).

Assuming that the earth loop supplied this heat pump with 45°F fluid, its heating capacity increases from 27,600 Btu/hr to 29,000 Btu/hr, if the entering load water temperature (e.g., the temperature of water returning from the heating load) was 80°F rather than 120°F.

However, this change in heating capacity is only part of the story. The same heat pump's efficiency,



expressed as coefficient of performance (COP), is significantly more dependent on the water temperature entering the condenser (ELWT), as seen in figure 2-5.

The higher the heat pump's COP, the more heat it delivers per unit of electrical energy consumption. Assuming the same 45°F entering fluid temperature from the earth

Figure 2-6

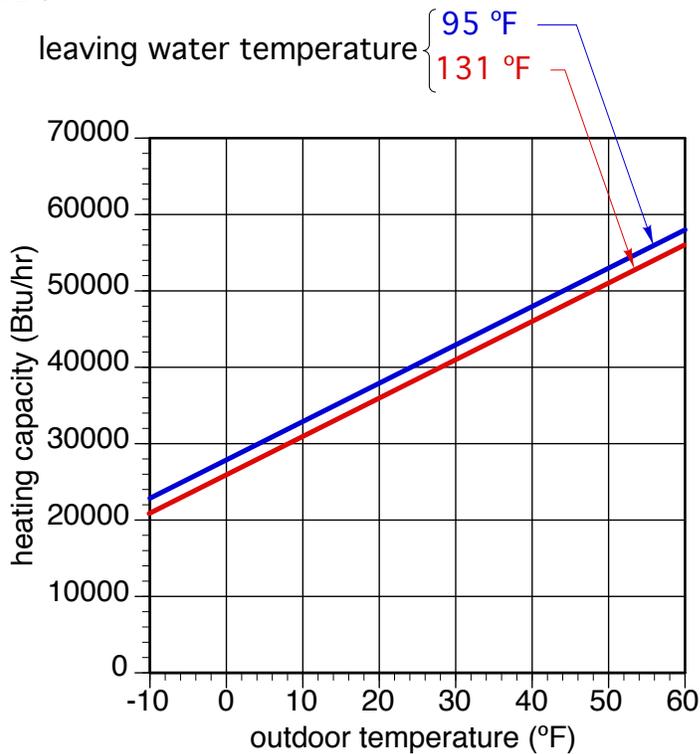
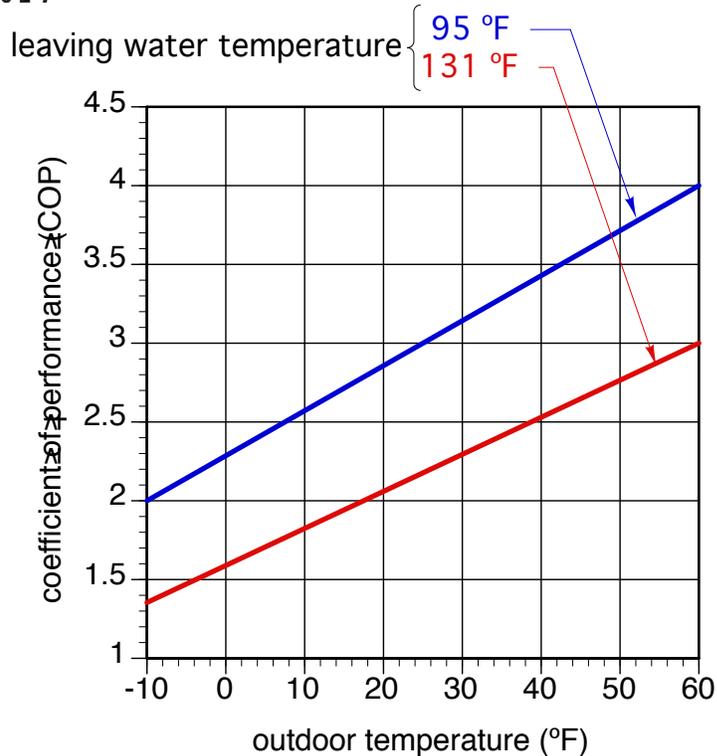


Figure 2-7



loop, the COP of this heat pump increases from 2.95 to 3.9 (a 32% increase) if the water temperature entering the heat pump's condenser (e.g., returning from the heating load) is 100°F rather than 120°F. The heat pump's COP increases from 3.9 to 5.2 if the temperature of water entering the condenser was 80°F rather than 100°F. That's another 33% gain.

One might argue that it's impossible to have water at 80°F returning from a space-heating load. While there's practical validity to that argument under design load conditions, it's not necessarily true under partial load conditions. A supply water temperature in the range of 90°F, and corresponding return temperature of 80°F, could likely meet the heating needs of an energy-efficiency building when the outdoor temperature is in the range of 30-40°F. The strategy of lowering a hydronic system's operating temperature under partial load conditions is called outdoor reset control. It has been used for several decades in combination with many types of heat sources and heat emitters, and will be discussed in more detail in later sections.

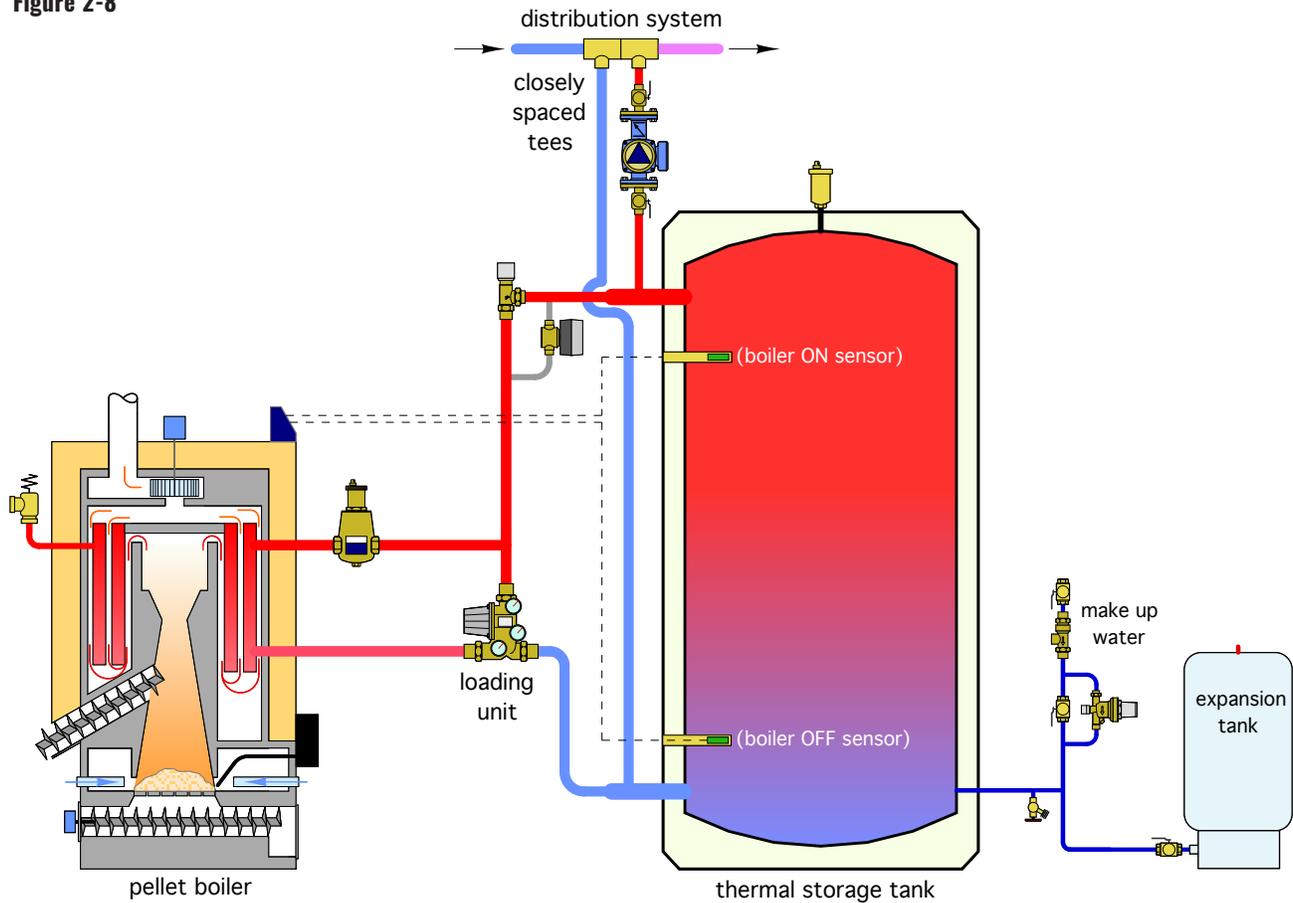
AIR-TO-WATER HEAT PUMPS:

The relationship between condenser entering water temperature, heating capacity and COP that applies to water-to-water heat pumps also holds true for air-to-water heat pumps. Figure 2-6 shows an example of heating capacity versus outdoor air temperature for two different water temperatures leaving the heat pump's condenser.

There is a slight drop in heating capacity when this heat pump operates at the higher condenser leaving water temperature (131°F).

Figure 2-7 shows the coefficient of performance (COP) of this heat

Figure 2-8



pump as a function of outdoor temperature and condenser leaving water temperature.

As was true with water-to-water heat pumps, there is a significant drop in COP when the heat pump is forced to operate at the higher condenser leaving water temperatures. This holds true across the full range of outdoor temperatures. Thousands of dollars in added electrical energy cost could be required over the life of the system if the heat pump is forced to operate at the high end of its temperature range. The remedy: Keep the water temperature in the heat pump's condenser as low as possible to maximize COP. This is done by carefully selecting low-temperature heat emitters, or by modifying existing high-temperature

distribution systems so that they can operate at significantly lower water temperatures.

BIOMASS BOILERS:

Cordwood gasification boilers and pellet boilers are both viable hydronic heat sources that use renewable carbon-neutral wood as fuel. Some can obtain steady state efficiencies in the mid-80 percent range. In contrast, the efficiencies of single stage atmospheric wood boilers is typically in the low to mid-40 percent range, especially if they are improperly operated or inadequately maintained. This implies that modern cordwood gasification and pellet boilers, when properly installed and operated, can extract nearly twice as much heat from a given weight of wood

fuel compared to early generation wood-fired boilers.

To attain high overall efficiency, cordwood gasification boilers and pellet boilers need to operate at high firing rates for continuous periods of 3 to 5 hours. During this time, they typically produce far more heat than is required by the load, especially under partial load conditions. The mismatch between boiler heat output and building load can be managed by incorporating thermal storage into the system.

Figure 2-8 shows a typical piping configuration for a pellet boiler with an associated thermal storage tank. The pellet boiler fires when the water temperature in the upper portion of the thermal storage tank

drops below some value. It continues to fire until the temperature in the lower portion of the tank increases to a relatively high setting. This boiler operating logic is completely independent of any heating demands from building thermostats. The goal is to make the firing cycles several hours long whenever possible to maximize the time over which the boiler operates at steady state conditions. A common design guideline for pellet boilers is to create a balance-of-system and control logic that provide an average boiler on-time of 3 hours per start.

The length of the boiler's firing cycle depends on the temperature cycling range of the thermal storage tank between boiler "ON" and boiler "OFF" conditions. The greater this " ΔT " is, the longer the operating cycle.

The upper temperature limit of the thermal storage tank is typically in the range of 170 to 200°F, depending on the design of the tank. "Open" tanks that hold water in liners made of EPDM rubber or polypropylene are typically limited to 175°F. Closed/pressurized steel tanks can, in theory, accept water at temperatures in excess of 212°F. However, temperature limitations of the boiler and safety concerns typically limit the upper tank temperature to 200°F.

The lower limit temperature of the thermal storage tank is determined by the distribution system and heat emitters used in the system. If the heat emitters are sized for design load output based on high water temperatures, such as is commonly done with fin-tube baseboard, the allowable temperature drop in the thermal storage, under design load conditions, will be minimal. For example, the pellet boiler may have to be fired when the temperature in the upper portion of the tank is still 170°F – just to ensure adequate heat output from the high-temperature heat emitters. The allowable ΔT of the tank between boiler "ON" and boiler "OFF" conditions is very small. This causes frequent short-cycles, a very undesirable condition from the standpoint of boiler efficiency. Short-cycles also create higher particulate emissions from the boiler. However, if the heat emitters used in the system can maintain design load output at much lower water temperatures, such as a radiant panel being supplied with 105°F water, the temperature cycling range of the tank is greatly expanded. This results in high efficiency and lower emissions, both of which are very desirable operating conditions.

In new systems, the lower-temperature emitters can be specified. In existing systems, it's often necessary to modify the distribution system and emitters, and use specific control strategies to allow low-temperature operation.

FUTURE-PROOFING HYDRONIC SYSTEMS:

The advantages of using low water temperature distribution systems to maximize the efficiency of several currently available hydronic heat sources has been discussed. However, those who design hydronic systems should also think beyond currently available heat sources. Two key concepts that should be considered are:

1. How long will a well-designed and properly installed hydronic heating or cooling system last?
2. Will the hydronic systems I'm currently designing be compatible with future heat sources?

The useful life of a well-planned, properly installed and adequately maintained hydronic system is typically longer, in some cases much longer, than the life of the original heat source. There are thousands of existing hydronic heating and cooling systems in both residential and commercial buildings that are still operating after 50 years or more of service. Most have had one or more replacement heat sources, and perhaps other items such as a new oil burner, circulator or thermostat, but most of the original hardware, such as piping, heat emitters and valves, are still in place and functioning well.

Modern modulating/condensing boilers have a typical service life of 12 to 18 years. Cast iron boilers typically have service lives of 30+ years, and geothermal heat pumps typically last 20 to 25 years, although their earth-loop piping could remain in service for many decades. Air-to-water heat pumps, which are exposed to the elements, typically last 15 to 20 years. There will always be exceptions to these anticipated service lives. Some heat sources will last longer, while others will fail prematurely due to improper installation, operation, lack of maintenance or severe opening conditions.

Odds are that a well-planned and properly installed hydronic distribution system will last significantly longer than its first heat source, perhaps longer than its second or even its third heat source. Prudent designers consider this and contemplate what can be done at present to ensure that a long-lasting hydronic distribution system will likely be compatible with future heat sources.

Although no one knows for sure what those future hydronic heat sources might be 20 years or more from now, there are current market trends that provide clues. One of those trends is a slow-but-steady migration away from traditional fossil fuels, such as natural gas, propane and fuel oil. This trend is influenced by economic, social and political factors, both in North America and globally. In the absence of major technological breakthroughs that could

mitigate or sequester carbon emissions from fossil fuels, it's increasingly likely that future hydronic heat sources and chillers will depend on electricity as their primary energy source. That electricity will come from a growing number of utility-scale renewable sources, such as large solar photovoltaic systems and wind farms. Smarter electrical distribution grids, in combination with utility-scale electrical energy storage, will better manage generation resources versus loads. Rate structures such as time-of-use and real-time pricing will incentivize customers to use electrical energy in ways that reduce strain on the grid and help distribute renewably sourced energy.

Electricity is a high-grade form of energy. As such, it's very easy to convert it directly into heat, a low-grade form of energy. However, doing so wastes the potential of electrical energy to provide a better outcome. For example, when electricity is converted directly into heat, one kilowatt•hour of electricity produces exactly one kilowatt•hour of heat. If that kilowatt•hour of electricity was instead used to operate a heat pump, the output could be 3, 4 or even more kilowatt•hours of useful heat. One kilowatt•hour of that heat would come from the electricity, and the rest would be gathered from a low-grade heat source, such as the ground, a lake or outside air.

The ability of electrically powered heat pumps to “multiply” useful heat yield in comparison to electric-resistance heating, as well as their ability to provide cooling, is likely to have a major influence on future building heating and cooling system choices, including those using hydronic distribution systems. To take maximum advantage of such heat sources, it's very important to combine them with low water temperature distribution systems. This also holds true for high-efficiency fossil-fuel hydronic heat sources.

Thus, it's highly probable that low water temperature hydronic distribution systems designed in the present will require the least amount of modification, if any, to remain compatible with future heat sources.

It makes little sense to design hydronic heating systems around high water temperatures in the present, knowing that major modifications or perhaps complete replacement of those systems will be required 20+ years from now, based on the likely operating requirements of future heat sources.

Designing new hydronic systems for low water temperature operation is a rational way to “future-proof” portions of the system that have the potential to last for many decades.

Another trend that provides clues to the future is the evolution of energy codes pertaining to building

construction. With every new update cycle, widely recognized codes such as the International Energy Construction Code (IECC), or standards such as ASHRAE 90.1 and 90.2, either mandate or highly recommend methods for reducing building heating and cooling loads. Buildings are using materials and glazings with higher and higher R-values. Air leakage rates are being reduced. Many programs at both state and federal levels are incentivizing measures that reduce building heating and cooling load.

These changes will inevitably continue to reduce the design heating and cooling loads of buildings. In single family homes, the loads could eventually be reduced to the point where combustion-type heat sources are not available with heating capacities low enough to provide high-efficiency operation. Heating loads could potentially become low enough that the monthly service charges associated with having a natural gas service to the building become a major portion of the utility invoice in comparison to the cost of the natural gas consumed. The economics of heating a low-energy-use home using natural gas will become increasingly unappealing, versus having a single utility service for electricity. This again suggests that electrically driven heat pumps are poised to gain market share. When those heat pumps are combined with hydronic distribution systems, it will be crucial that the latter can operate at low water temperatures.

GLOBAL TRENDS IN HYDRONICS:

The largest markets for hydronic heating are in Europe and Asia. These markets are ahead of North America in transitioning to low water temperature distribution systems. For example, the global demand for air-to-water heat pumps in 2017 was 2.66 million units. Most of this demand was in Asia and Europe, with North America being a very small fraction of the overall market.

The North American hydronics market will likely continue to be heavily influenced by new products and approaches developed in the Asian and European markets. One of the most consistent of these trends is increasing use of low water temperature distribution systems.

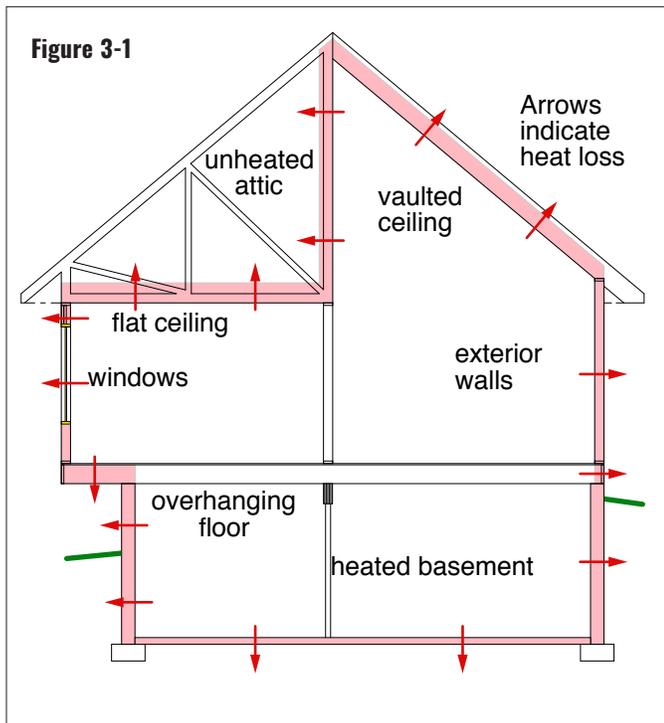
The stage is set for what's coming in hydronic heating technology. The necessary materials and methods for constructing new hydronic heating systems for low water temperature operation are well established in North America. However, there are tens of thousands of existing hydronic heating systems that are not currently able to operate at low water temperatures. The remainder of this issue will discuss ways of modifying those systems for such operation.

METHODS FOR REDUCING

This section describes concepts for reducing the water temperature in existing hydronic systems without compromising thermal comfort. Some are based on reductions in building heating load. Others involve the addition of supplemental heat emitters. Outdoor reset control is also discussed as a way to lower water temperatures under partial load conditions. These methods can be applied individually or in combination with each other. Section 4 then continues the process by showing specific ways to modify existing distribution piping when supplemental heat emitters are required.

REDUCING BUILDING LOAD:

The “thermal envelope” of a building is the collection of all surfaces that separate heated space from unheated space. Examples include exterior walls, windows, doors, ceilings under unheated attics, and overhanging or suspended floors having exterior air beneath them. Other examples include walls that separate heated space from an unheated attached garage, or heated space from unheated spaces within a roof truss. Basement walls and floors are also part of the thermal envelope. Figure 3-1 illustrates the concept of a building thermal envelope.



Building thermal envelope improvements reduce the heating load of a building. They also reduce the total space-heating energy required by a building during every subsequent year after they are made. In many cases, they are among the most cost-effective changes that can

be made to reduce heating and cooling costs, especially when the original building is old, poorly insulated or leaky.

These improvements could take the form of added insulation in the building envelope surfaces. Significant load reductions are also possible by replacing old single pane or early generation double pane windows with modern double or triple glazings that include low-E films and argon-filled cavities between panes. Modern methods of reducing air leakage, in combination with blower door testing to verify the results, can also markedly reduce heating loads. These methods are not mutually exclusive. In many cases, they are accomplished simultaneously as part of “deep energy retrofits” to existing buildings.

Figure 3-2



Source: <https://socialsecuritydisability-benefits.com>

There are several established methods for estimating the effect of adding insulation, changing the R-value of windows and evaluating the benefit of reduced air leakage. Most of these methods are based on software implementation of established heating load estimating procedures such as ACCA manual J.

The net result of all building thermal envelope improvements will be a new, lower value for the building’s design heat loss. This should be the starting point for assessing the impact on the water temperature required by the existing distribution system, assuming no changes to piping or heat emitters.

The change in supply water temperature at design load is proportional to the change in design load. The reduced supply water temperature under design load conditions can be determined using formula 3-1:

Formula 3-1:

$$T_{new} = T_{in} + \left(\frac{Q_{new}}{Q_{existing}} \right) \times (T_{De} - T_{in})$$

Where:

T_{new} = supply water temperature at design load after building envelope improvements (°F)

T_{in} = desired indoor air temperature (°F)

Q_{new} = design heating load after building envelope improvements (Btu/hr)

$Q_{existing}$ = existing design heating load (before improvements) (Btu/hr)

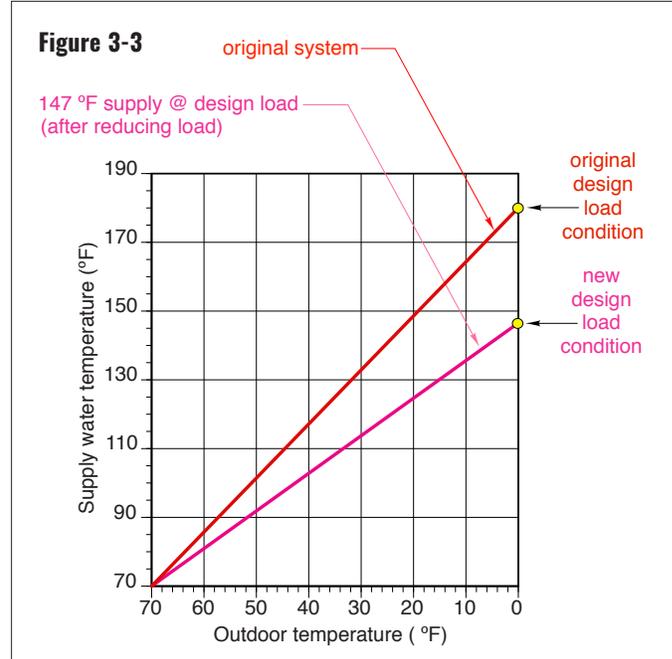
T_{De} = existing supply water temperature at design load (before improvements) (Btu/hr)

Consider a building with an existing design heating load of 100,000 Btu/hr, based on maintaining an interior temperature of 70°F. The existing hydronic distribution system uses standard fin-tube baseboard and requires a supply water temperature of 180°F at design load conditions. Assume that improvements to the building’s thermal envelope have reduced the design load from 100,000 Btu/hr to 70,000 Btu/hr. The new supply water temperature to the existing distribution system at design load is calculated as:

$$T_{new} = 70 + \left(\frac{70,000}{100,000} \right) \times (180 - 70) = 147^\circ F$$

Figure 3-3 shows the relationship between supply water temperature and outdoor temperature based on the original design load, and the reduced load after the building envelope improvements were made.

Reducing the design heating load from 100,000 Btu/hr to 70,000 Btu/hr reduces the required supply water temperature under design load conditions from 180°F to 147°F. Although this is certainly an improvement, it’s still above what some renewable energy heat sources can consistently provide. For example, most geothermal water-to-water heat pumps, as well as air-to-water heat pumps, cannot heat water to 147°F. Most current-generation hydronic heat pumps are limited to water temperatures of about 125°F. It would be possible to operate a modulating/condensing boiler at 147°F when required. It would also be possible to limit the lower



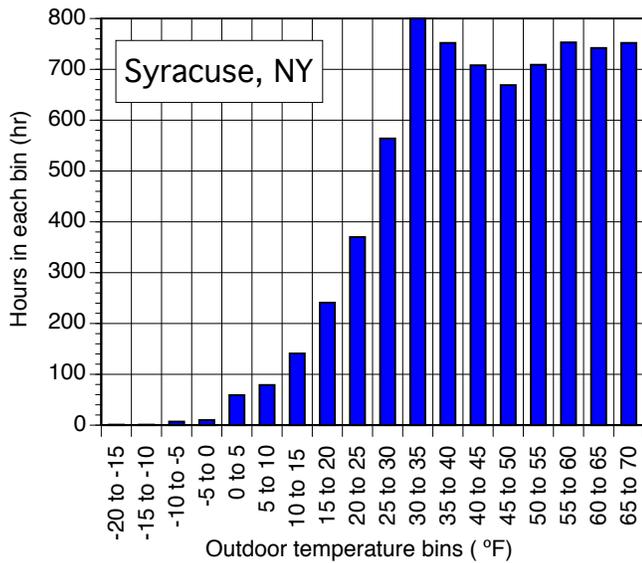
temperature of a thermal storage tank heated by a pellet boiler to not lower than 147°F. However, neither of these settings would allow these boilers to attain their highest potential efficiency or lowest emissions.

PARTIAL LOAD vs. DESIGN LOAD CONSIDERATION:

It’s important to consider that the 147°F supply water temperature determined in the previous example is only required during design load conditions. Under partial load conditions the supply water temperature could be significantly lower, as shown by the sloping lines in figure 3-3. For example, when the outdoor temperature is 40°F, the required supply water temperature indicated by the lower sloping line in figure 3-3 is about 104°F. This is easily within the operating range of the previously mentioned heat pumps. It would also improve the efficiency of a modulating/condensing boiler, and the temperature cycling range of thermal storage used with a pellet boiler.

Another way to view this situation is to consider how much space-heating energy is used during the coldest weather, when a heat source with a limited upper water temperature of perhaps 120°F can no longer provide sufficient supply water temperature. During that time, an auxiliary heat source could be used instead of the temperature-limited heat source. If the amount of energy used under such conditions is relatively small in comparison to the total seasonal heating energy required, it would be reasonable to use the lower-temperature heat source for the majority of the heating load, and rely on the auxiliary heat source for the peak load conditions.

Figure 3-4



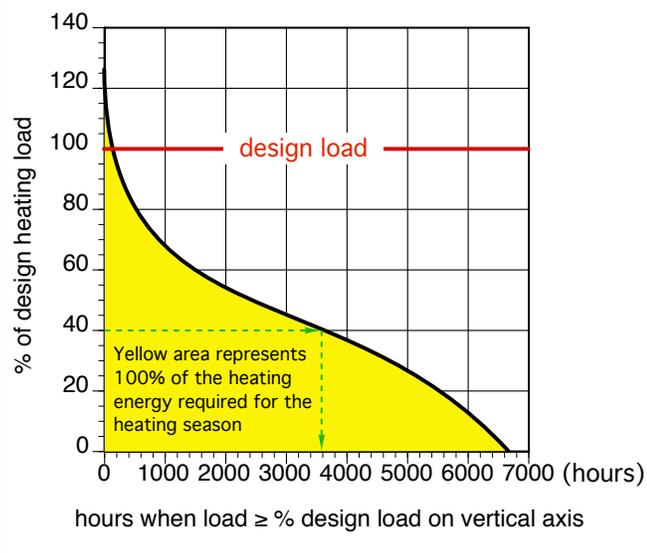
bulb temperatures are between 0 and 5°F. Other bins would show the hours where this temperature is between 10°F and 15°F, or 50°F to 55°F. The bins range all the way from the lowest outdoor temperature to the highest. Figure 3-4 shows bin temperature data for Syracuse, NY. Each blue bar represents a bin.

Since space-heating loads are approximately proportional to the temperature difference between inside and outside, and assuming that the indoor temperature will be maintained at 70°F, it is possible to use bin temperature data to create a “heating load duration graph,” as shown in figure 3-5.

This graph shows the percent of design heating load on the vertical axis versus the number of hours that the heating load is equal to or greater than the percent on the vertical axis. For example, in this location (Syracuse, NY), the space heating load is equal to or greater than 40% of design heating load approximately 3600 hours per year.

The yellow area under the curve in figure 3-5 is proportional to the total space-heating energy required for a complete year.

Figure 3-5



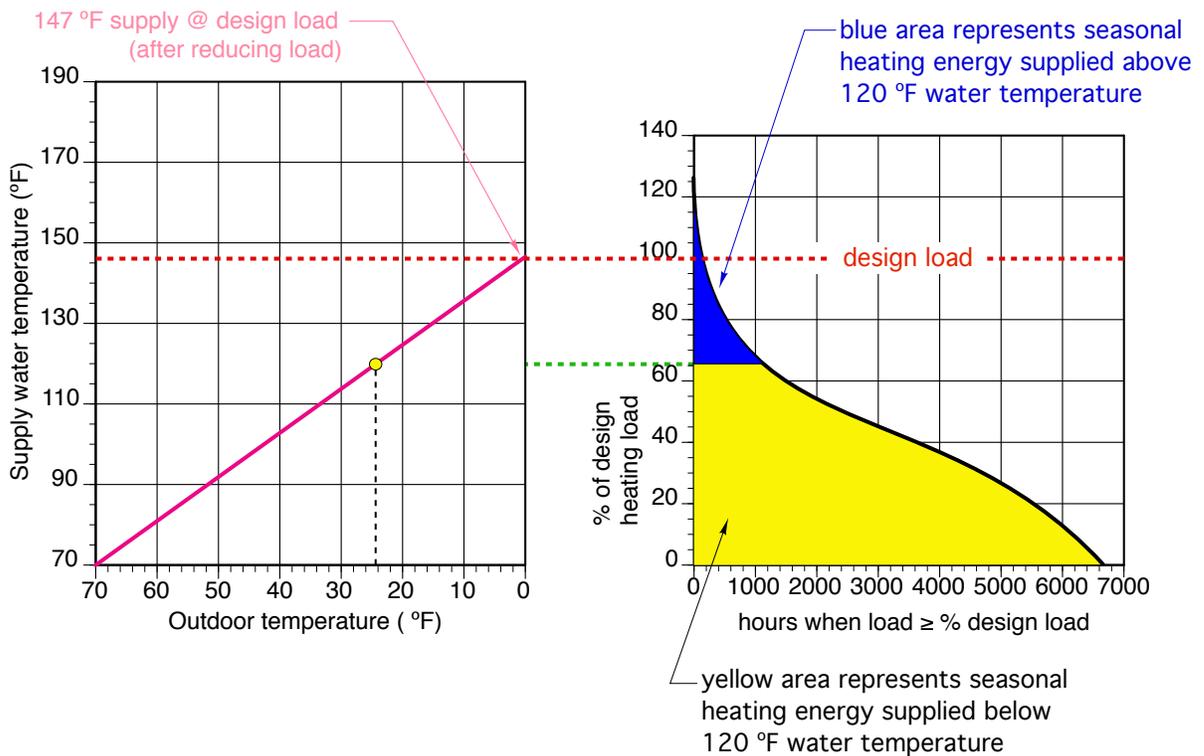
By combining and aligning figures 3-3 and 3-5, it's possible to determine the percent of total space-heating energy required when the supply water temperature must equal or exceed a specific value. Figure 3-6 shows this for a supply water temperature of 120°F.

Notice that the two design load conditions (e.g., 147°F supply water temperature on left graph, and 100% of design heating load on the right graph) are aligned. To determine the percent of total space-heating energy required when the supply water temperature must equal or exceed 120°F, a green line is drawn from the 120°F value on the vertical axis of the left graph horizontally across to the right graph. The blue shaded area above the green line is proportional to the space-heating energy required at or above 120°F supply water temperature. This area was calculated to be 6.5% of the total yellow shaded area in figure 3-5. Thus, only 6.5% of the seasonal energy required to heat the building with the reduced design load of 70,000 Btu/hr, and no modification of the distribution system or heat emitters, would require a supply water temperature at or above 120°F.

A supply water temperature of 120°F is within the operating range of most current generation hydronic heat pumps. This implies that an appropriately sized hydronic heat pump could supply about 93.5% of the total space-heating energy required by this building after its design load was reduced from 100,000 Btu/hr to 70,000 Btu/hr, with a corresponding reduction in design load supply

One way to evaluate such situations is by examining bin temperature data for the location of the system. Bin temperature data is available for many locations in North America from references such as ASHRAE and ACCA Manual J. Bin data indicates the number of hours in a year that long-term average outdoor temperatures fall within one of several “bins” that are 5°F “wide.” For example, at some locations one bin might indicate the hours per year when long-term average outdoor dry

Figure 3-6



water temperature from 180°F to 147°F. The remaining 6.5% could be provided by another heat source in the building. One example would be an existing fossil fuel boiler, and another would be a retrofitted electric boiler. The supplemental heat could even be supplied by plug-in space heaters or a wood stove, although these would require manual startup and shutdown.

The preceding example demonstrates that building envelope improvements have the potential to significantly reduce supply water temperature requirements, and thus bring existing buildings into the realm where current generation hydronic heat pumps or other renewable energy heat sources are viable.

Beyond this finding is the more obvious benefit that load reductions reduce space-heating energy needs. The 30% reduction in design load from 100,000 Btu/hr to 70,000 Btu/hr would infer a nominal 30% reduction in total space-heating energy over each subsequent heating season.

Energy conservation is a powerful and often cost-effective tool. Its impact should not be minimized in favor of immediately moving to other options for reducing system operating temperature.

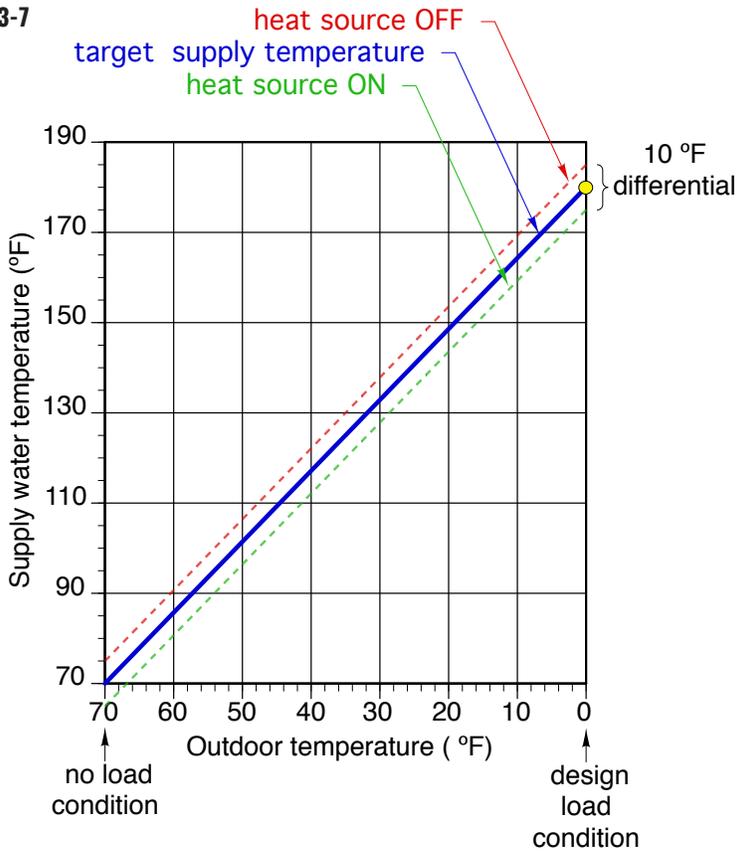
OUTDOOR RESET:

Figure 3-3 shows how the required supply water temperature for the building's heating system varies with outdoor temperature, both before and after thermal envelope improvements reduced the design heating load. The sloping lines in the graph in figure 3-3 are called "reset lines." They represent the theoretical proportional relationship between space-heating load and the difference between indoor and outdoor air temperature. They also represent an approximately proportional relationship between the heat output from any hydronic heat emitter and the supply water temperature to that emitter.

Using outdoor reset allows supply water temperatures that are lower than design load supply water temperature to maintain building comfort over much of the heating season. As previously shown, this often allows existing and unmodified hydronic distribution systems to operate at supply water temperatures that fall within the capability of lower-temperature heat sources – provided that some auxiliary heat source is available to meet peak heating demands.

One of the most common ways to implement outdoor reset is to add a relatively inexpensive controller to operate the heat source. That controller continuously monitors

Figure 3-7



outdoor temperature and uses that temperature, along with its settings, to calculate a “target” supply water temperature for the heat emitters. This target value varies with outdoor temperature and represents the lowest supply water temperature that could maintain comfort within the building based on the current outdoor temperature.

When the heat source is a simple on/off device, such as a single speed compressor in a heat pump, an oil burner or a basic gas valve, the outdoor reset controller turns it on when there’s a call for space heating and the measured supply water temperature is slightly below the target value. Assuming the call for heating persists, the controller keeps the heat source on until the measured supply water temperature is slightly above the target value. The differential below and above the target temperature is user-adjustable on most outdoor reset controllers. The narrower the differential, the closer the supply water temperature stays to the target temperature. However, if the differential is too narrow, the heat source will short-cycle, which should be avoided. Figure 3-7 shows a typical reset line for a “high-temperature” fin-tube baseboard system in which water temperature from the heat source is controlled by an outdoor reset controller that has been set for a 10°F differential between heat source “ON” and “OFF” conditions.

When the outdoor temperature is 0°F, and there is a call for space heating from one or more room thermostats, a system operating based on the controller settings shown in figure 3-7 would turn the heat source on when the supply water temperature was 175°F or less. Assuming the call for heating persists, the controller would turn the heat source off when and if the supply water temperature reaches

Figure 3-8

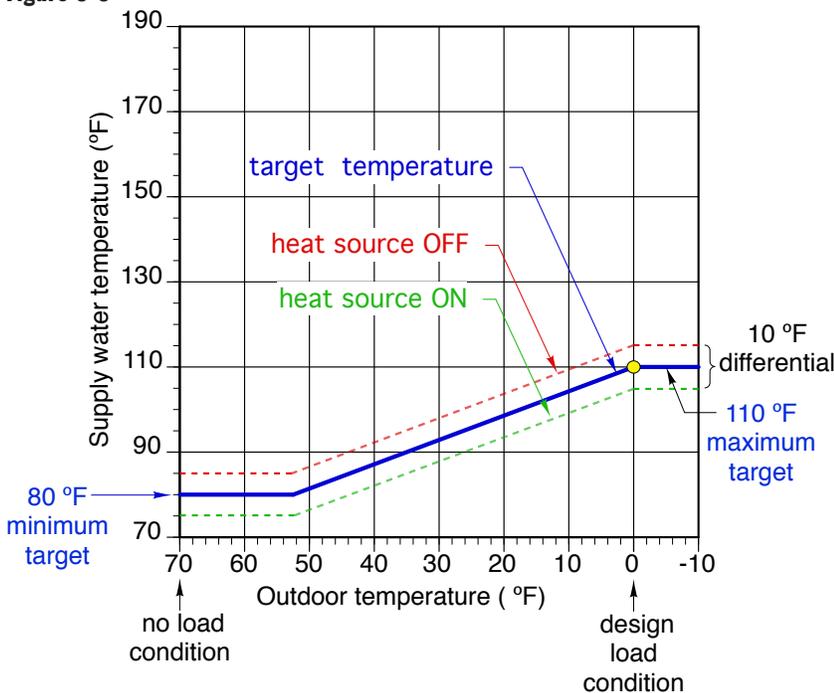
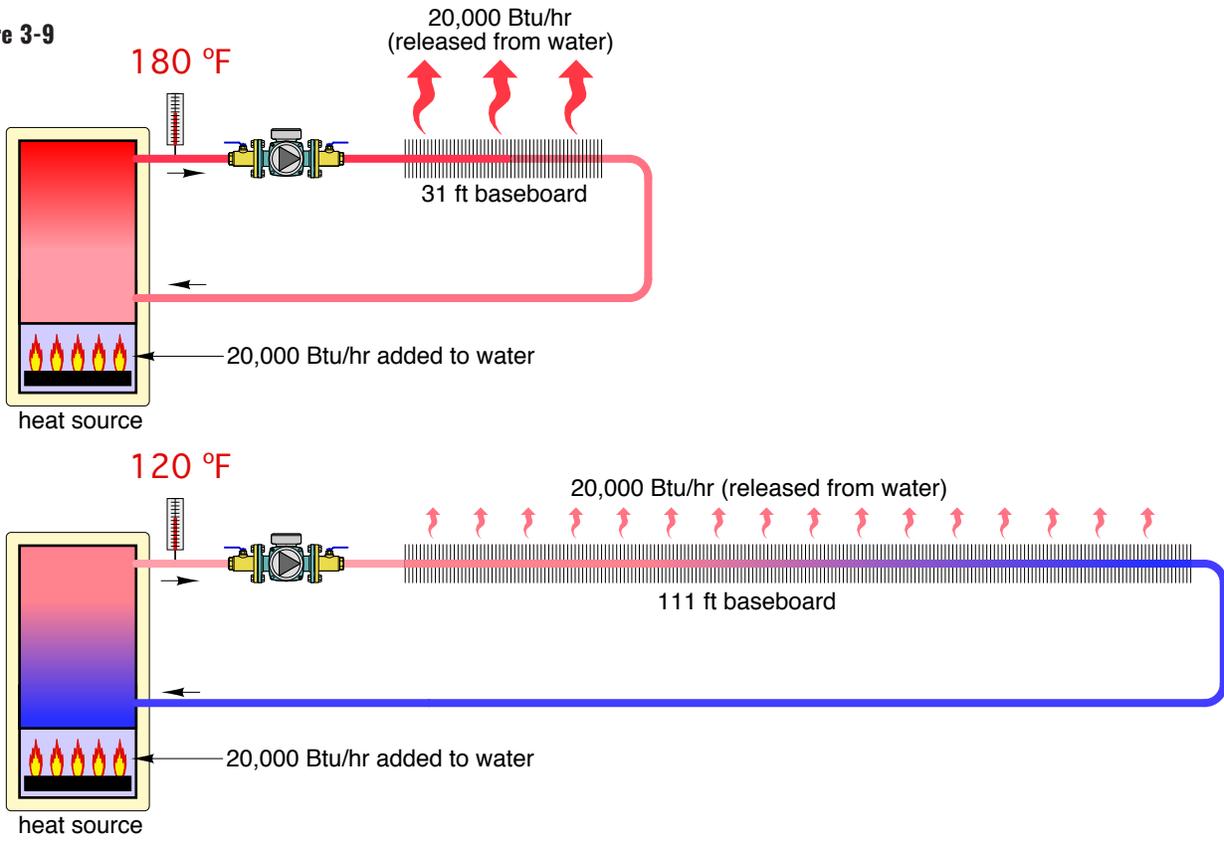


Figure 3-9



185°F. Because the 10°F control differential is centered on the target temperature (e.g., 5°F below and 5°F above) the average heat output from the heat emitters during the heat source on-cycle should be about the same as if the supply water temperature remained at a constant 180°F.

The slope of the reset line, the on/off differential, and both a minimum and maximum target temperature can all be set on modern reset controllers. Figure 3-8 shows another example where the reset line has been configured for a low-temperature slab heating system, along with minimum and maximum values for the target temperature.

The blue reset line is oriented such that it would pass through 70°F supply water temperature at 70°F outdoor air temperature. This, in theory, represents a “no load” condition. However, this outdoor reset controller has been set for a minimum target temperature of 80°F, and thus, when there’s a call for heating, the lowest the water temperature can get before the heat source is turned on is $(80-10/2) = 75^\circ\text{F}$.

Similarly, the controller has been set for a maximum target temperature of 110°F. Even if the outdoor temperature drops below 0°F (e.g., design load condition), the highest

the supply water temperature will get is $(110+10/2) = 115^\circ\text{F}$. Between outdoor temperatures of 0°F and approximately 52°F, the target temperature is based on the sloping reset line.

ADDING HEAT EMITTERS:

There will be situations where thermal envelope improvements as a means of lowering the supply water temperature are either impractical or otherwise limited. There are also situations where a new low-temperature heat source must provide design load without the assistance of any supplemental heater. In these cases, designers need to consider adding heat emitters to the existing distribution system as a means of lowering supply water temperature.

Any analysis of adding supplemental heat emitters must be based on the fundamentals of heat transfer. All three modes of heat transfer involve the area across which heat is being exchanged. When applied to a hydronic heating distribution system, this leads to a fundamental principle:

The greater the total surface exchange area of the heat emitters, the lower the required supply water temperature for a given rate of heat transfer.

A radiant floor, wall or ceiling panel is one example. For any specific panel construction (e.g., tube spacing, materials around tubing, floor covering, backside insulation, etc.), and any operating condition (e.g., water temperature, room temperature), the panel's heat output is always directly proportional to its area. Doubling the panel area without any change to panel construction or operating conditions doubles its heat output.

A similar tradeoff between heat exchange surface area and water temperature holds true for baseboard, panel radiators and fan-coils. Figure 3-9 shows two fin-tube baseboard circuits that each release heat at the same rate. The upper circuit contains 31 feet of baseboard and requires a supply water temperature of 180°F to dissipate 20,000 Btu/hr. The lower circuit contains 111 feet of the same baseboard. It only requires a supply water temperature of 120°F to dissipate 20,000 Btu/hr.

Figure 3-10 illustrates how increasing the physical size (height, width or depth) of a panel radiator reduces the water temperature required to meet both design load and partial load conditions.

The smallest panel radiator requires a supply water temperature of 180°F at design load. Increasing the panel size to "medium" reduces the required supply water temperature to 150°F, while providing the same design load heat output. The largest panel radiator further reduces the required supply water temperature to 120°F, again for the same design load heat output. The total surface area of a panel radiator can be increased by increasing the panel's width, height, depth, or any available combination of these dimensions. The depth dimension is typically determined by the number of water-filled "plates" in the radiator. One, two and three water plate panel radiators are commonly available. Most panel radiator suppliers offer a wide range of panel dimensions

to allow designers to fit the required heat output into available wall space. Some panel radiators are also available in tall, narrow configurations to make use of available vertical (rather than horizontal) wall space.

The sloping lines in figure 3-10 show how the required supply water temperatures for each size panel can be reduced as the outdoor temperature increases. These are supply temperature "reset" lines as discussed earlier in this section.

Figure 3-11 shows how increasing the number of tube passes within the coil of an air handler allows it to deliver a given rate of heat transfer with lower entering water temperature.

Companies that supply air handlers often provide options for the number of tube passes through the coil. As the number of tube passes increases, so does the size of the aluminum fins and the total surface area of the coil.

In addition to heat emitter surface area, the following factors need to be considered:

- Will the added heat emitters be the same make/model as the original heat emitters?
- What are the aesthetic preferences of the owners regarding additional heat emitters?
- Is there sufficient wall space to accommodate added heat emitters?
- How will piping from the new emitters be routed through the building?
- What are the installed costs of different supplemental heat emitters?
- Are any of the existing emitters at or near the end of their useful life?
- What piping modifications will be necessary to accommodate the added heat emitters?
- How will added heat emitters and modified piping affect the flow and head loss of the system?

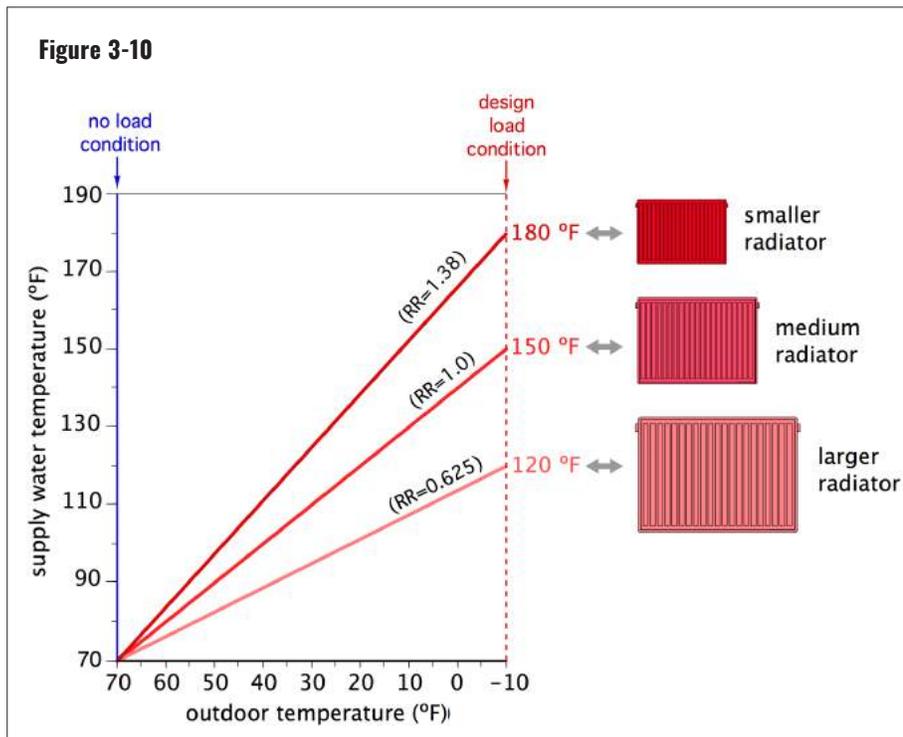
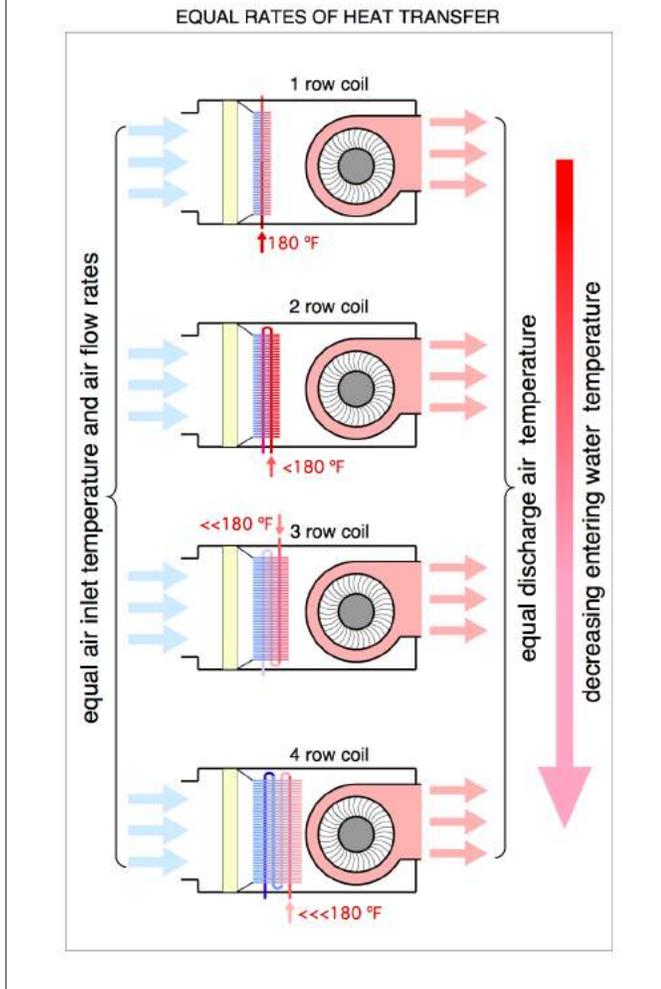


Figure 3-11



- Will the existing boiler be able to operate at lower water temperatures without sustained flue gas condensation? If not, what needs to be done to protect it?

ADDING MORE FIN-TUBE BASEBOARD:

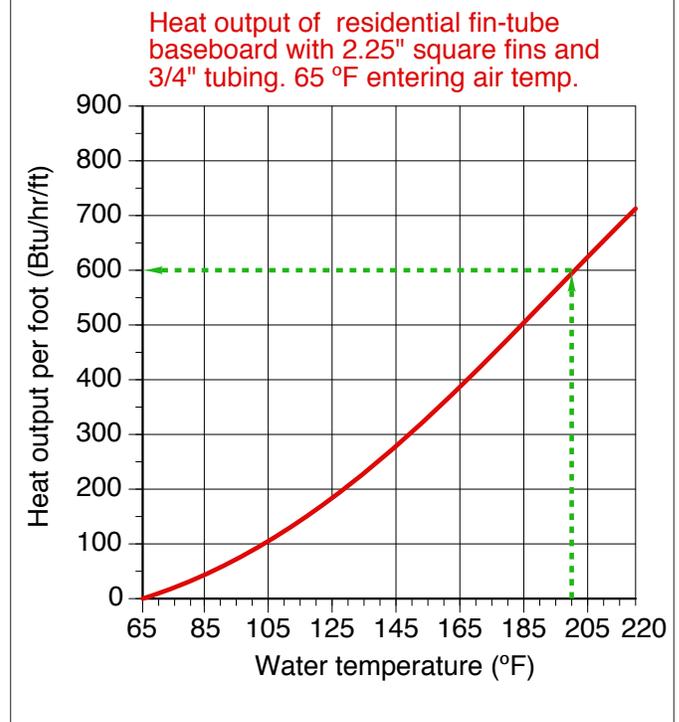
One of the most commonly encountered “legacy” hydronic distribution systems in North America uses one or more circuits containing multiple fin-tube baseboards connected in series. Many of these systems were installed at a time when high-temperature boiler water was the norm, and thus the fin-tube baseboard was often sized assuming available water temperatures of 180°F or higher. As discussed in section 1, many heat output rating tables for fin-tube baseboard still show outputs for water temperatures up to 220°F. These systems present a significant challenge when the goal is to change the building’s heat source from a traditional high-temperature boiler to a lower-temperature heat source such as a hydronic heat pump.

ADDING THE SAME TYPE OF FIN-TUBE BASEBOARD:

Consider an existing distribution system with fin-tube baseboard that requires a supply water temperature of 180°F at design load. A designer wants to evaluate how much of the same baseboard would be required to significantly lower the system’s required supply water temperature at design load. Once this is determined, the designer can evaluate the practicality of adding this supplemental baseboard to the system.

The existing baseboard is a standard “residential”-grade product. Its heating element has a rated output of 600 Btu/hr/ft based on 200°F water temperature and 65°F entering air temperature. Figure 3-12 shows how the heat output of this baseboard varies over a range of water temperatures, assuming the incoming air temperature at floor level of 65°F and a flow rate of 1 gallon per minute through the element.

Figure 3-12



The following procedure can be used to determine how much additional fin-tube baseboard is required:

Step 1: Accurately determine the building’s design heating load using ACCA Manual J or equivalent procedures.

Step 2: Determine the total length of fin-tube in the existing distribution system. Do not include the length of

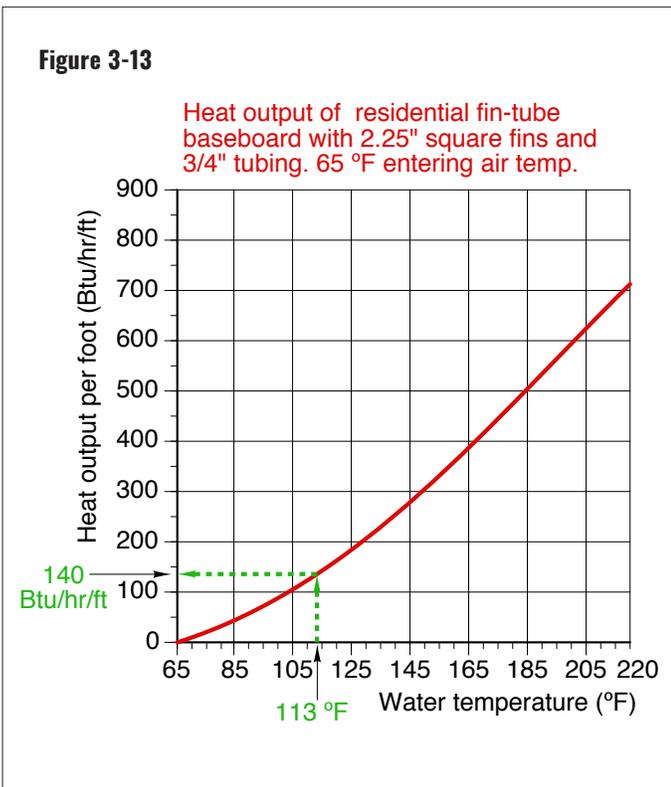
tubing that doesn't have fins on it. The existing fin-tube length will be designated as L_e .

Step 3: Determine the desired (lower) supply water temperature for which the system is to supply design load output.

Step 4: Estimate the lower average water temperature that would be present in the modified system. A reasonable estimate is to subtract 7°F from the supply water temperature determined in Step 3. This assumes the temperature drop in the distribution system at design load conditions will be approximately 14°F.

Step 5: Locate the estimated average circuit water temperature determined in Step 4 on the horizontal axis in figure 3-12. Draw a vertical line up from this point to the red curve. Draw a horizontal line from this intersection to the vertical axis and read the heat output of the baseboard at the average circuit water temperature. This number is designated as q_L . The green lines and numbers in figure 3-13 show how q_L is determined for an average circuit water temperature of 113°F (e.g., 120°F - 7°F). In this case, the value for q_L is 140 Btu/hr/ft.

Step 6: Determine the required length of baseboard to be added using formula 3-2.



Formula 3-2:

$$L_{added} = \frac{\text{design load}}{q_L} - L_e$$

Where:

L_{added} = length of fin-tube of same make/model baseboard to be added (feet)

design load = design heating load of building (Btu/hr)

q_L = output of baseboard at the lower average circuit water temperature (Btu/hr/ft)

L_e = total existing length of baseboard in system (feet)

Here's an example: Assume a building has a calculated design heating load of 40,000 Btu/hr, and its distribution system contains 120 feet of standard residential fin-tube baseboard. The goal is to reduce the supply water temperature to 120°F at design conditions, using more of the same baseboard. Also assume that the temperature drop of the distribution system under design load conditions will be 14°F. Determine the amount of baseboard that must be added:

Step 1: The design load has been calculated as 40,000 Btu/hr.

Step 2: The total amount of fin-tube element in the system is 120 feet.

Step 3: The lower supply water temperature at design load will be 120°F.

Step 4: The lower average circuit water temperature will be 120 - (14/2) = 113°F.

Step 5: The output of the fin-tube at an average circuit water temperature of 113°F is determined from figure 3-13 as 140 Btu/hr/ft.

Step 6: The required additional length of baseboard is now calculated using formula 3-2:

Formula 3-2:

$$L_{added} = \frac{\text{design load}}{q_L} - L_e = \left[\frac{40,000 \frac{\text{Btu}}{\text{hr}}}{140 \frac{\text{Btu}}{\text{hr} \cdot \text{ft}}} - 120 \right] = 166 \text{ ft}$$

Although it might be possible to add 166 feet of baseboard to the system, it would require lots of available wall space (e.g., space not already occupied by existing baseboard, cabinets, doorways, etc.). In most buildings, adding this much baseboard is not a practical solution.

Alternatives include using baseboard with higher heat output or using other types of heat emitters to achieve the necessary design load output.

ADDING HIGH OUTPUT FIN-TUBE BASEBOARD:

Due to the impracticality of adding 166 feet of additional standard “residential” baseboard, the designer wants to evaluate use of high-output fin-tube baseboard as a means of reducing system water temperature. An example of high-output fin-tube baseboard heat emitters is shown in figure 3-14.

Figure 3-14



Courtesy of Smith Environmental

This baseboard has significantly more fin surface area, as well as two tubes that supply heat to these larger fins. It’s a larger heat exchange element within a housing comparable in size to that of a standard baseboard. Figure 3-15 shows the heat output of this “high-output” fin-tube baseboard as a function of water temperature. The heat output of standard “residential” fin-tube is also shown for comparison.

The steps from the previous procedure can be modified to determine the amount of high-output fin-tube baseboard that is required to reduce the supply water temperature to the system under design load.

Steps 1-4: Same

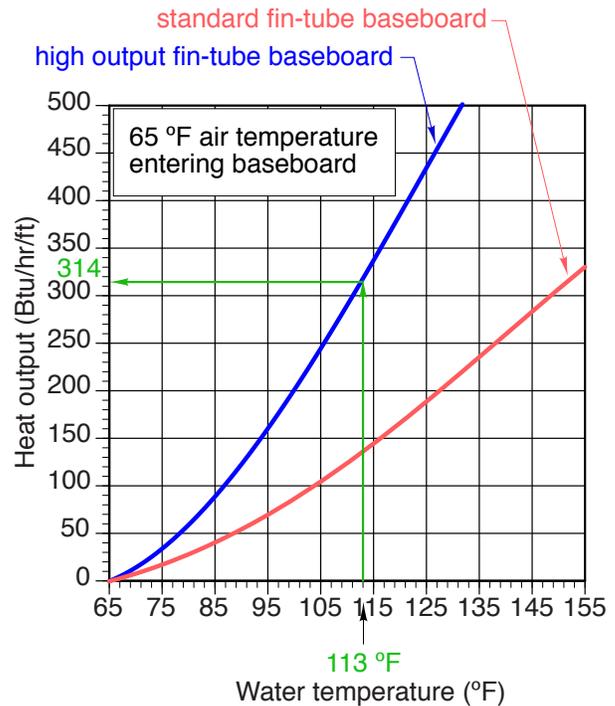
Step 5: Determine the output of high-output baseboard at the average circuit water temperature using figure 3-15 (or other product specifications for a specific baseboard make and model).

Step 6: The required length of high-output baseboard to be added is found using formula 3-2.

Formula 3-2:

$$L_{ho} = \frac{\text{design load} - (q_L)(L_e)}{q_{ho}}$$

Figure 3-15



Where:

- L_{ho} = length of high-output fin-tube baseboard to be added (feet)
- design load = design heating load of building (Btu/hr)
- q_L = output of existing baseboard at the lower average water temperature (Btu/hr/ft)
- L_e = total existing length of baseboard (finned-tube only) in system (feet)
- q_{ho} = output of high-output baseboard at the lower average water temperature (Btu/hr/ft)

Consider the previous building with a design heating load of 40,000 Btu/hr, and a distribution system containing 120 feet of standard residential fin-tube baseboard. The goal is to reduce the supply water temperature under design load to 120°F. Additional high-output baseboard will be added to the existing baseboard to allow this lower water temperature operation. Assume that the temperature drop of the distribution system at design load is 14°F. Also assume that the existing baseboard has an output of 140 Btu/hr/ft at average circuit water temperature of 113°F. Determine the amount of high-output baseboard required based on the performance shown in figure 3-15.

Step 1-4 already done in previous example.

Step 5: The output of high-output fin-tube at an average water temperature of 113°F is determined from figure 3-13 as 314 Btu/hr/ft.

Step 6: The required length of high-output baseboard to add to the system is found using formula 3-2.

$$L_{ho} = \frac{\text{design load} - (q_L)(L_c)}{q_{ho}} = \frac{40,000 - (140)(120)}{314} = 74 \text{ ft}$$

Although this is a substantial reduction compared to the 166 feet of additional standard baseboard required in the previous example, it is still a considerable total length. The building must be carefully evaluated to see if this additional length of baseboard can be accommodated.

ADDING OTHER HEAT EMITTERS:

A building with existing fin-tube baseboard doesn't necessarily have to use more fin-tube baseboard as supplemental heat emitters. Other options include panel radiators, fan-coils, and radiant floor, wall or ceiling panels.

Figure 3-16



Courtesy of Brian Just

Figure 3-16 shows an example of a panel radiator that was retrofitted into a small “knee wall” space in a home.

This lightweight radiator is suspended by three brackets fastened to the wall. Two 1/2” PEX-AL-PEX tubes for supply and return water flow connect at the lower right corner of the radiator. These tubes are routed back through the home’s framing to a central manifold station. The radiator is equipped with a thermostatic valve, seen in the upper right corner of the radiator, allowing for individual room temperature control.

FAN-COILS:

Fan-coils are another option for supplemental heat emitters. They are especially appropriate when available wall space is limited. For example, based on published ratings, one commercially available fan-coil that's only

2.5 feet wide has a heat output equivalent to 47 feet of residential-grade fin-tube baseboard when operated at the same supply water temperature and inlet air temperature.

Fan-coils equipped with condensate drip pans are also one of the only available heat emitters that can also be used for cooling. As such, they are well-suited when an existing distribution system is retrofitted with a hydronic heat pump. Not all hydronic fan-coils have drip pans, and those that don't should never be used for chilled-water cooling, in which water temperatures can be well below the room's dewpoint temperature.

Figure 3-17 shows an example of a small “console” fan-coil that's designed to be mounted to a wall at floor level. Room air enters the lower grill, passes up through the internal coil and is discharged through the top grill.

Figure 3-17



Courtesy of SpacePak

Heat outputs for the fan-coil shown in figure 3-17 that use 120°F entering water temperature range from 8,500 Btu/h for a 35-inch-wide unit to 17,800 for a 59-inch-wide unit. Associated cooling capacity using 50°F entering water temperature ranges from 4,789 Btu/hr for the 35-inch-wide unit to 10,900 Btu/hr for the 59-inch-wide unit.

Several manufacturers of console fan-coils also offer units that can be recessed into a wall cavity with only their front panel projecting a fraction of an inch outside the wall. Careful planning is needed to coordinate the dimensions of the required wall cavity and the specific fan-coil to be mounted into that cavity.

In addition to supply and return piping, most fan-coils require 120 VAC power. If used for chilled-water cooling, designers also need to plan for a condensate drainage tube with a continuous downward slope to a drainage point.

Another type of fan-coil is called a “high wall” unit. Figure 3-18 shows an example.

Figure 3-18



This unit is equipped with an internal drip pan, allowing it to be used for heating as well as chilled-water cooling. Room air is drawn into the top of the enclosure, passed by the coil and discharged through a slot at the base of the unit. This fan-coil has variable-speed blower and discharge louvers that open and oscillate to better mix discharge airflow with room air. Rated heating and cooling capacities depend on coil size. Models using the enclosure shown in figure 3-18 models are available with heating capacities ranging from 8,123 Btu/hr to 14,641 Btu/hr using 120°F entering water temperature, and cooling capacities from 5,085 to 9,279 Btu/hr using 50°F entering water temperature.

Small fan-coils can be controlled by a room thermostat or by controls integral to the fan-coil. Some modern units can also be controlled using handheld remotes. Some units have variable-speed ECM blowers that change speed depending on operating mode.

SIZING SUPPLEMENTAL HEAT EMITTERS:

The size and number of supplemental heat emitters required for a given reduction in supply water temperature is based on a simple and consistent procedure.

Step 1: Perform an accurate room-by-room heat loss estimate for the building.

Step 2: Select a supply water temperature for design load conditions. When the new heat source will be a hydronic heating pump, biomass boiler with storage or solar thermal collectors with storage, the maximum suggested value for this temperature is 120°F.

Step 3: Determine the total heat output available from all the existing emitters at the new (lower) design load supply water temperature.

Step 4: Subtract the output determined in Step 3 from the design load to determine the necessary heat output for the supplemental heat emitters.

Step 5: Use performance data from the manufacturer at the reduced average water temperature to determine how many heat emitters or how many square feet of radiant panel are needed.

The fundamental sizing concept for supplemental heat emitters can be expressed as formula 3-3:

Formula 3-3:

$$Q_n = \text{design load} - Q_e$$

Where:

Q_n = required heat output of the new heat emitters at lower supply water temperature (Btu/hr)

design load = the design heating load of the building (Btu/hr)

Q_e = heat output of existing heat emitters at the lower supply water temperature (Btu/hr)

Here’s an example: A building has a calculated design load of 40,000 Btu/hr, and its distribution system contains 120 feet of standard residential fin-tube baseboard. The designer’s goal is to reduce the supply water temperature to 120°F under design load by adding supplemental panel radiators. The interior air temperature under design load will be maintained at 68°F. A search of available products finds that panel radiators measuring 24 inches high by 72 inches long by 4 inches thick can release heat at 14,245 Btu/hr when the average water temperature in the radiator is 180°F and the surrounding room air temperature is 68°F. How many of these radiators would be necessary to supplement the existing baseboard, assuming all heat emitters will operate at an average water temperature of 113°F (e.g., 120°F supply minus half of 14°F temperature drop at design load)?

Solution: It’s common to find panel radiator and fan-coil heat output ratings based on relatively high average water temperatures. In such cases, the heat output at a lower water temperature needs to be reasonably estimated.

Formula 3-4 can be used to correct the heat output of typical panel radiators for different average water temperatures and different surrounding air temperatures.

The calculated correction factor must be applied to the heat output ratings of the radiators based on an average water temperature that's 112°F (50 °C) above the surrounding room air temperature; for example, if the average water temperature in the panel is 180°F and the surrounding room air temperature is 68°F.

Formula 3-4:

$$CF = 0.001882(T_w - T_{air})^{1.33}$$

Where:

CF = correction factor

T_w = average water temperature in the panel radiator (°F)

T_{air} = air temperature surrounding the panel radiator (°F)

1.33 = an exponent

Start by determining the output of the existing baseboard at 113°F. For simplicity, assume the air temperature entering the baseboard at floor level is 65°F. This output has already been determined as 140 Btu/hr/ft from a previous example, using figure 3-13. Thus, the heat output from the existing baseboard will be approximately:

$$120 \text{ ft} \times \left(\frac{140 \text{ Btu}}{\text{hr} \cdot \text{ft}} \right) = 16,800 \text{ Btu/hr}$$

The remainder of the design heating load that must be met by the panel radiators is therefore:

$$40,000 \text{ Btu/hr} - 16,800 \text{ Btu/hr} = 23,200 \text{ Btu/hr.}$$

Use formula 3-4 to determine the correction factor for the panel radiators at the lower average water temperature of 113°F.

$$CF = 0.001882(T_w - T_{air})^{1.33} = 0.001882(113 - 68)^{1.33} = 0.297$$

The corrected heat output from each panel will be its rated output multiplied by the correction factor. For the selected panel radiators this is:

$$Q = 14,254 \times 0.297 = 4,233 \text{ Btu / hr}$$

The number of radiators needed is found by dividing the total supplemental heating needed by the output of each panel at the lower average water temperature.

$$\frac{23,200 \text{ Btu / hr}}{4233 \frac{\text{Btu / hr}}{\text{radiator}}} = 5.5 \text{ radiators}$$

Since it's not possible to install half a radiator, the designer could choose to add 6 of these radiators. This would slightly lower the required average water temperature. The designer might also elect to use a slightly higher average water temperature and get the required number of radiators down to 5. The choice depends on where the panels can be accommodated within the building, accepting the compromise that a higher average water temperature will slightly reduce the efficiency of the low-temperature heat source, and the lower cost of installing 5 rather than 6 panel radiators.

THERMAL PERFORMANCE OF FAN-COILS:

A similar procedure can be used when sizing supplemental fan-coils. A given make and model will have a rated heat output. The designer needs to correct that rated output for the reduced average water temperature, and then divide the corrected output into the total amount of supplement heating needed.

The heat output of a fan-coil unit depends on the temperature of water in the coil, as well as the temperature of the entering air. It also depends on the flow rates of water and air passing through the coil. Manufacturers typically provide data in the form of tables that show the effects of these operating conditions. When such charts are available for low water temperature operating conditions, they are the best reference. In the absence of such charts, a general principle that applies to fan-coils is that heat output is approximately proportional to the difference between entering water temperature and entering air temperature. This relationship can be used to estimate the heat output of fan-coils at water temperatures that may not be listed in manufacturer's literature. It can be mathematically expressed as formula 3-5.

Formula 3-5:

$$q_2 = \frac{(T_{w2} - T_{air2})}{(T_{w1} - T_{air1})} [q_1]$$

Where:

q₂ = estimated heat output of fan-coil at lower entering water temperature (Btu/hr)

q₁ = rated heat output of fan-coil at higher entering water temperature (Btu/hr)

T_{w2} = higher entering water temperature (°F)

T_{air2} = entering air temperature at which heat output is rated (°F)

T_{w1} = lower entering water temperature (°F)

T_{air1} = entering air temperature at lower entering water temperature condition (°F)

For example, a specific fan-coil is rated to release heat at a rate of 12,000 Btu/hr when supplied with water at 180°F and entering air temperature of 68°F. The flow rate of water through the fan-coil is 2 gpm. Estimate the fan-coil's heat output when supplied with 120°F water at the same flow rate and same entering air temperature.

Solution: Putting the stated operating conditions into formula 3-5 yields the estimated heat output at the 120°F entering water temperature.

$$q_2 = \frac{(T_{w2} - T_{air2})}{(T_{w1} - T_{air1})} [q_1] = \frac{(120 - 68)}{(180 - 68)} [12,000] = 5,571 \frac{Btu}{hr}$$

It is also possible to use multiple sizes of panel radiators, multiple models of air handlers, or even a combination of panel radiators and fan-coil, if that combination would better fit the installation requirements of the project. The fundamental principle remains the same. Determine the output of the existing heat emitters at the reduced average water temperature. Subtract that output from the design load to find the total supplemental heat output needed. Select any combination of heat emitters that can provide that total supplemental output when they all operate at the same reduced average water temperature.

ADDING RADIANT PANELS:

Radiant floor, wall or ceiling panels can also be used for supplemental heating. It's also possible to use combinations of these panels, or mix them with fin-tube baseboard, panel radiators or fan-coils.

There are many types of radiant panel constructions. In retrofit situations, it's less likely that high-mass radiant panel constructions, such as slab-on-grade or thin-slab "overpours," can be used due to weight and thickness requirements.

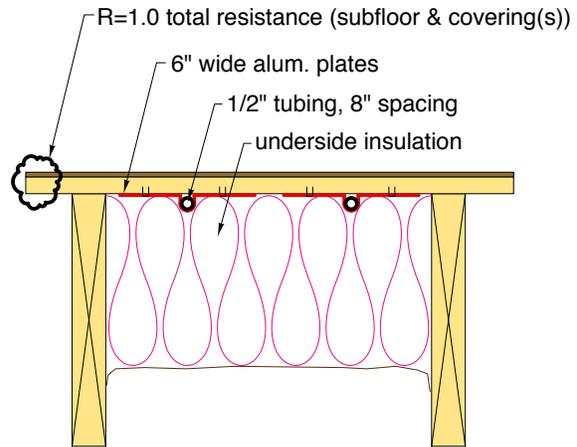
Three generic low thermal mass radiant panels that lend themselves to retrofitting are shown in figure 3-19.

All of the panels shown in figure 3-19 have tubing spaced 8 inches apart and use 6-inch wide aluminum heat transfer plates to diffuse heat away from the tubing and spread it across the panel surface. The radiant wall and ceiling panels use the same materials and assembly sequence. The radiant floor panel is specifically for situations where the tubing and heat transfer plates can be installed on the underside of a 3/4-inch-thick wood subfloor.

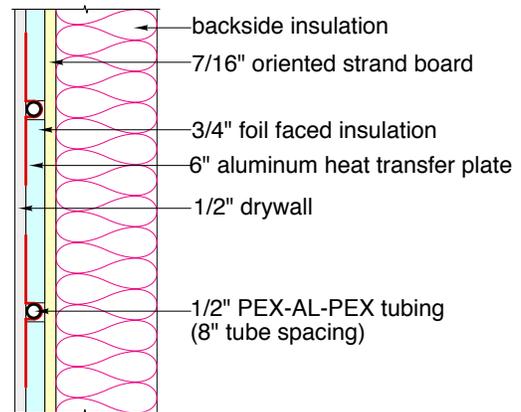
The thermal performance of each panel construction is shown in figure 3-20.

Figure 3-19

BELOW FLOOR TUBE & PLATE RADIANT FLOOR PANEL



RADIANT WALL PANEL



RADIANT CEILING PANEL

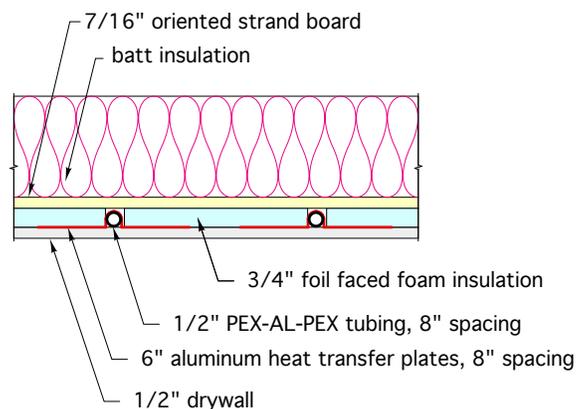
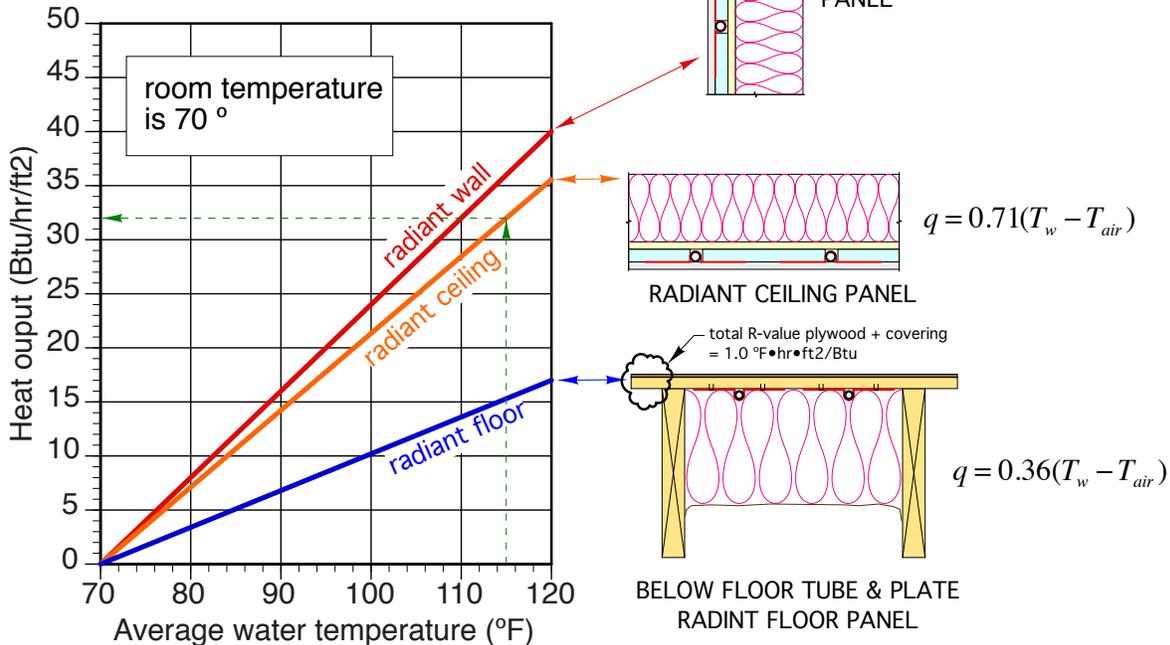


Figure 3-20



The graph shows the heat output of each panel, in Btu/hr/ft², based on the average water temperature in the panel circuit and an assumed room air temperature of 70°F. The radiant wall panel has the highest heat output. The radiant ceiling is a close second. The radiant floor panel has the lowest heat output due to the presence of the wood subfloor and some assumed covering (total R-value of the subfloor plus floor covering is assumed to be 1.0°F•hr•ft²/Btu).

To find the panel's output, start by finding the average water temperature on the horizontal axis. Draw a line up to the sloped line associated with the panel, and then a line from that intersection to the vertical axis. For example, the radiant ceiling panel shown, while operating with an average circuit water temperature of 115°F, in a room at 70°F, will release about 32 Btu/hr/ft².

The formulas in figure 3-20 can also be used to estimate the heat output per square foot of panel for different average water temperatures and room air temperatures.

The necessary area of each panel can be determined by dividing the required supplemental heat required by the output per square foot of panel determined from figure 3-20.

For example, assume the building with a design load of 40,000 Btu/hr, and 120 feet of standard residential baseboard will be retrofitted with radiant ceiling panels to provide supplemental heat output at an average water temperature of 120-7 = 113°F. At this average water temperature, the existing baseboard releases 140 Btu/hr/ft, or 16,800 Btu/hr total. That leaves a supplemental heat requirement of 40,000-16,800 = 23,200 Btu/hr. The designer wants to retrofit the building with the radiant ceiling panel construction shown in figures 3-19 and 3-20. At an average circuit water temperature of 113°F – the same as for the baseboard – and an assumed room air temperature of 70°F, the ceiling panel releases 0.71(113-70) = 30.5 Btu/hr/ft². Therefore, the total ceiling panel required for the supplemental heat requirement is: 23,200 / 30.5 = 761 square feet.

If the total amount of baseboard was divided up among the rooms in proportion to each room's design load as a percentage of the total building design load, then the same should be done for the supplemental radiant ceiling panel areas. For example, if the design load of a specific room was 20 percent of the total building design load, 20 percent of the total ceiling panel area, or (0.2x761=152 ft²) would be retrofitted into that room, if possible and if practical.

Figure 3-21



Courtesy of Chip Agnew

The radiant ceiling panel would not necessarily cover the entire ceiling of each room. When partial coverage is needed, the panel should generally start from the exterior wall and work toward the interior of the room. This places heat output closer to the higher heat loss areas of the room and helps increase the room's mean radiant temperature.

The panel area could end to form a coffered ceiling appearance, as shown in figure 3-21. Or the remaining non-heated area of the original ceiling could be furred out and covered with new drywall to provide a flush ceiling over the entire room.

EVALUATING EXISTING CAST IRON RADIATORS:

Many older homes have existing cast iron radiators. They may have been part of an original steam heating system, or they might have operated with water. A well-maintained cast iron radiator can last for many decades, and in some cases can be combined with a modern hydronic heat source such as an air-to-water heat pump or modulating/condensing boiler.

If the cast iron radiators were originally sized for a poorly insulated or uninsulated building, and that building was subsequently insulated, fitted with new windows and doors or air sealed, the existing radiators may only have to provide a fraction of the heat output that they were originally sized for. This may allow the radiators to operate at a much lower water temperature.

1-PIPE CAST IRON RADIATORS:

If the cast iron radiators were part of a 1-pipe steam system, they may or may not be suitable for conversion to a water-based system. The following points need to be considered:

1. Can the original pipe be removed from the radiator? In cases where the radiator is several decades old, there's a good chance that the pipe connection will be corroded. It might be able to be removed with a suitable pipe wrench, penetrating oil or after carefully heating the area around the connection with a hot air gun to slightly expand the metal. Be aware that any paint or other finish on the radiator could be damaged by such heating.

2. Assuming that the existing piping can be removed, the radiator needs to be checked internally. If heavily corroded, it's likely not suitable for reuse. If the internal corrosion is light, it might be able to be partially removed using commercially available hydronic descaling fluids. Such fluids may need to be circulated through the radiator for several hours for effective cleaning.

3. To be used in a hydronic system, it is necessary to have a connection at the radiator's upper header that can be fitted with an air vent such as shown in figure 3-22. The vent is needed to remove air from the radiator as it is filled with water.

4. Assuming that Steps 1, 2 and 3 above are doable, the radiator should also be pressure-tested to verify it can operate at a minimum of 30 psi pressure without leaking. Most steam heating systems used in smaller buildings only operate with 2-4 psi steam pressure, and thus the radiators are not subject to the pressures that could be experienced in a water-based system. Discovering that the radiator leaks after it's fully installed in a pressurized hot water system is obvious a costly problem.

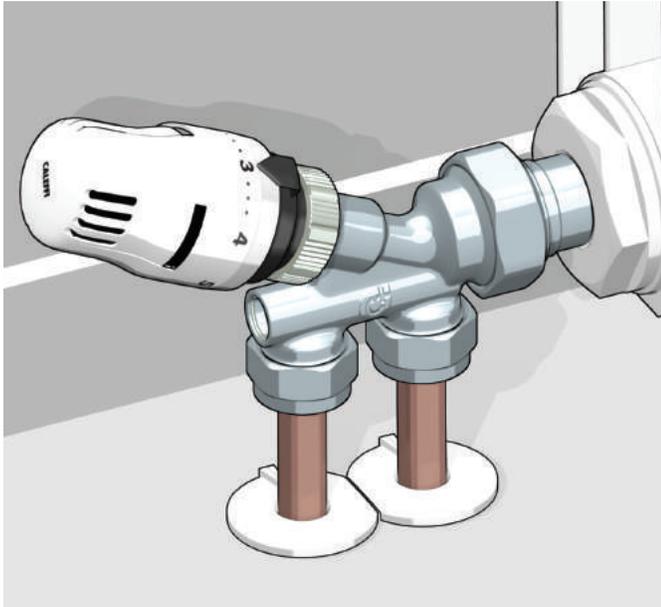
Figure 3-22



Figure 3-23a



Figure 3-23b



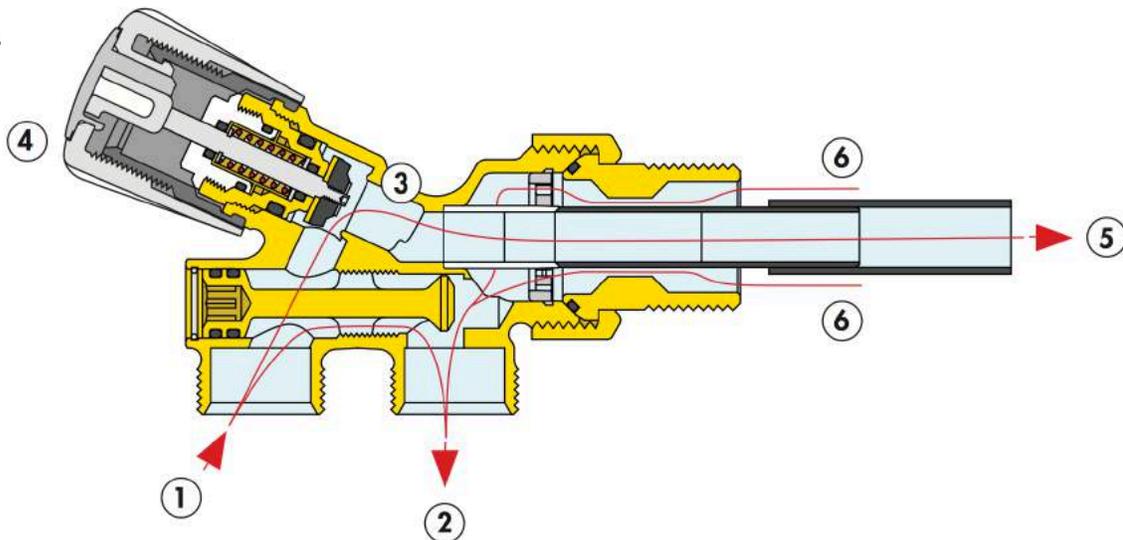
5. If the radiator has only one lower piping connection, it needs to be fitted with a “lance valve,” an example of which is shown in figure 3-23.

Lance valves have a tube that carries hot water to the far lower end of a radiator. The water then rises within the sections of the radiator by natural convection. Cooler water, which collects near the base of the radiator, is extracted through an annular opening at the threaded tailpiece of the valve. The portion of the valve that’s outside the radiator has connections for supply and return piping. Some lance valves can also be fitted with thermostatic operators. Some can also be configured with a bypass port, as shown in figure 3-24. This allows two or three radiators, each with their own lance valve, to be connected in a series circuit while also maintaining individual flow control through each radiator.

2-PIPE CAST IRON RADIATORS:

Existing cast iron radiators with two-pipe connections could have been used for steam or hot water heating. Their potential use in low-temperature systems also depends on evaluating internal corrosion (especially if previously used for steam heating), and having the radiator undergo and withstand a minimum 30 psi pressure test, and having a means of venting air from the upper portion of the radiator. If the piping connections on the radiator cannot be unthreaded, but the piping connected to the radiator is accessible, there is a possibility of connecting new piping

Figure 3-24



1. Heated water inlet
2. Valve outlet
3. Valve orifice controlling flow to radiator
4. Knob or thermostatic operator
5. Tube carrying heated water to far end of radiator
6. Flow returning from radiator

Figure 3-25



to existing piping (rather than directly to the radiator). Another possibility is connecting new piping to the inlet of an existing radiator valve, as shown in figure 3-25.

ESTIMATING HEAT OUTPUT FROM CAST IRON RADIATORS:

The heat output of cast iron radiators can be estimated for relatively low average water temperatures. The process requires that the surface area of the radiator be determined based on the radiator's type and dimensions. The output per square foot of surface area, which is based on the difference between room air temperature

Figure 3-26



Figure 3-27



Source: inspiredheating.com.uk

Figure 3-28



and average water temperature, can then be calculated. The radiator's surface area is multiplied by its heat output per square foot to determine its heat output at the lower water temperature.

There were many types of cast iron radiators created over several decades of common use. The most common three types are: column-type radiators, tube-type radiators and cast iron baseboard. The radiator shown in figure 3-26 is a tube radiator. The radiator in figure 3-27 is a column radiator. Figure 3-28 shows a cast iron baseboard.

The square footage of column-type and tube-type radiators is based on their height, width and the number of sections joined together to form the radiator. Figure 3-29 lists the square footage per section for several heights and widths of column-type cast iron radiators.

Figure 3-29

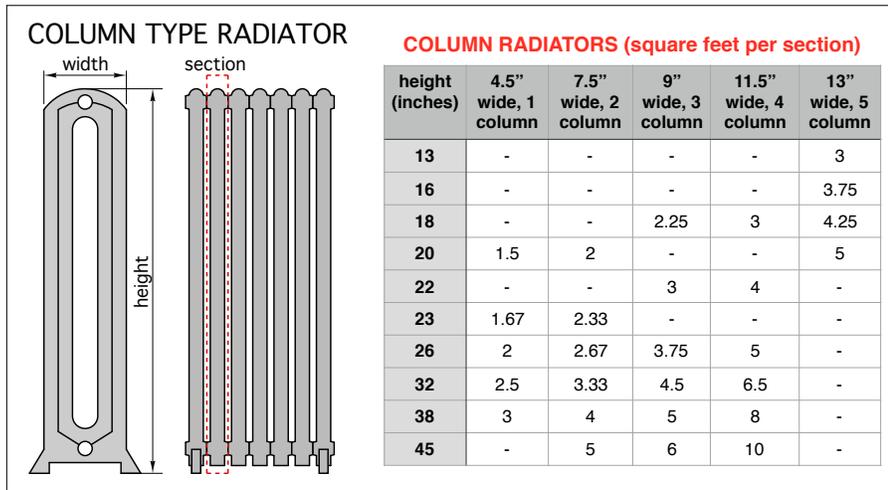
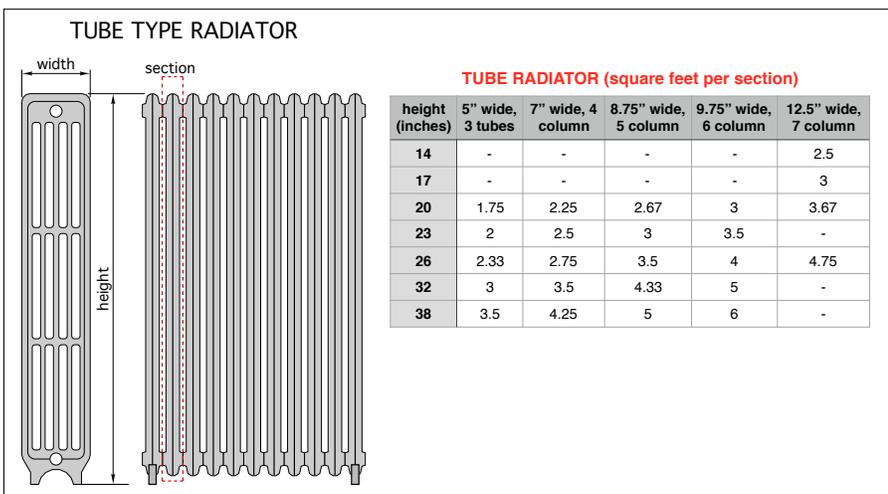


Figure 3-30



is estimated by multiplying the total square footage by the value of (q) determined from figure 3-31. (q) is the heat output rate per square foot of area for the radiator. It is based on the difference between the average water temperature in the radiator and the surrounding air temperature.

For example, if the previously discussed 10-section column radiator, having a total square footage of 65 ft², was operated in a room with an air temperature of 65°F, and the average water temperature in the radiator was 105°F, the quantity (T_w-T_{air}) on the horizontal axis in figure 3-17 would be (105-65) = 40°F. The value of (q) would be read from the vertical axis in figure 3-31 as 50 Btu/hr/ft². The total heat output of the radiator would be (65 ft²) x (50 Btu/hr/ft²) = 3,250 Btu/hr.

THE EFFECT OF FLOW RATE ON HEAT OUTPUT:

It is possible to slightly increase the heat output of any hydronic heat emitter by increasing the flow rate through it. Higher flow rates decrease the temperature drop (e.g., ΔT) across the heat emitter, and thus raise the average water temperature within the emitter.

For example, the square footage of each section of a column-type cast iron radiator that's 11.5 inches wide and 32 inches tall is 6.5 ft². If there are 10 sections in that radiator, its total square footage is 6.5 x 10 = 65 ft².

A similar procedure is used for finding the surface area of tube-type radiators. The data is given in figure 3-30.

Standard cast iron baseboard is 2.5 inches wide, and 10 inches tall. It has a surface area of 3.5 square feet per linear foot of length.

After the total square footage of a radiator is determined, its heat output

Figure 3-31

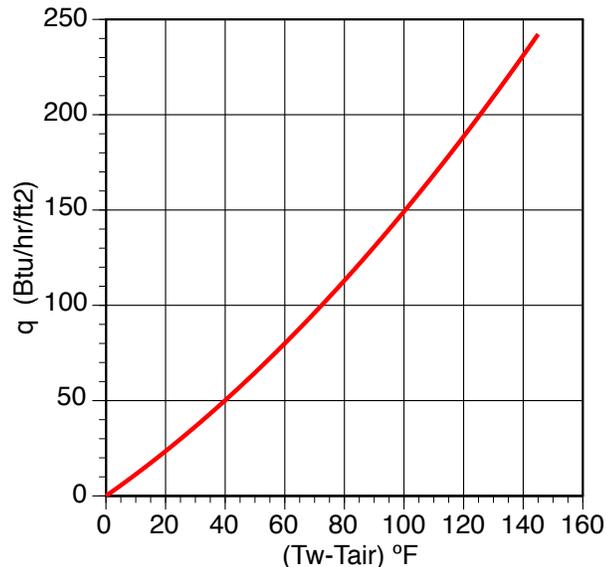
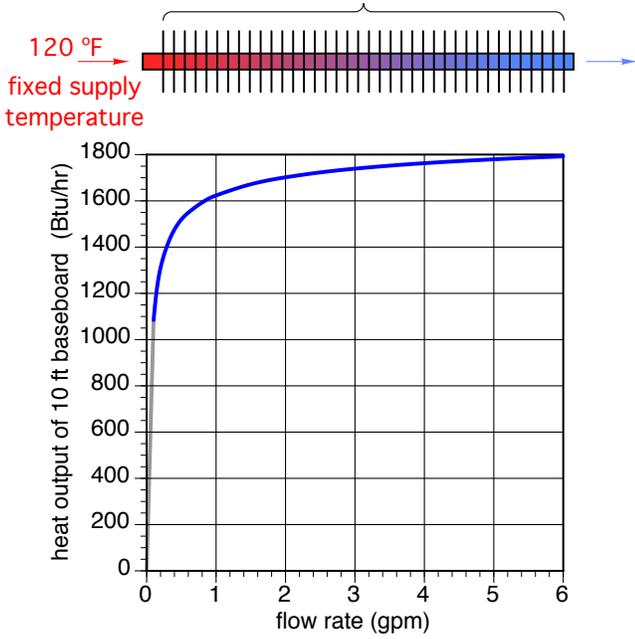


Figure 3-32

10 feet of finned-tube baseboard
rated at 600 Btu/hr/ft when
operating at 200°F water at 1gpm



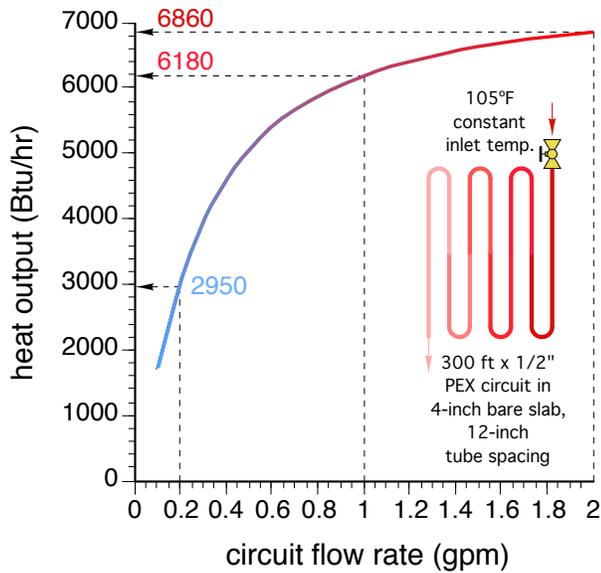
Note the rapid increase in heat output at flow rates below 1 gallon per minute, and very limited additional gains in heat output at flow rates over approximately 2 gallons per minute.

The designer must also consider that increasing flow rates create significantly higher head losses and associated higher circulator power input. As a guideline, flow rates over 2 gpm in residential fin-tube baseboard are difficult to justify.

This non-linear relationship between heat output and flow rate also holds true for other heat emitters. Figure 3-33 shows the relationship for a typical radiant floor panel circuit consisting of 300 feet of 1/2-inch PEX tubing in a 4-inch concrete slab.

Again, the increase in heat output is rapid at low flow rates, but marginal at flow rates over about 1 gallon per minute. Doubling the flow rate from 1 to 2 gallons per minute only increases heat output approximately 11 percent. In theory, the head loss through the circuit would increase by a factor of about 3.4 when operating at 2 gpm versus 1 gpm, and the power input to the circulator would increase by a factor of 6.7. Designers should carefully evaluate these compromises when evaluating intended operating flow rates for hydronic distribution systems.

Figure 3-33



The change in heat output with changes in flow rate are very “non-linear.” At very low flow rates, heat output increases rapidly with increasing flow. However, at flow rates above some nominal value, further gains in heat output are very limited. This effect is shown for fin-tube baseboard in figure 3-32.

PIPING MODIFICATIONS

Section 3 described methods for evaluating the thermal output of existing and supplemental heat emitters when operating at reduced water temperatures. This section focuses on ways to pipe supplemental heat emitters into existing distribution systems.

There are many possible approaches. Every existing system has nuances that must be considered. These include the cost of materials and labor, aesthetics, access to the existing piping, available wall space, and how the system will operate based on existing or newly created zones.

Many “legacy” residential hydronic heating systems in North America were constructed using fin-tube baseboards. The most common way these baseboards were connected is in series or “split-series” circuits. Figure 4-1 shows a series baseboard distribution system. Figure 4-2 shows a split-series baseboard system.

When several supplemental heat emitters are to be added to a system using series or split-series connected baseboards, they should not be simply cut into the series circuit. Doing so could substantially increase the flow resistance of the circuit(s), which will reduce flow rate, assuming the same circulator is used. Adding heat emitters in series also increases the temperature drop of the circuit. This reduces the heat output of heat emitters near the end of the circuit, especially when the supply temperature to the circuit is lowered.

Adding multiple heat emitters to a series or split-series circuit is best handled by making strategic cuts into the circuit where it can be reasonably accessed, and reconnecting the cut segments, along with new heat emitters, back into a parallel distribution system. These strategic cuts and reconnections might make each

room a separate branch circuit, which allows the possibility of separate zoning. These changes might also make a group of two or three rooms into a single zone. The choice depends on how well the existing piping can be accessed, as well as what the system’s owner wants to accomplish in the “makeover” of the original distribution system.

One of the easiest and most versatile ways to divide an existing series or split-series distribution system into multiple parallel branch circuits is by creating a homerun distribution system. Each heat emitter, or perhaps a grouping of two or three existing heat emitters or supplemental heat emitters, will be supplied by a separate branch circuit of PEX, PE-RT or PEX-AL-PEX tubing. This tubing is easy to route through cavities or along framing. It’s readily available in sizes from 3/8”

up to 1”, with 1/2” being the most commonly used size.

Each homerun circuit begins and ends at a manifold station. In some projects, a single manifold station is often sufficient. In larger buildings, or situations where the zoning will be spread across several building areas, multiple manifold stations can be used.

Figure 4-3 shows an example of how an existing series baseboard system was modified into a parallel system using this technique.

In this case, the existing series circuit was divided into four parallel branch circuits. Supplemental heat emitters were added to each of these circuits. Two of the circuits received high-output fin-tube baseboard (shown in orange). The other two received panel radiators. This illustrates the

Figure 4-1

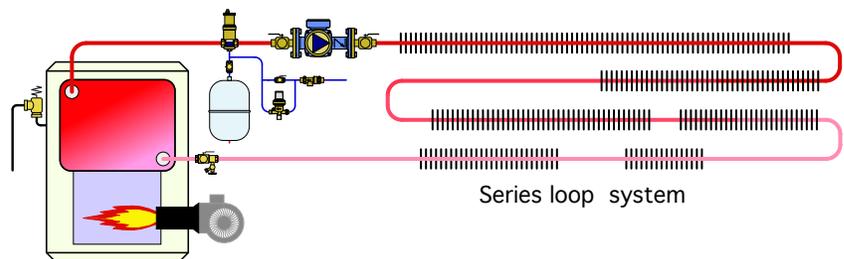
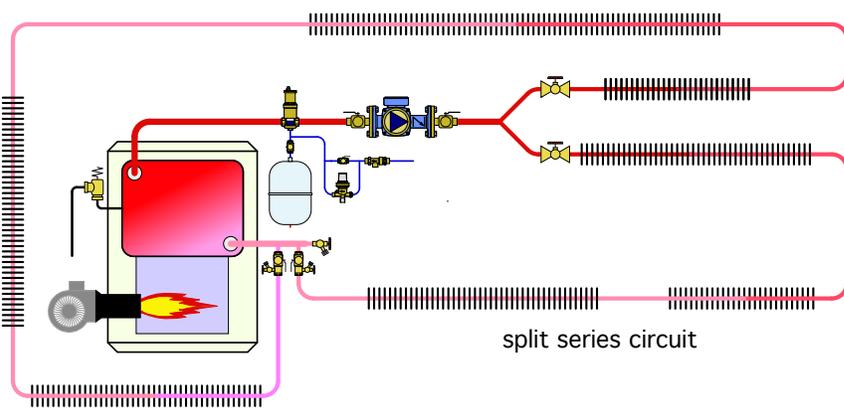


Figure 4-2



possibility of using multiple types of heat emitters, depending on available wall space, budget and aesthetic preferences.

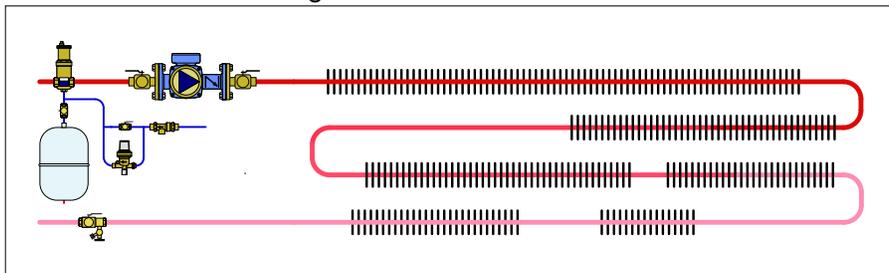
In one branch circuit, the supplemental baseboard was added upstream of the existing baseboard. In another branch circuit, the supplemental baseboard was added downstream of the existing baseboard. The choice depends on the available wall space and placement of the existing baseboard within each room.

The existing copper tube circuit was cut at accessible locations that preserved a reasonable amount of existing copper tubing, but also allowed convenient transition to 1/2" PEX, PE-RT or PEX-AL-PEX tubing.

Adapter fittings that transition from 1/2" or 3/4" copper to 1/2" PEX PE-RT or PEX-AL-PEX tubing are readily available. If the transition from copper tubing to one of these flexible tubes will be at the fin-tube element, it's convenient to use an "angle" adapter that provides

Figure 4-3

Existing series baseboard circuit



Modified distribution system

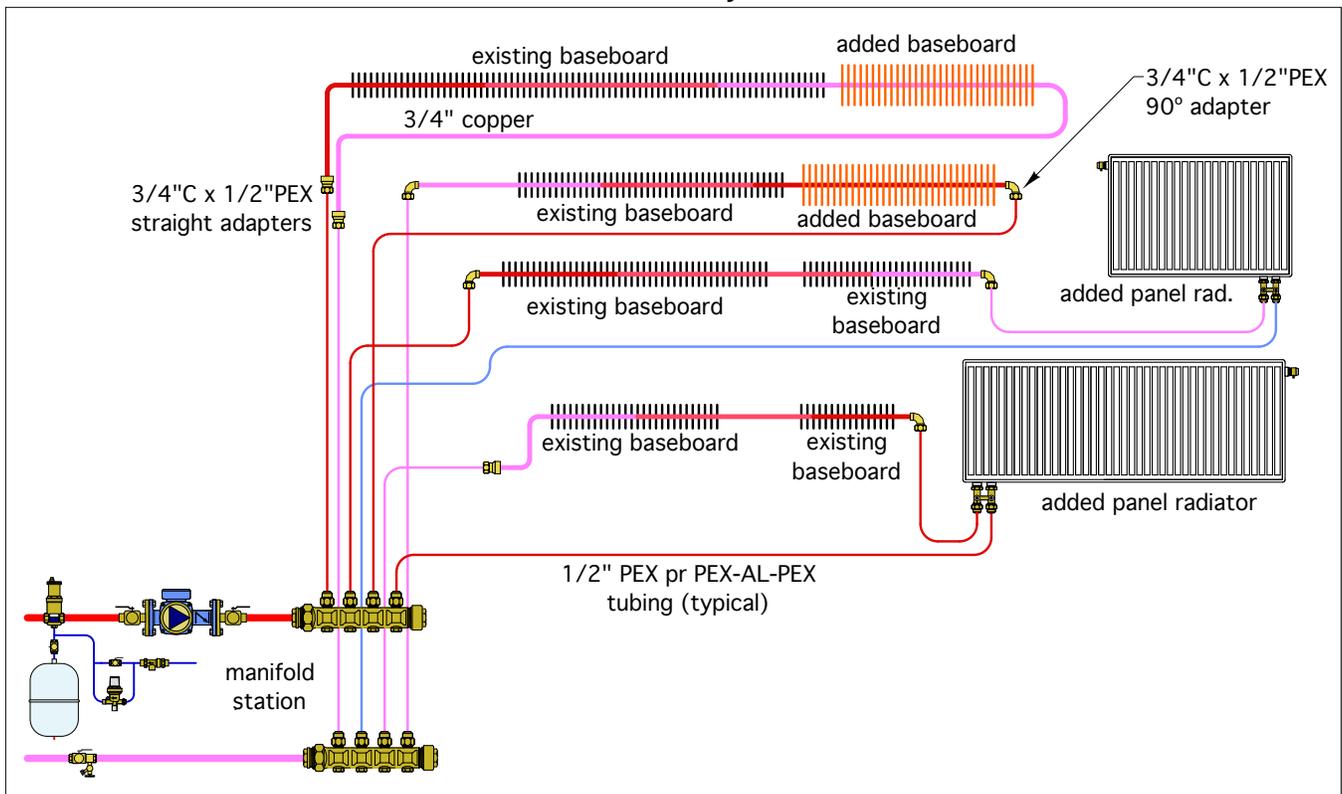


Figure 4-4a



Courtesy of Uponor

Figure 4-4b



Courtesy of MrPEX

the 90° turn from the flexible tubing that comes up through the floor, to the horizontal copper tube element. Figure 4-4a shows such a fitting. The small tapping at the top of this fitting is for mounting an air vent. This fitting is ideal for the outlet end of a copper fin-tube element. A similar transition fitting without the air vent tapping is also available.

PEX-to-copper adapters are also available for transitions along a common centerline, an example of which is shown in Figure 4-4b.

The 1/2" PEX, PE-RT or PEX-AL-PEX supply and return tubing connected to each heat emitter, or group of 2 to 3 emitters, runs back to a manifold station. If that manifold station is equipped with individual circuit-balancing valves, the flow through each branch circuit can be adjusted as necessary.

All four branch circuits in figure 4-3 operate simultaneously (e.g., they are not configured as individual zones). As such, this distribution system presents a constant flow resistance. Due to the parallel versus series configuration, that flow resistance is likely to be lower than that of the original series circuit. This should be verified by calculating the head loss or pressure drop of the path with the highest flow resistance using standard hydronic circuit analysis methods. If the head loss and total flow rate through the four circuits is comparable to the flow and head loss of the existing series circuit, the same circulator could be used. If the flow resistance of the modified system is significantly lower, consider replacing the existing circulator with an appropriately sized high-efficiency (ECM) circulator, set for constant speed operation. This could significantly lower the electrical power required to operate the distribution system.

One advantage of a parallel distribution system is that it supplies water at the same temperature to each branch circuit. This contrasts with a series distribution system in which each heat emitter receives a lower water temperature than the heat emitter upstream of it.

When a series circuit of heat emitters is converted to a parallel system, it's likely that heat emitters that were near the end of the series circuit will now operate at a higher average water temperature, which will slightly boost their heat output.

ZONING POSSIBILITIES:

Another advantage of parallel distribution systems is the ease of creating zones. If the existing series circuit is converted to multiple branches, each of those branches has the potential to operate as an independent zone. To do so, each

branch needs to be equipped with a valve that responds to a call for heat from the building space served by the branch circuit. Valve options include non-electric thermostatic radiator valves, electrically operated manifold valve actuators and electrically operated zone valves.

ZONING WITH NON-ELECTRIC THERMOSTATIC RADIATOR VALVES:

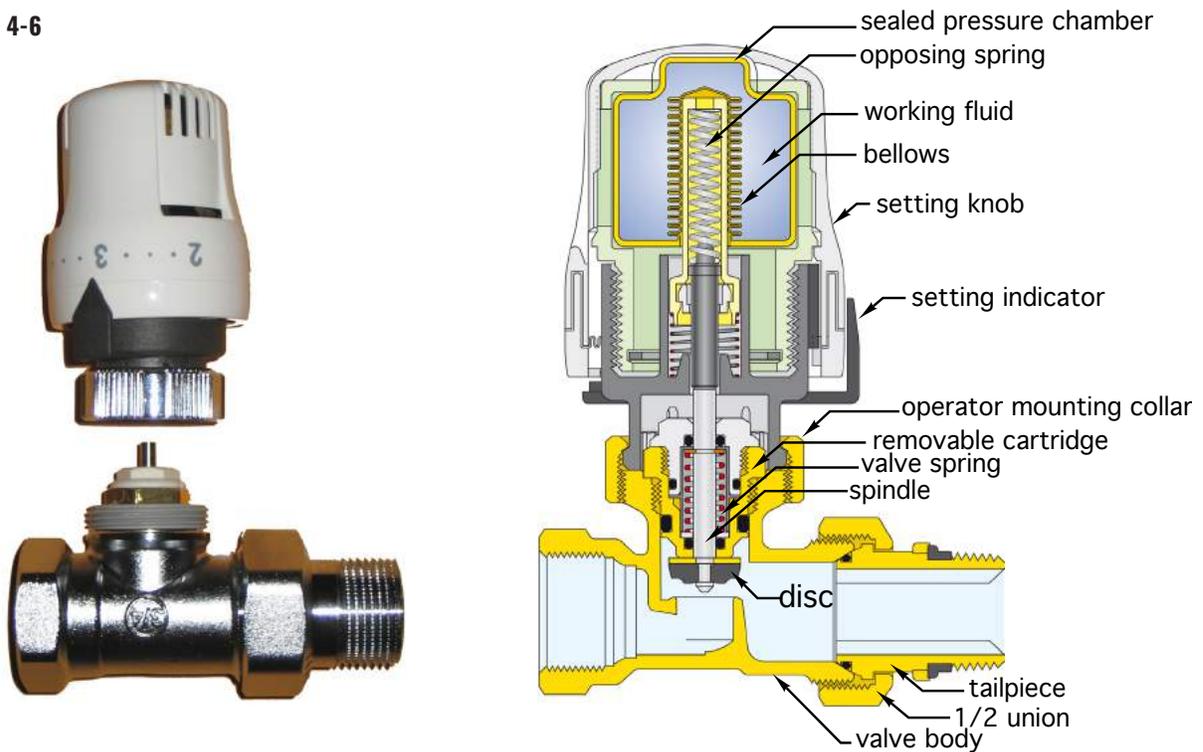
Non-electric thermostatic radiator valves combine a valve body with a thermostatic operator. Common valve bodies include "straight pattern" valves, where the inlet and outlet are on the same centerline, and "angle pattern" valves, where the outlet is rotated 90° relative to the inlet. Figure 4-5 shows examples of both valves.

These valves contain a disc that moves vertically relative to a seat (e.g., an opening within the valve through which all flow passes). The flow geometry within the valve body is similar to that of a globe valve. However, unlike a globe valve, the

Figure 4-5



Figure 4-6



shaft of the radiator valve doesn't rotate. It moves linearly and is spring-loaded to the normally open position. When the knob of the valve is turned clockwise, the spring-loaded shaft moves the disc closer to the seat and vice versa.

Although these valves can be used to manually regulate flow rate through a heat emitter, or group of heat emitters, their operation can be automated by adding a thermostatic operator, which simply screws onto the valve in place of the manual knob, as shown in figure 4-6.

A thermostatic operator contains a fluid sealed in a bellows chamber. As the air temperature surrounding the operator increases, the fluid expands, causing the bellows to push downward. This motion is passed to the valve's shaft, which moves the disc closer to the valve's seat, decreasing flow through the valve. When the air temperature around the operator decreases, the bellows contracts. This lifts the disc away from the seat and increases flow through the valve. These actions are fully modulating and continuously responsive to changes in room air temperature based on the number to which the knob is set. When the knob of the thermostatic operator is set to "5," flow through the valve is maximized. For any given supply water temperature, this setting maximizes heat output from the heat emitter. Flow through the

valve decreases when the knob is set for the lower numbers 4, 3, 2 and 1. Lower flow rates reduce heat output of one or more heat emitters in same the circuit path as the valve. The thermostatic operator knob can also be set to the "snowflake" position, where the flow is reduced to a "trickle" adequate to prevent freezing, but with very minimal heat output from the emitter.

Figure 4-7

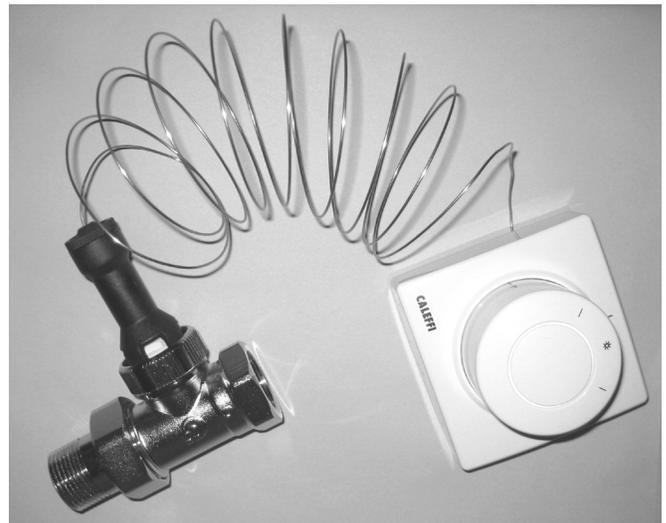
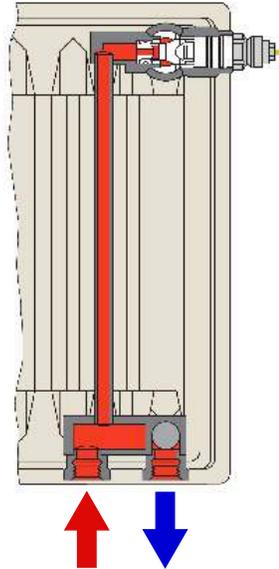
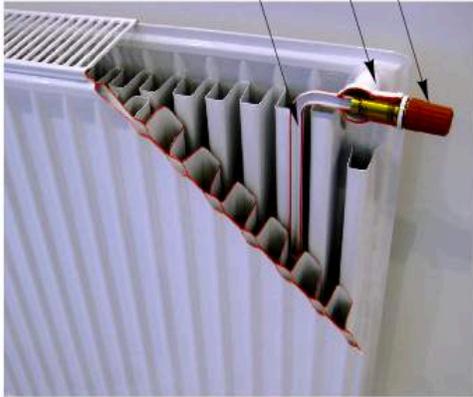


Figure 4-8

manual adjustment knob
integral valve
riser tube



This allows the setting dial to be mounted to a wall at the typical height for a room thermostat, while the valve and actuator remain at or below floor level.

Some panel radiators are supplied with integral valves, as shown in figure 4-8.

Water enters a connection at the bottom of the radiator, flows up through a riser tube and passes through this integral valve (assuming the valve is at least partially open). It then flows across the radiator's upper header, down through the vertical "flutes" and back across the lower header to the outlet connection at the bottom of the radiator.

This type of radiator is typically supplied with a simple plastic knob on the integral valve, as seen in figure 4-8. Although this knob can be manually adjusted to change the heat output of the radiator, such adjustments would need to be frequent to maintain stable room temperature in response to changes in room heat loss or internal heat gains.

Fortunately, this flow adjustment can be automated by replacing the manual knob with a thermostatic operator. The thermostatic operator simply screws onto the radiator valve in place of the manual knob, as shown in figure 4-9.

The thermostatic operator allows the valve to continuously fine-tune water flow so that heat output from the radiator is just what's needed to maintain the desired room temperature. By adding thermostatic radiator valves to the branch circuits created by dividing up an existing series circuit, each branch becomes an independent zone. Figure 4-10 shows this for the same piping configuration introduced with figure 4-3.

Figure 4-9

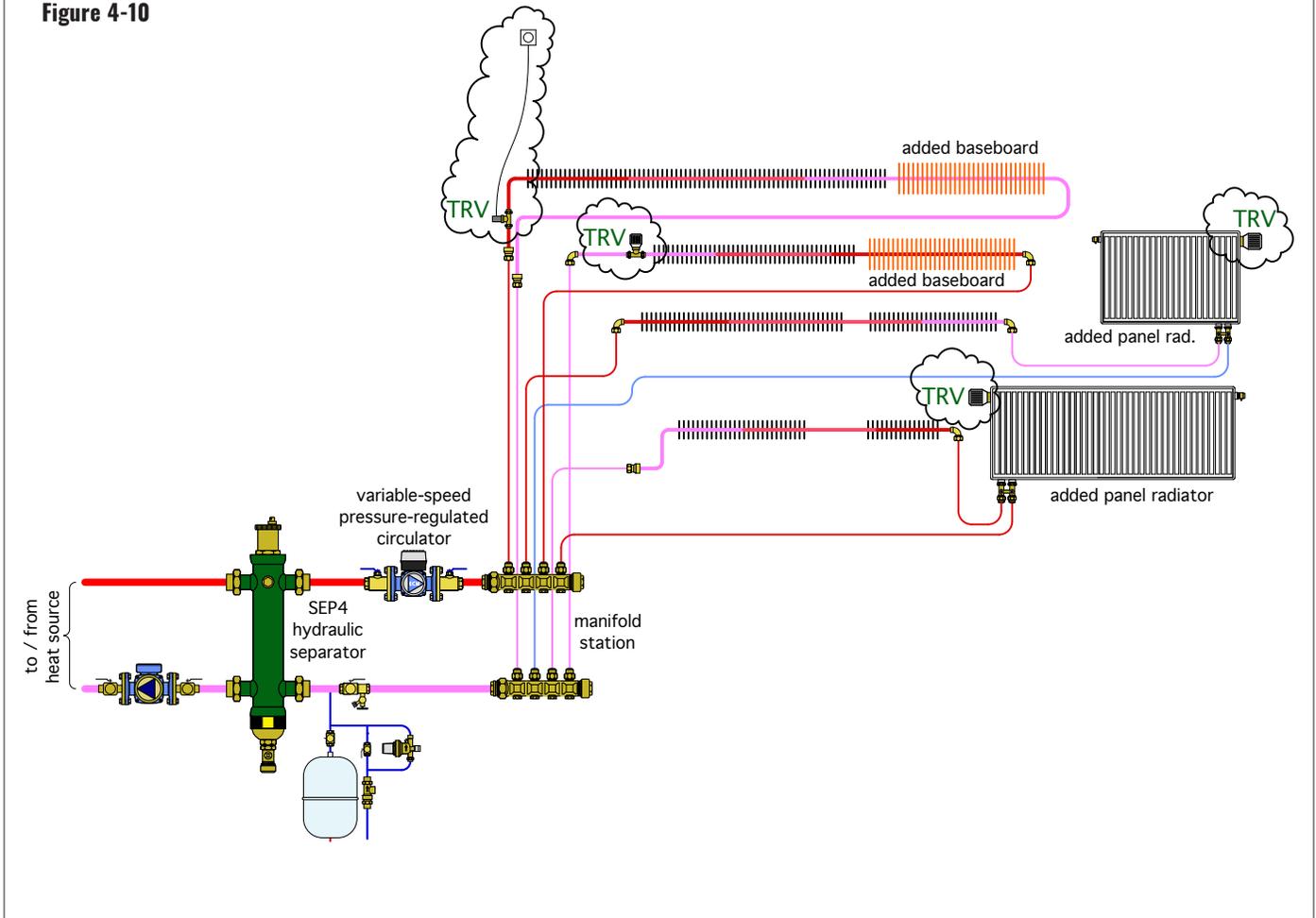


It's also possible to equip radiator valves with thermostatic operators that have remote setting knobs. The actuator screws onto the valve body and is connected to the setting dial

by a stainless steel capillary tube, as seen in figure 4-7.

The operator shown in figure 4-7 has a 2-meter-long capillary tube.

Figure 4-10



The upper branch circuit uses a thermostatic radiator valve installed below floor level and connected to a setting knob by a capillary tube. The next branch circuit down uses a standard straight pattern thermostatic valve connected into 3/4" copper just upstream from the first fin-tube baseboard. The lower two circuits have thermostatic operators added to the valves built into the panel radiators. Although these represent three different styles of thermostatic valves, they all have the same ability to modulate flow through their respective branch circuits in response to interior air temperature. Thus, what was previously a single zone series baseboard circuit is now a system with four independently controlled zones.

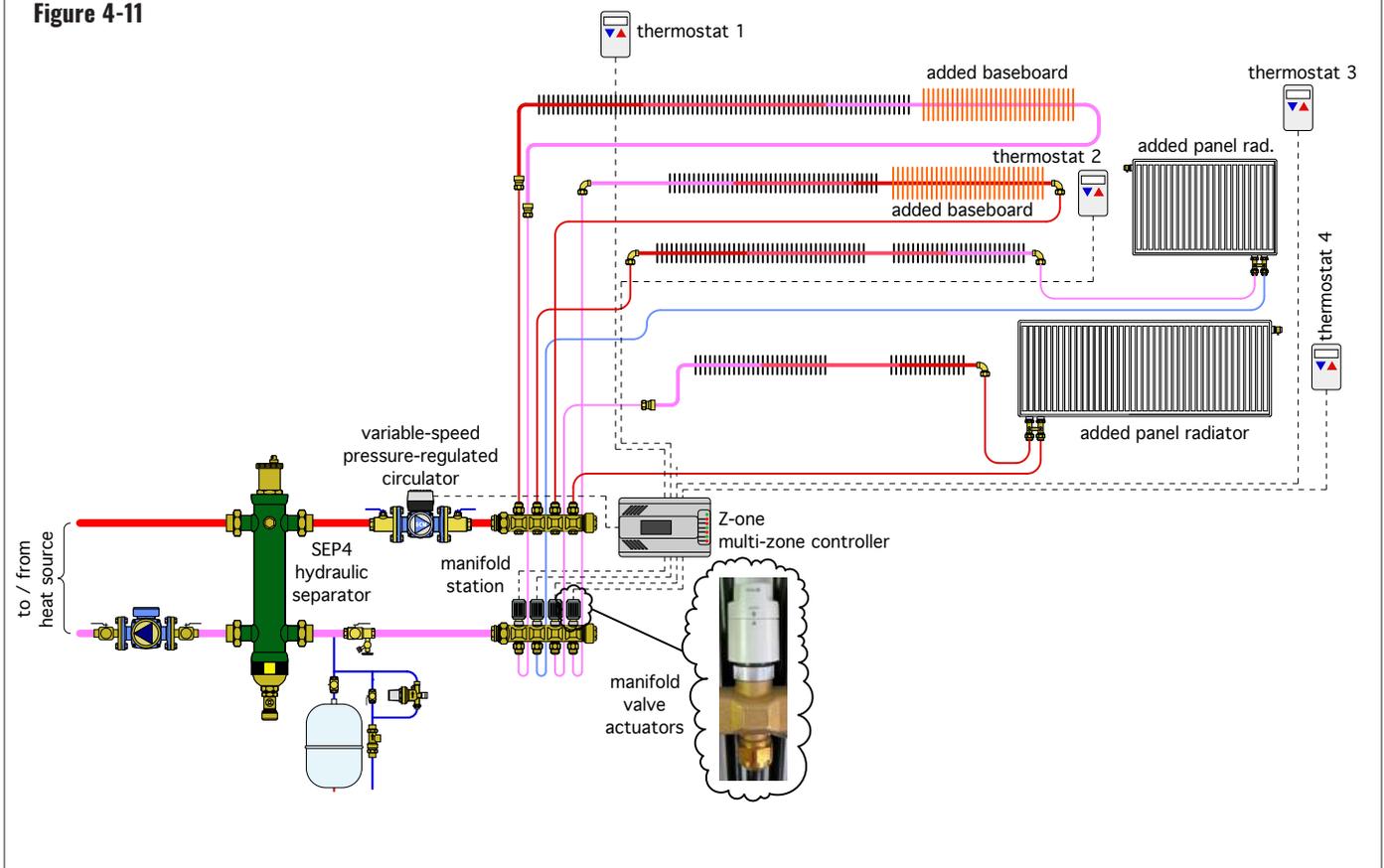
The original constant-speed circulator shown in figure 4-3 has been replaced by a variable-speed pressure-regulated circulator in figure 4-10. This circulator is set to maintain a constant differential pressure. As the four thermostatic radiator valves open, close or modulate, this circulator "feels" the resulting change in differential

pressure and automatically speeds up or slows down to counteract that change. This allows the flow rate through each branch to remain essentially constant regardless of flow changes in the other branches. Stable zone flow rates help ensure stable heat outputs.

The modified distribution system also includes a Caleffi SEP4 hydraulic separator, which isolates the pressure dynamics created by the variable-speed circulator from those of the heat source circulator, or any other circulators in the system. It also provides high-efficiency air separation, dirt separation and magnetic particle capture. The latter is recommended in any system using ECM (permanent magnet) circulators.

The original constant-speed circulator may or may not be useful in the modified system. It should be evaluated for possible use between the new heat source and the hydraulic separator.

Figure 4-11



ZONING WITH MANIFOLD VALVE ACTUATORS:

Another way to zone the modified 4-branch distribution system is by using manifold valve actuators in combination with four thermostats, as shown in figure 4-11.

Manifold valve actuators are low-voltage-operated heat motors that screw onto the valves already built into the manifold, as shown in figure 4-12

Each manifold valve actuator is wired to a Caleffi Z-one multi-zone controller. Each of the four zone thermostats are also wired to this controller. When any thermostat calls for heat, 24 VAC is passed from the Z-one multi-zone controller to the manifold valve actuator for that zone. The actuator opens the manifold valve, turns on the variable-speed circulator, and enables the heat source to operate. This provides flow of heated water to the branch circuit for that zone.

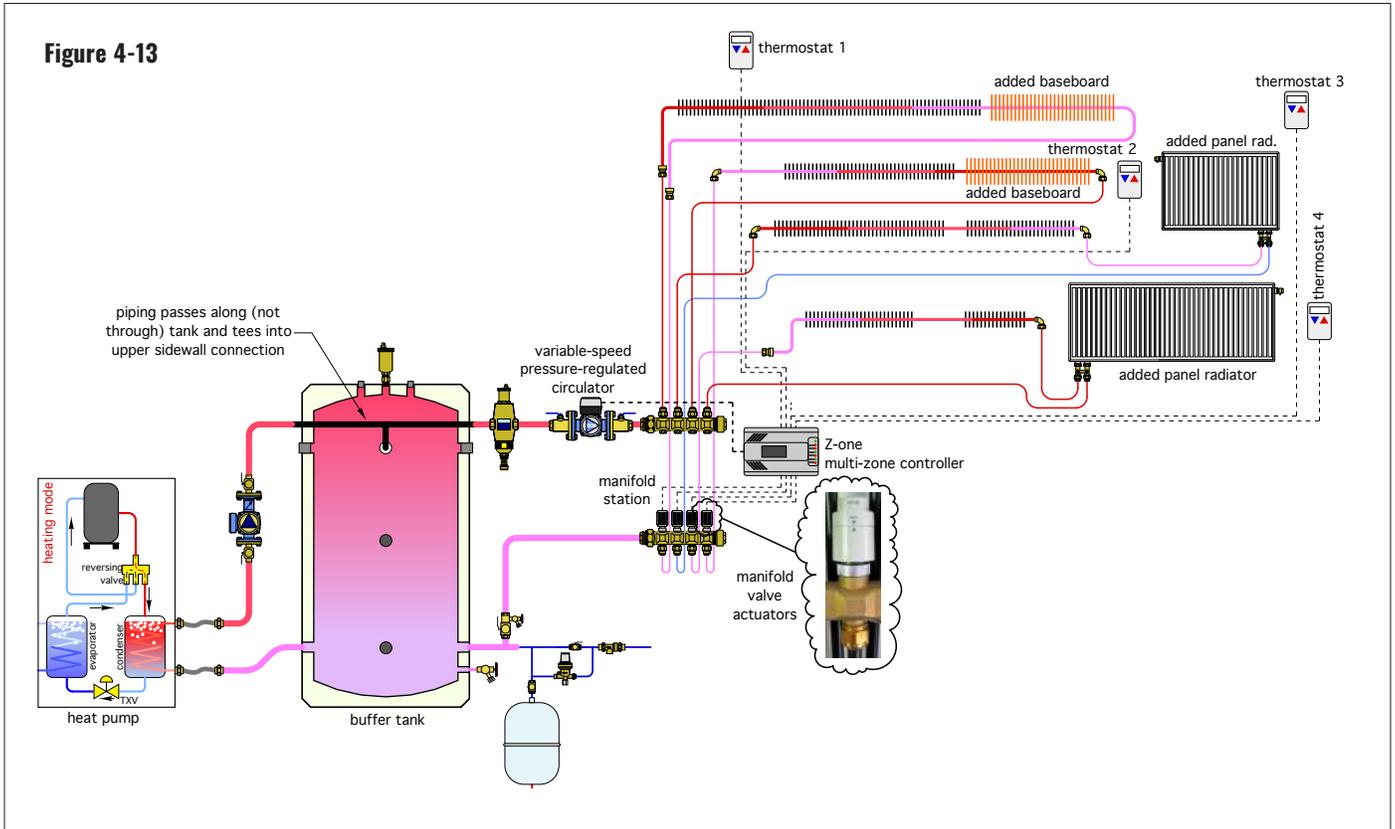
The variable-speed pressure-regulated circulator operates in constant differential pressure mode, the same as if thermostatic valves were being used.

Figure 4-12



If the low-temperature heat source to be used in the system is an on/off device (versus a modulating heat source), and the modified distribution system is highly zoned, the hydraulic separator should be replaced by a buffer tank, as shown in figure 4-13.

Figure 4-13



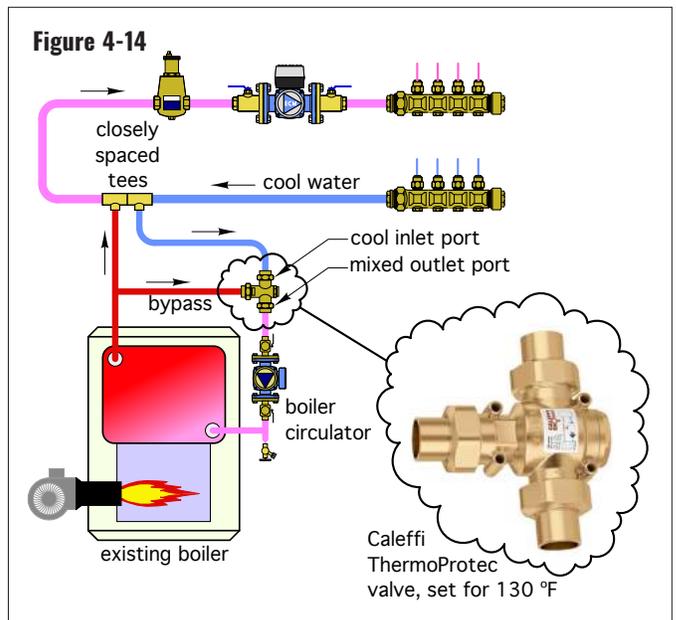
The buffer tank in Figure 4-13 is piped in a “3-pipe” configuration. The pipe from the outlet of the heat pump’s condenser is connected to the tank’s upper sidewall connection with a tee. This allows heated water to flow directly from the heat pump to any zone that’s operating simultaneously with the heat pump. The flow rate into or out of the tank’s upper sidewall connection is the difference between the flow rate from the heat pump and the flow rate to the load circuits. This reduces entering flow velocity and discourages mixing in the upper portion of the tank. It also helps ensure that the coolest water in the lower portion of the tank is returned to the heat pump, maximizing its COP. This piping also provides adequate hydraulic separation between the heat pump circulator and the variable-speed distribution circulator.

BOILER PROTECTION:

In many retrofit applications, the existing boiler is retained for use as a backup heat source, or a heat source that can help supply the building’s design heat load in situations where the new low-temperature heat source lacks sufficient capacity.

Since the modified distribution system will be operating at significantly lower water temperature, even under design load, it’s important to ensure that an existing conventional

Figure 4-14



boiler operating on fuel oil, natural gas or propane does not operate with sustained flue gas condensation.

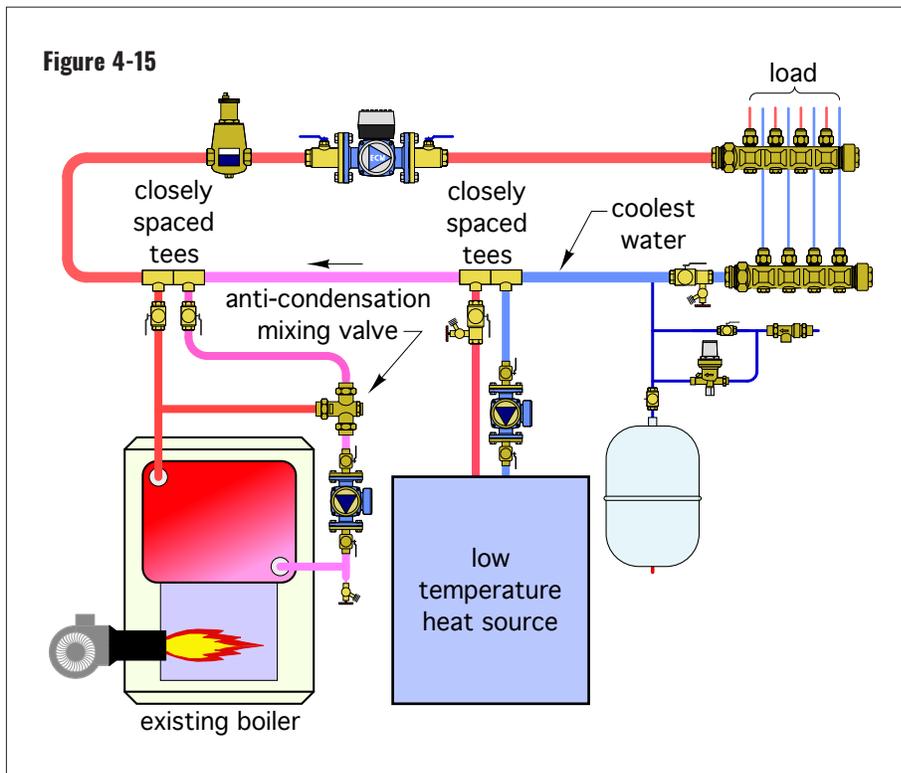
A common guideline is that flue gas condensation in boilers begins as the inlet water temperature drops below 130°F. A new distribution system that only requires 120°F

supply water temperature at design load, if directly connected to the boiler, would cause sustained flue gas condensation within that boiler.

Fortunately, it's easy to modify the near boiler piping to ensure that sustained flue gas condensation does not occur. One of the simplest approaches is to install a high-flow-capacity "anti-condensation" mixing valve, such as a Caleffi ThermoProtec valve, near the boiler, as shown in figure 4-14.

When the existing boiler is first fired, the cool inlet port on the ThermoProtec valve is completely closed. This blocks all flow to the closely spaced tees in the distribution system. All water leaving the boiler flows through the bypass, into the ThermoProtec valve, and back into the boiler. All the heat being produced is used to warm the boiler and its water content above the dewpoint of the flue gases as quickly as possible. No heat is flowing to the load.

As the boiler inlet temperature starts to climb above 130°F, the cool port on the ThermoProtec valve begins to open. Some flow of cool water returning from the distribution system flows into this port, as an equal amount of hot water flows up to the left side tee in the closely spaced pair. As the outlet temperature at the ThermoProtec valve continues to climb, greater flow rates of hot water are allowed into the distribution system at the closely spaced tees. However, this flow is always constrained so that the boiler inlet temperature remains at or above 130°F. If the outlet temperature of the ThermoProtec valve reaches 148°F, its bypass port is fully closed and the cool water return port is fully open. These actions occur in reverse as the boiler cools down after its burner cools down. This automatic valve



modulation allows a conventional boiler that was never intended to be used in a low-temperature system to operate at conditions that prevent sustained flue gas condensation, but still supply its full rated heat output to the low-temperature heat emitters. The high limit controller on the existing boiler should be set high enough to keep the burner operating well above the 130 °F minimum inlet temperature.

ThermoProtec valves can be ordered with other actuating temperatures for specialized applications such as biomass boilers.

HEAT SOURCE CONFIGURATION:

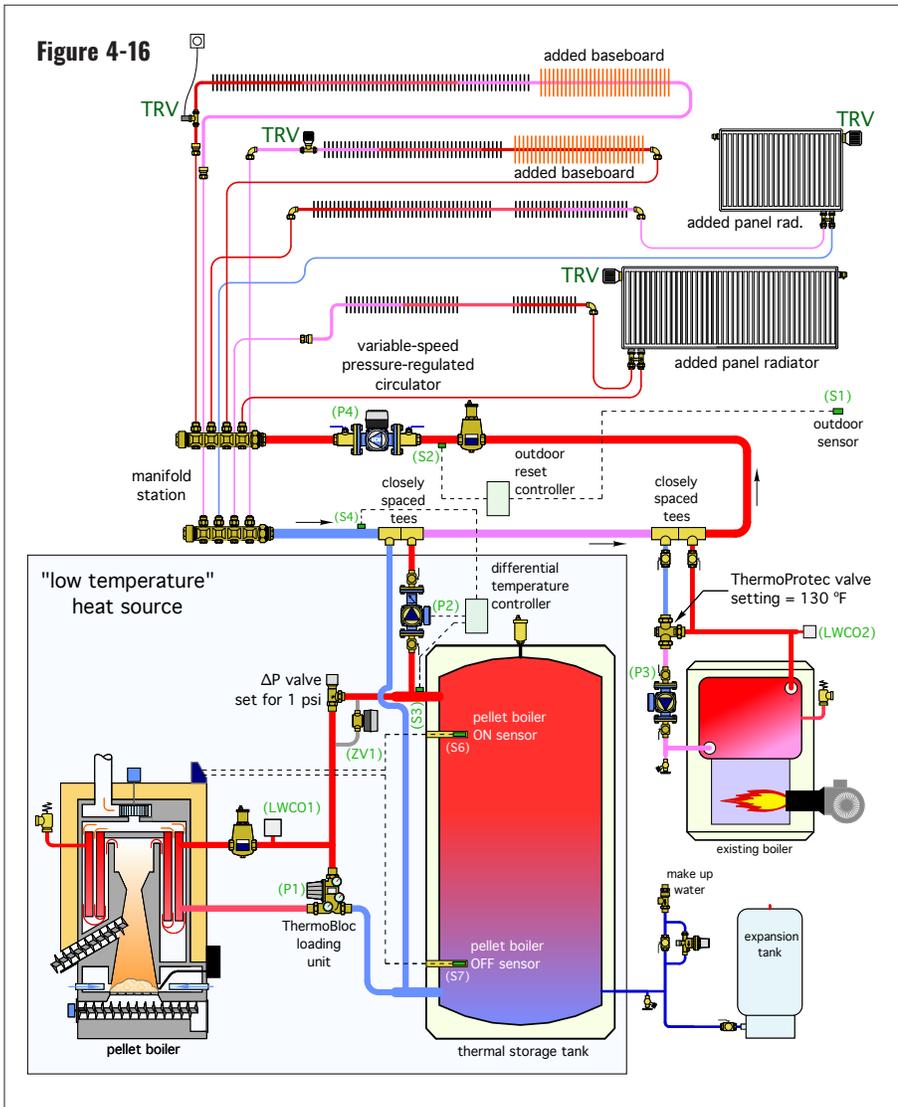
In applications where a new low-temperature hydronic heat source will be retrofitted into the system, and the existing conventional boiler will be retained as a backup, it's important to install the low-temperature heat source upstream of the existing boiler, as shown in figure 4-15.

This piping allows the coolest water in the system (e.g., the flow returning from the load), to be available to the low-temperature heat source. The lower the entering water temperature, the higher the thermal efficiency of that heat source.

If the existing boiler and its circulator are off, flow in the distribution system passes straight through the closely spaced tees. No flow passes through the auxiliary boiler. Likewise, if the low-temperature heat source and its circulator are off, it's hydraulically separated from the distribution system by the other pair of closely spaced tees. This piping also allows both heat sources to operate simultaneously if and when necessary. It also prevents heated water from passing through an inactive heat source, preventing that heat source from inadvertently dissipating heat to its surroundings.

Figure 4-16 shows another variation on this same concept for a system

Figure 4-16



that combines a pellet boiler, existing boiler and modified low-temperature distribution system.

The pellet-fired boiler and its associated thermal storage tank are considered the “low-temperature” heat source. The words “low temperature” may seem confusing, since most pellet boilers can produce water up to about 195°F. However, recall from section 1 that it’s advantageous for a thermal storage tank to have a wide operating temperature range. Since the upper temperature is typically limited by the boiler, the operating temperature range is dependent on the ability of heat emitters to

keep the building comfortable at the lowest practical water temperature. Wide temperature swings reduce the number of pellet boiler firing cycles, which improves efficiency and reduces emissions. This preferred “wide ΔT” operating mode has been proven in many systems that combine a pellet boiler system with low-temperature heat emitters.

As the low-temperature heat source, the pellet boiler and its thermal storage tank are connected as the “upstream” heat source, using a pair of closely spaced tees to provide hydraulic separation between circulators (P2) and (P4).

The existing boiler is connected as the “downstream” heat source, again using a pair of closely spaced tees to provide hydraulic separation between circulators (P3) and (P4).

Flow through the pellet boiler is created by the Caleffi ThermoBlock loading unit. It combines the functions of a circulator and thermostatic anti-condensation mixing valve, providing flow through the boiler and also protecting it from operating with sustained flue gas condensation.

A Caleffi differential pressure bypass valve set for 1 psi opening pressure prevents flow through the pellet boiler when it is off and circulator (P2) is on. This valve also prevents reverse thermosiphoning from the thermal storage tank through the pellet boiler when the loading unit is off.

A normally-open zone valve (ZV1) remains closed whenever utility power is available to the system. It opens during a power outage to allow forward thermosiphon flow between the pellet boiler and thermal storage tank. This flow safely dissipates the residual heat created by pellets that were burning inside the boiler when the power outage began.

Circulator (P2) injects hot water into the distribution system from either the pellet boiler (if it’s operating) or thermal storage tank (if the pellet boiler is off), or both, depending on flow rates.

Circulator (P3) injects heat from the existing boiler into the distribution system.

With proper controls, it is possible to operate circulators (P2) and (P3) simultaneously. However, during such operation, it is important that heat produced by the existing boiler doesn’t inadvertently end up in the thermal storage tank. This can be

prevented by a simple differential temperature control that only allows circulator (P2) to operate if the water temperature at the upper header on the thermal storage tank at sensor (S3) is a few degrees F above the temperature on the return side of the distribution system at sensor (S4).

The distribution system is the same as shown in figure 4-10. It's the result of converting an existing series baseboard distribution system into a four-branch parallel piping system and adding supplemental heat emitters. The variable-speed pressure-regulated circulator (P4) automatically changes speed to maintain a constant differential pressure on the distribution system as the four thermostatic valves open, close and modulate.

DESIGN GUIDELINES:

The piping modifications shown for converting a series or split-series baseboard distribution system into parallel branch circuits are just a few of many possibilities. Each conversion situation must consider the exact layout of the existing heat emitters and the practicality of modifying the system into parallel branches. The following guidelines should be used.

1. Determine the maximum water temperature that the new low-temperature heat source can produce at design load conditions. A suggested maximum supply water temperature at design load is 120°F. Even lower temperatures are preferred if the supplemental heat emitters that will be used can provide adequate heat output, are economically feasible and can fit in the available spaces.
2. Always determine the type and size of the supplemental heat emitter(s) to be used in each room and where it (they) will be located before modifying the piping. Follow the procedures given in section 3 for sizing.
3. In general, panel radiators and fan-coils can provide a given rate of heat output while occupying far less wall space compared to fin-tube baseboard.
4. Radiant floor, wall and ceiling panels are excellent low-temperature emitters, provided that they can be reasonably retrofitted into existing spaces.
5. From the standpoint of cost and installation time, it's best to use as much of the existing piping and heat emitters as possible.
6. Always consider the benefit versus cost of creating new zones when modifying the existing system. For example, if two bedrooms are typically maintained at

the same temperature, and the existing system has accomplished this, it's likely best to keep these two bedrooms together on the same zone after adding the necessary supplemental heat emitters. However, if the piping modifications to do so are comparable in cost/time to those required for creating two independent zones, then the latter is arguably a better choice.

7. Once all the supplemental heat emitters have been selected, and the proposed modifications to the distribution system have been sketched, always run a flow and head loss analysis for the modified system. This is used to confirm sufficient flow to each branch and to determine a suitable circulator for the modified system.

8. If the existing conventional boiler will be retained for use as a backup heat source for the low-temperature system, be sure the boiler is protected against sustained flue gas condensation by installing a thermostatic mixing valve near the boiler as described earlier in this section.

EXAMPLE SYSTEMS

This section brings the concepts and details discussed in earlier sections together to form complete systems. In each case, an original high-temperature system will be modified to make it suitable for lower water temperature operation, allowing the system's new heat source to operate under favorable conditions.

System #1:

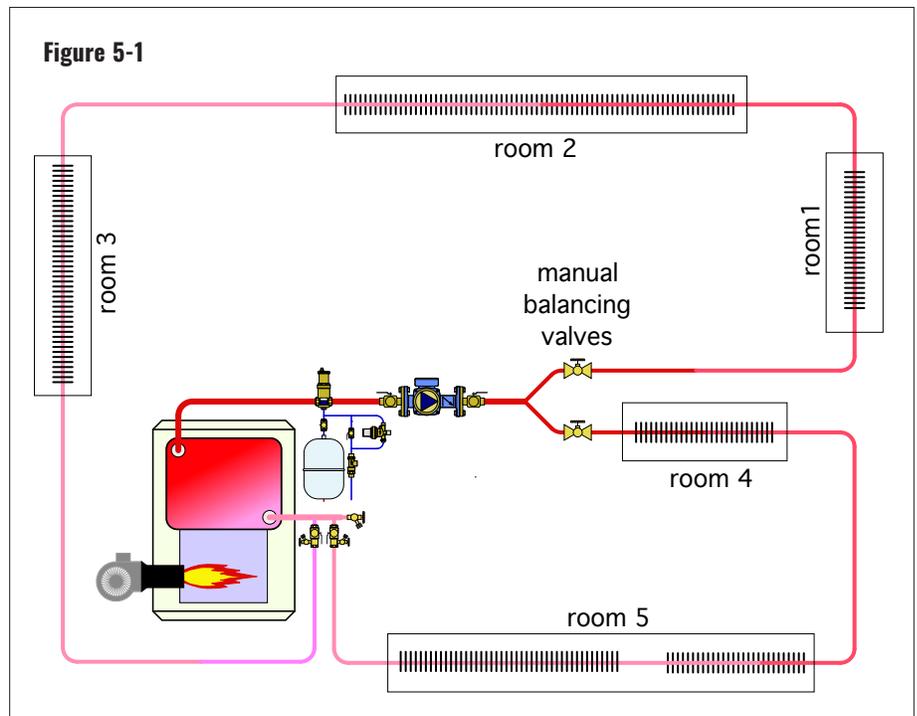
Figure 5-1 shows a “classic” split-series loop using fin-tube baseboard and an oil-fired boiler.

The supply water temperature at design load is 160°F. The original boiler is nearing the end of its service life, and the owner wants to change from fuel oil to natural gas and install a high-efficiency modulating/condensing boiler. To achieve very high seasonal efficiency, a decision is made to reduce the supply water temperature to 120°F under design load conditions.

As part of the home's energy upgrade, the owner hires an energy services company to perform a blower door test and fix any accessible air leaks. The owner also increases the attic insulation from R-19 to R-40 $^{\circ}\text{F}\cdot\text{hr}\cdot\text{ft}^2/\text{Btu}$. The combined effect of these envelope improvements is evaluated using formula 3-1. The results indicate that the supply water temperature at design load can be reduced from 160 to 145°F.

Further reductions in supply water temperature will require modification of the existing distribution system.

The owner and heat contractor have also discussed converting the system from a single zone to four independently controlled zones. They examine the existing copper tube distribution system and determine where it could be accessed to create the new zones. After performing an accurate



heating load estimate on each room, including the effects of the building envelope improvements, the heating contractor calculates how much “high-output” supplemental baseboard will be required in each room to bring the supply water temperature at design load down to 120°F. The contractor then measures available wall space in each room to be sure it's adequate for the added baseboard.

Rooms 1 and 2 will be treated as a single zone. The necessary supplemental baseboard will be supplied by using 1/2" PEX tubing and adapter fittings to transition to existing 3/4" copper tubing. Rooms 3, 4 and 5 will be treated as new independently controlled zones.

After draining the system, the heating contractor installs the supplemental baseboard and connects the piping segments that represent the four new zones back to a manifold station in the basement. The PEX tubing is routed along floor joists and held in place with CPVC “C-clips.” The

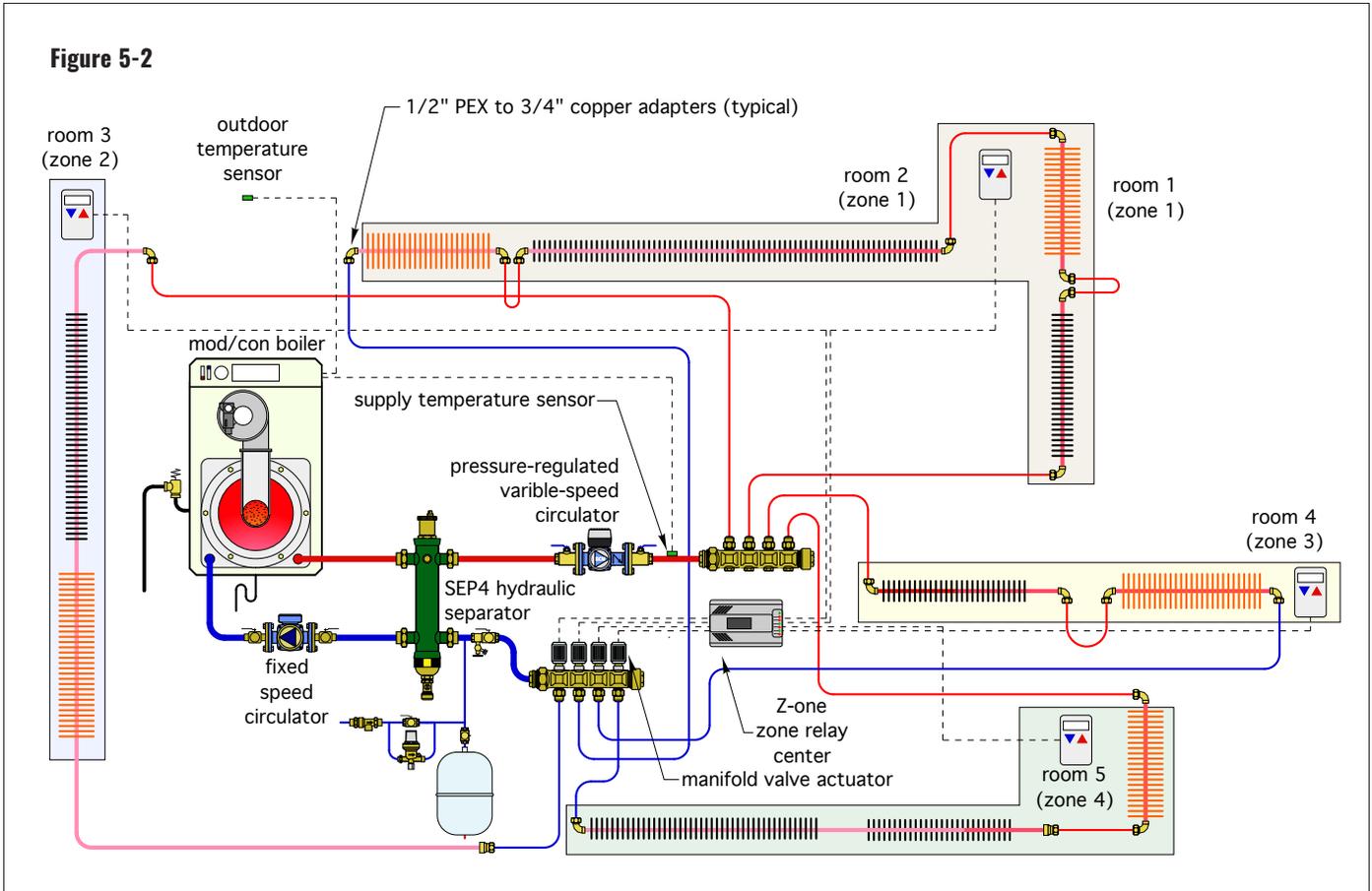
return manifold is equipped with four manifold valve actuators. These are wired to a Caleffi ZVR104 zone relay panel, as are the four new zone thermostats.

The piping changes are shown in figure 5-2.

The modified piping makes use of as much of the original copper tubing as possible while also taking advantage of the flexibility and installation speed of PEX tubing where necessary. One example is where piping needs to be extended from the original baseboard location to the location of the new high-output baseboard.

Because of the new zoning, the original fixed speed circulator is replaced by a variable-speed pressure-regulated circulator that automatically increases speed as zones turn on and vice versa. This circulator is set for constant differential pressure mode. This helps stabilize flow through any active zone, regardless of which other zones are

Figure 5-2



operating. The original circulator was inspected and found to be in good condition. It is repurposed within the system as the boiler circulator. A Caleffi SEP4 hydraulic separator is installed between the boiler and the distribution system. It provides hydraulic separation of the circulators, as well as high-efficiency air and dirt separation. It also captures magnetic iron oxide particles and allows them to be flushed from the system.

The mod/con boiler monitors the supply water temperature to the system and the outdoor temperature. It varies its firing rate to keep the supply water temperature just hot enough to maintain comfort in the building. This outdoor reset control action allows the boiler to operate at the lowest possible water temperature and

maximum efficiency. The boiler has a turndown ratio of 10:1, allowing it to deliver very low rates of heat output when only one or two zones are operating. This high turndown capability eliminates the need for a buffer tank.

System #2:

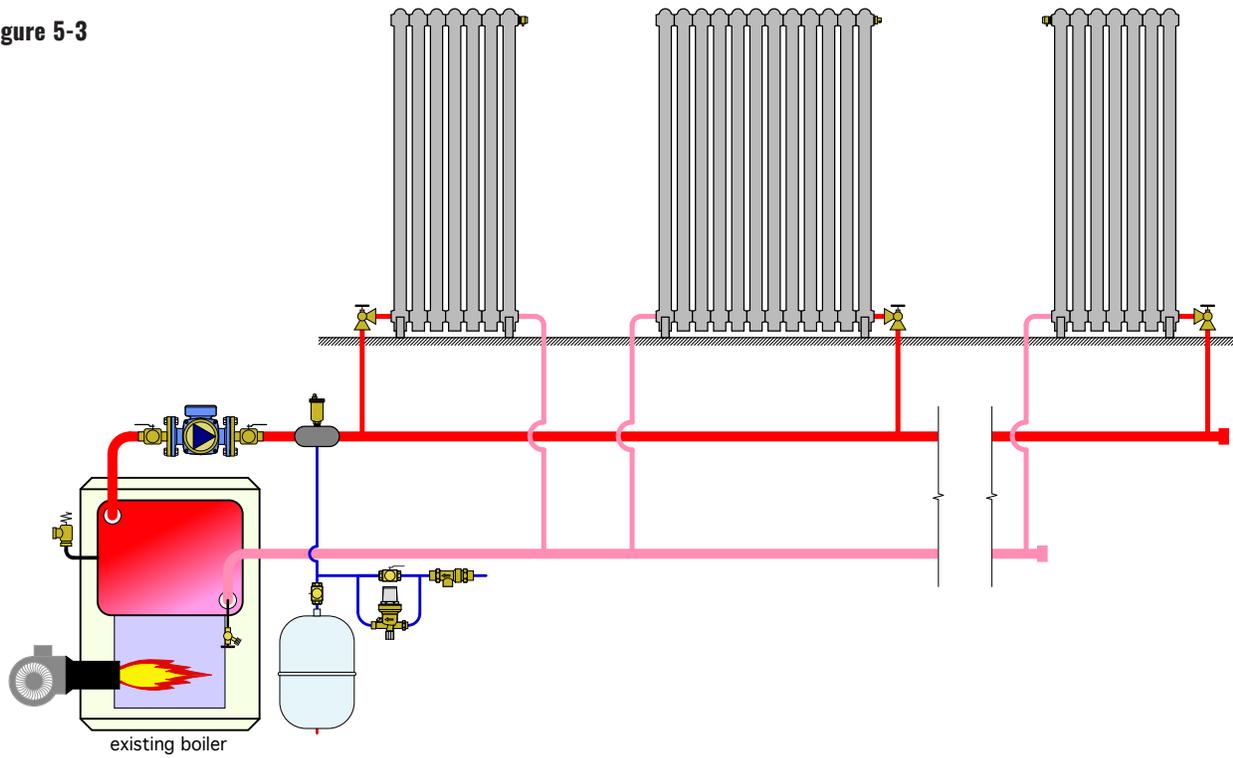
An older house has an existing “2-pipe” hot water distribution system consisting of several cast iron radiators connected with black iron pipe. Each radiator has a manually operated angle valve to regulate flow. The heat source is an oil-fired boiler. The supply water temperature required by the radiators at design load is 175°F. The existing piping is shown in figure 5-3.

The owner wants to install a geothermal heat pump, and retain

the oil-fired boiler as a backup heat source. To ensure that the heat pump will operate at good efficiency (e.g., high COP) the supply water temperature at design load needs to be lowered to 120°F. This will be done by adding panel radiators, piped in parallel with the existing cast iron radiators. The owner also wants to have separate thermostatic control of the temperature in each room.

The heating professional uses the methods from section 3 to size a supplemental panel radiator for each room. Once the panel sizes have been determined, the heating professional also measures to ensure that there is sufficient wall space for each panel radiator, and that the mounting locations allow access from beneath the floor.

Figure 5-3



To keep the panel radiators as small as possible, they need to operate at the same supply temperature as the existing cast iron radiators. This is possible by connecting the panel radiators in parallel with those existing radiators.

To provide separate zoning, each pairing of a cast iron radiator and its associated supplemental panel radiator needs to operate through a common zoning valve. After examining the piping in the basement, the heating professional determines that tees can be installed on the existing 3/4" iron pipe risers to each cast iron radiator. PEX tubing can be adapted to the side ports of these new tees to serve the new panel radiator. Zoning is accomplished by regulating the flow to each pair of radiators through a radiator valve installed on the 3/4" iron piping risers in the basement. A valve actuator with a capillary tube leading to a remote temperature control knob is installed on each radiator valve. The capillary tube and attached actuator are routed down through interior partitions and screwed onto the radiator valve. Figure 5-4 shows this new piping subassembly, which is repeated at each existing cast iron radiator.

Before the system piping is modified, the heating professional sets up valving, a 10-gallon plastic reservoir

Figure 5-4

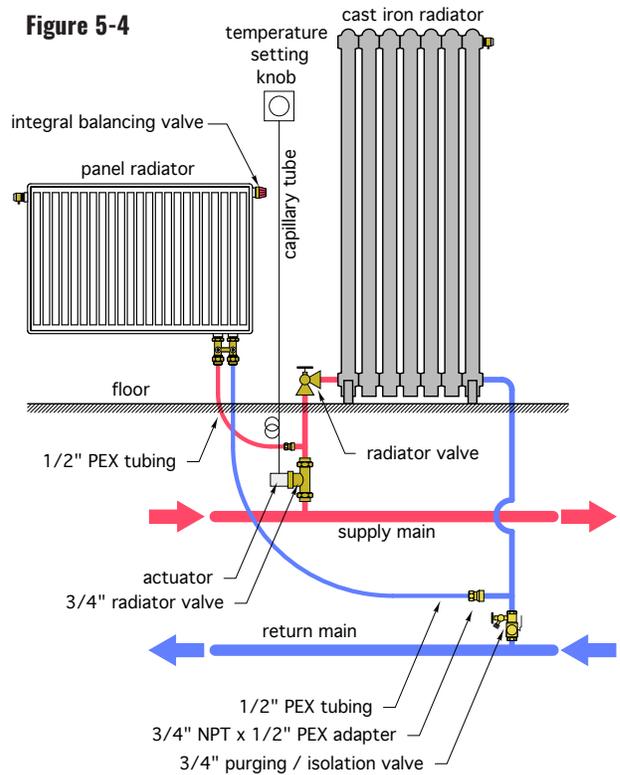
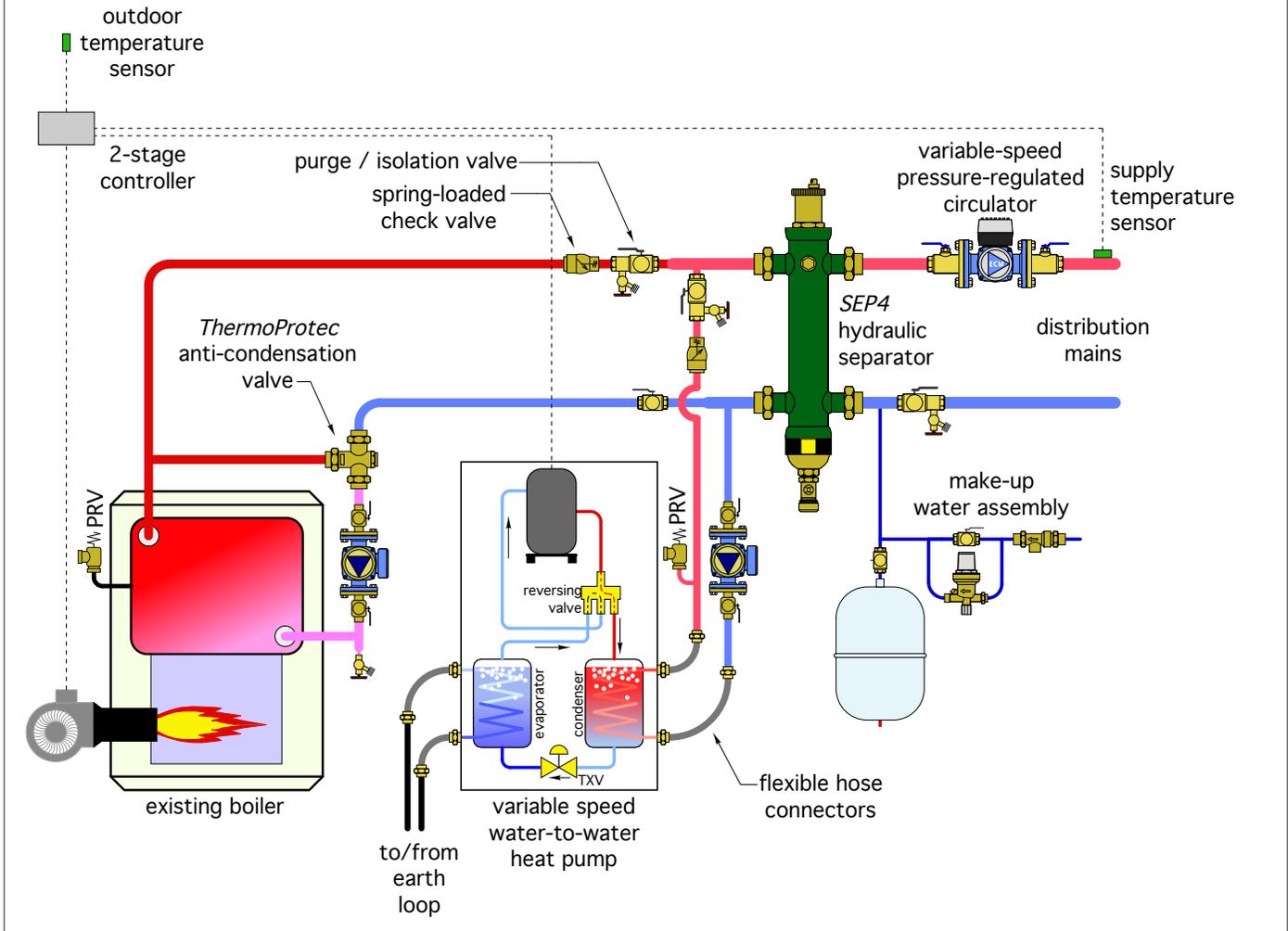


Figure 5-5



tank and a small sump pump to circulate a hydronic detergent through the system. This cleaning solution is circulated for several hours to thoroughly scour the inner surfaces of the iron piping and radiators. It is then drained from the system, and all piping and radiators are flushed with clean water.

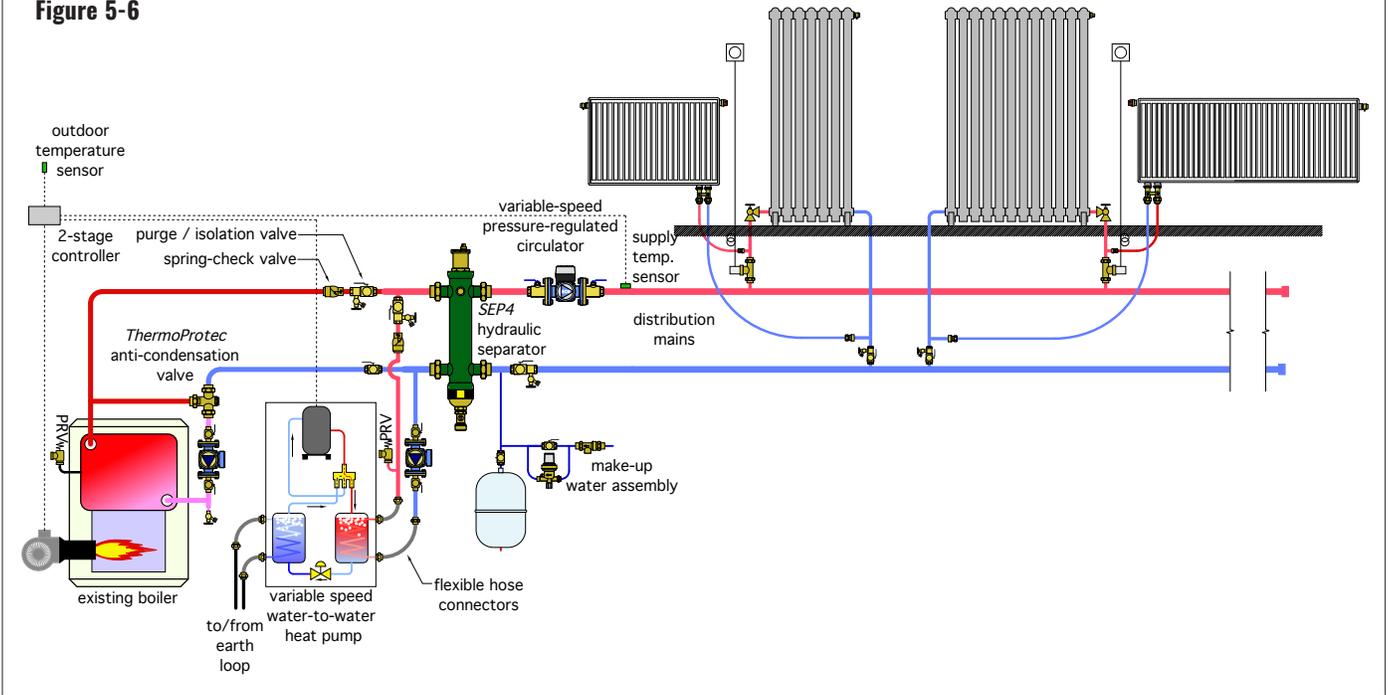
To ensure that any residual iron oxide particles will be removed, the heating professional installs a SEP4 hydraulic separator. The SEP4 also provides high-efficiency air and dirt separation, which were lacking on the original system. The SEP4 also isolates the pressure dynamics of the new variable-speed pressure-regulated distribution circulator from the circulators used with the heat pump and existing boiler.

The existing boiler and new variable-speed water-to-water heat pump are connected to the left side of the SEP4 hydraulic separator, as shown in figure 5-5.

Because it will now be part of a low-temperature system, the oil-fired boiler needs to have an anti-condensation mixing valve added to it to prevent operation with sustained flue gas condensation. A Caleffi ThermoProtec valve is used for this purpose. The existing boiler circulator can be reused, but must be located as shown in figure 5-5 to provide correct operation of the ThermoProtec valve.

The existing boiler and the heat pump are connected in parallel to a common set of headers leading to the SEP4 hydraulic separator. Each heat source is equipped with its own circulator that operates only when that heat source is running. The piping for each heat source includes a spring-loaded check valve to prevent flow reversal, and valves for purging and isolation. Because either heat source could be isolated from the balance of the system, each is equipped with a pressure-relief valve.

Figure 5-6



The variable-speed heat pump is operated as the first stage heat input for the system. The 2-stage controller provides a 0-10 VDC control signal to the heat pump to regulate its heat output in response to the supply water temperature. This controller also measures outdoor air temperature and lowers the required supply water temperature whenever possible based on the inferred heating load. Lower supply water temperatures increase the heat pump's COP. The variable output capability of the heat pump also eliminates the need to use a buffer tank.

Figure 5-6 shows the overall piping system and only three pairs of radiators with the modified piping. Additional pairs of radiators could be accommodated in the same manner.

The existing manually adjusted radiator valves on each cast iron radiator, along with the integral balancing valves on each panel radiator, allow flow through each radiator to be adjusted based on the location of the radiators along the distribution mains.

System #3:

An existing distribution system contains several fin-tube baseboards connected in a "1-pipe" manner using diverter tees. The heat source is a propane-fired cast iron boiler. The existing piping configuration is shown in figure 5-7.

The system operates as a single zone and currently requires a supply water temperature of 170°F at design load conditions.

Figure 5-7

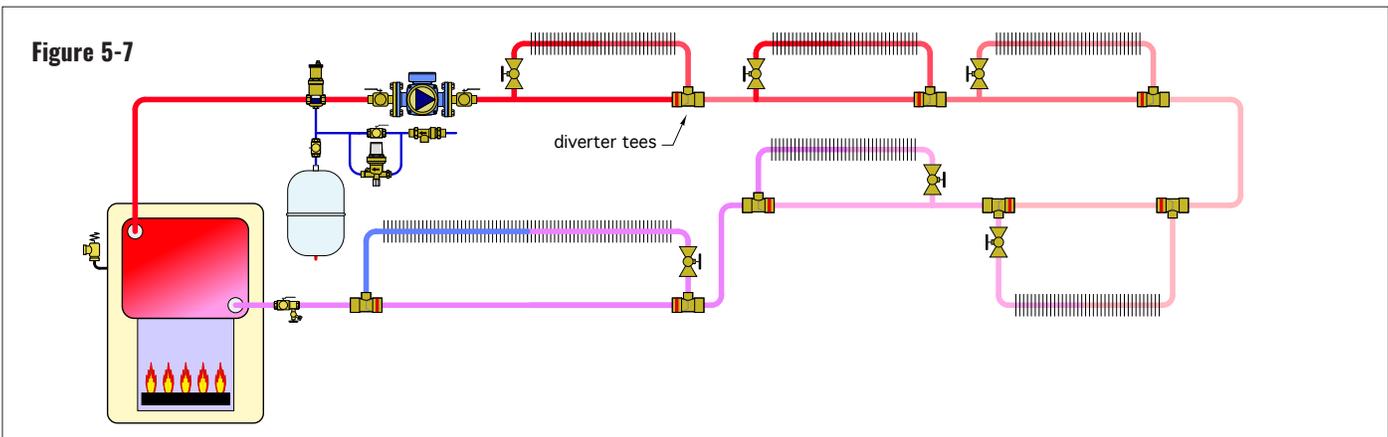
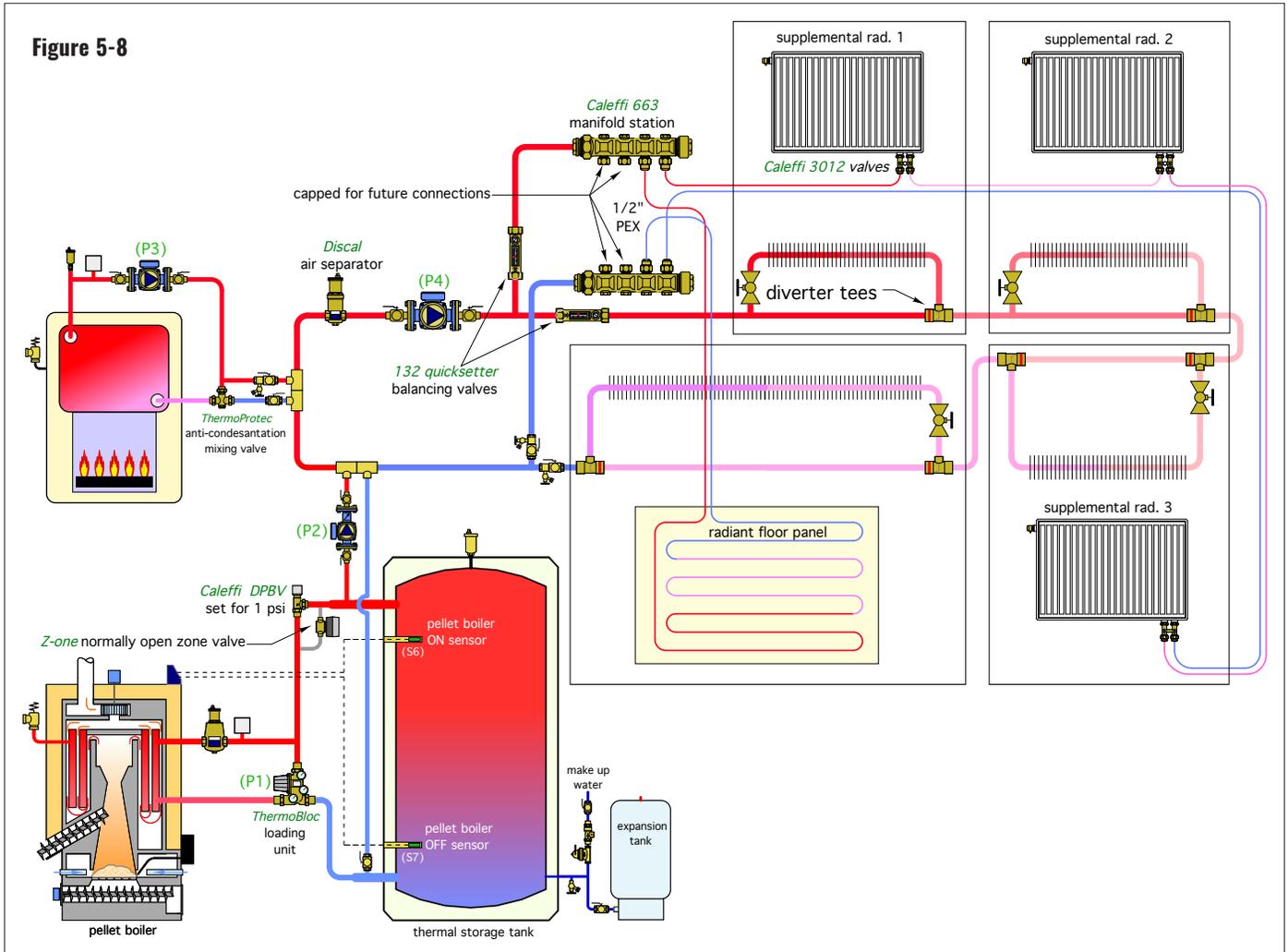


Figure 5-8



The owner plans to install a pellet-fired boiler as the primary heat source, and retain the propane boiler as a backup heat source.

To achieve maximum efficiency and minimal emissions, the pellet boiler needs to operate with on-cycles that average at about three hours per start. This necessitates a thermal storage tank sized to approximately 2 gallons per 1000 Btu/hr of pellet boiler heat output rating.

To extend the time between boiler firings, the temperature cycling range of the thermal storage tank should be as wide as possible. The upper temperature is limited to 180°F by the pellet boiler. The lower end of

the temperature cycling range is currently limited by the existing heat emitters to approximately 170°F to ensure design load output whenever necessary. The result is a very narrow temperature cycling range of only about 10°F. This greatly reduces the ability of the thermal storage tank to provide long boiler operating cycles and represents a significant “Achilles heel” in the overall system.

The temperature cycling range of the thermal storage tank can be significantly increased if the distribution system is modified so that it can maintain comfort at lower water temperatures under design load conditions. This can be done by adding supplemental heat emitters. Those emitters can

be selected and sized based on the methods described in section 3. In this application, the goal is to reduce the supply water temperature required at design load to 120°F.

The owner decides to use panel radiators as the supplemental heat emitters in three of the four spaces currently heated by the baseboards. In the fourth space, the owner wants to add floor heating to counteract what are currently very cold tile floors. To reduce the cost of piping modifications, the owner also wants to stay with a single zone system.

One piping approach that combines all these objectives is shown in figure 5-8.

The pellet boiler and its associated thermal storage tank can be thought of, collectively, as the “biomass heat source.” Heat can be supplied to the distribution system directly from the pellet boiler if it’s operating, from the thermal storage tank or from both at the same time. That heat is injected into the distribution system from a pair of closely spaced tees by circulator (P2). Those tees hydraulically separate circulator (P2) from the circulator for the existing boiler (P3) and the distribution circulator (P4).

The pellet boiler is protected against sustained flue gas condensation by a Caleffi ThermoBlock loading unit, which also creates circulation between the boiler and the thermal storage tank. A Caleffi 519 differential pressure bypass valve, set for 1 psi opening pressure, prevents forward or reverse heat migration between the pellet boiler and thermal storage tank. The normally open Z-one zone valve remains closed whenever utility power is available, but opens when a power outage occurs. This allows thermosiphon flow to carry any residual heat from the burning pellets from the boiler to the tank.

The existing boiler is also connected to the distribution system using a pair of closely spaced tees. However, because the modified distribution system will at times be running at low water temperatures, this boiler is retrofitted with a Caleffi ThermoProtec mixing valve to prevent sustained flue gas condensation.

With the exception of modification in the mechanical room, much of the existing distribution piping serving the baseboards remains the same. All baseboards continue to be served by the diverter tee system.

A four-port manifold station has been added in parallel to the existing distribution piping and has two circuits connected to it. One serves the new area of radiant floor heating. The other supplies a series of three panel radiators that are also connected in a “1-pipe” configuration using Caleffi 3012 dual isolation valves with adjustable bypass. These valves allow a portion of the heated water entering the valve’s left chamber to bypass to the right chamber without passing through the panel radiator. These valves allow the three panel radiators to be configured in what appears to be a series circuit, but still permit individual flow rate control through each radiator. This piping reduces the amount of interconnecting tubing relative to that required if each radiator was connected in a “homerun” circuit. However, unlike a homerun system, there is a drop in supply temperature as water passes through any panel radiator. That temperature drop should be accounted for when sizing each radiator in the “string.”

The remaining two ports of the manifold station are capped. This allows easy addition of other heating circuits if ever needed.

Caleffi 132 Quicksetter balancing valves are installed to allow flow proportioning between the existing distribution system and the manifold station.

The size of the system’s expansion tank was increased to accommodate the added volume of the thermal storage tank.

The modifications used in this system demonstrate several techniques, such as using multiple types of supplemental heat emitters, proper arrangement of the renewable heat source relative to the backup heat source, hydraulic separation of all circulators, the ability of either boiler to act as the system’s sole heat source, and protection of both boilers against sustained flue gas condensation.

SUMMARY:

As the North American market for high-efficiency boilers and renewable energy heat sources continues to expand, there is increasing need for hydronic delivery systems that complement those heat sources. In new construction situations, there are ample methods and materials for creating low-temperature hydronic heating distribution systems. However, tens of thousands of existing buildings have “high-temperature” distribution systems that severely limit the potential of these contemporary heat sources.

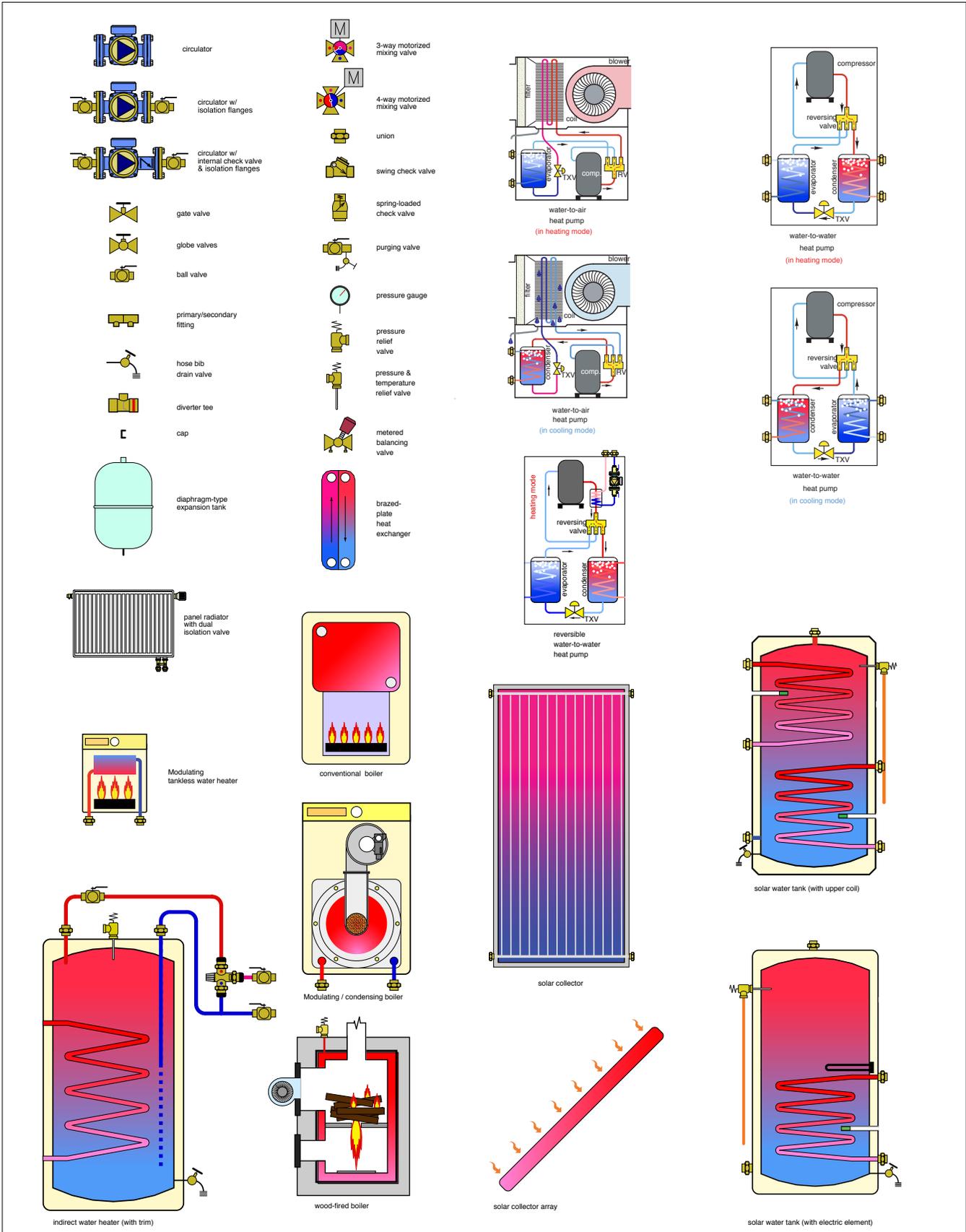
This issue of idronics has presented techniques to evaluate those existing systems for potential operation at significantly lower water temperatures. It has also shown practical methods of modifying them for low-temperature operation and expanded capabilities such as room-by-room zoning.

Every existing system has nuances that must be accounted for when planning these modifications. Likewise, every building owner will have preferences and budgets that must be worked into the proposed modifications.

Essential techniques such as outdoor reset control, boiler anti-condensation protection, homerun piping from manifold stations, pressure regulation in systems using valve-based zoning and hydraulic separation are indispensable tools in crafting hydronic solutions that meet all design objectives.

APPENDIX A: GENERIC PIPING SYMBOL LEGEND

GENERIC COMPONENTS



APPENDIX B: CALEFFI COMPONENT SYMBOL LEGEND

CALEFFI COMPONENTS

Float - type air vent

inline check valve

3-way thermostatic mixing valve

manifold station with balancing valves

DISCAL® air separators

QuickSetter™ balancing valve

geothermal manifold station

DIRTCAL® dirt separators

pressure reducing valve (3/4")

distribution station

DIRTMAG® dirt separators

Autofill™ backflow preventer comb

HYDROFILL™

backflow preventer

pressure-reducing valve

HYDROFILL™

pressure-reducing valve (3/4")

boiler tm kit

CONTECA™ heat calculator

DISCALDIRTMAG™ air & dirt separator

DISCALDIRT® air & dirt separator

multi-zone controls

differential pressure bypass valve

sensor fittings

Hydro Separator

Hydro Separator

Hydro Separator

single zone control

zone valve (2 way)

sensors

zone valve (3 way)

flow meter

ThermoCon buffer tank

LEGIOMIX®

thermoelectric zone valve (2 way)

Hydro Separator

TANKMIXER™

motorized ball valve (2 way)

THERMOSETTER™ adjustable balancing valve

motorized ball valve (3 way)

variable orifice balancing valve

thermostatic radiator valve

thermostatic radiator valve

FLOWCAL™ balancing valve

thermostatic radiator valve

FLOWCAL™ balancing valve

dual isolation valve for panel radiators

fixed orifice balancing valve

QuickSetter™ balancing valve w/ flowmeter

high-temperature solar DISCAL air separators

high-temperature solar pressure relief valve

high-temperature solar 3-way thermostatic mixing valve

HydroBlock mixing units

boiler protection valve

high-temperature solar air vent

isolar differential temperature controller

ThermoBloc™

high-temperature solar shut-off valve for solar air vent

high-temperature solar expansion tank

Hydroink™ (4 configurations)

solar circulation station

high-temperature solar expansion tank

symbols are in Visio library @ www.caleffi.com



Magnetic Separation Removes ferrous impurities

Ferrous oxide forms in hydronic systems when iron or steel corrodes. This abrasive, extremely fine sediment is difficult to remove; it can deposit onto heat exchanger surfaces and accumulate in pump cavities causing reduced efficiency and premature wear. Caleffi magnetic dirt separators accomplish 2½ times the ferrous oxide removal performance of standard dirt separators.



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