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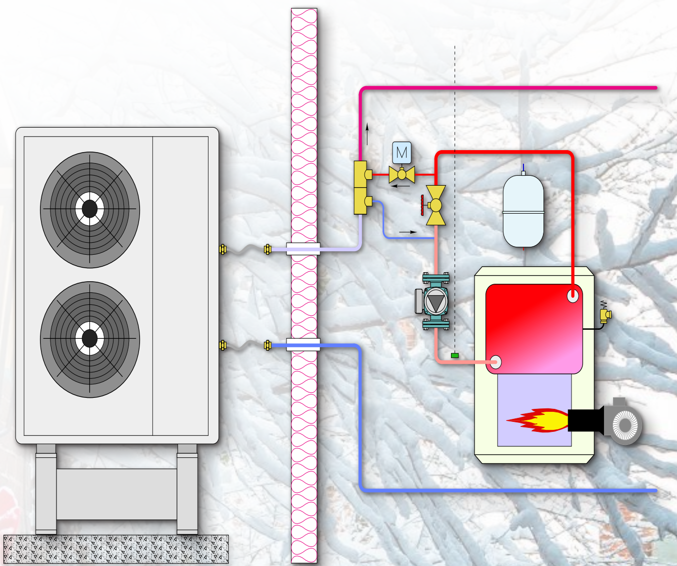

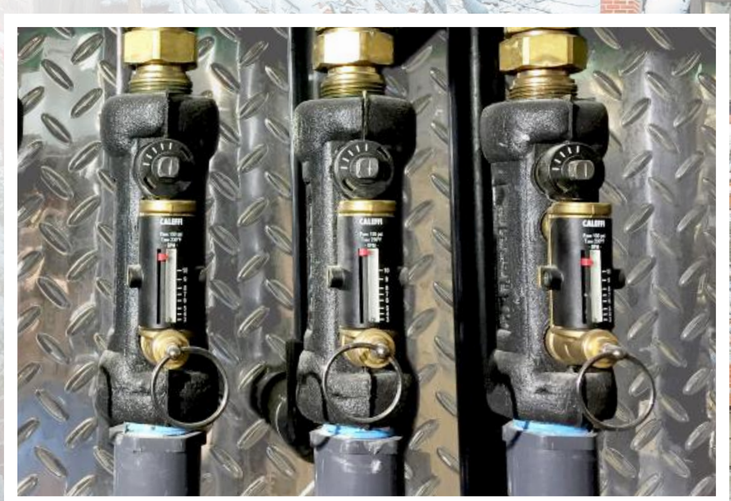

Evaluating Existing Hydronic Heating Circuits

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Best in the business dirt and magnetite removal! Caleffi leads the way with impurity separation technology to protect critical hydronic components. The DIRTMAG® Pro has dual magnetic fields and the XF™ adds a unique mesh filter and brush for first-pass cleaning. Don't settle for less, blowdown without the mess. **CALEFFI GUARANTEED.**

FROM THE CEO

Dear Plumbing and Hydronic Professional,

My name is Tina Gullickson and I am the CEO of Caleffi North America. As I joined the Caleffi team in 2023, one constant was immediately clear: we need to continue to uphold a high standard for the *idronics* journal series.

You can count on *idronics* to keep you abreast of critical energy efficiency concerns and technologies while delivering thermal comfort to your customers through well-designed hydronic systems. As a system component manufacturer, we don't write the policies or building codes. However, together we can identify best practices to develop systems that exceed the expectations defined by policymakers and the building occupants.

Our aim with this release, continuing in the same tradition of Excellence in Education, is to help evaluate the performance of existing hydronic circuits. Distribution efficiency doesn't carry an AFUE tag or ENERGY STAR sticker, but it is equally vital for whole-building performance.

The *idronics* journal is one of many ways we help support an essential industry that has the achievable potential to reshape the world in an energy efficient, comfortable way.

We hope you enjoy this issue and encourage you to send us any feedback by emailing us at idronics@caleffi.com. An entire collection of the journal series can be found at idronics.caleffi.com.

Tina Gullickson

Tina Gullickson

CEO, Caleffi North America

CALEFFI Hydronic Solutions

idronics

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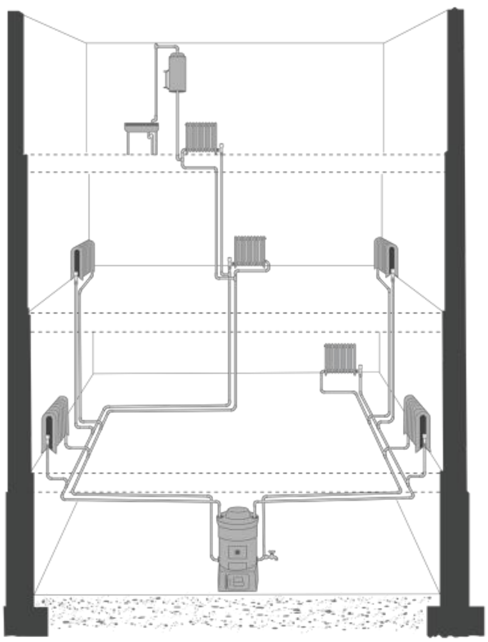
1. INTRODUCTION

Hydronic heating systems have been in use since the late 1800s. Like most technologies, the methods and materials used in these systems have continually advanced.

In the early 1900s, a new hydronic system likely burned coal in a cast iron boiler to heat the water to 190ºF or higher. It relied solely on the buoyancy of that water to carry heat to cast iron radiators. Although circulators were still several decades away, a rudimentary system, such as shown in figure 1-1, was able to move heat to where it was needed, even in multi-story buildings.

Figure 1-1



Today, a new hydronic system is likely to have a modulating/condensing boiler or heat pump as its heat source and may only heat water to 110ºF. Several components in the system, including the controls, the heat source, and the circulator(s), use microprocessor-based electronics to direct heat exactly where it's needed in a building, while optimizing fuel and distribution efficiency.

WHAT'S AHEAD?

There are an estimated 6 million hydronically heated homes in the United States. The vast majority of those systems have a boiler operating on fossil fuel as their heat source. Many of the systems that were installed 40+ years ago are likely operating with high supply water temperatures in the range of 160-200ºF. This was common practice when fuel was relatively inexpensive, and the efficiency of converting that fuel into heat was not a top priority. Higher water temperatures

Figure 1-2

also allowed for smaller heat emitters, which reduced installation cost.

As this issue of *idronics* is being written, global energy markets are changing rapidly. Social and political forces are incentivizing the use of electricity as a replacement for fossil fuels in a wide range of applications, including building heating. This trend is largely based on projections that electricity will be increasingly supplied by renewable sources, such as utility-scale solar photovoltaic systems, wind turbines and hydroelectric facilities.

These changes in energy markets will significantly alter the future of hydronics technology. Mechanical codes and other government regulations will increase the minimum acceptable thermal efficiencies of all hydronic heat sources. The price of fossil fuels is also likely to increase based on market demand, as well as directives such as carbon taxes.

Air-to-water heat pumps, as well as geothermal water-to-water heat pumps, will be increasingly used as heat sources for hydronic-based systems. In many retrofit situations, these heat pumps will be combined with an existing boiler, such as shown in figure 1-3. The heat pump serves as the primary heat source. The existing boiler serves as the supplemental and backup heat source.

Figure 1-3

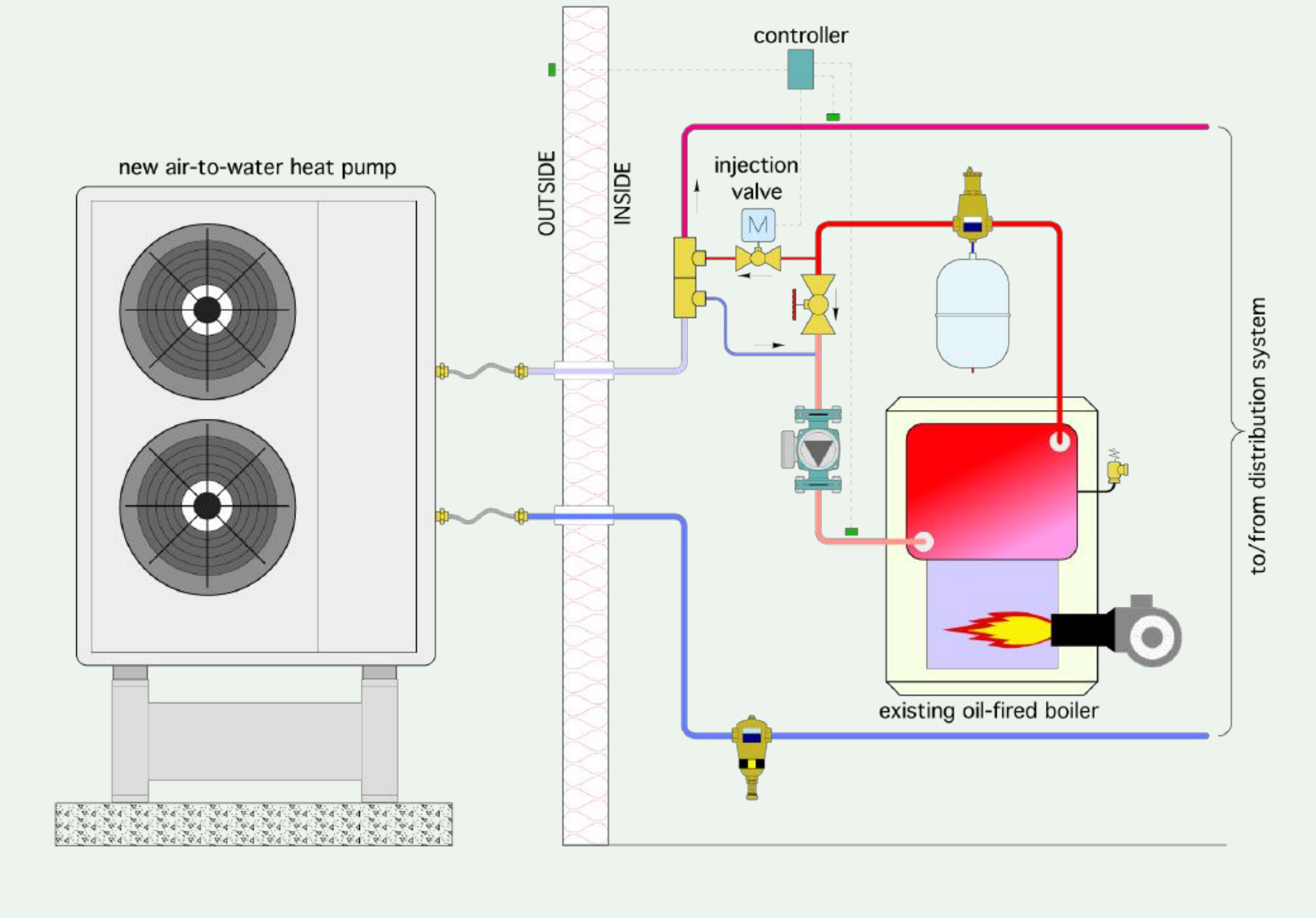


Figure 1-4

Conventional boilers designed to operate without sustained flue gas condensation, and having maximum thermal efficiencies in the mid-80% range, are likely to be limited to very specific applications, such as for steam production.

Depending on current and pending regulations, boilers used in future hydronic heating systems may be limited to those operating on electricity or a renewable fuel source, such as biogas, biodiesel, biomass (primarily wood-based) or hydrogen.

These changes will create many *opportunities* for transitioning an existing hydronic heating system supplied by a conventional boiler to a contemporary heat source, such as a hydronic heat pump or modulating/condensing boiler.

Some of these opportunities will be based on the existing boiler reaching a condition where keeping it in operation is impractical, impossible or even *illegal*. Others will arise from the owner's desire to reduce their carbon emissions by eliminating their existing fossil fuel boiler. Still others will be driven by the changing economics of fuel options.

Figure 1-5

Figure 1-6

ESTIMATING vs. GUESSING

Imagine someone who wants to build a race car. They've done research and determined that engines built by Ferrari offer excellent performance, so they make a major investment and purchase a 10-cylinder, twin-turbocharged Ferrari engine capable of delivering over 700 horsepower.

The engine arrives and gets installed in the vehicle shown in figure 1-6.

Do you think this combination will result in a high-performance race car?

It's very unlikely that the transmission in the 1957 Chevrolet could utilize the horsepower the Ferrari engine can deliver. The car's body shape is also poorly matched to the expectation of a vehicle that can cruise at speeds over 200 mph. In short, this is a very poor matching of an energy source (the engine) with a "balance-of-system" (the car body and transmission). The performance of such a matching would be far from optimal.

Although most readers recognize this as a far-fetched combination, it's analogous to installing a high-performance hydronic heat source, such as an air-to-water heat pump or mod/con boiler, into an existing hydronic distribution system that was never intended to operate at conditions that would optimize the performance of that heat source.

Professionals who design racing vehicles would never "guess" when it comes to matching an engine to a chassis, transmission or body. They would make calculations or perform tests and simulations to study the potential matchup.

In a similar manner, HVAC professionals need to study the potential matching of a high-performance heat source with an existing distribution system. In doing so, they can identify potential problems or performance deficiencies before committing to a specific design.

Simply removing the existing boiler and installing the new heat source, without understanding how it will interact with the existing distribution system can lead to problems such as:

- Short cycling of the new heat source
- Insufficient heat output in some areas of the building
- Inadequate flow rate to maintain proper operation of the new heat source
- Seasonal energy use that is much higher than anticipated
- Damage to the new heat source
- Premature failure of the new heat source
- Dissatisfied customers

This issue of *idronics* provides HVAC professionals with methods for evaluating the performance of existing hydronic heating systems. It begins with a discussion of different types of heat emitters and how their thermal performance varies with water temperature. It then presents relatively simple procedures for measuring temperatures and flow rates in existing hydronic circuits. It shows how to use those measurements to estimate the performance of the circuit over a range of operating conditions — such as at lower supply water temperatures. The latter serves as a starting point for evaluating the performance of those circuits with new heat sources. The issue concludes with a discussion of several design considerations when transitioning an existing distribution system to a new heat source.

2. ESTIMATING HEAT EMITTER OUTPUT

The term hydronic heat emitter applies to any hardware used to release heat from water into a space to be heated. Some simple hydronic systems use only one type of heat emitter. More complex systems can have two or more types of heat emitters. When evaluating an existing system, it's important to identify the types of emitters used and have a general understanding of how their output is affected by the water temperature and the surrounding air temperature at which they operate.

This section discusses the basic thermal performance characteristics of several types of heat emitters, including:

- Fin-tube baseboard
- Cast iron radiators
- Fan-coils and air handlers
- Panel radiators

The piping that connects these heat emitters together also emits heat, and as such, affects the total heat output of the circuit. The calculations described below also assume the heat emitters are clean, unobstructed and free of damage. When evaluating existing buildings, always check the condition of the equipment in occupied spaces. Here are some common existing conditions that can reduce heat output:

- Damage to fin-tube baseboards in high-traffic areas, typically bent/compacted fins from accidental impact
- Radiators covered by decorative woodwork
- Panel radiators that have dust or other debris blocking convective airflow
- Furniture placed in front of supply or return grills on convectors or fan-coils

THERMAL PERFORMANCE OF FIN-TUBE BASEBOARD

Fin-tube baseboard, such as shown in figure 2-1, is one of the most common heat emitters used in North American residential and light commercial hydronic systems.

Fin-tube baseboard had its origin in the 1930s as a new competitor to cast iron radiators. At that time, boilers operating at relatively high water temperatures were used in almost all hydronic heating systems. These combustion-based heat sources were capable of heating water to high temperatures with minimal effect on their efficiency. Fuels were relatively inexpensive as fin-tube baseboards gained market share. High water temperatures allowed shorter lengths of baseboard to achieve a given heat output, and thus reduced installation costs. Even today, the rating tables published by baseboard manufacturers list heat outputs at water temperatures up to 220ºF, and in some cases up to 230ºF. An example of such a rating table is shown in figure 2-2.

Figure 2-1

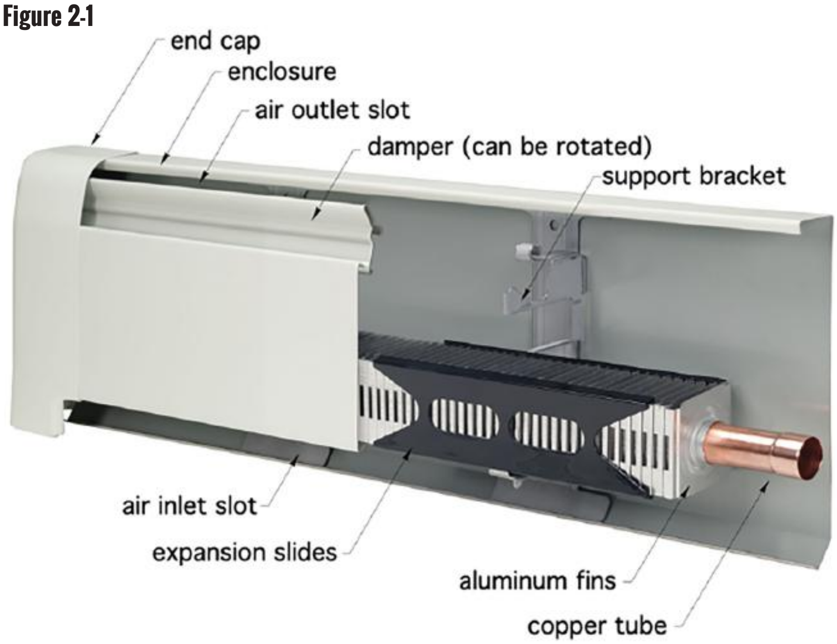


Figure 2-2

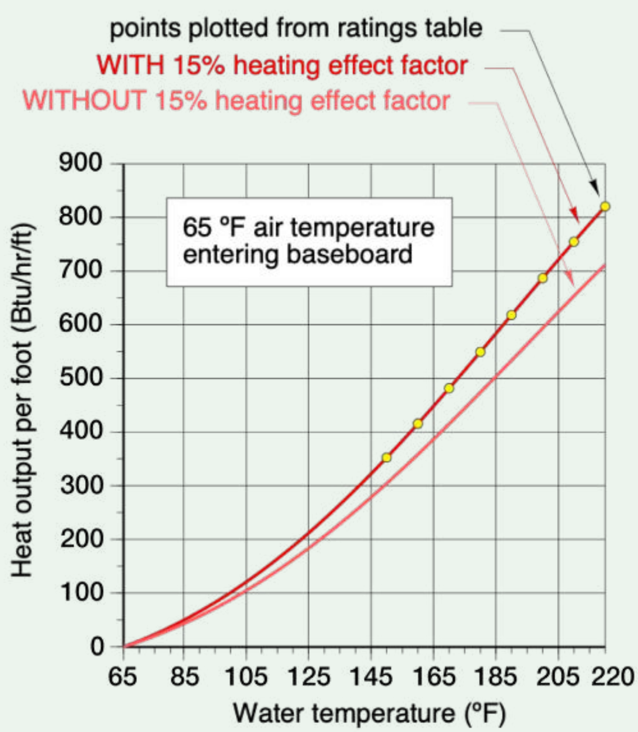
flow rate	AHRI WATER RATINGS BTUH Peer linear Foot At Average Water Temperatures Indicated								
	150 °F	160 °F	170 °F	180 °F	190 °F	200 °F	210 °F	220 °F	230 °F
4 gpm	340	390	450	510	560	610	670	720	770
1 gpm	320	370	430	480	530	580	630	680	730

A footnote that often accompanies a fin-tube rating table states that the ratings include a 15 percent "heating effect factor." When conceived decades ago, this factor was meant to boost the tested heat output by 15 percent to account for the baseboard being installed at floor level, where surrounding air temperatures were often significantly cooler than air temperatures higher in the room. Today's homes do not have such drastic room air temperature stratification, and as such, the heating effect factor is not appropriate. Conservative design practice is to divide any heat output ratings that include the heating effect factor by 1.15 to remove it and use the corrected value when sizing the baseboard.

Based on typical rating tables similar to figure 2-2, one might assume that a fin-tube baseboard cannot operate at water temperatures under 150ºF. This is not the case. Any heat emitter will release heat as long as the fluid inside it is warmer — even slightly — than the surrounding air temperature.

Figure 2-3 shows a graph of the heat output per foot of fin-tube length for typical residential grade fin-tube baseboard as a function of the average water temperature in the fin-tube, and assuming an air temperature of 65ºF at floor level.

Figure 2-3



The lower curve is based on removing the 15 percent heating effect factor from the manufacturer's ratings.

Although the outputs are relatively small, the curves on this graph show some heat output even at water temperatures barely above the 65ºF assumed floor-level air temperature.

The lower curve on the graph can be very helpful in evaluating the potential performance of fin-tube baseboard in systems operating at low water temperatures and supplied by contemporary heat sources, such as hydronic heat pumps, mod/con boilers and thermal storage systems.

Example: A baseboard manufacturer lists the output of their fin-tube baseboard as 680 Btu/hr/ft at an average water temperature of 200ºF. That rating includes the 15 percent heating effect factor. Use figure 2-3 to estimate the output of a 12-foot length of that baseboard when operated at an average water temperature of 105ºF, and assuming air enters the baseboard at 65ºF.

Solution: First remove the 15% heating factor by using the lower curve in figure 2-3. Draw a line up from 105ºF on the horizontal axis until it meets the curve. Then draw a line to the vertical axis to read an output of approximately 100 Btu/hr/ft. Finally, since the baseboard is 12 feet long, its total output under the stated conditions would be about 12 ft x 100 Btu/hr/ft = 1200 Btu/hr.

It's also possible to *estimate* the heat output of a fin-tube at different entering water and air temperatures using Formula 2-1.

Formula 2-1:

$$q \approx a\left(T_w - T_{air}\right)^{1.4}$$

Where:
q = heat output of fin-tube per unit of length (Btu/hr/ft)
a = a number to be found from rating data or figure 2-3
Tw = average water temperature in baseboard (ºF)
Tair = air temperature entering baseboard (ºF)
1.4 = an exponent

The value of (a) in Formula 2-1 needs to be "calibrated" for a specific make and model of baseboard. This is done by choosing an output rating about midway between the highest and lowest ratings on the chart, removing the 15 percent heating effect factor (if the footnote with the rating chart indicates it's included), and then putting all the known values into a rearranged form of Formula 2-1:

Formula 2-1 (rearranged):

$$a = \frac{q}{\left(T_w - T_{air}\right)^{1.4}}$$

Example: The rating chart in figure 2-2 indicates a heat output of 510 Btu/hr/ft at an average water temperature of 180ºF. Assume that the 15 percent heating effect factor is included in this rating. Estimate the true heat output rate of the fin-tube if operated at an average water temperature of 110ºF in a basement where the air temperature entering the baseboard is 58ºF.

Solution: First, remove the 15 percent heating effect factor:

$$q_{corrected} = \frac{510}{1.15} = 443 \frac{Btu}{hr \bullet ft}$$

Next, find the value of (a) by putting the data from the rating table into the rearranged form of Formula 2-1:

$$a = \frac{q}{\left(T_w - T_{air}\right)^{1.4}} = \frac{443}{\left(180-65\right)^{1.4}} = \frac{443}{252.6} = 0.577$$

Finally, use Formula 2-1 along with the new operating conditions, and value of (a) to get the estimated heat output.

$$q \approx a\left(T_w - T_{air}\right)^{1.4} \approx 0.577\left(110-58\right)^{1.4} = 145.7 \frac{Btu}{hr \bullet ft}$$

ESTIMATING HEAT OUTPUT OF CAST IRON RADIATORS

Many older hydronic systems have cast iron radiators. The physical design of such radiators varies widely. The two most common configurations are “column-type” and “tube-type” radiators. Both types consist of multiple cast iron sections that are joined together to create the overall radiator. Cast iron sections are made with different heights and widths, but a given radiator is assembled using identical sections.

Figure 2-4 shows the geometry of a column-type radiator, along with a chart giving common heights and section widths. The numbers in the table indicate the *square footage surface area of each section.*

Figure 2-5 shows the geometry of tube-type radiators with a similar table listing the height, width and square footage surface area of each section.

To determine the total heat output of a given cast iron radiator, multiply the square footage surface area of each section (from figure 2-4 or 2-5) by the number of sections in the radiator, and then use Formula 2-2.

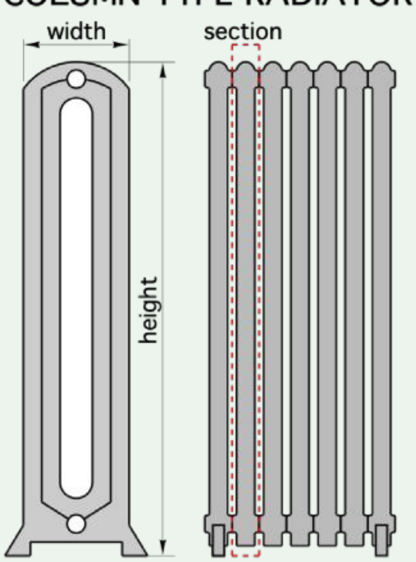
Formula 2-2:

$$Q = 0.3748\left(A_{total}\right)\left(T_w - T_{room}\right)^{1.3}$$

Where:
Q = heat output from radiator (Btu/hr)
A_{total} = total surface area of all sections in radiator (ft^2)
T_w = average water temperature in radiator (ºF)
T_{room} = air temperature surrounding radiator (ºF)
1.3 = an exponent

Figure 2-4

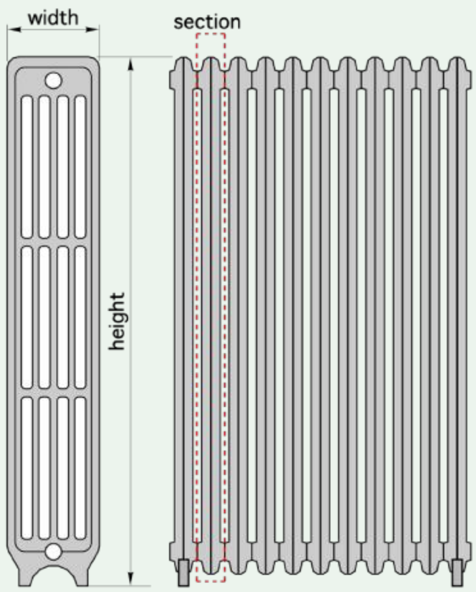
COLUMN TYPE RADIATOR



COLUMN RADIATORS (square feet per section)

height (inches)	4.5” wide, 1 column	7.5” wide, 2 column	9” wide, 3 column	11.5” wide, 4 column	13” wide, 5 column
13	-	-	-	-	3
16	-	-	-	-	3.75
18	-	-	2.25	3	4.25
20	1.5	2	-	-	5
22	-	-	3	4	-
23	1.67	2.33	-	-	-
26	2	2.67	3.75	5	-
32	2.5	3.33	4.5	6.5	-
38	3	4	5	8	-
45	-	5	6	10	-

Figure 2-5 TUBE TYPE RADIATOR



TUBE RADIATOR (square feet per section)

height (inches)	5" wide, 3 tubes	7" wide, 4 column	8.75" wide, 5 column	9.75" wide, 6 column	12.5" wide, 7 column
14	-	-	-	-	2.5
17	-	-	-	-	3
20	1.75	2.25	2.67	3	3.67
23	2	2.5	3	3.5	-
26	2.33	2.75	3.5	4	4.75
32	3	3.5	4.33	5	-
38	3.5	4.25	5	6	-

Example: determine the heat output of a tube-type cast iron radiator having a section height of 32 inches, a section width of 7 inches, and made using 10 sections. The average water temperature in the radiator is 115ºF, and the room air temperature surrounding the radiator is 70ºF.

Solution: From figure 2-5, the surface area of each section of this radiator is 3.5 ft^2. The total surface area is 10 x 3.5 = 35 ft^2. Putting this value and the other stated conditions into Formula 2-2 yields:

$$Q = 0.3748 A_{total} (T_w - T_{room})^{1.3} = 0.3748(35)(115-70)^{1.3} = 1850 \frac{Btu}{hr}$$

ESTIMATING HEAT OUTPUT OF FAN-COILS & AIR HANDLERS

The heat output of "fan-forced" convectors (e.g., fan-coils & air handlers) is approximately proportional to the difference between the temperature of the fluid entering the coil and the temperature of air entering the coil. This can be mathematically described using Formula 2-3.

Formula 2-3:

$$Q = c(T_f - T_{air})$$

Where:
Q = rate of heat output (Btu/hr)
c = a constant to be determined for a specific fan-coil or air handler (Btu/hr/ºF)
T_f = temperature of fluid entering coil (ºF)
T_{air} = temperature of air entering coil (ºF)

Because they are produced in many styles and sizes, the only way to determine the value of "c" is to insert a rating point from the manufacturer into Formula 2-3.

Example: An air handler is rated by its manufacturer to output 50,000 Btu/hr when the water temperature entering the coil is 140ºF and the air temperature entering the air handler is 65ºF. Determine the value of "c" for this air handler.

Solution:

$$c = \frac{50{,}000 \frac{Btu}{hr}}{(140^\circ F - 65^\circ F)} = 667 \frac{Btu}{hr \bullet^\circ F}$$

Once the value of "c" is determined, the output of the air handler can be estimated for other entering fluid and air temperatures.

Further "corrections" to estimated heat output may be possible based on the type and extent of performance data provided by the manufacturer. For example, the heat output of any fan-coil or air handler will increase as the flow rate through the coil increases. This happens because higher flow rates increase the average fluid temperature within the coil, and thus increase its heat transfer rate. When the coil is operating at typical flow rates suggested by the manufacturer, the gain in heat output at higher flow rates is relatively small. However, if the coil is operating at flow rates well below nominal suggested conditions, the change in heat output can be substantial and must be accounted for.

Some manufacturers may provide correction factors for such conditions.

One "approximation" that describes the change in heat output of a coil as a function of flow rate, assuming the constant entering fluid and entering air temperatures, is that the coil will attain about 90% of its rated heat output when operated at about 50% of its rated flow rate. Figure 2-6 shows a typical relationship between heat output from a coil and flow rate.

Other hydronic heat emitters have similar relationships between heat output rate and flow rate. The "non-linear" characteristic of the graph representing this relationship shows that operating any heat emitter at abnormally high flow rates produces very little thermal benefit. Doing so also substantially increases pumping power requirements.

ESTIMATING HEAT OUTPUT OF PANEL RADIATORS

In North America, it is common to find the heat output of panel radiators based on an average water temperature of 180ºF, and assumed room air temperature of 68ºF. Figure 2-7 shows an example of a typical rating table, where the heat output is based on this temperature difference (e.g., 180 - 68 = 112ºF).

It's possible to estimate the heat output of panel radiators at other average water temperatures and surrounding air temperatures using the graph in figure 2-8.

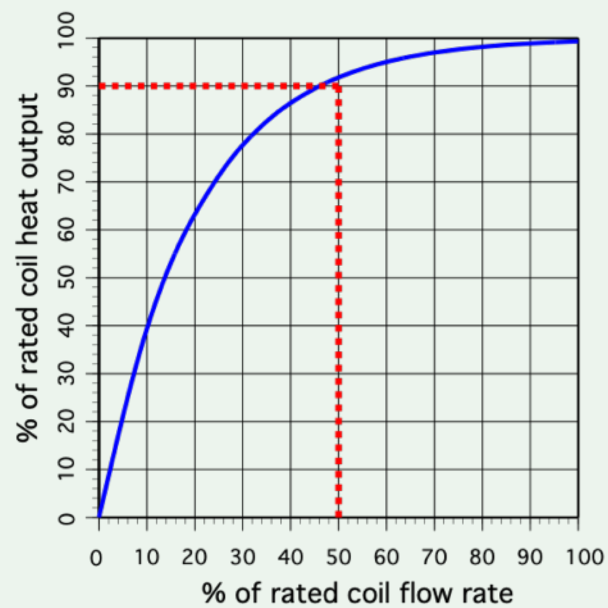
To estimate heat output, first determine a value for the difference between the average water temperature in the panel radiator and the surrounding room air temperature. Find the value on the horizontal axis. Follow a vertical line up to the red curve, and then a horizontal line to the vertical axis. The value on the vertical axis is a correction factor.

Multiply the rated heat output from figure 2-7 (which is based on 180ºF average water temperature, and 68ºF room air temperature) by the correction factor from figure 2-8 to get the estimated heat output at the new condition.

Example: Determine the heat output of a double water plate panel radiator that is 24 inches high and 48 inches long, when operated at 110ºF average water temperature and 70ºF surrounding room air temperature.

Solution: The difference between average water temperature and room air temperature is (110ºF - 70ºF) = 40ºF. From the graph in figure 2-8, the correction factor is approximately 0.26. The table in figure 2-7 lists the output at the reference conditions to be 9,500 Btu/hr. So, the output at the lower water temperature condition is (0.26) x 9,500 = 2,470 Btu/hr.

Figure 2-6



Although the heat output at the lower water temperature is only about one quarter of the output at the reference condition, that output is still adequate to heat a room of about a 250 square foot floor area having a design heating load of 10 Btu/hr/ft^2.

Panel radiators can work well with low-temperature heat sources, such as hydronic heat pumps and mod/con boilers. Large panels are required at lower water temperatures.

As the average water temperature decreases, the percentage of heat leaving a panel radiator as radiant output rather than convective output increases. This is beneficial, since radiant output helps warm surrounding surfaces, increasing the mean radiant temperature of the room. Figure 2-9 shows the percentage of total heat output that is radiant heat for different panel radiator configurations and as a function of the difference between average water temperature and surrounding room air temperature.

Figure 2-9 shows that the highest ratio of radiant heat output is provided by panel radiators without fins. As fins are added, the amount of heat leaving the panel due to convection increases and radiant heat output decreases. Using panels with more fins always increases total heat output.

Figure 2-7

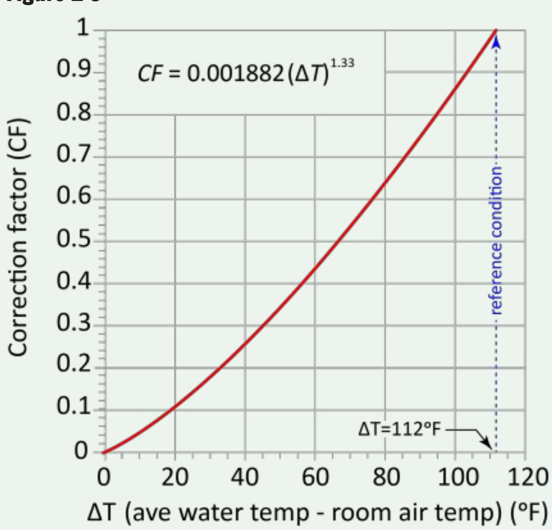
Heat output ratings (Btu/hr)
at reference conditions:
Average water temperature in panel = 180°F
Room temperature = 68°F
temperature drop across panel = 20°F

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

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

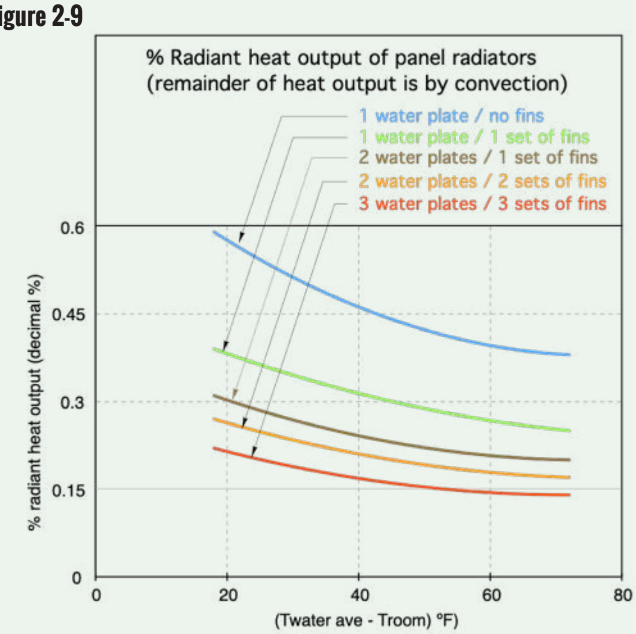
	3 water plate panel thickness					
	16" long	24" long	36" long	48" long	64" long	72" long
24" high	4531	6830	10247	13664	18216	20494
20" high	3934	5937	9586	11870	15829	17807
16" high	3320	4978	7469	9957	13277	14938
10" high	2191	3304	4958	6609	8811	9913

Figure 2-8



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

Figure 2-9



3. ON-SITE MEASUREMENTS OF CIRCUIT PERFORMANCE

There are different ways to evaluate the hydraulic and thermal performance of existing hydronic circuits.

One approach relies on *theoretical modeling* of the circuit and its circulator. It starts by finding the circuit's hydraulic operating point (e.g., the flow rate where head energy added by the circulator equals the head energy loss of the circuit). Once the flow rate is calculated, a thermal analysis of the circuit, based on heat transfer models for the heat emitters and connecting piping, finds a relationship between supply water temperature and heat output. .

This *analytical* approach requires a substantial amount of on-site work to document lengths and sizes of piping, fittings count, type and size of heat emitters, and pump curves for all circulators in the system. If the circuit operates with antifreeze, the concentration of that antifreeze also needs to be determined. All this information is required to build hydraulic and thermal models for the circuit.

Another approach is based on *physically measuring* the thermal performance of a circuit as it operates. This method requires heat input to the circuit, as well as time to allow the circuit to settle to (or very close to) steady state conditions. It also requires accurate measurement of flow rate and temperature drop across the circuit. This approach requires some instrumentation but less data gathering for each circuit. This section describes the overall process.

MEASUREMENTS NEEDED

There are two quantities that must to be measured as part of evaluating the heat output of a hydronic circuit:

- Temperature drop (ΔT) (ºF)
- Flow rate (gpm)

These measurements will be combined, along with two physical properties of the fluid flowing through the circuit, to calculate its heat dissipation rate.

TEMPERATURE MEASUREMENTS

The temperature drop of the circuit can be measured using two identical temperature sensors, such as thermocouples, thermistors or other resistance temperature detectors (RTDs), that are temporarily fastened to the outside of the piping at the beginning and end of the circuit being tested, as shown in figure 3-1.

Temperature-measuring instruments that use thermocouple sensors are widely available and relatively inexpensive.

Thermocouples are made by bonding the ends of two wires made of dissimilar metals to form a junction. That junction generates a very small but very repeatable voltage that changes as the temperature of the junction changes. The sensing area of a thermocouple is very small, as shown in figure 3-2.

To make a temperature measurement, the end of a thermocouple can be temporarily secured to the outer surface of a pipe using electrical tape. When two thermocouples are used

Figure 3-1

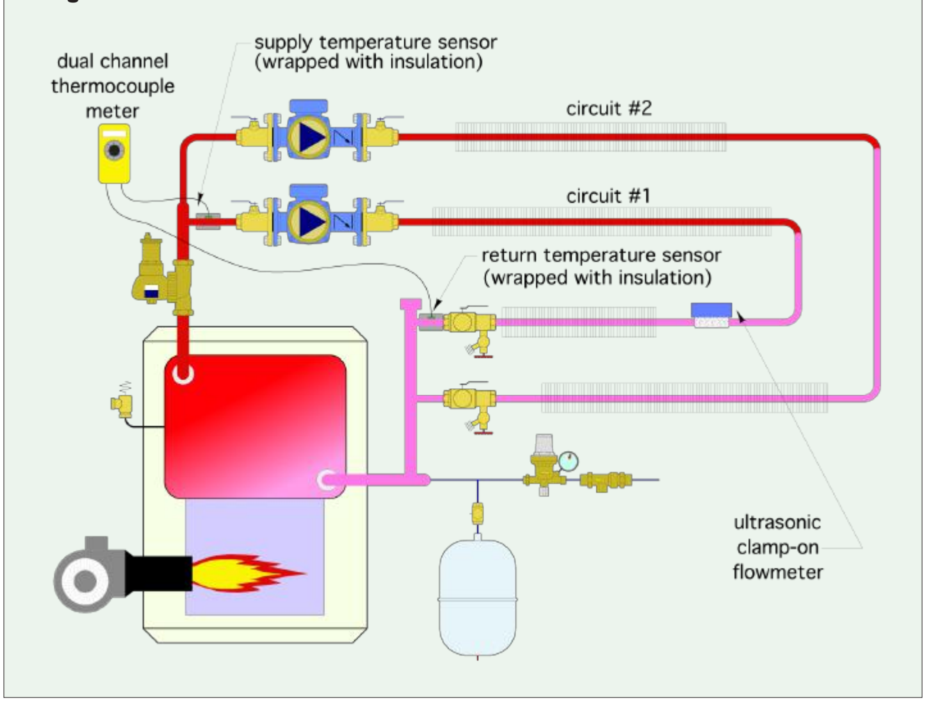


Figure 3-2

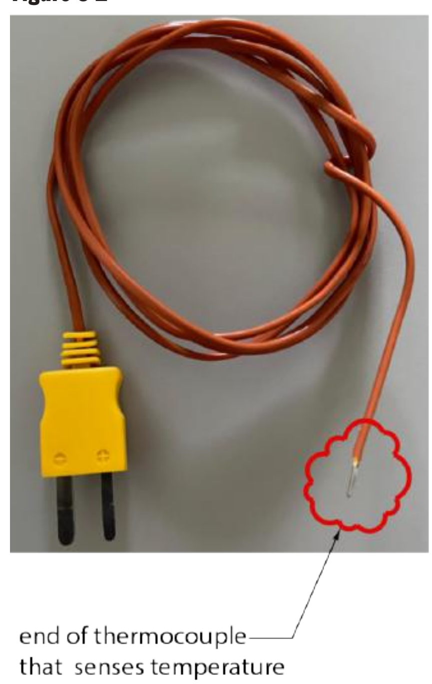


Figure 3-3a

to measure a temperature difference in a hydronic circuit, they should both be mounted to pipe surfaces *using identical methods.*

After mounting, both thermocouples should also be wrapped with a nominal 3-inch length of elastomeric foam insulation, as shown in figure 3-3. This minimizes the influence of surrounding air temperature on the readings.

Thermocouples are very small, and as such, have very low thermal mass. This allows them to stabilize to the temperature of the pipe very quickly.

An ideal instrument for reading the temperature from both thermocouples is a dual-channel thermocouple meter, such as shown in figure 3-4.

Dual channel thermocouple meters typically have a setting for temperature difference (a.k.a., ΔT). These meters can read both sensors at the same instant and calculate the temperature difference. This improves accuracy, since it eliminates the need to change the connections between each sensor and the meter, as would be the case with a single channel meter.

Figure 3-3b

FLOW MEASUREMENT

There are several ways to measure flow rate in a hydronic circuit. The method used depends on the required accuracy of the measurement and the hardware installed in the existing system.

If the circuits for which flow rate needs to be measured have existing balancing valves, such as the Caleffi QuickSetter™ or Flo-Set™, they can be used to measure flow rate to an accuracy of +/-10 to +/-15%.

The QuickSetter valve shown in figure 3-5a uses a direct-reading flowmeter and has a specified accuracy of +/-10%.

The Flo-Set valve, shown in Figure 3-5b, is available with either a fixed orifice or variable orifice. The fixed orifice valve has a specified accuracy of +/-10%. The variable orifice valve has a specified accuracy of +/-15%. Both versions of the Flo-Set valve require a manometer to measure the pressure drop across the valve's orifice, which can then be combined with the valve's setting to infer flow rate.

Another option is a circulator with an ECM motor, and either a direct

Figure 3-3c

Figure 3-4

Figure 3-5a

Figure 3-5b

readout or a Bluetooth interface to a smart phone or tablet. An example of each is shown in figure 3-6.

ECM circulators do not directly measure flow rate, they infer flow rate

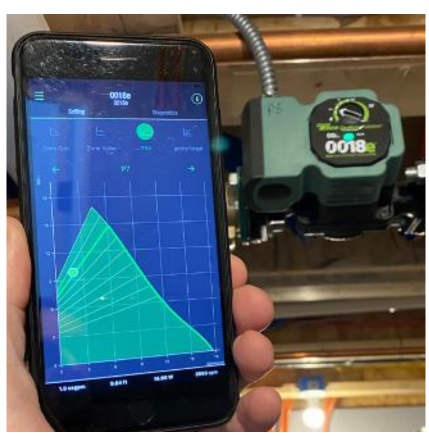
Figure 3-6a

based on the circulator's power input, its RPM, and the internal calculation algorithms. They are capable of +/-5% accuracy when operated under the following conditions:

- They are free from any oxide accumulation or debris.
- They are operating with water (glycol solutions will increase wattage and reduce accuracy).
- They operate to the left of their maximum hydraulic efficiency point.

Another way to measure circuit flow rate is a flowmeter directly installed in the fluid stream. A wide variety of these "inline" flowmeters are available.

Figure 3-6b



Some are classified as "float-type" meters (rotameters), which operate based on flow passing through a variable cross sectional area, which lifts a metal "float" within a calibrated transparent tube. Others are based on a turbine or paddle wheel that spins as flow passes through. In general, higher accuracy meters cost more than lower accuracy meters.

However, it's uncommon to find inline flowmeters permanently installed in existing residential and light commercial hydronic systems. Doing so would add cost, create additional points of potential leakage, and require the system fluid to be free of particles or other debris that could cause the metering device to jam, become difficult to read or otherwise affect its accuracy.

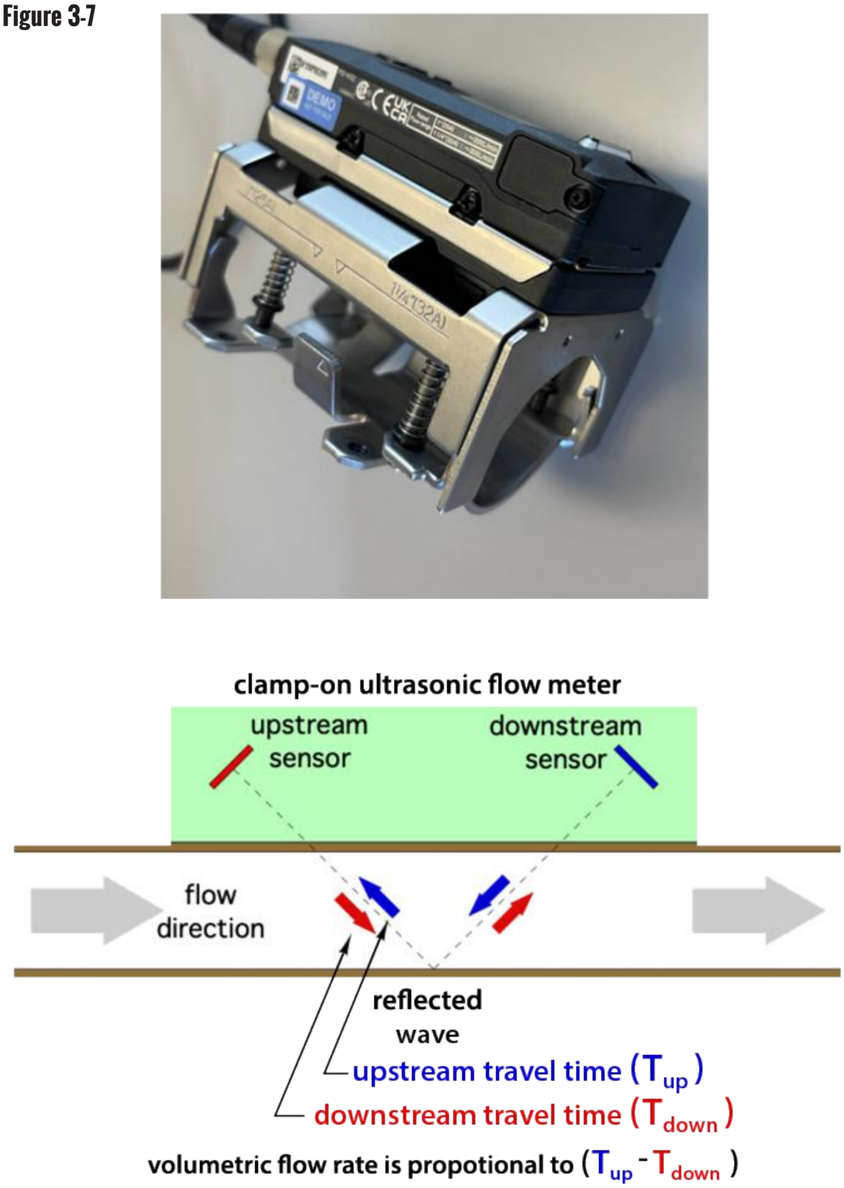
For purposes of calculating heat transfer rates in residential and light commercial systems, flowmeters with a resolution of 0.1 gpm are suggested. This is especially important for smaller circuits with nominal flow rates of 1 to 3 gpm.

One of the most convenient and accurate devices for measuring flow rates in existing circuits is an ultrasonic flowmeter. *These meters do not have flow passing through them. Instead, they clamp to the outside of a pipe.*

Ultrasonic flowmeters have two sensors that emit and detect a high frequency (ultrasonic) pulse in the range of 0.5 to 4 Mhz. The two sensors are displaced a short distance along the length of the pipe, as shown in figure 3-7.

During operation, the upstream sensor emits an ultrasonic pulse, which travels through the pipe wall and the fluid stream, reflects off the opposite pipe wall, goes back through the fluid stream, through the opposite pipe wall, and is finally

Figure 3-7



detected by the downstream sensor. The time between the pulse being emitted and detected in the direction of flow is very accurately measured.

The function of the two sensors then reverses. The downstream sensor emits an ultrasonic wave at the same frequency, which again passes through the pipe walls and fluid, but in a direction opposite the flow. The time between emission and detection is again very accurately measured. The *difference* between these two “flight times” is directly proportional to the fluid’s volumetric flow rate.

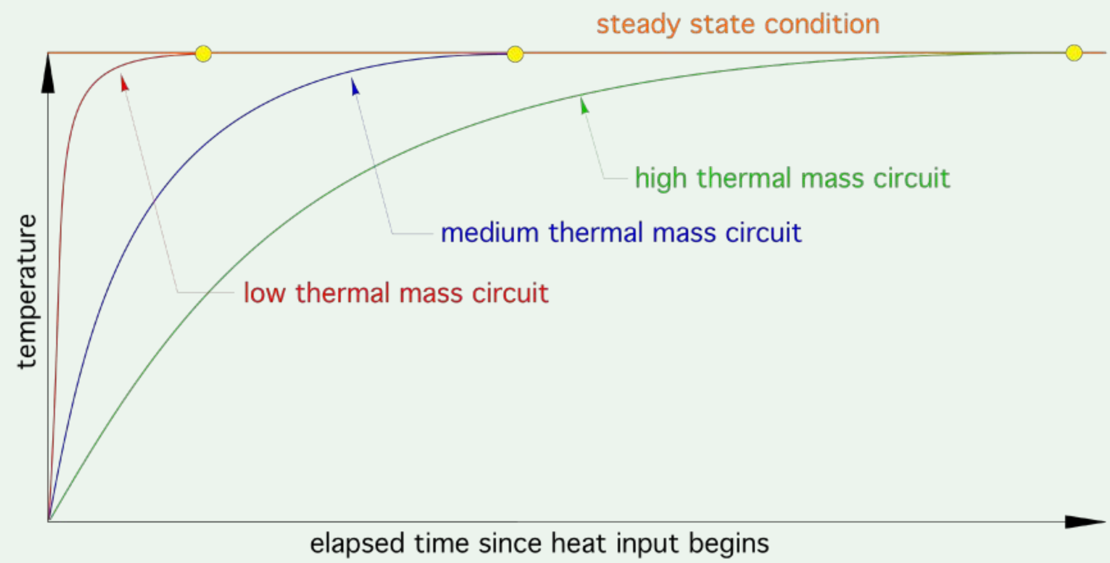
Ultrasonic flowmeter technology has advanced in terms of improved accuracy and lower cost. Meters such as the one shown in figure 3-8 are now available and affordable for heating technicians needing a simple, fast and non-invasive way to accurately measure flow rates.

Figure 3-8

This meter can be quickly mounted to the outside of a pipe. It can be used with a wide range of pipe materials and sizes, as well as different fluids, and over a wide range of temperatures. Reported accuracy is +/-3% of the reading value. When equipped with optional platinum RTD temperature sensors, the meter shown in figure 3-8 can directly measure the ΔT of a hydronic circuit and combine that measurement with flow rate to calculate the rate of heat transfer.

The meter can be mounted at any location in the circuit since the flow rate is constant throughout. Some meters require minimum straight piping length both upstream and downstream of the meter. Be sure to check the placement requirements when deciding where to mount the meter.

Figure 3-9



STEADY STATE CONDITIONS

Whenever a hydronic circuit and its associated heat source are turned on, the temperature of everything in the circuit begins to rise. Components with low thermal mass and high thermal conductivity, such as a copper water tube, warm rapidly. Components with much higher thermal mass, such as a cast iron boiler or a concrete floor slab, warm at much slower rates. As these changes occur, the system is said to be operating under transient conditions. *Measurements of temperature drop and flow rate under transient conditions do not accurately reflect the performance of the circuit.*

If the rate of heat input is stable and continuous, all components in the system, including the fluid at any given location, reach a stable temperature, as shown in figure 3-9. This is called a steady state.

There can be wide variations in the time required for a circuit to reach steady state conditions. Short piping circuits of small-diameter pipe, serving low thermal mass heat emitters such as fin-tube baseboard, and supplied by a low thermal mass heat source, may reach steady state in 10-15 minutes. Longer circuits built with larger piping and serving heat emitters such as cast iron radiators, could take an hour or more to reach steady state. Systems that heat concrete floor slabs can take several hours to reach steady state.

At steady state, the rate of heat input to the circulating fluid equals the rate of heat dissipation from the circuit. *This is the condition at which temperature and flow rate measurements should be taken.*

ADJUSTING CIRCUIT OPERATING CONDITIONS

When the heat output of a heat source is greater than the heat dissipation rate of the distribution circuit — which is almost always true for multi-zone systems with non-modulating heat sources — the heat source will reach its high-temperature limit setting before the circuit reaches steady state conditions. When this occurs, the heat source stops injecting heat to the circuit and the circuit's temperature starts to decrease as heat is dissipated. After the circuit cools through the differential temperature setting of the heat source's high limit controller, the heat source will start again. These on/off cycles of the heat source will repeat as long as there is a call for heating. This makes it difficult to attain steady state conditions.

The swing in supply water temperature from the heat source can be reduced by reducing the high-limit temperature setting of the heat source, *and also reducing the temperature differential between when the heat source turns on and off.* The goal is to find a "warm" fluid temperature, perhaps 20-25ºF above room air temperature, that the heat source can maintain as steady as possible. *The fact that this temperature may be well below the normal operating temperature of the circuit can be dealt with using methods presented later in this section.* Keeping this temperature relatively low and stable also reduces the time required for the circuit to achieve steady state.

CALCULATING HEAT TRANSFER RATES

When the flow rate and temperature drop of a circuit are stable (e.g., the circuit is operating at steady state

Figure 3-10

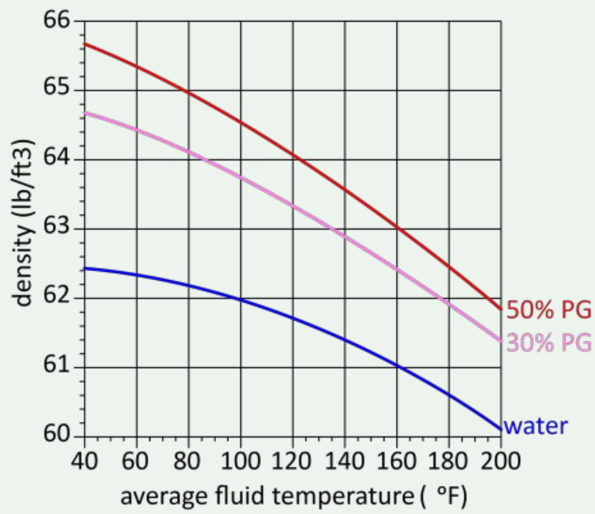
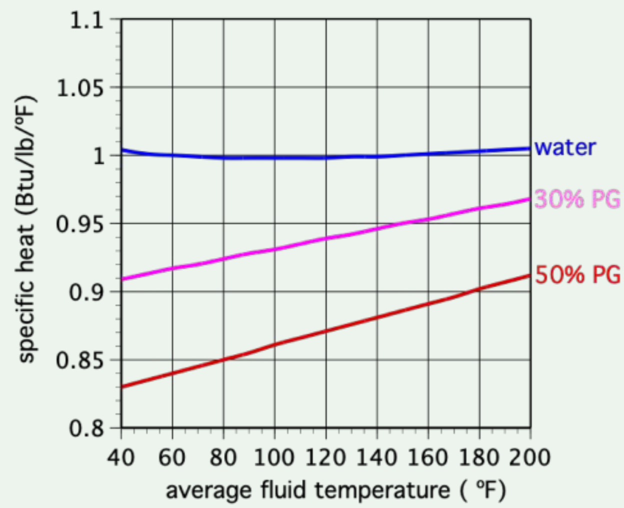


Figure 3-11



conditions), the rate of heat transfer from that circuit can be calculated using formula 3-1.

Formula 3-1:

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

Where:
Q = rate of heat dissipation from circuit (Btu/hr)
D = density of the fluid at the average temperature of the circuit (lb/ft^3)
c = specific heat of the fluid at the average temperature of the circuit (Btu/lb/ºF)
f = flow rate through circuit (gpm)
ΔT = temperature drop of the circuit at steady state conditions (ºF)
8.01 = a constant based on the units in the formula

Both fluid properties (e.g., density and specific heat) are functions of the fluid's temperature. Figure 3-10 shows the density of water as well as 30% and 50% solutions of propylene glycol as a function of temperature. Figure 3-11 shows the specific heat of these same fluids, again as a function of temperature. The density and specific heat of other concentrations of propylene glycol can be estimated from these graphs by interpolation. Use the average temperature of the circuit being tested (e.g., the average of the measured supply and return temperatures) when using these graphs to estimate the fluid's density and specific heat.

Example: Circuit #1 in the system shown in figure 3-12 is turned on and allowed to reach a steady state condition. A dual temperature meter is used to measure the supply and return temperature of the circuit. The circuit's supply temperature is 120.6ºF. The circuit's return temperature is 108.1ºF. A strap-on ultrasonic flowmeter indicates a flow rate of 4.2 gpm. The system operates with a 30% solution of propylene glycol. Estimate the heat output of the circuit using formula 3-1.

Solution: Start by finding the average fluid temperature in the circuit.

$$T_{ave} = \frac{120.6 + 108.1}{2} = 114.35 \simeq 114^{\circ} F$$

Use figures 3-10 and 3-11 to estimate the density and specific heat of the fluid at this average temperature. D = 63.5 lb/ft^3,c = 0.935 Btu/lb/ºF.

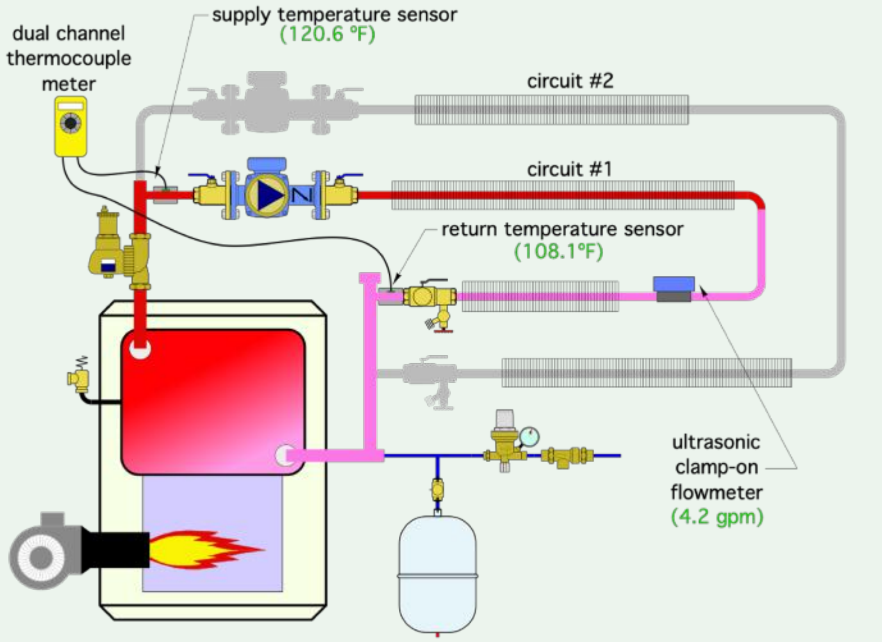
Putting the measured data along with the fluid properties into formula 3-1 yields:

$$Q = 8.01Dc(f)(\Delta T) = 8.01(63.5)(0.935)(4.2)(120.6 - 108.1) = 24968 \simeq 25{,}000\frac{Btu}{hr}$$

The components used for this test do not require depressurization of the system and can be added and removed easily.

The calculations are an instantaneous "snapshot" of the circuit's thermal performance.

Figure 3-12



MEASURING CIRCUIT HEAT OUTPUT USING AN EXTERNAL HEAT SOURCE

It's also possible to measure the thermal performance of a hydronic circuit *without operating that circuit's heat source.* This technique is helpful in situations where the heat source is not operational, or during warm weather testing in which minimal heat release into the space is desirable. The testing method uses a small electrically powered heating device to provide a very stable heat input to the circuit being tested, which expedites achieving steady state conditions.

The procedure is based on temporarily removing the circuit's circulator and installing the test apparatus shown in figure 3-13.

The test apparatus consists of a set of circulator flanges combined

Figure 3-13

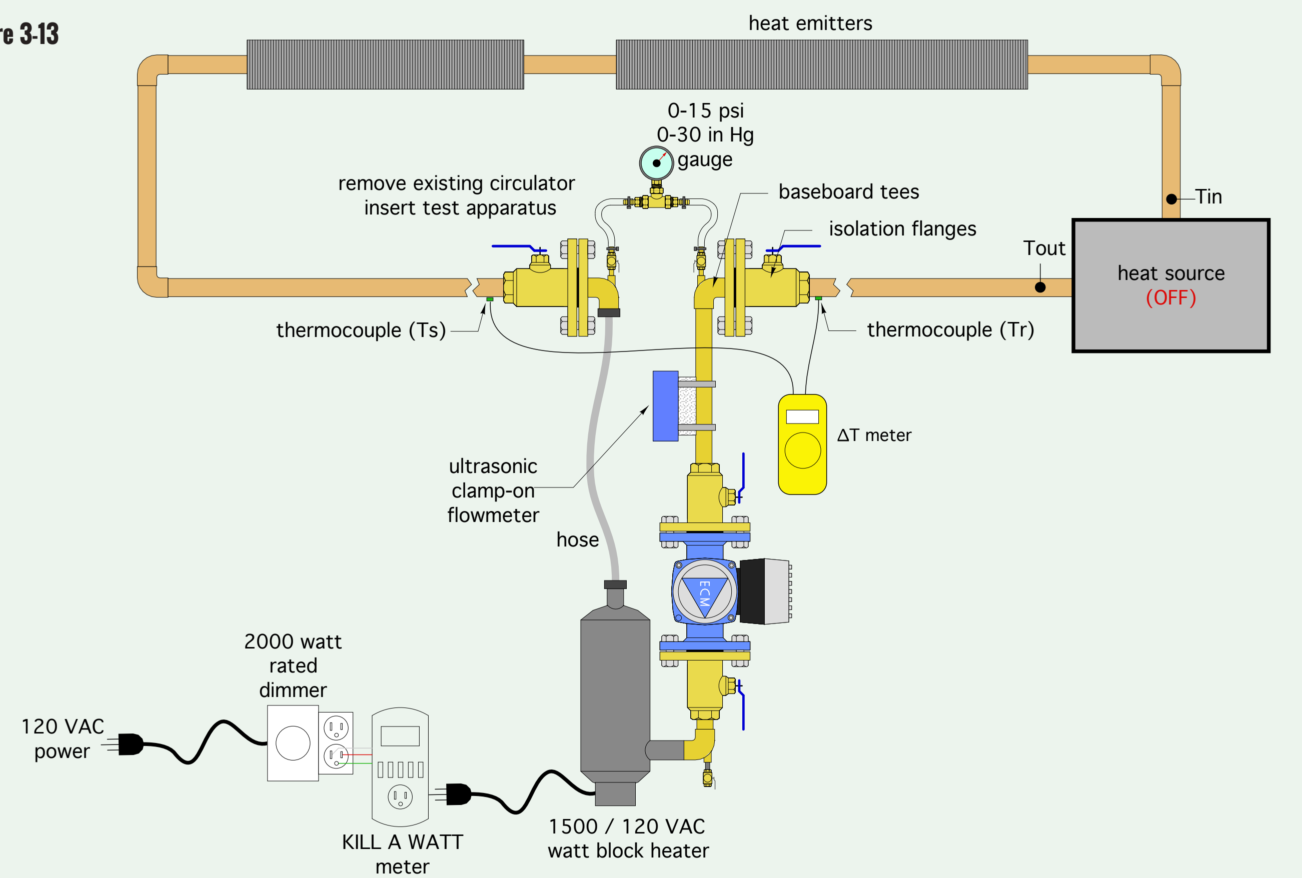


Figure 3-14

Figure 3-15

Figure 3-16

with a circulator and an electric block heater. The flanges of the test apparatus connect directly to the isolation flanges of the original circulator, which would be closed prior to removing the original circulator.

Ideally, the test apparatus is filled with the same fluid that's in the circuit before it is connected to the isolation flanges. Doing so minimizes air entry into the system during the test. If pre-filling is not possible due to the mounting orientation, a small amount of fluid can be added to the system using the feed water valve or glycol fluid feeder.

Each flange of the apparatus connects to a baseboard tee fitting. The small 1/8" FPT port on each baseboard tee can be connected to a pressure gauge using two small ball valves and small-diameter tubing.

In systems with relatively low static pressure at the location of the test apparatus, a pressure gauge rated for 0-15 psi should work. In systems with higher static pressure at the location of the test apparatus, a 0-30 psi gauge can be used.

The purpose of the gauge is to measure pressure drop around the circuit at different flow rates, which enables a circuit head loss curve to be developed. The latter can be used to evaluate how the circuit might perform using a different circulator, or a specific speed setting on a multi-speed circulator.

A procedure for operating this apparatus to determine the circuit head loss curve is given in Appendix C. If the differential pressure characteristics of the circuit are not needed, the 1/8" FPT ports on the baseboard tees can be closed with threaded plugs.

An electric block heater, such as the one shown in figure 3-14, is used to thermally "excite" the circuit.

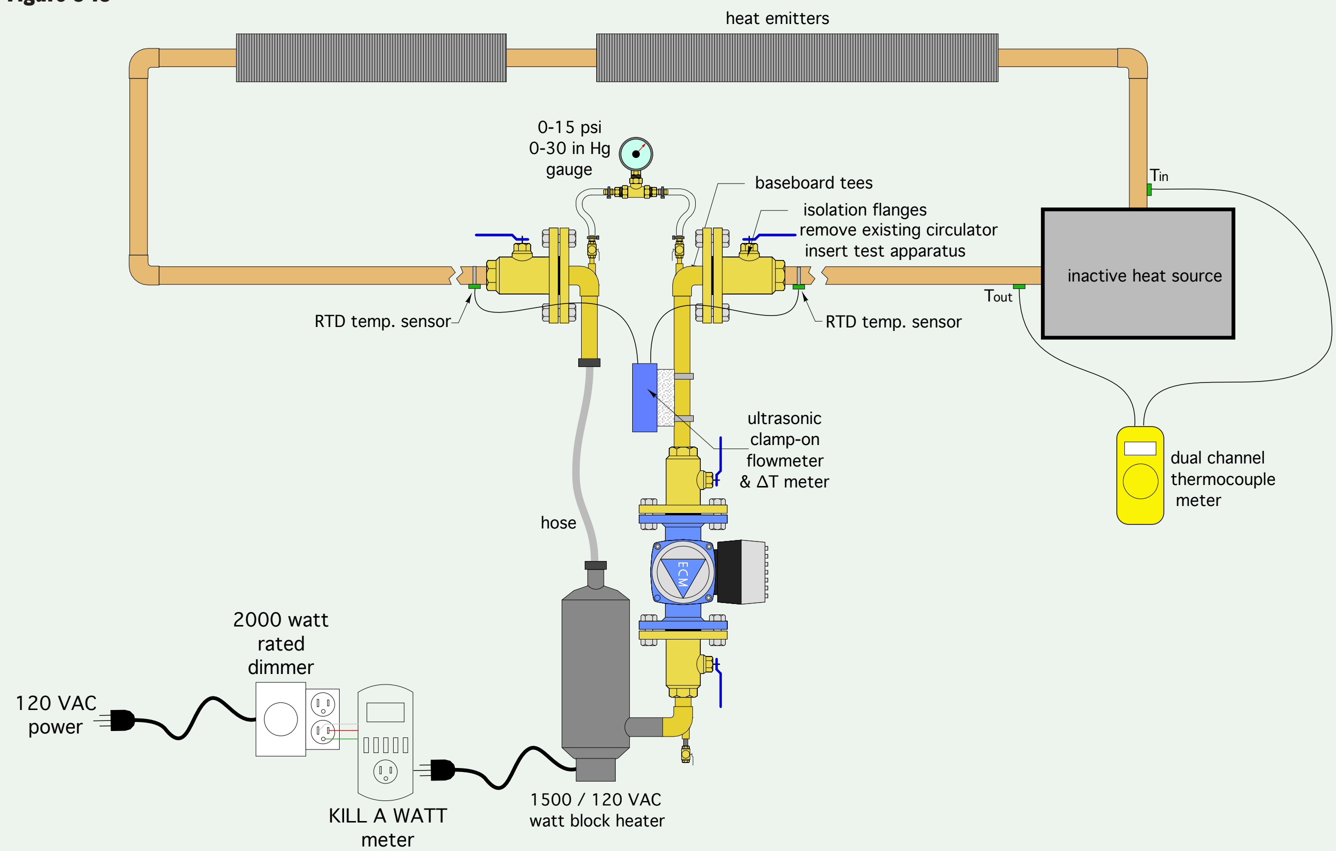
These block heaters are intended to preheat the coolant in vehicle engines during very cold weather. They consist of an electric heating element housed in a small vessel. Most are designed to be connected with hoses and have nominal heating capacity of 1,000 to 1,500 watts (3,413 to 5,120 Btu/hr).

Figure 3-17

Power input to the block heater can be controlled by a suitably rated dimmer switch, such as the 2,000 watt rated dimmer shown in figure 3-15.

Power input can be measured using a consumer-level electrical testing device such as the KILL A WATT meter shown in figure 3-16. Be sure that the watt meter used is rated to

Figure 3-18



handle the full wattage at which the block heater can operate.

A dual channel thermocouple meter, such as shown in figure 3-17, can be used to measure the temperature drop of the circuit.

Some ultrasonic flowmeters are also capable of monitoring two temperature sensors. As such they can simultaneously capture flow rate and differential temperature, making them capable of calculating the instantaneous rate of heat transfer. If this type of meter and its associated sensors are mounted on the test apparatus, and a dual channel thermocouple meter is also available, mount the thermocouples for the dual channel meter across the inactive heat source as shown in figure 3-18.

Figure 3-19 shows an assembled test apparatus based on the concept shown in figure 3-18.

THERMAL TEST PROCEDURE (FOR ZONE CIRCULATOR SYSTEMS)

The following procedure is for systems using zone circulators, such as those shown in figure 3-20.

Before installing the test apparatus, secure the ultrasonic flowmeter to one zone circuit, turn the existing heat source off, and *turn on all zone circulators.* This allows measurement of the flow rate in the zone being tested under hydraulic conditions that simulate design load (e.g., assuming all zone circulators are on under design load). With all zone circulators operating, there is maximum flow, and thus, maximum head loss through the common piping. This condition results in the lowest flow rate through the zone being tested.

If the zone circulators are hydraulically separated from each other, there will be a minimal drop in zone flow rate with all zone circulators operating. However, if the degree of hydraulic separation is marginal, there may be a noticeable drop in the zone flow rate with all zones operating relative to what the zone flow rate would be as the only operating zone. The most “conservative” condition for measuring flow rate is with all zone circuits operating.

After recording the flow rate in the zone being tested under the above conditions, move the ultrasonic flowmeter to another zone and repeat the process. Do this for each

Figure 3-19

zone in the system, recording the flow rate produced within each zone by its associated zone circulator.

Next, install the test apparatus in place of the existing circulator in one zone, and follow these steps.

1. Turn on the test circulator and adjust its speed while monitoring the flow rate through the ultrasonic flowmeter. Establish a flow rate approximately equal to the previously recorded flow for that zone.

2. Plug in and power up the electric block heater. Adjust the dimmer controlling power input to the block heater to 500 watts input, as indicated by the watt meter.

3. Allow the circuit to stabilize to steady state, then measure the supply and return fluid temperatures.

4. If necessary, adjust the power input to the block heater so that the supply fluid temperature is at least 20ºF above the room air temperature where the heat emitters are located.

5. Allow the circuit to stabilize to steady state whenever there is a change in power input.

6. Read and record the supply temperature, return temperature, zone room air temperature, and flow rate.

7. If the zone flow passes directly through the inactive heat source, use the procedure in the next subsection (ACCOUNTING FOR HEAT LOSS THROUGH AN INACTIVE HEAT SOURCE) to measure the temperature drop across that heat source.

8. Move the thermocouples and flowmeter to another zone and repeat this procedure for all zones in the system.

9. After gathering the temperatures, flows and power input for each zone, use formula 3-2 to convert the input power for each zone to Btu/hr.

Figure 3-20

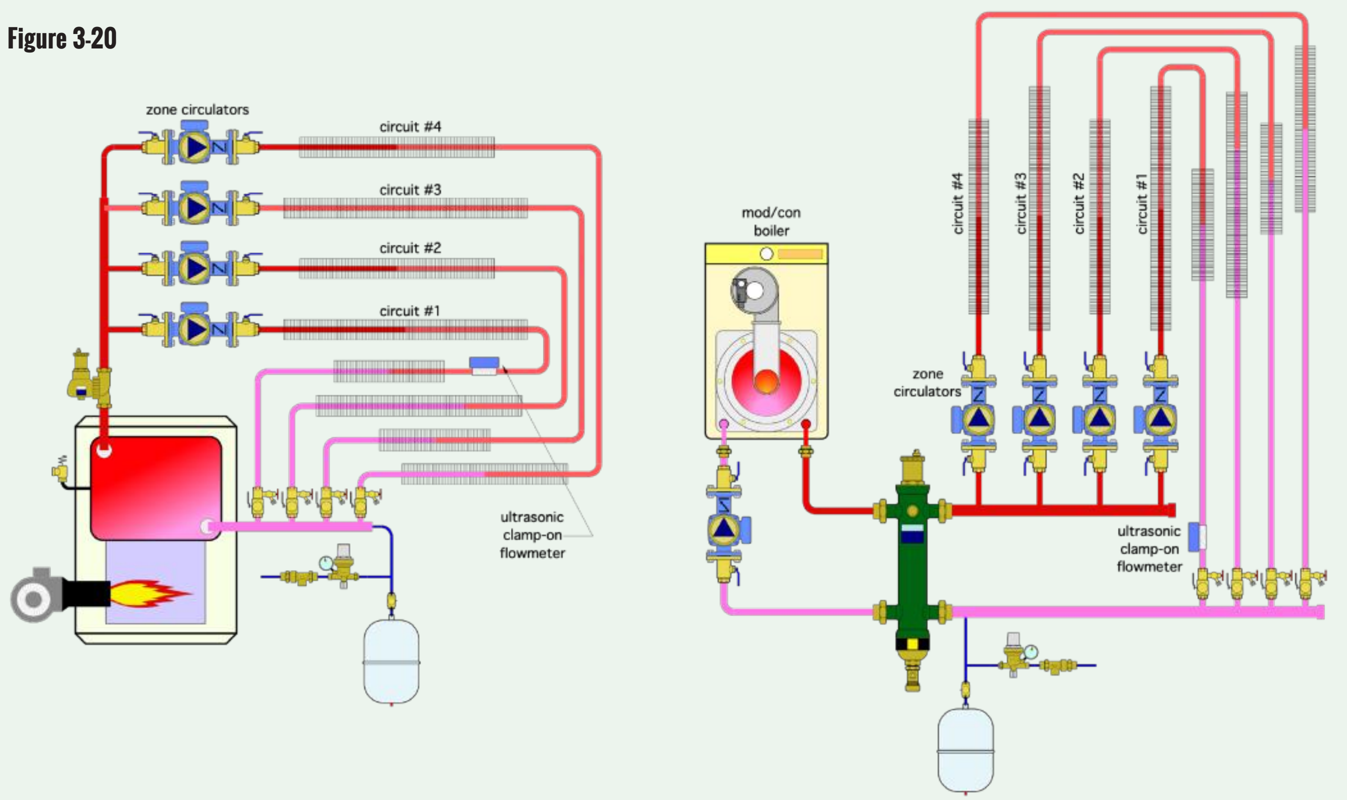
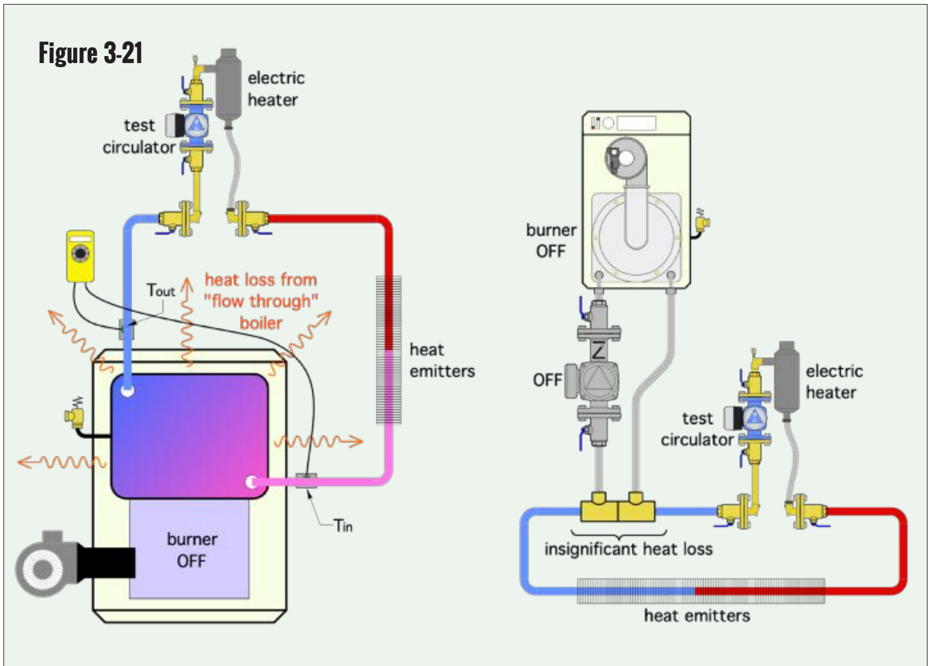


Figure 3-21



Formula 3-2:

$$\frac{Btu}{hr} = (wattage)3.413$$

Where:
Btu/hr = heat dissipation rate for zone (Btu/hr)
wattage = power input to the electric block heater at steady state (watts)
3.413 = conversion factor

ACCOUNTING FOR HEAT LOSS THROUGH AN INACTIVE HEAT SOURCE

If the electric block heater is being used to heat the circuit, *AND* the circuit *passes directly through* an inactive heat source, such as a cast iron boiler, there will be heat dissipation from that heat source. This heat loss would not be present under normal operating conditions when the boiler is operating and supplying heat to the circuit.

If the heat source is indirectly connected to the circuit, as shown in figure 3-21, there should be no flow through it when its dedicated circulator is off. In this case, the heat loss from the inactive heat source, although not zero, should be very small, and can be ignored.

When the circuit flow passes directly through the heat source, the temperature drop across that heat source should be measured using a dual channel thermocouple meter. This measurement should be taken as soon as possible after measuring the supply and return temperature of the circuit, and with the circuit operating at steady state conditions. Having an extra set of thermocouples for the dual channel meter, installed across the inactive boiler as shown in figure 3-18, allows for a rapid switch of sensors. After these measurements are taken, the heat loss across the inactive heat source can be calculated using formula 3-3:

Formula 3-3:

$$q_{hs} = 8.01Dc(f)(T_{in} - T_{out})$$

Where:
q_{hs}= heat loss of inactive heat source (Btu/hr)
D = density of the fluid at the average temperature of the circuit (lb/ft^3) (from figure 3-10)
c = specific heat of the fluid at the average temperature of the circuit (Btu/lb/ºF) (from figure 3-11)
f = flow rate through the circuit (gpm)
T_{in} = fluid temperature entering the inactive heat source (ºF)
T_{out} = fluid temperature leaving the inactive heat source (ºF)
8.01 = a constant based on the units in the formula

The values for density (D), specific heat (c), and flow rate (f) are the same as those used in formula 3-1. So is the flow rate. The only difference between formulas 3-3 and 3-1 are the temperatures.

The "net" heat output of the circuit would be the value from formula 3-1

Figure 3-22a

	existing flow rate w/ all zones on (gpm)	flow rate in zone during test (gpm)	power input to electric heater during test (watts)	Heat input to zone during test. (watts x 3.413)	SUPPLY temp. to zone during test (°F)	RETURN temp. from zone during test (°F)	AIR TEMP. surround zone circuit (°F)	inactive heat source inlet temp. (°F)	inactive heat source output temp. (°F)
zone 1	4.2	4.1	800	2730	95.0	82.6	68.1	82.8	82.1
zone 2	3.5	3.6	750	2560	99.1	80.2	70.6	80.4	80.3
zone 3	2.8	2.8	550	1877	103.3	85.1	70.1	85.3	85.1
zone 4	8.3	8.2	1020	3481	105.6	86.0	72.4	86.2	86.1

minus the value from formula 3-3. This can be expressed as formula 3-4.

Formula 3-4:

$$q_{cnet} = (8.01cD)f\left[(\Delta T_{circuit}) - (\Delta T_{ihs})\right]$$

Where:
q_{cnet} = net rate of heat output from circuit (Btu/hr)
8.01 = a unit constant
c = fluid specific heat (Btu/lb/ºF)
D = fluid density (lb/ft^3)
f= flow rate in circuit (gpm)
$\Delta T_{circuit}$ = the temperature difference across the test flanges (T_S - T_R) (ºF)
ΔT_{ihs} = the temperature drop across the inactive heat source (T_{in}-T_{out}) (ºF)

A spreadsheet table such as shown in figure 3-22a can be used to keep the test data organized.

The system fluid is water.

The **green** cells in figure 3-22a are all measurements taken during the testing.

Once this data is collected, additional columns can be added to process the data using formula 3-4. This formula requires a value for the fluid density (D) and fluid specific heat (c) at the average temperature of each circuit. These can be found using figures 3-10 and 3-11.

Figure 3-22b is an extension of the spreadsheet in figure 3-22a. It shows the values for average water temperature, as well as the density and specific heat of water at the average water temperature in each circuit. The purple column on the far right is the value for net heat output of each circuit, calculated using formula 3-4.

Notice that the density and specific heat of water have the same values for all zones. This is the result of the accuracy of estimating these values from figures 3-10 and 3-11, as well as the small spread of the average circuit temperatures and the inherent characteristics of water. Antifreeze solutions, especially in systems that have a wider spread in average circuit temperatures, would likely reveal changes in density and/or specific heat from one zone to another.

Figure 3-22b

	AVERAGE fluid temp. in zone during test (ºF)	DENSITY of fluid at average circuit temp (lb/ft³)	SPECIFIC HEAT of fluid at average circuit temp (Btu/lb/ºF)	net heat output of zone (Btu/hr), formula 3-4
zone 1	82.45	62.0	1.00	23,823
zone 2	80.35	62.0	1.00	33,611
zone 3	85.2	62.0	1.00	25,030
zone 4	86.15	62.0	1.00	79,410

THERMAL TEST PROCEDURE (FOR ZONE VALVE SYSTEMS)

It's also possible to estimate the thermal performance of systems that use zone valves using a procedure like that used for systems with zone circulators.

In systems with zone valves, the flow rate through individual zones can change whenever a different zone turns on or off. The change in flow rate for a given zone as other zones turn on and off depends on the flow resistance of the common piping (e.g., the supply header, return header, and heat source). It also depends on the ability of the system to minimize changes in differential pressure across the circulator as zones turn on and off.

An "ideal" zone valve distribution system would have very low hydraulic resistance through the common piping and a variable-speed circulator that can maintain constant differential pressure regardless of the number of zones operating.

Before installing the test apparatus, secure the ultrasonic flowmeter to one zone circuit, turn the existing heat source off, *and manually open all zone valves*. Turn on the existing circulator. If a variable-speed circulator is used, set it to operate at maximum speed. This allows the flow rate in the zone being tested to be very close to its value under design load conditions (e.g., assuming that all zones are on, and that a variable-speed circulator — if present — is at full speed under design load). When all zone valves are open, there is maximum head loss through the common piping and the lowest operating flow rate through the zone being tested.

After recording the flow rate in the zone being tested, and under the above conditions, move the ultrasonic flowmeter to another zone and repeat the process.

Do this for each zone in the system, recording the flow rate produced in each zone by its associated zone circulator.

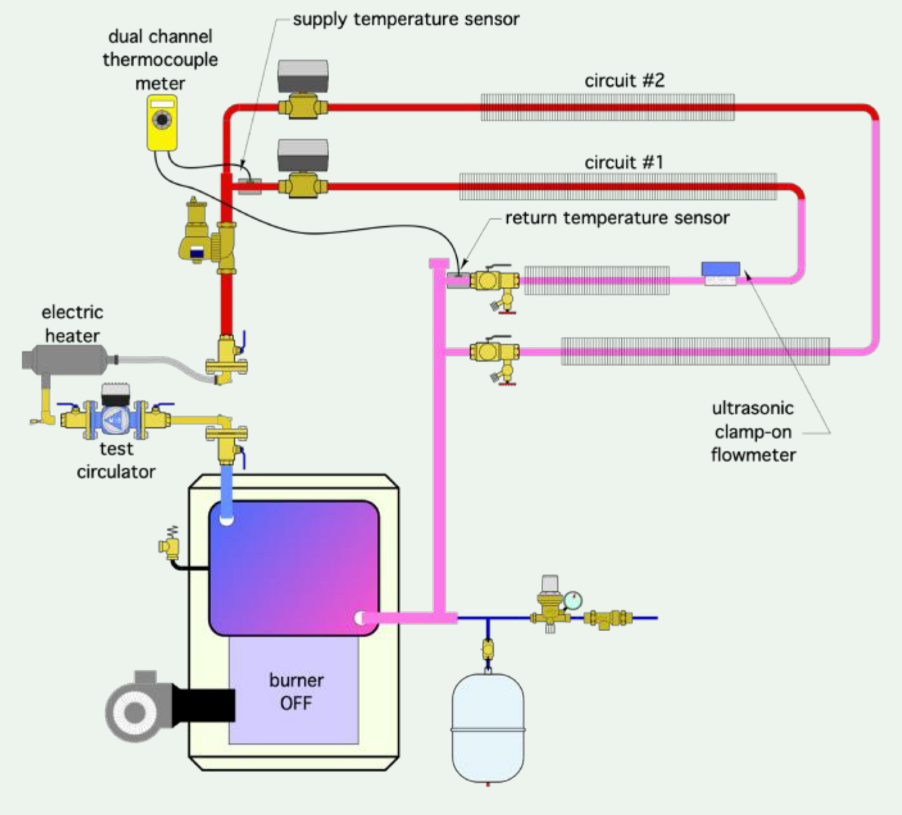
At this point, the nominal flow rate that each zone would operate at under design load conditions has been established.

Next, install the test apparatus in place of the existing circulator as shown in figure 3-23, and follow these steps.

1. Manually open *ALL* zone valves.

2. Attach the ultrasonic flowmeter to the piping in one zone.

Figure 3-23



3. Turn on the test circulator and adjust its speed while monitoring the flow rate in the zone. Establish a flow rate approximately equal to the previously recorded flow rate for that zone.

4. Plug in and power up the electric block heater. Adjust the dimmer switch controlling power to the block heater to 500 watts input, as indicated by the watt meter.

5. Allow the circuit to stabilize to a steady state, then measure the supply and return fluid temperatures.

6. If necessary, adjust the power input to the block heater so that the supply fluid temperature is at least 20ºF above the room air temperature where the heat emitters are located.

7. Allow the circuit to stabilize to steady state conditions after making any changes to the circuit's power input.

8. Read and record the supply temperature, return temperature, zone air temperature, and flow rate.

9. If the zone flow passes directly through the inactive heat source, use the procedure in the previous subsection (ACCOUNTING FOR HEAT LOSS ACROSS AN INACTIVE HEAT SOURCE) to measure the temperature drop across that heat source.

10. Move the thermocouples and flowmeter to another zone and repeat this procedure for all zones in the system.

11. Use formula 3-2 to convert the input wattage for each zone to Btu/hr.

Formula 3-2:

$$\frac{Btu}{hr} = (wattage)3.413$$

Where:
Btu/hr = heat dissipation rate for zone (Btu/hr)
wattage = power input to the electric block heater at steady state (watts)
3.413 = conversion factor

12. Set up tables like those in figure 3-22a and 3-22b to keep the measurements and subsequent calculations organized.

ESTIMATING CIRCUIT PERFORMANCE AT OTHER TEMPERATURES

At this point, the circuit testing and subsequent calculations have produced a data point (net heat output (q_{net})) and the associated fluid supply temperature at which that heat output occurred.

This information can be used to "calibrate" a simplified relationship between the supply water temperature to a circuit and its heat output. That relationship is given by formula 3-5.

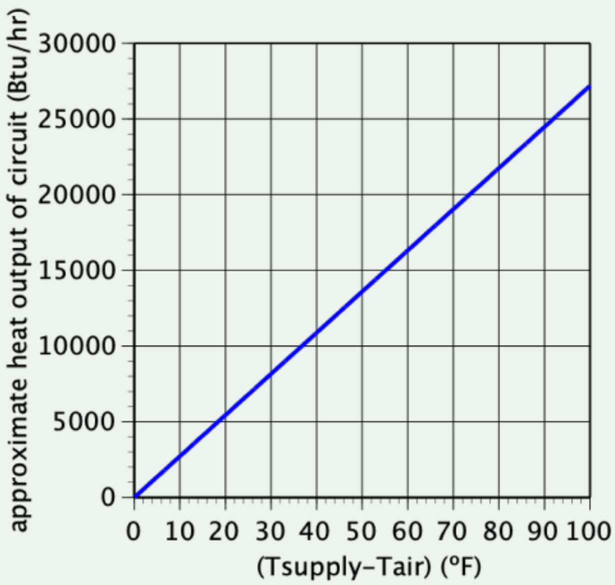
Formula 3-5:

$$q_{circuit} \approx k(T_s - T_{air})$$

Where:
$q_{circuit}$ = estimated heat output of the circuit (Btu/hr)
k = a constant to be determined for the circuit (Btu/hr/ºF)
T_s = supply water temperature to the circuit (ºF)
T_{air} = air temperature in space where the circuit is located (ºF)

Put in words, formula 3-5 states that the heat output of a circuit is approximately proportional to the difference between the supply water temperature and the air temperature surrounding the circuit.

Figure 3-24



There are some simplifying assumptions and approximations in formula 3-5.

First, the formula “linearizes” some effects that are primarily driven by natural convection heat transfer. These effects are more pronounced for fin-tube baseboard and panel radiators, both of which transfer some of their heat output by natural convection. The linearization has minimal effect on heat emitters such as radiant floor or ceiling panels, fan-coils or air handlers.

Second, most hydronic circuits have piping in both conditioned (e.g., intentionally heated) spaces, as well as unconditioned (e.g., unintentionally heated) spaces. Thus, some of the piping will not be losing heat to surrounding air at the same temperature as the air in conditioned space. Formula 3-5 is based on a single air temperature. Circuits with minimal amounts of piping in unconditioned space, or well-insulated piping in unconditioned space, will be well modeled by formula 3-5. Many current energy codes require minimum insulation levels on all piping carrying heated fluids through unconditioned spaces.

These assumptions and approximations are generally acceptable based on the intended purpose of the circuit testing and the accuracy of the measurements from the testing process.

Based on the tests described earlier, there is a known value for the circuit’s net heat output (qnet) and the supply fluid temperature (Ts) at which this output occurred. This can be used to find the value of (k) in formula 3-5.

Example: Based on testing, a hydronic heating circuit has a net heat output (q_{net}) of 15,500 Btu/hr when supplied with fluid at a temperature of 125ºF. Use formula 3-5 to find the value of k for this circuit, assuming that the air temperature surrounding the majority of the heat emitters and piping is 68ºF.

Solution: Rearrange formula 3-5 and insert the known information to find the value of k.

$$k = \frac{q_{net}}{\left(T_s - T_{air}\right)} = \frac{15{,}500}{\left(125-68\right)} = 271.9\frac{Btu}{hr \bullet {}^\circ F}$$

The value of (k) can be interpreted as follows: This specific circuit releases approximately 272 Btu/hr for each ºF its supply fluid temperature is above the air temperature surrounding the circuit.

After the value of (k) has been determined, it’s possible to estimate the heat output of the circuit at other fluid supply temperatures and other air temperatures. This can be done analytically using the value of (k) in formula 3-5, along with the fluid supply temperature and the desired surrounding air temperature.

Example: Assuming the (k) value of 271.9 Btu/hr/ºF from the previous example, estimate the heat output of the same circuit when supplied with fluid at 105ºF, and within a space where the air temperature surrounding the heat emitters and piping is 70ºF.

Solution: Putting the information into formula 3-5 and calculating yields:

$$q_{net} = k\left(T_s - T_{air}\right) = 271.9\left(105-70\right) = 9{,}517\frac{Btu}{hr}$$

It’s also possible to make a graph of formula 3-5 using the value of (k) for the specific circuit, as shown in figure 3-24.

Example: Zone circuit #2 in the hydronic system shown in figure 3-25 was tested using the methods described in this section.

The circuit achieved a steady state condition when the power supplied to the electric block heater was 950 watts. At this point, the supply temperature to the circuit was 104ºF, and the return temperature was 96ºF. The air temperature surrounding most of the circuit during the test was 68ºF. The ultrasonic flowmeter attached to the circuit indicated 3.7 gpm flow. The circuit operated with a 30%

solution of propylene glycol antifreeze. The circuit passed through an inactive boiler, across which the temperature drop was 0.5ºF. Determine:

a. The net heat output of the circuit including that from the inactive boiler.
b. The approximate heat output of the circuit if it operated with a supply temperature of 120ºF, and a surrounding air temperature of 70ºF.

Solution: Start by finding the average fluid temperature in the circuit:

$$T_{ave} = \frac{T_{supply} + T_{return}}{2} = \frac{104 + 96}{2} = 100^{\circ} F$$

Using figure 3-10 to find the density of the 30% propylene glycol solution at 100ºF.

D = 63.7 lb/ft^3

Use figure 3-11 to find the specific heat of the 30% propylene glycol solution at 100ºF.

c = 0.93 Btu/lb/ºF

The temperature drop around the circuit is: $\Delta T_{circuit}$ = 104.0 - 96.0 = 8.0ºF

The temperature drop across the inactive heat source is: ΔT_{ihs} = 0.5ºF

Putting these values along with the measured flow rate into formula 3-4 yields:

$$q_{cnet} = (8.01cD)f\left[(\Delta T_{circuit}) - (\Delta T_{ihs})\right] = (8.01)(0.93)(63.7))(3.7)\left[(8.0) - (0.5)\right] = 13{,}168 \frac{Btu}{hr}$$

The value of (k) for this circuit is found using formula 3-5:

$$k = \frac{q_{net}}{(T_s - T_{air})} = \frac{13168}{(104 - 68)} = 365.8 \frac{Btu}{hr \bullet^{\circ} F}$$

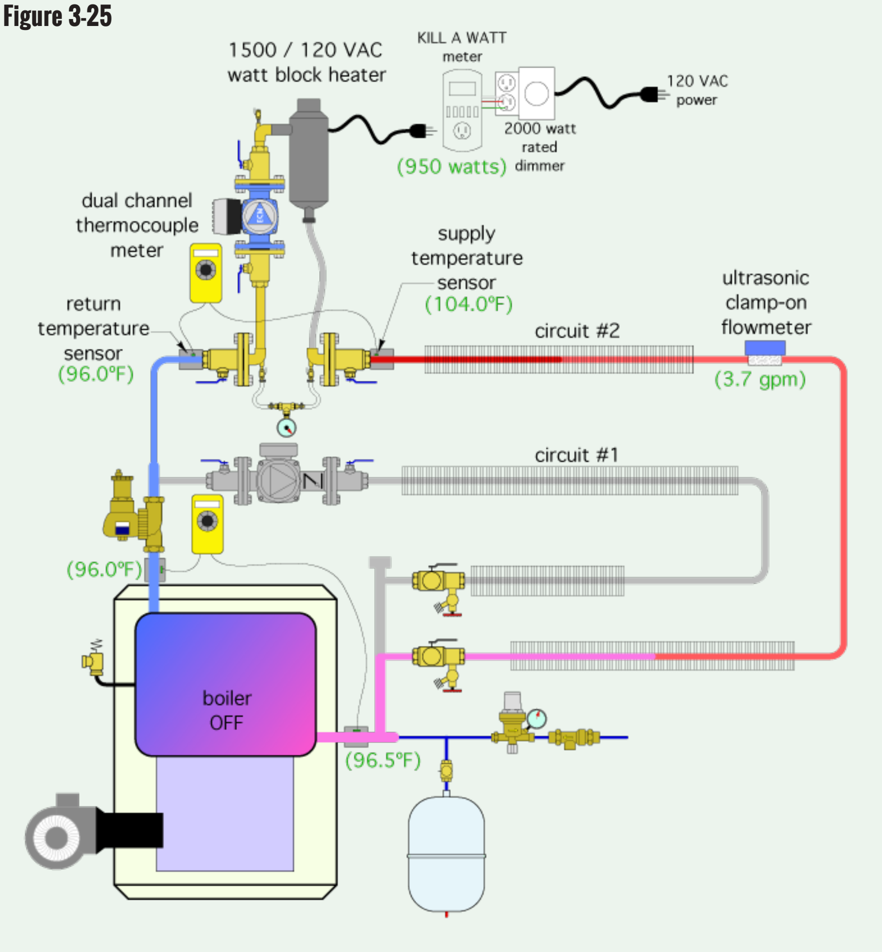


Figure 3-25

Formula 3-5 can now be used to estimate the circuit's heat output at the new stated conditions:

$$q_{net} = k(T_s - T_{air}) = 365.8(120 - 70) = 18{,}290 \frac{Btu}{hr}$$

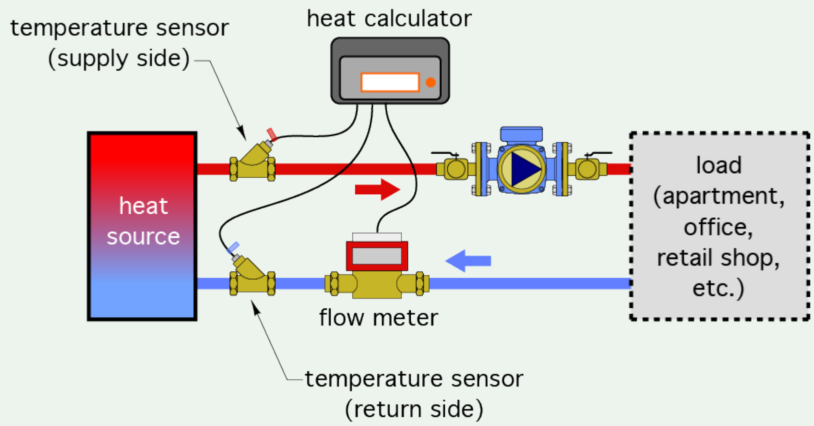
The techniques discussed in this section represent a minimally invasive approach for physically testing existing hydronic heating circuits to estimate their performance over a range of fluid supply temperatures and surrounding air temperatures.

The objective in getting this information is to estimate the performance of the existing circuit at conditions other than those at which it currently operates, such as when a new heat source will be used and the circuit operated at a reduced supply water temperature.

Consider the following scenario: An existing circuit of fin-tube baseboard operates at an average supply water temperature of 170ºF. A designer is considering adding an air-to-water heat pump to the system to either replace the boiler or act as a primary heat source, with the boiler serving as a supplemental and backup heat source. The designer wants to estimate the output of the existing circuit at a supply temperature of 120ºF. They know that the heat output will be lower but not how much lower. Based on the methods described, the designer can determine what modifications to the circuit or what changes to the building might be necessary to allow the circuit to maintain comfort in the space at the lower fluid supply temperature.

For more information on reducing water temperature in existing hydronic heating systems, see idronics #25.

Figure 3-26



MEASURING ENERGY TRANSFER OVER TIME

While this section has focused on how to measure the *rate* of heat output from hydronic circuits, some situations require ongoing measurement of total heat transferred over time. That information could then be the basis for invoicing for thermal energy usage.

Figure 3-26 shows an energy metering assembly that can be installed to measure both instantaneous rate of heat transfer, as well as the total amount of heat transferred over time.

The two temperature sensors and the flowmeter provide information to a heat calculator that can determine the instantaneous rate of heat transfer and the total amount of heat that has passed from the heat source to the load over time.

For more information on Fundamentals of Heat Metering in Hydronic Systems, see idronics #24.

4. OTHER CONSIDERATIONS WHEN EVALUATING EXISTING SYSTEMS

In addition to thermal performance estimating, several other factors need to be considered when evaluating the suitability of bringing a new heat source into an existing hydronic heating system. This section discusses those factors.

SEASONAL PERFORMANCE ESTIMATES

Modern heat sources, such as modulating/condensing boilers, air-to-water heat pumps and geothermal water-to-water heat pumps, deliver higher thermal efficiencies when operated at lower water temperatures.

If the required supply water temperature of existing zone circuits is relatively high, such as 160 ºF at design load, it's unlikely that most current-generation air-to-water or water-to-water heat pumps can supply the system at design load conditions. Heat pumps using R290 (propane) or CO_2 (carbon dioxide) refrigerants are capable of higher temperature operation, but at present have limited availability in North America.

However, even distribution systems that require high water temperature at design load can be served by air-to-water or geothermal water-to-water heat pumps. Those heat pumps can provide much of the space heating energy required under *partial load conditions*, with an existing (or new) boiler providing the higher temperature water required at or near design load conditions.

Simulations of modern "cold climate" air-to-water heat pumps have shown that they can supply upwards of 90 percent of the total seasonal energy required for space heating, even when the existing distribution system requires 180ºF water at design load conditions.

Figure 4-1 shows the results of such simulations for a house with a design space heating load of 36,000 Btu/hr in a location with an outdoor design temperature of -9ºF. The required indoor temperature at design load is 70ºF.

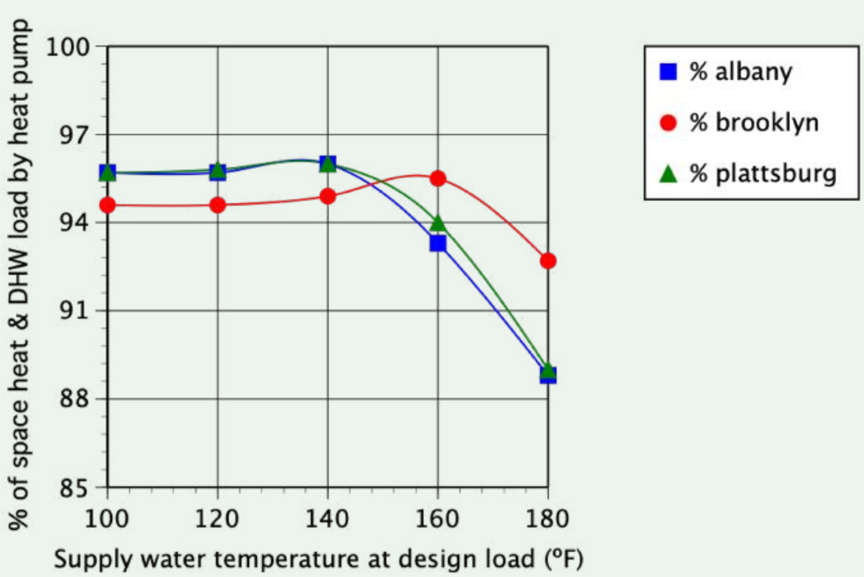
The simulations were performed for three different locations in New York State (Plattsburg, Albany, and Brooklyn). These locations represent a range of climates from approximately 4,900 ºF•days in Brooklyn, to just over 8,000 ºF•days in Plattsburg. The air-to-water heat pump modeled in the simulations had an enhanced vapor injection refrigeration system, and a nominal capacity of 4 tons. The simulation also included a year-round domestic water heating load of 60 gallons per day, heated from 50ºF to 120ºF.

The graph in figure 4-1 shows the percentage of the space heating + domestic water heating load supplied by the air-to-water heat pump as a function of the supply water temperature required at design load conditions.

The simulation model was based on the use of outdoor reset control of the water temperature supplied to the hydronic distribution system. *This well-established control technique is crucial in leveraging the ability of the heat pump under partial load conditions.* It allows the water temperature leaving the heat pump to be just high enough to meet the heating load of the building, with no "excess" temperature that would reduce the efficiency of the heat pump. The reset schedule was also adjusted to account for modest internal heat gains. This helps improve the accuracy of the simulation, especially in low-energy houses.

The results show that the space heating contribution by the heat pump is relatively insensitive for systems that have *design load* supply water temperatures under 140ºF. This contribution decreases for systems that require higher water temperatures at design load. *However, even systems that require 180ºF water at design load can have the majority of their seasonal space heating energy supplied by the heat pump.*

Figure 4-1



LOWERING WATER TEMPERATURE REQUIREMENTS THROUGH LOAD REDUCTION

Since the thermal efficiency of modern hydronic heat sources improves with decreasing supply water temperature, any measures that decrease the required supply water temperatures of existing systems, while still maintaining building comfort, will boost efficiency and decrease operating costs.

One way to reduce the required supply water temperature of an existing hydronic distribution system, or a zone within that system, is to reduce the heating load through building weatherization. Measures such as improved insulation, reduced air leakage or window upgrades can reduce heating (and cooling) loads. Such measures also reduce the total space-heating energy required by a building during every subsequent year. In many cases, they are 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.

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 4-1:

Formula 4-1:

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

Where:

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

T_{in} = desired indoor air temperature (ºF)

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) (ºF)

Example: Consider a building with an existing design heating load of 100,000 Btu/hr, based on maintaining an interior temperature of 70ºF. The existing hydronic distribution system uses standard fin-tube baseboard and requires a supply water temperature of 180ºF 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:

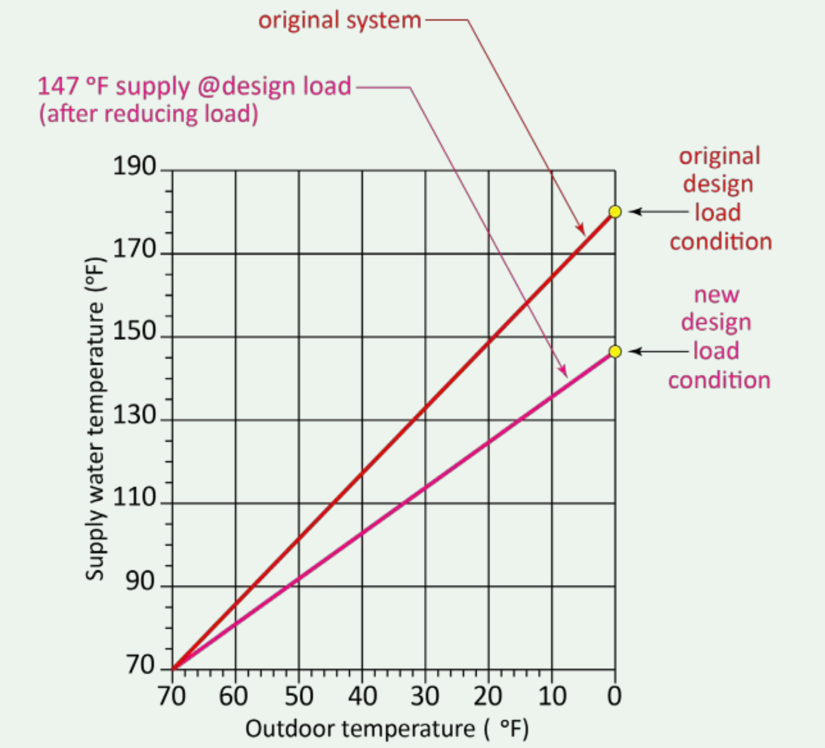
$$T_{new} = 70 + \left(\frac{70{,}000}{100{,}000} \right) \times (180 - 70) = 147^{\circ} F$$

Figure 4-2 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 180ºF to 147ºF. Although this is certainly an improvement, it's still above what some renewable energy heat sources can *consistently* provide. For example, most current generation air-to-water or geothermal water-to-water heat pumps cannot heat water to 147ºF.

However, it's important to remember that the 147ºF 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

Figure 4-2

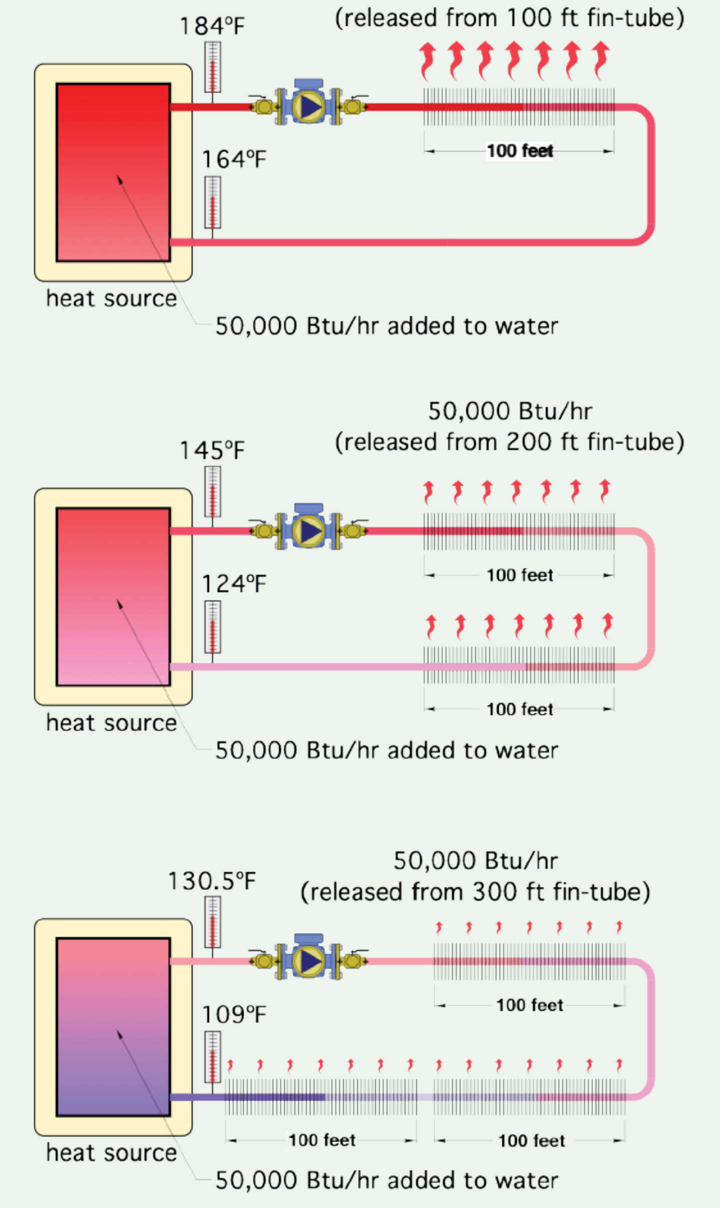


the sloping lines in figure 4-2. For example, when the outdoor temperature is 40ºF, the required supply water temperature indicated by the lower sloping line in figure 4-2 is about 104ºF. This is easily within the operating range of current generation hydronic heat pumps.

LOWERING WATER TEMPERATURE REQUIREMENTS BY ADDING HEAT EMITTERS

Another way to reduce the water temperature in a hydronic system (or zone) is by adding one or more heat emitters to a zone. *Any heat emitter that increases the total surface area separating the heating water from the room air will reduce the required supply temperature, while maintaining a given rate of heat output.* This concept, along with representative numbers, is illustrated in figure 4-3.

Figure 4-3



The type of heat emitter that's added to the system does not necessarily have to be the same as the existing heat emitters. For example, one or more panel radiators could be added to a circuit with existing fin-tube baseboard. A length of fin-tube or a fan-coil could be added to a system with cast iron radiators. An area of lower temperature radiant floor, wall or ceiling heating could be added to an existing system originally designed to operate at a relatively high temperature, but doing so would require a mixing device to reduce water temperature to the radiant panel.

When a decision is made to add heat emitters, the "topology" of the existing system can also be modified. A long circuit containing several series-connected heat emitters can be split into two or more parallel circuits. This concept is shown in figure 4-4.

In this example, the original series circuit of the baseboard was cut into four segments. Those segments were then reconnected to a manifold station using 1/2" PEX or PEX-AL-PEX tubing. These flexible tube options are easier to install and less expensive than reconnecting with rigid copper tubing. Fittings for transitioning between copper tubing and either PEX or PEX-AL-PEX tubing are readily available.

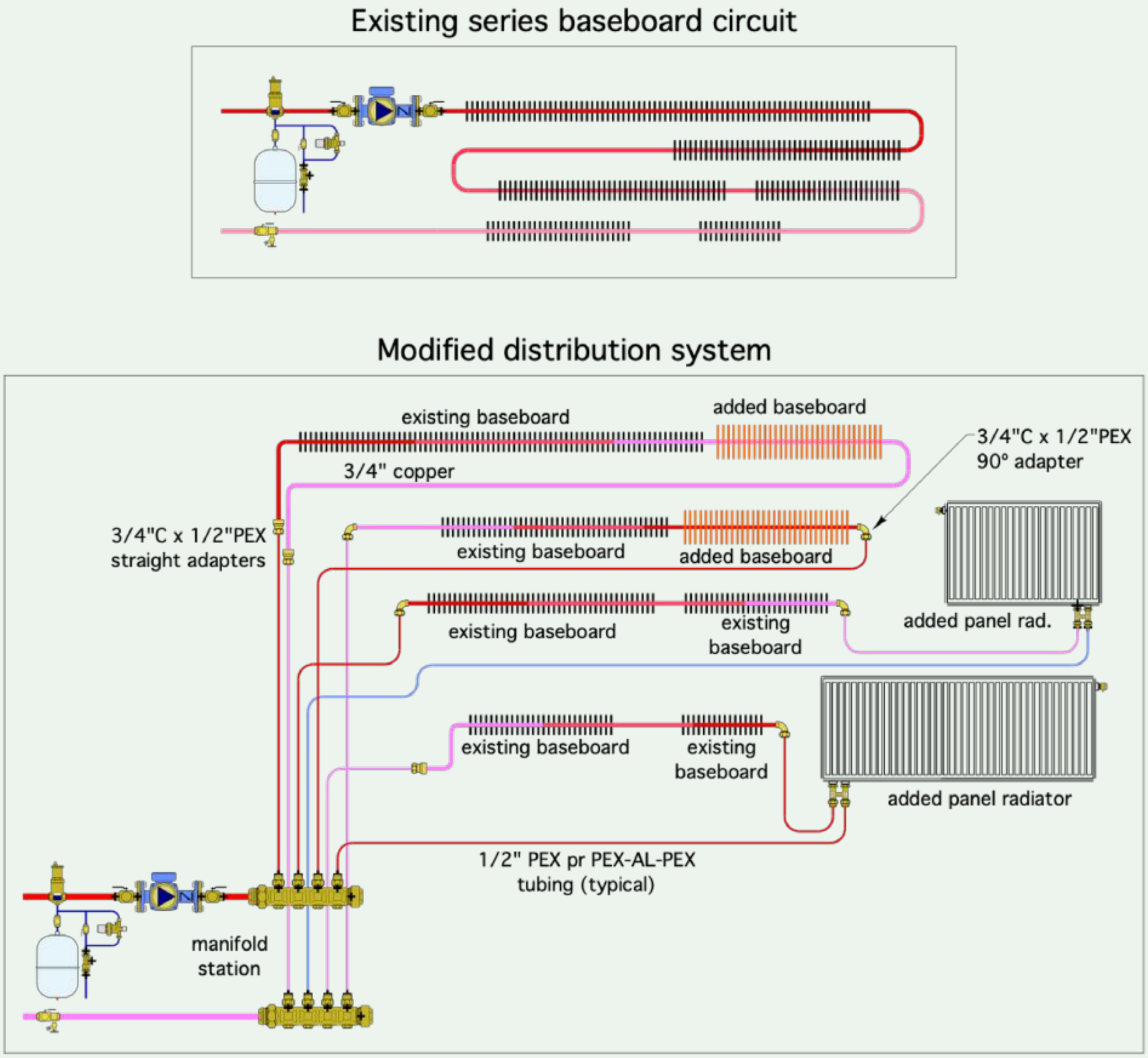
Dividing the original series circuit into branches also opens the possibility of zoning. Individual zones can be controlled using non-electric thermostatic radiator valves or manifold valve actuators. Figure 4-5 shows an example of the latter.

This system uses manifold valve actuators operated by zone thermostats. All wiring is routed through a Caleffi Z-one™ relay center. A variable speed pressure-regulated circulator set for constant differential pressure mode automatically adjusts its speed as different zones turn on and off.

DEALING WITH "OUTLIER" ZONES

In a multi-zone system, there's a possibility that one or more of the zones requires a fluid supply temperature significantly higher than the other zones. The methods of analysis presented in section 3 allows such zones to be identified.

Figure 4-4



Consider a four-zone system where the supply fluid temperatures at design load are as follows:

Zone 1: supply fluid temperature at design load = 123ºF
Zone 2: supply fluid temperature at design load = 165ºF
Zone 3: supply fluid temperature at design load = 118ºF
Zone 4: supply fluid temperature at design load = 120ºF

The average supply fluid temperature for zones 1, 3, and 4 is 120.3ºF. The maximum deviation of any of these temperatures from this average is 2.7ºF. Given this limited "spread" of required supply temperatures, it's likely that these three zones could be served from a common source device, at the same temperature, without creating detectable differences in comfort in their respective spaces. However, the supply temperature for zone 2 is well above this average. If it were supplied at, say, 121ºF, it's unlikely that comfort could be maintained under design load conditions.

Upon seeing this "outlier" zone, the designer could inform their client about the situation and make recommendations for correcting the problem through weatherization of the building's thermal envelope associated with that zone, or by adding heat emitters, or a combination of these modifications.

Figure 4-5

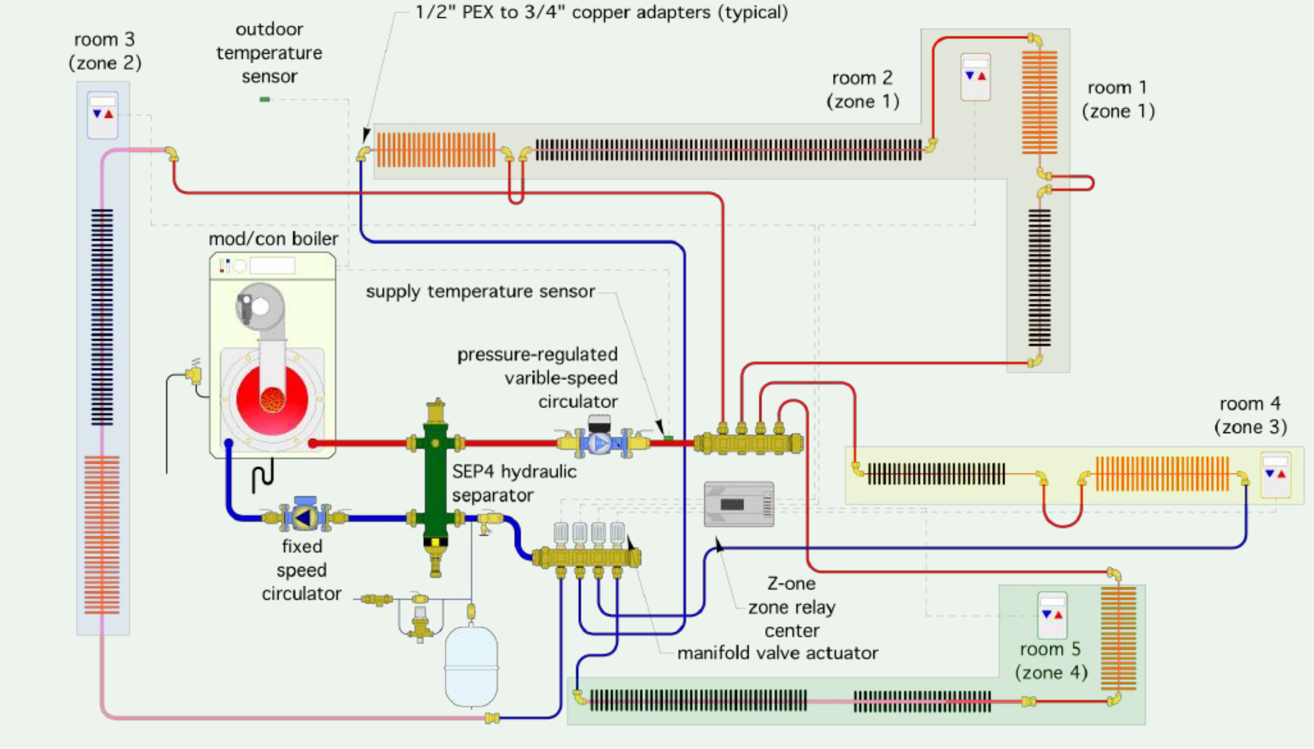


Figure 4-6

Courtesy of Harvey Youker

Identifying these issues prior to installing a new or alternative hydronic heat source and advising clients of the consequences and possible corrective actions is professional. It helps ensure satisfaction with the modernized system. More than a new heat source would be required for the system shown in Figure 4-6.

PROTECTING CONVENTIONAL BOILERS FROM SUSTAINED FLUE GAS CONDENSATION

When a modern heat source such as an air-to-water heat pump is combined with an existing conventional boiler, and the water temperature in the system is controlled based on outdoor reset, there will be many hours of operation where the water temperature on the return side of the distribution system is well below the dewpoint temperature of the boiler's flue gasses. If this water returns directly to a conventional boiler, that boiler and its venting system can be seriously damaged by corrosion.

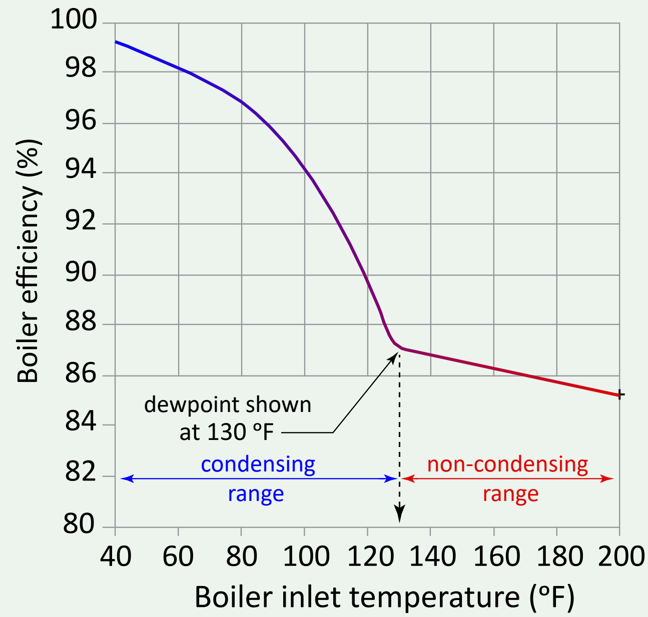
The galvanized steel vent connector piping shown in figure 4-7 was corroded to the point of failure due to sustained flue gas condensation. That failure occurred after only 6 months of boiler operation.

This type of failure could result in the release of toxic flue gasses, such as carbon monoxide, into occupied spaces and must be avoided.

Although the exact boiler inlet temperature that can create sustained flue gas condensation varies with the fuel and fuel/air ratio, a

Figure 4-7

Figure 4-8



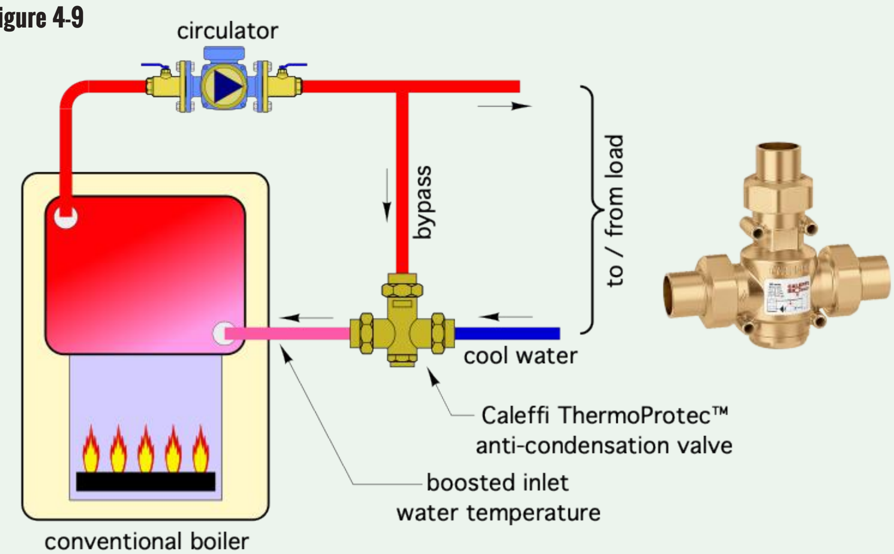
generally accepted guideline is to maintain the inlet water temperature to a boiler burning natural gas at or above 130ºF.

Figure 4-8 shows how a conventional boiler transitions from "non-condensing" to "condensing" mode operation as the boiler's inlet water temperature decreases.

There are several ways of protecting a conventional boiler against sustained flue gas condensation. They involve different hardware and control methods. One of the simplest approaches is to install a high-flow-capacity thermostatic mixing valve such as the Caleffi ThermProtec™, as shown in figure 4-9.

When the water temperature leaving the ThermoProtec valve is below the calibrated temperature of its sensing element, the valve's cold port is closed. This blocks any flow into or out of the load. Heated water leaving the boiler recirculates through the valve and back into the boiler. The full heat output of the boiler is temporarily focused on heating the water in this recirculating piping loop, allowing the water temperature to quickly rise above the dewpoint of the flue gasses. As this temperature rise progresses, the cold port of the ThermoProtec valve begins to open, and

Figure 4-9



the valve's bypass port begins to close. This allows flow to develop between the boiler and the load, while keeping the boiler operating in a "non-condensing" mode.

EVALUATING WATER QUALITY IN EXISTING SYSTEMS:

Older hydronic systems, especially those converted from steam to hot water, often contain sludge formed by oxidized iron. This sludge contains very fine particles that can become lodged between moving parts in valves and circulators, which impedes or totally stops the required motion. Figure 4-10 shows an accumulation of iron oxide particles on the permanent magnet rotor of a modern circulator. Figure 4-11 shows debris and scale that have jammed within a valve.

Figure 4-10

Figure 4-11

Dirt and magnetic particles can be removed using a magnetic dirt separator, such as the Caleffi XF "extra filtration" separator shown in figure 4-12.

The dirt separator should be installed upstream of both the existing and new heat source, as well as upstream of the circulators serving those heat sources.

The "chemistry" of the water in the existing system should also be considered when adding a new heat source. A sample of the water should be drawn and sent to a company specializing in water treatment for hydronic systems. There it can be analyzed, and a water treatment plan developed based on the results of that analysis.

The water chemistry of systems that have been operating with an antifreeze solution should also be checked, especially if those systems have operated at higher temperatures with minimal (or no) maintenance of the antifreeze solution. Over time, glycol-based antifreeze becomes acidic, which can accelerate certain forms of internal corrosion. If the fluid sample has a dark color or pungent odor, the system should be drained and internally cleaned with a hydronic detergent/descaling solution. After cleaning, draining and rinsing, the ideal refill fluid would be a mixture of demineralized water mixed with new antifreeze.

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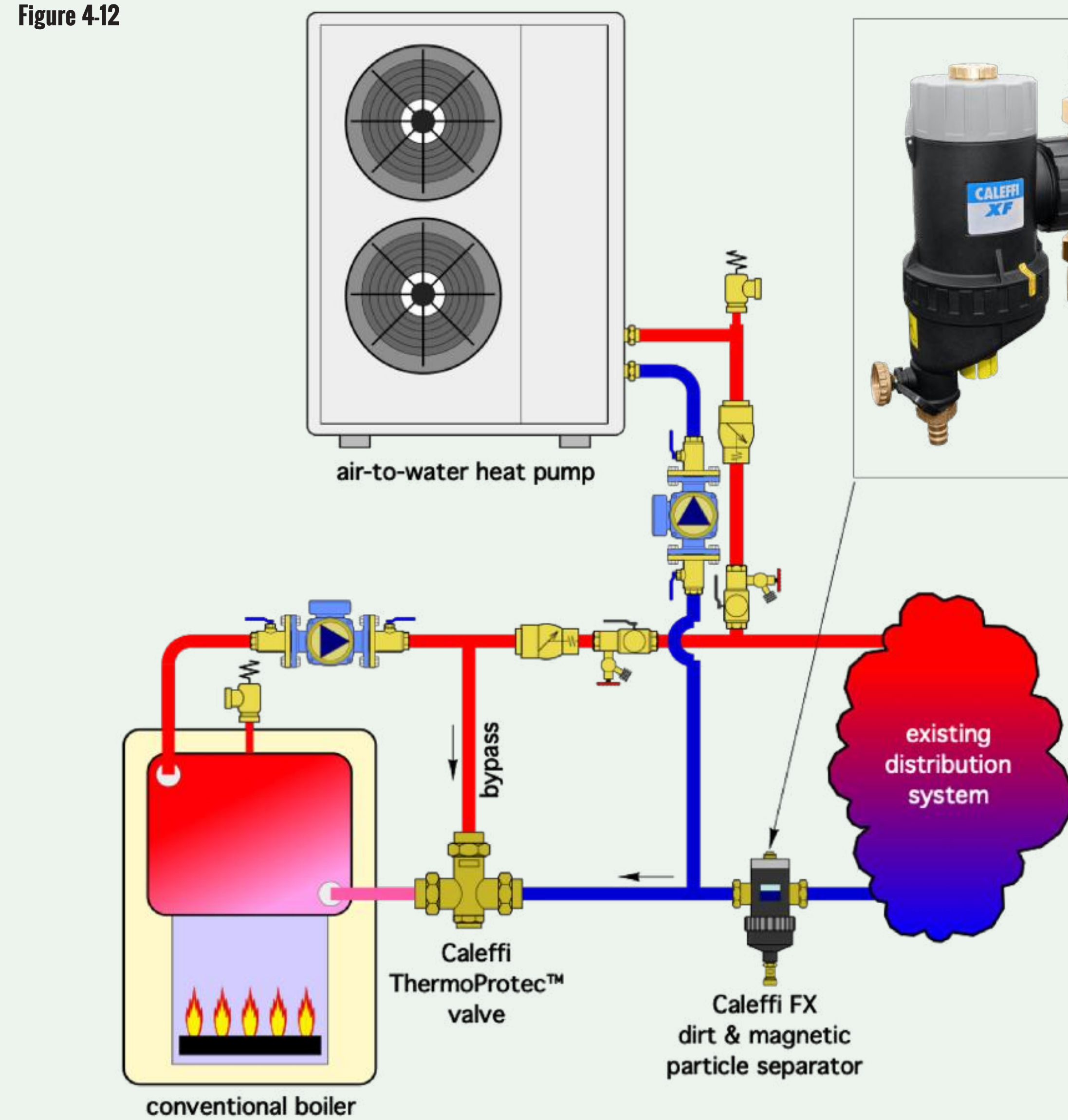
For more information on water quality in hydronic heating systems, see idronics #18.

FLOW REQUIREMENTS FOR HEAT PUMPS

When evaluating an existing hydronic system for a new heat source, such as an air-to-water heat pump or mod/con boiler, it's important to remember that these modern heat sources have *minimum flow rate requirements*. This is in contrast to cast iron boilers that can usually operate at very low flow rates. Older cast iron boilers equipped with tankless coils for domestic water heating can even be fired with no flow.

In many older systems, the water and metal content of a cast iron boiler or steel fire-tube boiler is such that no buffer tank is needed for a multi-zone application. The boiler is able to operate at low flow rates, such as might occur when only one zone is active. However, hydronic heat pumps typically require 2-3 gallons per minute of flow per ton (12,000 Btu/hr) of output capacity. Directly

Figure 4-12



connecting such a heat pump to an extensively zoned hydronic distribution system, such as shown in figure 4-13, is likely to create short cycling issues, especially for heat pumps with fixed speed compressors.

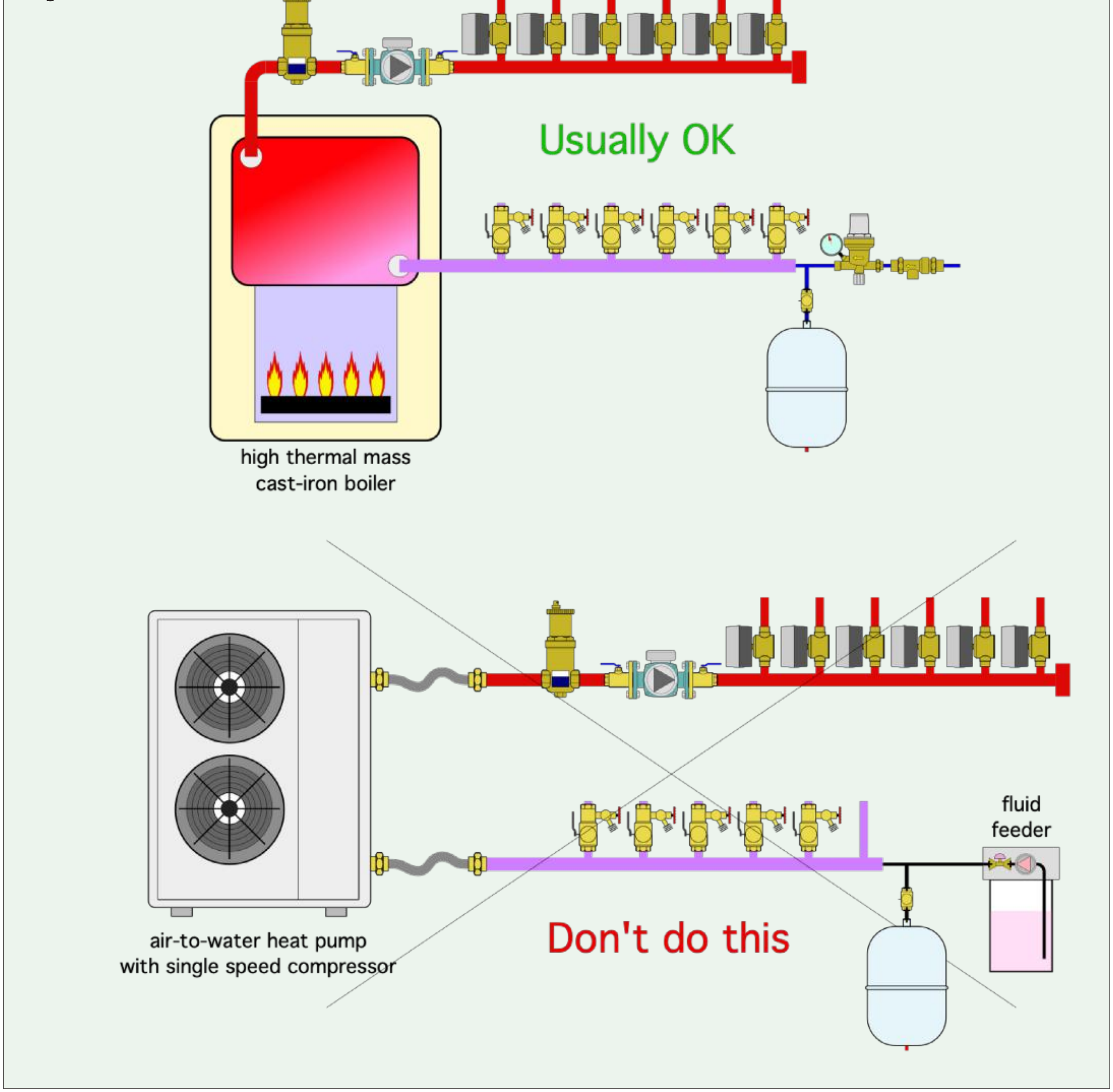
A buffer tank is typically required when retrofitting a hydronic heat pump that has a fixed speed compressor to an extensively zoned hydronic distribution system, as shown in figure 4-14.

The buffer tank allows the heat pump's heat output rate to be much higher than the load from a single zone circuit. A properly sized buffer tank can prevent short cycling. The heat pump is configured to respond to the temperature within the buffer tank, rather than directly from room thermostats.

When a heat pump with a variable-speed "inverter" compressor is used, it may be possible to use a hydraulic separator between the heat pump and zoned load circuits, as shown in figure 4-15.

This configuration is based on the ability of the heat pump to reduce its heat output rate to potentially match the requirement of a single zone circuit, and thus not require the added thermal mass of a buffer tank. The

Figure 4-13



SEP4 hydraulic separator allows the heat pump flow rate to be different from that of the distribution system. It also provides air, dirt and magnetic particle separation for the system. The latter functions are especially important when retrofitting older systems that may contain dirt or magnetite from cast iron or steel components.

For more information on air-to-water heat pumps systems, see idronics #27.

Figure 4-14

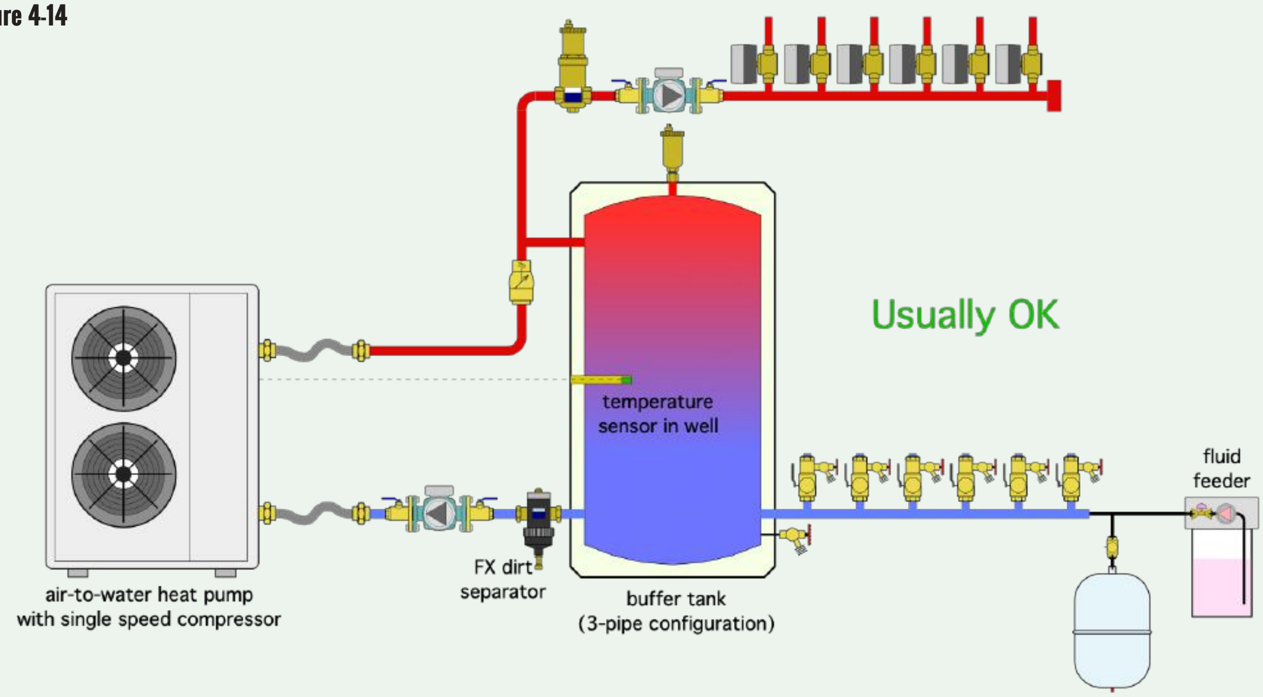
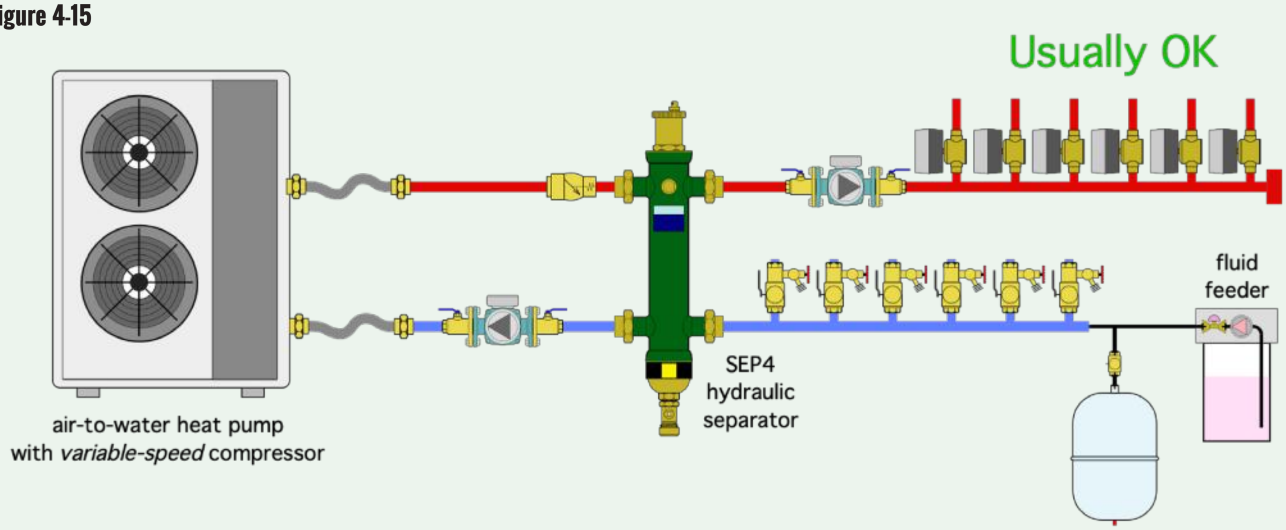


Figure 4-15



SUMMARY

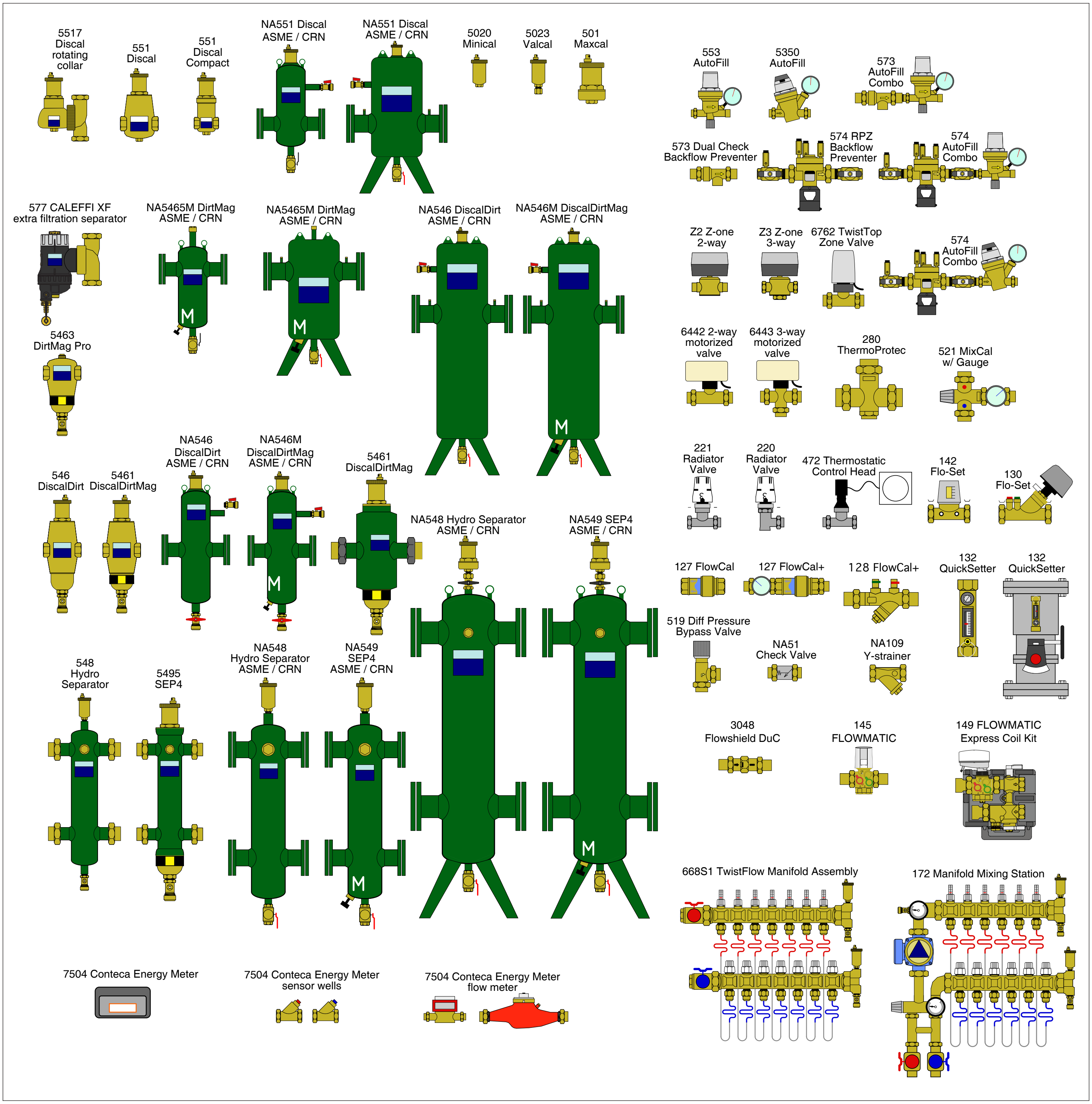
There are millions of existing hydronic heating systems in North America. The vast majority of them were designed around boilers operating on fossil fuel and capable of providing relatively high water temperatures.

As energy markets trend toward electrically powered HVAC systems, heat pumps (both air-to-water and geothermal water-to-water) will play an increasingly important role within the hydronics industry. There will be many situations where an existing system, supplied by a boiler, will be examined for possible inclusion of a heat pump, either as a replacement for the boiler or as the primary heat source in a dual fuel system.

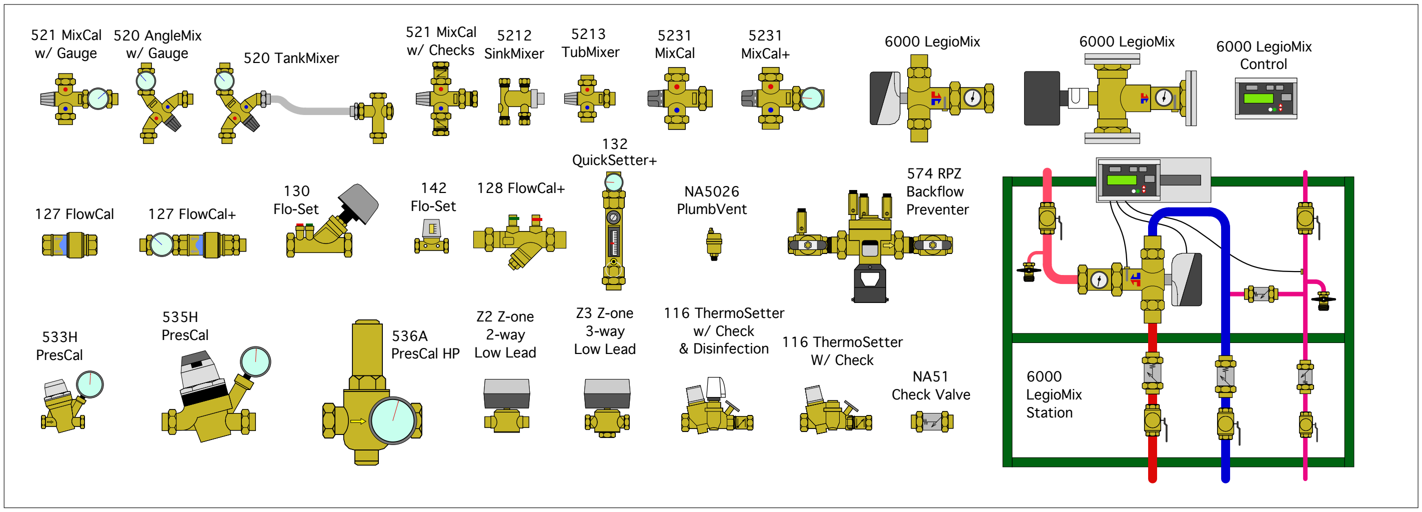
Having a procedure for evaluating the thermal performance of an existing hydronic distribution system for possible operation at lower water temperatures is crucial in deciding how best to integrate the heat pump and what its expected performance will be.

This issue of *idronics* has presented a method for on-site testing of existing hydronic heating circuits, as well as a way to use the results of that testing to infer the performance of those circuits at different — typically lower — supply temperatures. It has also provided guidance on how to configure a modified system to avoid issues that cause unstable operation, corrosion or zones with widely different supply temperatures.

APPENDIX A: CALEFFI HYDRONIC COMPONENTS

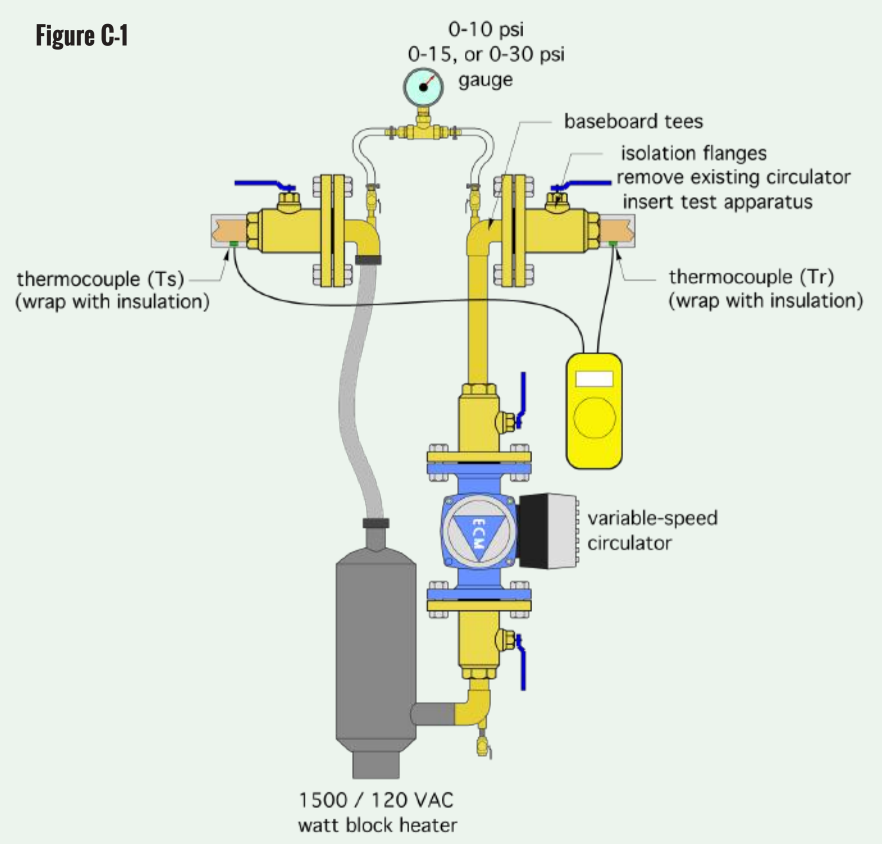


APPENDIX B: CALEFFI PLUMBING COMPONENTS



APPENDIX C: PROCEDURE TO ESTABLISH A CIRCUIT HEAD LOSS CURVE

Figure C-1



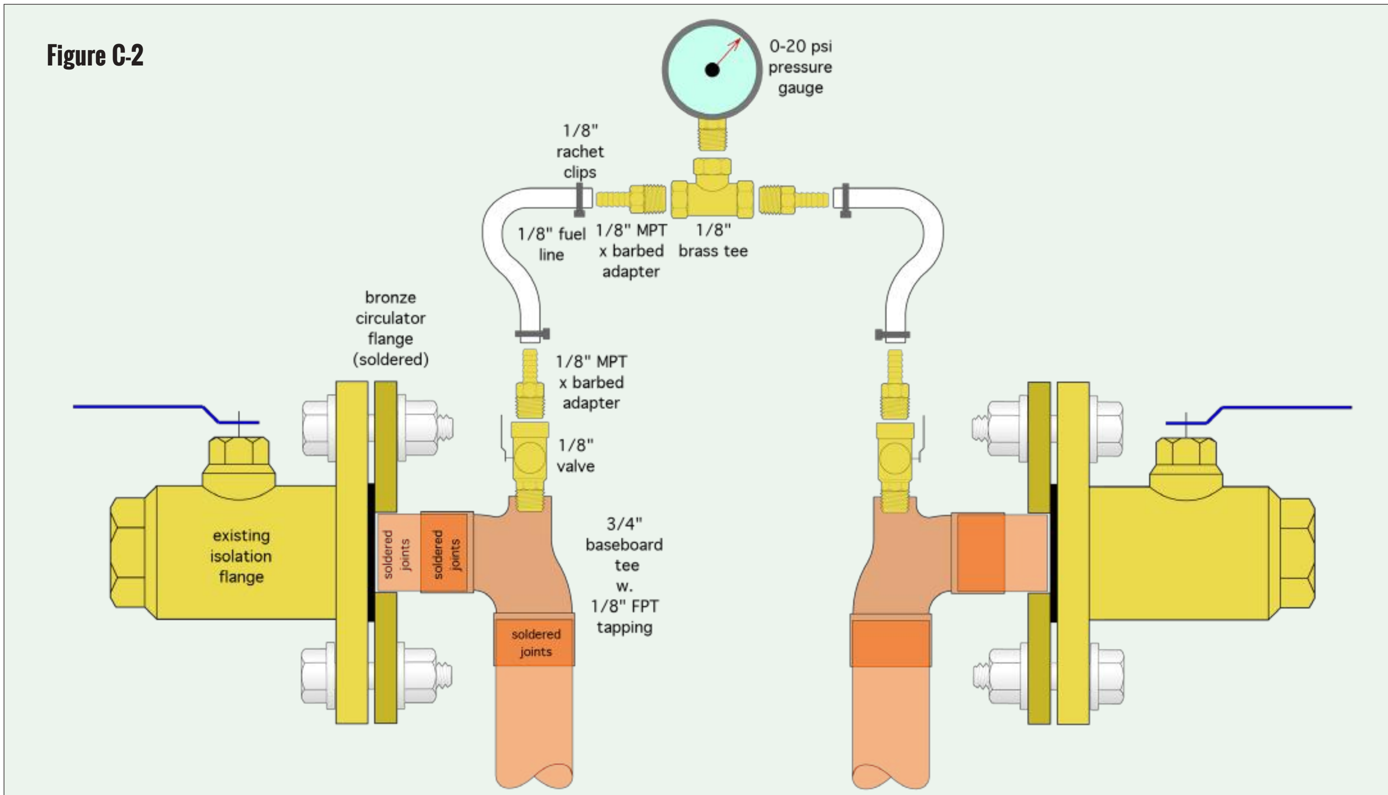
The test apparatus shown in figure C-1 was described in section 3. A procedure for using it to estimate the thermal performance of a circuit was also given.

This apparatus can also be used to gather hydraulic performance information for the circuit. That information can be plotted as a circuit head loss curve, which in turn can be used to estimate the flow rate in the circuit when powered by a different circulator, or a different speed setting on a multi-speed circulator.

Figure C-2 is a closeup view of one way to make the pressure-measuring subassembly shown in figure C-1.

To gather pressure drop data, a pressure gauge needs to be connected to the 1/8” FPT ports on each baseboard tee. Those connections are made with flexible tubing, such as fuel line, combined with small (1/8”) brass fittings. A small valve needs to

Figure C-2



be installed between each baseboard tee and the tee connected to the pressure gauge.

The range of the pressure gauge should be as small as possible but also high enough to handle the static pressure in the circuit. Pressure gauges are available with ranges of 0-10, 0-15, and 0-30 psi. The smaller the range of the pressure gauge, the more accurately it can be used to take pressure readings that then determine the differential pressure across the circuit when it's operating at its normal flow rate. Keeping the static pressure in the circuit as low as possible during the test allows for gauges with smaller pressure ranges to be used.

It may also be necessary to use a 1/4" x 1/8" brass adapter coupling, along with a short 1/8" brass nipple, to connect a pressure gauge with 1/4" MPT threads to the 1/8" FPT connections on the tee shown in figure C-2.

The hydraulic test procedure is relatively simple:

1. Close both valves connecting the baseboard tees to the pressure gauge.

2. Turn on the variable-speed circulator and adjust its speed so that a relatively low flow rate such as 1-2 gpm is indicated on the circulator's display or on a smart phone connected to the circulator via Bluetooth. If an ultrasonic flowmeter is being used for flow rates, it's not necessary to read flow rates from the circulator display. In general, ultrasonic flowmeters yield higher accuracy than the inferred flow rates displayed on small variable-speed circulators.

3. Open the valve on the supply side of the circuit and record the pressure on the gauge. *If the gauge is calibrated in inches of Hg, convert the pressure to psi (1 inch Hg = 0.49115 psi).*

4. Close the value on the supply side of the circuit and open the valve on the return side. Record the reading on the pressure gauge. *If the gauge is calibrated in inches of Hg, convert the pressure to psi (1 inch Hg = 0.49115 psi).*

5. Subtract the pressure read on the return side of the circuit from the pressure read on the supply side of the circuit. This difference is the pressure drop of the circuit at that flow rate.

6. Increase the speed of the circulator until the flow rate settles at a value that's at least 2 gpm higher than the previous flow rate.

7. Repeat steps 3, 4 and 5.

Figure C-3

flow rate (gpm)	ΔP (psi)	**head loss H_L, (ft of head)**
3	0.7	
5	1.7	
7	3.0	
9	4.7	
11	6.6	
13	8.9	

8. Repeat step 6 and 7 until at least 5 flow rates and corresponding differential pressures have been recorded. Figure C-3 shows how to keep the test reading organized in a table.

9. Measure the supply and return temperatures of the circuit and average them.

10. Use figure 3-10 to estimate the density of the fluid in the circuit at the average temperature found in step 9.

11. Convert each differential pressure reading to head using formula C-1.

Formula C-1:

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

Where:
H_L = head loss (ft of head)
ΔP = differential pressure (psi)
D = density of fluid at average circuit temperature (lb/ft^3) (see figure 3-10)
144 = constant required by units

The table in figure C-4 shows the head loss values (in red), which are based on water at 60ºF, where the value of (144/D) in formula C-1 is 2.31.

Figure C-4

flow rate (gpm)	ΔP (psi)	head loss H_L, (ft of head)
3	0.7	**1.6**
5	1.7	**3.9**
7	3.0	**7.0**
9	4.7	**10.8**
11	6.6	**15.3**
13	8.9	**20.6**

12. Plot the head loss numbers on the vertical axis with their corresponding flow rate on the horizontal axis, as shown in figure C-5.

Once the head loss curve for the circuit is established, the pump curve for any "candidate" circulator can be plotted on it. The point where the head loss curve crosses over the pump curve is the operating point. Draw a line straight down from the operating point to the horizontal axis to read the estimated flow rate in the circuit, as shown in figure C-6.

The curves shown in figure C-6 intersect at a flow rate of 8.5 gpm.

Figure C-5

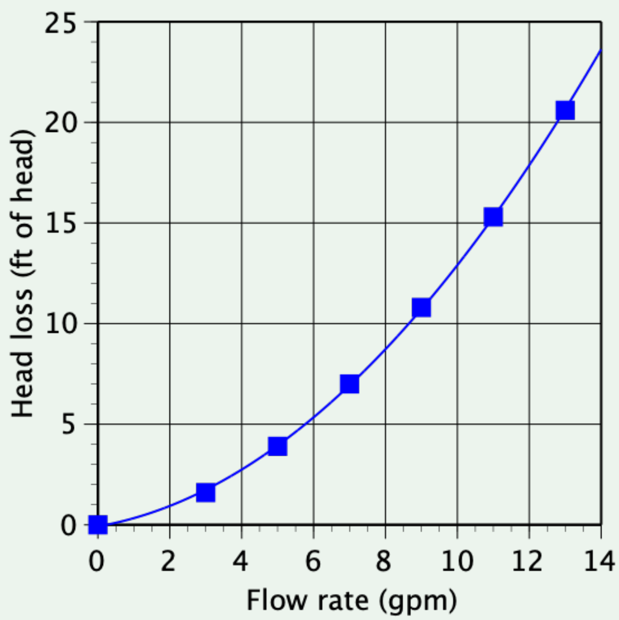
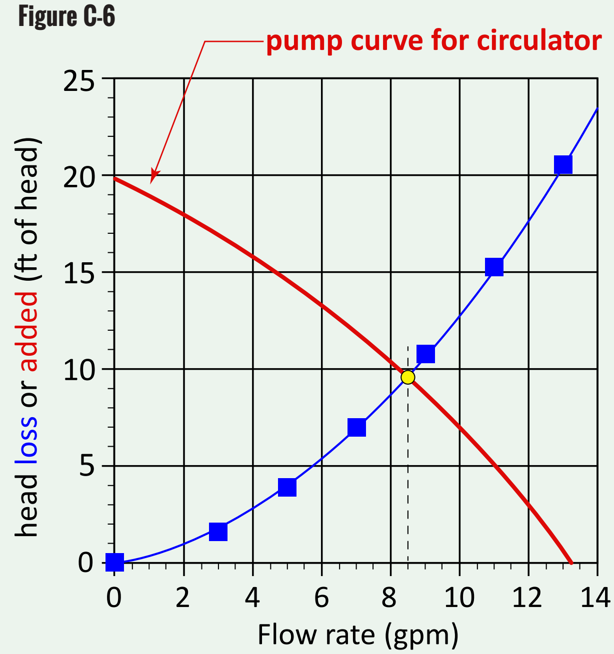


Figure C-6

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