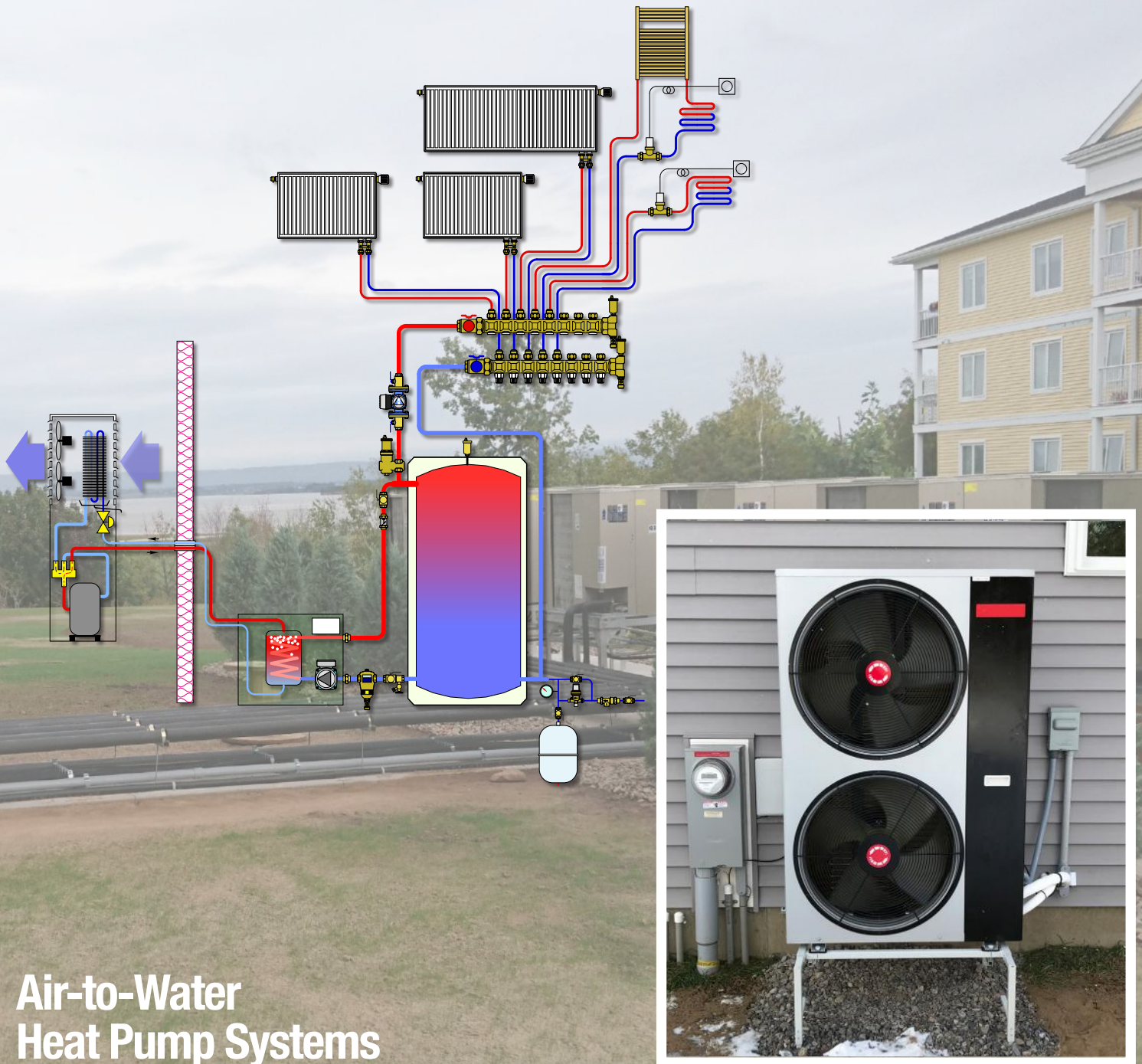


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# FROM THE GENERAL MANAGER & CEO

Dear Plumbing and Hydronic Professional,

There are many types of heat pumps. The one most of us are familiar with is our kitchen refrigerator. It removes heat from food and pumps it back to the kitchen.



Over the past three decades refrigerator manufacturers have continually introduced new ways to reduce the energy used by their products. No longer is a homeowner's decision to replace their refrigerator based mostly on its age. Now, the electrical energy savings offered by the latest technology can justify the purchase - even though their current refrigerator is still functioning.

The design of heat pumps that supply space conditioning and domestic water heating has also improved dramatically. Enhanced vapor injection, variable speed inverter compressors, electronic expansion valves and other state-of-the-art technologies have been integrated into many modern heat pumps. The resulting efficiency gains now allow air-source heat pumps to be used in cold Northern climates, even when outside temperatures fall below 0 °F. And because they operate on electricity, rather than fossil fuel, they are well-positioned for today's focus on carbon reduction driven by changing social attitudes and government policies.

This issue of idronics deals with "air-to-water" heat pumps that heat buildings by absorbing low temperature heat from outside air and delivering it, at higher temperatures, to a hydronic distribution system. These heat pumps combine the advantages of modern air-source heat pump technology with the unsurpassed comfort of hydronic heating and cooling. They are widely used in Europe and Asia, and represent a growing market opportunity within North America.

We hope you enjoy this issue of idronics and encourage you to send us any feedback by e-mailing us at [idronics@caleffi.com](mailto:idronics@caleffi.com).

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Mark Olson

A handwritten signature in black ink that reads "Mark Olson".

General Manager & CEO

A Technical Journal

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# TABLE OF CONTENTS

## 5 Introduction

What is a heat pump?  
History of heat pumps  
Performance limitations

Cold climate air-source heat pumps  
Modern air-to-water heat pumps

## 10 Heat pump operating fundamentals

Non-reversible vs. Reversible heat pumps  
Heating mode thermal performance  
Cooling mode thermal performance

Energy efficiency ratio  
What does "tons" mean?  
Enhanced vapor injection  
Defrosting

## 20 Energy trends & their effect on hydronic heat sources

1. Government policy supporting decarbonization:  
2. Financial incentives:  
3. Strong interest and impressive market growth for "net-zero" buildings:  
4. Decreasing heating and cooling loads:

5. Lack of suitable combustion-based heat sources:  
6. Basic service charges & pricing for natural gas:  
7. Safety issues:  
8. Moratoriums on natural gas expansion:

### Advantages of air-to-water heat pumps

1. Significantly lower installation cost:  
2. Noninvasive installation:  
3. Ground water protection:

4. Independent of incentives:  
5. Higher net COP:  
6. Savings decrease as loads decrease:

## 28 Air-to-water heat pump configurations

7. Influence of global markets:  
Monobloc air-to-water heat pumps  
Freeze protection options for monobloc heat pumps  
Antifreeze-based freeze protection  
Split system air-to-water heat pumps  
interior air-to-water heat pumps

Other air-to-water heat pump system configurations  
Integrated systems  
Balance point  
Climate considerations  
Spr

## 46 Heat emitter options for air-to-water heat pump systems

Seasonal average cop  
Heated floor slabs  
Underfloor tube & plate radiant panels

Radiant wall panels  
Radiant ceiling panels  
Panel radiators

## 58 Design details for air-to-water heat pump

Low-temperature fin-tube baseboard  
Cast iron radiators  
Reducing water temperature in existing systems  
Mounting outdoor heat pumps piping connections  
Buffer tanks  
Buffer tank piping  
Dirt separation  
Auxiliary heating provisions  
Controlling auxiliary heat  
2-Stage room thermostat

2-Stage setpoint control  
2-Stage outdoor reset control  
Auxiliary heat source control logic within the heat pump  
Monitoring heat pump performance  
Domestic water heating options  
desuperheater option  
Indirect tank option  
Reverse indirect tank option  
On-demand external heat exchanger option  
chilled water cooling details chilled water terminal units

## 73 System examples

Preventing unwanted condensation  
System #1  
System #2  
System #3

System #4  
System #5  
System #6  
Summary

## 82 Appendix A: component symbol legend

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# INTRODUCTION

## WHAT IS A HEAT PUMP?

Heat, by nature, always moves from an area of higher temperature to an area of lower temperature. This “natural” heat transfer takes place constantly all around us. Examples include:

- Heat leaving our skin and clothing surfaces, and transferring to cooler air surrounding us.
- Heat transferring from the inside of buildings to outside air whenever the inside temperature is warmer than the outdoor temperature.
- A glass of cold iced tea placed on a countertop continually absorbing heat from warmer air surrounding it, as well as from the countertop, both of which are at higher temperature than the tea.

No machines or special techniques are needed to move heat from materials at higher temperature to materials at lower temperature.

Heat pumps were developed to reverse the “natural” direction of heat transfer. Their function is to move (e.g., “pump”) heat from materials at lower temperature to materials at higher temperature.

The low-temperature heat is gathered from some material called the “source,” and then concentrated and released into another material called the “sink.”

In some respects, a heat pump is similar to a refrigerator. The latter absorbs low-temperature heat from the food placed inside it. It then raises the temperature of the absorbed heat and releases it in the surrounding air. Most heat pumps and refrigerators use a chemical called a refrigerant that circulates within a closed circuit of components and changes phase between liquid and vapor to “pump” heat from low-temperature materials into higher-temperature materials. The refrigerant is pushed through the closed loop of components by an electrically operated compressor. The details of this refrigeration cycle are discussed in section 2.

Although there are similarities between heat pumps and refrigerators, there are also very distinct differences. Most heat pumps are designed to operate at higher rates of heat transfer compared to a common refrigerator. Most heat pumps can also reverse which material supplies the

low-temperature heat and which material receives the higher-temperature heat. This makes it possible for heat pumps to heat and cool buildings. There are also many different configurations of heat pumps available depending on the material from which low-temperature heat is being absorbed, and the material into which higher-temperature heat is being released.

When used to heat buildings, heat pumps can gather low-temperature heat from sources such as outdoor air, ground water, lakes or ponds, or tubing buried in the earth. All of these sources provide “free” low-temperature heat.

Heat pumps that extract low-temperature heat from outside air are common in North America. They are appropriately called “air-source” heat pumps. The vast majority of air-source heat pumps currently in service are configured to deliver higher-temperature heat through a forced-air distribution system within the building. This leads to the more specific classification of “*air-to-air*” heat pump.

Heat pumps that extract low-temperature heat from geothermal sources such as lakes, ponds, wells or tubing buried in the earth use water or an antifreeze solution to convey heat from those sources to the heat pump. They are thus classified as water-source heat pumps. Water-source heat pumps that deliver heat through a forced-air system are more specifically called “*water-to-air*” heat pumps. Those that deliver heat using a hydronic distribution system are known as “*water-to-water*” heat pumps.

This issue of *idronics* deals with a specific heat pump configuration that absorbs low-temperature heat from outside air and delivers that heat, at higher temperatures, to a stream of water within a building. This type of heat pump is more specifically called an “*air-to-water*” heat pump.

## HISTORY OF HEAT PUMPS

Heat pumps are based on the principles of refrigeration, which were first demonstrated by Scottish physician William Cullen in 1755. Cullen developed an apparatus to create a vacuum over a container of ether immersed in water. The vacuum caused the ether to boil, and in doing so, absorb heat from the water to create a small amount of ice.

The thermodynamic principles underlying heat pumps are partially credited to Lord Kelvin, who contributed to the formulation of the first and second laws of thermodynamics and proposed the concept of an absolute temperature scale. The French engineer Sadi Carnot also contributed

to the thermodynamic underlying “heat engines,” which are devices that extract energy from some higher-temperature material and convert that energy into a combination of mechanical work and lower-temperature heat. From a thermodynamic perspective, a heat pump can be thought of as a heat engine operating in reverse. It combines heat from a low-temperature source material with mechanical work to produce heat at a higher temperature. Carnot, building on the work of Kelvin, also developed a formula that sets the theoretical performance limits for any heat pump.

Crude heat pumps were developed in the early 1900s but remained little more than science experiments at a time when fossil fuels, especially coal and petroleum, were the dominant energy source for heating buildings.

The first heat pumps to be mass produced were based on machines used for central air conditioning. The Carrier Corporation is widely recognized as one of the first companies to commercialize residential central cooling using vapor-compression refrigeration systems. During the 1950s, Carrier Corporation provided over 700 early-generation central air-conditioning systems for one of the first large-scale housing developments in Levittown, Pennsylvania.

Although often taken for granted today, the advent of central air conditioning at that time allowed scarcely populated areas in the southwestern U.S. to develop into major population centers. Some historians have even cited air conditioning as one of the most impactful technical accomplishments of the 20th century.

Early-generation air-conditioning systems were only able to cool buildings, absorbing heat from interior spaces and dissipating it to outside air. The next technological hurdle was finding ways to reverse the direction of heat flow, and thus convert low-temperature heat in outside air into higher-temperature heat to maintain comfort in buildings. This was the advent of air-to-air heat pumps.

The basic configuration of a “split system” air-to-air heat pump, operating in heating mode, is shown in Figure 1-1. The outside unit, often called the “condenser” because of its origin in air-conditioning systems, connects to an

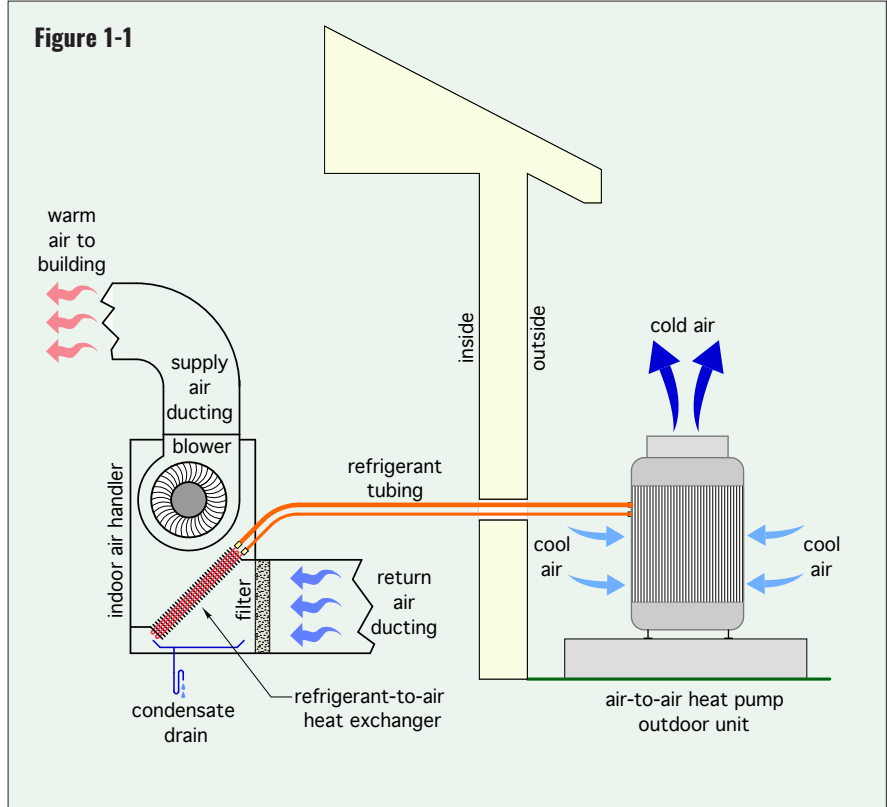


Figure 1-2



Courtesy of Allied A/C & Heating

interior air handler using two copper refrigerant tubes. The compressor is located in the outdoor unit. The indoor unit contains a refrigerant-to-air heat exchanger and blower.

As is often the case with new technologies, early experiences with air-to-air heat pumps were mixed. First generation products experienced higher than acceptable compressor failure rates. In 1964, this reliability issue led



the U.S. Department of Defense to issue a ban on the use of heat pumps in military facilities due to the severity of maintenance problems. Fossil fuel continued to be the dominant energy source for heating buildings.

The OPEC oil embargo, which began in 1973, reinvigorated efforts to develop reliable electrically powered heat pumps that could lessen dependence on petroleum-based heating fuels. Manufacturers of air-conditioning systems worked on methods of reversing the direction of heat flow, and thus allow low-temperature heat in outside air to be raised to temperatures sufficient for heating buildings.

One of the earliest attempts at creating a vapor-compression machine that could heat as well as cool simply reversed the direction of the entire air conditioner within an opening in an exterior wall. Another used multiple dampers to change airflow directions.

Manufacturers eventually discovered that the refrigerant flows used in vapor-compression air conditioners could be reversed using a combination of four hand-operated valves. In time, this approach was replaced by two valves operated by electrical solenoids. Further development led to a single 4-port, electrically operated “reversing valve.” This type of valve, which is discussed in more detail later in this issue, is now used in a wide variety of heat pumps that provide heating and cooling.

**Figure 1-3**



Reversing valves made it practical to heat and cool homes using air-to-air heat pumps. Sales of residential air-to-air heat pumps grew rapidly during the 1970s. The primary markets were southern states with relatively mild winter temperatures and a definite need for summer cooling. Air-to-air heat pumps became heavily promoted by southern electric utilities, as well as by manufacturers such as Carrier, Westinghouse and General Electric.

During the 1970s, the reliability of air-to-air heat pumps continually improved through revised compressor design, better lubrication details and techniques to reduce liquid “slugging” of compressors. By 1975, the U.S. Department of Defense lifted their previous ban on heat pumps in military facilities. A surge of interest in heat pumps during 1976 led to an annual sales growth rate of 96%. Manufacturers were having difficulty keeping up with demand. By 1978, it was estimated that air-to-air heat pumps were installed in over 1.4 million U.S. homes.

### **PERFORMANCE LIMITATIONS**

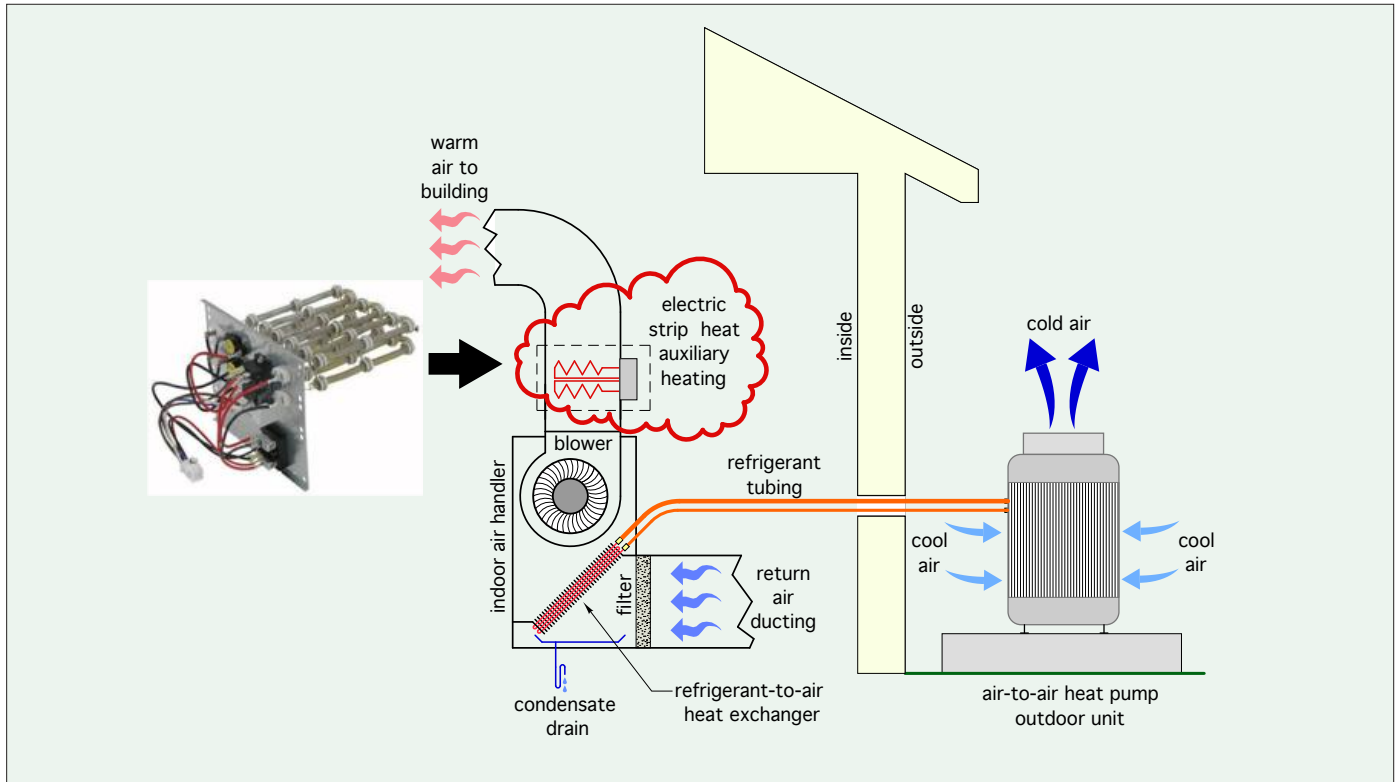
Early generation air-to-air heat pumps could not operate well at the low outdoor temperatures experienced in the Northern U.S. and Canada. Many were limited to minimum outdoor temperatures in the range of 15-20°F. If the outdoor temperature dropped below this limit, the heat pump would operate at grossly insufficient output or simply turn off. The heating load would then be assumed by electric resistance “strip heaters,” which are heating elements mounted in the supply air plenum on the heat pump’s interior unit, as shown in Figure 1-4.

Strip heat was usually activated by the second stage of a 2-stage thermostat as room air temperature dropped slightly below the desired setting. Although reliable as a supplemental heat source, strip heat, like all electric-resistance heating, is expensive to operate. Some early-generation air-to-air heat pumps were also installed along with gas-fired furnaces that would assume the full heating load if the heat pump could not operate due to low outdoor temperature or some other condition.

The inability to operate at the low outdoor air temperatures experienced in many northern states, and much of Canada, created a stigma that air-source heat pumps were only suitable for heating in mild climates. This limitation was one of the largest factors leading to the emergence of geothermal heat pumps during the 1980s. Because water returning from earth loops, wells or large open bodies of water was always above 32°F, even when outdoor air temperatures were extremely cold, geothermal heat pumps could provide predictable heating performance in northern climates. In milder climates,

Courtesy of AD Cooper

**Figure 1-4**

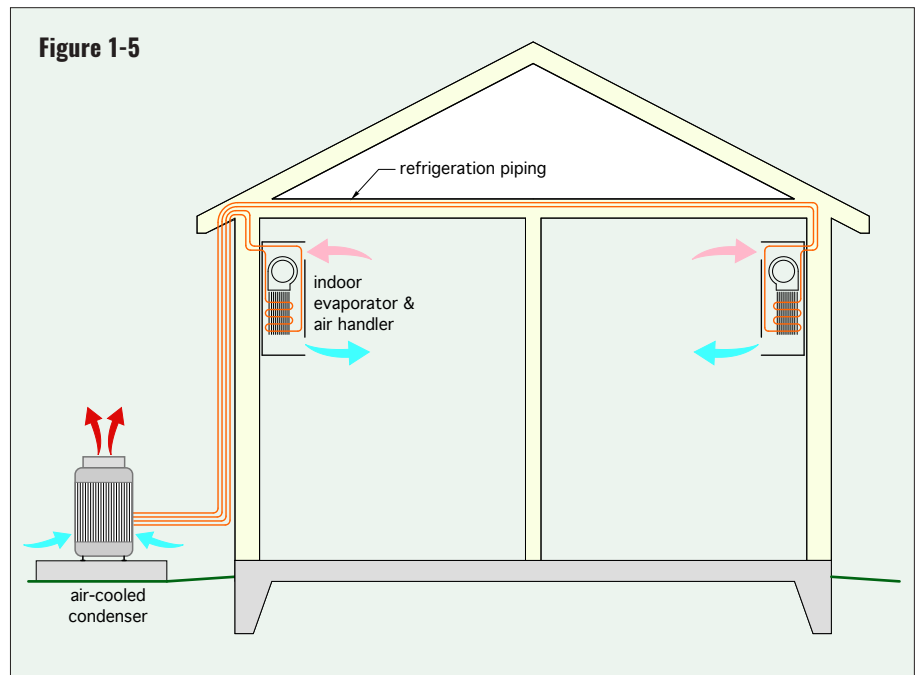


geothermal heat pumps also provided higher-efficiency cooling performance compared to early-generation air-source heat pumps. The North American market for geothermal heat pumps has grown steady over the last 30 years, largely driven by the potential for high efficiency and thus lower operating cost.

### **COLD CLIMATE AIR-SOURCE HEAT PUMPS**

As the market for geothermal heat pumps increased over the last 20 years, so did efforts to improve the performance of air-source heat pumps. New refrigeration technologies such as enhanced vapor injection (EVI), variable-speed “inverter” compressors, and electronic expansion valves, none of which were available for use in early-generation heat pumps, now allow modern air-source heat pumps to achieve significantly higher thermal performance at cold outdoor temperatures, in some cases as low as -22°F.

**Figure 1-5**



Many “cold climate” air-to-air heat pumps (a.k.a. “low ambient” air-source heat pumps) are currently available as “ductless” split systems. A single outdoor unit connects to

one or more indoor air handlers using refrigeration piping. Figure 1-5 shows the concept for a ductless split air-to-air heat pump system with two interior wall-mounted air handlers.



**Figure 1-6**



**Figure 1-7**



An example of a typical outdoor unit for a modern air-to-air heat pump is shown in Figure 1-6. One of the indoor air handler units connected to this outdoor unit is shown in Figure 1-7.

Although “ductless” split system heat pumps can provide reasonably good thermal performance, they are limited to space heating or cooling. They are also limited by the compromises associated with forced air distribution. These include drafts, dispersal of dust and allergens, potential for clogged air filters, cool air collecting at floor level as warm air rises to ceiling level, possible aggravation of respiratory illnesses and objectionable interior noise.

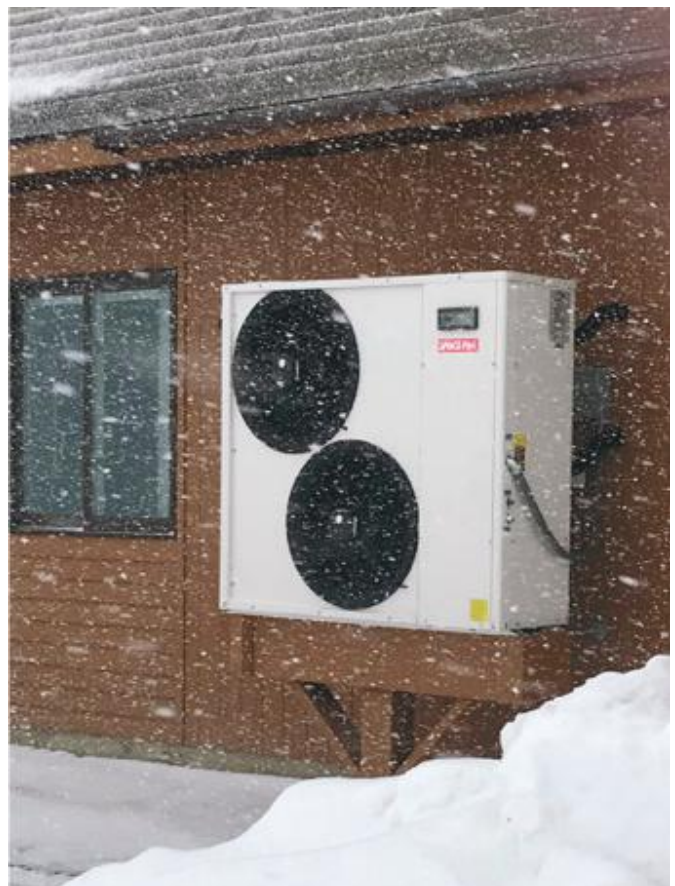
## **MODERN AIR-TO-WATER HEAT PUMPS**

The same innovations that now make ductless air-to-air heat pumps viable in cold climate applications have been used to create air-to-water heat pumps. When operating in heating mode, these units absorb heat from outdoor air, concentrate that heat to increase its temperature, and transfer it to a stream of water or an antifreeze solution. The heated water can be used for a wide variety of loads such as hydronic space heating, domestic water heating or pool heating. Air-to-water heat pumps can also be reversed to extract heat from an interior stream of water and dissipate it to outside air. As such they can be used to supply several types of chilled-water cooling distribution systems. Modern air-to-water heat pumps provide an ideal combination of low ambient thermal performance along with the unsurpassed comfort and energy efficiency afforded by modern hydronics technology.

One example of a modern air-to-water heat pump is shown in Figure 1-8.

The remainder of this issue will discuss the details for applying modern air-to-water heat pumps in a variety of heating and cooling applications.

**Figure 1-8**



# HEAT PUMP OPERATING FUNDAMENTALS

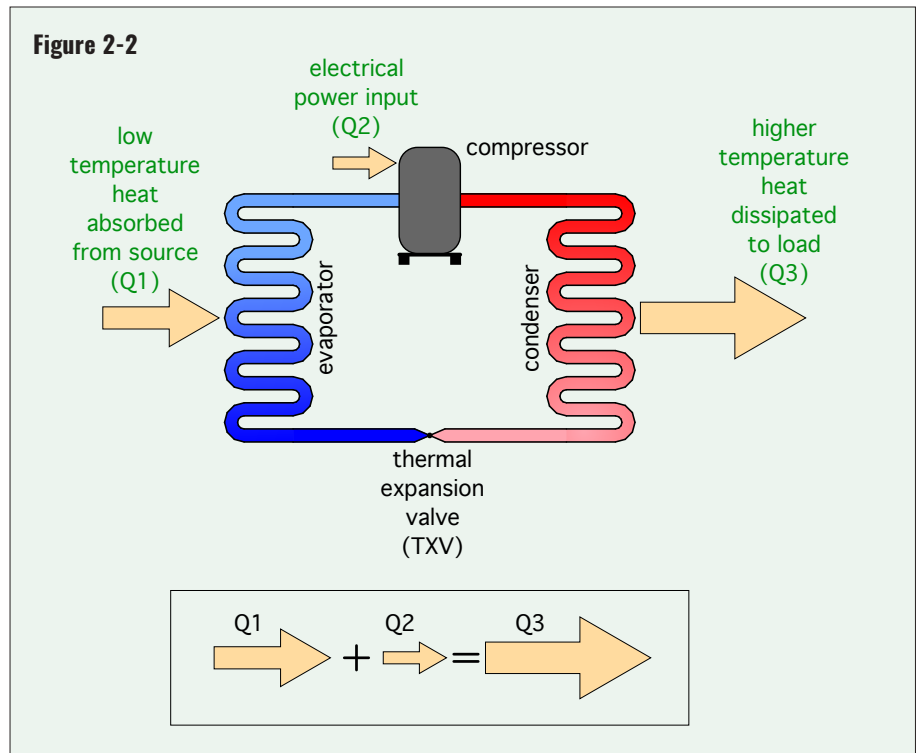
The refrigeration cycle is the basis of operation of all vapor-compression heat pumps. During this cycle, a chemical compound called the refrigerant circulates around a closed piping loop passing through all major components of the heat pump. These major components are named based on how they affect the refrigerant passing through them. They are as follows:

- Evaporator
- Compressor
- Condenser
- Thermal expansion valve (TXV)

The basic arrangement of these components to form a complete refrigeration circuit are shown in Figure 2-1.

To describe how this cycle works, a quantity of refrigerant will be followed through the complete cycle.

The cycle begins at station (1) as cold liquid refrigerant within the evaporator. At this point, the refrigerant is colder than the source media (e.g., air or water) passing across the evaporator. Because of this temperature difference, heat moves from the higher-

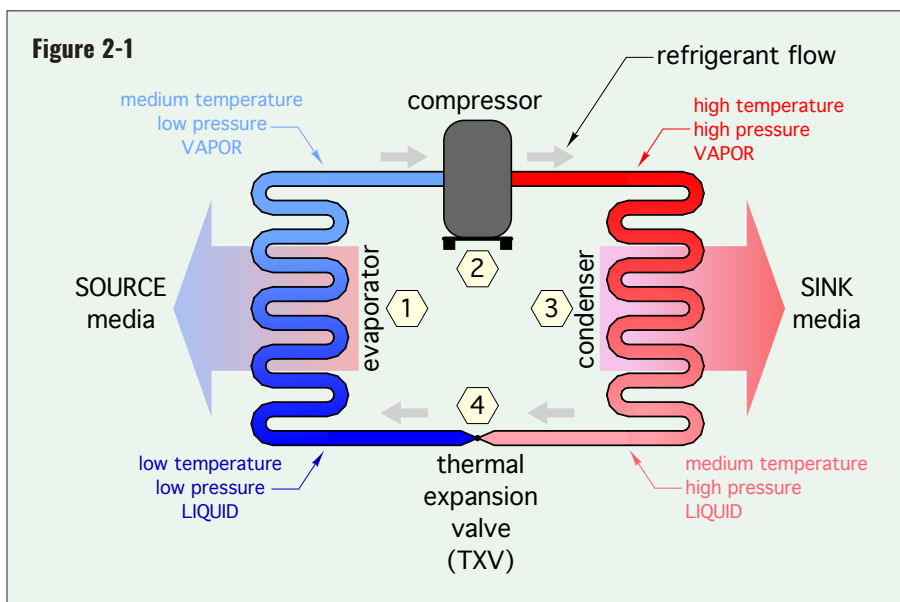


temperature source media into the lower-temperature refrigerant. As the refrigerant absorbs this heat, it changes from a liquid to a vapor (e.g., it evaporates). The vaporized refrigerant continues to absorb heat until it is slightly warmer than the temperature at which it evaporates. The additional heat required to raise the temperature of the refrigerant

above its saturation temperature (e.g., where it vaporizes) is called *superheat*, which also comes from the source media.

This vaporized refrigerant then flows on to the compressor at station (2). Here a reciprocating piston or an orbiting scroll driven by an electric motor compresses the vaporized refrigerant. This causes a large increase in both pressure and temperature. The electrical energy used to operate the compressor is also converted to heat and added to the refrigerant. The temperature of the refrigerant gas leaving the compressor is usually in the range of 120° to 170°F depending on the operating conditions.

The hot refrigerant gas then flows into the condenser at station (3). Here it transfers heat to a stream of water or air (e.g., the sink media) that carries the heat away to the load. As it gives up heat, the refrigerant changes from a high-pressure, high-temperature vapor into a high-pressure, somewhat cooler liquid (e.g., it condenses).



The high-pressure liquid refrigerant then flows through the thermal expansion valve at station (4), where its pressure is greatly reduced. The drop in pressure causes a corresponding drop in temperature, restoring the refrigerant to the same condition it was in when the cycle began. The refrigerant is now ready to repeat the cycle.

The refrigeration cycle remains in continuous operation whenever the compressor is running. This cycle is not unique to heat pumps. It is used in refrigerators, freezers, room air conditioners, dehumidifiers, water coolers, vending machines and other heat-moving machines.

Figure 2-2 shows the three primary energy flows involved in the refrigeration cycle. The first energy *input* is low-temperature heat absorbed from the source media into the refrigerant at the evaporator. The second energy *input* is electrical energy flowing into the compressor whenever it is operating. The third energy flow is the heat *output* into the sink media at the condenser.

The first law of thermodynamics dictates that, under steady state conditions, the total energy input rate to the heat pump must equal the total energy output rate. Thus, the sum of the low-temperature heat absorption rate into the refrigerant at the evaporator, plus the rate of electrical energy input to the compressor, must equal the rate of energy dissipation from the refrigerant at the condenser. This is depicted by the arrows in Figure 2-2.

### NON-REVERSIBLE VS. REVERSIBLE HEAT PUMPS

Heat pumps always move heat from a lower-temperature source media to a higher-temperature “sink” media. The basic non-reversible heat pump described in Figures 2-1 and 2-2 can be used as a dedicated heating device or a dedicated cooling device.

**Figure 2-3**



As a dedicated heating device, the evaporator side of the heat pump will always gather low-temperature heat from some source where that heat is freely available and abundant. The condenser side will always deliver higher-temperature heat to the load.

One example would be a heat pump that always delivers energy for space heating a building. Another would be a heat pump that always delivers energy to heat domestic water. Still another would be a heat pump that always delivers heat to a swimming pool.

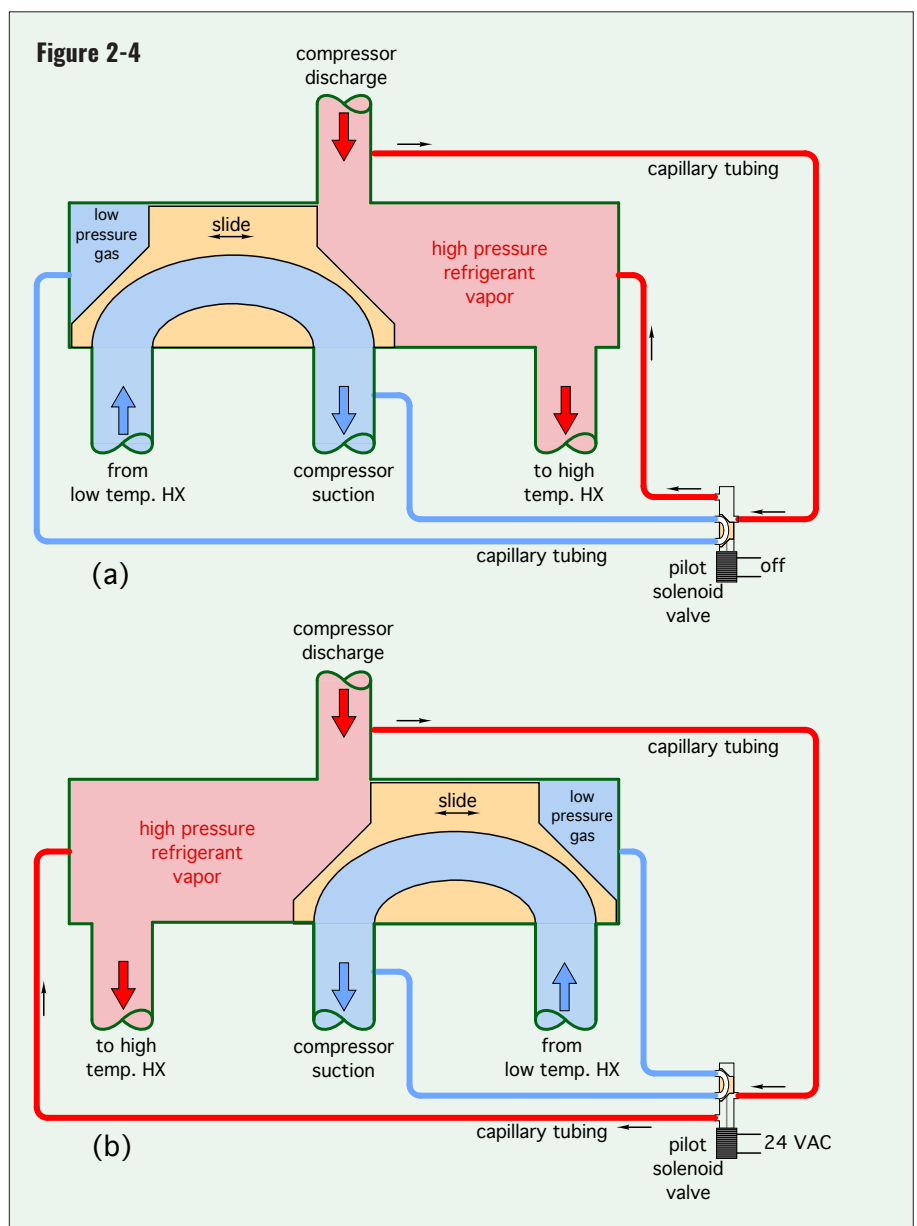
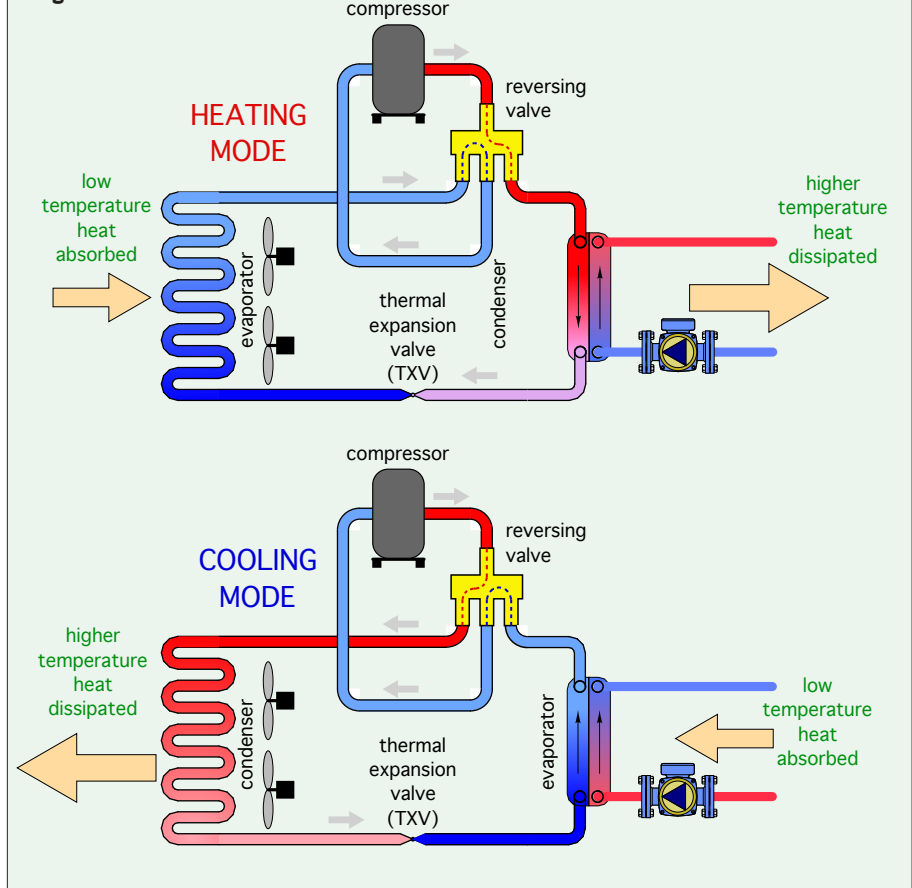




Figure 2-5



As a dedicated cooling device, the evaporator side of a non-reversible heat pump always absorbs heat from a media that is intended to be cooled. Examples would be heat extraction from a building during warm weather, or heat extraction from water that will eventually be converted into ice. The condenser side of such a heat pump will always dissipate heat to some media that can absorb it (e.g., outside air, ground water or soil).

There are several applications where non-reversible heat pumps can be applied. However, one of the most unique benefits of modern heat pumps is that the refrigerant flow can be reversed to quickly convert the heat pump from a heating device to a cooling device. Such heat pumps are said to be “reversible.” A reversible heat pump that heats a building

in cold weather can also cool that building during warm weather.

Reversible heat pumps contain an electrically operated device called a reversing valve. Figure 2-3 shows an example of a modern reversing valve.

The type of reversing valve used in most heat pumps contains a slide mechanism that is moved by refrigerant pressure. The direction of movement is controlled by a small “pilot” solenoid valve. When the heat pump is in heating mode, the magnetic coil of pilot solenoid valve is not energized. This allows the high-pressure refrigerant leaving the compressor to position the slide within the reversing valve so hot refrigerant gas from the compressor goes to the condenser, as shown in Figure 2-4a.

When the heat pump needs to operate in cooling mode, the pilot solenoid valve is energized by a 24 VAC electrical signal. This allows the refrigerant pressure to immediately move the slide within the reversing valve to the opposite end of its chamber. Hot gas leaving the compressor is now routed to the heat pump’s other heat exchanger (e.g., what was the evaporator now becomes the condenser.) This is illustrated in Figure 2-4b.

Figure 2-5 shows where a reversing valve is installed in an air-to-water heat pump.

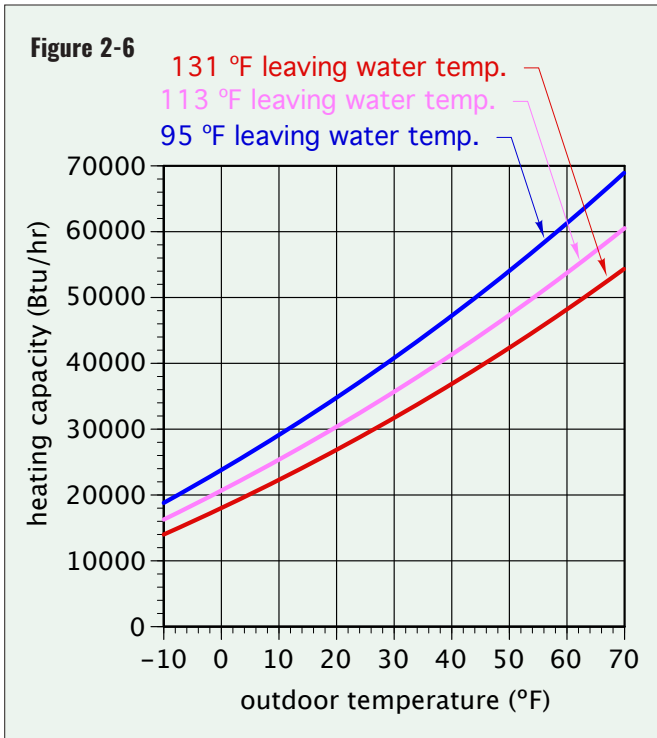
The reversing valve effectively “swaps” the functions of the heat pump’s two heat exchangers. The heat exchanger that serves as the evaporator in the heating mode serves as the condenser in the cooling mode. Similarly, the other heat exchanger that served as the condenser in the heating mode acts as the evaporator in the cooling mode.

The most common configuration for a reversible heat pump is one in which two thermal expansion valves are used in combination with two check valves. One thermal expansion valve functions during the heating mode, while the other functions during the cooling mode. Some heat pumps also use a single electronically controlled “bi-directional” thermal expansion valve. For simplicity, the heat pump refrigeration piping diagrams shown assume a single bi-directional thermal expansion valve.

### HEATING MODE THERMAL PERFORMANCE

In the heating mode, there are two indices used to quantify heat pump performance:

- a. Heating capacity
- b. Coefficient of performance (COP)



Heating capacity is the *rate* at which the heat pump delivers heat to the load. As such, it is similar to the heating capacity of a boiler. However, *the heating capacity of any heat pump is very dependent on its operating conditions, specifically the temperature of the source media and the temperature of the sink media.* The greater the temperature difference between the source media and the sink media, the lower the heat pump's heating capacity. Figure 2-6 shows how heating capacity of a specific air-to-water heat pump varies as a function of outdoor air temperature and the water temperature leaving its condenser.

A heat pump's heating capacity also depends on the flow rate of the source and sink media through the evaporator and condenser. The higher these flow rates are, the greater the heating capacity will be. This is the result of increased convection heat transfer at higher flow velocities. However, the gains in heating capacity are not proportional to the increase in flow rate. Heating capacity increases incrementally at high flow rates. In some cases, the gains in heating capacity do not justify the significantly higher electrical power input to larger circulators, higher speed operation of variable-speed circulators, or higher fan speeds. Water flow rates in the range of

2 to 3 gpm per 12,000 Btu/hr of rated heating capacity are generally recommended.

The coefficient of performance (COP) of a heat pump is a number that indicates the *ratio of the beneficial heat output from the heat pump, divided by the electrical power input required to operate the heat pump.* The higher the COP, the greater its rate of heat output per unit of electrical input power.

Formula 2-1 shows this relationship in mathematical form. The factor 3.413 in this ratio converts watts into Btu/hr. This makes COP a unitless number.

**Formula 2-1**

$$\text{COP} = \frac{\text{heat output (Btu/hr)}}{\text{electrical input (watt)} \times 3.413}$$

COP can also be visualized as the ratio of the heat output arrow divided by the electrical power input arrow, as shown in Figure 2-7.

Another way to think of COP is the number of units of heat output energy the heat pump delivers per unit of electrical input energy. Thus, if a heat pump operates at a COP of 4.1, it provides 4.1 units of heat output energy *per equivalent unit* of electrical input energy.

COP can also be considered as a way to compare the thermal advantage of a heat pump to that of an electric

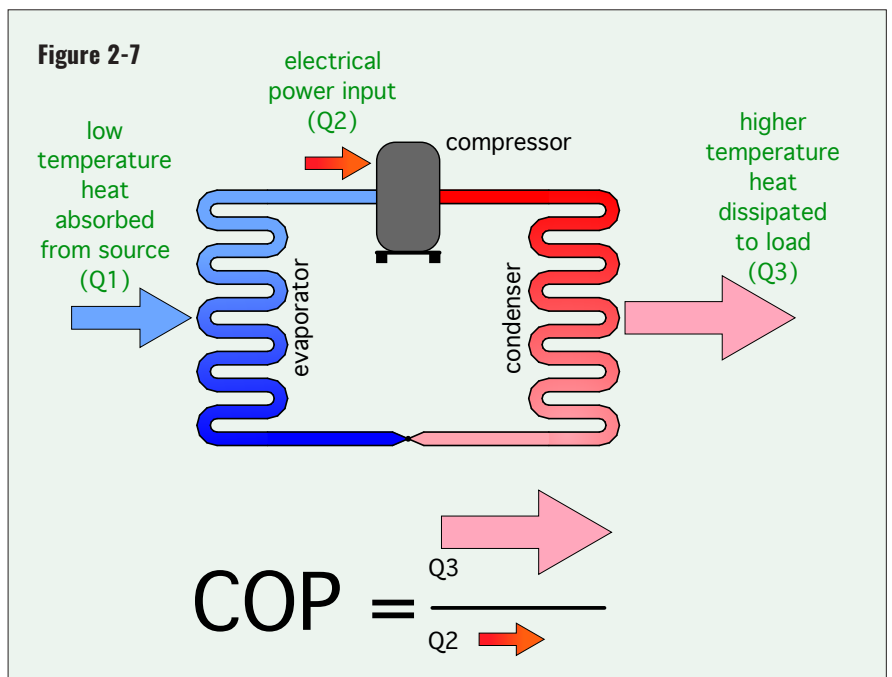


Figure 2-8

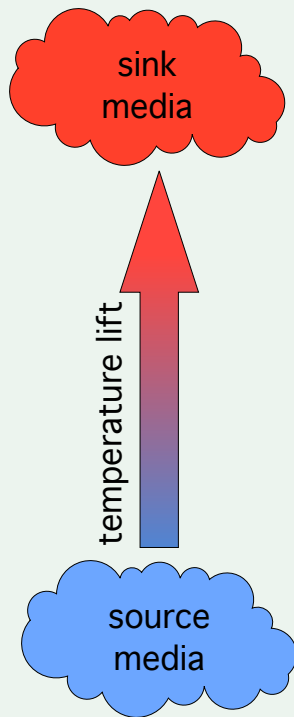
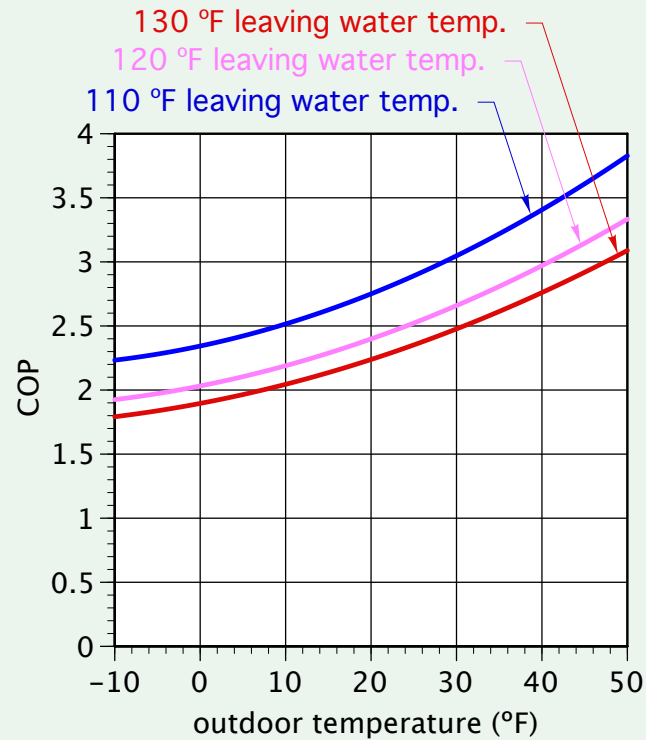


Figure 2-9



resistance heating device that provides the same heat output. For example, if an electric resistance space heater is 100% efficient, then by comparison, a heat pump with a COP of 4.1 would be 410% efficient. Some would argue that no heat source can have an efficiency greater than 100%. This is true for any heat source that simply converts a fuel into heat. However, much of the heat released by a heat pump is heat that was *moved* instead of created through combustion or direct conversion of electrical energy to heat. As such, its beneficial effect is equivalent to a heat source that would have an efficiency much higher than 100%.

The COP of all heat pumps is highly dependent on operating conditions. This includes the temperature of the source media, as well as the media to which the heat pump dissipates heat. *The closer the temperature of the source media is to the temperature of the sink media, the higher the heat pump's COP.*

One can visualize the difference between the source and sink temperatures as the “temperature lift” the heat pump must provide, as shown in Figure 2-8.

The smaller the lift, the higher the heat pump's COP.

The theoretical maximum COP for any heat pump was established by nineteenth century scientist Sadi Carnot and is appropriately called the Carnot COP. It is based on the *absolute temperatures* of the source media and sink media and is given in Formula 2-2.

Formula 2-2

$$COP_{Carnot} = \frac{T_{sink}}{(T_{sink} - T_{source})}$$

$COP_{Carnot}$  = Carnot COP (the maximum possible COP of any heat pump)

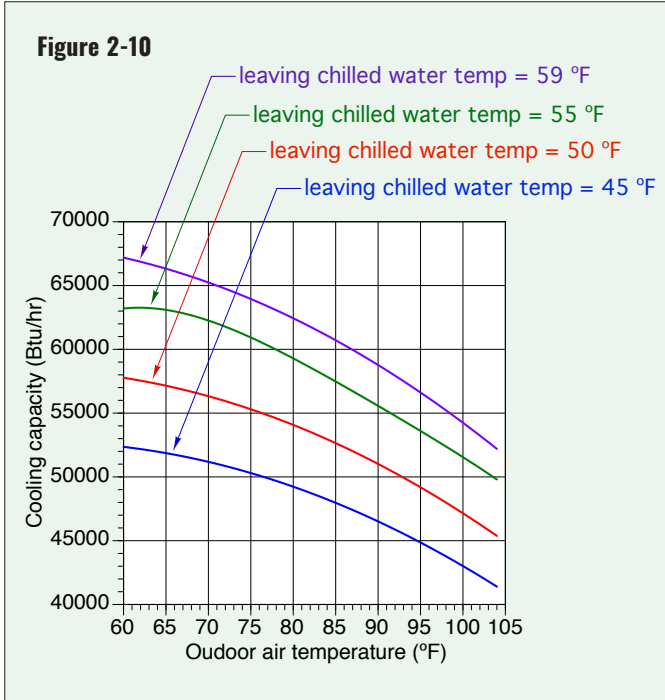
$T_{sink}$  = *absolute* temperature of the sink media to which heat is delivered (°R)

$T_{source}$  = *absolute* temperature of the source media from which heat is extracted (°R)

°R = °F + 458°

This Carnot COP is based on a hypothetical heat pump that has no mechanical energy losses due to friction or electrical losses due to resistance. It is also based on “infinitely sized” source and sink that remain at exactly the same temperatures as they give up and absorb heat. No real heat pump operates under such idealized conditions, and thus *no real heat pump ever attains the Carnot COP.*





The COPs of currently available heat pumps, even when operated under very favorable conditions, is substantially lower than the Carnot COP. Still, the Carnot COP serves as a way to compare the performance of evolving heat pump technology to a theoretical limit. It also demonstrates the inverse relationship between the “temperature lift” of a heat pump and COP.

The COP of air-to-water heat pumps decreases as the outside air temperature decreases. It also decreases as the temperature of the water leaving the heat pump’s condenser increases. Figure 2-9 shows a typical relationship between COP versus outdoor temperature and the water temperature leaving the condenser for a modern “low ambient” air-to-water heat pump.

**COOLING MODE THERMAL PERFORMANCE**

In the cooling mode, the two indices used to quantify the performance of air-to-water heat pumps are:

- a. Cooling capacity
- b. Energy Efficiency Ratio (EER)

For air-to-air and water-to-air heat pumps, both of which use forced-air delivery systems, cooling capacity is divided into two parts: sensible cooling capacity and latent cooling capacity. Sensible cooling capacity is based on the temperature drop of the interior air stream passing through the heat pump’s evaporator coil. Latent cooling capacity is based on the ability of the interior coil to remove water

vapor from the air stream. However, because air-to-water and water-to-water heat pumps both deliver a stream of cool water as their output, there is only one rating for cooling capacity, which in North America is usually expressed in Btu/hr.

Cooling capacity is significantly influenced by the temperature of the air entering the heat pump’s condenser, and the temperature of water entering the heat pump’s evaporator. Cooling capacity increases when the temperature of the water delivering unwanted heat to the heat pump’s evaporator increases. Cooling capacity also increases when the temperature of the air absorbing heat from the heat pump’s condenser decreases. So, as was true for both heating capacity, and COP, the closer the source temperature is to the sink temperature, the higher the cooling capacity of the heat pump. This is shown, for a specific heat pump, in Figure 2-10.

**ENERGY EFFICIENCY RATIO**

In North America, the common way of expressing the instantaneous cooling efficiency of a heat pump is called **Energy Efficiency Ratio (EER)**, which can be calculated using Formula 2-3.

**Formula 2-3**

$$EER = \frac{Q_c}{w_e} = \frac{\text{cooling capacity (Btu/hr)}}{\text{electrical input wattage}}$$

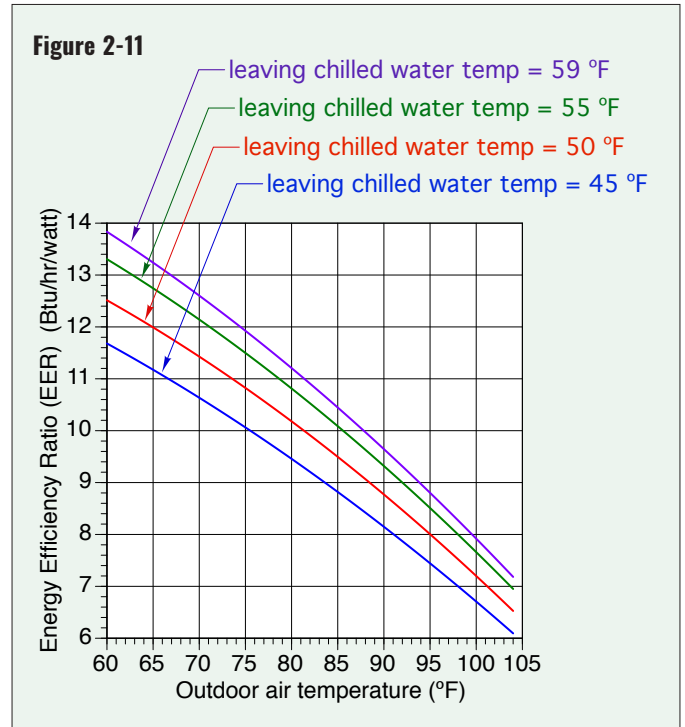
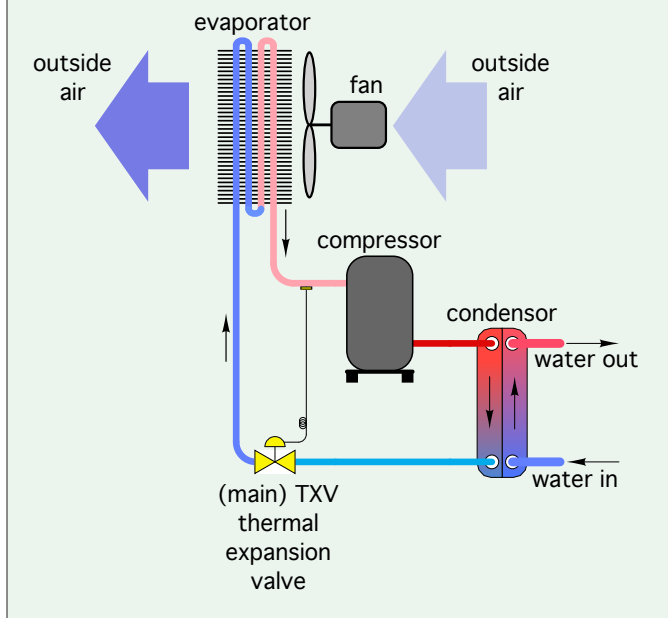


Figure 2-12



Where:

EER = Energy Efficiency Ratio

$Q_c$  = cooling capacity (Btu/hr)

$W_e$  = electrical power input to heat pump (watts)

The higher the EER of a heat pump, the lower the electrical power required to produce a given rate of cooling.

Like COP, the EER of an air-to-water heat pump depends on the source and sink temperature. The warmer the source media temperature is compared to the sink media temperature, the higher the heat pump's EER. Figure 2-11 shows how the outdoor air temperature and leaving chilled-water temperature affect the EER of a specific air-to-water heat pump.

*To maximize EER, designers of chilled-water cooling systems using either air-to-water or water-to-water heat pumps should use the highest possible chilled-water temperature that still allows adequate dehumidification. EER is also slightly influenced by flow rates. Higher flow rates of either the source media or the sink media produce small increases in EER. This is the result of increased convection on both the air-side and water-side heat exchangers.*

## WHAT DOES "TONS" MEAN?

In North America, the heating and cooling capacity of a heat pump is often stated in "tons." In this context, a ton describes a *rate* of heat flow. More specifically, 1 ton equals 12,000 Btu/hr. Thus, a "4-ton" heat pump implies a nominal heating or cooling capacity of 4 x 12,000 or 48,000 Btu/

hr. The tonnage of a heat pump has nothing to do with the heat pump's weight. The unit of "ton" originated during the transition from stored natural ice as a means of cooling to mechanical refrigeration. It represents the average heat transfer rate associated with melting one ton of ice over a 24-hour period.

A description of a heat pump heating or cooling capacity based on tons is usually a *nominal* rating at some specific set of operating conditions. Thus, a "3-ton" rated heat pump could yield a heat output rate significantly higher than 3 tons when operated under more favorable conditions, and significantly less than 3 tons when operated under unfavorable conditions.

## ENHANCED VAPOR INJECTION

One of the developments that has significantly improved the ability of air-source heat pumps to operate at low outside air temperature is called *enhanced vapor injection* (EVI). This refers to a modified refrigeration circuit that lowers the temperature of liquid refrigerant entering the outdoor evaporator when the heat pump is operating in heating mode. The lower the liquid refrigerant temperature entering the evaporator, the lower the air temperature at which the heat pump can operate. EVI also increases the refrigerant mass flow through the compressor, which helps in maintaining heating capacity at low outdoor air temperatures.

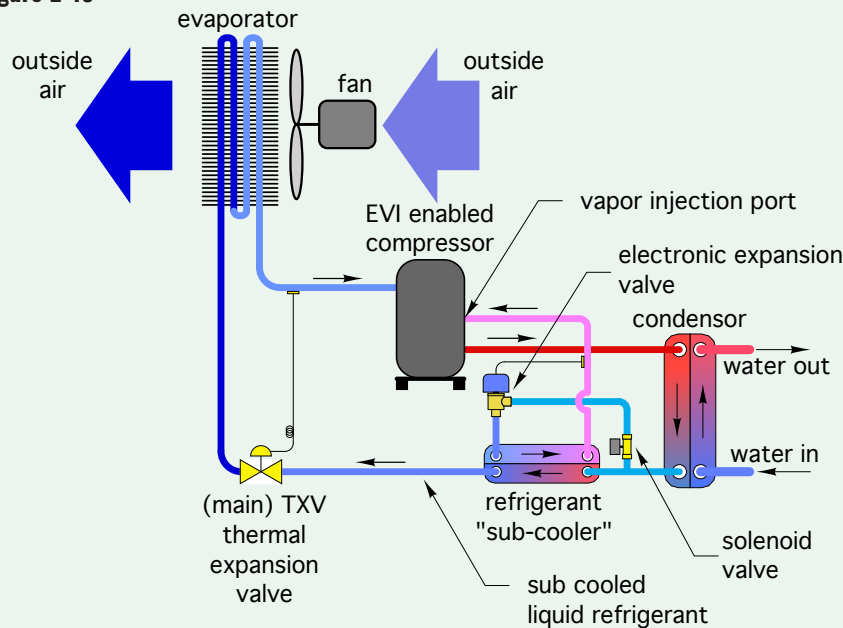
To understand EVI, it is helpful to consider a basic refrigeration circuit of a heating-only air-to-water heat system, as shown in Figure 2-12.

The temperature and liquid/vapor proportions of the refrigerant leaving the condenser, in part, determine the extent to which the thermal expansion valve can lower the refrigerant temperature entering the outdoor evaporator. This, in turn, limits the low ambient heating capacity and COP of the heat pump.

Figure 2-13 shows how the basic refrigeration circuit of Figure 2-12 is modified to allow EVI functionality.

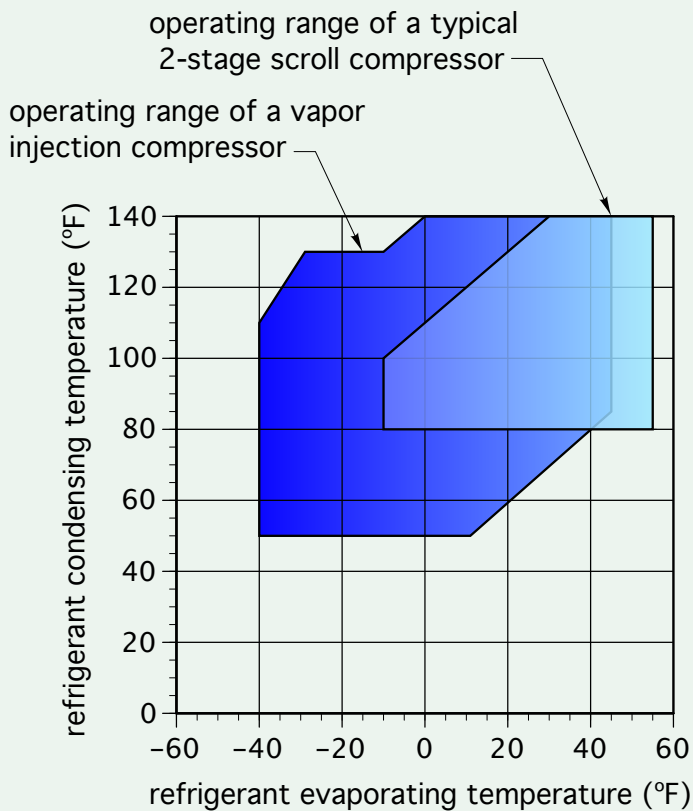
EVI works by routing the refrigerant leaving the condenser through an intermediate heat exchanger called a "sub-cooler." A portion of the refrigerant passes directly through one side of the sub-cooler. The other portion passes through an electronic expansion valve that lowers the refrigerant's pressure and temperature prior to flowing through the other side of the sub-cooler. This portion of the refrigerant evaporates in the sub-cooler, absorbing heat from the other portion. This reduces the temperature of the liquid entering the thermal expansion valve, and thus, the temperature entering the evaporator. The lower the

Figure 2-13



refrigerant temperature entering the evaporator, the better it can absorb heat from cold outside air. The vapor formed as part of the refrigerant expands within the sub-cooler is at a pressure higher than that at the suction port of the compressor. This medium-pressure vapor is routed back into the refrigeration cycle using a specially designed scroll compressor with a medium-pressure vapor injection port. The medium-pressure refrigerant vapor enters at a specific location within the scroll set. That location prevents the injected vapor from flowing toward the low-pressure side of the scroll set. This effectively increases the compression ratio beyond the mechanical ability of the compressor alone. Figure 2-14 compares the refrigerant temperature operating range of

Figure 2-14



a typical 2-stage scroll compressor versus a scroll compressor using EVI. Notice that the vapor-injected compressor can achieve much lower refrigerant evaporating temperatures. The lower the refrigerant evaporating temperature, the lower the outdoor air temperature from which useable heat can be extracted.

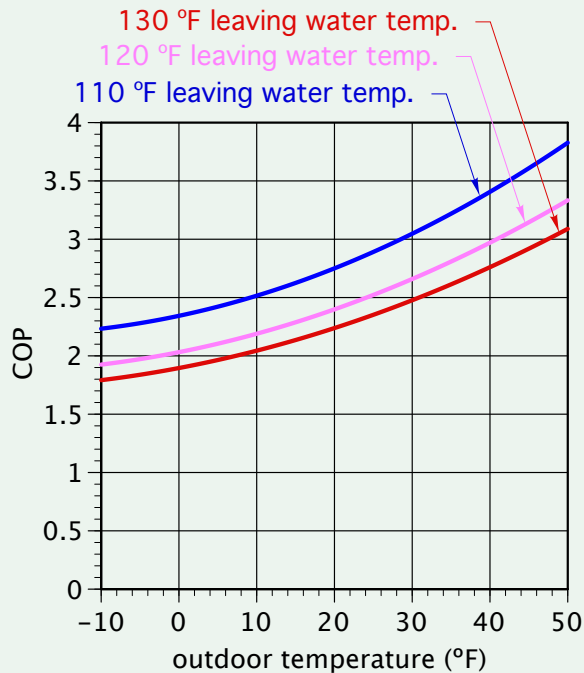
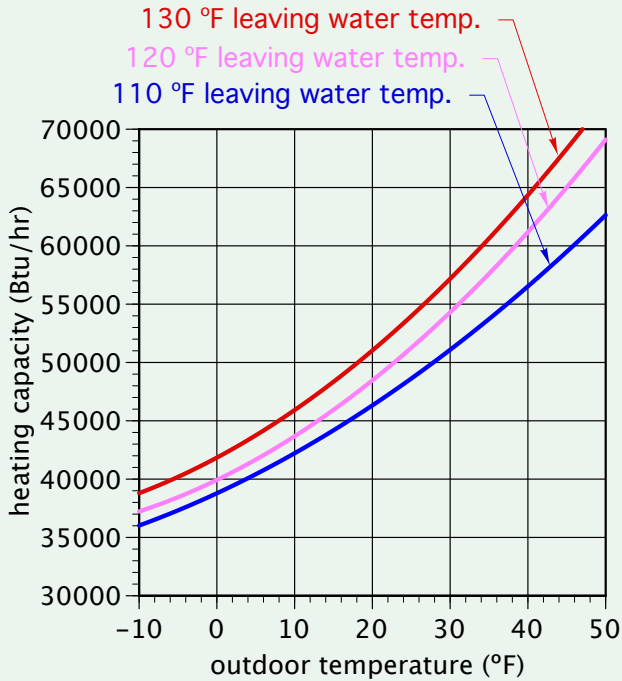
One characteristic of air-to-water heat pumps using EVI refrigeration systems is an *increase* in heating capacity as the temperature of the water leaving the condenser increases. This is shown in Figure 2-15.

This characteristic is counterintuitive because it is opposite from the decrease in heating capacity of non-EVI refrigeration systems as the water temperature leaving the condenser increases. However, as is true with non-EVI refrigeration circuits, there is a significant drop in COP as the water temperature leaving the condenser increases. Since the principal goal is to keep the heat pump's COP as high as possible, it's always best to operate the hydronic system at the lowest water temperature that maintains comfort in the heated space.

Many contemporary air-to-water heat pumps, especially those intended for use in cold climates, now use EVI refrigeration systems. These heat pumps are sometimes called "cold climate" or "low ambient" heat pumps to emphasize their suitability



Figure 2-15



for use in cold locations. Many of these heat pumps are capable of operating with reasonable performance at sub 0°F air temperatures.

## DEFROSTING

All air-source heat pumps (e.g., air-to-air and air-to-water) used in climates where outdoor temperatures drop below a nominal 50°F will, at times, accumulate frost on the outdoor air-to-refrigerant heat exchanger, which operates as the evaporator during heating mode. The rate at which frost accumulates depends on several factors, such as relative humidity, concurrent precipitation and the evaporating temperature of the refrigerant. Figure 2-16a shows an example of a heavily frosted evaporator coil on a monobloc air-to-water heat pump.

Figure 2-16b shows this evaporator partially defrosted. Figure 2-16c shows the fully defrosted evaporator, with melt water draining from the bottom of the enclosure.

As frost builds on the evaporator, airflow is reduced, which reduces the ability of the refrigerant to absorb heat from outside air. To restore heating performance, it's necessary to melt the frost off the evaporator. This is done automatically by temporarily switching the refrigerant flow direction using the reversing valve. This forces hot refrigerant gas through the evaporator, which rapidly melts the frost. In effect, the heat pump is temporarily switched to cooling mode operation while defrosting.

On most air-to-air heat pumps, the heat needed to melt frost comes from indoor air. This often results in cool air being discharged from the indoor portion of the heat pump. Although a typical defrost cycle may only last a few minutes, cool air discharging from the indoor portion of an air-to-air heat pump during cold weather is arguably a significant compromise in comfort.

However, most air-to-water heat pumps are connected to a buffer tank. Heat for defrosting comes from this tank. Even in systems without buffer tanks, the attached hydronic distribution system has much greater thermal mass relative to air, and thus, any deviation in the temperature of the distribution system during defrost is small and of short duration. In most systems, there is no detectable effect on indoor comfort. This is a significant advantage of air-to-water over air-to-air heat pumps.

Heat pump manufacturers offer different methods for defrosting. Sometimes defrosting occurs on a fixed elapsed time basis. It may also be "demand-controlled" defrost, which is usually based on low refrigerant pressure at the suction side of the compressor. Some modern air-to-water heat pumps also take the ambient air temperature into account when determining the need for defrosting. The goal is to clear the evaporator of frost with minimum required heat.

Figure 2-16a



Figure 2-16b



Figure 2-16c



As the frost melts from the evaporator, water drops from and runs to the base of the heat pump. *It is very important to ensure that this melt water doesn't accumulate at the base of the heat pump.* Accumulated melt water will eventually refreeze, and in doing so, can physically damage the evaporator, possibly even rupturing a refrigerant tube.

Most air-to-water heat pumps have a means of draining the melt water well away from the evaporator coil. Some use heated drip pans; others count on the water dripping several inches below the base of the heat pump. In cold climates, the outdoor portion of the heat pump should be mounted a few inches above expected snow depth to ensure that melt water can drain away from the unit. In cold climate applications, the outdoor portion of the heat pump should not be placed on a solid surface low to the ground.

# ENERGY TRENDS & THEIR EFFECT ON HYDRONIC HEAT SOURCES

The market for hydronic heat sources, like many products that use energy to provide a benefit, is shaped by public perception, government policy and economic competition. As the 21st century unfolds, social attitudes and government policies around the world are increasingly focused on climate change, with a prevailing emphasis on reducing carbon emissions.

There are widely varying opinions on how “decarbonization” should be dealt with. They range from complete dismissal of any need to act on the subject, to proposals that would radically change how an average person would eat, remain comfortable in their home or workplace, travel, or even use their leisure time.

This issue of *idronics* is not meant to endorse any specific view on how carbon reduction should be dealt with. However, one objective of *idronics* is to provide information on *trends* that are likely to influence the market in which HVAC professionals work. To that end, *it appears highly likely that the energy used by future hydronic heating and cooling systems will be increasingly supplied through electricity and less by the burning of fossil fuels.*

There are several observations and market trends that support this statement. They include:

### 1. Government policy supporting decarbonization:

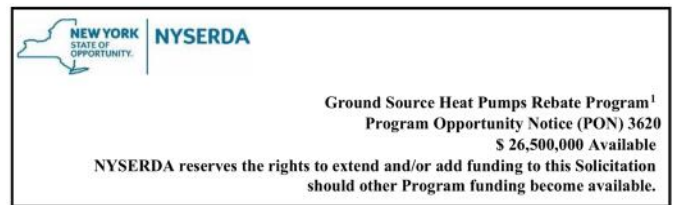
Currently there are expanding government policies that discourage use of carbon-based fuels in favor of electricity. These policies are being implemented by a wide spectrum of government agencies, including federal, provincial, state and municipal. *They are based on the premise that an increasing percentage of electricity will be generated by renewable sources such as utility-scale solar photovoltaic systems, wind energy farms and hydroelectric facilities.* These policies are having an effect. There is steady growth in the amount of renewably sourced electricity available to the North American power grid. Since 2010, solar electrical generating capacity in the U.S. has increased 80-fold. Electricity generated by utility-scale wind farms has tripled over the same period. Electrification is also being increasingly implemented in the transportation sector.

### 2. Financial incentives:

Financial incentives have been implemented by government agencies and private utilities to encourage use of electric heat pumps. Past experience consistently proves that markets adapt to leverage these incentives *while they are available.* That experience also proves that the phaseout and eventual elimination of these financial incentives can lead to rapid market adjustments.

**3. Strong interest and impressive market growth for “net-zero” buildings:** Although there are several ways that “net-zero” can be defined, most consumers understand the concept as a building that produces all the energy it requires to support an average lifestyle, provide an adequate work environment or otherwise allow the building to sustain its intended function. Since virtually all buildings have electrical loads, some of the energy has to be produced on site, usually by solar photovoltaic systems, or by a “community” solar cooperative using a larger-scale solar photovoltaic system.

Figure 3-1



Net zero housing units within the U.S. increased by 59% from 2017 to 2018. Over that same period, the increase in net zero housing units in Canada was 240%. Multi-family projects currently represent 71% of the total net zero housing stock. One market research firm anticipates that the global market for net zero buildings will increase at a compound annual growth rate of 39% by 2021.

The only practical approach to achieving net zero is through use of electricity generated by renewable sources such as solar photovoltaic systems. Unlike burning fossil fuels, which is a non-reversible form of energy conversion, electricity can be manipulated in many ways and over many “scales.” Electricity generated at a building site can be accurately metered back to the utility grid and instantly used by any of the millions of electrical loads served by that grid. Techniques

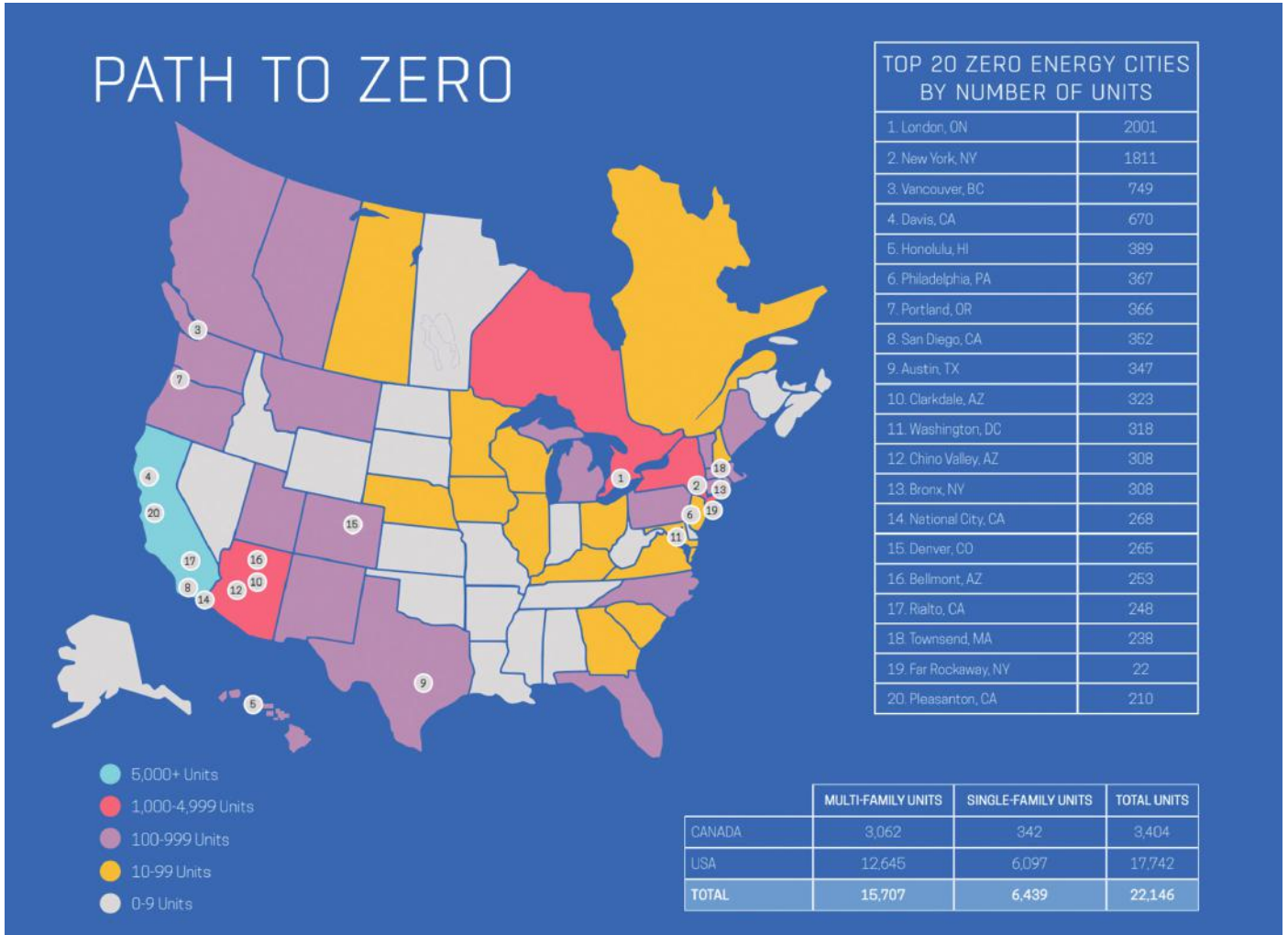
Figure 3-2



Courtesy of Revision Energy



Figure 3-3



such as net metering, real-time pricing, time of use rates, and the aggregation of thousands of residential battery storage systems into a “virtual utility” provide flexibility in how electrical energy can be favorably managed. This flexibility, and its associated benefits, are far beyond those which could be practically implemented using fossil fuels.

**4. Decreasing heating and cooling loads:**

Residential building and energy codes continue to mandate lower heating and cooling loads. Reduced residential heating and cooling loads are also encouraged through programs such as ENERGY STAR®, Certified Passive House and R-2000 (Canada). Reduced heating and cooling loads in commercial structures are encouraged through programs such as LEED (Leadership in Energy and Environmental Design).

Residential design heating loads in the range of 10 to 15 Btu/hr per square foot of floor area are becoming more common in new construction or deep energy retrofits. Thus, a modern 2,000 square foot home could have

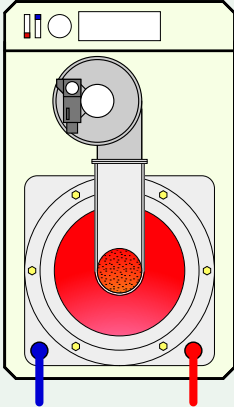
Figure 3-4





**Figure 3-5**

gas-fired  
modulating /  
condensing boiler



boiler model	MAXIMUM HEAT OUTPUT (Btu/hr)	MINIMUM HEAT OUTPUT (Btu/hr)
model 55	51,000	7,700
model 85	79,000	7,900
model 110	102,000	10,200
model 155	144,000	14,400
model 200	185,000	18,400
model 285	264,000	26,400
model 399	377,000	75,600

**Figure 3-6**

<b>GAS RESIDENTIAL</b>			
Meter Number: [REDACTED]			
Dec 19 reading (Actual)		9524	
Nov 17 reading (Actual)		-9349	
		<hr/>	
Total Usage CCF 32 days		175	
<b>Delivery Charges</b>			
Basic Service Charge			<b>\$20.00</b>
Includes	3CCF	@ 0.000¢ each	\$00.00
Next	47 CCF	@ 65.285¢ each	\$30.68
Next	125 CCF	@ 62.835¢ each	\$78.54
Month Gas Adj		@ 12.22455¢ each	\$21.39
NY Assessment	175 CCF	@ 0.96528¢	\$1.69
RDM Adjustment	175 CCF	@ 0.72803¢	\$1.27
SBC Charge	175 CCF	@ 0.51500¢	\$0.90
Government surcharges - Delivery			\$5.22
<b>Total Delivery Charges</b>			<b>\$159.69</b>
Merchant Function Chg	175 CCF @ 2.9453¢		\$5.15
Government surcharges - Delivery			\$0.17
<b>Merchant Function Charges</b>			<b>\$5.32</b>
<b>Gas Supply Chg</b>	<b>175 CCF @ 33.00655¢</b>		<b>\$57.75</b>
Government surcharges - Commodity			\$0.75
Total Supply Charges			\$58.51
<b>CURRENT GAS CHARGES</b>			<b>\$223.52</b>

a design heating load of only about 20,000 Btu/hr. This is about 1/3 the design load of a typical 2,000 square foot house constructed during the 1980s.

**5. Lack of suitable combustion-based heat sources:**

As building design heating loads continue to drop, it becomes difficult (or impossible) to find combustion-based heat sources that would not be significantly oversized for those loads. The smallest gas-fired mod/con boiler currently available has a rated output of approximately 50,000 Btu/hr and a turndown ratio of about 6.7 to 1. Even at its lowest stable firing rate, the heat output from this boiler would provide 39% of a 20,000 Btu/hr *design* heating load. Oversized combustion-type heat sources tend to short cycle which lowers their net efficiency. By comparison, there are more selections for electrically operated heat pumps that can better match lower heating loads.

**6. Basic service charges & pricing for natural gas:**

For houses with natural gas service, the monthly basic service charge associated with having the gas meter becomes an increasingly larger percentage of the gas invoice as the building's design heating load decreases. For modest homes, constructed to ENERGY STAR® or other contemporary energy conservation standards, the total monthly service charges may even exceed the cost of the gas consumed.

Many gas utilities also charge a higher rate per CCF (100 cubic feet) for some initial quantity of natural gas, with a decreasing rate as usage increases. This tends to increase the average price per CCF of gas required for homes with low heating loads.

Both of these factors can be seen in the residential gas service invoice in Figure 3-6.

The basic service charge and sliding price scale discourage use of natural gas in low-load homes in favor of an “all-electric” home. For rural houses, the leasing or amortized purchase cost of a large propane storage tank adds to monthly energy bills. These costs are eliminated when the only metered energy source is electricity.

**Figure 3-7**



**Figure 3-8**



## **7. Safety issues:**

Considering that they are used in tens of millions of buildings, natural gas and fuel oil have relatively good safety records. Still, natural gas is flammable and potentially explosive under the right conditions. A leak in a residential fuel oil tank, if not immediately detected and corrected, can require environmental remediation that costs tens of thousands of dollars. Furthermore, any combustion process produces some amount of carbon monoxide, which if leaked into buildings in sufficient quantities can cause asphyxiation. The choice of fuel can also affect insurance rates. Insurance companies often require higher premiums for homes using fuel oil or firewood for home heating. These potential concerns are not present when electrically operated heat pumps provide heating and cooling.

## **8. Moratoriums on natural gas expansion:**

In some highly populated areas, an increase in building construction, combined with conversion of oil-based heating to natural gas in existing buildings, has increased demand for natural gas. This has strained the ability of existing gas distribution networks, causing some U.S. utilities to enact (or threaten to enact) moratoriums on any expansion of natural gas service in their territory. This has forced developers to look at alternatives for heating and cooling. Electrically operated heat pumps are an attractive alternative.

All of these trends and circumstances are generally beyond what an individual, or even a sizable HVAC corporation, can control. Without some degree of adaptation to the changing energy landscape, it will be difficult or impossible to meet customer expectations, comply with codes and regulations, or work with future product offerings.

The North American hydronic heating and cooling market will be affected by these trends and circumstances. Increasing use of electrical energy will support the growth of heat pumps as hydronic heating and cooling sources. When paired with hydronic distribution systems, both water-to-water and air-to-water heat pumps offer the benefits of low operating cost, high distribution efficiency, design flexibility and unsurpassed comfort. Of these two types, the air-to-water heat pump offers several compelling advantages.

## **ADVANTAGES OF AIR-TO-WATER HEAT PUMPS**

### **1. Significantly lower installation cost:**

Air-to-water heat pumps can be typically be installed at costs of 30 to 50% those of equivalent capacity and unsubsidized geothermal heat pump systems. The latter require extensive excavation (for installing horizontal earth



Figure 3-9



Source of drilling photo: thegeoexchange.org



VS.

Figure 3-10



loops) or extensive drilling (for installing vertical earth loops). Although costs vary regionally and with specific site conditions, current estimated vertical earth loop installation costs range from about \$2,000 to \$3,500 per ton of heat pump capacity. At an average of \$2,750 per ton of capacity, a typical 4-ton rated vertical earth loop field would cost \$11,000. This is exclusive of the heat pump and any other interior portion of the system. In contrast, a typical air-to-water heat pump is installed on a small support stand adjacent to the building it serves. The total exterior installation cost, excluding the heat pump, is likely in the range of \$500-800. The cost of the geothermal earth loop is eliminated, and thus a major portion of the installation cost associated with a geothermal heat pump is avoided.

### 2. Noninvasive installation:

Installing geothermal earth loops is highly disruptive to existing landscapes. Although the equipment needed for horizontal and vertical earth loop installations is available in most areas of North America, the use of that equipment, especially on established landscapes, is very invasive. Complete restoration of established landscapes can take weeks or months. The cost of such restorations should be included as part of the geothermal heat pump installation.

### 3. Ground water protection:

Laws intended to protect ground water can impact the installation of geothermal earth loops. Deep boreholes have the potential to reach ground water aquifers. Surface runoff or spillage of potentially toxic materials during installation is a possibility, as is surface seepage into boreholes that are not properly sealed. Although proper techniques minimize these threats, they should be evaluated and factored into the logistics and cost of earth loop installations. These factors are not present when an air-to-water heat pump is installed.

After installation there remains a possibility, albeit small, that the earth loop could be damaged by future excavations. Although obviously unintentional, these situations have occurred due to lack of accurate mapping of where the buried piping is located. Spillage of the antifreeze solution from ruptured piping into surrounding soil is possible. When non-toxic antifreeze such as propylene glycol is used, it's unlikely that such spillage would warrant emergency actions. However, methyl alcohol and ethylene glycol,

**Figure 3-11**



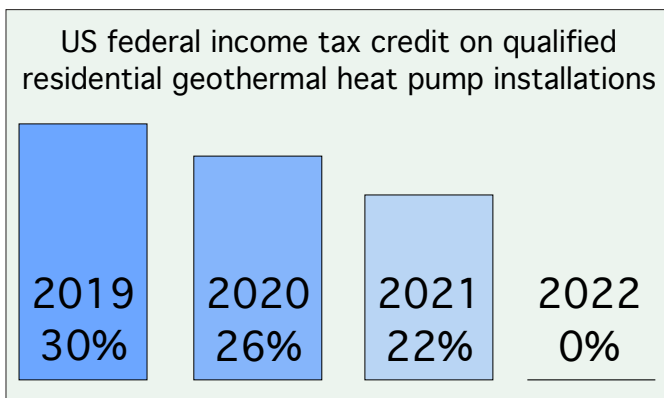
Courtesy of REHAU

both of which have been used as antifreezes in earth loop systems, are toxic materials. Spillage of many gallons of these materials could require legally mandated remediation.

**4. Independent of incentives:**

In most areas of the U.S. and Canada, air-to-water heat pumps are *not* currently subsidized through utility or government incentives. This is in contrast to geothermal heat pump systems, which are significantly subsidized in many areas of North America. Although the availability of government incentives that underwrite a significant portion of the installation of geothermal heat pumps seems like a decisive *advantage*, it is a *temporary* condition that artificially skews the market. As this issue of *idronics* is being

**Figure 3-7**



written, the U.S. federal income tax credit on geothermal heat pump systems stands at 26% of qualified geothermal equipment installation cost. This is down from 30% in 2019. Current tax law will reduce this credit to 22% through the end of 2021. At that point, barring legislation that would extend these tax credits, they will end. The phaseout of these tax credits will significantly increase the installed cost and decrease the economic merit of geothermal heat pump systems relative to alternatives. This will improve the life cycle economics of air-to-water compared to those of geothermal heat pumps.

**5. Higher net COP:**

Both geothermal water-to-water heat pumps and air-to-water heat pumps should be evaluated based on “net COP” rather than the COP of the heat pump as a standalone device. Net COP is calculated by dividing the rate of heat output by the total electrical power required to run the heat pump *and its associated circulators*. Net COP is a better indicator of the true operating cost of either type of hydronic heat pump since the power required by the required circulators is factored in.

For a geothermal water-to-water heat pump, this would include the circulator(s) used to create flow in the earth loop and the circulator that provides flow through the load side of the heat pump. Some earth loops require multiple circulators connected in series to provide sufficient head and flow. The total power required by these circulators can be several hundred watts. Both the geothermal water-to-water heat pump and an air-to-water heat pump require a load side circulator.

To see the effect of net COP, consider a geothermal water to water heat pump with a heating capacity of 48,000 Btu/hr. The heat pump compressor draws 4.2 KW when operating. Its earth loop requires two circulators, piped in series, operating at 220 watts each. The load side circulator draws 180 watts. Calculate the COP of the heat pump itself, and the net COP of the heat pump with its associated circulators.

The COP of the heat pump as a standalone device is:

$$COP_{HPonly} = \frac{48,000 \text{ Btu / hr}}{[4200] \text{ watt} \times 3.413 \frac{\text{Btu / hr}}{\text{watt}}} = 3.35$$

The net COP of the heat pump and its associated circulators is:

$$COP_{net} = \frac{48,000 \text{ Btu / hr}}{[(2 \times 220) + 4200 + 180] \text{ watt} \times 3.413 \frac{\text{Btu / hr}}{\text{watt}}} = 2.92$$



In this example, the net COP is about 13% lower than the COP of the heat pump as a standalone device.

Assume that an air-to-water heat pump has an output of 48,000 Btu/hr, with a corresponding electrical input of 5.5 KW. The load side circulator requires 180 watts. Calculate the COP of this heat pump alone and the net COP of the heat pump and its load side circulator.

The COP of the air-to-water heat pump alone is:

$$COP_{HP\text{only}} = \frac{48,000 \text{ Btu / hr}}{[5500]_{\text{watt}} \times 3.413 \frac{\text{Btu / hr}}{\text{watt}}} = 2.56$$

The net COP of the heat pump and its load side circulator is:

$$COP_{\text{net}} = \frac{48,000 \text{ Btu / hr}}{[5500 + 180]_{\text{watt}} \times 3.413 \frac{\text{Btu / hr}}{\text{watt}}} = 2.48$$

In this example, the net COP is only about 3% lower than the COP of the air-to-water heat pump as a standalone device. Even though the air-to-water heat pump draws an additional 1.3 kilowatts of electrical power compared to the geothermal heat pump, its net COP, in this example, is only about 12% lower than that of the geothermal heat pump. It's also important to remember that the COP of any heat pump is highly dependent on its operating conditions, and thus, there are scenarios where the net COP of an air-to-water heat pump could exceed the net COP of a geothermal water- to-water heat pump.

## 6. Savings decrease as loads decrease:

As the design heat loss of buildings decreases, so do the energy cost savings associated with a heat pump system having a "high" seasonal COP relative to those of a heat pump system with a "lower" COP.

The seasonal COP of a heat pump is a weighted average value that accounts for the wide range of operating conditions that a specific heat pump undergoes during an entire heating season, in a specific system at a specific location. Seasonal COP is typically estimated using computer simulation and is discussed in more detail in section 5.

Formula 3-1 can be used to calculate the annual heating energy reduction realized when a heat pump with a higher seasonal coefficient of performance is used instead of a heat pump with a lower seasonal coefficient of performance.

### Formula 3-1

$$E_s = E_r \left[ \frac{1}{COP_L} - \frac{1}{COP_H} \right]$$

Where:

$E_s$  = energy savings (MMBtu/yr)

$E_R$  = seasonal space heating energy required by building (MMBtu/hr)

$COP_L$  = seasonal average coefficient of performance of lower-performing heat pump

$COP_H$  = seasonal average coefficient of performance of higher-performing heat pump

Consider a 2,000 square foot house with a design heating load of 15 Btu/hr/ft<sup>2</sup>. The design heating load is 30,000 Btu/hr. Assume that this load corresponds to an interior temperature of 70°F and an outdoor design temperature of 5°F. The house is located in a relatively cold 6000°F•day climate. It's estimated annual space heating energy requirement is 39.9 MMBtu/yr (1 MMBtu = 1,000,000 Btu). The house is equipped with a low-temperature hydronic heating distribution system. A computer simulation of a geothermal water-to-water heat pump systems indicates an average seasonal COP of 3.4. Another simulation for a low ambient air-to-water heat pump system projects a seasonal COP of 2.5. Estimate the annual energy saved by using the geothermal heat pump rather than the air-to-water heat pump. Also estimate the cost savings assuming that the local cost of electricity is \$0.11 per kilowatt-hour.

The energy saved is easily calculated using Formula 3-1:

$$E_s = E_r \left[ \frac{1}{COP_L} - \frac{1}{COP_H} \right] = 39.9 \left[ \frac{1}{2.5} - \frac{1}{3.4} \right] = 4.22 \text{ MMBtu / season}$$

If the air-to-water heat pump was used, the additional 4.22 MMBtu/season required would have to be supplied by electricity. At a flat rate of \$0.11 per KWHR the additional energy would cost:

$$4.22 \text{ MMBtu} \left( \frac{293 \text{ KWHR}}{\text{MMBtu}} \right) \left( \frac{\$0.11}{\text{KWHR}} \right) = \$136 / \text{season}$$

This is certainly a non-trivial savings, especially considering that it is a recurring annual cost that could escalate if the price of electricity increases. However, this energy cost should also be considered in the context of spending several thousands of dollars more to install the geothermal heat pump earth loop, especially after installation subsidies are no longer available. Based on a previously mentioned cost estimate, a vertical earth loop for a 2.5-ton rated geothermal heat pump would cost between \$5,000 and \$8,750. At an average estimated cost of \$6,875, an annual savings of \$136 would yield a simple payback of 50.6 years. That is about twice the life expectancy of current generation geothermal heat pumps.

This example demonstrates that even though the geothermal heat pump has a higher seasonal COP compared to the air-to-water heat pump — 36% higher in this case — the annual energy cost savings of operating the geothermal heat pump in a modern energy-efficient house is such that the higher installed cost cannot be recovered in a reasonable time. This example is not meant to imply that similar economic results occur in all situations. Each must be evaluated based on specific heating loads, estimated performance for the installed systems and local cost of electrical energy.

It should also be mentioned that cooling costs were not factored into this comparison since the location was assumed to be a relatively cool climate with minimal cooling requirements. Any cooling load would be met at a slight lower cost by the geothermal heat pump compared to the air-to-water heat pump, but the associated savings would be very small.

### **7. Influence of global markets:**

Although they currently represent a small portion of the heat pump market in North America, air-to-water heat pumps are widely used in Asia and Europe.

The Japanese publication JARN reported that the European market for air-to-water heat pumps reached

368,900 units in 2018, which represented a 14.3% increase over the previous year. France and Germany are the two largest markets. Split system air-to-water heat pump configurations represent about 70% of the overall market. Several European companies that are well known as boiler manufacturers have made significant investments in air-to-water heat pump technology and now offer a range of products.

JARN also reported the following number of air-to-water heat pump installations during 2018: China, 1,280,000 units, and Japan, 475,000 units.

Heat pump sales in Europe are incentivized by directives aimed at carbon reduction. In China there are also incentives aimed at converting coal-fired heating systems to electricity.

JARN did not report an estimate for the number of air-to-water heat pumps (for space heating and cooling applications) for the U.S. or Canada. The number of air-to-water heat pumps installed for building heating and cooling in the U.S. and Canada during 2018 is very small in comparison to the European and Asian market, perhaps under 1,000 units. Still, past trends indicate that many aspects of hydronics technology migrate from Asian and European markets into North America.

# AIR-TO-WATER HEAT PUMP CONFIGURATIONS

Most air-to-water heat pumps have an outside unit, similar to that of an air-to-air heat pump or central air-conditioning system. However, the heat generated while operating in the heating mode is delivered to a hydronic distribution system within the building. When operating in the cooling mode, air-to-water heat pumps deliver a stream of chilled

water, or chilled antifreeze solution, that flows to the balance of the system. The use of a hydronic distribution system for heating and cooling creates many possibilities that are not possible with forced-air distribution systems.

**Figure 4-1**



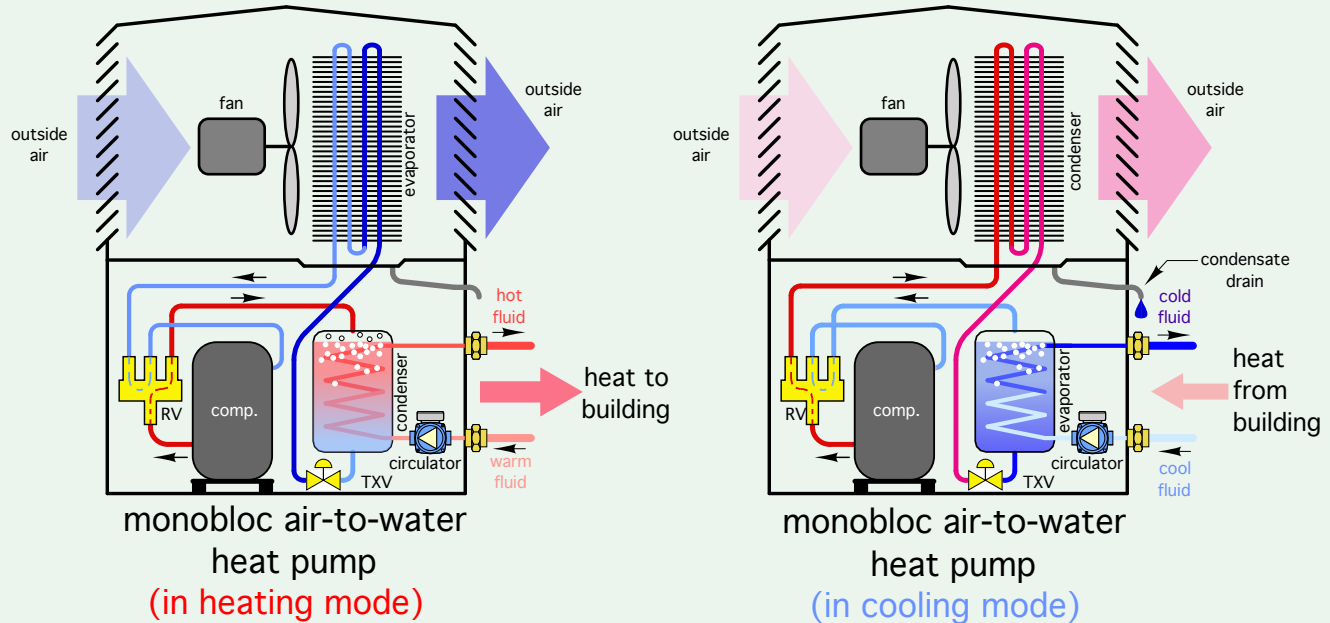
## MONOBLOC AIR-TO-WATER HEAT PUMPS

Figure 4-1 shows a modern “monobloc” air-to-water heat pump. The insulated pipes, seen penetrating the building wall behind the unit connect to the interior portion of the system.

When operating, variable-speed fans pull outside air across the air-to-refrigerant heat exchanger on the rear of the unit. In heating mode, this heat exchanger serves as the evaporator. Low-temperature heat is absorbed from the air, and the cooled air is discharged through the fan grills at the front of the unit. In cooling mode, the same heat exchanger serves as the condenser for dissipating heat to outside air.

Monobloc air-to-water heat pumps house all the refrigeration system components and most of the electrical controls within the outdoor unit. Some monobloc heat pumps also contain hydronic components such as a circulator, expansion tank or flow verification switch. Monobloc heat pumps are supplied pre-charged with the correct amount of refrigerant, which implies that they can be installed without need of refrigeration service equipment. Figure 4-2 illustrates the basic internal components in a monobloc heat pump as they would operate in heating and cooling modes.

**Figure 4-2**



Systems designed with monobloc air-to-water heat pumps vary depending on the severity of the winter climate. In mild climates, where outdoor temperatures below freezing are rare, it is usually acceptable to install the heat pump with water in the piping circuit between the outdoor unit and indoor distribution system. Most heat pumps of this type have a controller that automatically turns on a circulator and an electric heating element when necessary to protect the unit from freezing. These components can add enough heat to maintain the water-filled portion of the heat pump above freezing, even when there is no load calling for heat pump operation.

However, in situations where significant temperature setbacks or prolonged power outage are possible during subfreezing outdoor temperatures, and no backup generator capable of running the heat pump is available, it is advisable to install the unit as part of an antifreeze-protected hydronic circuit. Some manufacturers mandate that antifreeze solutions be used in *all* installations of their monobloc air-to-water heat pumps.

### FREEZE PROTECTION OPTIONS FOR MONOBLOC HEAT PUMPS

All systems designed around monobloc air-to-water heat pump systems need to consider freeze protection. Even systems in southern states could experience a situation in which a prolonged power outage accompanied by subfreezing temperatures could allow water to freeze within the exterior piping and heat exchanger. Another possibility is when the heat pump is not operating due to a service issue while subfreezing temperatures occur.

### ANTIFREEZE-BASED FREEZE PROTECTION

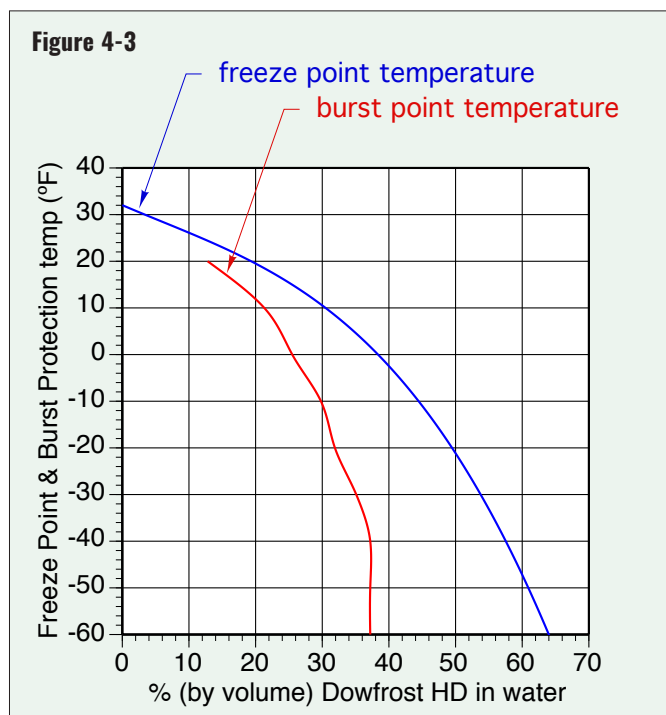
One of the simplest methods of preventing freezing is to fill the entire system with an antifreeze solution of adequate concentration to prevent damage to the system under the coldest expected temperatures. The amount of antifreeze needed depends on the expected function of the solution during cold weather. Is that function to allow flow through the system under the coldest expected temperature, or is it to prevent bursting of one or more components due to expansion of the solution when it freezes solid? This requires differentiation between the freeze point and burst point temperatures.

The *freeze-point* temperature of an antifreeze solution is the minimum temperature at which the solution remains flowable. Small ice crystals are just beginning to form in the fluid when it drops to its freeze-point temperature. This temperature is well below the normal operating temperature of a typical air-to-water heat pump. However, if it is possible that the heat pump may have to start after being off for several hours in very cold ambient conditions, the antifreeze

solution used should have a freeze-point temperature as low as the minimum ambient air temperature at which the cold start could occur.

The *burst-point* temperature of an antifreeze solution is the lowest temperature at which the piping and piping components that contain the solution will not be subject to expansion forces that could rupture them. The antifreeze solution will be mostly ice crystals when it drops to burst point temperature, and thus, *not* flowable.

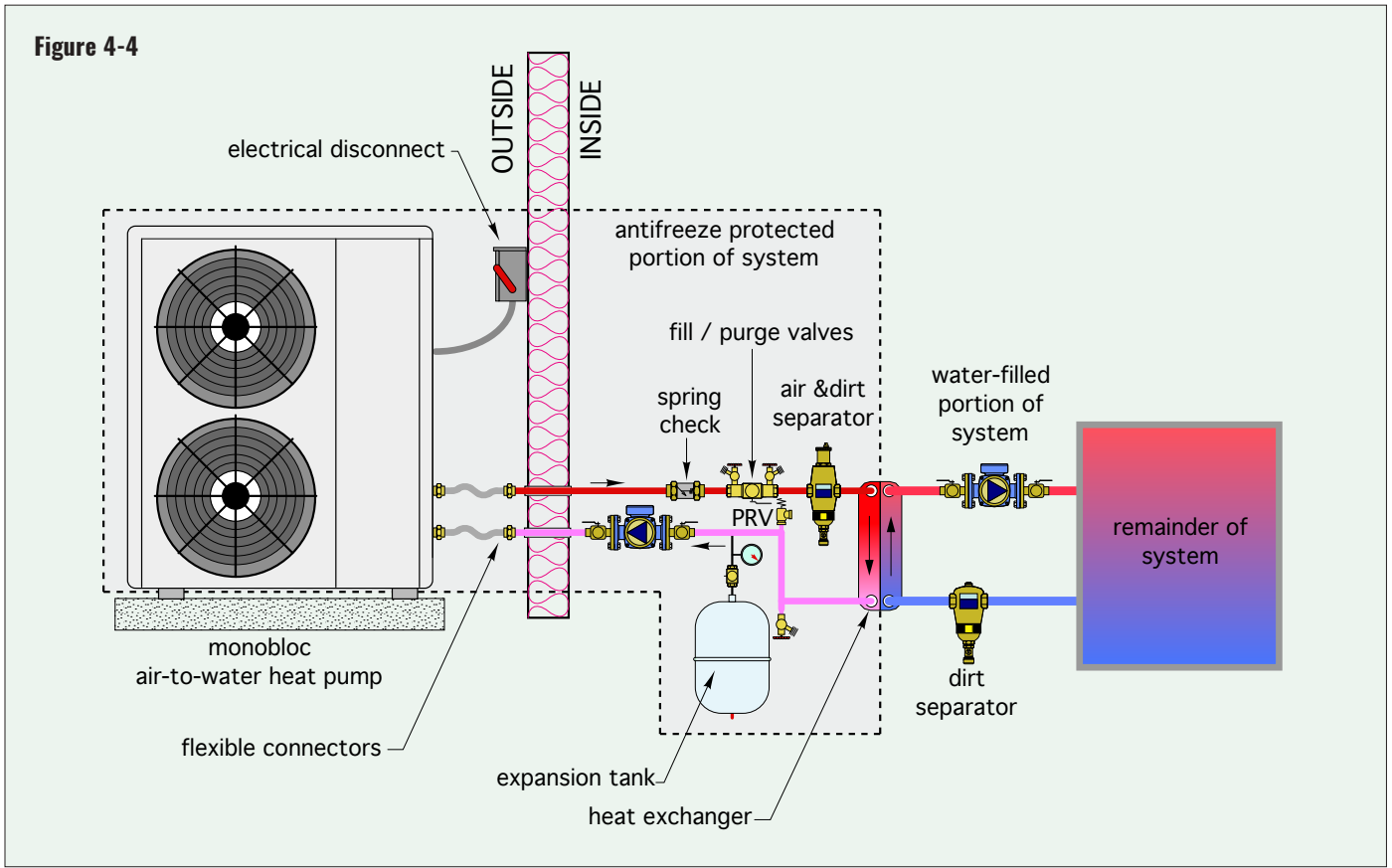
Figure 4-3 compares the freeze protection temperature and burst protection temperature for a range of volumetric concentrations of an inhibited propylene glycol antifreeze.



Based on Figure 4-3, a 35% solution of propylene glycol antifreeze remains flowable down to 5°F and protects the heat pump and piping against bursting down approximately -30°F. A 50% solution of the same antifreeze remains flowable down to -20°F and provides burst protection to temperatures below -60°F. The objective is to select a concentration that adequately protects the system but doesn't use excessive amounts of antifreeze. The higher the antifreeze concentration, the higher the viscosity of the fluid, and the greater the flow resistance of the hydronic circuit. Adding antifreeze to water also decreases the heat transfer capacity of the solution. A 50% solution of propylene glycol antifreeze lowers the specific heat of the solution to approximately 90% that of water. This can be compensated for by using higher flow rates, but that can significantly increase circulator power requirements.



Figure 4-4



Setting up the entire system to operate with an antifreeze solution has advantages and disadvantages. One advantage is the elimination of any heat exchanger between the heat pump and balance of the system. A heat exchanger is required in situations where the heat pump circuit operates with an antifreeze solution while the balance of the system operates on water. The presence of a heat exchanger between the heat pump and balance of the system forces the heat pump to operate at higher temperatures compared to where it would operate without the heat exchanger. This is necessary to create a temperature difference across the heat exchanger that's adequate to transfer heat from the antifreeze solution to the water at the rate at which the heat pump supplies heat. Operating the heat pump at higher fluid temperatures lowers its COP. This increases the amount of electrical energy required per unit of heat delivered.

Another advantage of operating the entire system with antifreeze is that the entire system is protected in the event of prolonged malfunction or power outage during cold weather.

One disadvantage of adding antifreeze is higher power requirements for circulators relative to those required for water only. A 50% solution of propylene glycol antifreeze requires approximately 60% higher circulator input power

to achieve the same heat conveyance rate (e.g., the combined effect resulting from changes in specific heat, viscosity, density and flow rate). Adding antifreeze to systems with large volumes such as those with extensive radiant panel circuits and/or large buffer tanks increases cost. Systems with antifreeze should also be checked annually to ensure that the pH buffers in the antifreeze are adequate to prevent thermal breakdown.

Another option for freeze protection is to add a heat exchanger between the heat pump and remainder of the system. This is illustrated in Figure 4-4.

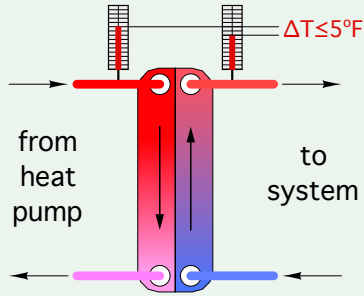
The piping circuit between the heat pump and the primary side of the heat exchanger is filled with an antifreeze solution. The secondary side of the heat exchanger and the balance of the system operate with water.

If a heat exchanger will be used, it is essential that it is generously sized. One suggested criterion is to select a heat exchanger that can pass the full rated heat output of the heat pump without exceeding an approach temperature difference of 5°F, as shown in Figure 4-5.

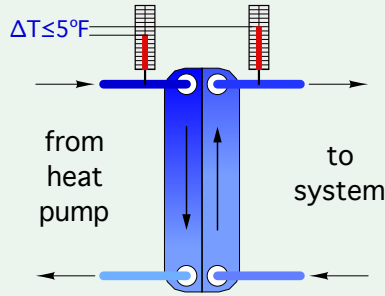
For heating mode operation, the approach temperature difference is calculated by subtracting the temperature

**Figure 4-5**

maximum approach temperature difference  
(heating mode)



maximum approach temperature difference  
(cooling mode)



**Figure 4-6**



It also adds to operating cost. Use of high-efficiency ECM-circulators helps minimize the total electrical energy required for pumping.

In situations where the heat pump will provide cooling, it is necessary to fully insulate the heat exchanger to prevent surface condensation. Figure 4-6 shows an example of a relatively large brazed plate stainless steel heat exchanger that is used between a nominal 4-ton air-to-water heat pump and the balance of the system.

The heat exchanger shown in Figure 4-6 uses plates measuring 5 inches

wide and 12 inches tall. There are 100 plates “stacked” together to make this heat exchanger. It is capable of transferring 60,000 Btu/hr with an approach temperature difference of 5°F. The heat exchanger is completely wrapped with a layer of 1/2-inch elastomeric foam insulation. The insulation is sealed to the piping at all connections. All piping to and from the heat exchanger will eventually be covered with elastomeric foam pipe insulation and sealed to prevent surface condensation when operating in cooling mode.

### SPLIT SYSTEM AIR-TO-WATER HEAT PUMPS

Another common configuration for air-to-water heat pumps is known as a “split system.”

Some of the refrigeration system components are located in the outdoor unit, while the remainder are located inside the building. Two copper tubes transport refrigerant between the inside and outside units. Figure 4-7 shows one example of an outdoor unit and its matching indoor unit. The refrigeration tubing routed between these units is covered with white insulation.

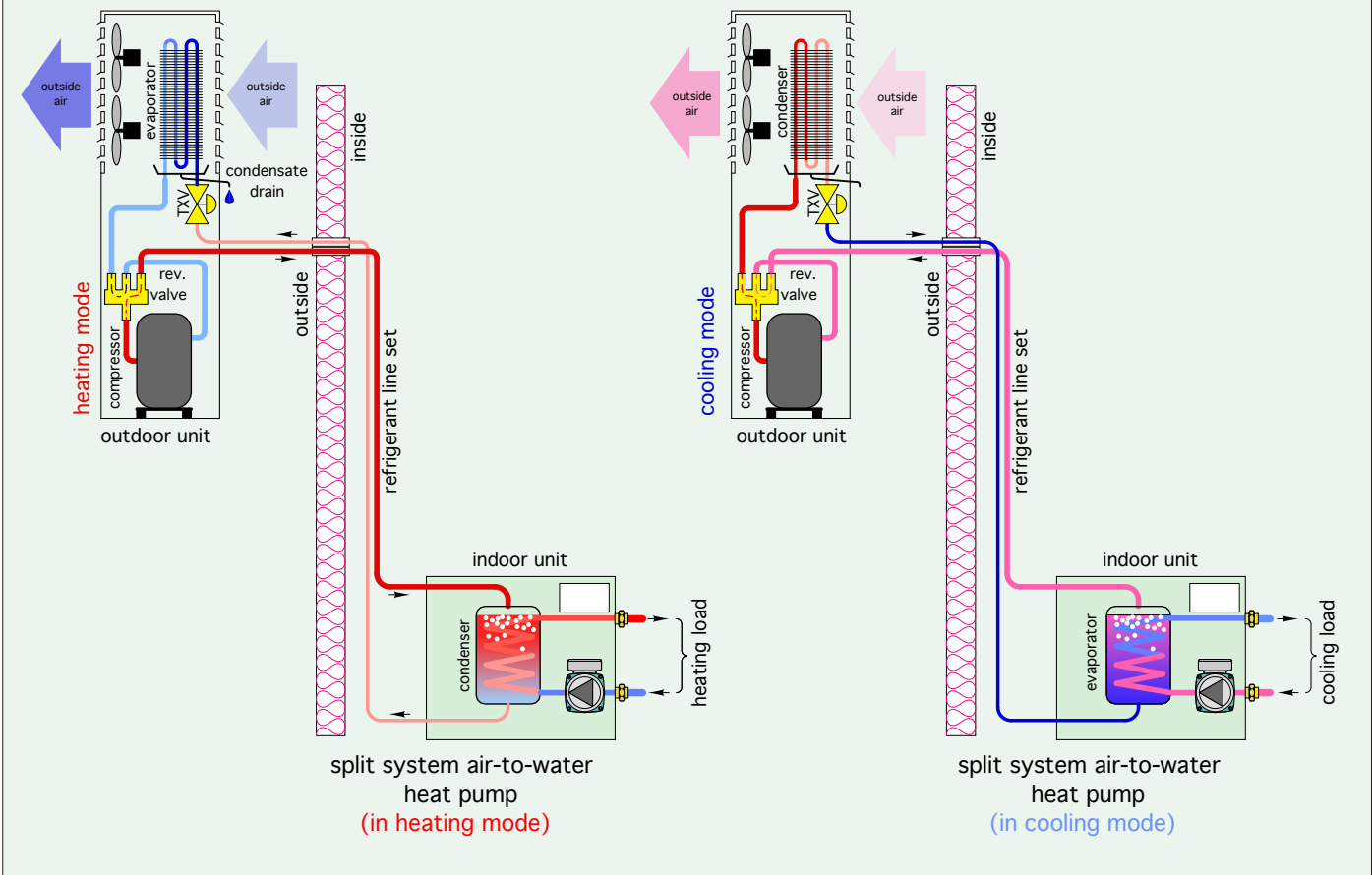
of the water leaving the secondary side of the heat exchanger from the temperature of the antifreeze entering the primary side of the heat exchanger. The greater the internal area of the heat exchanger, the lower the approach temperature difference will be for a given rate of heat transfer. When practical and cost effective, approach temperature differences lower than 5°F are beneficial.

The use of a heat exchanger between the heat pump and balance of the system also requires two circulators: One between the heat pump and heat exchanger, and the other between the heat exchanger and balance of the system. This adds to installation cost.

**Figure 4-7**



Figure 4-8



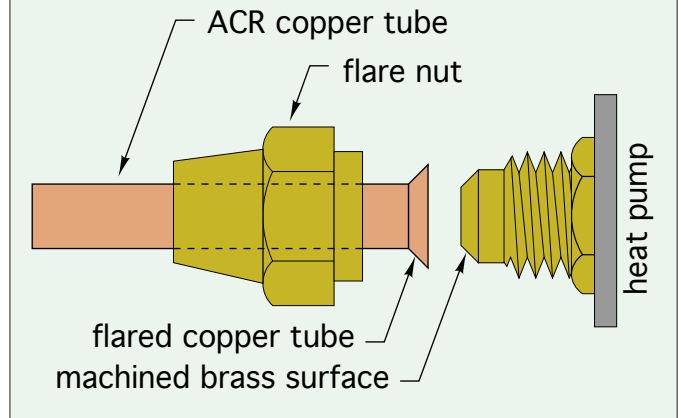
In this system, the compressor is located in the outdoor unit. The indoor unit houses the refrigerant-to-water heat exchanger, as well as the circulator and the user interface.

Figure 4-8 illustrates the major components in a split system air-to-water heat pump as they would operate in heating and cooling modes. These illustrations show a circulator integrated into the heat pump's interior unit. Some heat pumps provide this circulator, while others do not.

Split system air-to-water heat pumps require refrigeration service tools to install. A typical installation involves running the ACR copper tubing lines between the indoor and outdoor units. The larger tube is called the suction line and the smaller tube is called the liquid line. They typically connect to the indoor and outdoor units using flared joints, as illustrated in Figure 4-9.

After the copper tube set has been connected, it needs to be pressure tested to verify that no leaks are present. This is usually done by introducing compressed nitrogen into the tubing until the internal pressure reaches 250 psi. The system should remain under this test pressure for at least

Figure 4-9



an hour while a technician checks for leaks. Any leaks must be repaired and the pressure test repeated until the system test pressure remains stable for at least one hour.

The next step is to release the test nitrogen and connect a vacuum pump to the system. The vacuum pump is used to remove essentially all gases and moisture from the

refrigeration tubing. The vacuum pump typically connects to the refrigeration service valves on the outdoor unit. It is operated until the vacuum level inside the tubing drops to 50 microns. One micron is the very small absolute pressure needed to raise a column of mercury 1/1000th of a millimeter in height.

The outdoor unit of modern split system heat pumps is factory charged with refrigerant. The amount of factory-charged refrigerant corresponds to some maximum allowed length for the refrigeration tubing set between the indoor and outdoor units. Maximum allowed lengths in the range of 15 to 30 feet are common. If the refrigeration tubing set is longer, more refrigerant must be added to the system.

Once the refrigerant tubing has been brought to the required vacuum, the service valves on the outdoor unit are opened to allow the factory-charged refrigerant to fill the remainder of the refrigeration system components. The refrigeration system is now ready to operate.

Currently, the most commonly used refrigerant in residential heat pumps is R410a. Speculation exists that R410a will eventually be replaced by other refrigerants that have

less effect on the atmosphere (e.g., lower global warming potential). These include carbon dioxide (CO<sub>2</sub>), propane and other hybrid mixtures. Given the amount of currently installed systems operating on R410a, it's very likely that at least some future refrigerants will allow "drop-in" replacement for R410a, if and when as it is phased out.

Split system heat pumps have the advantage that no water or water-based antifreeze solutions are used in the outdoor portions of the system. There is nothing in the outdoor portion of the system that can freeze. All components containing water are housed in the indoor unit.

In some split system air-to-water heat pumps the compressor is housed in the outdoor unit. This reduces interior sound levels. In other split system configurations, the compressor is housed in the interior unit. The latter approach minimizes the number of components in the outside unit, which allows for easier servicing during inclement weather, and likely extends the average service life of components that would otherwise be housed in the outside unit. Figure 4-10 shows an example of a split system air-to-water heat pump with an interior compressor.

**Figure 4-10**

**outdoor unit**  
(contains refrigerant-to-air heat exchanger and fan)



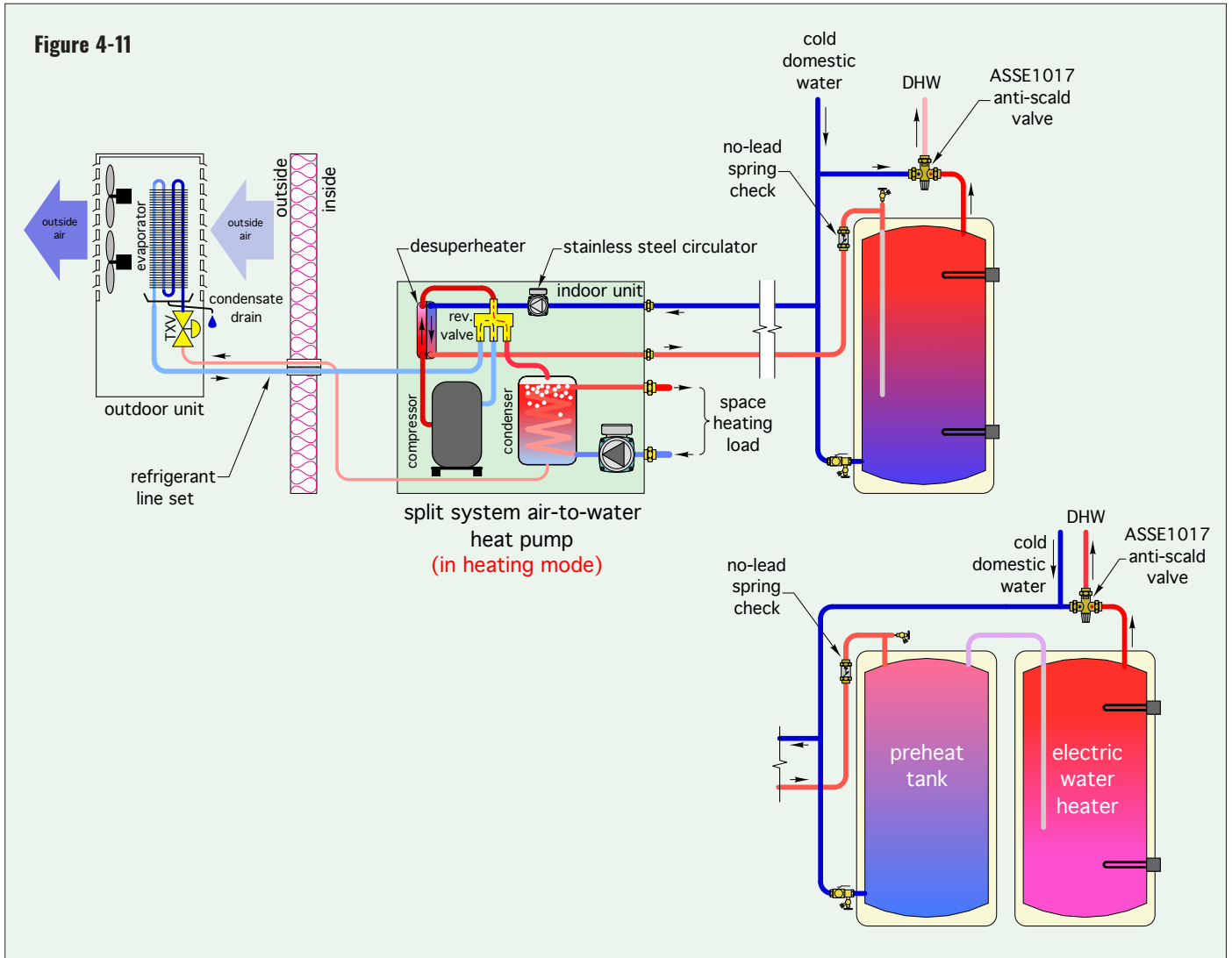
**indoor unit**  
(contains compressor, refrigerant-to-water heat exchanger, controls, DHW desuperheater)



Courtesy of Nordic Heat Pumps



Figure 4-11



Split system heat pumps with interior compressors also allow the possibility of adding a desuperheater heat exchanger to the indoor unit. The desuperheater is used to heat domestic water by absorbing heat from the hot refrigerant gas leaving the compressor before that gas reaches the reversing valve or condenser. Figure 4-11 shows how a desuperheater is used in this type of heat pump.

Because it receives hot refrigerant gas directly from the compressor, the desuperheater is able to heat domestic water whenever the compressor is running, in heating or cooling mode. In cooling mode, heat transferred to domestic water is heat that would otherwise be wasted by dissipation from the condenser coil into outside air. As such, it is “free” heat. Some heat pumps have controls that limit the domestic hot water temperature produced by the desuperheater to approximately 130°F. When the water leaving the desuperheater reaches this temperature, the small stainless steel circulator inside the heat pump’s

indoor unit turns off. The hot refrigerant gas leaving the compressor just passes through the desuperheater and flows on to the reversing valve and condenser.

Some designers prefer to use *two* domestic water tanks in systems where the heat pump has a desuperheater. The tank that receives the cold domestic water is a “preheat” tank. The cooler water in this tank circulates through the heat pump’s desuperheater, allowing it to operate at a lower temperature. This improves the heat pump’s performance, especially in systems where the hydronic distribution system operates at low water temperatures. Warm water from the preheat tank flows into the electric water heater whenever there’s a draw of domestic hot water. The elements in the electric water heater operate, if necessary, to raise the water to the desired setpoint temperature. An ASSE 1017-rated thermal mixing valve should always be installed to ensure that the water temperature sent to the hot water fixtures doesn’t exceed 120°F.

## INTERIOR AIR-TO-WATER HEAT PUMPS

There are also air-to-water heat pumps that are designed to be located *inside* buildings. They use short lengths of ducting to bring outside air to their air-handling section, as well as to discharge that air back outside. Having the heat pump inside has the following advantages:

- There is no outdoor equipment beyond air intake and discharge grills on the side of the building
- A desuperheater can be incorporated for domestic water heating
- The system can be operated without antifreeze
- Degradation of equipment due to weather exposure is eliminated
- There is less potential for debris to collect on heat transfer coil surfaces

Disadvantages of interior air-to-water heat pumps include:

- Require more interior space
- Brings compressor sound inside the building
- Requires careful coordination with building design to ensure that adequately sized ducting can be accommodated and terminated above snow level.

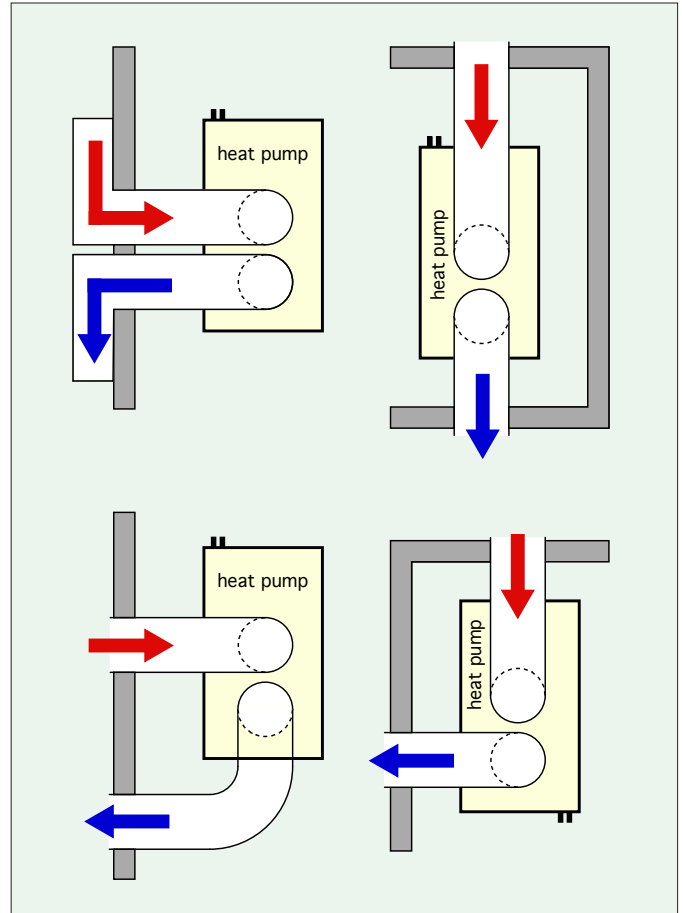
Figure 4-12 shows an example of an interior air-to-water heat pump. The large diameter insulated flexible ducting brings outdoor air to the unit and exhausts it back outside.

It's important to locate the air intake and discharge openings to prevent cross flow of discharge air into the

**Figure 4-12**



**Figure 4-13**



intake opening. Manufacturers of interior air-to-water heat pumps typically specify minimum separation distances and provide options such as exterior hoods specifically designed to eliminate crossflow. Some possible options are shown in Figure 4-13.

Interior air-to-water heat pumps are currently offered in Europe, but as of this writing, are not available in North America.

## OTHER AIR-TO-WATER HEAT PUMP SYSTEM CONFIGURATIONS

Another approach to heating and cooling based on an air-to-water heat pump involves use of a “hub” module that connects to and controls several peripheral devices, such as the outdoor condenser, buffer tank and distribution system. Figure 4-14 shows an example of such a product.

The white box mounted to the mechanical room wall is the “hub” of this air-to-water heat pump system. It connects to the buffer tank,

Figure 4-14a

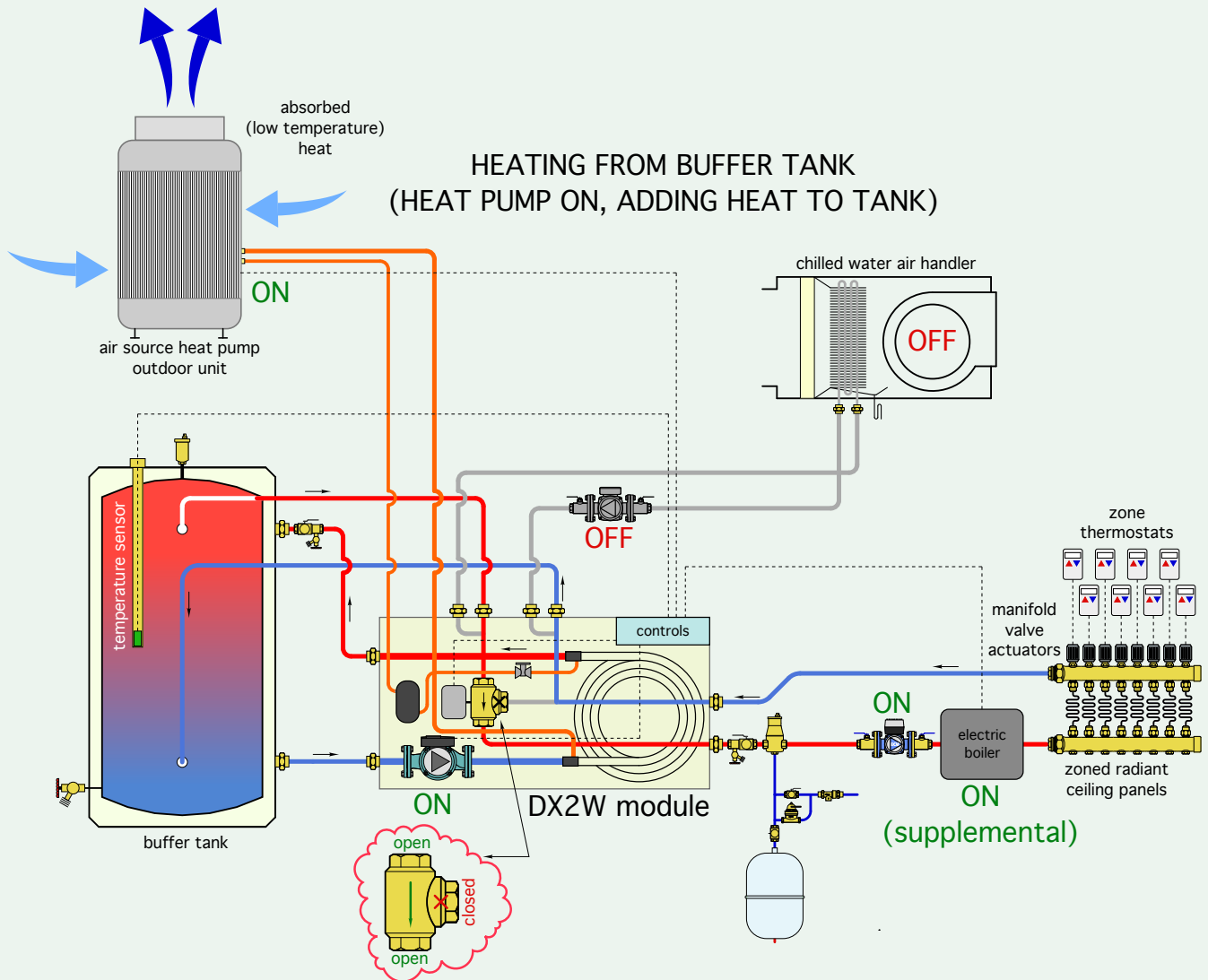


Figure 4-14b



Courtesy of ThermAtlantic

Figure 4-15





**Figure 4-16a**



Courtesy of ThermaAtlantic

**Figure 4-16b**



**Figure 4-17**



Courtesy of Groupe Atlantic

the outdoor condenser unit and the hydronic distribution system. It contains a refrigerant-to-water heat exchanger, a motorized mixing valve, a circulator and a suction line accumulator (for the refrigerant circuit). Figure 4-15 shows how the internal components of this “hub” interface with the surrounding equipment.

As shown in Figure 4-15, the refrigeration system is on. Hot refrigerant gas is circulating from the outdoor unit through the refrigerant-to-water heat exchanger, which is functioning as the condenser. Heated water is flowing into the buffer tank. Heated water is also being extracted from one of the upper connections on the buffer tank and routed through the 3-way mixing valve to the radiant panel distribution system. This water flows through an electric boiler which may or may not be operating as a supplemental heat source depending on the required supply water temperature to the radiant panel circuits.

This system can also provide chilled water for cooling. The coldest chilled water is passed to the air handler, where it provides some sensible cooling and

dehumidification. Slightly warmer chilled water is provided to the radiant panel circuits for added sensible cooling. The motorized mixing valve in the “hub” maintains the temperature of the chilled water flowing to the radiant panel circuits above the dewpoint temperature of the space being cooling. This prevents surface condensation on the radiant panel.

This approach has also been used to construct multiple stage air-to-water heat pump systems to provide space heating, cooling and domestic water heating for multi-family housing buildings. Figure 4-16 shows one example where three 2-stage air-to-water heat pump “hubs” connect to three buffer tanks. Each of these hubs connects to two independently controlled outdoor condenser units.

### **INTEGRATED SYSTEMS**

In addition to monobloc, split systems, interior heat pumps and “hub” configurations, some manufacturers supply integrated air-to-water heat pumps. These “appliance” products contain all the hardware necessary to provide heated water for space heating, chilled water for space cooling, and heat for domestic hot water. They also contain electric resistance heating elements that can provide supplemental heating when necessary. Figure 4-17 shows an example of the components used in one of these systems.



## SIZING CONSIDERATIONS FOR AIR-TO-WATER HEAT PUMPS

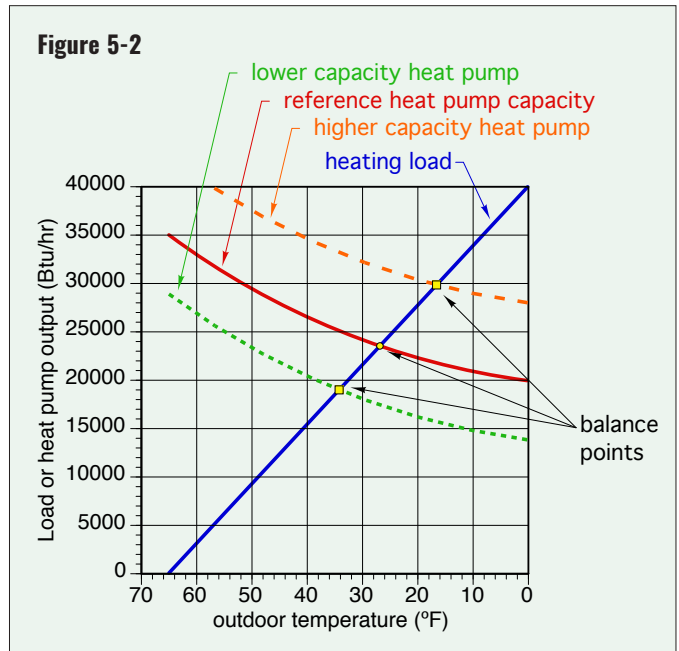
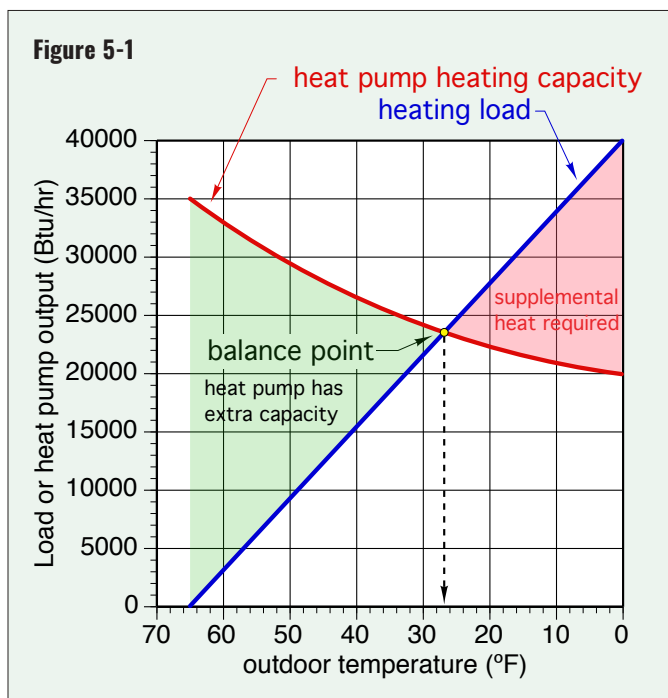
Previous sections have discussed how the heating and cooling capacity of air-to-water heat pumps varies with changes in outdoor temperature and the water temperature leaving the heat pump. This section presents information to help select a heat pump that is suitable for a given application. The primary focus is on heating mode operation, which typically presents the greater load requirement.

### BALANCE POINT

A common way to compare the heating output of an air-source heat pump to a building's heating load is to plot the heat pump's heating capacity and the load as a function of outdoor temperature. Figure 5-1 shows an example.

The heating load, represented by the blue line in Figure 5-1, is assumed to increase linearly as the outdoor temperature drops. The heating capacity of a representative air-source heat pump, represented by the red curve in Figure 5-1, decreases as the outdoor temperature drops. The point where the load line and heating output curve cross is called the balance point. This is where the heating output of the heat pump exactly equals the load. For the scenario shown in Figure 5-1, this occurs at an outdoor temperature of about 27°F, and a corresponding heat transfer rate of about 24,000 Btu/hr.

The green shaded area to the left of the balance point represents conditions where the heat pump has excess heating capacity relative to the load. Heat pumps with single



speed compressors would cycle on and off under these conditions to avoid overheating the building. Heat pumps with variable-speed compressors will typically reduce heating output to match the load under these conditions.

The red shaded area to the right of the balance point represents conditions where there is insufficient heat output from the heat pump to meet the load. Some heat pumps would be operating continuously under these conditions, with the "deficit" between heat output and heating load provided by a supplemental heat source. Other heat pumps might turn off at some lower limit of outdoor temperature. In this case, all the heat required by the building would have to come from a supplemental heat source.

If a heat pump with a higher heating capacity is used, the heating capacity curve would shift upward and the balance point would shift to a lower outdoor temperature. If a heat pump with a lower heating capacity was used, the red curve would shift down, and the balance point would shift to a higher outdoor temperature. These effects are illustrated in Figure 5-2.

It's somewhat intuitive that selecting a heat pump with a larger heating capacity will reduce the amount of supplemental heating required. For the heat pumps assumed in Figure 5-2, the heat pump with the higher heating capacity shifted the outdoor balance point from about 28°F down to about 16°F. Still, this information is not, by itself, helpful in answering questions about seasonal heating performance, such as:

- How much of the building's seasonal heating requirement can be supplied by a given heat pump?
- How should the heat pump's heating capacity compare to the design load of the building?
- How does the cost of supplying heat using the heat pump compare to another fuel option, such as natural gas?
- How does the variation in the heat pump's heating capacity and COP affect its seasonal performance in a given system installed at a specific location?

To show how these questions can be addressed, consider the following example.

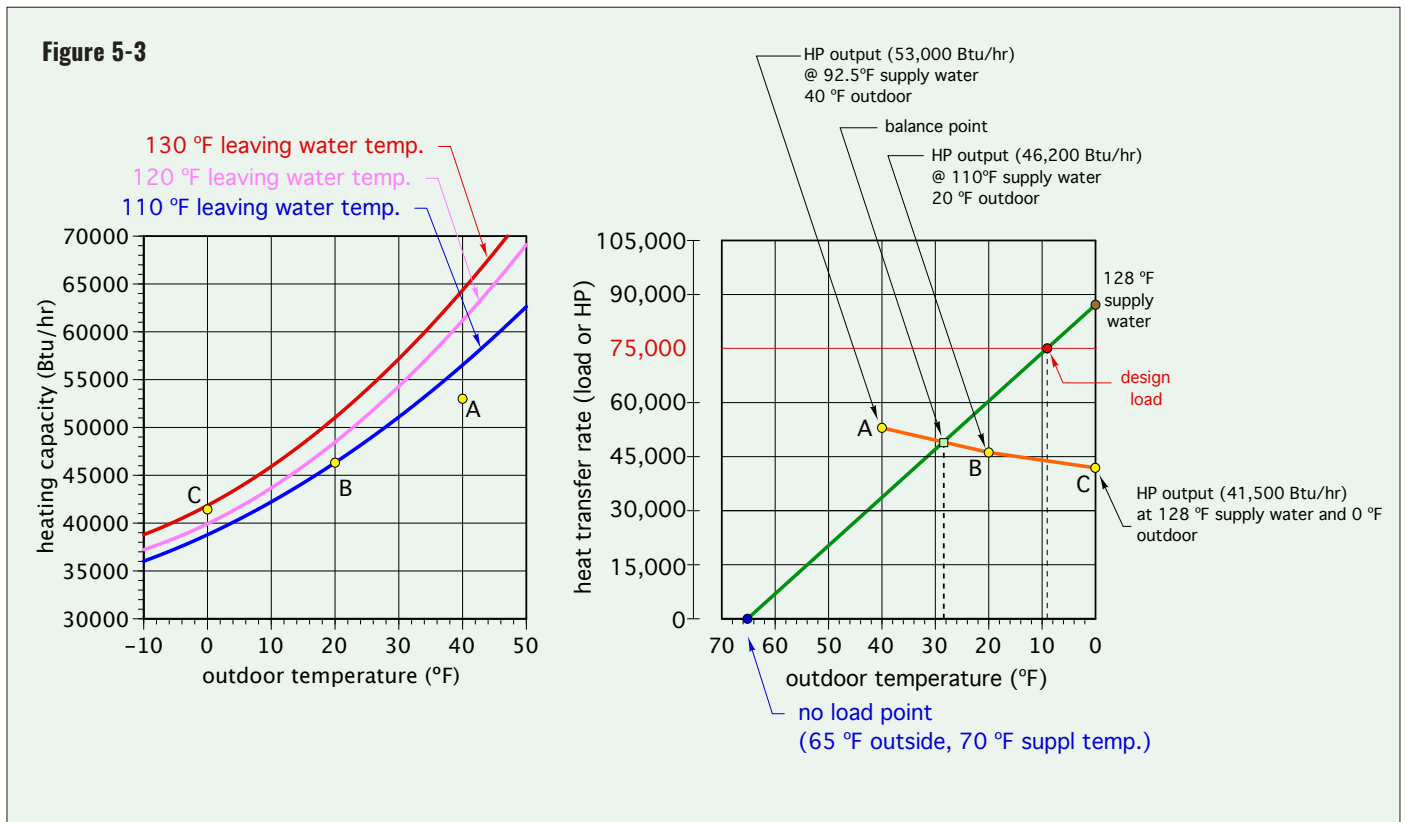
A house located near Boston, MA, has a design heat loss of 75,000 Btu/hr. The house has a low-temperature hydronic distribution system that can supply design heating load using a supply water temperature of 120°F. The water temperature supplied to the distribution system is based on full-range outdoor reset control. At outdoor temperatures above 65°F, the net heating load is zero due to internal heat gains. A low-ambient air-to-water heat pump having the performance shown in Figure 2-15 is being considered for use in this house. Determine the balance point for this

house and heat pump. Also determine what percentage of the seasonal heating energy can be supplied by this heat pump. Compare the cost of operation of the heat pump system to that of a natural gas boiler using local utility rates.

Figure 5-3 shows the heating capacity graph from Figure 2-15 alongside a graph showing the supply water temperature required based on outdoor reset control. This graph also shows a portion of the heating capacity curve for the heat pump.

The heating capacity curve shown in orange on the right side graph was generated by determining the heat pump's heating capacity at three randomly selected outdoor temperatures (0°F, 20°F and 40°F), and at the supply water temperature required by the hydronic distribution system at those outdoor temperatures (e.g., based on outdoor reset control). These temperature combinations are represented by points A, B, and C in both graphs. Some interpolation between the three capacity curves was required. The green line on the right side graph is the outdoor reset line for supply water temperature.

In this case, the balance point temperature is about 28°F. When the outdoor temperature is warmer than 28°F, the heat pump has more than sufficient capacity



to meet the house's heating load. When the outdoor temperature is lower than 28°F, some amount of supplemental heat will be required. Taking this analysis further requires data for outdoor temperatures over the heating season.

### CLIMATE CONSIDERATIONS

The next step in answering questions related to seasonal performance is to combine the performance information for the heat pump and house with climate data for Boston.

Because the heating capacity of an air-to-water heat pump is highly dependent on outdoor air temperature, its seasonal performance will depend on the time during which the heat pump operates at different outdoor temperatures. Data for long-term averages of hourly outdoor temperature is readily available for hundreds of locations in North America. Sources for this data include:

- ACCA manual J
- ASHRAE Weather Data Viewer software

Figure 5-4 shows an example of “bin” temperature data for Boston, MA. In this case, the “bins” are 5°F wide. Each vertical bar gives the number of hours that the outdoor temperature falls within the specified range, based on long-term average values for those temperatures. On any specific year, the number of hours in each bin may be higher or lower than the long-term average.

The outdoor design temperature in Boston is 9°F. Notice how many hours correspond to outdoor temperatures significantly higher than this design temperature. This implies that most of the heating occurs under partial load conditions.

If internal heat gains are minimal, and thermostat setbacks are not used, space heating loads can be modeled as being approximately proportional to the difference between the indoor setpoint temperature and the outdoor temperature. This allows the heating load to be calculated as a percentage of design heating load using the average outdoor temperature for each bin. Figure 5-5 shows a

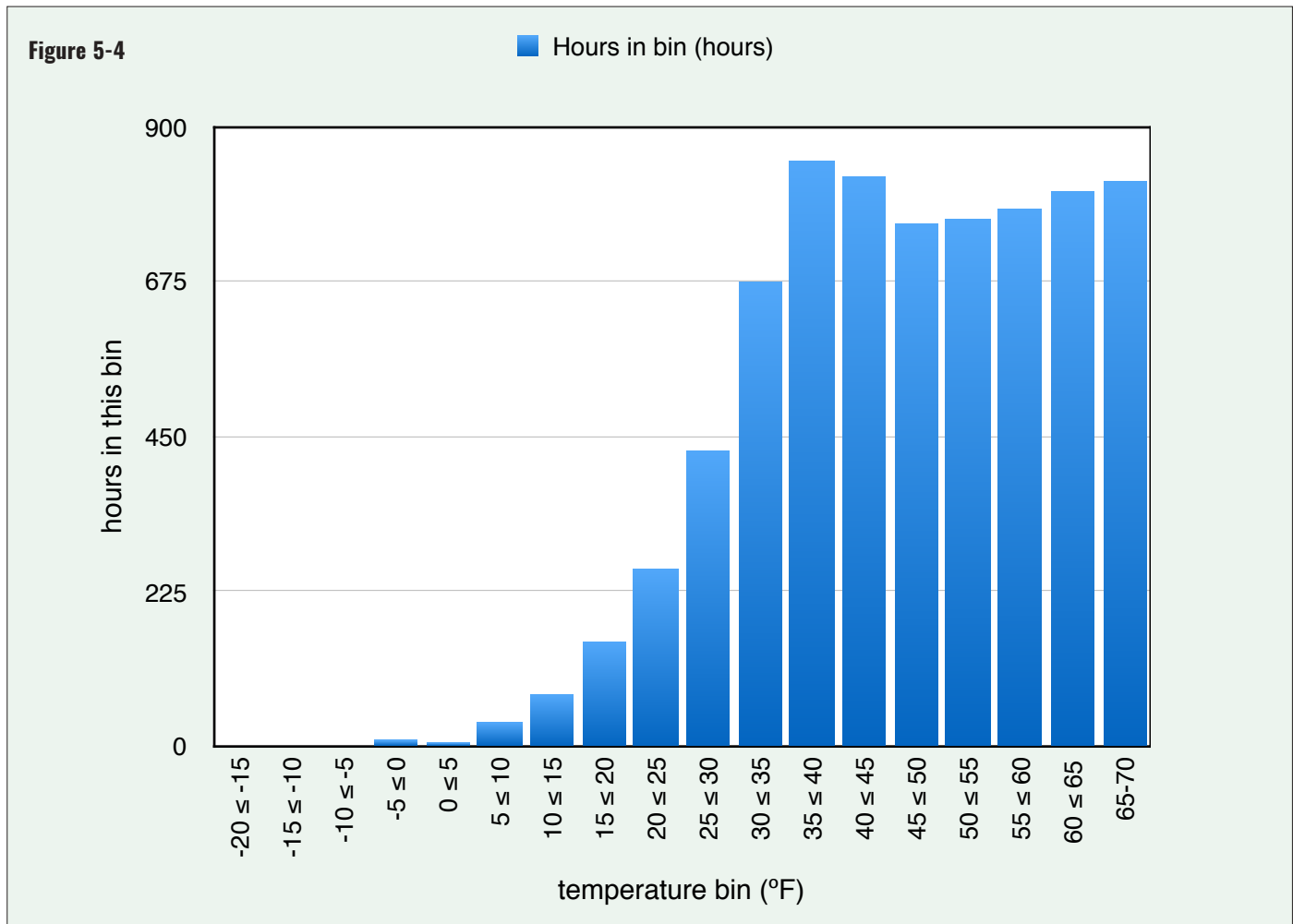


Figure 5-5

Bin range (°F)	Hours in bin (hours)	Average outdoor temp of bin (°F)	% of design load	Hours during which load $\geq$ this % of design load
-10 $\leq$ -5	1	-7.5	127	1
-5 $\leq$ 0	9	-2.5	118.9	10
0 $\leq$ 5	4	2.5	110.7	14
5 $\leq$ 10	35	7.5	<b>100</b>	49
10 $\leq$ 15	74	12.5	94	123
15 $\leq$ 20	151	17.5	86	274
20 $\leq$ 25	256	22.5	78	530
25 $\leq$ 30	429	27.5	70	959
30 $\leq$ 35	674	32.5	62	1633
35 $\leq$ 40	848	37.5	53	2481
40 $\leq$ 45	828	42.5	45	3309
45 $\leq$ 50	757	47.5	37	4066
50 $\leq$ 55	766	52.5	29	4832
55 $\leq$ 60	781	57.5	20.5	5613
60 $\leq$ 65	804	62.5	12.3	6417

spreadsheet where this has been done, based on the Boston bin temperature data shown in Figure 5-4.

The right side column in Figure 5-5 gives the hours during which the space heating load equals or exceeds each given percentage of design load. Plotting the last two columns of this spreadsheet up to outdoor temperatures of 65°F yields a “heating duration curve,” as shown in Figure 5-6.

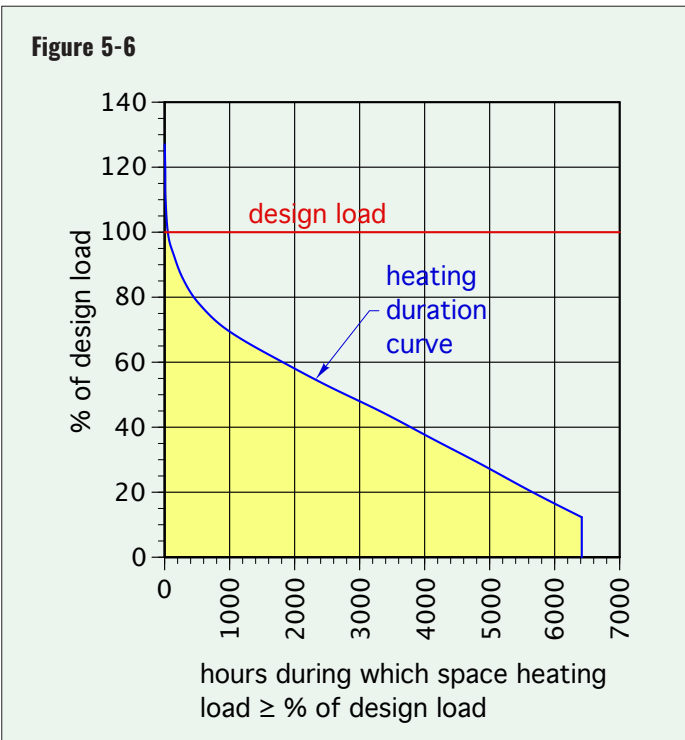
The yellow shaded area under the heating duration curve is approximately proportional to the total space heating energy needed over an average heating season. This is based on the assumption that internal heat gains alone supply the home’s heat loss at outdoor temperatures of 65°F or higher.

The relationship between the area under the heating duration curve and the total seasonal space heating energy required can be very useful when combined with other building and heat pump performance data. Figure 5-7 shows one example.

The graph of heat pump heating capacity and building supply water temperature (on right) is shown next to the heating duration curve for Boston. The graphs have been scaled so that the design heating load of 75,000 Btu/hr on the right graph aligns with the 100% of design load on the left graph. A line has been drawn horizontally from the balance point on the right graph to the vertical axis of the other graph. The green shaded area under this line represents about 94% of the total area under the heating duration curve. This implies that, on an average year, the selected heat pump *alone* supplies about 94% of the total space heating energy while operating at or above the outdoor balance point temperature. The remaining space heating energy is supplied through the combined operation of the heat pump and a supplemental heat source.

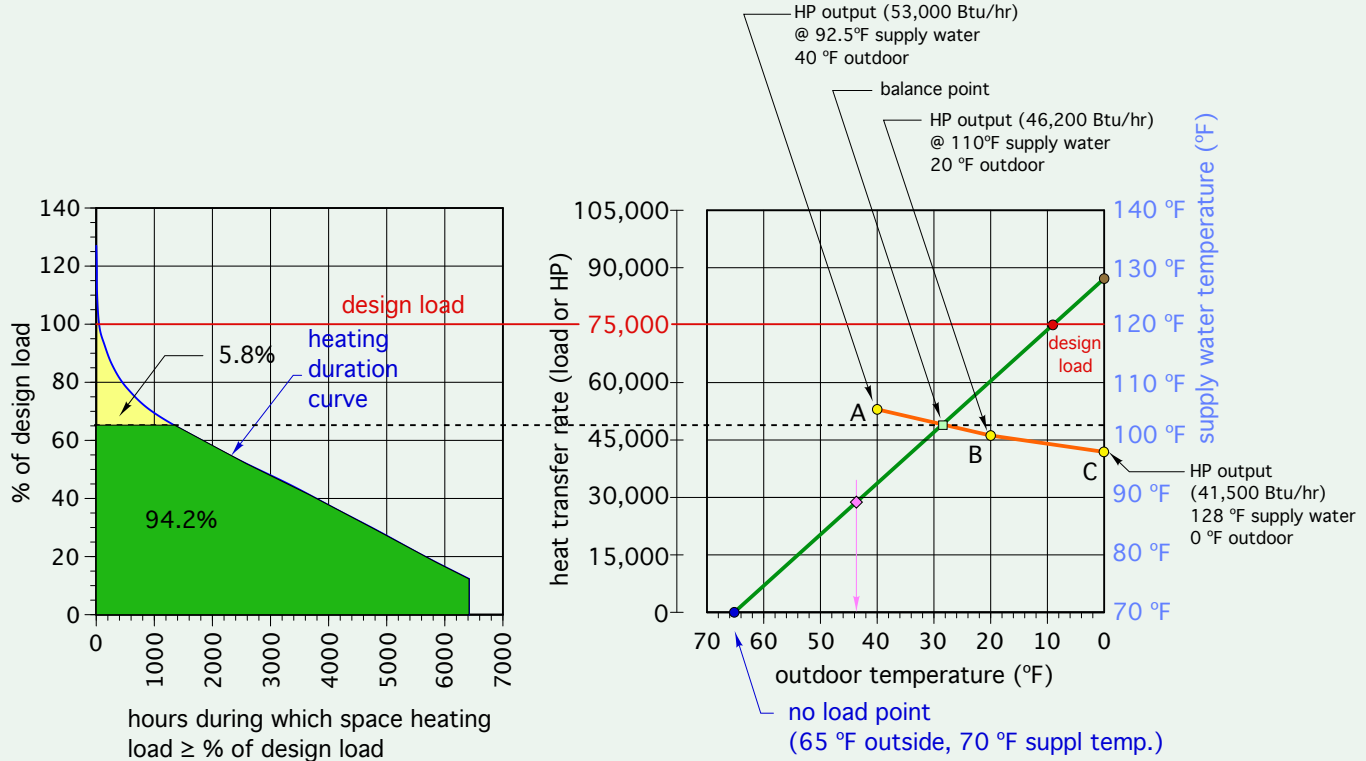
The same horizontal line can be extended to the supply water temperature scale on the right graph. This shows that for 94% of the heating season, the required supply water temperature is at or below 103°F, a temperature that allows modern air-to-water heat pumps to run at relatively high COP.

The low-ambient air-to-water heat pump assumed in this example could operate down to an outdoor temperature of -5°F. On an average year in Boston, there is only 1 hour of outdoor temperature lower than -5°F. Thus, the heat pump would remain in operation over essentially the full range of outdoor air temperature. The required supply water temperature corresponding to an





**Figure 5-7**



outdoor temperature of  $-5^{\circ}\text{F}$  would be about  $132^{\circ}\text{F}$ . The low-ambient heat pump used in this example would have a COP of about 1.7 under this condition.

### SPREADSHEET MODELING OF SYSTEM PERFORMANCE

Previous discussions have shown the variability of heating capacity and COP based on changes in the temperature of water leaving the heat pump's condenser and the outside air temperature.

The building that the heat pump serves also experiences wide changes in heating load over the heating season.

Hydronic distribution systems that use outdoor reset control to avoid "overheating" the water supplied to the distribution system will minimize the required supply water temperature, and thus, enhance the heat pump's performance under partial load conditions. These systems will experience a wide range of supply water temperatures.

The geographic location where the heat pump is installed will have specific average values for each temperature bin. All these variables make it impossible to provide an

accurate "rule of thumb" for sizing an air-to-water heat pump on a given project. The preferred approach is to *simulate* a proposed system configuration by building a spreadsheet that includes reasonable mathematical models for the building load, performance of the heating distribution system, performance of the heat pump and outdoor air temperature for the location of the project. Once the simulation spreadsheet is built, the designer can experiment with various "what if" scenarios on equipment size, supply water temperatures, etc., to determine overall seasonal performance and make decisions on system design.

The following mathematical models were used to build a simulation spreadsheet that merges the thermal performance of the previous discussed building, a nominal 4-ton low-ambient air-to-water heat pump, the building's heating distribution system and climatic data for Boston. These calculations were performed for each bin of outdoor temperature. The outdoor temperature ( $T_{\text{amb}}$ ) is taken as the average temperature of each bin (i.e., for the  $15\text{--}20^{\circ}\text{F}$  bin,  $T_{\text{amb}}$  would be by  $17.5^{\circ}\text{F}$ ).

Figure 5-8

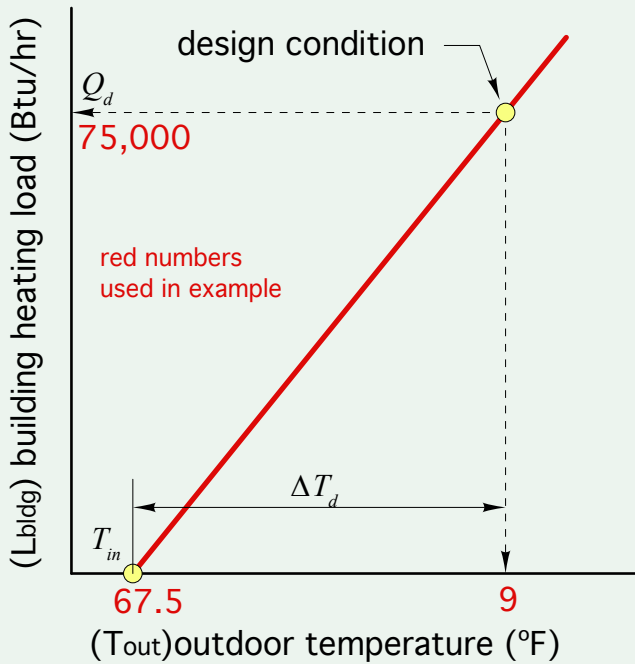
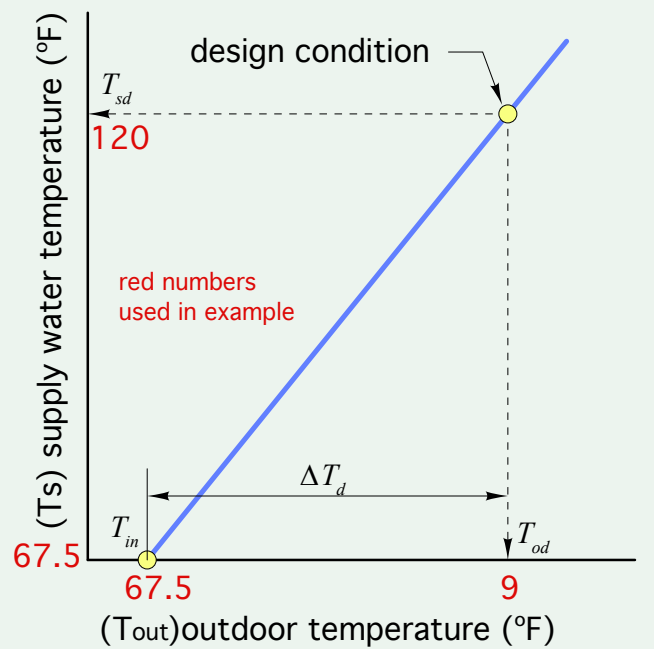


Figure 5-9



**Instantaneous building space heating load:**

$$L_{bldg} = \frac{Q_d}{\Delta T_d} (T_i - T_{out}) - Q_i$$

Where:

- $L_{bldg}$  = instantaneous building space heating load (Btu/hr)
- $Q_d$  = building’s design space heating load (Btu/hr)
- $\Delta T_d$  = difference between inside and outside temperature at design load (°F)
- $T_{in}$  = indoor setpoint temperature for space heating (°F), typically assumed at 70°F
- $T_{out}$  = current outdoor temperature (°F)
- $Q_i$  = current rate of internal heat gain (Btu/hr)

**Supply water temperature to space heating distribution:**

$$T_s = \left( \frac{T_{sd} - T_{in}}{T_{in} - T_{od}} \right) (T_{in} - T_{out}) + T_{in}$$

Where:

- $T_s$  = currently required supply water temperature (°F)
- $T_{sd}$  = required supply water temperate at design load (°F)
- $T_{in}$  = indoor setpoint temperature for space heating (°F), typically assumed at 70°F
- $T_{od}$  = outdoor temperature at design load (°F)
- $T_{outb}$  = current outdoor temperature (°F)

**Heating capacity of heat pump:**

$$Q_h = [k_1 + k_2(T_s)] + k_3(T_{out}) + k_4(T_{out})^2$$

Where:

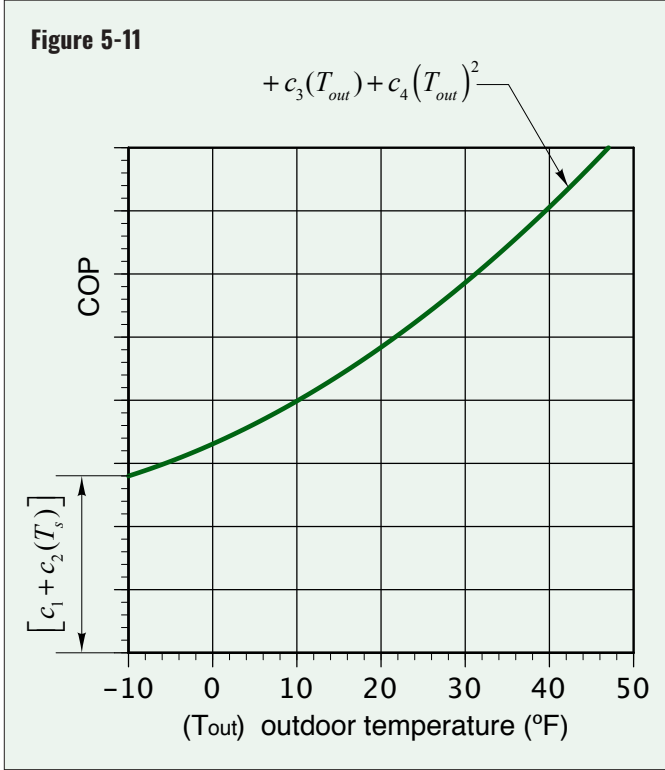
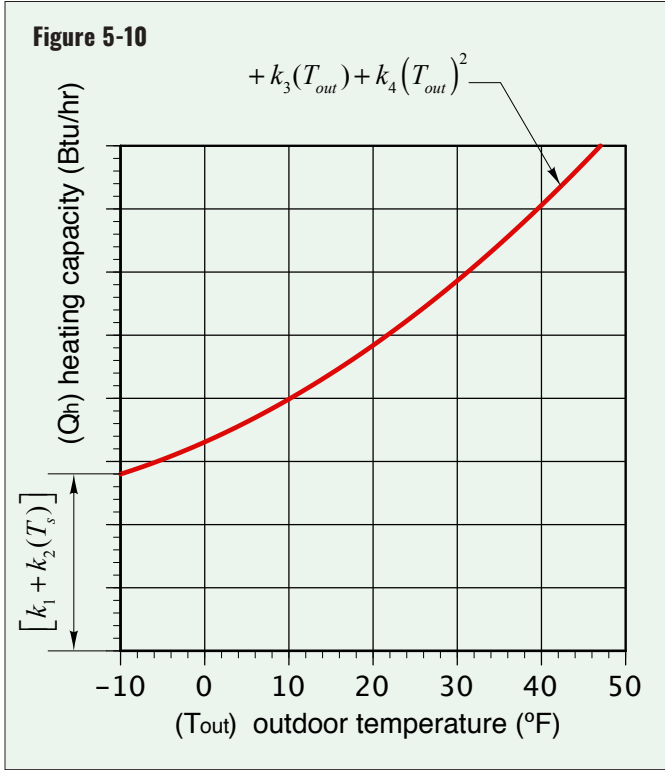
- $Q_h$  = “instantaneous” heating capacity of heat pump (Btu/hr)
- $T_s$  = currently required supply water temperature to distribution system (°F)
- $T_{out}$  = current outdoor temperature (°F)
- $k_1, k_2, k_3, k_4$  = constants determined by curve fitting to manufacturer’s published heat capacity curves.

**COP of heat pump:**

$$COP = [c_1 + c_2(T_s)] + c_3(T_{amb}) + c_4(T_{amb})^2$$

Where:

- COP = “instantaneous” COP of heat pump (Btu/hr)
- $T_s$  = currently required supply water temperature to distribution system (°F)
- $T_{amb}$  = current outdoor temperature (°F)
- $c_1, c_2, c_3, c_4$  = constants determined by curve fitting to manufacturer’s published heat capacity curves.



The modeling equations for the heat pump’s heating capacity and COP are based on curve fitting, which provides an approximation of performance over a range of operating conditions. *These modeling equations may not accommodate the performance characteristics of all air-to-water heat pumps, or the characteristics of all building loads or heating distribution systems. In some cases, designers may need to use other modeling methods to achieve accurate simulation.* Any models developed should be checked to see if they can replicate published performance information with reasonable accuracy. It’s also important to remember that curve fitting is based on specific ranges of data. The equations developed from curve fitting should only be applied within those ranges of data unless specific constraints are imposed, such as limiting maximum heating capacity and COP values under very desirable operating conditions. Check with heat pump manufacturers to verify such constraints.

**SEASONAL AVERAGE COP**

It is possible to use performance information for the heat pump and the building’s hydronic heating distribution system, combined with bin temperature data for a given location, to calculate a “project-adjusted” seasonal average COP. This involves determining the required supply water temperature for each bin, using this temperature along with the corresponding outdoor temperature and the heat pump’s COP data (or model)

to determine the COP for that bin. The operating hours of the heat pump in each bin are calculated. These values cannot exceed the actual hours in each bin. The operating hours in each bin are multiplied by the COP for that bin. These multiplications are summed and then divided by the total heat pump operating hours over the heating season to get a project-adjusted average COP for the heating season. These calculations are best done using a spreadsheet.

When this method was applied to the previously discussed building near Boston, along with the constraint that the maximum COP could not exceed 4.5, and the heating capacity could not exceed 72,000 Btu/hr, the seasonal average COP of the heat pump was 3.49. *This is an excellent performance number that is comparable to, if not higher than, what the seasonal average COP of a geothermal heat pump of the same capacity, and applied under the same conditions, might be.* This seasonal average COP is based on the full outdoor temperature range for an average Boston winter, which ranges from -10°F to a high of 70°F.

Additional spreadsheet-based analysis of this example project indicates that the total space heating energy required for the building for an average Boston winter is 194.3 MMBtu ( 1 MMBtu = 1,000,000 Btu). Of this, the heat pump supplied 185.0 MMBtu (about 95% of total),

and the auxiliary heat source supplied 9.3 MMBtu (about 5%) of total.

As of this writing, the cost of electricity supplied to residential customers just west of Boston, where the building is located, is \$0.149/kwhr. This is slightly higher than the current U.S. national average rate of \$0.118/kwhr. If electricity costing \$0.149/kwhr was directly converted to heat, such as in an electric boiler, the cost per million Btus would be \$43.66/MMBtu. This would make the estimated seasonal heating cost of the building \$8,483.

Assuming that the air-to-water heat pump, operating at a seasonal average COP of 3.49, supplies 95% of the seasonal heat — both of which were projected by the spreadsheet analysis — and that an electric boiler supplies the remaining 5%, the estimated seasonal heat cost would be:

$$\text{seasonal heating cost} = \left[ \frac{0.95 \times 194.3 \times 43.66}{3.49} \right] + \left[ \frac{0.05 \times 194.3 \times 43.66}{1} \right] = \$2,733$$

95% of load by heat pump  
 seasonal average COP provided by heat pump  
 provided by aux elec. heat source

The current price of natural gas in Boston is about \$1.55 per therm, which at an assumed 92% seasonal average boiler efficiency equates to \$16.85/MMBtu. The cost of heating the example house for an average winter using natural gas at this rate and conversion efficiency would be:

$$\text{seasonal heating cost} = \left( \frac{\$16.85}{\text{MMBtu}} \right) \times \left( \frac{194.3 \text{ MMBtu}}{\text{season}} \right) = \$3,274$$

In this project, the cost of space heating energy using the air-to-water heat pump was 16.5% lower than using natural gas. Furthermore, if a basic service charge of \$20 per month for having a natural gas meter on the building was eliminated so that the house was all electric, an additional savings of \$240 per year would be achieved. The total annual savings would be \$781 per year.

This example has shown that the majority (95%) of the space heating energy for a house near Boston with a 75,000 Btu/hr design load, and a hydronic distribution system that requires 120°F water at design conditions, can be supplied by a nominal 4-ton low-ambient air-to-water heat pump, and at seasonal cost about 16% lower than if the heating was done using a high-efficiency boiler operating on natural gas.



## HEAT EMITTER OPTIONS FOR AIR-TO-WATER HEAT PUMP SYSTEMS

One of the most important design details for successful application of air-to-water heat pumps is keeping the required load water temperature as low as possible. Low load temperatures allow the heat pump to operate at high COPs in heating mode.

*Space heating distribution systems that can provide design heating output using supply water temperatures no higher than 120°F allow the majority of currently available air-to-water heat pumps to deliver reasonably good performance. Systems that can operate at even lower supply water temperatures will further improve heat pump performance.*

Distribution systems that supply each heat emitter using parallel piping branches rather than series configurations are also preferred because they provide the same supply water temperature to each heat emitter.

Examples of heat emitters and other techniques that allow air-to-water heat pumps to provide good performance include:

- Heated floors with low-resistance coverings
- Radiant wall and ceiling panels
- Generously sized panel radiator systems
- Fan-assisted panel radiators

- Fan coils (allowing for heating and cooling)
- High-output fin-tube baseboard
- Existing cast iron radiators
- Reducing building heating loads, or adding a heat emitter to existing systems

This section discusses these heat emitter and design options in more detail.

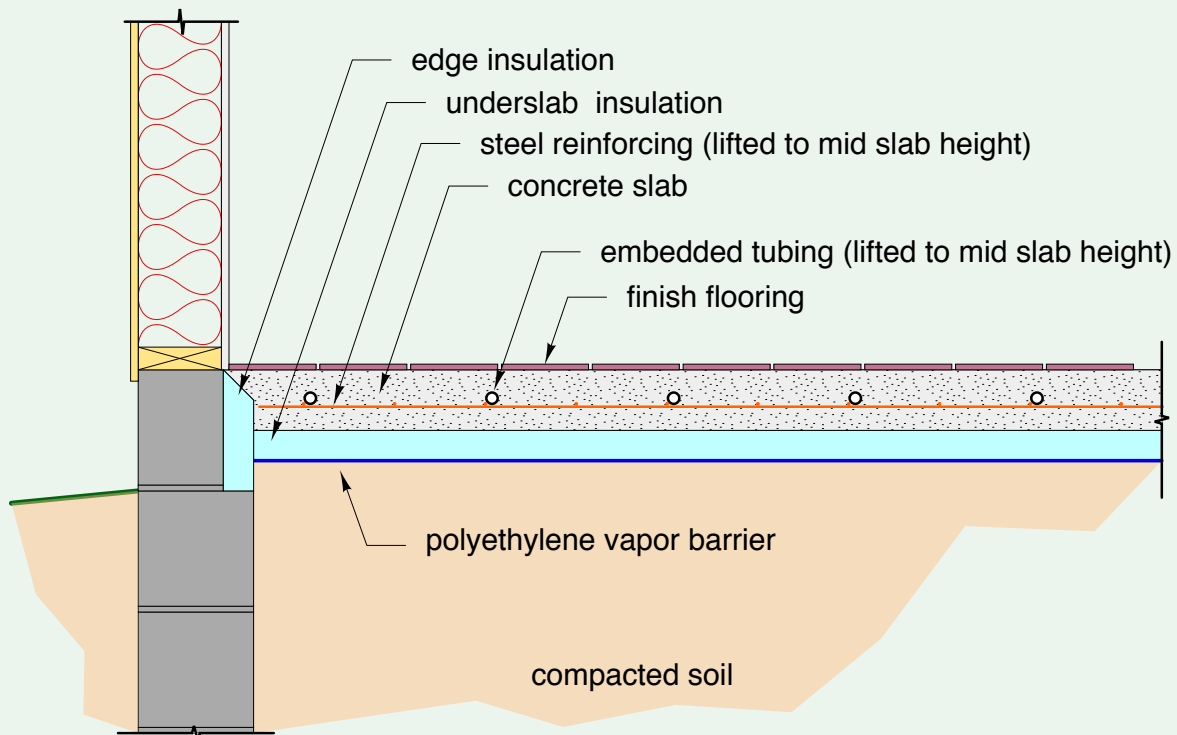
### HEATED FLOOR SLABS

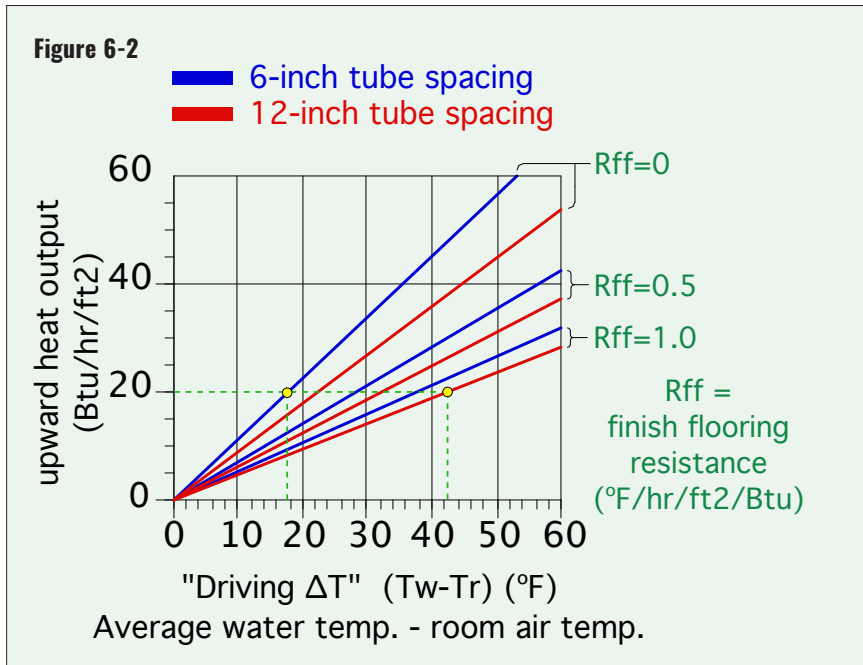
Heated floor slabs with relatively close tube spacing and low finish floor resistances are well-suited for use with air-to-water heat pumps. Figure 6-1 shows a cross section of a heated floor slab.

The graph in Figure 6-2 shows upward heat output from a heated slab using tube spacings of 6 inches and 12 inches, and for finish floor resistances ranging from 0 to 1.0 (°F•hr•ft<sup>2</sup>/Btu). The steeper the line, the better suited the distribution system is for use with a heat pump.

Figure 6-2 shows that achieving an upward heat output of 20 Btu/hr/ft<sup>2</sup> from a slab with no covering (e.g., R<sub>ff</sub> = 0) and 6-inch tube spacing requires the “driving ΔT” (e.g., the difference between the average water temperature in floor circuit and room air temperature) to be 17.5°F.

Figure 6-1





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

- Any finish flooring layers installed on the slab should have a combined total R-value of 1.0 or less.

### UNDERFLOOR TUBE & PLATE RADIANT PANELS

Another possible radiant panel construction is called an underfloor tube & plate system. Figure 6-3 shows a typical cross section for this panel.

This panel provides a way to heat wood-framed floors. It relies on aluminum heat transfer plates to diffuse heat away from the tubing and spread it across the floor area. These plates are critically important for good performance. To achieve compatibility with low water temperatures, tube spacing should not exceed 8 inches.

Thus, in a room maintained at 70°F, the average water temperature in the circuit needs to be 87.5°F. The supply water temperature to the circuit would likely be in the range of 95–98°F. This is a relatively low supply water temperature that would allow an air-to-water heat pump to operate at high COPs.

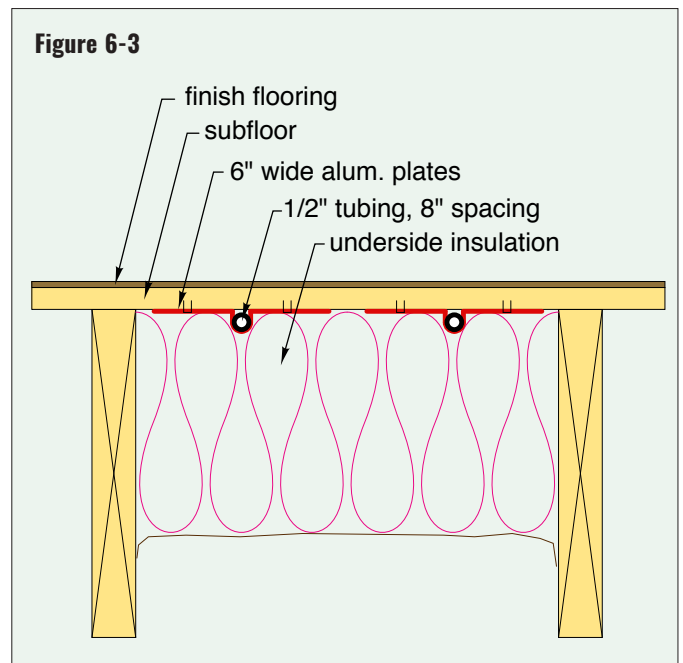
For comparison, consider supplying the same 20 Btu/hr/ft<sup>2</sup> load using a heated floor slab with 12-inch tube spacing and a finish floor resistance of 1.0°F•hr•ft<sup>2</sup>/Btu. The driving ΔT must now be 42.5°F. The average circuit water temperature required to maintain a room temperature of 70°F would be 70 + 42.5 = 112.5°F and the supply temperature likely in the range of 120–123°F. This higher temperature would significantly reduce the heat pump's COP.

The following guidelines are suggested in applications where a heated floor slab will be used to deliver heat derived from a hydronic heat pump:

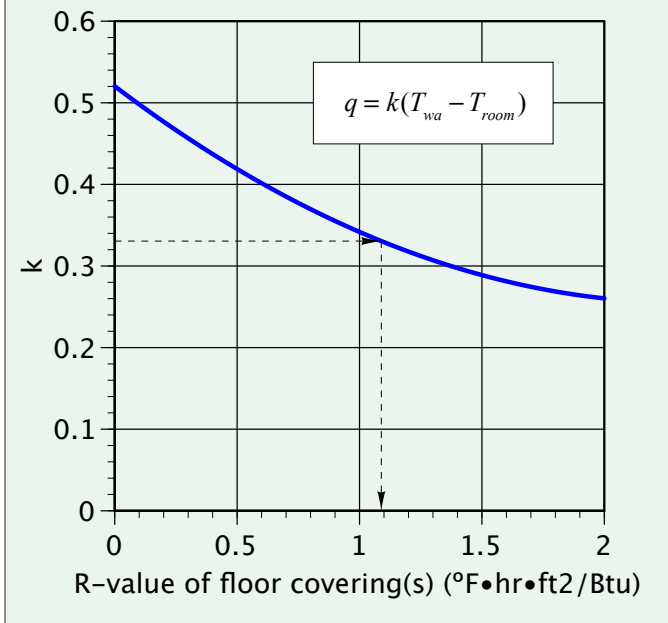
- The tube spacing within the slab should not exceed 12 inches.
- The slab should have a minimum of R-10 underside and edge insulation.
- The tubing should be placed at approximately 1/2 the slab depth below the surface, as shown in Figure 6-1. This placement decreases the required water temperature required for a given rate of heat output relative to tubing placed at the bottom of the slab. Lower water temperatures improve heat pump performance.

The maximum allowed R-value of the subfloor and finish floor coverings should also be kept as low as possible. Figure 6-4 plots a coefficient called "k," versus the R-value of finish floor covering installed over a 3/4-inch thick plywood subflooring and assumes 8-inch tube spacing.

The heat output of this floor panel can be estimated using Formula 6-1.



**Figure 6-4**



**Formula 6-1:**

$$q = k(T_{wa} - T_{room})$$

where:

q = heat output of the underfloor tube & plate panel (Btu/hr/ft<sup>2</sup>)

k = value from Figure 6-4

T<sub>wa</sub> = average water temperature in tubing circuit (°F)

T<sub>room</sub> = room air temperature (°F)

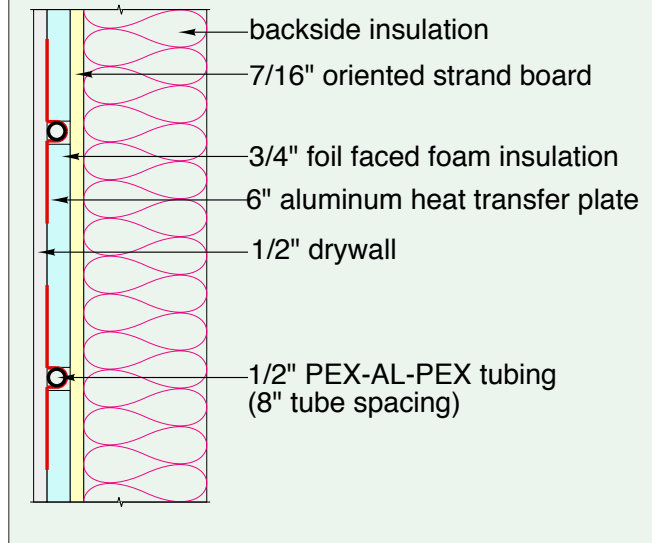
A suggested guideline in evaluating floor coverings for this type of radiant panel is to constrain the average water temperature to no higher than 115°F. This would allow the supply water temperature to be 120°F (assuming a 10°F circuit temperature drop). If this constraint is entered into Formula 6-1, a relationship between panel output, room air temperature and allowed finish floor covering resistance is established.

Here's an example. Assume that a designer has determined that the radiant floor needs to release 15 Btu/hr/ft<sup>2</sup> under design load conditions. To achieve good heat pump performance, the designer wants to limit the average water temperature in the panel to 115°F under design load conditions. What is the maximum thermal resistance of the floor coverings?

$$q = k(T_{wa} - T_{room}) = 15 = k(115 - 70)$$

$$k = \frac{15}{115 - 70} = 0.333$$

**Figure 6-5**



Using Figure 6-4, the maximum thermal resistance of the finish floor covering(s) allowable in this situation is R-1.1°F•hr•ft<sup>2</sup>/Btu.

**RADIANT WALL PANELS**

Radiant panels can be integrated into walls and ceilings as well as floors. Several of these configurations are suitable for use with air-to-water heat pumps. The key is ensuring that the radiant panel can deliver design load output while operating at a relatively low water temperature. This favors radiant panels that provide high surface areas relative to their rate of heat delivery. It also favors panels that have low internal thermal resistance between the tubing and the surface area releasing heat to the room.

One example is a radiant wall panel constructed as shown in Figure 6-5.

When finished, this “radiant wall” is indistinguishable from a standard interior wall. Its low thermal mass allows it to respond quickly to changing internal load conditions or zone setback schedules. The rate of heat emission to the room can be estimated using Formula 6-2:

**Formula 6-2:**

$$q = 0.8 \times (T_{wa} - T_{room})$$

where:

q = heat output of wall panel (Btu/hr/ft<sup>2</sup>)

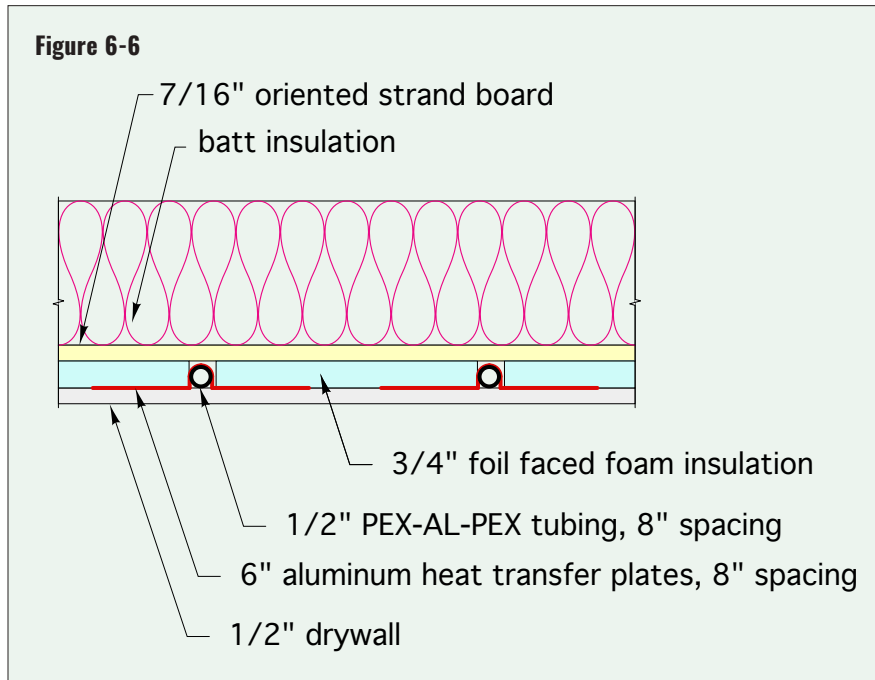
T<sub>wa</sub> = average water temperature in panel (°F)

T<sub>room</sub> = room air temperature (°F)

Thus, if the radiant wall panel operates with an average water temperature of 110°F in a room with 70°F air temperature, each square foot of heated wall would release about  $0.8 \times (110 - 70) = 32$  Btu/hr/ft<sup>2</sup>.

## RADIANT CEILING PANELS

Another possibility is a radiant ceiling panel using the same materials and construction methods as the radiant wall panel. Figure 6-6 shows a cross section of this construction.



**Figure 6-7**



Figure 6-7 shows an example of a radiant ceiling under construction.

In this construction, the 1/2" PEX-AL-PEX tubes are spaced 8 inches apart. The plates are 5 inches wide. The 3/4-inch foil-faced polyisocyanurate foam insulation strips are "held back" near the U-bends in the tubing. A 3/4-inch thick furring board runs adjacent to the upper plate of the partition. This assembly will be covered with 1/2-inch drywall. 2.5-inch long drywall screws will be installed halfway between adjacent tubes and spaced 12 inches apart in the direction parallel with the tubes.

The rate of heat emission to the room can be estimated using Formula 6-3:

### Formula 6-3:

$$q = 0.71 \times (T_{wa} - T_{room})$$

where:

q = heat output of ceiling panel (Btu/hr/ft<sup>2</sup>)

T<sub>wa</sub> = average water temperature in panel (°F)

T<sub>room</sub> = room air temperature (°F)

Heated ceilings have the advantage of not being covered or blocked by coverings or furniture, and thus, are likely to retain good performance over the life of the building. They are also ideal surfaces for radiant cooling using chilled water supplied from an air-to-water heat pump. However, when radiant cooling is used, it is critical to maintain the chilled water temperature above the dewpoint of the interior air.



More information on radiant cooling is available in *idronics* #13.

## PANEL RADIATORS

Generously sized panel radiators can also provide good performance when used as part of a hydronic heat pump system. *The suggested guideline is to*



**Figure 6-8**



Courtesy of Andrew Jones

size panels so they can deliver design space heating output using a supply water temperature no higher than 120°F. An example of a panel radiator with integral thermostatic radiator valve is shown in Figure 6-8.

Panel radiator manufacturer's provide "reference" output ratings for their panels assuming that they operate at a specific average water temperature and in rooms with a specific air temperature. Correction factors are then given that allow the heat output rate to be adjusted for other average water temperatures and room air temperatures.

As an approximation, a panel radiator similar to the one shown in Figure 6-8, operating with an average water temperature of 110°F, and in a room with 70°F air temperature, provides approximately 27% of the heat output it would have at a reference average water temperature of 180°F and room air temperature of 70°F. Larger panels (e.g., longer, taller and deeper) are available to increase surface area to compensate for lower average water temperatures. The panel radiator shown in Figure 6-8 will release about 3,400 Btu/hr into a room at 70°F, when operated at an average water temperature of 110°F.

### **FAN-ASSISTED PANEL RADIATORS**

One of the newest low-temperature heat emitters is a fan-assisted panel radiator. These units use an array of small, low-power fans installed between the front and back surfaces of the panel. The fans automatically change speed based on room temperature relative to setpoint. They significantly increase convective heat output at low supply water temperatures. Each fan only requires about 1.5 watts of electrical power at full speed, and thus, electrical energy consumption is negligible, especially when compared to

**Figure 6-9**



Courtesy of Rettig Group

the electrical energy savings associated with operating the heat pump at lower water temperatures and higher COPs.

Fan-assisted panel radiators can operate at water temperatures as low as 95°F. They have integral controls that can be set for a "boost" mode (e.g., full-speed fan operation) when the system is recovering from a setback temperature.

### **FAN-COILS**

One of the benefits offered by air-to-water heat pumps is the ability to create chilled water for warm weather cooling. Successful implementation of hydronic cooling must address sensible cooling (e.g., lowering the temperature of interior air), as well as latent cooling (e.g., lowering the moisture content of interior air). The latter process implies that some of the water vapor in the air must be condensed into a liquid.

Condensation will occur on any surface that is below the dewpoint temperature of the surrounding air. The lower the surface temperature relative to the dewpoint temperature, the faster the rate of condensation. On humid days, the dewpoint of untreated interior air can reach well into the low to mid-70s °F range. This is much higher than the chilled water temperatures that air-to-water heat pumps can produce, which are often in the range of 45 to 60°F. Panel radiators, fin-tube baseboard and radiant panels are not intended to operate under condensing conditions.

**Figure 6-10**



Courtesy of Myson

Allowing condensation to occur on these heat emitters can quickly lead to water stains, corrosion, mold, and in the case of radiant panels — major damage to the materials making up the panels.

One heat emitter that *can* serve as a “cooling emitter” as well as a *heat emitter* is a fan-coil equipped with a condensate drip pan. One example of such a product is shown in Figure 6-10.

This fan-coil combines a large surface “coil” made of copper tubing with aluminum fins, with a low-power tangential blower located under the coil. A drip pan is also located under the blower. It catches water droplets that form on the coil and eventually drip from it. The captured condensate is drained away from the unit by gravity through a small tube and disposed of outside the building or into a suitable drainage system. If no drain is available at a lower elevation than the drip pan, the condensate can be routed to a condensate pump, which will move it upward to a suitable drain.

Modern fan-coils designed for heating and cooling operation can be sized to operate at a relatively low supply water temperature in heating (e.g., 120°F or less). Their cooling performance is based on the temperature of chilled water supplied to the coil. Lower water temperatures improve both sensible and latent cooling capacity. Designers need to assess the sensible and latent ratings of perspective fan-coils to ensure adequate overall cooling.

Heat pump performance (e.g., both cooling capacity and EER) increases as chilled water temperatures increase. To achieve the best heat pump performance, designers should use the highest chilled water temperature that can ensure adequate sensible and latent cooling. Chilled water supply temperatures in the range of 50 to 60°F are possible with some fan-coils and are well within the operating range of air-to-water heat pumps.

### **LOW-TEMPERATURE FIN-TUBE BASEBOARD**

Fin-tube baseboard was originally developed for the high water temperatures available from conventional boilers. It was often sized for supply water temperatures of 180°F to 200°F, and in some cases even higher. This is much higher than the water temperatures air-to-water heat pumps can produce. Thus, traditional fin-tube baseboard is not recommended in such applications.

However, as the global hydronics industry moves toward low water temperature distribution systems, some manufacturers have developed “low-temperature” fin-tube convector products. The fin-tube element shown in Figure 6-11 has significantly greater fin area compared to that of a standard element. It also has two tubes passing through the fins. This allows significantly higher heat output at lower water temperatures. The rated output of the fin-tube element in Figure 6-11, with both tubes operating in parallel, is 272 Btu/hr/ft at an entering water temperature of 90°F, and 532 Btu/hr/ft at a water temperature of 120°F, both at a total flow rate of 1 gallon per minute.

**Figure 6-11**



(Source: Stirling Environmental)

### **CAST IRON RADIATORS**

Many older homes have existing cast iron radiators. They may have been part of an original steam heating system, or they might have operated with water.

If the cast iron radiators were originally sized for a poorly insulated or uninsulated building, and that building was

subsequently insulated and fitted with new windows and doors or air sealed, the existing radiators may only have to provide a fraction of the heat output for which they were originally sized. This may allow the radiators to operate at a much lower water temperature, such as those available from an air-to-water heat pump.

The heat output of cast iron radiators can be estimated for relatively low average water temperatures. The process requires that the surface area of the radiator be determined based on the radiator's type and dimensions.



***idronics #25 provides detailed information for evaluating the suitability of using existing cast iron radiators in low-temperature distribution systems.***

## **REDUCING WATER TEMPERATURE IN EXISTING SYSTEMS**

There are many "legacy" hydronic heating systems designed to operate at water temperatures much higher than are possible when an air-to-water heat pump serves as the primary heat source. The required water temperature in those systems can be reduced through two methods:

1. Reducing the building's design heating load.
2. Adding heat emitters to the distribution system.

It is also possible to use a combination of these methods.



***idronics #25 provides analytical methods for assessing the impact of both methods for reducing water temperature. It also shows several techniques for adding heat emitters to existing distribution systems.***



## DESIGN DETAILS FOR AIR-TO-WATER HEAT PUMP

This section discusses several design details that enhance the operation of air-to-water heat pumps. Some help the heat pump operate at high COPs and high EERs. Others improve system stability and reduce potential for short cycling. Still others improve control of heat generation and distribution.

### MOUNTING OUTDOOR HEAT PUMPS

Most air-to-water heat pumps have outdoor units that serve as the evaporator in heating mode and condenser in cooling mode. Although outdoor units are made and tested to withstand relatively harsh environments, thoughtful mounting will extend their service life and help ensure good performance.

Outdoor units should be mounted above the deepest expected snow level. This allows for adequate airflow regardless of weather. Elevating the unit at least one foot above the ground also reduces accumulation of grass clippings and leaves. Figure 7-1 shows the outdoor unit of a split system mounted on a stand that keeps it at about two feet above the

**Figure 7-1**



Courtesy of Brian Just

**Figure 7-2**



ground in a location that experiences deep snow.

This stand is bolted to four 6-inch diameter concrete piers that provide stability.

Outdoor units should be mounted in locations where runoff from roofs will not fall on them. If mounted under an eave, a gutter should be installed to catch roof runoff and route it away from the outdoor unit, as shown in

**Figure 7-3**



Courtesy of Revision Energy

Figure 7-2. Also keep the unit away from locations where icicles could form above the unit.

In some situations, it may be possible to mount the outdoor unit under a roof or wide overhang. Always verify that there is adequate space behind the unit for proper airflow. Manufacturers typically specify a minimum distance from the rear of the unit to a wall. Figure 7-3 shows an installation where two outdoor units are mounted under a wide overhang, and on a base that elevates them well above ground level.

Some outdoor units can be mounted on heavy-duty wall brackets, an example of which is shown in Figure 7-4. Always verify that the bracket system is rated to support the weight of the condenser unit, and that adequate fastening to the wall framing is specified.

**Figure 7-4**



Outdoor units should also be mounted so that the melt water from defrosting can drain clear of the unit, as seen in Figure 7-5. If melt water accumulates at the bottom of the unit, it can refreeze and eventually damage the outdoor coil or other components.

In heating-dominated climates, and when other factors can be accommodated, it's preferable to mount the outdoor unit at the south



**Figure 7-5**



side of the building where solar radiation can create a “microclimate” that slightly warms the surrounding air.

Always consider that any outdoor unit will emit some sound when operating. Units with inverter compressors and variable-speed fans tend to produce less sound than those with fixed-speed compressors. Still, avoid mounting locations near outdoor patios, decks or bedroom windows.

The length of the refrigeration line set should be factored into the decision on mounting location. In general, the shorter the line set, the better the heat pump’s heating and cooling performance. Manufacturers typically charge the outdoor unit with sufficient refrigerant to accommodate a specified length of line set. Length allowances of 15 to 30 feet are common. If the line set has to be longer, additional refrigerant must be added to the system when it is commissioned.

Although there are instances where condenser units for air-conditioning systems have been mounted on low concrete slabs or plastic pedestals, *this is not recommended for air-to-water heat pumps*, especially in cold northern climates. Mounting the outdoor unit near ground level tends to increase debris accumulation on the coil and make the unit more

vulnerable to insects, rodents and animal urine. Snow accumulation can interfere with air flow through the evaporator coil and fans. Melt water from defrosting can also accumulate and refreeze on slabs, possibly damaging the unit.

Once the outdoor unit of a split system is connected to the refrigeration piping, it is good practice to close off any gaps where this piping joins the outdoor cabinet. Gaps at this location provide an entry point for mice, which can chew on wiring and insulation, build nests or otherwise degrade internal components.

Figure 7-6 shows an outdoor unit mounted on a simple commercially available steel stand that elevates it above the ground.

The legs of this stand are bolted to pressure-treated 2x6 boards that are leveled and placed slightly below finish grade. The area under the unit was then covered with a landscaping fabric to prevent weed growth. Crushed stone was placed over the landscape fabric to keep it in place. All fasteners are stainless steel.

The national electric code requires a service disconnect adjacent to every outdoor heat pump or air-conditioning condenser. It should be

**Figure 7-6**



mounted close to, but not behind the unit, as seen in Figure 7-6.

Most outdoor units are supplied with vibration-dampening mounting studs, which incorporate a layer of stiff rubber between the base of the unit and its mounting frame.

If the unit will be mounted in a high wind area, it is preferable to orient it so that prevailing winds don’t directly impinge on the front face where airflow from the fans is released. It’s also advisable to secure the mounting frame to the building foundation so that the unit is not excessively buffeted by high wind gusts.

### **PIPING CONNECTIONS**

All monobloc style air-to-water heat pumps require two piping connections to the balance of the system. There will be some slight movement in the outdoor unit due to temperature changes, wind buffeting and even the potential that someone accidentally “bumps” into the unit. Outdoor units that house a compressor also generate some vibration. Given these issues, it’s advisable to provide some flexibility and vibration-absorbing capacity between the outdoor unit and the piping it connects to inside the build.

**Figure 7-7**



One approach is to use reinforced flexible piping connectors, as shown in Figure 7-7.

These 1-inch pipe size flexible connectors have a swivel fitting at one end, allowing for fully tightened connections without stressing the connector. They have 1-inch MPT threads that connect to the heat pump and the rigid copper tubing at the wall. Note the slight offset of the connections, which minimizes stress when the outdoor unit moves slightly.

Another option is to connect the heat pump to piping at the wall using corrugated stainless steel tube (CSST) assemblies, as shown in Figure 7-8.

**Figure 7-8**



Courtesy of Enertech Global

Piping penetrations in exterior walls should be done in ways that prevent entry of precipitation or insects. They should also prevent any condensation that occurs during cooling mode operation from leaking into the wall cavity.

One approach is to create sleeves of PVC piping that pass from the inside wall surface to the outside wall surface. The sleeves should be sealed to the wall surfaces at both ends. The pipes to and from

**Figure 7-9**



the heat pump pass concentrically through these sleeves, as shown in Figure 7-9.

After the piping connection has been verified as leak-free, the space between the PVC sleeve and the piping can be filled with an expanding foam sealant. Any excess

**Figure 7-10**



**Figure 7-11**



foam can be neatly trimmed away with a hacksaw blade, as shown in Figure 7-10.

After all piping connections have been pressure-tested to ensure there are no leaks, all outdoor piping should be wrapped with a suitable insulation. The outside surface of the insulation should be protected against ultraviolet degradation using a UV-resistant tape or coating. This has not yet been done in Figure 7-11.

## **BUFFER TANKS**

Anytime an on/off heat source is combined with a zoned hydronic distribution system, there is potential for short cycling of that heat source. That potential can be reduced by using a buffer tank in the system. An example would be a 4-ton on/off air-to-water heat pump supplying several individually controlled panel radiators. Each radiator represents a “micro-zone.” The output of such a zone might only be 5 to 10% of the heating capacity of the heat pump. If several micro-zones are connected to a header that leads directly to the heat pump, even a heat pump that can modulate down to perhaps 30% of rated capacity, short cycling is very likely.

Beyond this primary function, buffer tanks can also provide hydraulic separation between simultaneously operating circulators and an energy reserve for domestic water heating.

The size of a buffer tank is based on two considerations:

1. What is the minimum run time of the heat pump that avoids their definition of “short cycle?”
2. What is the allowed temperature change of the buffer tank during the minimum on-cycle time?

Once values for these two parameters are chosen, the minimum size of a buffer tank can be determined using Formula 7-1.

**Formula 7-1:**

$$V = \frac{t(Q_{\text{heat source}} - q_{\text{load}})}{500(\Delta T)}$$

where:

- V = required volume of the buffer tank (gallons)
- t = desired duration of the heat source’s “on-cycle” (minutes)
- Q<sub>heat pump</sub> = maximum expected heat output rate of the heat pump (Btu/hr)
- q<sub>load</sub> = rate of heat extraction from the tank (can be zero) (Btu/hr)
- ΔT = temperature rise of the tank from when the heat source is turned on to when it is turned off (°F)

Here’s an example. An air-to-water heat pump with a maximum expected heat output of 48,000 Btu/hr is to operate with a minimum on-cycle of 10 minutes while supplying a towel warmer radiator releasing heat at 2,000 Btu/hr. The heat pump responds to the buffer tank temperature. It turns on when the buffer tank temperature drops to 100°F, and off when the tank reaches 120°F. What is the necessary buffer tank volume to achieve this performance?

Solution: Just put the values into Formula 7-1 and run the calculation.

$$V = \frac{t(Q_{\text{heat source}} - q_{\text{load}})}{500(\Delta T)} = \frac{10(48000 - 2000)}{500(120 - 100)} = 46 \text{ gallons}$$

Larger buffer tanks can provide longer heat source on-cycles. They can also allow a narrower temperature change over a specific on-cycle. It’s easy to evaluate the trade-offs between on-cycle length and tank temperature swing using Formula 7-1. Larger buffer tanks obviously cost more, take up more room in a mechanical room and usually have higher standby heat loss.

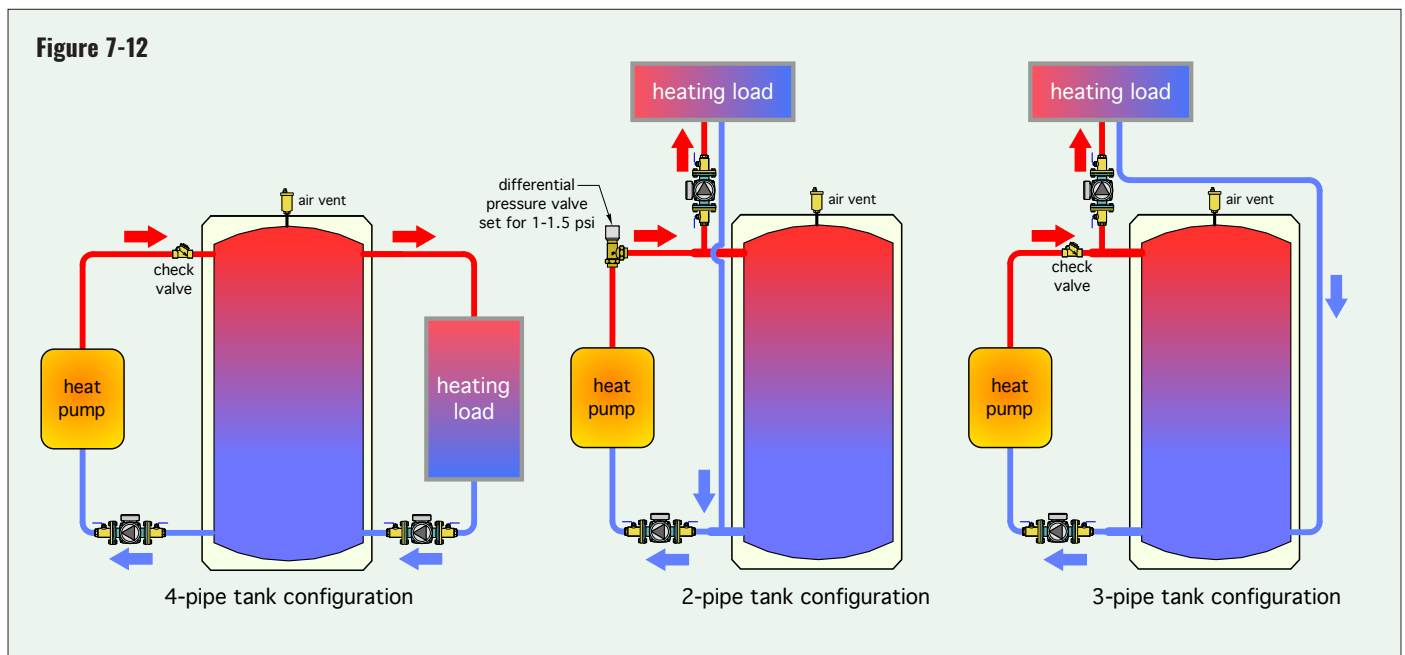
*In systems where the buffer tank will contain chilled water for cooling, it is essential that all piping connections are insulated and vapor sealed. This is especially true for metal tanks. If moist room air comes in contact with cold piping, fittings or the pressure vessel, condensation will form and quickly cause staining or superficial corrosion.*

**BUFFER TANK PIPING**

There are several ways to pipe buffer tanks. They are called “4-pipe”, “3-pipe” and “2-pipe” configurations. All three configurations are shown in Figure 7-12.

The 4-pipe configuration is the “classic” way to pipe buffer tanks in hydronic systems. The heat source adds heat on one side, while the load removes heat from the other side. This piping configuration provides excellent hydraulic separation between the heat source circulator and the load circulator(s).

One characteristic of a 4-pipe configuration is that all heat from the heat source must pass through the tank on its way to the load. This isn’t necessarily a problem when the buffer tank temperature is being maintained by the heat source, independently of calls from the load. However, this



arrangement retards heat transfer from the heat source to the load if the tank is allowed to cool substantially.

A check valve should be installed on the heat pump side of the buffer tank to prevent reverse thermosiphoning through the heat pump when it is off.

The 2-pipe configuration places the load *between* the buffer and the heat pump. This allows the possibility of passing heat directly from the heat source to the load without that heat having to first pass through the

buffer tank. This enables faster recovery from setback conditions.

One limitation of the 2-pipe configuration is that a differential pressure valve that creates a forward opening resistance of 1 to 1.5 psi, or a motorized ball valve, needs to be installed in the heat pump piping to prevent a portion of the flow returning from the load from passing through the heat pump when it's off. It's also necessary to keep the tees that connect to the load as close as possible to the tank to allow for good hydraulic separation.

**Figure 7-13**

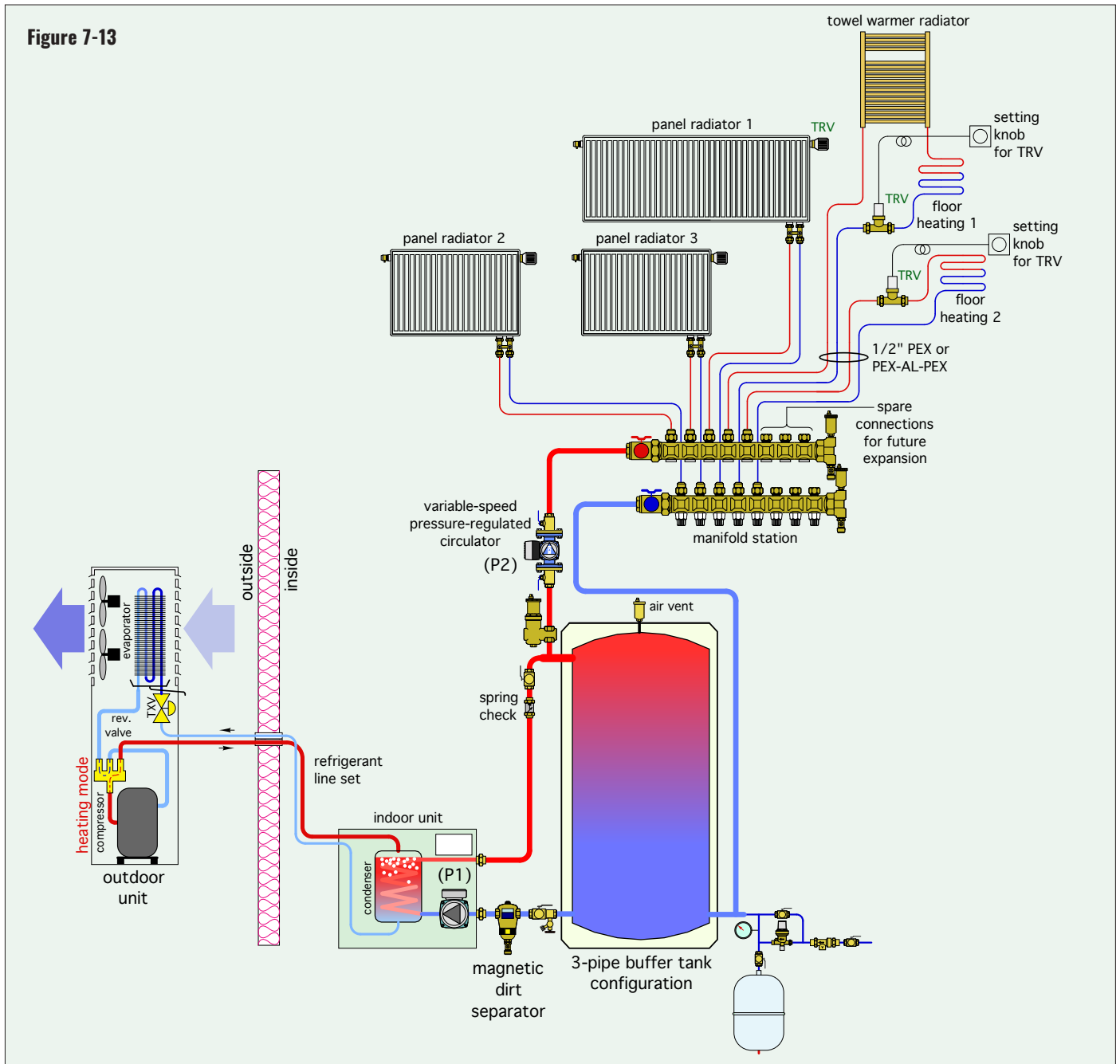




Figure 7-14

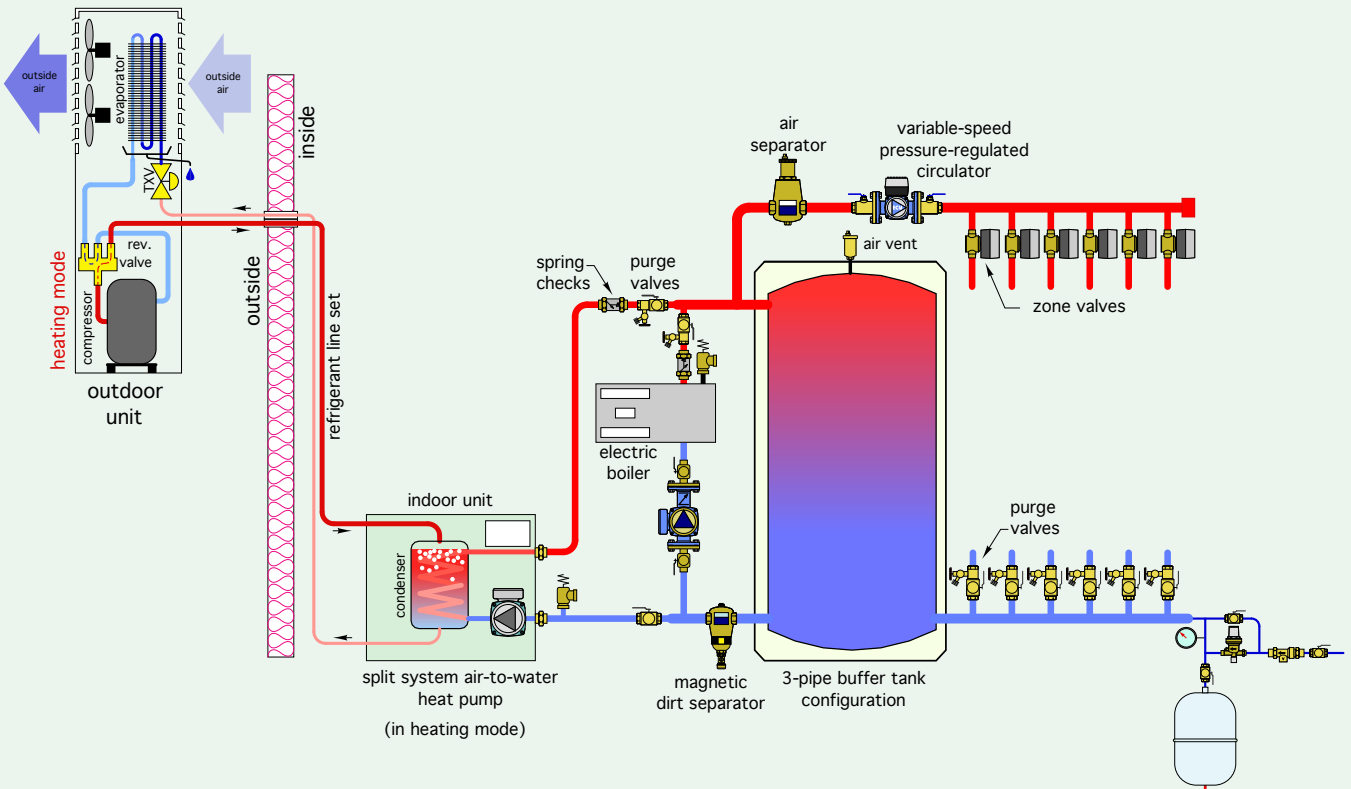
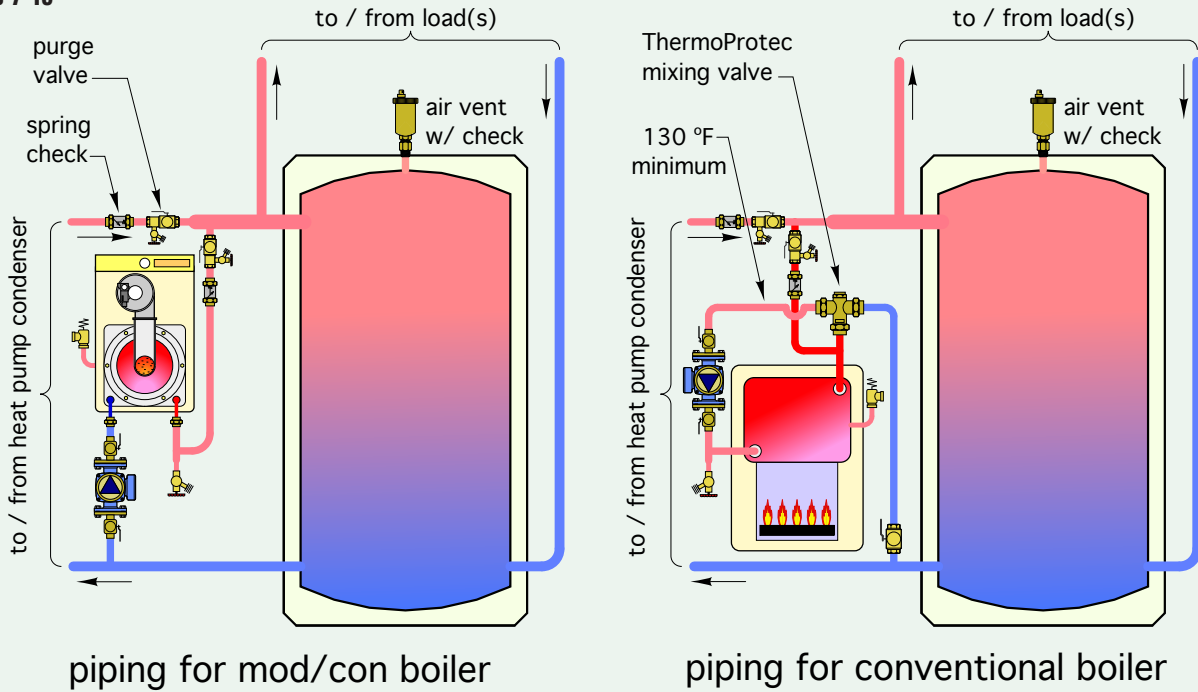


Figure 7-15



2-pipe buffers should only be used when the heat source is turned on and off based on buffer tank temperature. The reasoning is as follows: if the heat pump flow rate and load flow rate are similar, there will be very little flow through the tank. This could cause the heat pump to shut off based on satisfying the space heating thermostat, without adding much heat to the tank. In this scenario, the tank is not well “engaged” in the heat flow process. However, when the heat pump is controlled directly from tank temperature, the heat pump should continue to run even after the space heating thermostat is satisfied, storing heat that’s immediately ready to flow to the next zone requesting it.

The 3-pipe buffer tank configuration is an excellent compromise between the strengths and limitations of the 4-pipe and 2-pipe configurations. It provides a way to pass heat directly from the heat pump to the load when both are operating. It also forces flow returning from the load through the lower portion of the tank, and thus, ensures that the tank’s thermal mass is engaged. This “direct-to-load” configuration has been modeled using TRNSYS simulation software and found to increase the seasonal COP of the heat pump by keeping the lower portion of the tank slightly cooler than is possible with other piping configurations.

Buffer tanks connected to heat pumps tend to have minimal temperature stratification. This happens because most heat pumps have recommended flow rates of 3 gpm per ton (12,000 Btu/hr) of capacity. A typical 4-ton air-to-water heat pump operating at these conditions would “turn over” an 80-gallon buffer in less than 7 minutes. Those flow rates, especially if introduced vertically into the tank, create lots of internal mixing.

Figure 7-13 shows an example of an air-to-water heat pump supplying a highly zoned distribution system through a buffer tank with a 3-pipe configuration.

The split system air-to-water heat pump supplies a combination of low-temperature panel radiators and radiant floor panel circuits. The heat output of each radiator and radiant panel circuit is regulated by a non-electric thermostatic valve. A variable-speed pressure-regulated circulator automatically adjusts speed based on the status of these valves. The heat pump is turned on an off to maintain the water temperature at the middle of the buffer tank between 100 and 110°F. In addition to buffering the heat pump against short cycling, the tank provides hydraulic separation between the heat pump’s internal circulator (P1) and the variable-speed distribution circulator (P2).

## DIRT SEPARATION

The refrigerant-to-water heat exchangers used in modern air-to-water heat pumps often have narrow flow passages. It is essential that the fluid passing through these heat exchangers is clean. A low-velocity zone dirt separator, as shown in Figure 7-13, is one solution. If the system has one or more ECM-based circulators, a magnetic dirt separator is recommended.

## AUXILIARY HEATING PROVISIONS

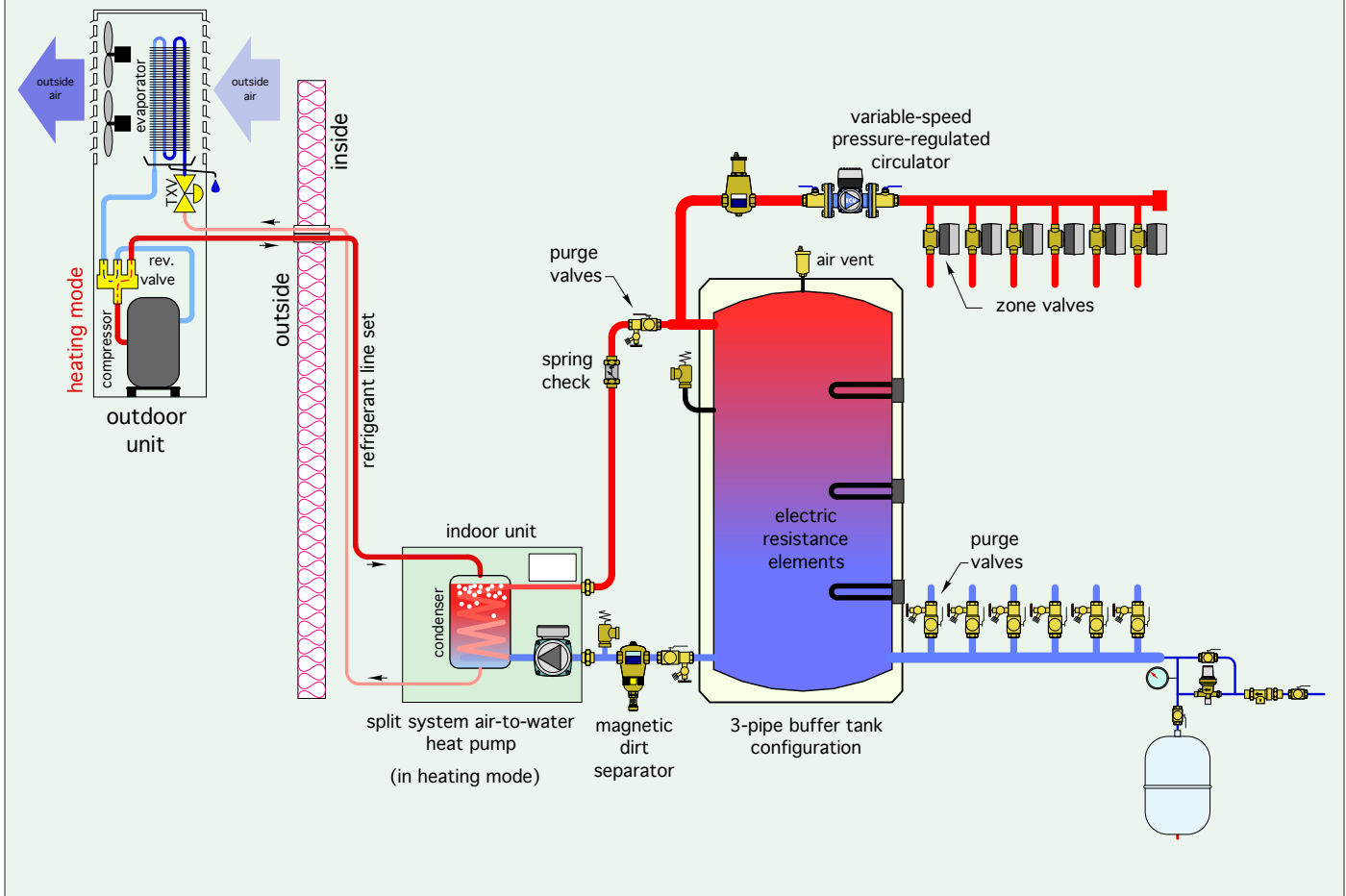
Although it’s possible to size an air-to-water heat pump such that it can supply design heating load, or even for that load plus a safety factor, such sizing is not always economically justified. The Boston house example described in section 5 demonstrated that a low-ambient air-to-water heat pump having a balance point capacity of about 65% of the design load could supply about 96% of the total seasonal heating energy. The nominal 4-ton heat pump would have to be increased to a nominal 6-ton unit to meet the design load. Under typical partial load conditions, the larger heat pump would be more subject to short cycling. The suggested solution was to install an electric boiler that could provide supplemental heat during the few hours in a typical year when it was needed, while still maintaining the heating system as “all electric.”

Figure 7-14 shows an electric boiler piped in parallel with the air-to-water heat pump.

Using an electric boiler for auxiliary heat has the advantage of allowing a wider selection of possible buffer tanks and boiler heating capacity. A wider selection of buffer tanks is possible because of the limited number of buffer tanks currently available with internal electric heating elements. The boiler could be sized to provide *supplemental* heating in combination with the heat pump, or it could be sized to provide 100% backup heating, even at design loads should the heat pump be down for service. Large electric boilers operating on typical residential 240 VAC single phase power can require high amperage. The minimum electric service panel rating should be 200 amps. Electric boilers also allow easy servicing since they are not integrated into a tank or other system component.

One prerequisite of using an electric boiler is that a separate circulator is required. The system shown in Figure 7-14 has this circulator. It also has a check valve, purging valve and pressure-relief valve in each heat source circuit. These components allow the buffer tank to be supplied by either heat source or both at the same time. They also allow either heat source to be completely isolated from the remainder of the system without having to shut down heat delivery.

Figure 7-16



New or existing boilers fueled by natural gas or propane and designed to operate with sustained flue gas condensation can be piped the same as an electric boiler. Conventional boilers with heat exchangers made of cast iron, carbon steel or finned copper tubing are generally not designed to operate with sustained flue gas condensation. If one of these conventional boiler types is used in a low-temperature distribution system, it must be protected by a thermostatic mixing valve. Figure 7-15 shows piping options for combining a mod/con or a conventional boiler into a system where an air-to-water heat pump supplies a 3-pipe buffer tank.

Another possibility is to use a buffer tank that has built-in electric heating elements, as shown in Figure 7-16.

This approach eliminates the need to pipe an external boiler and its associated circulator. It also reduces the space needed in the mechanical room. The downside to this approach is that servicing a failed electric heating element

requires the tank to be taken offline and at least partially drained. There can be no heat flowing to the system when this type of service is required.

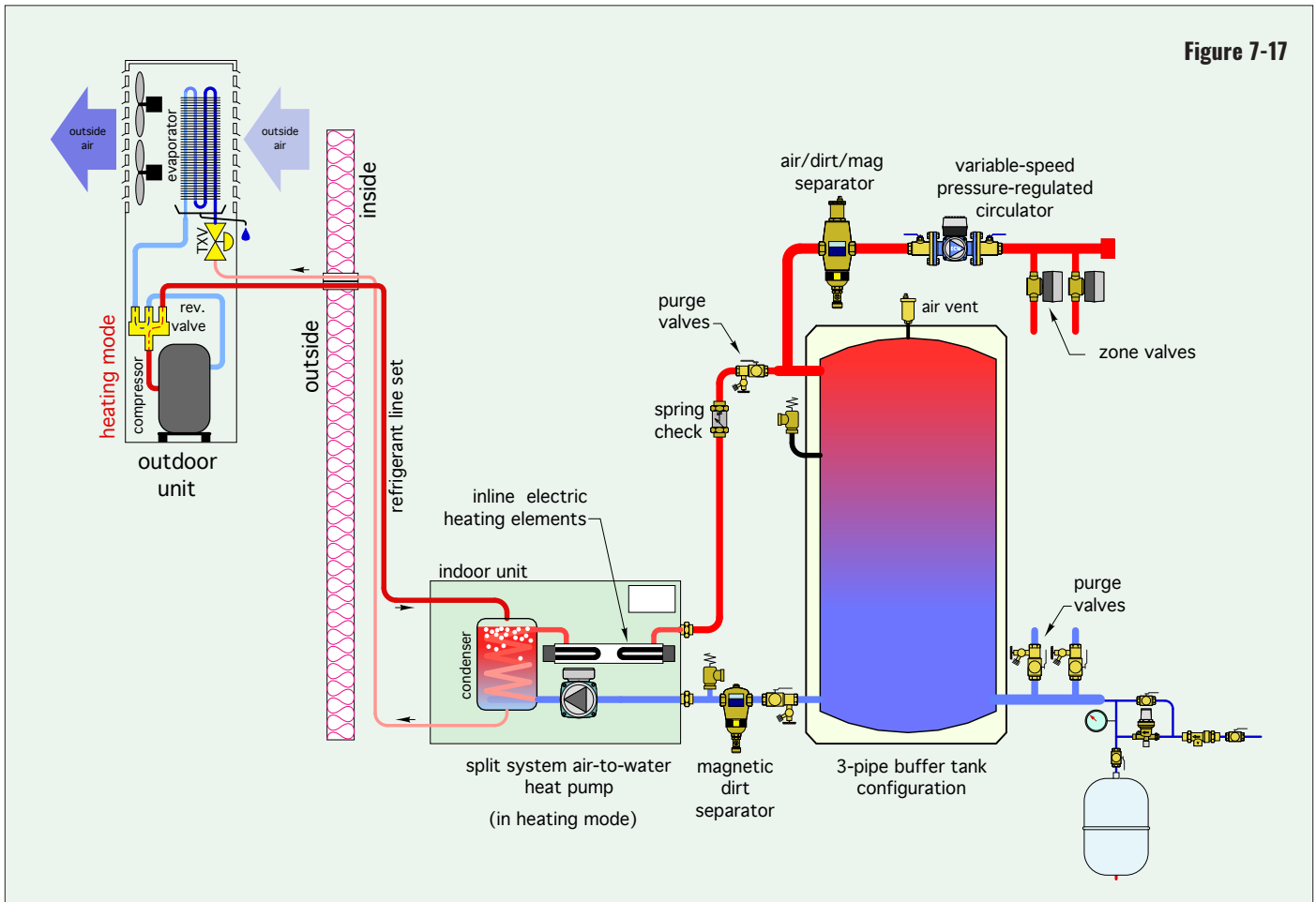
Yet another option is to use a heat pump with an inline electric heater incorporated into the indoor unit, as shown in Figure 7-17.

### CONTROLLING AUXILIARY HEAT

The auxiliary heat source can be automatically controlled in several ways. They include:

- Use of 2-stage room thermostat
- Use of 2-stage setpoint controller based on buffer tank temperature
- Use of 2-stage outdoor reset controller while monitoring buffer tank temperature
- Directly from the heat pump's internal controller

Figure 7-17



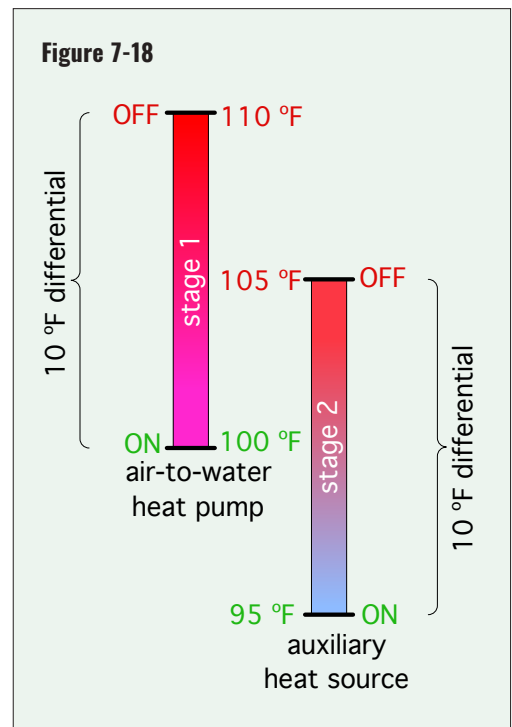
## 2-STAGE ROOM THERMOSTAT

Two-stage room thermostats have been extensively used in single zone forced-air systems using air-to-air heat pumps equipped with electric strip heat. If the room thermostat was unable to maintain the room setpoint temperature using the heat pump alone, the 2nd stage of the thermostatic would operate a contactor that in turn turned on the electric strip heaters in the supply air plenum. It was also possible to set the thermostat to “emergency” mode when the heat pump was down for service. This would operate the electric strip heat as the system’s sole heat source. Two-stage room thermostats were designed for this type of low thermal mass, single zone system. As such they are not well-suited for higher thermal mass hydronic systems, especially those with multiple zones.

## 2-STAGE SETPOINT CONTROL

Two-stage setpoint control based on the temperature in a buffer tank is a simple control technique for a hydronic system where the air-to-water heat pump serves as the first stage heat source with a boiler serving as second stage heat source. A single temperature sensor in the buffer tank provided feedback to the 2-stage controller. Each stage of heat input has a setpoint and differential, both of which are adjustable. One possible control configuration for a low-temperature distribution system is given in Figure 7-18.

Figure 7-18





With these settings, the air-to-water heat pump would turn on when the tank temperature sensor dropped to 100°F. It would continue to run until the tank temperature increased to 110°F. The auxiliary heat source would only turn on if the tank temperature dropped to 95°F. Once on, the auxiliary heat source would continue to run until the tank temperature climbed to 105°F.

Modern digital temperature controllers allow the setpoint and differential of both stages to be adjusted over a wide range. The operating ranges can overlap, as shown in Figure 7-18, or not. The exact settings for a given system are usually determined by experience. They represent a compromise between not allowing excessively wide temperature swings that could negatively affect comfort, and preventing short cycling of the heat sources, which increases wear and ultimately shortens service life.

Some two-stage setpoint controllers also allow an interstage time delay to be in effect. This sets the minimum time that must elapse between when the stage 1 contacts close, and when the stage 2 contacts close. Providing a nominal 3- to 5-minute interstage time delay allows time for the air-to-water heat pump to reach a stable operating condition, and thus establish the likely temperature trending at the tank sensor (increasing, decreasing or stable). This helps prevent short cycling of the second stage heat source at times when the air-to-water heat pump will eventually be able to satisfy the load requirement.

Two-stage setpoint control is a good strategy when the buffer tank is expected to maintain a minimum temperature, regardless of the prevailing space heating load. One example is when the buffer tank will be used to preheat domestic water.

## 2-STAGE OUTDOOR RESET CONTROL

Two-stage outdoor reset control is similar in some ways to 2-stage setpoint control. The goal is to operate both heat sources as necessary to maintain a target water temperature in the buffer tank. That target temperature increases as the outdoor temperature decreases, and vice versa. Allowing lower target temperatures in the buffer tank under partial load conditions increases the heat pump's seasonal COP. *This is a good control strategy when the buffer tank is only used to supply space heating.*

Some 2-stage outdoor reset controllers are capable of rotating the operating order of the two heat sources. When an air-to-water heat pump is used, it should always be the stage 1 heat source. Thus, the “boiler rotation” function of the controller should be disabled.

Most 2-stage outdoor reset controllers use PID (Proportional Integral Derivative) logic to determine when the contacts for each stage are closed. The farther the water temperature in the buffer tank varies from the target value, the longer this deviation exists, and the faster the temperature is changing, the more “aggressive” the controller gets in operating the heat sources. The controller's goal is to keep the measured water temperature in the buffer tank as close to the target value as possible.

## AUXILIARY HEAT SOURCE CONTROL LOGIC WITHIN THE HEAT PUMP

Some air-to-water heat pumps have relay contacts that automatically close when auxiliary heat is needed. These heat pumps have user settings to fine-tune when the auxiliary heating contacts will operate. These contacts may or may not be designed to directly operate 240 VAC heating elements. Designers should always verify the maximum ampacity of the auxiliary relay contacts. If the contacts do not have suitable ampacity, the auxiliary relay in the heat pump can be used as a “pilot relay” that energizes the coil of another contactor having contacts with sufficient ampacity.

## MONITORING HEAT PUMP PERFORMANCE

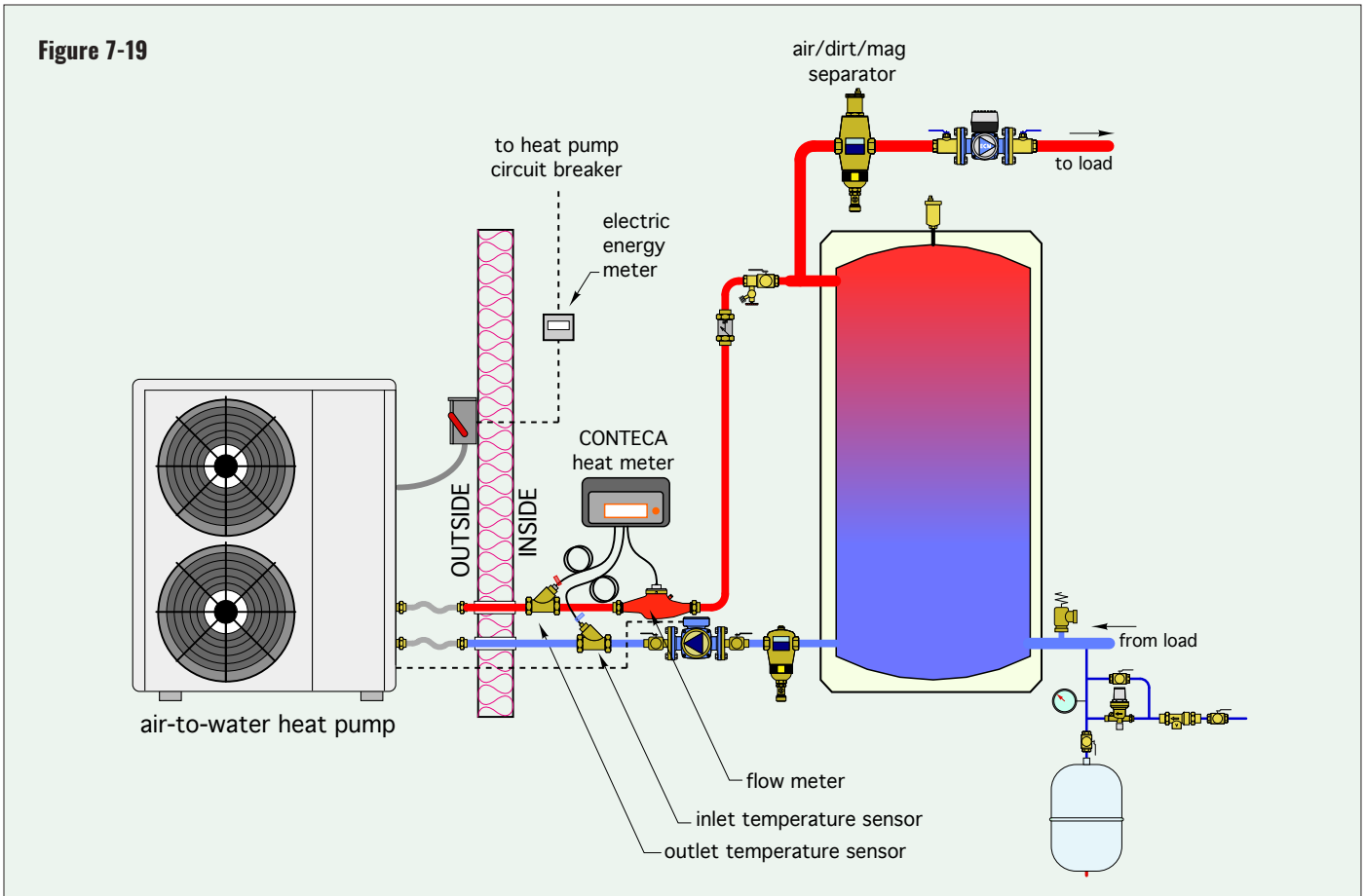
Modern heat meters, used in combination with electrical energy meters, make it possible to determine the heating output and COP of an air-to-water heat pump over a wide range of time intervals. Figure 7-19 shows how these meters would be installed.

The Caleffi CONTECA heat meter uses two precision temperature sensors to measure the difference between the fluid temperatures entering and exiting the heat pump (e.g.,  $\Delta T$ ). It also accepts a signal from the flowmeter. By combining the instantaneous values of  $\Delta T$  and flow rate, the CONTECA meter can calculate the rate of heat transfer. It can also integrate this calculation over time to determine the total thermal energy supplied by the heat pump. If a monobloc heat pump is used, the system will likely be operating with an antifreeze solution. Be sure the CONTECA meter has been configured for the fluid properties of that antifreeze solution.

A small and relatively inexpensive electric meter, such as shown in Figure 7-20 can be used to gather electric energy use.

This particular meter doesn't display instantaneous power. However, it does deliver 800 pulses per kilowatt-hour of electrical energy use. When connected to a typical residential heat pump drawing 3,000 to 4,000 watts, the average power over a time increment of 1 to 2 seconds could be determined by a pulse counting device.

**Figure 7-19**



**Figure 7-20**



The heat pump's average COP over a given time increment can be calculated using Formula 7-2.

**Formula 7-2**

$$COP_{ave} = \frac{\text{Btu delivered}}{(\text{kwhr used}) \times 3413}$$

Where:

COP<sub>ave</sub> = average COP of some time increment (hour, day, month, etc.)

kwhr used = number of kwhr metered over the same time interval

3,413 = number of Btus per kwhr



**idronics #24 gives more in depth information on how to apply heat meters in a variety of situations.**

## DOMESTIC WATER HEATING OPTIONS

There are several possible ways to use an air-to-water heat pump for domestic water heating. They include:

- Using a desuperheater
- Using an indirect water heater with large internal coil
- Using a reverse indirect water heat
- Using an “on-demand” external heat exchanger with flow switch

### DESUPERHEATER OPTION

Some split system air-to-water heat pumps can be ordered with desuperheater heat exchangers. These were discussed in section 4. The desuperheater absorbs heat from the hot refrigerant vapor leaving the heat pump’s condenser and transfers it into a stream of domestic water. This heat transfer takes place whenever the heat pump’s compressor is operating, in heating or cooling mode. Some

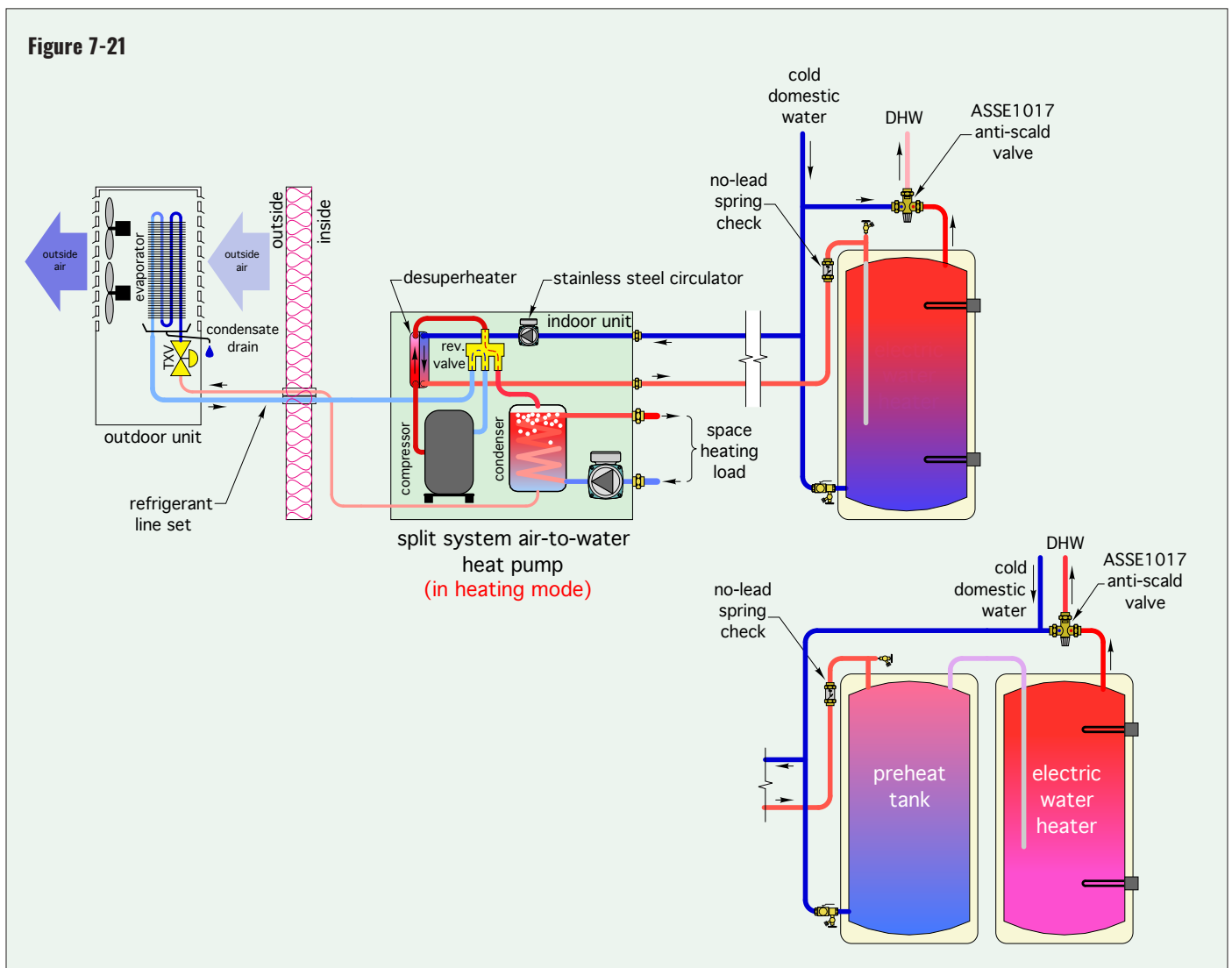
heat pumps turn off their internal desuperheater circulator when the leaving domestic water temperature reaches 130°F. Figure 7-21 shows a typical piping arrangement for this approach to domestic water heating.

When the space heating system operates at low water temperature, or when the domestic hot water delivery temperature is high, the “two-tank” configuration is preferred. It allows the refrigeration system to operate at lower temperatures, which improves the heat pump’s COP.

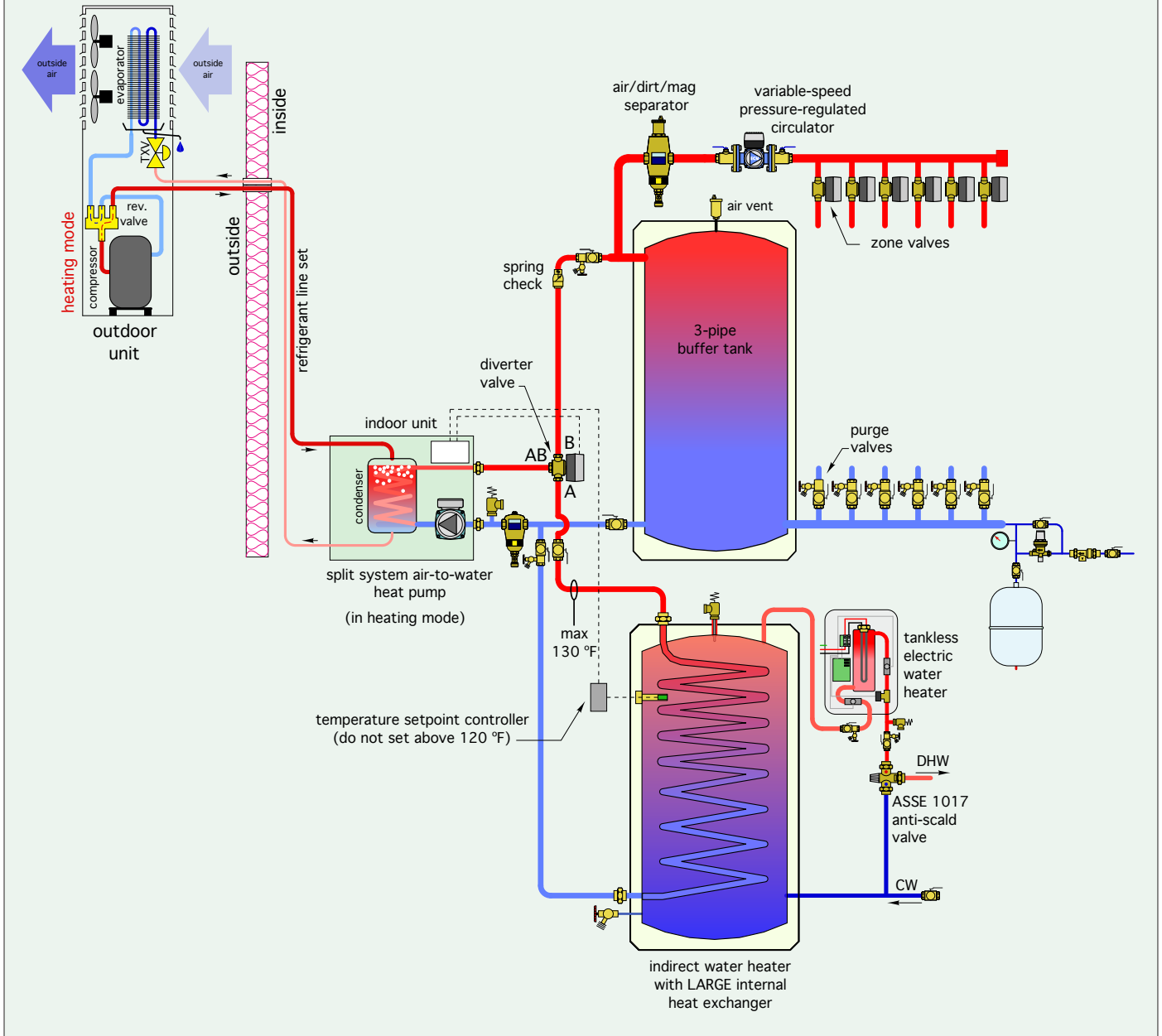
### INDIRECT TANK OPTION

Another option is to connect an indirect water heater as a priority zone. Since the water heater has several gallons of thermal mass, the buffer tank is bypassed in this mode of operation. One possible configuration is shown in Figure 7-22.

Figure 7-21



**Figure 7-22**



When this approach is used, it is critically important to specify an indirect tank with a large internal coil that allows the full rate of heat production from the heat pump to dissipate into the domestic water without forcing the heat pump's condenser temperature above 130°F. Failure to do this will cause the heat pump's operating temperature to quickly rise to a value at which an internal safety control turns off the compressor.

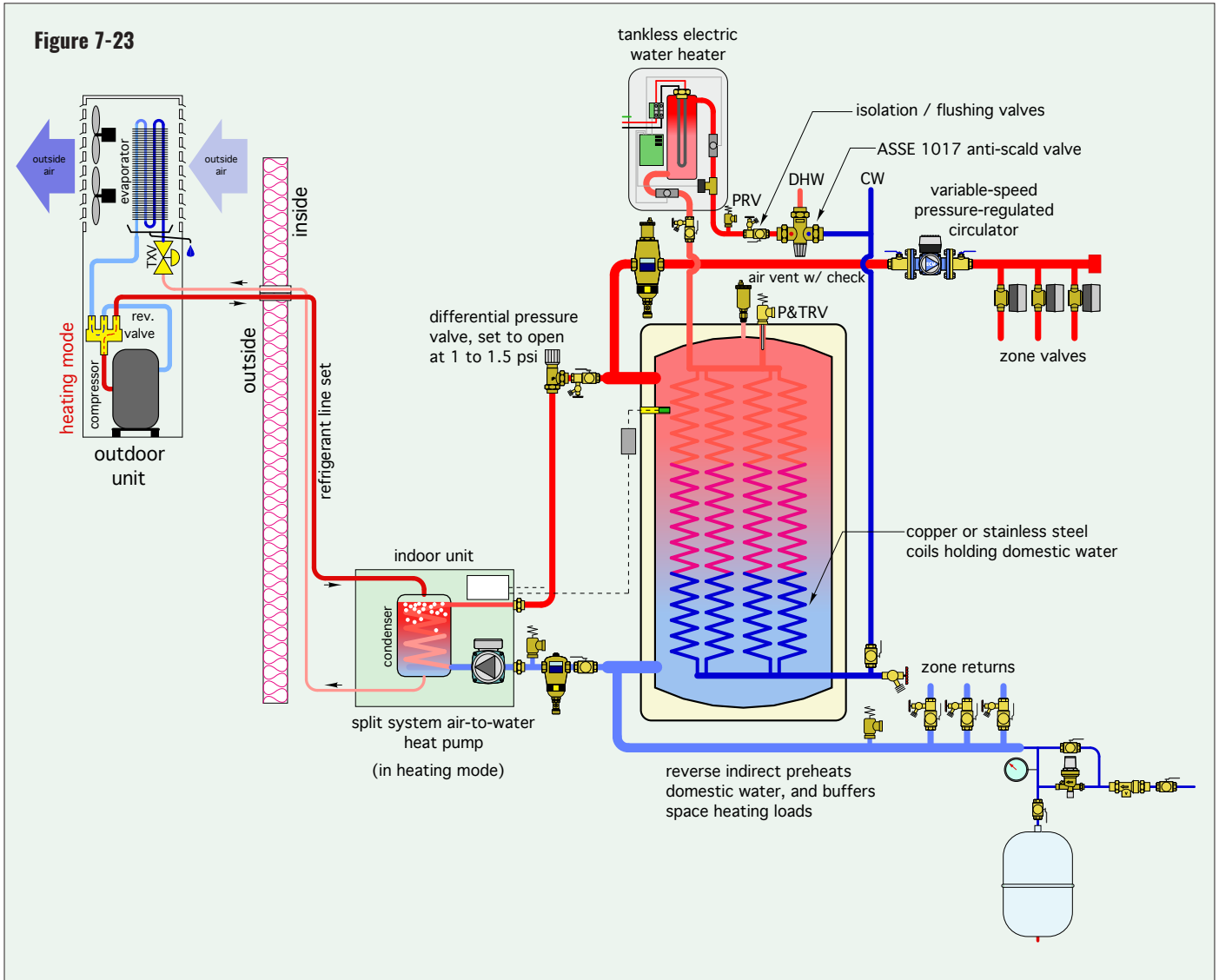
It is also important to realize that the heat pump alone cannot generate domestic water temperature above

approximately 120°F. When higher DHW delivery temperatures are needed, some form of supplemental heat input will be required. An electric tankless water heater is one option, as shown in Figure 7-22.

Some air-to-water heat pumps have internal controls that can treat domestic water heating as a priority load. When a contact in the setpoint controller monitoring water temperature in the indirect tank calls for heating, the heat pump powers on the diverting valve. This directs all flow from the heat pump through the internal coil of the



**Figure 7-23**



indirect tank. Some air-to-water heat pumps can produce a leaving water temperature of up to 130°F under this mode of operation. Be sure that the setpoint controller monitoring the water temperature in the tank is not set so high that the heat pump can never achieve the setting. A maximum suggested setting for the tank thermostat is 5°F lower than the maximum leaving water temperature from the heat pump.

One advantage of this approach is that the buffer tank can function in both heating and cooling modes. If the heat pump is operating in cooling mode when a call for domestic water heating occurs, the heat pump switches to heating mode. The diverter valve directs the heated water through the coil of the indirect water heater. Chilled water in the buffer tank is still available to flow to cooling loads while the indirect tank is being heated.

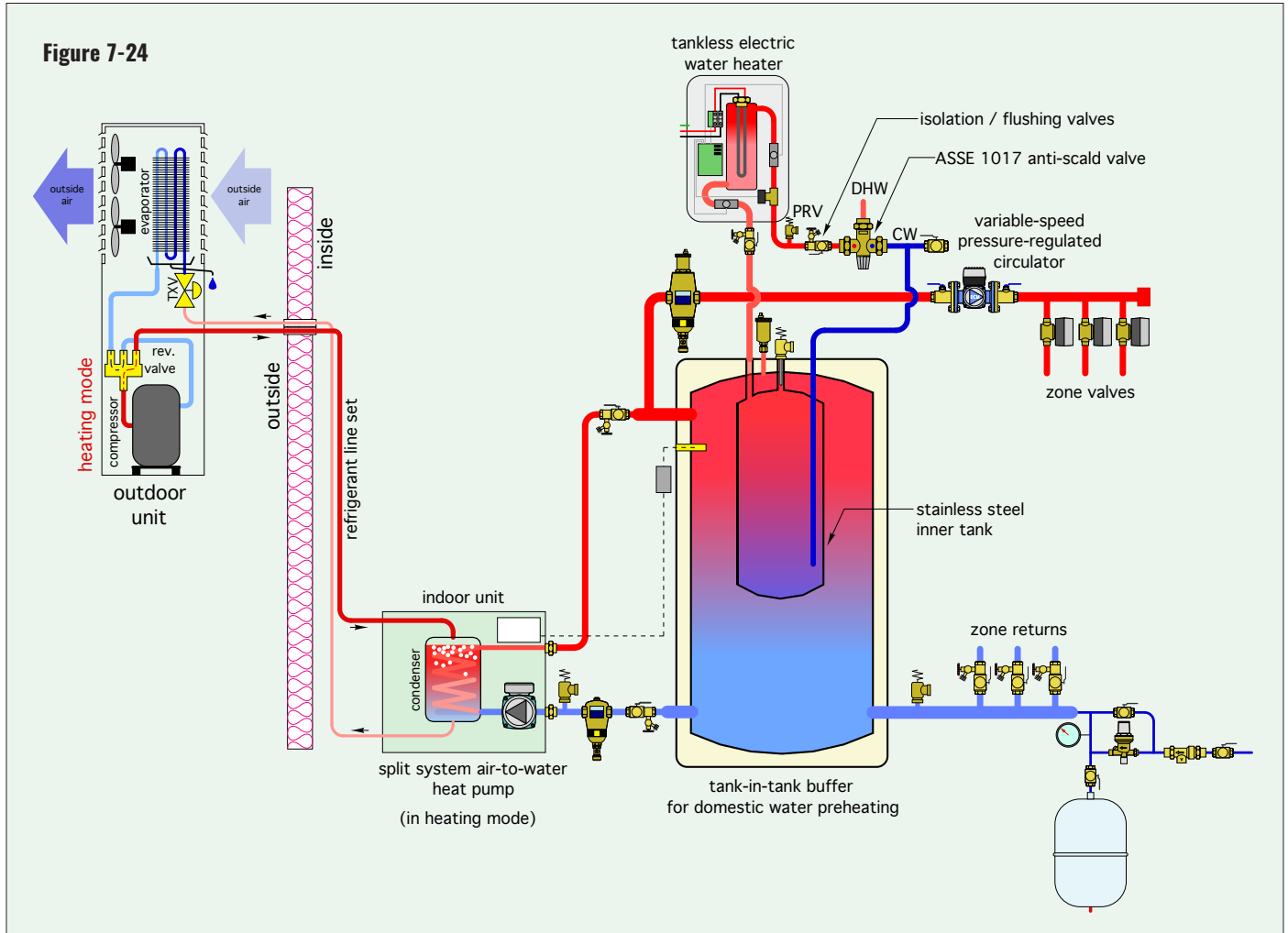
### REVERSE INDIRECT TANK OPTION

Another approach to domestic water heating uses a “reverse” indirect tank to preheat domestic water. A tank-type or tankless electric water heater serves as the supplemental heater. An example of this approach in a “heating only” system, is shown in Figure 7-23.

This approach allows the thermal mass of the reverse indirect to buffer space heating as well as domestic water heating. It eliminates the need of a second tank. The tank should be equipped with a large internal coil heat exchanger to minimize the temperature difference between the water in the tank shell and the domestic water inside the coils.

The system shown in Figure 7-23 configures the tank as a “2-pipe” buffer. A motorized ball valve is installed in the

Figure 7-24



pipework from the heat pump to the tank. This valve opens when the heat pump operates and closes at all other times. The valve has two purposes. First, it prevents water returning from the space heating zones from passing through the heat pump when it is off. If otherwise allowed, this flow would needlessly dissipate heat into the mechanical room and lower the temperature of the water supplied to the space heating zones. The motorized ball valve also prevents reverse thermosiphoning from the tank through the heat pump when the latter is off. This second function eliminates the need to install a check valve to prevent reverse thermosiphoning.

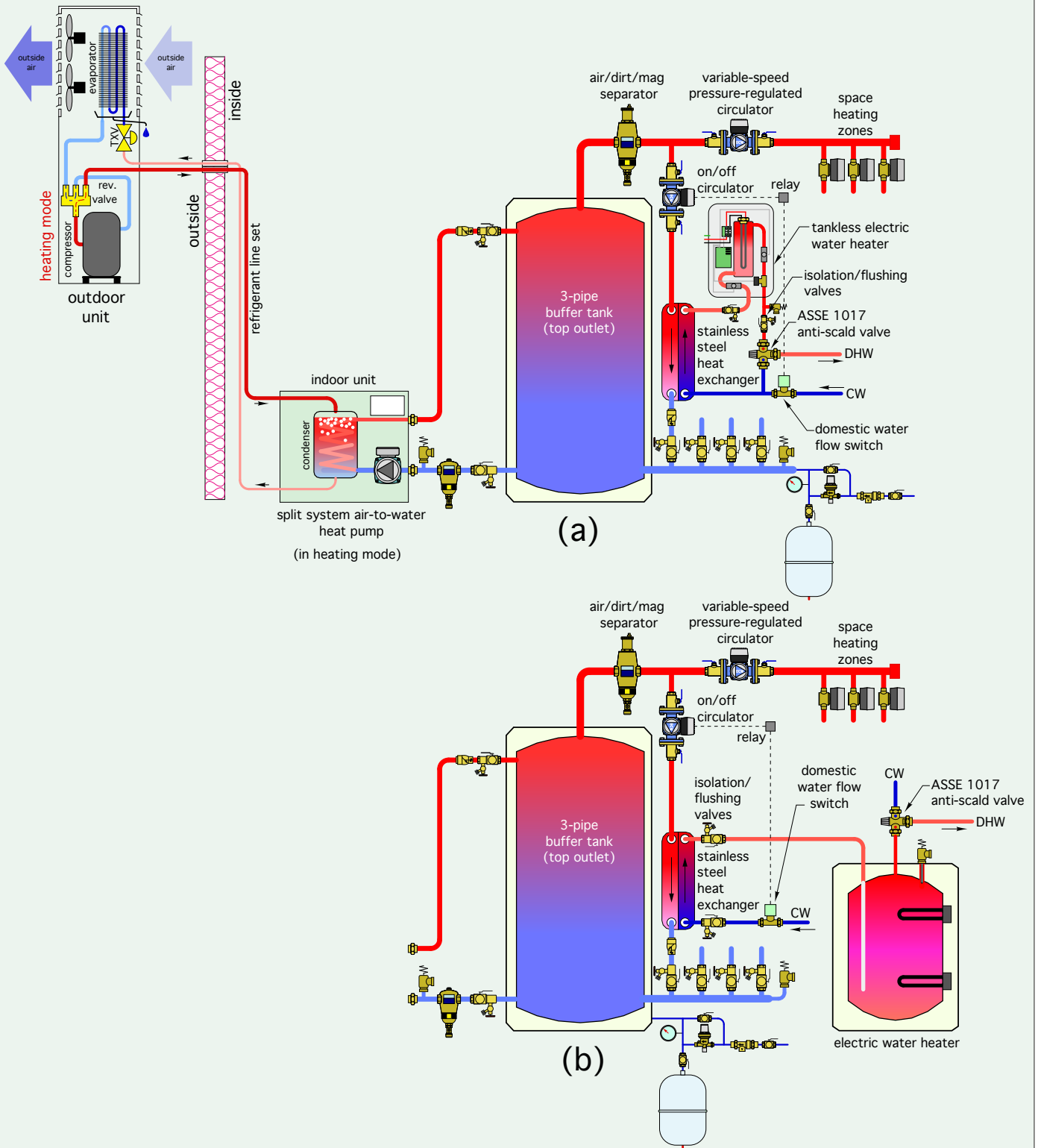
Another option is to use a “tank-in-tank” buffer, as shown in Figure 7-24.

The buffer tank has an internal stainless steel tank which holds domestic water. The inner tank is surrounded by heated water supplied from the heat pump and used for the space heating distribution system. Cold domestic

water flows into the inner tank whenever hot water is drawn at a fixture. The temperature of the heated domestic water leaving the inner tank depends on the temperature of the water in the outer tank, as well as the rate at which domestic hot water is drawn. An electric tankless water heater is shown as a supplemental heater to boost the domestic water to the required delivery temperature. An ASSE 1017-rated mixing valve is used to ensure that the temperature of the water delivered to the plumbing distribution system doesn't exceed 120°F.

These approaches to domestic water heating could be used in a system that supplies chilled water for cooling. However, if the cooling distribution system is zoned, and the heat pump uses a fixed-speed compressor, a separate chilled water buffer tank would be required. If a variable-speed heat pump is used, and the cooling distribution system is not highly zoned, the chilled water buffer tank could likely be eliminated.

Figure 7-25



## ON-DEMAND EXTERNAL HEAT EXCHANGER OPTION

It's also possible to use an *external* heat exchanger to preheat domestic water using heat from a buffer tank. This option is illustrated in Figure 7-25.

Whenever there is a draw for domestic hot water that exceeds some minimum threshold (typically 0.6 to 0.7 gpm in residential systems), a flow switch closes its contacts. This energizes a relay, or one zone in a multi-zone relay center, to power up a circulator that creates flow from the top of the buffer tank through the primary side of a stainless steel brazed plate heat exchanger. Cold domestic water passes in counterflow through the other side of this heat exchanger and absorbs heat. The temperature of the domestic water leaving the heat exchanger depends on the temperature at the top of the buffer tank and the size of the heat exchanger. A suggested guideline is to size the heat exchanger for a 5°F approach temperature difference based on the average anticipated water temperature at the top of the tank and the design flow rate of cold domestic water entering the heat exchanger.

In Figure 7-25a, the preheated water leaving the heat exchanger passes through a tankless electric water heater, which supplies any necessary boost in temperature. Domestic hot water leaving the tankless heater passes through an ASSE 1017-rated mixing valve before flowing to hot water fixtures. It's also possible to use a tank-type water heater in place of the tankless heater, as shown in Figure 7-25b.

In systems supplying heating and cooling, the circulator associated with the domestic water heat exchanger would be disabled during cooling mode operation. This would transfer the full domestic water heating load to the tankless water heater.

## CHILLED WATER COOLING DETAILS

One of the foremost benefits of air-to-water heat pumps is their ability to provide both heating and cooling. The latter is provided as a stream of chilled water typically ranging in temperature from 45 to 60°F. There are several potential ways to use chilled water to reduce the temperature of interior air, as well as lower its moisture content. Some of those details are discussed below.

## CHILLED WATER TERMINAL UNITS

The most common way of using chilled water for building cooling is to route it through copper tubing in a water-to-air heat exchanger, often called a "coil," while room air passes by aluminum fins attached to the copper tubing. The air movement across the coil is created by a fan or a blower. This combination of hardware has several different names depending on its size and mounting location. They include:

- Fan-coil
- High wall cassette
- Air handler

A fan-coil is intended to be mounted within a finished interior space. In most cases, it is located at the base of a wall just above the floor. Some fan-coils are surface mounted, others are mounted into a recessed cavity in the wall. Figure 7-26 shows an example of a modern surface-mounted fan-coil.

**Figure 7-26**



Courtesy of Myson

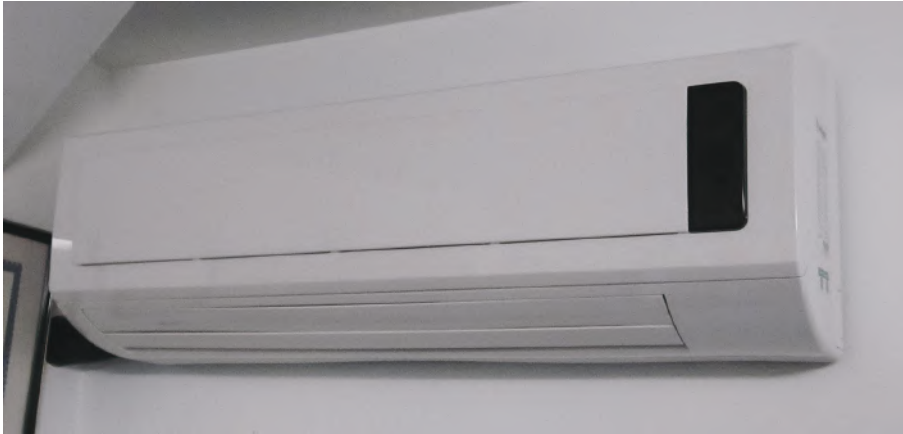
*Any fan-coil used for chilled water cooling must include a condensate drip pan.* This pan catches water droplets falling from the coil and routes them to a piping connection. A plastic drain tube then carries the condensate to a suitable drain.

Most fan-coils must be supplied by line voltage to operate the fan or blower. Some can be turned on and off by a single contact closure, while others are operated by a handheld remote. Most have integral controls that allow the fan speed to be set as desired.

A high wall cassette is a special type of fan-coil designed to be mounted a few inches below ceiling level. An example is shown in Figure 7-27.



**Figure 7-27**



High wall cassettes draw in air at the top of the unit and discharge it through a slot at the front base. Most have an oscillating damper that opens when the unit is operating and closes when it is off. The movement of the damper is designed to mix discharge air with room air. Most high wall cassettes are operated by a handheld remote. Internal contacts within the cassette allow it to turn on a circulator or valve for either heating or cooling mode.

Air handlers are designed to be mounted in utility spaces and connected to ducting systems. Many modern air handlers can be mounted in either vertical or horizontal orientations. Figure 7-28 shows an example of the latter.

This air handler has a nominal cooling rating of 3 tons (36,000 Btu/hr). It is supplied with chilled water from a heat pump by a set of 3/4" pre-insulated PEX tubes. The condensate drain with trap is seen in the lower front of the unit. The condensate trap prevents the air handler from sucking in air through the drainage pipe. This trap is also equipped with an overflow switch that turns off the blower if a blockage within the trap prevents condensate from properly draining. Notice that the 3/4" PVC drain pipe is sloped downward for gravity drainage. This air handler is also mounted above a secondary drain pan, which would catch any water leaking from the primary drain pan.

**Figure 7-28**



The air handler is powered through a disconnect switch seen on the wall at the left of the opening. This air handler has a removable filter between the air intake plenum at the right and the air intake. The air handler is fully accessible by removing a panel not shown in the photo. This air handler could also be configured for vertical mounting by rotating the orientation of the chilled water "A-coil" within the cabinet.

The cooling ratings of fan-coils and air handlers are typically divided into two categories: sensible cooling and total cooling. Sensible cooling is the ability of the air handler to lower the temperature of air passing through it at a specific combination of entering water temperature, entering air temperature, water flow rate and airflow rate.

Latent cooling is the ability of the unit to remove moisture from the airstream passing through it. The latent cooling capacity of an air handler or fan-coil is found by subtracting sensible cooling capacity from total cooling capacity.

Cooling load calculations develop values for sensible and latent cooling capacity. Designers need to reasonably match the ratio of sensible-to-total cooling capacity of an air handler to the ratio of sensible cooling load-to-total cooling load. The latter would be determined through cooling load calculations.

## **PREVENTING UNWANTED CONDENSATION**

Perhaps the most critical detail on chilled water cooling systems is preventing condensation on piping and other components conveying chilled water. If surrounding air comes in contact with chilled surfaces that are below the dewpoint of that air, condensation will occur. This will cause superficial oxidation of any ferrous metal components, even those with a painted or coated surface, as seen in Figure 7-29.

**Figure 7-29**



The accumulating condensate will eventually drip from the piping and possibly damage whatever is under it. Such damage includes stained ceilings, deteriorating materials and eventual growth of mold and mildew.

To prevent condensation, the outer surface of the pipe insulation must be maintained above the dewpoint of the surrounding air. Insulation manufacturers typically provide tables, such as shown in Figure 7-30, that relate the minimum insulation wall thickness required to prevent surface condensation to the temperature of the fluid in the tube, and the air conditions surrounding the insulation, including dry bulb temperature and relative humidity.

It is also important to use insulation that has a low vapor permeability, or to wrap insulation materials, such as fiberglass, that have relatively high

**Figure 7-30**

Suggested wall thickness of elastomeric foam insulation to prevent surface condensation (R-value of approximately 7.0 °F•hr•ft <sup>2</sup> /Btu per inch of wall thickness).			
fluid temperature in pipe →		50 °F	35 °F
normal conditions 85°F / 70% RH	3/8" ≤ d ≤ 1.25"	3/8"	1/2"
	1.25" < d ≤ 2"	3/8"	1/2"
	2" < d ≤ 2.5"	3/8"	1/2"
	2.5" < d ≤ 6"	1/2"	3/4"
mild conditions 80°F / 50% RH	3/8" ≤ d ≤ 2.5"	3/8"	3/8"
	2.5" < d ≤ 6"	1/2"	1/2"
severe conditions 90°F / 80% RH	3/8" ≤ d ≤ 1.25"	3/4"	1"
	1.25" < d ≤ 3.5"	3/4"	1"
	3.55" < d ≤ 6"	3/4"	1"

← dry bulb air temperature     
 ← relative humidity     
 ← nominal pipe size

vapor permeability with a continuous jacket that provides a vapor barrier. If such measures are not taken, the vapor pressure differential between the air surrounding the insulation and the cooler air near the pipe wall will cause vapor diffusion through the insulation. This vapor will condense on the pipe surface or within the insulation, and it could eventually saturate it with liquid water, leading to dripping, mold and significant loss of thermal resistance.

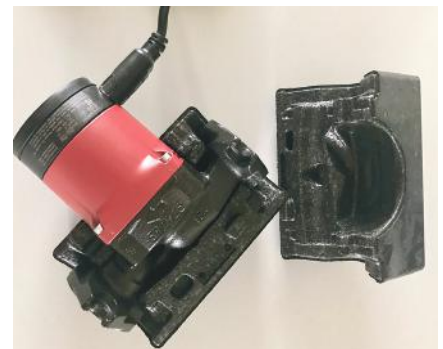
Closed-cell elastomeric foam insulations typically have low vapor permeability, as does cellular glass insulation. Acceptable vapor barriers for fiberglass or open-cell foam insulation include aluminum foil, PVC or composites using layers of paper, fiberglass scrim and aluminum foil.

Care should be taken to maintain the integrity of the insulation and its vapor barrier at all joints, as well as components such as valves, circulators, buffer tanks, unions and other pipe fittings. This often requires

**Figure 7-31a**



**Figure 7-31b**



Courtesy of Grundfos

**Figure 7-32**



use of sheet insulation products, carefully fitted to irregular shapes and applied with compatible adhesives.

The volute of a circulator is one of the more difficult components to insulate. Some circulator manufacturers offer preformed insulation shells for specific models of circulators. One example of such a shell is shown in Figure 7-31.

To be effective, these insulation shells must be fit tightly to the circulator volute and *be sealed at all edges and seams* to prevent air entry. It's also important to insulate isolation flanges or valves. Carefully cut and fit elastomeric foam insulation is one possibility. Self-adhering elastomeric foam tape is another. The motor on a circulator should never be insulated because it inhibits heat dissipation, especially if the same circulator operates during heating mode.

These insulation and vapor sealing techniques also apply to devices that combine electrical components with fluid conveying components. Examples include zone valves, diverter valves and mixing valves.

Figure 7-32 shows adequately insulated zone valves. Notice that the valve bodies are insulated, but the electrical actuators are not.

It is also important to support insulated piping that conveys chilled water so that the insulation does not undergo significant compression due to the forces transferred between the piping and its supports. Figure 7-33 shows one type of polymer tube support designed so that insulation can be nearly continuous along the copper tubing.

**Figure 7-33a**



**Figure 7-33b**



Courtesy of Andy Jones

The polymer tube support hangs from a piece of channel strut and closes around the copper tube. Elastomeric foam insulation slides into the support from both directions. A final detail is to caulk the perimeter seam where the insulation and tube support meet, which is not shown in Figure 7-33b.



**idronics #13 provides a more extensive discussion of chilled water cooling.**



## SYSTEM EXAMPLES

This section combines many of the details from previous sections into complete systems. These systems are configured around monobloc as well as split system air-to-water heat pumps. Some are “heating only” systems. Others include domestic water heating and chilled water cooling. Different approaches to details such as buffer tank piping, domestic water heating and auxiliary heat input are mixed into a variety of systems. Many additional combinations of these details and subassemblies are possible.

### SYSTEM #1

This system shown in Figure 8-1 is a single zone “heating only” application where a monobloc air-to-water heat pump supplies a heated floor slab. This simple arrangement would be ideal for a garage or shop environment.

The high thermal mass of the slab provides a substantial buffering effect, eliminating the need for a buffer tank. This system assumes that the heat pump can monitor the temperature of the fluid supplied to the floor heating manifold station. If the heat pump has a fixed-speed compressor, it would turn on and off to keep the supply

water temperature within some differential above and below a target temperature. If the heat pump had a variable-speed compressor, it would modulate to keep the supply temperature as close to the target value as possible. The target temperature could be a fixed value, suitable to meet design heating load, or it could be based on outdoor reset control. The latter has a significant advantage because it allows the system to operate at lower fluid temperatures as the outdoor temperature increases. This would significantly improve the heat pump’s seasonal COP.

The power to operate circulators (P1) and (P2) is supplied through the heat pump. For the highest seasonal COP, both of these circulators should have electronically commutated motors (ECM).

A SEP4 hydraulic separator provides high-efficiency air, dirt and magnetic particle separation for the system. It also provides hydraulic separation between circulators (P1) and (P2), allowing for stable, but potentially different flow rates. This configuration also allows continuous operation of circulator (P2), while circulator (P1) turns on and off with

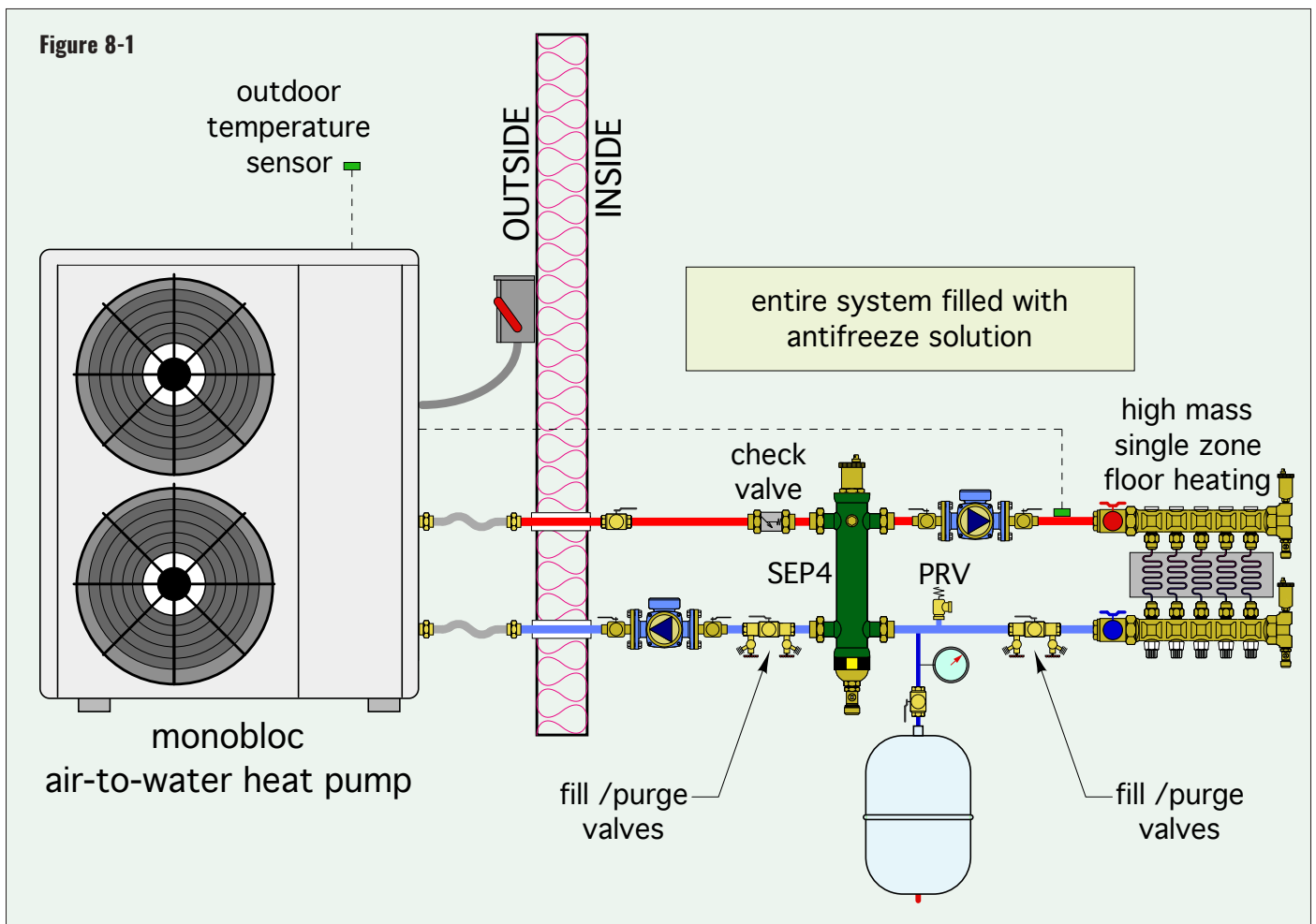
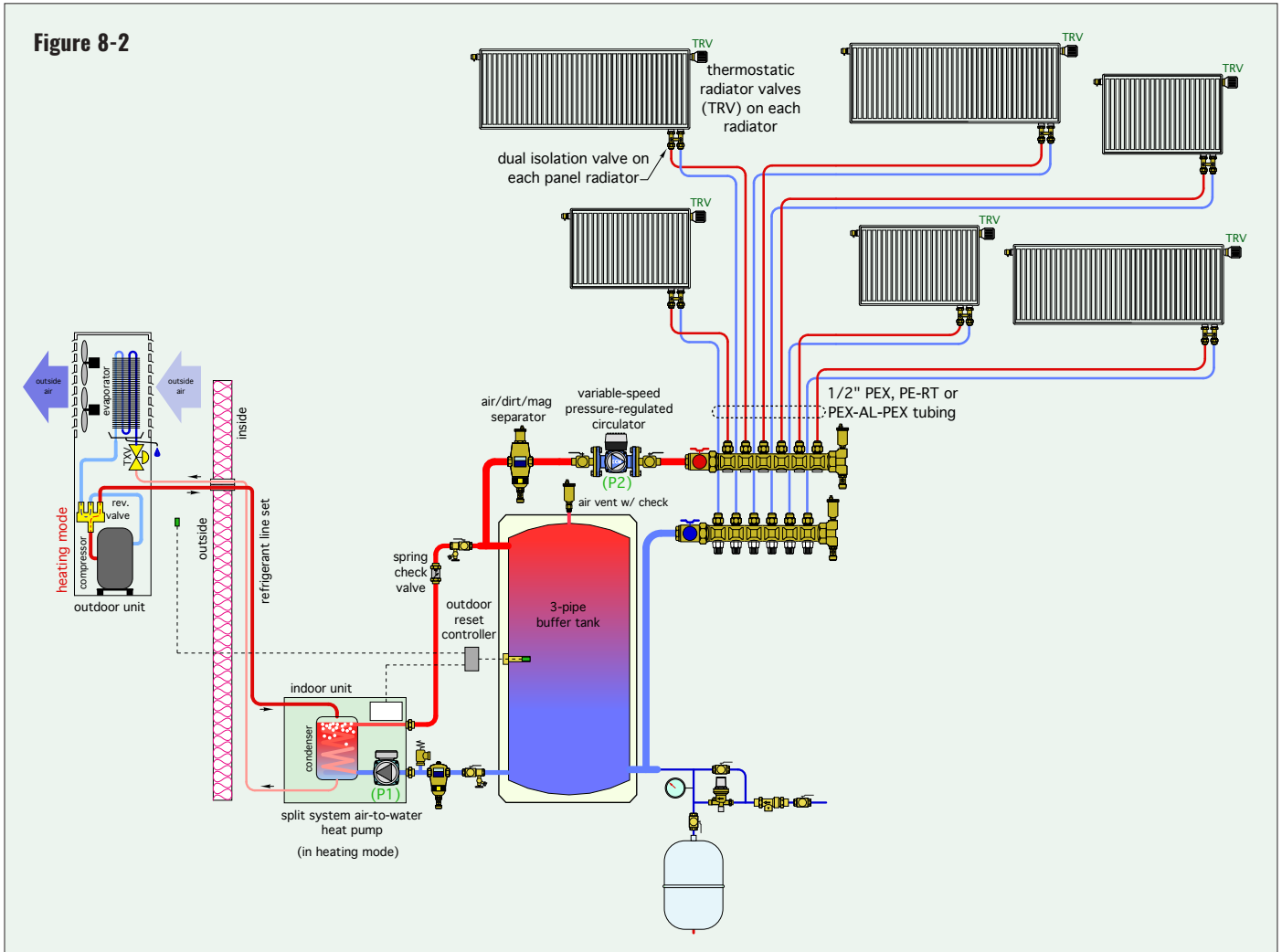




Figure 8-2



the heat pump. Continuous operation of the load circulator helps reduce variations in slab surface temperatures.

This system would operate with a solution of propylene glycol antifreeze. It is equipped with two bidirectional filling/purging valves to quickly fill the system and flush out bulk air during commissioning.

The spring check valve near the upper left connection on the SEP4 prevents reverse thermosiphoning through the heat pump when it is off.

The entire system can be operated from a simple wall thermostat.

## SYSTEM #2

The system shown in Figure 8-2, is another “heating only” application, but with extensive zoning of the heating distribution system.

The split system air-to-water heat pump operates to keep the water temperature in the buffer tank at a temperature based on outdoor reset control. This water supplies a homerun distribution system in which a single manifold station supplies six independently controlled panel radiators, each equipped with a thermostatic valve operator. Each panel radiator is sized to provide design heat output to its associated space when supplied with water at 120°F. As the outdoor temperature increases, the target temperature in the buffer tank decreases. This significantly increases the heat pump’s seasonal COP.

The buffer tank is piped in a 3-pipe configuration, which allows heat to flow directly from the heat pump to the heat emitters, as required, when the heat pump is operating. The balance of the flow leaving the heat pump passes through the buffer tank. All flow returning from the panel radiators passes into the lower portion of the tank. This helps ensure that the thermal mass of the tank is well “engaged” in the energy flow processes.

The heating distribution system is very simple. A single variable-speed pressure-regulated circulator operates continuously during the heating season. Its speed automatically increases and decreases to maintain a constant differential pressure as the thermostatic valves on the panel radiators open, close or modulate.

The thermostatic valve at each radiator allows it to operate as a separate zone, maintaining the desired comfort level in each space.

The split system heat pump allows the hydronic system to operate without need of antifreeze.

A magnetic dirt separator protects the heat pump's condenser. A combined air/dirt/magnetic particle separator protects the permanent magnet motor in the ECM circulator for iron oxide. It also provides high-efficiency air separation for the system.

A spring check valve prevents reverse thermosiphoning between the buffer tank and heat pump. This reduces extraneous heat loss through the piping when the heat pump is off.

The heat pump can be isolated from the balance of the system for service if necessary.

This system also has very simple control requirements. A single switch can be used to "enable" the system at the start of the heating season. The heat pump and its associated circulator turn on and off as necessary to maintain the target water temperature in the buffer tank. The distribution circulator (P2) operates continuously but is always tracking the differential pressure present based on the status of the thermostatic radiator valves. With current ECM circulator technology and proper component sizing, this distribution system could precisely deliver over 1,500 Btu/hr *per watt* of electrical energy supplied to circulator (P2). *This is much higher distribution efficiency than what could be attained with forced-air delivery.*

### SYSTEM #3

The system shown in Figure 8-3 supplies multiple zones of space heating with a mixture of heat emitters. It also provides domestic hot water.

A split system air-to-water heat pump is the primary heat source. An electric boiler, piped in parallel with the heat pump's condenser, provides a second stage of heat input if needed. It also provides a backup to the heat pump should it be down for service. Each heat source is equipped with a pressure relief valve and can be fully isolated if necessary. Depending on local codes, the

electric boiler may require a low water cutoff and a manual reset high limit controller.

Both heat sources supply a "tank-in-tank" buffer. The inner tank is constructed of stainless steel and holds 40 gallons of domestic water. It is surrounded by an outer tank that receives heat from the heat pump or the electric boiler. Heat flows from the system water in the outer tank to domestic water in the inner tank whenever the former is at a higher temperature than the latter. Domestic cold water enters the inner tank whenever there is a draw from a hot water fixture. The temperature of the domestic water leaving the inner tank depends on the temperature maintained in the tank shell. A thermostatically controlled tankless electric water heater boosts the domestic water to the desired supply temperature.

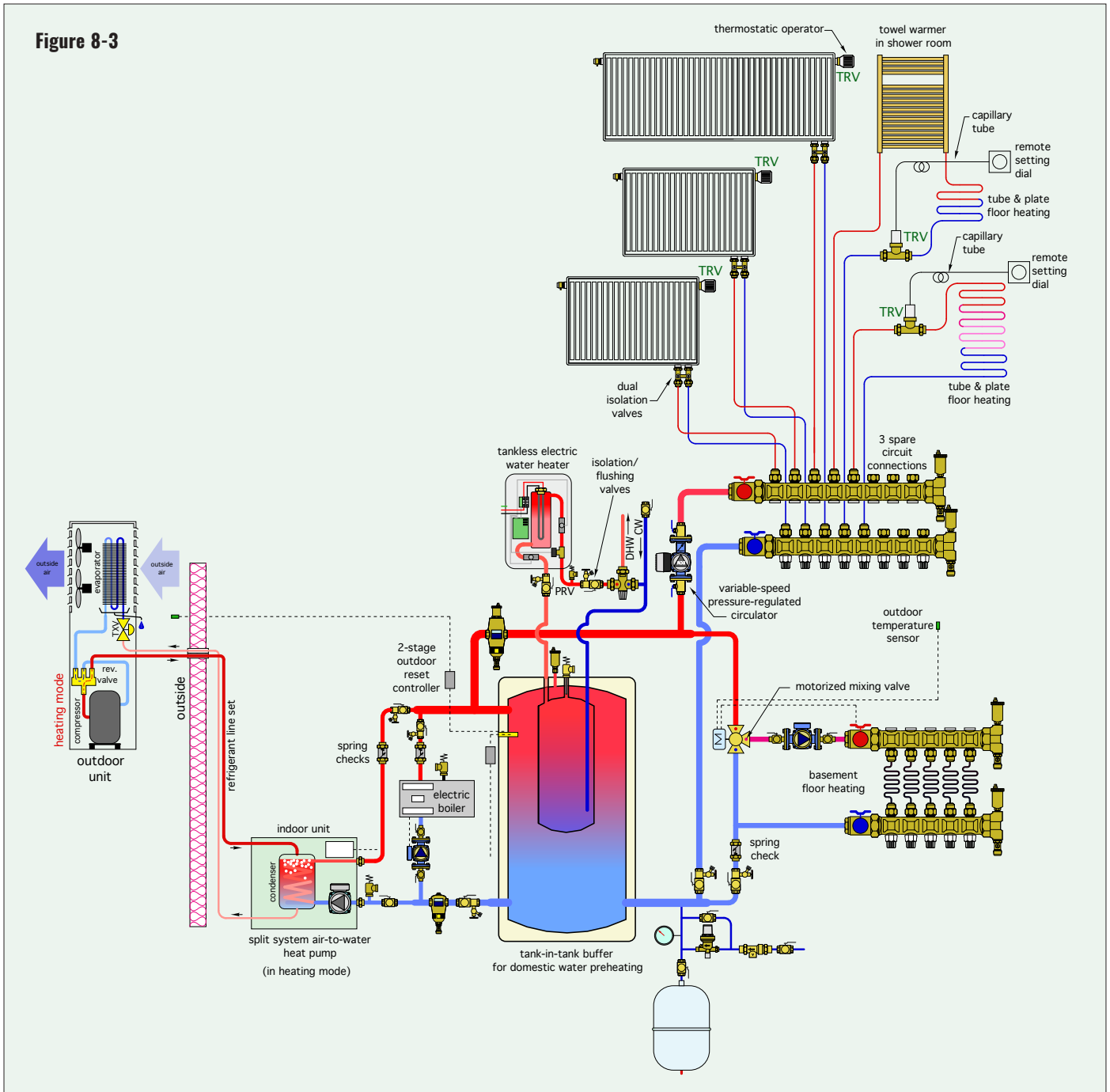
The buffer tank has a 3-pipe configuration. Heated water from the heat pump or electric boiler can flow directly to one or more of the space heating zones when either heat source is on at the same time as one or more of the zones. Any difference in flow rates between the heat source(s) and zones passes through the buffer tank. All return flow passes into the lower portion of the buffer tank, and thus keeps its thermal mass well engaged.

A spring-loaded check valve is installed in the piping leaving the heat pump and the electric boiler. These valves prevent reverse thermosiphon flow from the heated tank through either heat source when they are off.

The water temperature in the tank is controlled by a 2-stage outdoor reset controller. At design load conditions, the target water temperature in the tank is 120°F. The target temperature decreases as the outdoor temperature increases. The minimum target temperature is 100°F. Maintaining the tank temperature in this range allows the heat pump to operate with relatively good COPs. It also allows the heat pump to provide the majority of the "temperature lift" required for domestic hot water. If the heat pump is not able to maintain the necessary target temperature, the controller operates the electric boiler for supplemental heat input. There is a 5-minute interstage time delay to allow time for the heat pump to stabilize its operation before turning on the electric boiler.

Space heating is supplied through several types of heat emitters. The home's main floor uses a combination of panel radiators, a towel warmer, and two areas of tube & plate underfloor heating. All of these emitters have been sized for design load output at 120°F, and thus they can all be supplied as parallel circuits. The towel warmer is combined in series with a short tubing circuit that provides a small area of floor heating in the master bathroom. The three panel radiators each have integral thermostatic

Figure 8-3



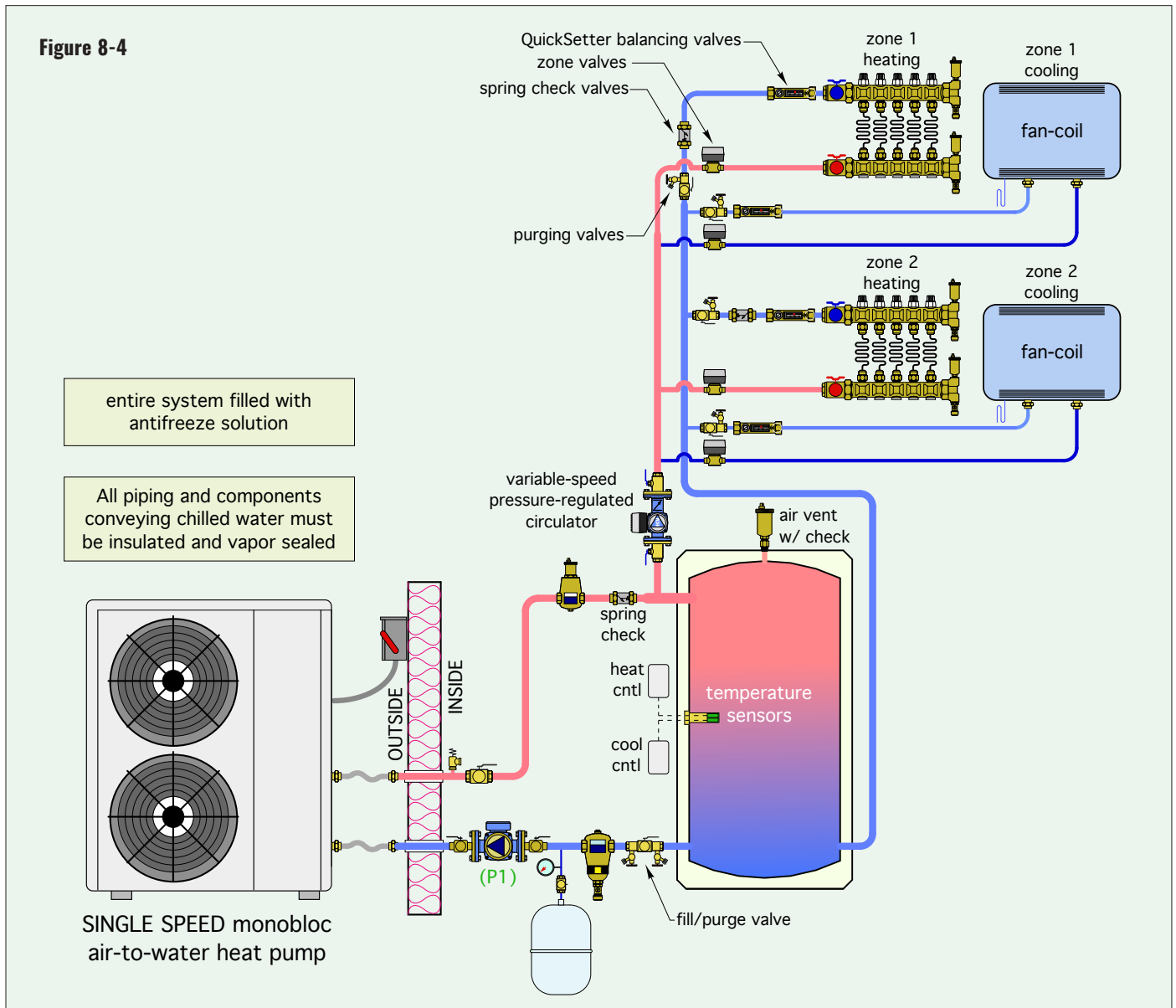
radiator valves and operate independently. The master bathroom and one other area of floor heating are equipped with non-electric modulating valves that are coupled to remote setting dials by capillary tubes.

Flow to all the main floor circuits is provided by a variable-speed pressure-regulated circulator operating in constant differential pressure mode. All circuits begin and end at a common manifold station that is equipped with three

“extra” connections. Those connections are initially capped but allow additional panel radiators or other emitters to be easily added in the future.

The home’s basement slab is also heated. The slab circuits require lower water temperature compared to the main floor circuits (100°F water at design load). This water temperature is provided by a 3-way motorized mixing valve, also operated using outdoor reset control.

**Figure 8-4**



### SYSTEM #4

The system shown in Figure 8-4 provides heating and cooling to two zones. The system can only operate in one mode (e.g., heating or cooling) at a time. The buffer tank and heat pump are shown in heating mode.

The heating and cooling source is a single speed monobloc air-to-water heat pump. The buffer tank is piped in a 3-pipe configuration and is used in both heating and cooling modes. The entire system operates with a solution of propylene glycol antifreeze.

In heating mode, a variable-speed pressure-regulated circulator supplies two independent zones of radiant panel

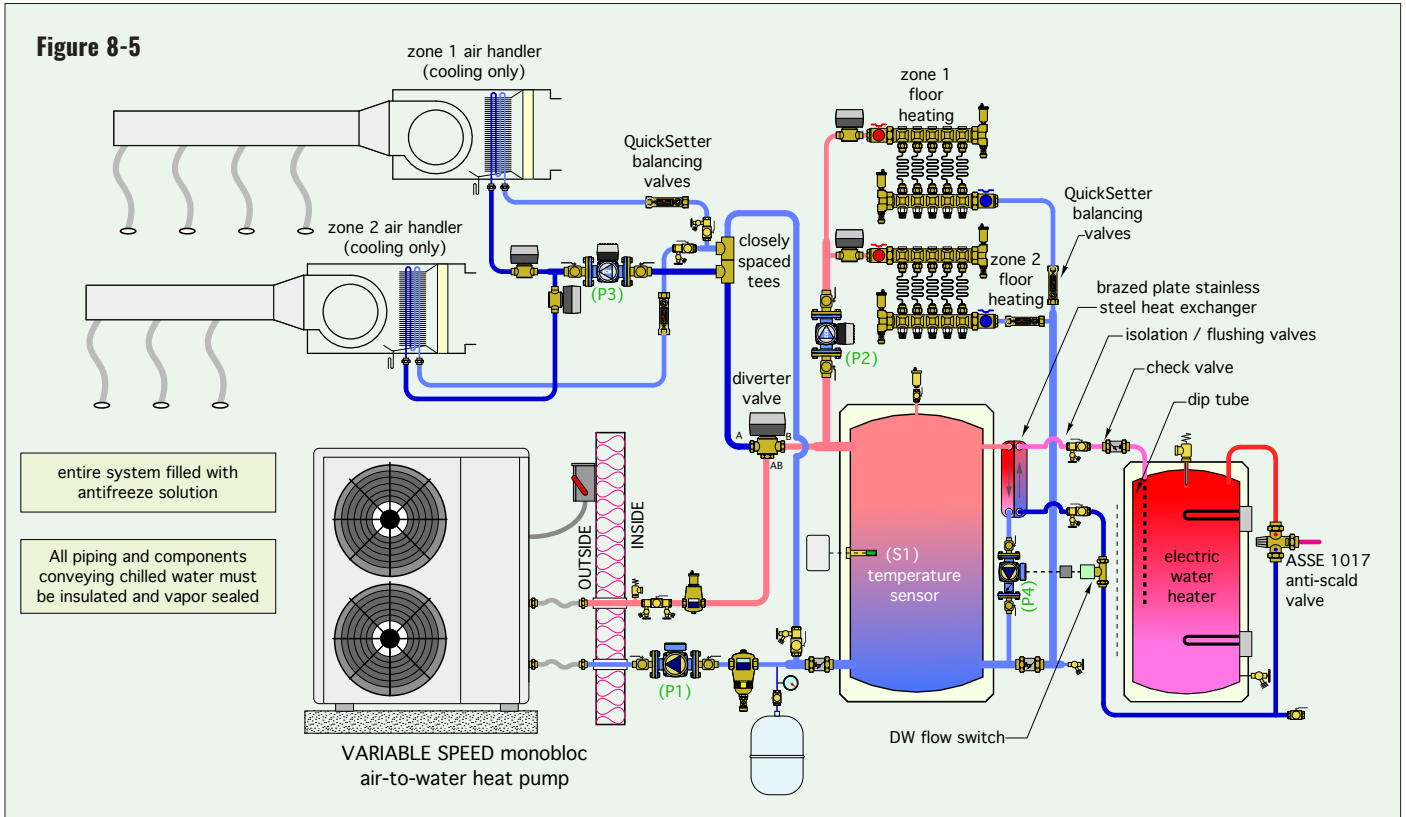
heating. Flow through each manifold station is controlled by a zone valve. Each manifold station also has a flow balancing valve. The zone valves to the fan-coils remain closed during heating mode.

In cooling mode, the same circulator supplies two independently controlled fan-coils. Each fan-coil circuit has a flow balancing valve. The zone valves to the heating manifold stations remain closed during cooling mode.

During heating mode, the heat pump and its associated circulator are turned on and off based on an outdoor reset controller that monitors the temperature at the mid-point of the buffer tank. This control action is independent of any call for heat from the room thermostats. The warmer the outdoor



Figure 8-5



temperature, the lower the target temperature calculated by this controller. The maximum water temperature during heating mode is 110°F. The minimum temperature is 80°F. The reset controller operates on a 10°F differential centered on the target temperature. These temperatures allow the heat pump to operate at a relatively high COP.

During cooling mode, the heat pump and its associated circulator are turned on and off based on a temperature setpoint controller that monitors another sensor at the mid-point of the tank. The heat pump is turned on when the tank temperature is 60°F or higher. The heat pump is turned off when the tank temperature drops to 45°F. The heat pump operates independently of calls for cooling from the room thermostats.

Spring check valves are used to prevent reverse thermosiphoning from the tank, and to limit chilled water migration into the heating portions of the distribution system. A magnetic dirt separator is used to protect the heat pump and high-efficiency circulator for dirt and iron oxides.

### SYSTEM #5

The system shown in Figure 8-5 provides two independently controlled zones of floor heating, as well as two associated zones of chilled water cooling. It also provides domestic hot water.

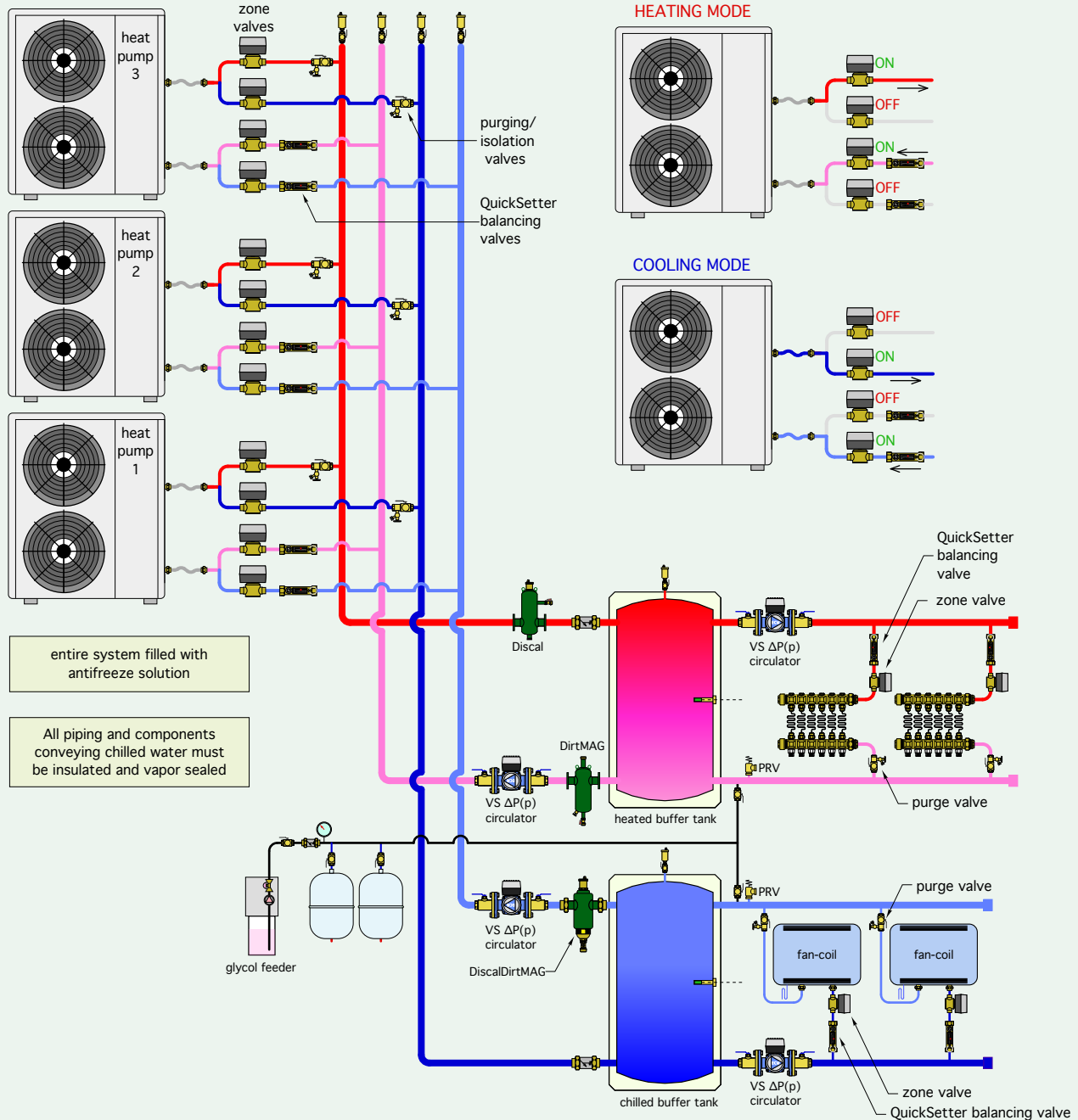
This system is supplied by a *variable-speed* monobloc air-to-water heat pump. Its *variable speed capability* allows the cooling portion of the system to operate without a buffer tank. In cooling mode, the speed of the heat pump's compressor automatically adjusts as necessary to maintain a chilled water temperature between 45° and 60°F when either (or both) of the chilled water air handlers is operating. *Note: Zoned cooling distribution systems should not be used with a fixed-speed heat pump unless a buffer tank is provided.*

A motorized diverter valve is used to route the heat pump's output to the balance of the system associated with each mode of operation.

It is possible for this system to deliver limited amounts of simultaneous heating and cooling. The buffer tank could deliver some amount of heating to the floor circuits as well as preheat domestic water while the heat pump is operating in cooling mode.

One possible control scenario would be to set up controls to prioritize cooling operation during the cooling season and heating mode operation during the heating season. The heat pump would switch to the lower priority load only after temporarily satisfying the higher priority load.

**Figure 8-6**



Another control possibility would be to set the target temperature of the buffer tank using outdoor reset control during the heating season, and switch that target temperature to a higher, narrower range during warm weather when the heat pump's COP is relatively high. This would allow a higher percentage of the energy needed for domestic hot water to be supplied by the heat pump rather than the resistance heating elements in the supplemental water heater.

### SYSTEM #6

It is possible to use multiple air-to-water heat pumps in systems that require more heating or cooling capacity than can be supplied by a single heat pump. The use of multiple heat pumps also adds "redundancy" to the system in the event that one of the heat pumps is down for service.

The system in Figure 8-6 shows three monobloc air-to-water heat pumps piped so that each heat pump can independently operate in heating or cooling mode.

This system has a “hot” buffer tank and a “cold” buffer tank. The system controls monitor the temperature of each tank and call for a heat pump to turn on when the temperature of either tank deviates slightly from its target temperature or outside of a set temperature range. The water temperature in the “hot” buffer tank would be regulated based on outdoor reset control. The water temperature for the “cold” buffer tank would be maintained between 45° and 60°F whenever a cooling load is present.

When a heat pump is called to operate, the zone valve pairs associated with its mode of operation open. The status of the heat pump’s reversing valve is also set. A variable-speed pressure-regulated circulator operates to create flow through the appropriate buffer tank. The speed of the circulator is based on *proportional* differential pressure control. The speed automatically increases or decreases depending on how many heat pumps are operating.

The valving at each heat pump is also arranged so that the zone valves or heat pump can be isolated from the balance of the system if necessary for service.

Each buffer tank provides hydraulic separation between the heat pump circulators and the load circulators.

The heating zones are supplied by low-temperature radiant panels. Flow to each manifold station is controlled by a zone valve. Each manifold station piping assembly is also equipped with a balancing valve and purging valve. These valves are arranged so that each manifold station and its associated zone valve can be completely isolated from the balance of the system if necessary for service.

The cooling zones are supplied by fan-coils. Each fan-coil piping assembly is equipped with a balancing valve and purging valve. These valves are arranged so that each fan-coil and its associated zone valve could be completely isolated from the balance of the system if necessary for service.

Because this system can simultaneously supply heating and cooling, one mode of operation must take priority when staging the heat pumps. Several possibilities exist. For example, during heating season, the ability to maintain adequate water temperature in the “hot” buffer would be a likely priority. Once that temperature is established, at least one of the heat pumps would be allowed to operate in cooling mode, if a cooling load is present. During the cooling season, it’s likely that all heat pumps would be prioritized to satisfy the cooling load.

Although not shown in Figure 8-6, it would be possible to operate a water-to-water heat pump between the two buffer tanks when a simultaneous demand for heating and cooling exists. The air-to-water heat pumps would supplement the rate of heating or cooling needed by the system.

The piping and control concepts discussed for this system could be extended to additional numbers of heat pumps if needed. The “hot” buffer could be used to supply a mix of heat emitters, such as radiant panels and panel radiators. Likewise, the “cold” buffer could supply a mix of terminal units such as fan-coils, air handlers and radiant cooling panels. The latter requires special piping and controls to ensure that the panel always remains above the dewpoint of the interior air.



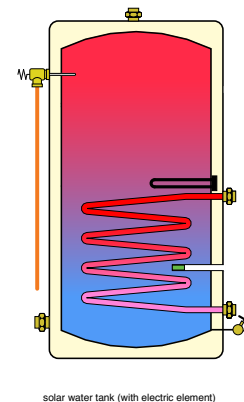
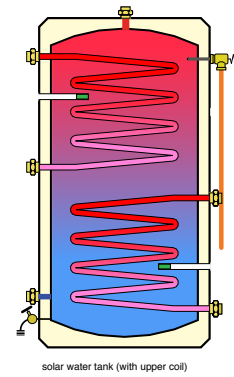
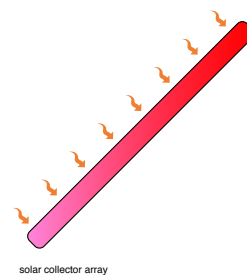
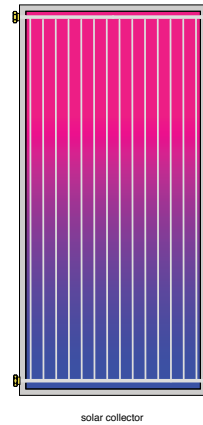
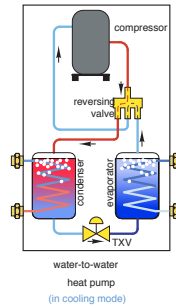
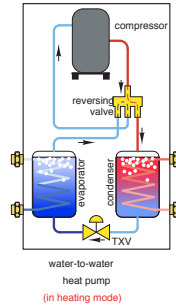
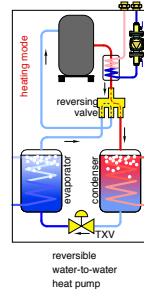
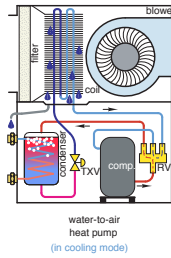
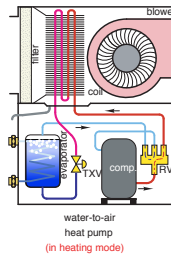
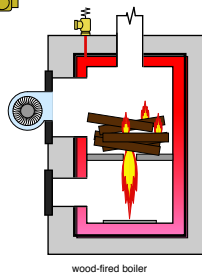
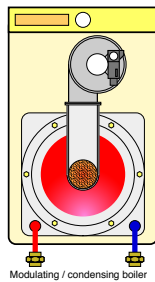
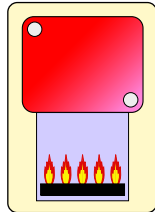
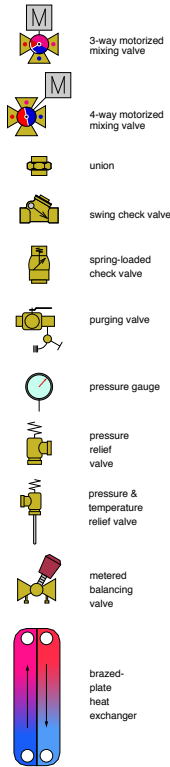
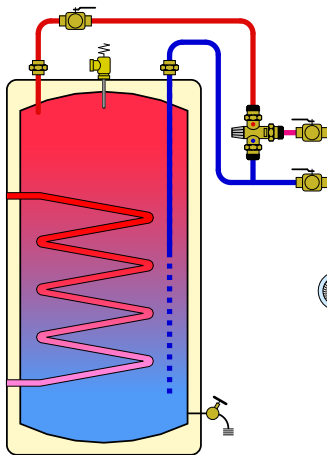
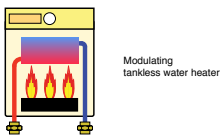
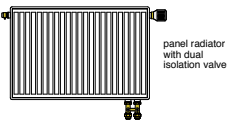
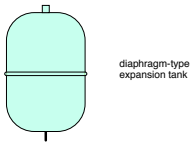
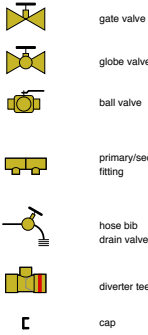
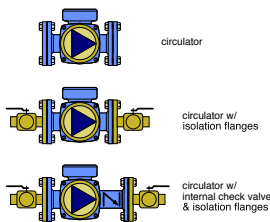
**More information on radiant cooling is available in *idronics* #13.**

## SUMMARY

21st century refrigeration techniques now make it possible for air-source heat pumps to operate in cold winter climates, with significantly improved performance compared to earlier-generation heat pumps. By combining low-ambient air-source heat pump technology with the versatility and high distribution efficiency of modern hydronics, designers can create systems for unsurpassed heating and cooling comfort, as well as domestic hot water production. Those systems are ideally suited for use with renewably sourced electricity, carbon reduction goals and “net zero” building projects. When properly selected and applied, low-ambient air-to-water heat pumps can approach the performance of geothermal heat pump systems at significantly lower installation cost and complexity. Air-to-water heat pump systems offer hydronic heating professionals a means of responding to evolving market trends and constraints without compromising quality, comfort or efficiency.

# APPENDIX A: COMPONENT SYMBOL LEGEND

## GENERIC COMPONENTS





# DIRTMAG®

## Dirt separator with magnet

### 5463, NA5453, 5465M, NA5465M series



#### Function

In heating and air conditioning control systems, the circulation of water containing impurities may result in rapid wear and damage to components such as pumps and control valves. It also causes blockages in heat exchangers, heating elements and pipes, resulting in lower thermal efficiency within the system.

The dirt separator removes these dirt particles, collecting them in a large collection chamber from which they can be flushed even while the system is in operation. This device is capable of efficiently removing even the smallest particles, with very low head loss.

The DIRTMAG® magnetic dirt separator removes both ferrous and non-ferrous impurities continuously, featuring powerful removable magnets that remove up to 100% of the ferrous impurities, including magnetite, that can form in a hydronic system. The DIRTMAG has 2 ½ times the removal performance of a standard dirt separator.

Insulation shells are available separately for brass models.

#### Product range

- 5463 series DIRTMAG dirt separator with magnet for horizontal pipes, in brass.....connections 1" to 2" NPT female, sweat & press
- 5465AM series DIRTMAG dirt separator with magnet for horizontal pipes, in steel.....connections 2" to 4" ANSI 150 flange
- NA5465AM series DIRTMAG dirt separator with magnet for horizontal pipes, in steel, ASME and CRN.....connections 2" to 14" ANSI 150 flange  
-CRN approval for 2" – 12" sizes; consult factory for 14" size.

#### Technical specifications

##### Brass body magnetic dirt separators

###### Materials

- body, dirt collection chamber and top plug: brass
- internal element: glass reinforced nylon PA66G30
- hydraulic seal: EPDM
- drain valve: brass
- magnet: neodymium rare-earth

###### Performance

- Suitable fluids: water, glycol solution
- Max. percentage of glycol: 50%
- Max. working pressure: 150 psi (10 bar)
- Temperature range: 32–250°F (0–120°C)
- Particle separation capacity: to 5 µm (0.2 mil)
- Ferrous impurities separation efficiency: up to 100% removal

SIZE	MAXIMUM FLOW RATE				
	¾"	1"	1¼"	1½"	2"
GPM	6	9	15	24	36
Cv	19	32	56	73	81

- Connections** - main: ¾", 1", 1¼", 1½" and 2" NPT female  
1", 1¼", 1½" and 2" integral sweat  
1", 1¼", 1½" and 2" integral press
- lay length (press connections): see page 2
  - top: ½" NPT female (with plug)
  - drain: ¾" garden hose connection

##### Steel body magnetic dirt separators

###### Materials

- body: epoxy resin painted steel
- top cap: brass
- hydraulic seal: non-asbestos fiber
- drain valve: brass
- internal element: stainless steel and HDPE
- magnet: neodymium rare-earth
- magnet probe drywell: brass
- (8" - 14" sizes use quantity 3 magnets)

###### Performance

- Suitable fluids: water, glycol solution
- Max. percentage of glycol: 50%
- Max. working pressure: 150 psi (10 bar)
- Temperature range (vessel): 32–270°F (0–132°C)
- Particle separation capacity: to 5 µm (0.2 mil)
- Ferrous impurities separation efficiency: up to 100% removal

###### Connections

- flanges: 2" - 14" ANSI B16.5 150 CLASS RF
- top: ¾" NPT male (with cap)
- thermo well tap (8" - 14" only):  
Inlet/outlet flanges: ½" NPT female
- drain valve with lever: sizes 2 - 6 inch: 1" NPT  
sizes 8 - 14 inch: 2" NPT

###### Agency approval

Series NA5465AM designed and built in accordance with Section VIII, Div. 1 of the ASME Boiler and Pressure Vessel Code and tagged and registered with the National Board of Boiler and Pressure Vessel Inspectors, stamped for 150 psi (10 bar) working pressure with ASME U-Stamp. 2 - 12 inch sizes are CRN Registered, consult Caleffi for 14 inch.

SIZE	MAXIMUM FLOW RATE					
	2"	2½"	3"	4"	5"	6"
GPM	89	150	227	355	816	904
Cv	88	176	211	328	520	842

SIZE	8"	10"	12"	14"
GPM	1,570	2,450	3,525	4,800
Cv	1,055	1,400	1,755	2,075

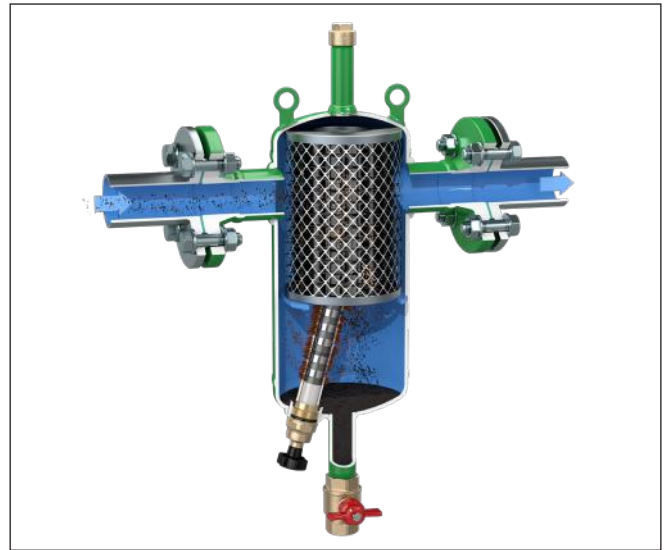
## Operating principle DIRTMAG

Non-ferrous and ferrous impurities, including magnetite, in hydronic systems can deposit onto heat exchanger surfaces and accumulate in pump cavities causing reduced thermal efficiency and premature wear. The small and often microscopic magnetic particles, called magnetite, form when iron or steel corrodes. Highly abrasive, the extremely fine particles are difficult to remove by traditional means. DIRTMAG separators offer highly efficient separation of typical dirt as well as magnetite. The versatile DIRTMAG magnetic dirt separator removes all impurities, including ferrous, continuously. In addition to removing sand and other impurities with an internal element in a low-velocity-zone chamber, the DIRTMAG features a powerful removable magnet below the flow line for fast and effective capture of ferrous impurities. The magnet removes up to 100% of the magnetic debris that can form in a hydronic system.

For the brass DIRTMAG, the ferrous impurities are captured by a strong neodymium rare-earth magnetic field created by a powerful removable magnet around the body below the flow line.



In the size 8-14 inch steel DIRTMAG, the ferrous impurities are captured by a concentrated magnetic field created by a stack of neodymium rare-earth magnets positioned inside three brass dry-wells installed from the underside of the separator and below the flow stream. Non-magnetic dirt particles are separated by colliding with an internal element in the flow stream and settling to the bottom. The deep collection chamber keeps the dirt from re-entering the flow stream. All collected debris are flushed out while the system is operating, by removing the magnets and opening the purge valve.

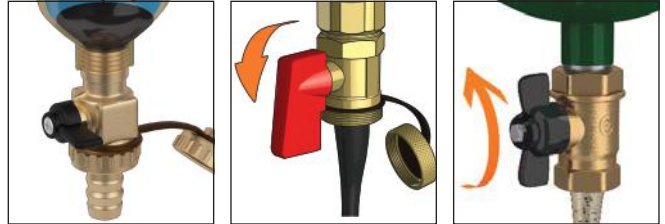


For the size 2-6 inch steel DIRTMAG, the ferrous impurities are captured by a concentrated magnetic field created by a stack of neodymium rare-earth magnets positioned inside one brass dry-well below the flow stream.

## Draining off debris

The dirt separator collection chamber has a drain valve. Using the handle provided it is possible to drain off the accumulated dirt particles even with the system in operation.

For the brass DIRTMAG, captured debris are easily flushed by unclamping the magnetic collar and purging.

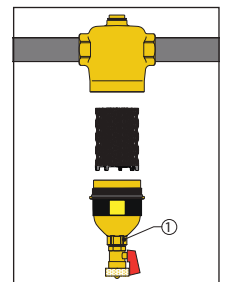


To purge the ferrous impurities in the steel DIRTMAG, the flexible magnetic stack is removed from the brass dry-well and the drain valve is opened. Aided by the system pressure, the captured debris, including magnetite, flushes out quickly and effectively.



## Maintenance

To perform maintenance, simply use a 26 mm hexagon wrench (1) to unscrew the dirt collection chamber, of the brass DIRTMAG, to which the inner mesh element is connected for removal and cleaning.





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