Air Source Heat Pump Design Guide

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Foreword

The aim of this Technical Guide is to serve as a reference work that is useful to both practicing engineers and during different training courses. The Technical Guide deals with technical issues related to design and application of air source heat pump heating and cooling systems but does not claim to be comprehensive.

It is entirely possible to build poorly functioning systems even with the best products available on the market. Consequently, it is extremely important to have knowledge of what is required to build well-functioning systems with maximum possible comfort and the lowest possible energy consumption.

A client rarely has the necessary skills to assess and predict the consequences of a specific decision. It is therefore the responsibility of the consultant to present the technical and financial possibilities of decisions made during a project.

Appendices at the end of this guide include definitions, designations, conversion factors, formulas, glycol properties, and piping details.

For more information on heating and cooling plants please refer to other Swegon publications (www.Swegon.com) or the American Society of Heating Refrigerating and Air-Conditioning Engineers (www.ASHRAE.org).

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1 Air Source Heat Pump Overview

Figure 1.1 | *Swegon Air Source Heat pump (ASHP)*

Introduction

This guide focuses on using air source heat pumps (ASHP) to provide most or all of the heating for the building. Essentially, the boiler that would be typically used is replaced by an ASHP. ASHPs behave quite differently from boilers so it is important for a successful project to understand how ASHPs behave and operate in the hydronic system. This chapter will provide a refresher on heat pumps and explain how air source heat pump performance is influenced by ambient temperature.

Heat Pump Refrigeration Cycle

Figure 1.2 shows a basic refrigeration circuit of a reversible heat pump in both cooling (left) and heating (right) mode. In cooling mode, heat collected cooling the chilled water from 12 to 7 °C (54 to 44 °F)[1] in the Refrigerant-to-Water Brazed Plate Heat Exchanger (BPHX) boils the refrigerant. The gaseous refrigerant is then compressed raising its temperature close to 90 °C (194 °F). The refrigerant is directed by the reversing valve to the Refrigerant-to-Air Heat Exchanger Coil where it cools and condenses back into a liquid to repeat the cycle. The heat that leaves the refrigerant includes both the heat collected from the chilled water plus the heat from the work of compression (about 25% of the chilled water heat). The heat is added to the ambient air used to cool the condenser, raising its temperature from around 35 °C (95 °F) to 50 °C (122 °F) as it is drawn through the Ref-to-Air coil by the fans.

In heat pump mode, the heat flow is reversed. The reversing valve redirects the refrigerant flow to move heat from the outside ambient air into the water loop. Return hot water is heated from 43 to 49 °C (110 to 120 °F) in the Ref-to-Water BPHX. This cools and condenses the hot refrigerant to a liquid. The refrigerant flows through the heating expansion valve where the pressure and temperature drops to around -14 °C (6.8 °F). Heat from -8.3 °C (17 °F) ambient air boils the refrigerant in the Ref-to-Air HX coil.

ASHPs need a signal to know which mode to operate in. Typically, a digital signal or Digital Input (DI) to the unit controller will instruct the unit to operate in either heating or cooling mode and controls the reversing valve position. It is common that the reversing valve will fail in the heating position to be able to continue to provide heating if required. As the pressure in the unit must be reversed upon a mode change between heating and cooling mode,

[1] The temperatures shown here are typical to get a sense of the operating conditions. The actual temperatures will depend on the product and the application.

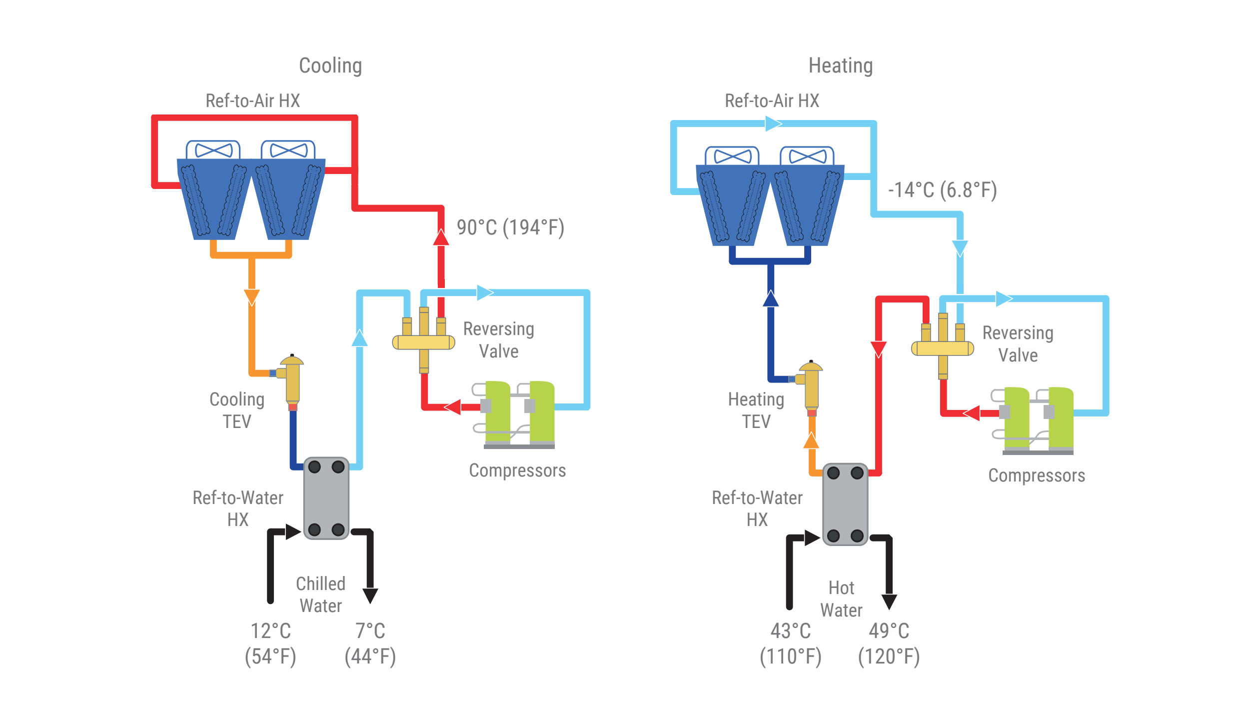

Figure 1.2 | *Heat Pump Refrigeration Circuit*

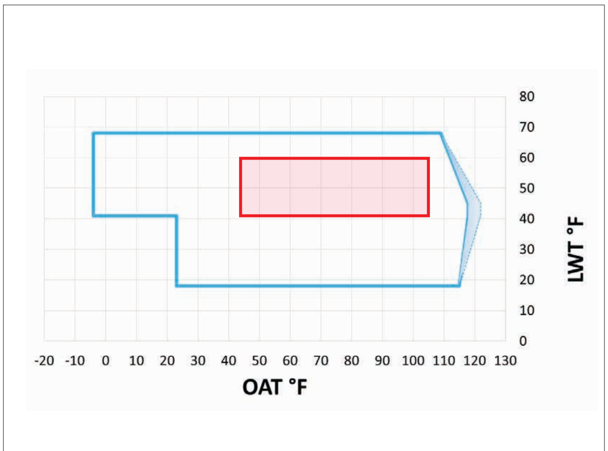

Figure 1.3 | *Chiller Operating Envelope*

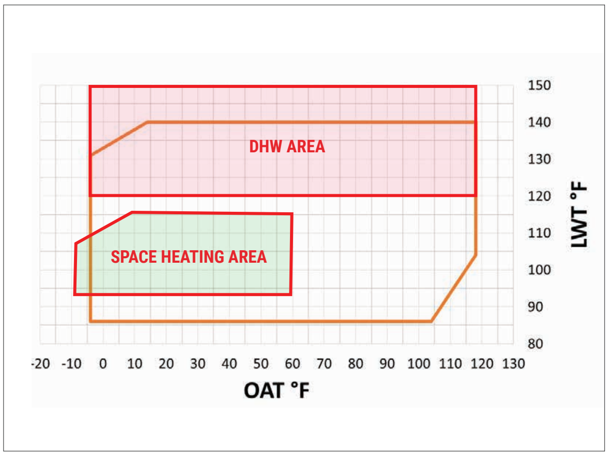

Figure 1.4 | *ASHP Operating Envelope*

it should not be expected that a reversible ASHP will cycle back and forth from heating and cooling repeatedly.

Operating Envelopes

Figure 1.3 shows the operating envelope for a typical air cooled chiller. The edge of the envelope indicates the chilled water supply temperature that is possible based on the ambient air temperature. Operation outside the envelope is not possible; the red area shows typical operating conditions for most chilled water applications. Chillers usually operate in the middle of their operating envelopes providing reliable operation throughout their lifespan.

Figure 1.4 shows the operating envelope for a typical ASHP in heating mode. Using a heat pump for heating or domestic hot water (DHW) is by nature, a more demanding application. As there is less room for error, more care should be taken regarding the application and equipment selection to ensure reliable performance and equipment longevity.

Heating Capacity vs. Ambient Temperature

Figure 1.5 displays one of the most critical points when using ASHPs. ***The capacity of the ASHP decreases as the ambient air temperature decreases.*** While equipment design, compressor type, and refrigerant selection affect the rate of capacity loss, all heat pumps lose capacity as ambient temperature drops. The sizing of the heat pump capacity relative to the system must be carefully thought through and will be further discussed in subsequent chapters.

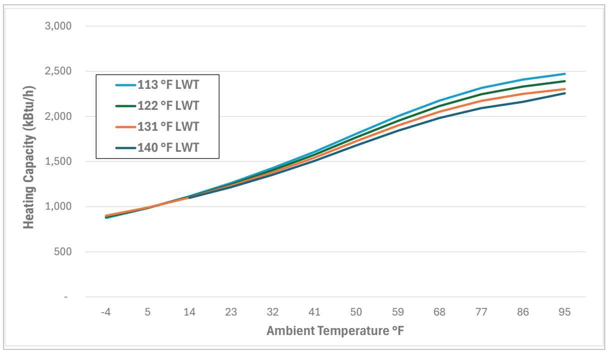

Figure 1.5 | *Heating Capacity vs. Ambient Temperature for ASHP*

Figure 1.6 shows the impact to an ASHP's coefficient of performance[2] (COP) as the ambient temperature decreases. In the case of the 60 °C (140 °F) water curve, operation below -10 °C (14 °F) is not even possible as the unit in this example has reached the operating envelope limit.

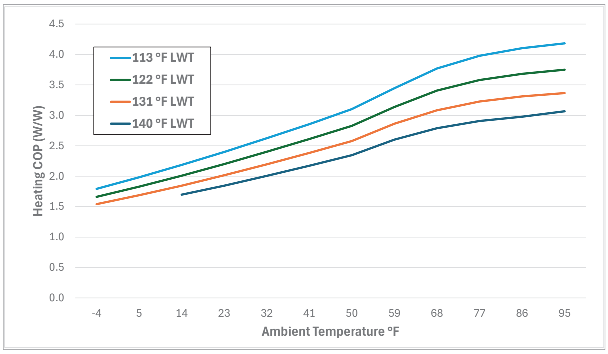

Figure 1.6 | *Heating COP vs. Ambient Temperature for ASHP*

Figure 1.6 also illustrates that the higher the lift (the difference between the ambient air temperature and the water temperature) the lower the COP. Most ASHP units operating at their lower limit have a COP around 2. This is still twice as good as an electric boiler with a COP of 1.

Reviewing Figures 1.4, 1.5 and 1.6 shows that is possible to make a statement "this ASHP can operate at -20 °C (-4 °F)". It is also possible to state, "this unit can produce 60 °C (140 °F) supply water". It is rare, that both statements can be true at the same time. Further, designing this close to the edge of the operating envelope for the operating life of the unit is not advised where reliability is critical. By reducing the operating supply temperature, the reliability and longevity of the equipment can be improved, while also improving efficiency and energy savings.

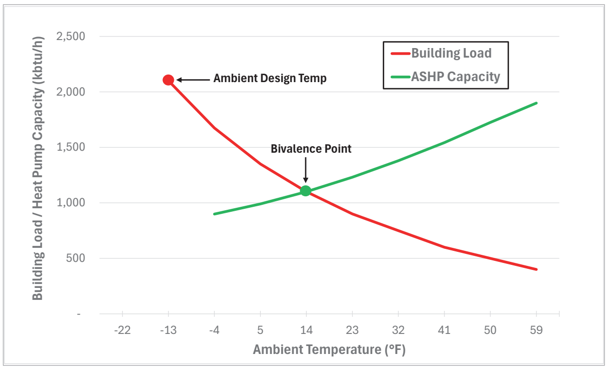

Figure 1.7 | *Building Load vs. ASHP Capacity*

The best advice is to get specific selections for the units being considered and mark where the design operating point will be on the unit's operating envelope similarly as would be done for a fan operating point on a fan curve.

The green line in Figure 1.7 shows the ASHP capacity as the ambient temperature decreases for 55 °C (131 °F) supply water temperature. A rule of thumb[3] is there will be a 2% capacity loss in capacity for every 1 °C drop in ambient temperature (1% loss for every 1 °F).

[2] COP (Coefficient of Performance) is the heating or cooling effect delivered divided by the power required to operate the unit. A unit that delivers 3 kW of heat and requires 1 kW of electricity to operate has a COP of 3.

[3] Rules of Thumb are simply estimates. The best action is to get real selections at the expected operating conditions and perform building load calculations at various ambient temperatures.

The red curve shows the building heating load profile as the ambient temperature drops. The point where the two lines cross is known as the bivalence point. *The bivalence point is the lowest ambient temperature where the heat pump can meet the building load.* To the left of the bivalence point, supplemental heat will be required. Designers must recognize that manufacturer catalog data for capacity and efficiency are typically rated at AHRI 550/590 ***Standard Rating Conditions***, which is based on an ambient temperature of 8.3 °C (47 °F) and sometimes is also published at -8.3 °C (17 °F). Therefore, when designing an air-source heat pump system it is important to understand the equipment performance at various ***Application Rating Conditions***.

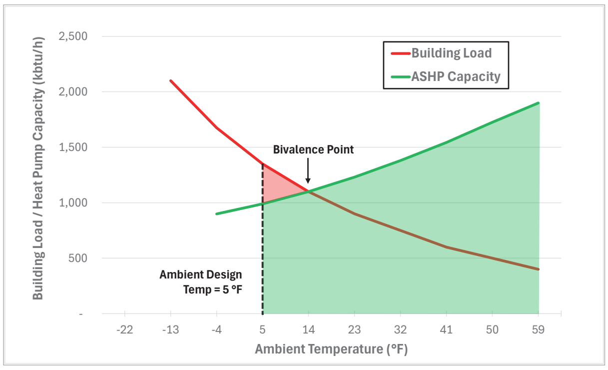

Figure 1.8 | *ASHP w/ Supplemental Heat*

Figure 1.8 shows an ASHP heating plant with supplemental heat to make up the difference between what the ASHP can deliver and the building load. The Supplemental heat source could be an electric or gas boiler. The minimum boiler size is the difference between the ASHP capacity at design conditions and the design heating load.

One option that may be possible in medium winter climates is to oversize the ASHPs. Figure 1.9 shows the bivalence point moving to the left as the ASHP capacity is increased. Where the winter design condition is close to the bivalence point, this solution may be cost effective. However, there are a few things to consider. First, it implies the system design is moving closer to the operating limits of the ASHP (depending on water temperature selection) and reliability should be considered.

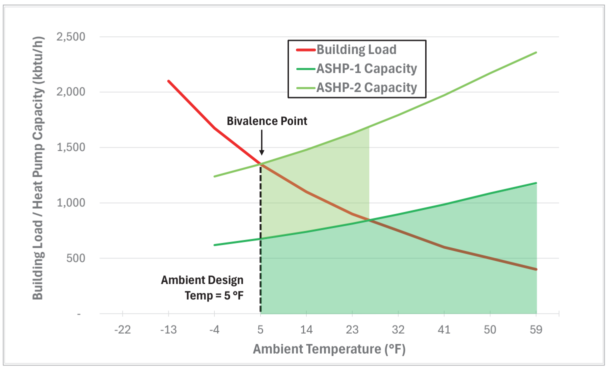

Figure 1.9 | *Oversizing ASHPs to Reduce Auxiliary Heat*

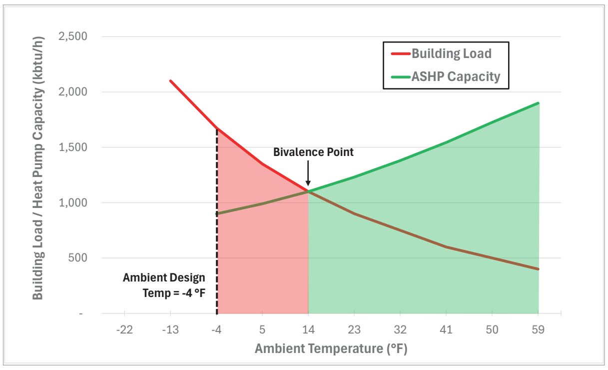

Figure 1.10 | *Using Auxiliary Heat Below the Bivalence Point*

Another consideration is as the ambient air temperature increases (moving to the right in Figure 1.9), the capacity of the ASHP also increases while the building load is decreasing. A situation where the ASHP becomes "oversized" for the reduced load may occur such that the ASHP will be operating at minimum flow and minimum capacity. This could result in compressor short cycling.

Additionally, the ASHP will be oversized (possibly significantly) for the cooling load. To manage the oversizing issue, use multiple ASHPs. In Figure 1.9, two 245 kW (70 Ton) ASHPs can meet the winter load. As the ambient temperature increases in heating or the units are operating in cooling, one of the units can be cycled off to better match unit capacity to the building load.

In colder climates, especially when the winter design ambient temperature is below the operating envelope of the ASHP, it will be necessary to shut down the ASHPs and switch to auxiliary heat sized for the full heating load. Figure 1.10 shows a boiler sized for the entire heating load with a changeover point matching the bivalence point.

If an electric boiler is used, it becomes important to try and use the ASHPs as much as possible because of their efficiency. This would mean the ASHPs would operate below the bivalence point supplemented by the electric boiler. Once the ambient temperature drops to a temperature close to the ASHP operating envelope, the ASHPs are shut down and the electric boiler is used to meet the entire heating load. This approach requires the electric service to be sized to operate the ASHPs and the boiler. Depending on the possible energy savings, it may not be worth the capital cost.

It is important to check the pump sizing whenever the design approach requires operating a full sized boiler plant concurrently with ASHPs. It is possible that the minimum flow of the ASHPs and boiler plant may exceed the system design flow. This will be discussed in later chapters.

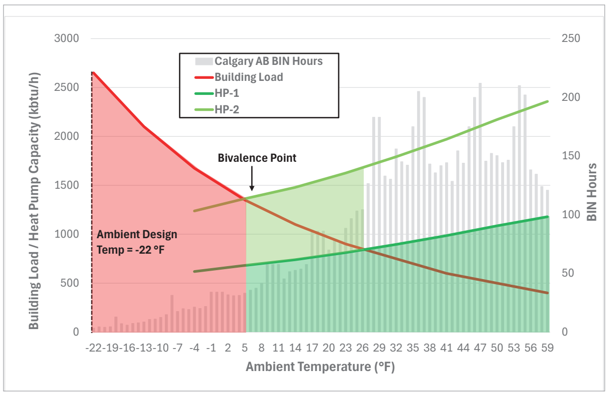

Figure 1.11 | *Operating Hours Below the Bivalence Point*

From an energy point of view, using the ASHPs as much as possible will deliver the best Energy Use Intensity (EUI). Even at the lower operating range of the ASHP, the COP is still close to 2 while a gas boiler is 0.9 and an electric boiler is 1.

From a carbon point of view, switching to a gas boiler will increase the carbon footprint but the impact may not be significant. Figure 1.11 shows that for Calgary, Canada, there are only 462 hours below the bivalence point.

From an operating cost point of view, when to switch from ASHPs to gas boilers will depend on the cost of natural gas and electricity. Where electricity costs are high, it may make sense to switch to the gas boiler for cost savings even though the ASHP can meet the heating load.

Adding a boiler in addition to ASHPs to meet the heating load might, at first, appear as requiring two heating systems. However, this is a hydronic heating system sharing the same pumps, piping etc. The choice is whether to purchase more ASHPs or a boiler and generally ASHPs are more expensive. The gas boiler will require a gas service but offers the owner resiliency in heating energy sources with a small impact to the carbon footprint. For a full electric solution, an electric boiler can be used. Here the capital cost consideration will be the electrical service to support the boiler.

Air Source Heat Pump Overview Summary

- Heating with an ASHP is a more demanding application than cooling and care should be taken to not design the system to consistently operate at the edges of the operating envelope.
- The ASHP capacity decreases as the ambient temperature decreases and the building heating requirement increases.
- Where the ASHP capacity meets the heating load requirement is known as the bivalence point
- For mild winter conditions, oversizing the ASHPs to meet the heating design load is an option but consider using multiple ASHPs to better match heating and cooling capacity output relative to the building load.
- Where winter design conditions are close or beyond the operating range of the ASHP, an auxiliary boiler plant sized for the design heating load will be required.

2 Defrost and Heat Pumps

Introduction

This chapter will introduce the need for defrost cycles in ASHPs and the impact of defrost cycles on the plant design.

Defrost

Even in winter with ambient temperatures well below freezing, there is some moisture in the air. Since the refrigerant-to-air coil is colder than the ambient air dewpoint during heat pump mode, the moisture will freeze directly on all cold surfaces in a process known as Deposition[1]. If the ice formation is not addressed, the coil will freeze to the point where the unit will not operate. In severe cases the frost can actually damage the coil.

Figure 2.1 | *Frost on Refrigerant-to-Air Coil*

All ASHP products have some form of defrost. This includes heat pump/chillers, variable refrigerant flow (VRF) systems, commercial refrigeration, etc. The most common method of defrost is ***hot gas defrost***.

Hot gas defrost is a means of melting the ice formed on the refrigerant-to-air by switching from heat pump to cooling mode (reversing the heat flow). Heat is extracted from the hot water loop and used to heat the coil and melt the ice. Once the ice is melted, the unit returns to heat pump mode. *During the time the unit is in defrost mode, it is cooling the hot water loop, not heating it.*

While a basic unit might operate defrost on a timed cycle, most commercial ASHPs will recognize ice formation by a change in the refrigerant pressures and temperatures. Ice insulates the coil, so it becomes harder for the refrigeration circuit to extract heat from the ambient air. This has the effect of changing the refrigerant temperature/pressure.

The unit will operate in defrost for 3 to 10 minutes to melt the ice and varies widely between different equipment manufacturers. Advanced products can have very sophisticated algorithms to recognize frost formation and when the frost has melted it can return to heat pump mode. Minimizing defrost duration is critical; the longer the unit remains in defrost mode beyond what is necessary, the more heat is extracted from the hot water loop. The best systems are dynamic and can defrost in as little as 4 minutes.

[1] Deposition is the process where water vapor converts to ice without first becoming a liquid. It is the opposite of Sublimation.

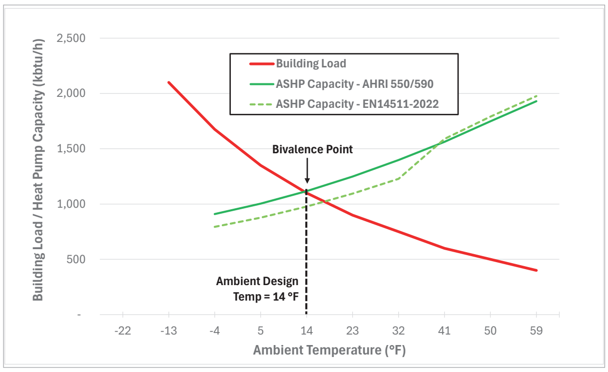

Figure 2.2 | *Defrost Adjusted Capacity*

Impact of Defrost on Unit Capacity

It is important to consider the impact of defrost on an ASHP's capacity. A unit that is rated to deliver 100 kW (34.1 kBtu/h) of heat will not actually deliver that much heat to the building when defrost is considered. A rule of thumb is the capacity will be reduced by 8 to 12% when defrost is taken into account. Hence a 100 kW (34.1 kBtu/h) unit will only deliver around 90 kW (30.7 kBtu/h) to the building.

The European heat pump rating standard EN 14511-2022 requires the impact of defrost to be recognized in the unit capacity. Essentially, the EN 14511 standard accounts for the effects of defrost and performance is measured across an entire cycle of heating with at least one defrost cycle. The AHRI standard, on the other hand, takes the performance measurement after a certain period of steady state operation, after a defrost cycle has already been performed (i.e. ideal coil conditions without frost).

Defrost Cycles

The question of how often a unit will need to defrost is dependent on several factors including ambient conditions (both temperature and humidity) and unit load. ASHPs move a significant amount of air across the coils to either reject or gather the heat. A typical 352 kW (100 ton) unit will move 42,500 l/s (90,000 cfm) of ambient air through the refrigerant-to-air coil. While there may not be much moisture in the air below freezing temperatures there is enough that defrost will be required around once an hour[2].

[2] Defrost could happen in as little as 30 minutes or take several hours. The one hour comment is given so that it is recognized that defrost is not a daily or weekly event. It is happening all the time when the ambient temperature is near or below freezing.

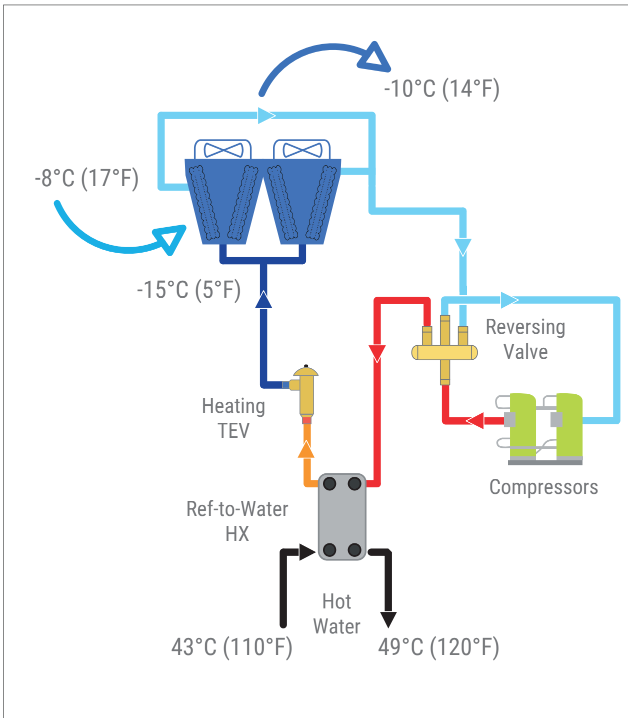

Figure 2.3 | *Condensation at -8°C (17°F) Ambient*

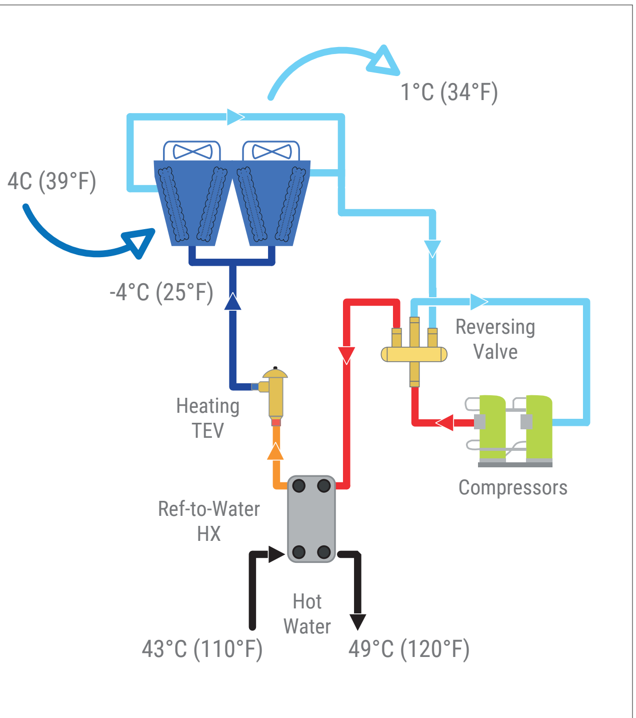

Figure 2.4 | *Condensation at 4°C (39°F) Ambient*

Figure 2.3 shows a 350 kW (100 ton) ASHP operating at -8 °C (17 °F) Ambient with 99% RH. This unit will condense 76 l/h (20 gallons/h) of water vapor which will freeze on the outdoor coil.

Figure 2.4 shows the same unit operating at 4 °C (39 °F) Ambient with 99% RH (i.e. it is raining). The unit will condense 159 l/h (42 gallons/h) of water vapor which will freeze on the outdoor coil. This occurs because the refrigerant-to-air coil temperature is below freezing.

Figures 2.3 and 2.4 show as the ambient temperature drops, the defrost cycles will reduce due to the lower moisture content in the ambient air. The worst case is near freezing (4 °C (39 °F)) conditions. Near freezing, the building will not be at design heating load, and the additional ASHP plant capacity can be used to minimize the defrosting impact on the hot water supply temperature to the building.

Condensate Management

The Figure 2.3 example above shows that 76 l/h (20 gallons/h) of liquid water will be formed during defrost when it is -8 °C (17° F) outdoors. This water will refreeze. In a very short order, there will be a significant amount of ice formed on, under or around the ASHP that can create an unsafe work area and possibly damage the unit and the building.

ASHPs that are designed to operate in cold climates such as northern Europe or North America should include many features for condensate management that should be called out for a successful project design. These include;

» Sloped drain pans (Figure 2.5)

» Heat traced drain pans and possibly coil frames (Figure 2.6)

» Heat Traced Condensate drainpipes (Figure 2.7)

» Field heat traced piping from the condensate drainpipes to a roof drain

Figure 2.5 | *Sloped Drain Pans*

Figure 2.6 | *Heat Traced Drain Pans and Coils*

Figure 2.7 | *Heat Traced Down Pipes*

Defrost and Heat Pumps Summary

- ASHPs need to defrost and generally use hot gas defrost as the method. Hot gas defrost operates the unit in cooling mode to melt the frost off the air coil.
- While the ASHP is in defrost mode, it is extracting heat from the building. This must be taken into account when sizing equipment.
- Managing the condensate during defrost is critical. The condensate should be collected and run through heat traced piping to a drain to avoid ice forming around the unit.

3 Importance of Thermal Mass

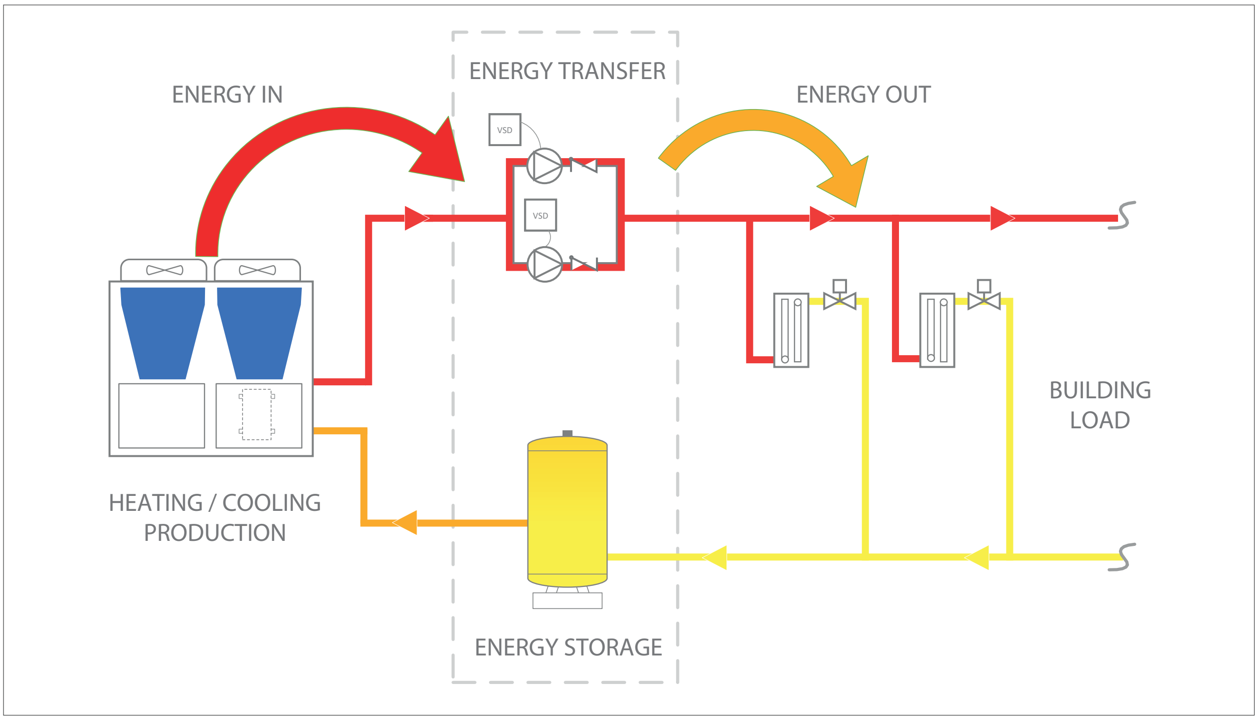

Figure 3.1 | *Building Energy Balance*

Introduction

This chapter will introduce the importance of understanding the system thermal mass. It is one of the most important concepts to get correct to achieve the desired plant performance.

The whole purpose of heating and cooling plants is to offset heat gain or loss in the building and thus maintain an energy balance. It is not realistic that the energy balance will be perfect in real-time. This is due to the response time of the equipment, its operating range etc. What enables EN_{in} to not match EN_{out} is the thermal mass of the hydronic system.

When there is insufficient thermal mass in the system, supply water temperature control can be problematic. This can lead to poor space temperature control and poor occupant satisfaction. In extreme cases it can be detrimental to the equipment. Short cycling and defrost control are two examples where insufficient thermal mass can be an issue for ASHPs.

In the case of chilled water systems, thermal mass is usually not an issue but still should be checked. There is a large amount of water volume in a typical chilled water loop providing a lot of thermal mass. Applications that require some scrutiny include a large cooling load (i.e. a large air handling unit) close coupled to a chiller or process loads. Chiller controls are typically set up for HVAC loads which do not change dramatically in a short time. A process load may not behave anything like a HVAC load and can lead to poor performance.

Heat Pumps, Defrost and Thermal Mass

Air source heat pumps are far more sensitive to the available thermal mass. The ASHP defrost cycle will greatly impact the thermal mass calculation as heat will be quickly removed from the loop to melt the frost.

Figure 3.2 shows an ASHP operating in heating mode. Over time frost forms on the refrigerant-to-air coil. At some point the unit will need to defrost. Figure 3.3 shows the unit in defrost mode.

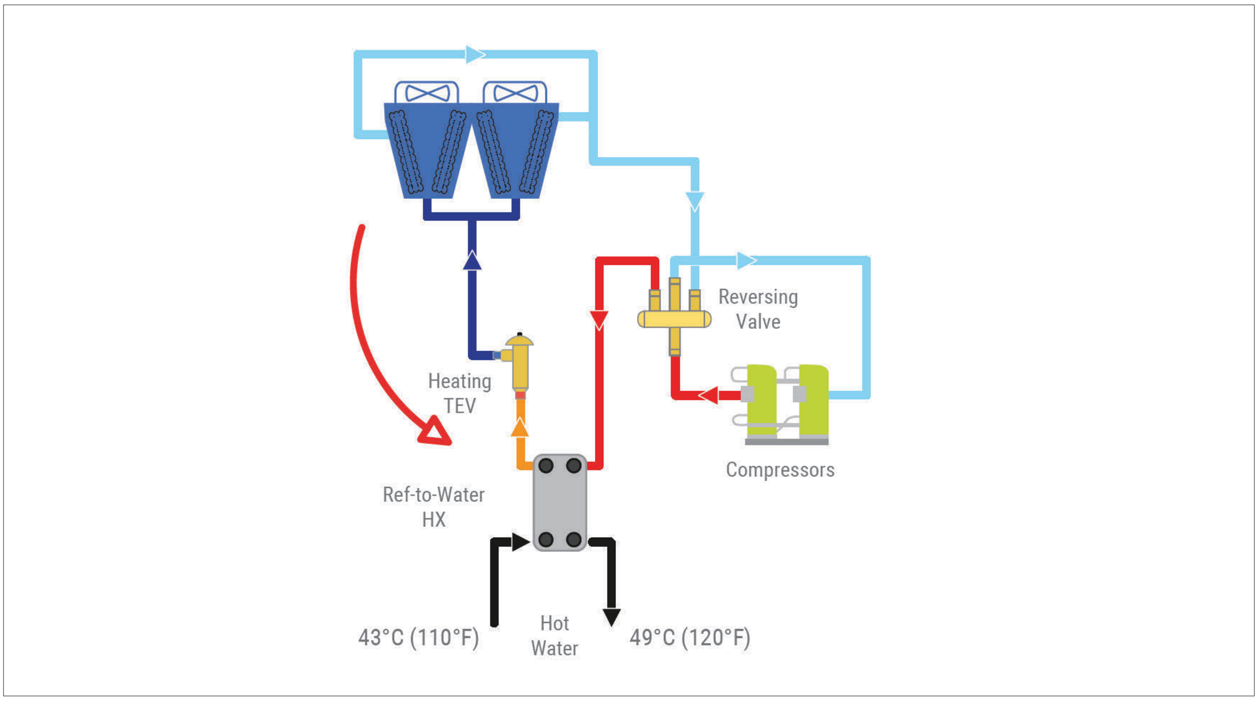

Figure 3.2 | *Single Circuit ASHP in Heating Mode*

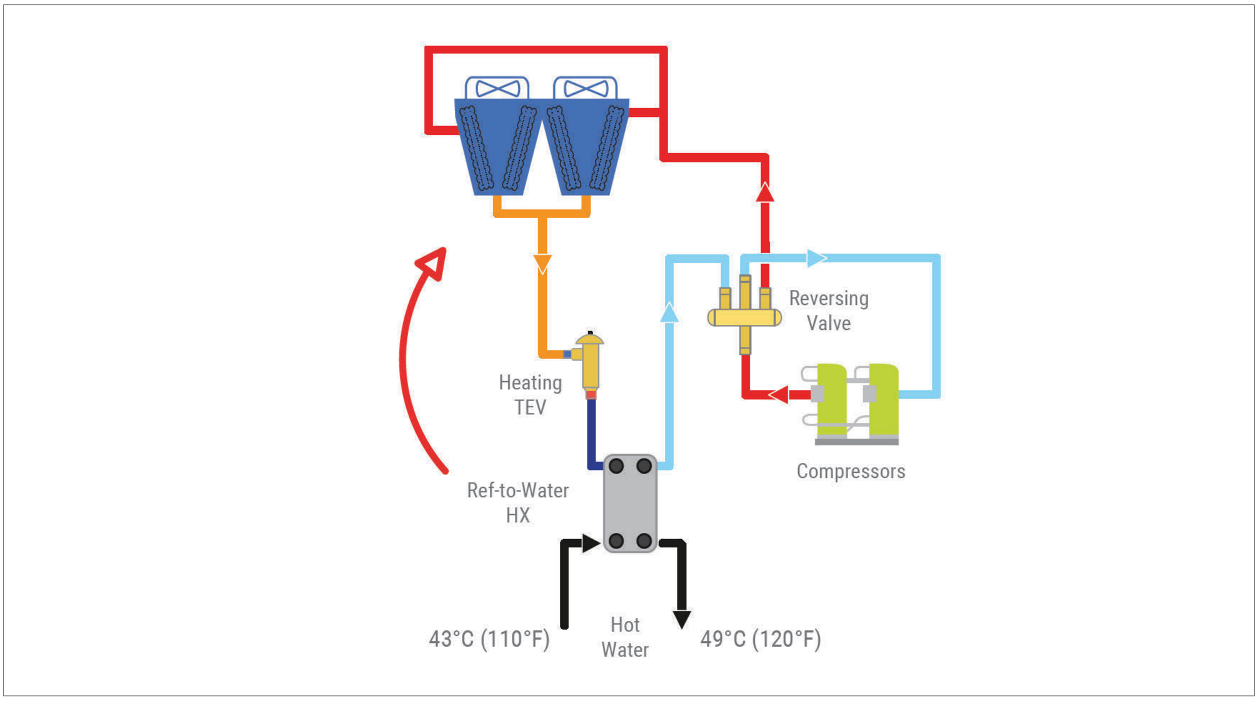

Figure 3.3 | *Single Circuit ASHP in Defrost Mode*

The unit is now removing heat from the hot water loop to melt the frost.

Note the supply water temperature is now below the return temperature. During defrost there is no energy balance in the building. Assuming the defrost cycle does not carry on too long, it is unlikely the occupants will be affected. The hot water is still warmer than the room air and heat will still be flowing from the hot water into the occupied space through the terminal units in the indoor spaces, as long as the thermal mass is sufficient.

The bigger issue is to ensure a complete defrost cycle. When the ASHP operates in defrost (i.e. chiller) mode, it is very efficient at removing heat from the water loop. Consider a nominal 352 kW (100 ton) ASHP. The cooling capacity is based on summer

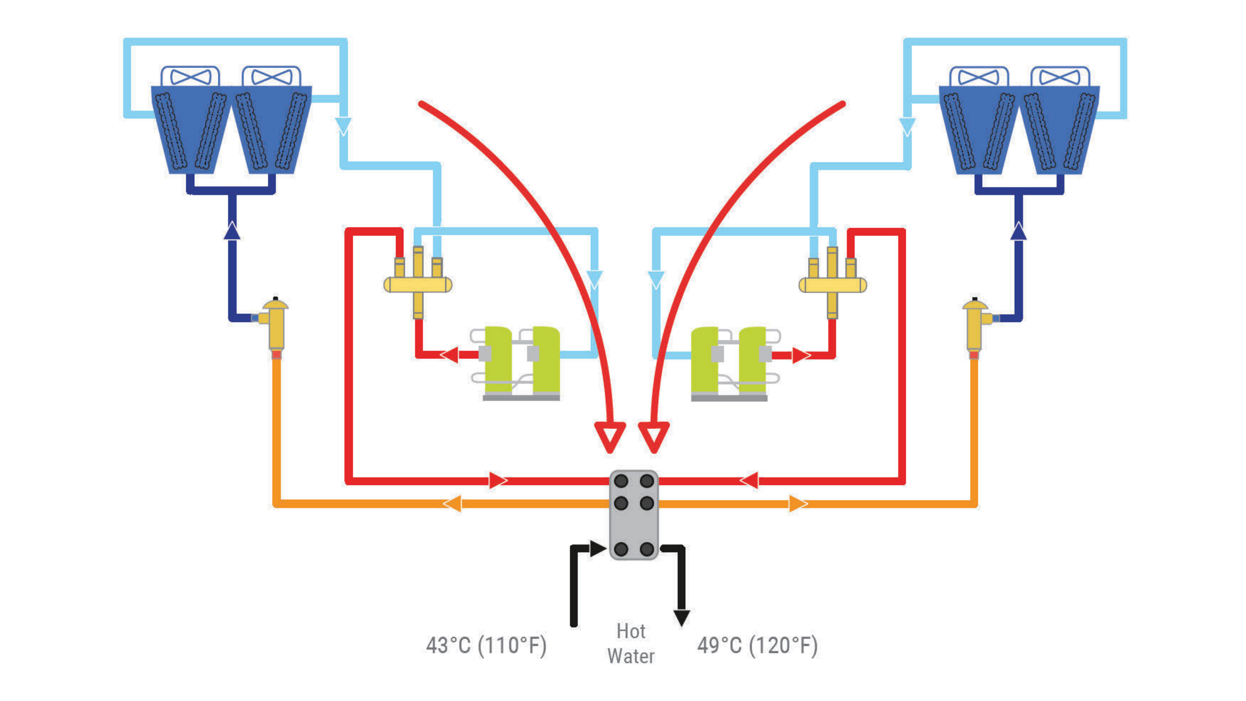

Figure 3.4 | *Dual Circuit ASHP in Heating Mode*

conditions, typically rated at 35 °C (95 °F). When operating in chiller mode below freezing, the capacity will increase making it very effective at removing heat from the hot water loop.

If there is not enough thermal mass, the lack of available heat will result in an incomplete frost melt and over the course of several defrost cycles the outdoor coil can become blocked by ice. The unit will shut down on an alarm and require service to resolve. It can lead to serious damage to the unit.

The solution is to ensure there is enough water volume to provide the thermal mass and this will likely mean adding a buffer tank in the primary heat pump loop. The tank size is the difference between the required water volume and the water volume in the piping that is available during a defrost cycle (i.e. water connected to the heat pump loop that may flow through the ASHP).

Figure 3.4 shows the same example as Figure 3.3 above but now the ASHP has two completely independent refrigeration circuits. Both circuits are operating in heat pump mode with frost slowly forming on both refrigerant-to-air coils.

At some point, one of the circuits will need to defrost as shown in Figure 3.5. The unit controller should not allow both circuits to defrost at once. The difference between this and the previous example is one circuit is operating in heating, while the other circuit is operating in defrost (cooling). In the previous example the entire capacity of the single circuit ASHP unit went into defrost.

While it appears that the two circuits will “cancel each other out” neither adding nor removing heat from the loop, that is not likely what will happen.

Figure 3.6 shows the capacity output of each circuit during defrost according to ambient temperature. When it is approximately 9 °C (48 °F) outside, the heating and cooling capacities “cancel out” leading to no net heating or cooling to the system. As the ambient temperature continues to decrease, the cooling capacity of the circuit undergoing defrost increases while the heating circuit capacity decreases. Below the cross over temperature, the ASHP is drawing heat out of the system and the thermal mass must be increased to offset the impact.

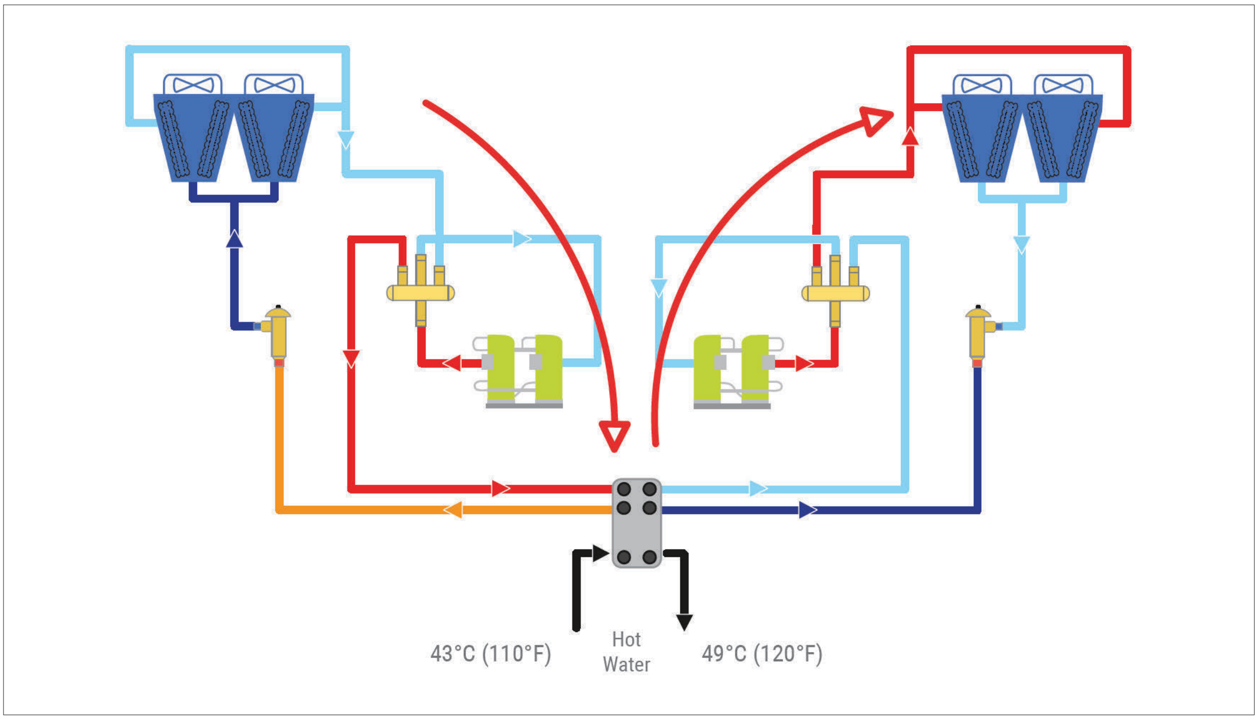

Figure 3.5 | *Dual Circuit ASHP in Heating Mode (Defrost Context)*

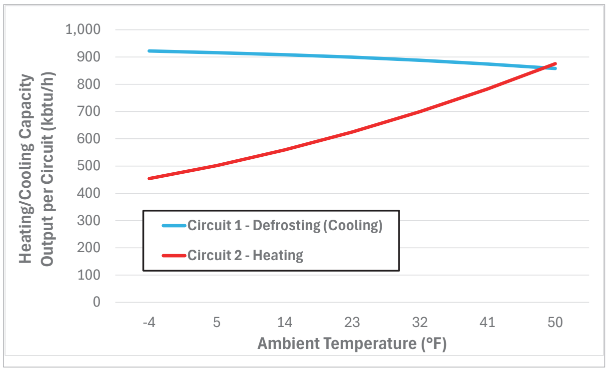

Figure 3.6 | *Capacity Output of Independent Circuits During Defrost*

There are a couple of key notes to be gained by comparing single-circuit to multiple circuit units.

First, dual circuit units require much less thermal mass to operate properly than a single circuit ASHP.

Second, the amount of thermal mass is a dynamic calculation and cannot be simply based on system size with a rule of thumb. For example, consider a building in Seattle, WA with an ambient design temperature around 8 °C (17 °F). Based on Figure 3.6, the defrost cycle will have a minor effect on the thermal mass of the system. Now move the building to Chicago where the ambient temperature is -20 °C (-4 °F). The imbalance between heating and cooling is now 50% which will greatly affect the thermal storage requirement.

$$\boldsymbol{EN}_{\boldsymbol{in}} = P_{units}[_{kW}] * t\,[_{sec}] = waterflow\left[\frac{m3}{sec}\right] * \rho water\left[\frac{kg}{m3}\right] * DT[_{°C}] * Cpwater\left[\frac{kJ}{Kg * °C}\right] * t\,[_{sec}]$$

$$\boldsymbol{EN}_{\boldsymbol{storage}} = Cp_{water}\left[\frac{kJ}{Kg*°C}\right] * TankVolume\,[m^3] * \rho_{water}\left[\frac{kg}{m^3}\right] * \mathbf{DT}_{\mathrm{storage}}[°C]$$

$$\boldsymbol{EN}_{\boldsymbol{out}} = Pabsorbed[kW] * t\,[sec] = \boldsymbol{Max\ Load}[\boldsymbol{kW}] * \boldsymbol{p}\%[\%] * t[sec]$$

Figure 3.7 | *Thermal Mass Equations*

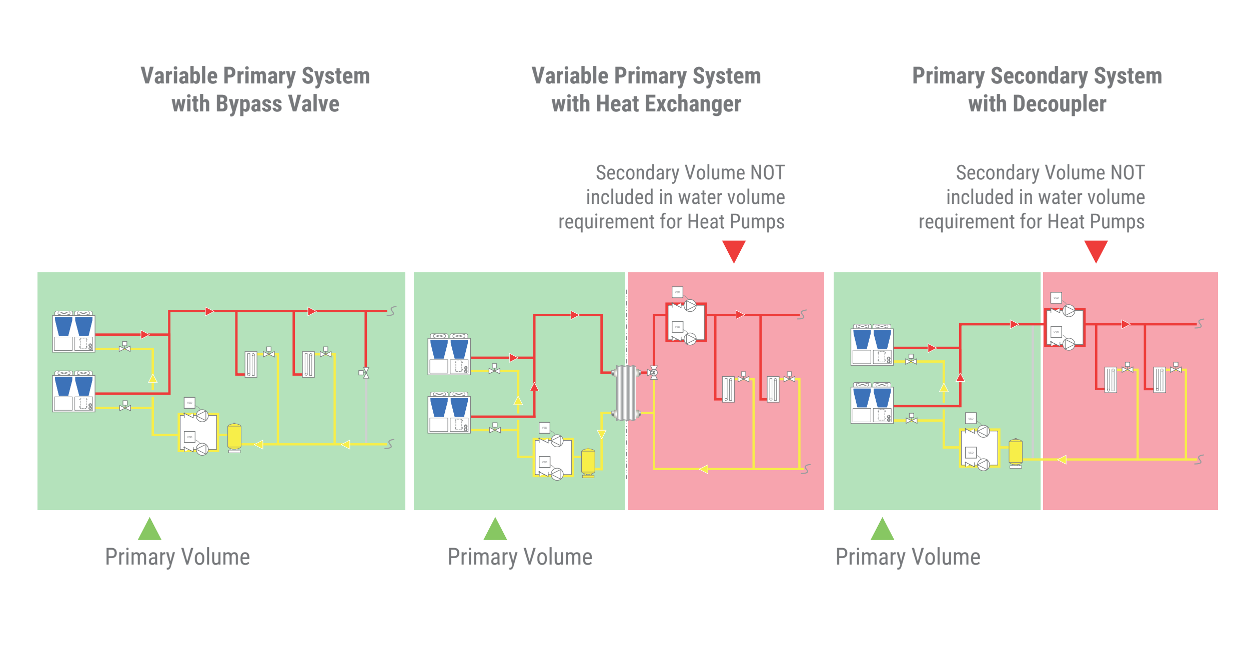

Figure 3.8 | *Primary Water Volume Available for Thermal Mass*

Calculating Thermal Mass

Accurate calculation of thermal mass requires several variables and operating parameters. Often this is reduced to "8-10 liters/kW (10 gal/ton)" but as demonstrated above, this is only an approximation. Proper calculations require the following;

Delivered Water Stability – What water temperature change $DT_{storage}$ is acceptable both for the building and the unit defrost cycle. The range will depend on the equipment and application, but it is likely to be 7 °C (14 °F) or less to keep the hydronic system operating within an acceptable range at all times. If the application requires a closer temperature tolerance, then the thermal mass will have to increase.

ASHP Properties – Compressor type, number of compressors, number of circuits, heating capacity, cooling capacity etc.

Design Ambient Temperature and Bivalence Ambient Temperature – These temperatures are used to account for the imbalance between heating and defrost capacities as discussed above.

Tank Water Temperature – If the tank temperature can drop significantly during unoccupied hours, then the impact of having to heat up the tank up should considered. This is often referred to as a "cold start".

Defrost Time and Compressor Staging Time – Both time cycles will impact the thermal mass requirements.

Number of Units in Plant – How many total refrigeration circuits there are across all the units and how many can be in defrost at any given time will impact the thermal mass. However, it requires the units to be in communication to stage defrost cycles across multiple machines. If they are not coordinated, then multiple units are not helpful in reducing the system volume as it can be expected that multiple units will defrost at the same time.

A lot of effort is being spent on the details of thermal mass and ASHPs because experience has shown that this is one of the most critical issues to get correct to ensure a successful project. It is recommended that the equipment manufacturer provide guidance on the required water volume based on the actual application. Also, once the system volume is established, all potential equipment providers confirm their equipment will work properly with the volume provided by the design.

Water Volume Definition

The amount of water volume available for defrost needs to be carefully considered. While the building hydronic loop may contain a significant amount of water, it may not be readily available for a defrost cycle which can occur in just a few minutes. Figure 3.8 shows three common hydronic systems highlighting the primary volume which should be considered for confirming sufficient water volume. Decoupler/bypass lines and heat exchangers have proven to short cycle or isolate the primary water volume available to ASHPs during defrost. If the primary water volume is less than the required thermal mass, a buffer tank should be added to make up the difference.

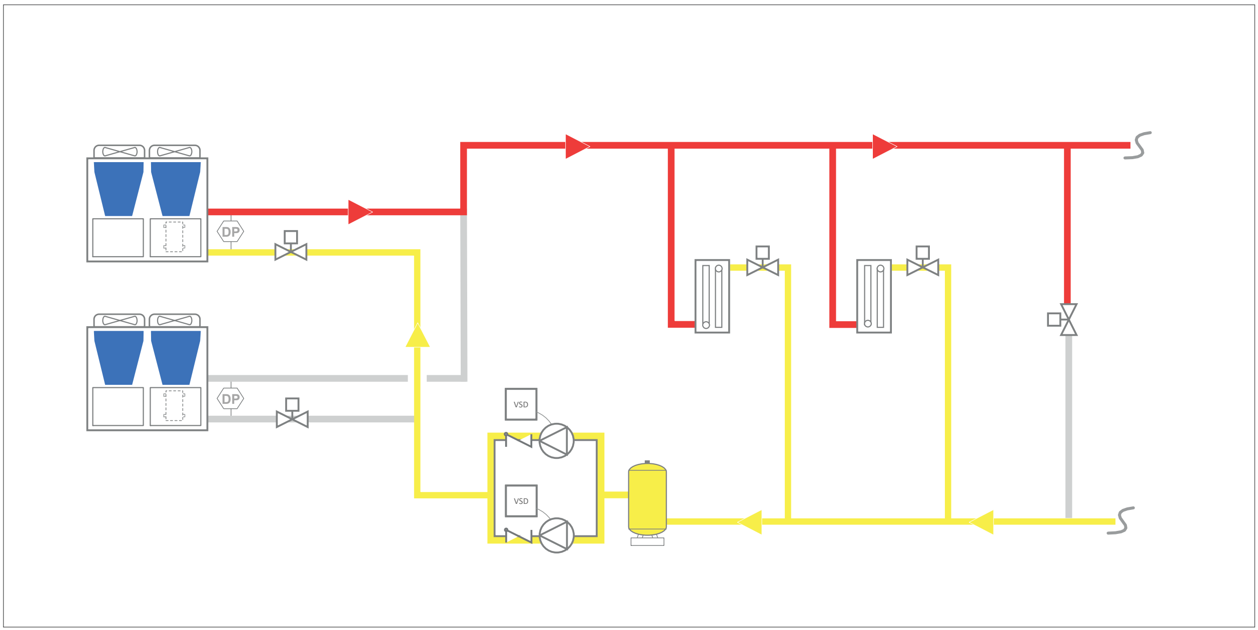

Figure 3.9a | *Buffer Tank Locations*

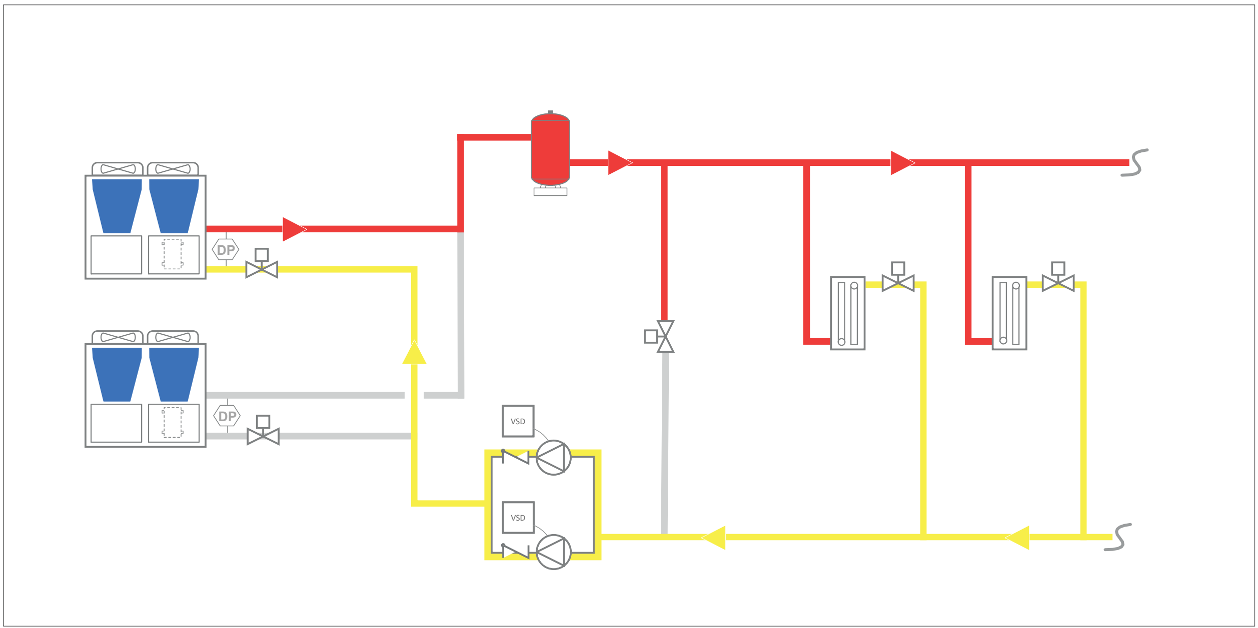

Figure 3.9b | *Buffer Tank Locations*

Buffer Tank Location Considerations

Figure 3.9a shows a buffer tank in the return line of the primary loop and Figure 3.9b shows a buffer tank in the supply line of the primary loop. It is critical that the tank be in the primary loop to keep the thermal mass near the ASHPs for defrost.

Placing the tank in the return line provides a large source of hot water when the ASHPs enter defrost mode. This improves the defrost cycle performance. Placing the tank in the supply line minimizes the impact of the cold water from defrost on the building loop.

Both locations have advantages. Generally, even during defrost, the supply water temperature to the building will be warm enough to still provide heat and not affect the occupants. In this case, the priority should be given to ensuring a complete defrost cycle and the tank should be in the return line.

Thermal Mass Summary

» Sufficient thermal mass is more important in heating than cooling due to defrost.

» Ensuring a complete defrost is generally a bigger issue than any impact on occupant comfort.

» Rules of thumb such as “10 gallons per ton” are rough estimates and a more thorough approach should be used taking into account many parameters especially ambient temperature and whether the equipment can stage defrost cycles.

» The primary water volume available for defrost should be carefully considered and is generally less than the hydronic system volume. The primary water volume available for defrost should be carefully considered and is generally less than the hydronic system volume.

4 4-Pipe Multifunctional Units

Figure 4.1 | *Air Source 4-Pipe Multifunctional Unit*

Introduction

This chapter introduces an advanced form of ASHP that can deliver simultaneous heating and cooling. Integrating this type of unit into plant applications can greatly improve performance, efficiency, and reduce energy consumption and carbon emissions. It will also compare the multifunctional unit to standard heat recovery chiller units.

4PMF units

Reversible ASHP units can only operate in either heating or cooling mode and so lend themselves to 2-pipe hydronic systems with spring and fall changeover. A 4-pipe multifunctional unit (4PMF) can operate in heating and cooling modes simultaneously while meeting heating and cooling set-points independently, thus supporting 4-pipe hydronic systems. See Figure 4.2.

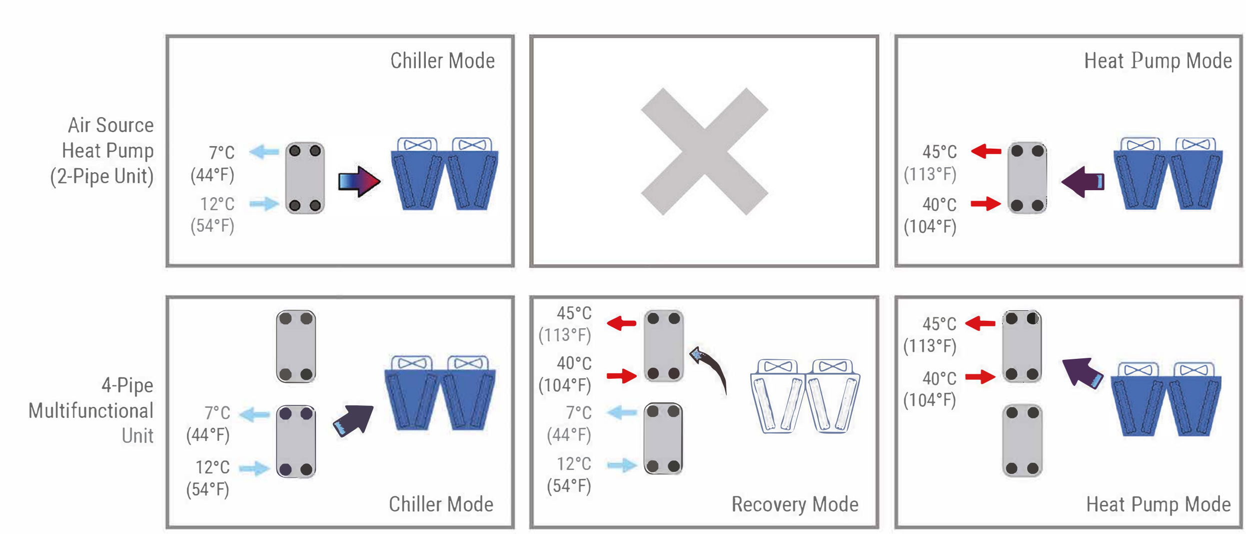

Figure 4.2 | *ASHP vs. 4-Pipe Multifunctional Unit*

A performance metric used to rate the efficiency for 4-pipe multifunction units is referred to as the Total Efficiency Ratio, or TER. In a traditional reversible ASHP, the COP (heating output divided by power input) typically ranges from 3 to 4 W/W (10 to 13.6 Btu/h-W) at nominal conditions. The TER takes into account the recovered energy and is the heating output plus the cooling output, divided by the total power input, which will typically be in the range of 6-8 W/W (20.5- 27.3 Btu/h-W).

Figure 4.3 shows a 4-pipe multifunctional unit schematic. The unit has two sets of water connections, one for hot water and one for chilled water. The unit can operate as outlined below;

100% Heating — The unit operates the same as a 2-Pipe ASHP in heating mode, extracting heat from ambient air to heat the hot water loop.

75% Heating / 25% Cooling — The heat extracted from the chilled water loop is sent to the hot water loop. The additional heat required to meet the heating load is extracted from the ambient air.

50% Heating / 50% Cooling — Whenever the heating and cooling loads are balanced, the unit operates as a water-to-water unit moving the heat from the chilled water loop to the hot water loop. The power used to deliver both the heating and cooling capacity is very low so the Thermal Energy Recovery (TER)[1] can reach 6 to 8 depending on the hot water temperature.

25% Heating / 75% Cooling — 25% of the heat extracted from the chilled water loop is sent to the hot water loop. The remaining 50% heat is rejected to ambient air.

100% Cooling — The unit operates the same as a 2-Pipe ASHP in cooling mode, extracting heat from the chilled water loop and rejecting it to ambient air.

[1] TER is equivalent to a COP for 4 pipe multifunction units. Basically, it is the heating plus cooling capacity divided by the electrical power consumed.

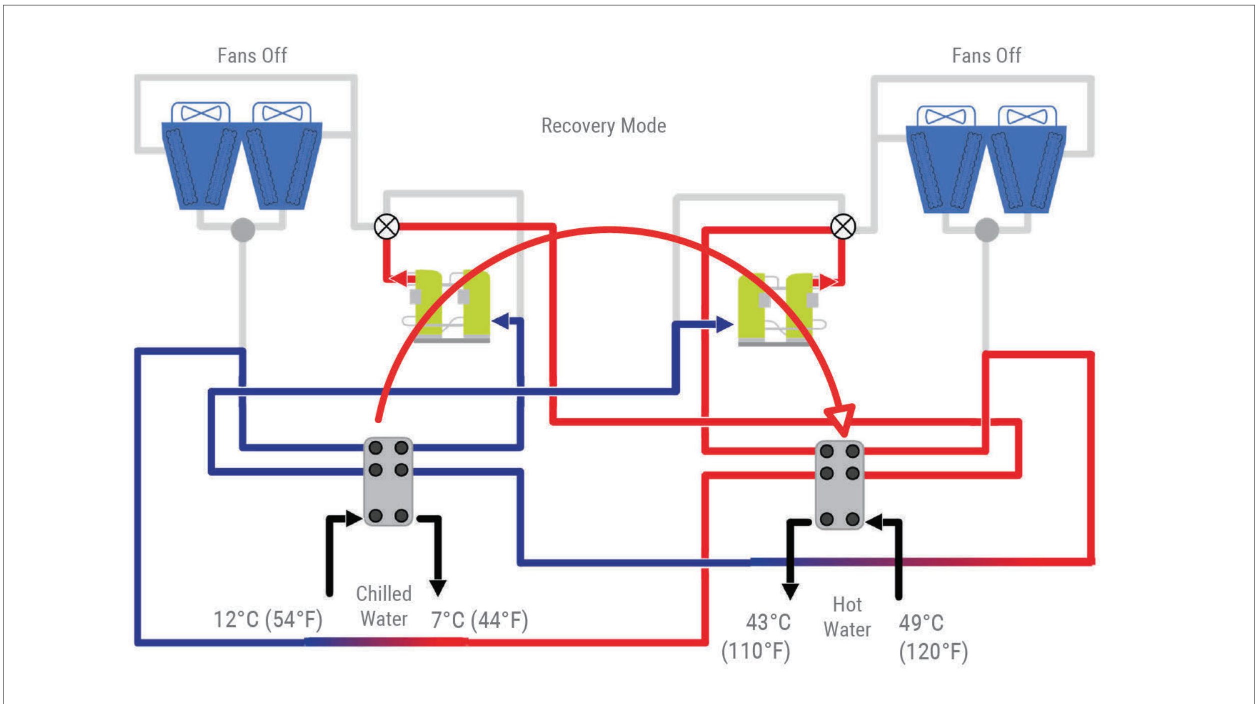

Figure 4.3 | *4-Pipe Multifunctional Schematic*

Desuperheater Energy Recovery

A common option to get heat recovery with a chiller is to use a desuperheater. This device will heat water by cooling the discharge gas from the compressor to the condenser. There is no capacity control on the heating side of the desuperheater. The available heat will depend on the operating conditions being experienced by the chiller. Further, the unit control algorithm is based on meeting the cooling load only. If the unit is operating at 25% load, then that is all the heat that is available regardless of the actual heating demand.

A good application for a desuperheater is to preheat domestic hot water (DHW). It will likely not be enough to meet the DHW heating demand, but any recovered heat will help reduce the building energy use.

Air Source Heat Recovery Chiller

An Air source heat recovery chiller has both a refrigerant-to air condenser and a Refrigerant-to-water condenser piped in parallel in the refrigeration circuit. It will have 4-pipe hydronic connections. The controls logic provides as much refrigeration capacity as required to meet the cooling load. All heat collected from the chilled water loop, plus the heat of compression, is available for heating. For example, a 350 kW (100-ton) unit running at 50% cooling load would have 175 kW (50 tons) of heat plus the heat of compression available for heating. If the heating load is greater than 175 kW (50 tons), the unit will not meet the load, and a supplemental energy source will be required.

Many heat recovery units cannot modulate heat intended for the hot water loop at all. If there is 175 kW (50 tons) of heat being collected in the chilled water loop, all of it must go into the hot water loop even if the heating load is less. This can lead to temperature control issues.

4-Pipe Multifunctional Units vs. Heat Recovery Units

4-pipe multifunctional units and heat recovery units operate differently. A 4-pipe multifunctional unit can operate from 0 to 100% heating output and 0 to 100% cooling output independently and simultaneously, while recovering heat between the chilled water loop and hot water loop. There are two temperature sensors for control located in both the hot water and chilled water loops and two control algorithms looking to satisfy the load in each loop.

A heat recovery chiller can operate from 0-100% cooling out put but the heat output is limited to what can be recovered from the cooling load. If the heating load exceeds the cooling load, the heat recovery chiller will not meet the heating load.

4-Pipe Multifunctional Unit Summary

A 4-pipe multifunctional unit can match the heating and cooling loads simultaneously. The TER (efficiency) during simultaneous heating and cooling is twice the normal COP. The real advantage occurs when there is simultaneous heating and cooling in the building. It is possible to meet any combination of heating and cooling that could occur throughout the year. This is not possible when only ASHPs or heat recovery units are used. This will be discussed in greater detail in future chapters.

5 Equipment Selection

Introduction

This chapter will consider the impact of the building heating and cooling load profiles on the plant design including equipment size, type and quantity while delivering performance and reliability.

Design Example

Figure 5.1 shows a load profile for a building located in Seattle, WA. In addition to the load profile;

Design Cooling Load: 400 kW (114 tons)

Design Heating Load: 280 kW (955 kBtu/h)

Preliminary Design

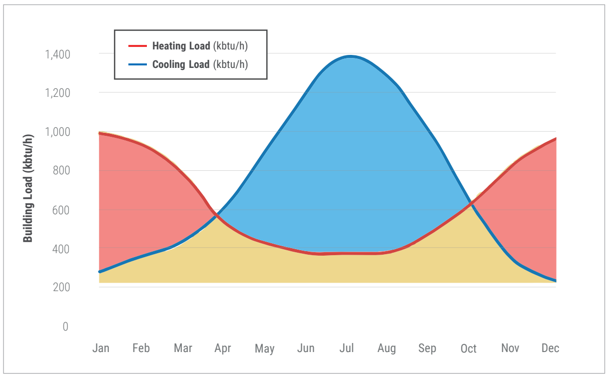

Figure 5.1 | *Annual Building Load Profile*

One of the first steps is to decide if the heating load can be met with just ASHPs or if some form of supplementary heat will also be required. The winter design condition in Seattle is mild at -8.3 °C (17 °F). Most commercial ASHPs can operate down to -20°C (-4°F) so Seattle is well within the operating envelope. In this case the heating load can be met with just ASHP units.

Effectively every project is either heating or cooling dominant. As it is the same piece of equipment providing the heating and the cooling it can only be sized to meet one of these two requirements. Generally a good place to start is by selecting equipment that meets the cooling load and seeing what heating capacity it will deliver at the design ambient conditions.

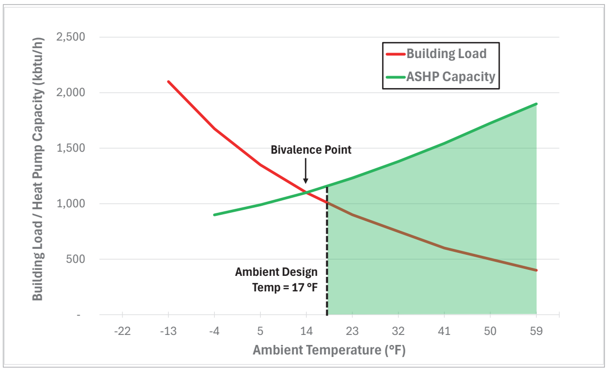

Figure 5.2 | *Seattle Example Bivalence Point*

For Seattle, (see Figure 5.2) building load is summer dominant. It means the heating design point is to the right of the bivalence point. Sizing for cooling will mean the ASHP are large enough for heating.

Had the project been in a slightly colder climate than Seattle, the winter design load would fall slightly to the left of the bivalence point. Additional heat would be required as the ASHPs could not meet the winter design condition.

One approach (see Figure 5.3) would be to size the ASHPs for the winter load. As discussed in Chapter 1, care should be taken as the plant can become oversized during periods of low heat requirement and will be oversized for the peak cooling design condition. The recommendation is to use multiple ASHPs so as the winter ambient temperature increases and in summer cooling mode, capacity steps from different units can be staged off to better match the actual load.

An alternative to sizing the ASHP to meet the winter design condition is to add supplemental heat sized for the difference between what the ASHP can deliver, and the building heating load as shown in Figure 5.4.

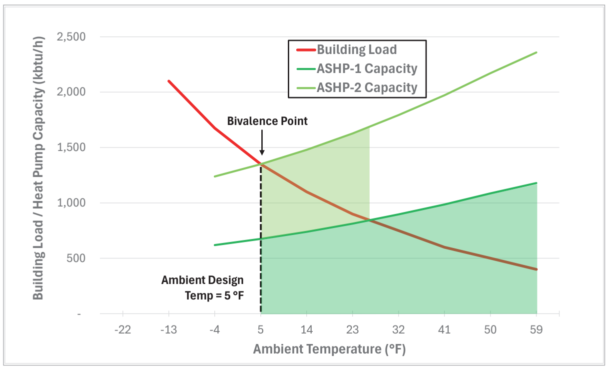

Figure 5.3 | *Sizing for Winter Load*

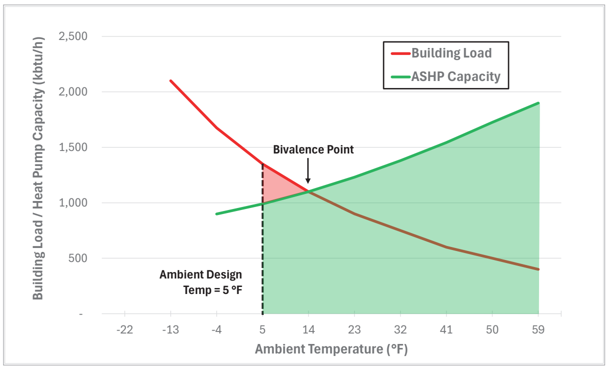

Figure 5.4 | *Simultaneous Supplemental Heat*

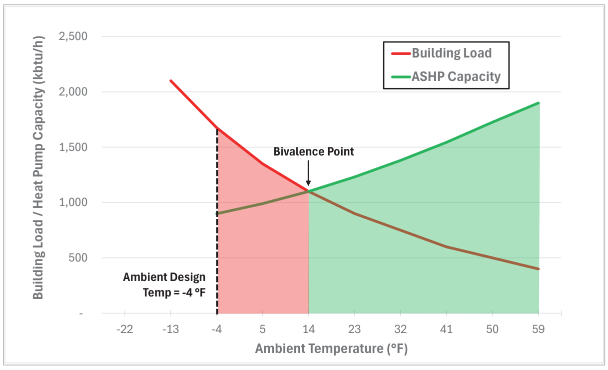

Figure 5.5 | *Full Supplemental Heat*

Both increasing the ASHP size to meet the load and adding simultaneous supplemental heat will deliver an acceptable solution. Things to consider when choosing between the two solutions include;

» If the project requires an all-electric solution, then size the ASHP for winter load.

» Using supplemental gas heat offers a second primary energy source for the project which will improve building resiliency. While a gas boiler can add to the project carbon footprint there are not a lot of operating hours below the bivalence point so the impact is minimal. Supplemental gas heat is likely a lower capital cost solution when the smaller electrical service and the cost difference between a gas boiler and ASHP are considered.

» As the ambient temperature drops, the ASHP COP also drops. It may cost more to operate an ASHP at low ambient temperatures than a gas boiler. This will depend on the local cost of natural gas and electricity.

In very cold climates where the winter design temperature is near or below the minimum ambient temperature the ASHP can operate at, a supplemental heat source sized for the entire heat load will be required as shown in Figure 5.5. It is not recommended to operate the ASHPs down to the operating envelope limit prior to switching to the boiler. A common solution is to switch over at the bivalence point but consider how many operating hours there are to the left of the bivalence point.

If there are a lot of BIN hours between the bivalence point and the minimum ASHP operating temperature, then it may justify operating the boilers and ASHPs simultaneously in this zone. Once the ambient drops to the minimum ASHP temperature the ASHP are shut down. Care should be taken in sizing the pumps to meet the minimum flows of both the ASHPs and the boilers. It may be possible that the minimum flows exceed the system design flow.

A final consideration is redundancy. Generally, it is advantageous to have at least two units on the project so there is some capacity should one of the units not operate. If the building application is more critical then a N+1 scenario may be considered.

Advanced Design Considerations

In the previous section the equipment was sized to meet the summer and winter design points. However, the plant will rarely ever operate at these points so consideration to the entire operating range should be given.

Referring again to the Seattle example, a few key observations can be made. There is an overlap in the heating and cooling curves indicating that the building will have simultaneous heating and cooling loads. If the hydronic system is a 2-pipe changeover design, then there will be times when not all the building zones can be satisfied. The heat-cool units will only be ASHPs (the hydronic system can only be in heating or cooling) as 4PMF or heat recovery units require 4-pipe systems. The ASHPs are piped into the water loop and a signal from the building Automation System (BAS) will indicate whether the units operate in heating or cooling.

If the hydronic system is 4-pipe, then all the building zones have access to both hot and chilled water and can be satisfied. There several approaches in equipment selection to meet the load profiles where both heating and cooling occur simultaneously.

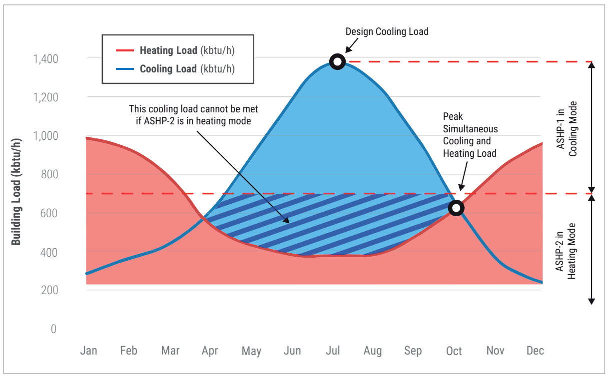

Figure 5.6 | *Multiple ASHPs*

Option 1 — Multiple ASHPs

Figure 5.6 shows multiple ASHP units in which the combined capacity meets the design load. Consider two 200 kW (57 tons) units for the Seattle example. In winter months, both units operate in heating mode. In summer months, both units operate in cooling mode. During the shoulder season where there is simultaneous heating and cooling, one unit is valved into the heating loop to provide heat while the other unit is valved to the chilled water loop to provide cooling.

The more units in the plant, the closer the capacity can be correctly allocated to the hot water or chilled water loop. In practical terms, there will be times when one ASHP is not enough for the hot or chilled water loop. For example, when the heating load is 150 kW (512 kBtu/h) and the cooling load is 300 kW (86 tons), one ASHP operating in heating can meet the heating load but the second unit operating in cooling will not meet the cooling load (see blue-hatched area in Figure 5.6). Adding more smaller ASHPs will allow the units to be assigned to heat or cool and better match the heating and cooling load profiles.

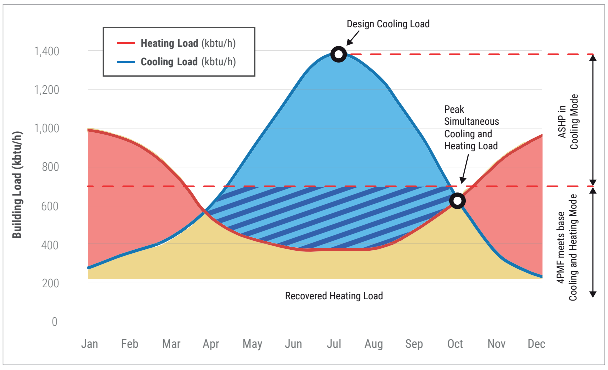

Figure 5.7 | *ASHPs Plus 4PMF Units*

Option 2 — ASHPs Plus 4-Pipe Multifunctional Units

Figure 5.7 illustrates a combination of 2-Pipe Reversible ASHPs and 4-pipe multifunctional units to meet the load throughout the year. The 4-pipe multifunctional unit is permanently connected to both the hot water and chilled water loops. The multifunctional unit(s) are sized to meet the largest coincident simultaneous heating and cooling load. The remaining required capacity is met with 2-Pipe Reversible ASHPs that are switched over seasonally. The ASHPs can be connected to either the hot and chilled water loops through valving to meet the additional load in summer and winter.

In the Seattle example there would be one 200 kW (57 ton) 2-pipe reversible ASHP and one 200 kW (57 ton) 4-pipe multifunctional unit. The 2-pipe ASHP can be valved to connect it to either the hot water or the chilled water loop as needed to meet the seasonal peaks, while the 4-pipe multifunctional unit will meet both the cooling and heating loads during simultaneous heating and cooling periods. The 4-pipe unit will move the heat collected from the chilled water loop to the hot water loop offering significant energy savings (see Chapter 4: 4-Pipe Multifunctional Units). When either the heating or cooling load exceeds the capacity of the multifunctional unit, the 2-pipe ASHP is valved to the appropriate loop to provide the additional capacity.

The controls logic is simplified as the 4-pipe multifunctional unit is always enabled for both the heating and cooling loads to satisfy the base-loads of both heating and cooling. The ASHP can be switched to the appropriate operating mode (and hydronic loop) whenever the 4-pipe multifunctional unit cannot meet the load.

The major advantage with the 4-pipe multifunctional unit is that it will provide energy recovery by moving heat from the chilled water loop to the hot water loop for a longer period of the year, often with a TER as high as 8 W/W (26 Btu/W-h). This means that the energy efficiency of the building can be significantly improved.

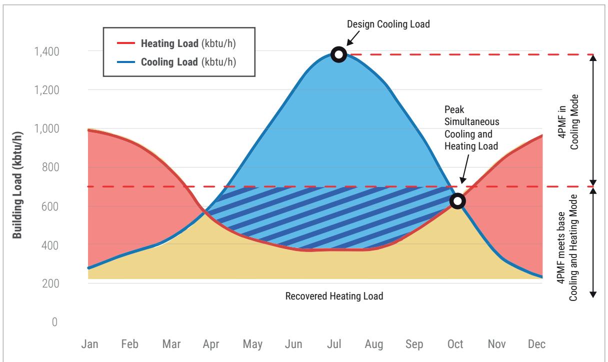

Figure 5.8 | *4PMF Units*

Option 3 — 4-Pipe Multifunctional Units

Figure 5.8 uses 4-pipe multifunctional units only. All the units are connected to both the hot water and chilled water loops so no hydronic switchover valving and controls are required. The units can provide simultaneous heating and cooling. When there are simultaneous loads, the heat collected in the chilled water loop is transferred to the hot water loop offering significant energy savings.

In the Seattle example, there would be two 200 kW (57 tons) 4-pipe multifunctional units.

A drawback to this selection is that the equipment is more expensive. Another complication is oil management within the 4-pipe multifunctional unit.

If the unit operates for an extended period in cooling only, then it will be necessary to operate the unit in heat recovery mode for a brief period to move the oil that has migrated to the refrigerant-to hot water heat exchanger[1]. This requires the hot water pump to start for a short period so the unit heating mode can be engaged.

Option 1 is typically the lowest cost solution, but care must be taken to ensure the simultaneous heating and cooling loads can be met. Option 2 is a good choice when there is a small simultaneous load relative to the design load, and in very large systems where the cost savings of using a combination plant can become significant. Option 3 is a good choice when the simultaneous load is a large percentage of the total load and where simplicity of system design and operation are important.

[1] Oil flows with the refrigerant throughout the unit. Some amount of refrigerant and oil will collect in parts of the refrigerant circuit that are closed/idle. At some point this oil needs to be returned to the compressor sumps.

Equipment Selection Summary

» When using ASHPs, it is the same piece of equipment for both heating and cooling so the units can only be sized for one load or the other.

» A key decision is whether the ASHP can meet the heating load. In some cases, oversizing heating is an option. Alternatively adding supplemental heat to operate simultaneously can work. For very cold climates, a boiler plant sized for the entire heating load will be required.

» A good place to start is to size the ASHPs in cooling and see if they will meet the heating load.

» A 2-pipe hydronic system only needs ASHPs

» A 4-pipe hydronic system can include 4-pipe multifunctional units

» Most buildings have simultaneous heating and cooling. This can be met by;

- Multiple ASHPs. Caution should be taken when considering this approach as it may not be possible to meet all the combinations of heating and cooling the building may experience. An additional ASHP may be required.
- ASHPs and 4PMF units. This approach can meet any combination of heating and cooling the building may experience. During simultaneous heating and cooling, the 4PMF unit will operate as a water to water unit doubling its COP so this is also a very efficient design.
- Multiple 4PMF units. This approach offers the same advantages as the ASHP and 4PMF option but is easier to design and operate. It may have a higher capital cost.

6 Water Temperature and Flow Ranges

Introduction

Selecting chilled and hot water supply temperature and flow ranges is a critical part of optimizing a hydronic system. Using ASHPs limits the options, particularly on the hot water system due to the operating envelope. This section will cover the basics, show the issues, and offer common solutions.

Supply Water Temperature

Figure 6.1 shows typical operating envelopes for ASHPs. Most chilled water systems are designed with 7 °C (44 °F) supply temperature which is well within the range of an ASHP.

Hot water supply temperatures with ASHPs are more complicated. Most ASHPs can deliver 60 °C (140 °F) hot water, but it is not advised to design at the limit of the operating envelope. Typically, 50-55 °C (120-130 °F) offers good results and equipment longevity.

In new construction or major renovations, using 50-55 °C (120-130 °F) hot water is straightforward. It may require slightly larger heating coils which can lead to higher fan static pressures but the energy savings from operating the ASHP at a lower temperature will generally more than offset the fan work penalty.

Another consideration is whether the system includes a gas boiler. Condensing boilers offer high efficiency performance. To achieve this, the return water should be below the dew point of the flue gases—around 50 °C (120 °F)—which aligns well with ASHP operating temperatures.

Projects with existing hot water systems designed around 82 °C (180 °F) are a challenge for heat pumps. Heat pumps utilizing Propane (R-290) as the refrigerant (where they are allowed to be used) have a wider operating envelope that can come close to these higher temperatures. Alternatively, some form of cascading system can be used. ASHPs are used to operate a loop around 50-55 °C (120-130 °F) and a second water-to-water heat pump boosts the temperature to the 82 °C (180 °F) range.

Temperature Ranges

Most ASHPs and 4-pipe multifunctional units are designed to operate with 6 °C (10 °F) temperature ranges. The refrigerant-to-water heat exchangers have been selected to offer a reasonable water pressure drop while allowing the flow to be reduced to approximately 50% of the design flow rate without serious laminar flow performance[1] issues. While 6 °C (10 °F) is a common range for cooling it is traditionally smaller than the range used with boilers (typically 11-22 °C (20-40 °F)).

[1] Laminar flow occurs in heat exchangers when the Reynolds number drops below 2,000. At this condition, the fluid velocity is low enough that turbulence is not possible. The heat exchanger will not transfer much heat and the refrigeration unit will not operate properly, if at all.

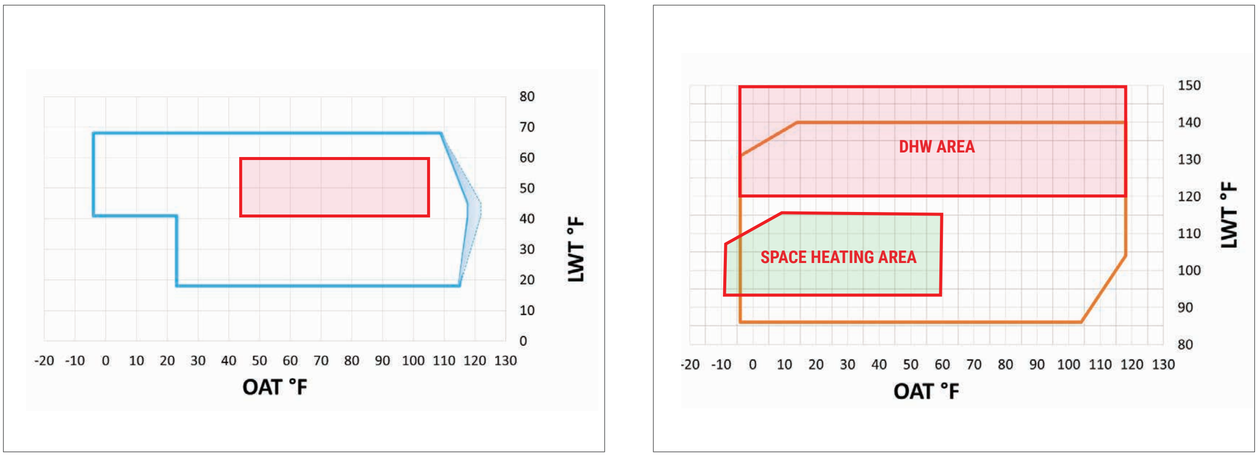

Figure 6.1 | *ASHP Operating Envelopes*

	Cooling	Heating
Ambient Air Temperature	35 °C (95 °F)	-10 °C (14 °F)
Capacity	427 kW (122.1 tons or 1465 kBtu/h)	328 kW (1118 kBtu/h)
Supply Water Temperature	7 °C (44 °F)	54 °C (130 °F)
Water Flow	20.4 l/s (323.3 USgpm)	20.4 l/s (323.3 USgpm)
Delta T	5.5 °C (10 °F)	3.8 °C (6.9 °F)
Water Flow		15.3 l/s (243 USgpm)
Delta T		5.5 °C (10 °F)
Minimum Flow	12.3 l/s (195 USgpm)	12.3 l/s (195 USgpm)

Figure 6.2 | *ASHP Performance in Heating and Cooling*

An additional challenge with ASHPs is that the capacity in heating and cooling is different. The colder the winter design temperature is, the greater the difference will be.

Figure 6.2 shows a nominal 135 ton ASHP performance in cooling and heating with a winter ambient of -10 °C (14 °F). At this ambient the heating capacity is only 76% of the cooling capacity. If the primary water flow is fixed (e.g. a primary -secondary system) then the heating delta T drops to 3.8 °C (6.9 °F). If the system is variable primary flow, then the delta T can be maintained but the flow drops from 20.4 l/s (323.3 USgpm) to 15.3 l/s (243 USgpm).

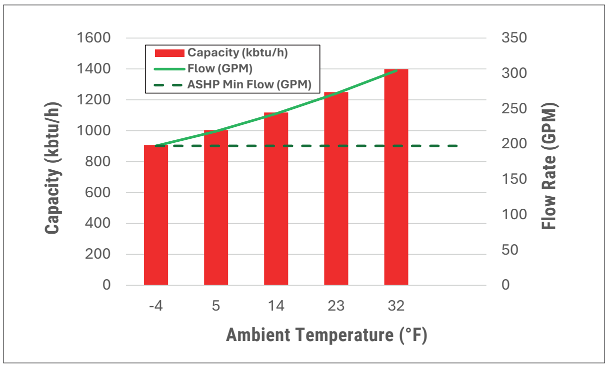

Figure 6.3 | *ASHP Capacity and Flow with 5.5 °C (10 °F) Delta T*

Figure 6.3 shows the same unit at various ambient temperatures while maintaining a 5.5 °C (10°F) delta T. The delta T is possible throughout the ASHP operating envelope.

Figure 6.4 shows the ASHP with a 8.3 °C (15 °F) Delta T. In this case, as the ambient drops and the heating capacity is reduced, the ASHP reaches its minimum flow rate around -5.5 °C (22 °F). Even with a variable primary flow hydronic system, the minimum flow will need to be maintained so below this ambient, the delta T will start to drop.

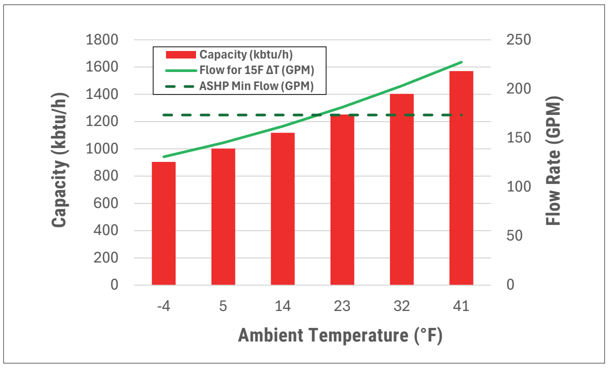

Figure 6.4 | *ASHP Capacity and Flow with 8.3 °C (15 °F) Delta T*

If a higher delta T is used, minimum flow will be reached sooner and at that point the delta T will start to drop. While these curves may change for different models, the same issue will occur with any ASHP.

While it should be confirmed with project to specific selections, it can be assumed that using a 5.5 °C (10 °F) in heating and cooling throughout the operating envelope is possible. In mild winter conditions, it may be possible to achieve 8.3 °C (15 °F) or greater delta Ts but it will all depend on the ASHP minimum flow. Adding glycol to the system will effectively raise the minimum flow requirement.

While it is possible to design an ASHP with a larger delta T (and minimum flow rate) it would also affect the cooling Delta T as well. Since it is the same piece of equipment for both heating and cooling some compromises is required. It is possible to increase the secondary loop delta T based on the system design. This will be covered in future chapters.

Water Temperature and Flow Ranges in a 2-Pipe system

2-pipe changeover systems use the same distribution piping for either heating or cooling. This reduces the capital cost by avoiding a second hydronic loop for a 4-pipe system. 2-pipe systems are commonly used with fancoils, unit ventilators, chilled beams and air handling unit systems.

A 2-pipe system can only be in heating or cooling mode. It is generally a seasonal decision by a building operator or the BAS system. Heating up or cooling down the entire hydronic loop makes any short term changeover strategy not possible.

	Cooling	Heating
Capacity	43kW (147 kBtu/h)	39 kW (134 kBtu/h)
Entering Water Temperature	7 °C (44 °F)	54 °C (130 °F)
Delta T	5.5 °C (10°F)	20 °C (36 °F)
Water Flow Rate	1.9 l/s (29.4 USgpm)	0.43 l/s (7.5 USgpm)

Figure 6.6 | *Hydronic Coil Rated in Heating and Cooling*

Additionally, the terminal unit control valves must reverse their action[2]. This can be accomplished either by a local sensor that detects whether the supply water is hot or cold or a global command from the BAS to the terminal unit controllers.

Using the same components for both heating and cooling creates performance issues. The components can only be optimized for either cooling mode or heating mode, therefore compromises are necessary.

2-pipe changeover systems are optimized for cooling. When switched to heating mode, system performance is suboptimal. Figure 6.6 shows a 4 row, 12 fin per inch coil selected to meet the cooling load. The coil is oversized for heating with 54 °C (130 °F) hot water so the flow is reduced by the AHU

[2] Chilled water valves open more as the supply air temperature rises while hot water valves close as the supply air temperature rises. Since 2 pipe systems use the same valve and coil, the valve action must be reversed

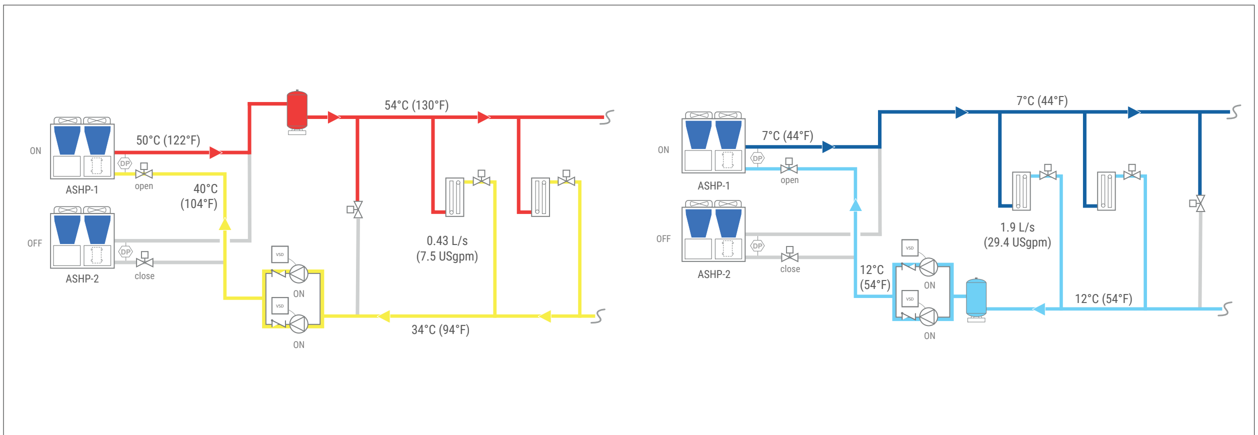

Figure 6.5 | *2-Pipe Change Over System in Heating and Cooling*

control valve from 1.9 l/s (29.4 USgpm) to 0.43 l/s (7.5 USgpm) and the delta T goes from 5.5 °C (10 °F) to 20 °C (36 °F).

The overall effect on the secondary side of the hydronics is a very low flow rate. This will happen with a boiler-chiller system or an ASHP system. The ASHP system is further complicated by the loss of heating capacity due to ambient (will require more units to operate to deliver enough capacity) and the minimum flow requirements of the ASHPs. The system will find a balance for both flow and capacity, but the exact parameters are hard to estimate.

Consider the following when designing 2-pipe changeover systems.

» Use the lowest design hot water supply temperature possible. If there are just 2-pipe changeover AHUs, consider using a hot water supply temperature around 5.5 °C (10 °F) warmer than the required hot air supply temperature for coils picked in cooling. If there are hydronic parameter heating or reheat coils, then keep the hot water supply temperature as low as possible while meeting the heating requirements.

» Sizing the decoupler for a primary secondary system with ASHPs requires special attention. It is quite possible to have multiple ASHPs operating to produce enough heating at low ambient. These units will require their primary flow. At the same time, the secondary flow may be greatly reduced if the coils are optimized for cooling. The result may be a large amount of flow through the decoupler. This can be minimized with variable flow systems.

Water Temperature and Flow Ranges Summary

» Operate ASHPs at typical chilled water temperatures and ranges as chillers.

» Since the ASHP serves as both the heating and cooling unit, the operating flow rates and delta Ts need some consideration. This is made more challenging when considering the ASHP heating capacity changes with ambient temperature.

» ASHP hot water supply temperatures will be close to condensing boilers 49-54 °C (120-130 °F)

» Heating temperature range of 5.5 °C (10 °F) should be possible throughout the ASHP operating range.

- Larger delta Ts may be possible at high winter ambient temperatures.
- Larger delta T design may be accomplished in the hydronic design

» It is difficult to predict the actual heating operating parameters in a 2-pipe changeover system as the equipment is optimized for cooling. It is important to use as low a supply water temperature as possible.

7 Primary-Secondary Systems

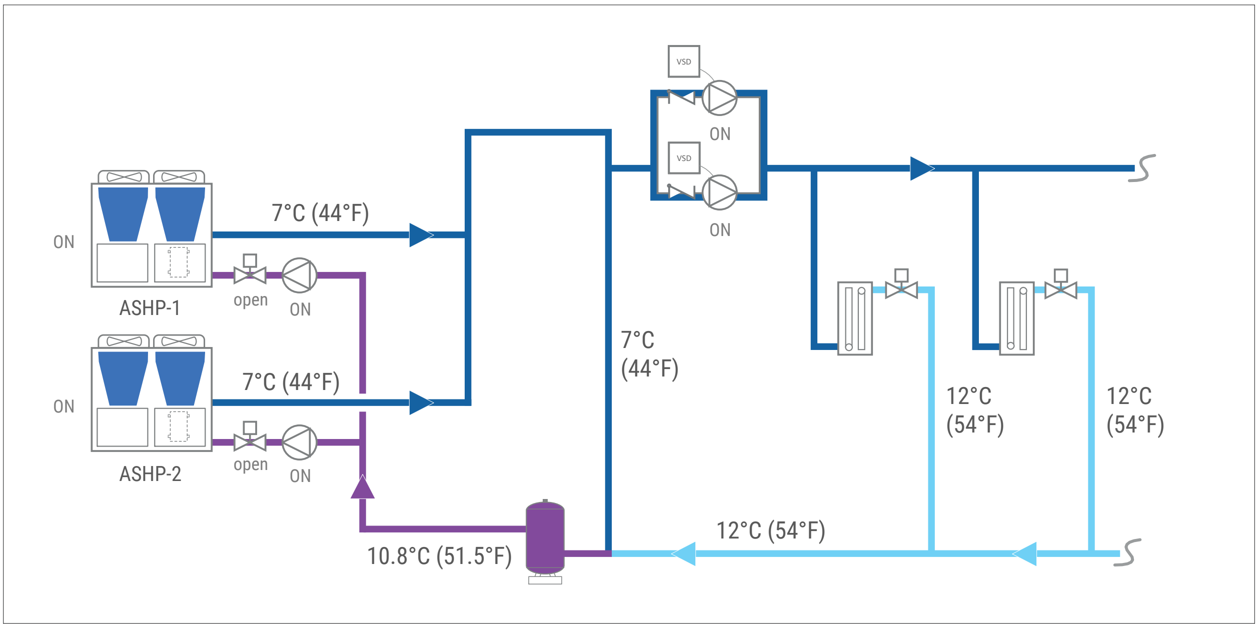

Figure 7.1 | *Basic Primary-Secondary Hydronic System*

Introduction

To reduce pump energy use, most modern hydronic solutions are variable flow. ASHRAE Standard 90.1 requires variable flow solutions when the pump size exceeds 7.46 kW (10 HP). Variable flow systems use 2-way control valves at the terminal units (air handling units, fancoils, chilled beams etc.). As the heating or cooling requirement decreases at a terminal unit, the valve closes, and flow is reduced. In an ideal system the water delta T would remain constant, and the flow rate would be proportional to the demand. For example, an air handling unit with a design cooling capacity of 35 kW (10 tons) and a water delta T of 6 °C (10 °F) would require 0.06 l/s (24 gpm). At 50% load, the air handling unit would consume 0.03 l/s (12 gpm) of chilled water but maintain 6 °C (10 °F).

As buildings operate at part load most of the time, the pump energy savings from reduced water flow is significant. The design variable flow rate is based on the design load and not the connected load. If the building diversity is 80% then the pump size and piping system will be 20% smaller than if the system had constant flow design (using three-way control valves and the plant flow rate being the sum of the connected loads). This represents a significant capital cost savings. The challenge with variable flow systems is that they are more complex to design and operate.

This chapter will cover primary-secondary plant design including piping and controls details and the impact of Low Delta T on system performance.

Primary-Secondary Basics

Traditionally, varying the water flow through a heat pump/chiller was not recommended. Without modern digital controls and algorithms, it was difficult to maintain stable unit control. The primary-secondary system maintains constant flow through the chiller or heat pump when they are operating while using variable flow throughout the building. The temperature range is established for the building and both terminal units and heat pumps/chillers are selected using this range.

Figure 7.1 shows a basic primary-secondary system. The primary pumps are constant-flow and sized for the design flow rate of the heat pump/chillers they serve. The pumps can be dedicated to a specific unit or headered to allow pumps and heat pump/chiller to be individually selected.

The secondary pumps are sized to meet the design flow rated based on the design cooling and/or heating load (not connected load) and the system temperature range. Secondary pumps are variable flow. The terminal units vary the flow rate to meet the load while maintaining the system temperature range using two way valves.

Decoupler sizing

The decoupler is a pipe that separates the primary loop from the secondary loop. Excess primary flow not required in the building passes through the decoupler from supply to return and mixes with the return water. The operating heat pump/chillers see a constant water flow but the return water temperature varies according to the load.

A properly designed decoupler will provide appropriate flow at a reasonable water pressure drop while avoiding unintended mixing of supply and return water.

For chilled water systems, the decoupler pipe size should be based on the primary flow of the largest chiller. The logic is there is almost no cooling load in the building, but the largest chiller is started. The secondary flow will be minimal and most of the primary flow will need to pass through the decoupler. The pipe size should be smaller than the main lines.

For heating, the decoupler size will likely need to be larger. The primary pump flow rate is based on the cooling capacity of the ASHPs. Since ASHP heating capacity is typically less than cooling capacity (depending on the design heating ambient temperature), ASHPs stage on faster in heating mode than in cooling mode. This in turn requires the primary pumps to operate and the extra flow needs to pass through the decoupler.

The worst case occurs when the last ASHP just starts in heating. The primary flow will be the sum of all the primary pumps. The secondary flow will be based on (N-1 ASHP) X ASHP flowrate at the design heating capacity that maintains the system Delta T. The decoupler should be sized for the difference between primary flow and secondary flow.

For Example: A System has three 350 kW (100 ton) ASHPs and three 15.1 l/s (240 USgpm) primary pumps. The system design Delta T is 5.5°C (10°F). At the winter design ambient the ASHP heating capacity is 70% of the cooling capacity (245 kW (835 kBtu/h)).

When the building heating load reaches 491 kW (1681 kBtu/h) the third ASHP will start. All three

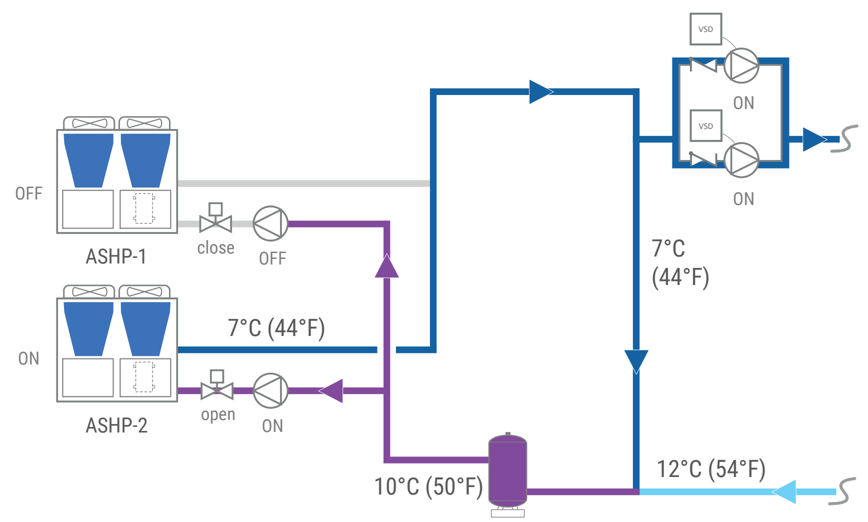

Figure 7.2 | *Decoupler*

primary pumps will produce 45.3 l/s (720 USgpm) flow. The secondary flow is based on 5.5 °C (10 °F) delta T and the 491 kW (1681 kBtu/h) heating load so the flow will be 21.1 l/s (336 USgpm). The decoupler heating design flow rate is 45.3 l/s (720 USgpm) - 21.1 l/s (336 USgpm) = 24.2 l/s (384 USgpm).

Note that the heating decoupler flow rate is greater than the cooling decoupler flow rate (15.1 l/s (240 USgpm)).

In addition to decoupler pipe sizing the following criteria should be met:

- The decoupler pipe size should be based on the primary flow of the largest unit in the plant. The pipe size should be smaller than the main lines
- The pressure drop should not exceed 4.5 kPa (1.5 ft. w.c.)
- The connections should be as shown in Figure 7.2 to avoid unexpected mixing
- The decoupler length should be approximately 3 pipe diameters
- There should be at least 10 pipe diameters to the first ASHP

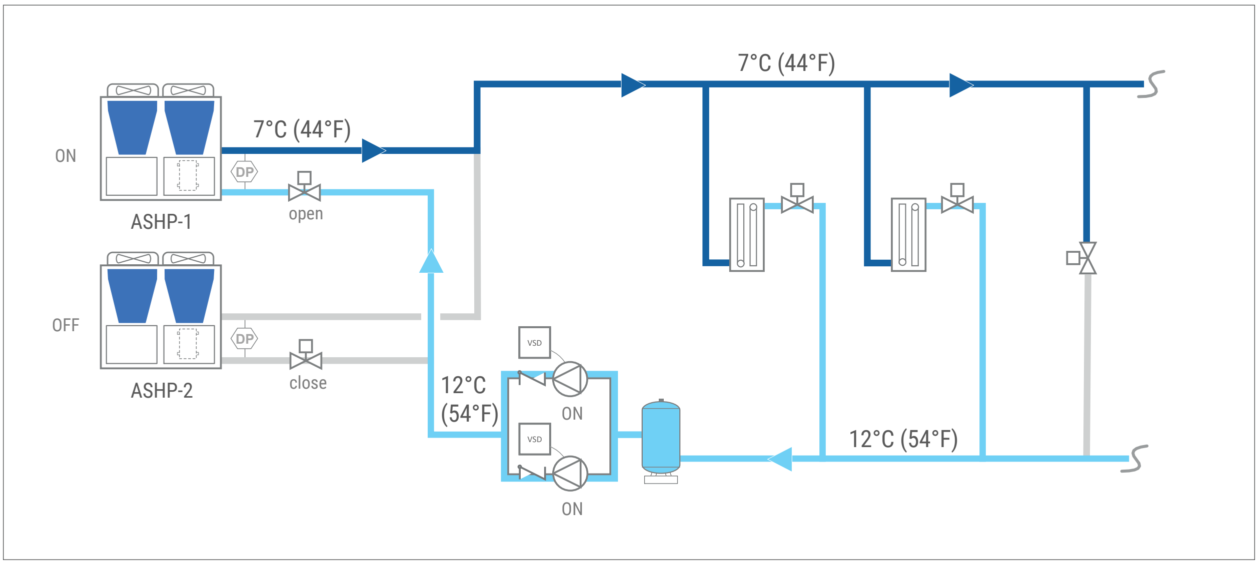

Figure 7.3 | *2-Pipe Changeover Primary-Secondary Heating and Cooling Plant*

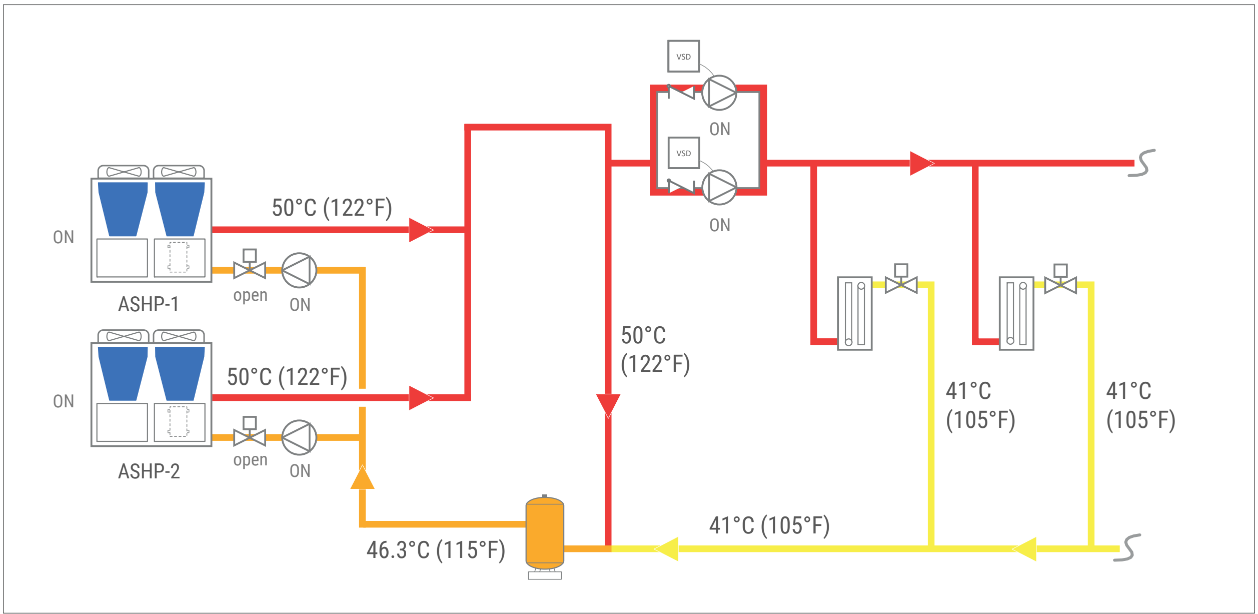

Figure 7.4 | *2-Pipe Changeover Primary-Secondary Heating Plant*

Using ASHPs in 2-pipe systems in heating mode are even more problematic. It is possible for multiple ASHPs to be operating along with the primary pumps. At the same time the secondary flow may be very low with a large delta T. The imbalance in flows between the primary and secondary loops results in a very high flow through the decoupler. It may be necessary for the decoupler to be sized for most or all the design flow rate.

2-Pipe Changeover Primary-Secondary Design Considerations

Figure 7.3 and 7.4 show a 2-pipe changeover primary-secondary system for mild or medium winter conditions. Referring to the Seattle example from chapter 5, this system will use two 200 kW (57 ton) ASHPs to meet the winter heating requirement without supplemental heat.

ASHP Sizing

ASHP sizing is covered in more detail in Chapter 5. If the building load is summer dominant, the ASHPs are sized to meet the summer design condition. If the building load is winter dominant, the ASHPs are sized to meet the winter load (assuming it is a mild winter location and ambient condition will not exceed ASHP operating envelope). Where the ASHP sizing is based on winter load, it is strongly recommended that at least two units are used to better match unit building capacity at low heating loads and the summer load requirements with adequate capacity control. Unless there is a compelling need, all ASHPs should be the same size. It makes system design, balancing, commissioning and operation much easier.

System Temperature Range and Pump Sizing

System supply water temperatures and ranges are discussed in Chapter 6.

For 2-pipe changeover systems the secondary pumps should be sized using the highest flow rate of the cooling or heating design load (usually it is dictated by cooling), system temperature range and the secondary loop head loss. Secondary pumps are variable flow (VFDs). Multiple pumps are recommended for a wider flow range and some redundancy. N+1 arrangements can be used to improve redundancy. All pumps should be the same size.

Primary pumps should be sized for the flowrate of the ASHP they serve and the head loss in the primary loop. Primary pumps are constant flow so every time an ASHP is enabled, a primary pump is started. Generally, the sum of the primary pump flows should match the secondary pump design flow rate. In principle it is acceptable for the primary flow to be greater than the secondary flow. However, it is not acceptable for the primary flow to be less than the secondary flow. At full load conditions, the secondary flow will exceed the primary flow rate resulting in water flowing backward through the decoupler and mixing warm return water with chilled water (in chilled water mode).

Winter Operation Considerations

The challenges of 2-pipe changeover systems are discussed in Chapter 6. Primary-secondary systems add to the challenge because the primary flow is fixed (constant speed pumps) and the ASHP cooling capacity will not match the heating capacity. The fixed flow means the heating and cooling temperature ranges will be different.

Figure 7.4 shows a 2-pipe change over primary-secondary system operating in heating. The fixed flow primary pumps matched with ASHP heating-cooling capacity difference leads to hot water temperature range dropping to 3.7 °C (6.6 °F). The lower capacity of the ASHPs requires multiple units to operate to meet the load. This requires more primary flow which increases pump work in the system. It also means there is a lot more primary flow than secondary flow and the decoupler must be sized to manage the flow. (See Decoupler sizing.)

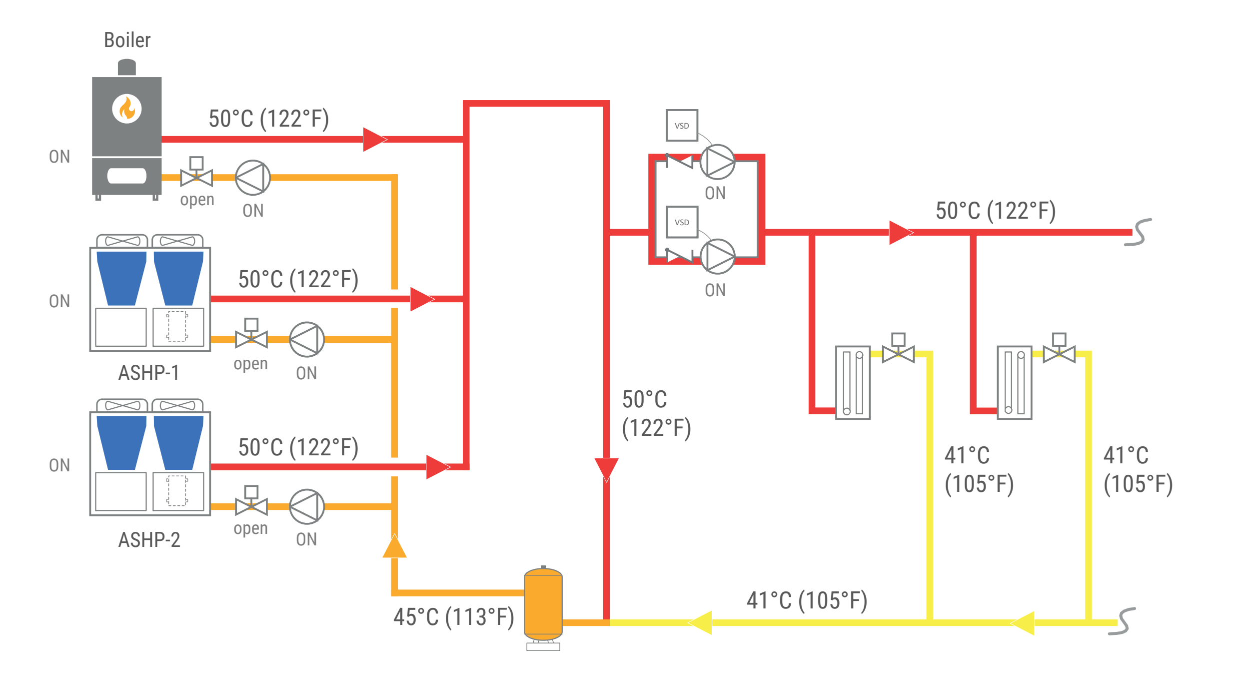

Figure 7.5 | *Primary-Secondary System with Simultaneous Parallel Auxiliary Heat*

2-Pipe Changeover Primary-Secondary with Auxiliary Heat Design Considerations

Figure 7.5 shows a 2-pipe changeover primary-secondary system where winter design conditions require auxiliary heat. In some applications, the auxiliary heat will supplement the heat pump capacity with both operating simultaneously. For very cold climates, the auxiliary heat will meet the entire heating demand and the heat pumps will be off. In this case the heat sources are sequential (it is one heat source or the other).

ASHP and Boiler Sizing

Where the auxiliary heat works simultaneously with the ASHPs, the boiler is sized for the difference between the winter design load and the heating capacity of the ASHPs at the winter design condition. Where the boilers work sequentially with the ASHPs, the boilers are sized for the full design heating load.

Boiler Placement Considerations

The boilers can be piped in parallel with the ASHPs or in series. Figure 7.5 shows the boilers in parallel. In this application the boilers and ASHPs typically have the same hot water supply temperature. The sum of primary flows though the boilers and ASHPS must be equal to or greater than the secondary flow to ensure correct flow through the decoupler. The primary pump serving the boiler should match the required flow rate for the boiler. If this is different than the ASHP flow rate, then either dedicate the primary pump to the unit it serves or header pumps specifically for the boilers and use a second set of headered pumps for the ASHPs.

When the auxiliary heat is simultaneous with the ASHPs, the heating load is first met with the ASHPs. Once all the ASHPs are operating and the load cannot be met, the boilers are staged on (see Figure 7.5).

When the auxiliary heat is used sequentially, as shown in Figure 7.6, the ASHPs operate until the changeover temperature is reached. In most cases the bivalence point is the most convenient time to change over. If the ASHPs operate below the bivalence point, then the boilers and ASHPs will need to operate simultaneously to meet the build heating load. This may be a good option if an electric boiler is being used. In which case, supplementing the electric boiler with the ASHPs as much as possible will improve the energy and operating cost.

The need to operate the ASHPs below the bivalence point may also be justified if there are significant operating (BIN) hours near but below the bivalence point. In this case using the ASHPs will help reduce the carbon footprint and energy use. However, the design will need to be carefully considered. It is recommended that the BIN hours below the bivalence point be reviewed, and the potential additional savings only be pursued if they are significant.

Figure 7.7 shows a 2-pipe primary-secondary system where the boiler is piped in series with the heatpumps. The series arrangement makes it possible to increase the hot water delta T but this is not recommended in a 2-pipe system (see Chapter 6: Water Temperature and Flow Ranges).

In an application where the ASHPs and the boiler operate simultaneously, the ASHPS are first used to meet the heating load. When the ASHPs can no longer maintain the supply water temperature (cannot meet the heating load), the boiler is enabled.

Using a boiler in a series configuration for applications where the boiler and the ASHPs are operated sequentially is problematic. Once the changeover point is reached the ASHPs will be switched off along with the primary pumps. Some means is required to ensure the primary flow meets or exceeds the secondary flow which is difficult to do when the boiler is piped in series.

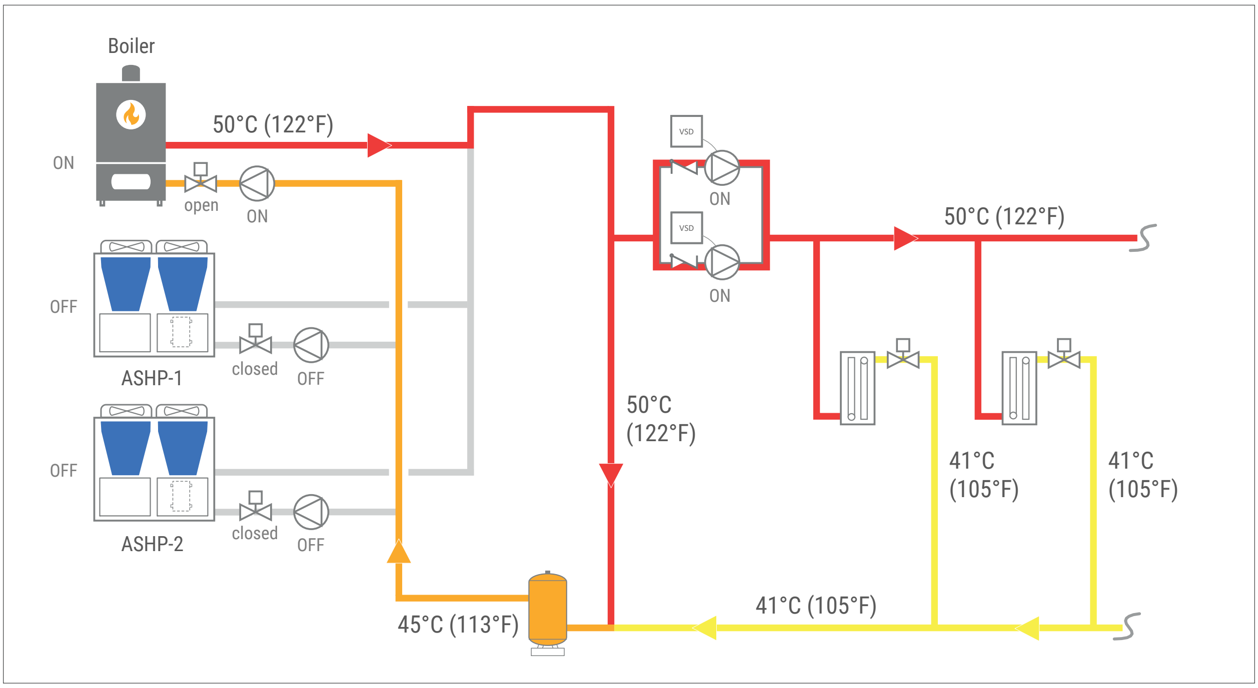

Figure 7.6 | *Primary-Secondary System with Sequential Parallel Aux. Heat*

4-Pipe Changeover Primary-Secondary Design Considerations

A 4-pipe hydronic system provides both hot and chilled water to the terminal units at the same time allowing for heating or cooling at the zone level as required. This improves occupant satisfaction but increases the capital cost for the additional piping. Integrating the ASHPs into a 4-pipe system is more complex than a 2-pipe system. To illustrate this, the following examples are given.

4-Pipe System with ASHPs — Cooling

Figure 7.8 shows the 4-pipe system using ASHPs in cooling only mode. Both ASHPs have been signaled to operate as chillers by the BAS (Building Automation System) and the necessary changeover valves have been arranged to connect both units to the chilled water loop.

4-Pipe System with ASHPs — Heating and Cooling

Figure 7.9 shows simultaneous heating and cooling loads. ASHP 1 operates as a chiller and is valved to connect it to the chilled water loop. ASHP 2 operates as a heat pump and is valved to connect it to the hot water loop. The BAS tells the ASHPs which mode to operate in. As long as the load in either the hot water or chilled water loop does not exceed the capacity of their respective ASHP, this system will meet the building requirements. If for example, the cooling load increased beyond the capacity of ASHP 1, then a choice must be made if ASHP 2 should continue as a heating unit or be switched to cooling mode.

Switching an ASHP back and forth between loops cannot be done quickly or repeatedly.

If the system shown in Figure 7.9 had 4 ASHPs rather than 2, it would be possible to more accurately assign ASHPs to either heating or cooling to better match the simultaneous heating and cooling load. Alternatively, an additional ASHP (go from 2 units to 3) would allow the simultaneous loads to be properly matched. This approach will increase the capital cost of the project.

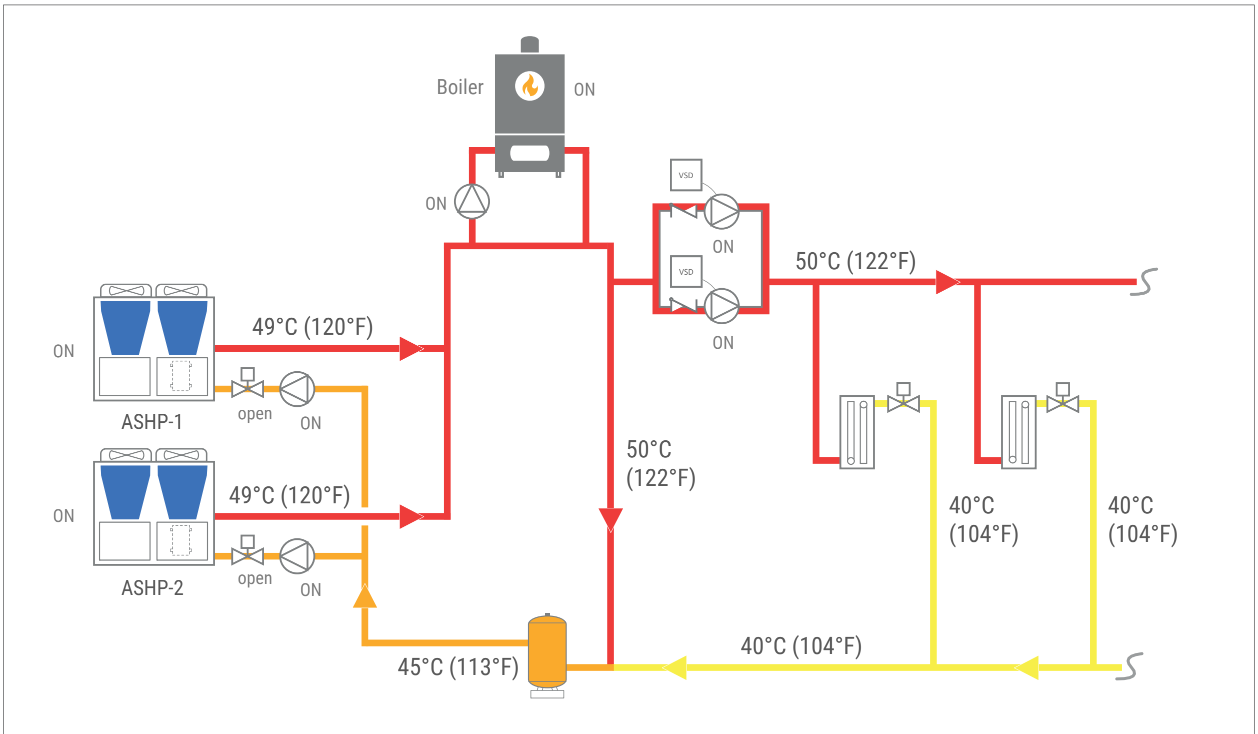

Figure 7.7 | *Primary-Secondary System with Aux. Heat in Series*

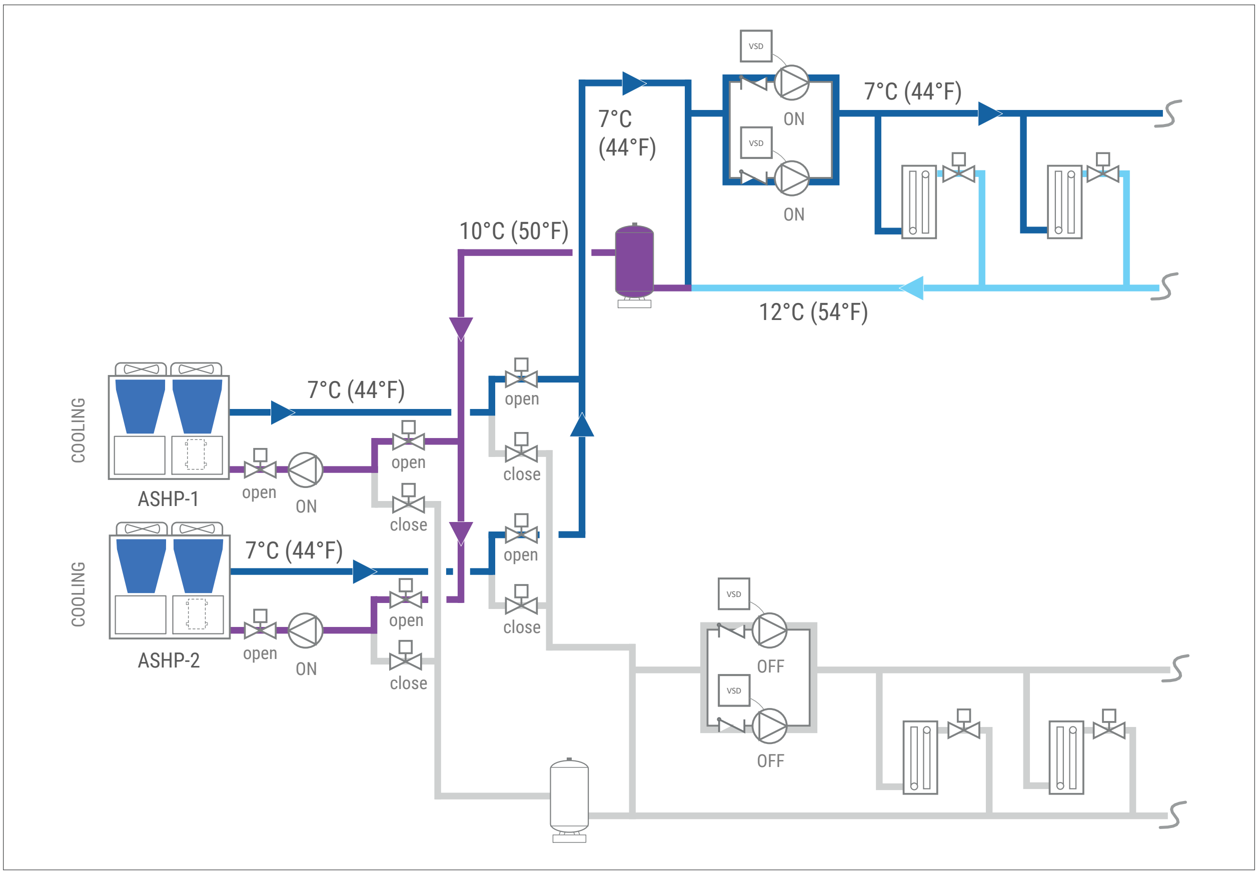

Figure 7.8 | *4-Pipe Primary-Secondary System with 2-Pipe ASHPs - Cooling Only*

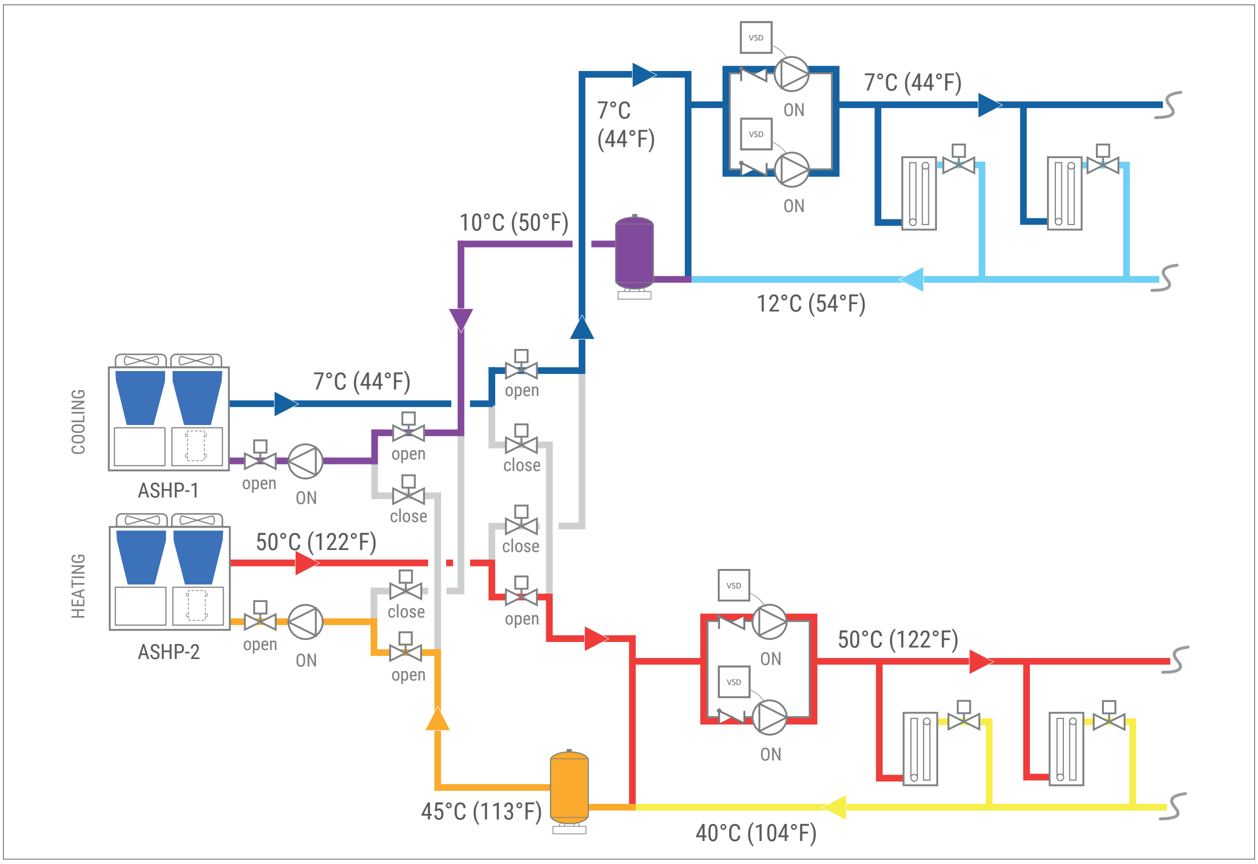

Figure 7.9 | *4-Pipe Primary-Secondary System with ASHPs - Cooling and Heating*

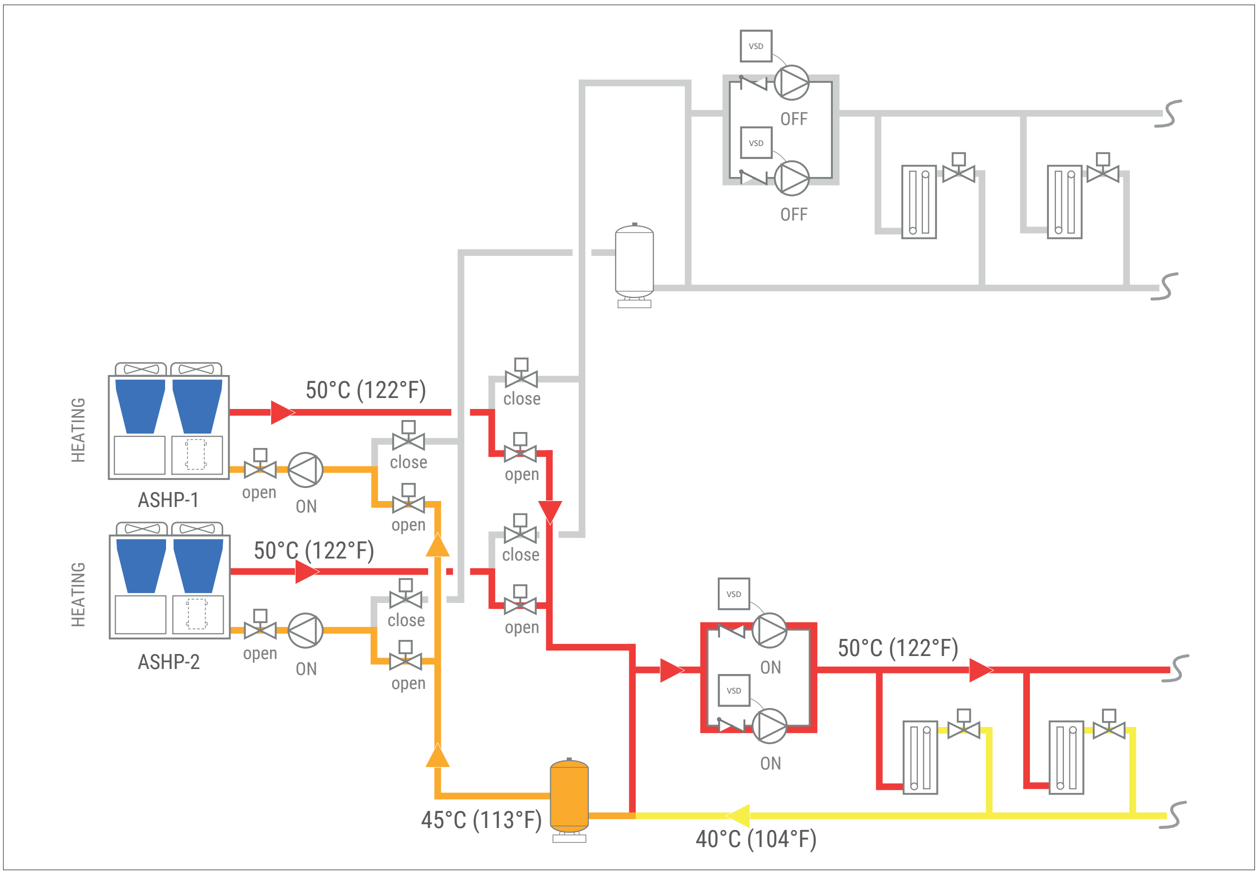

Figure 7.10 | *4-Pipe Primary-Secondary System with ASHPs - Heating Only*

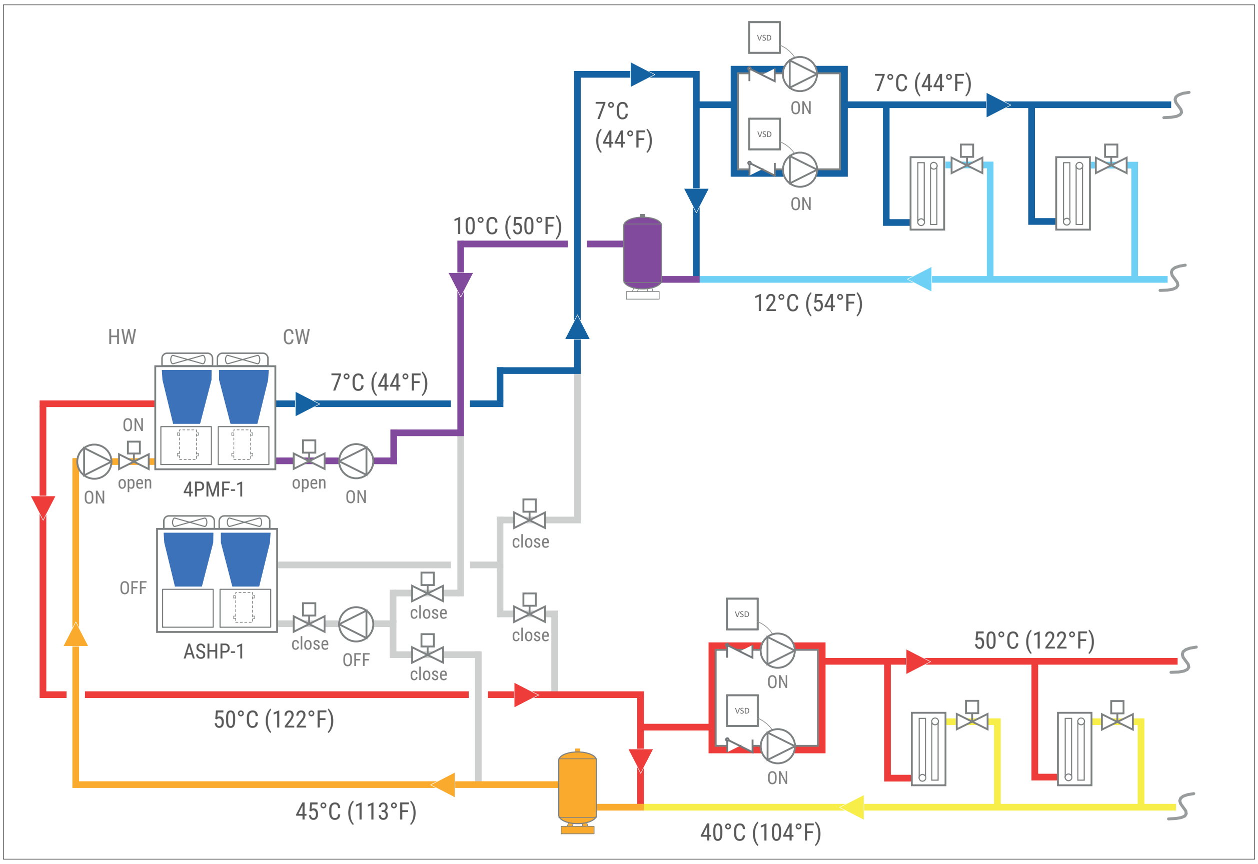

Figure 7.11 | *4-Pipe Primary-Secondary System with 2P ASHP and 4PMF- Cooling and Heating*

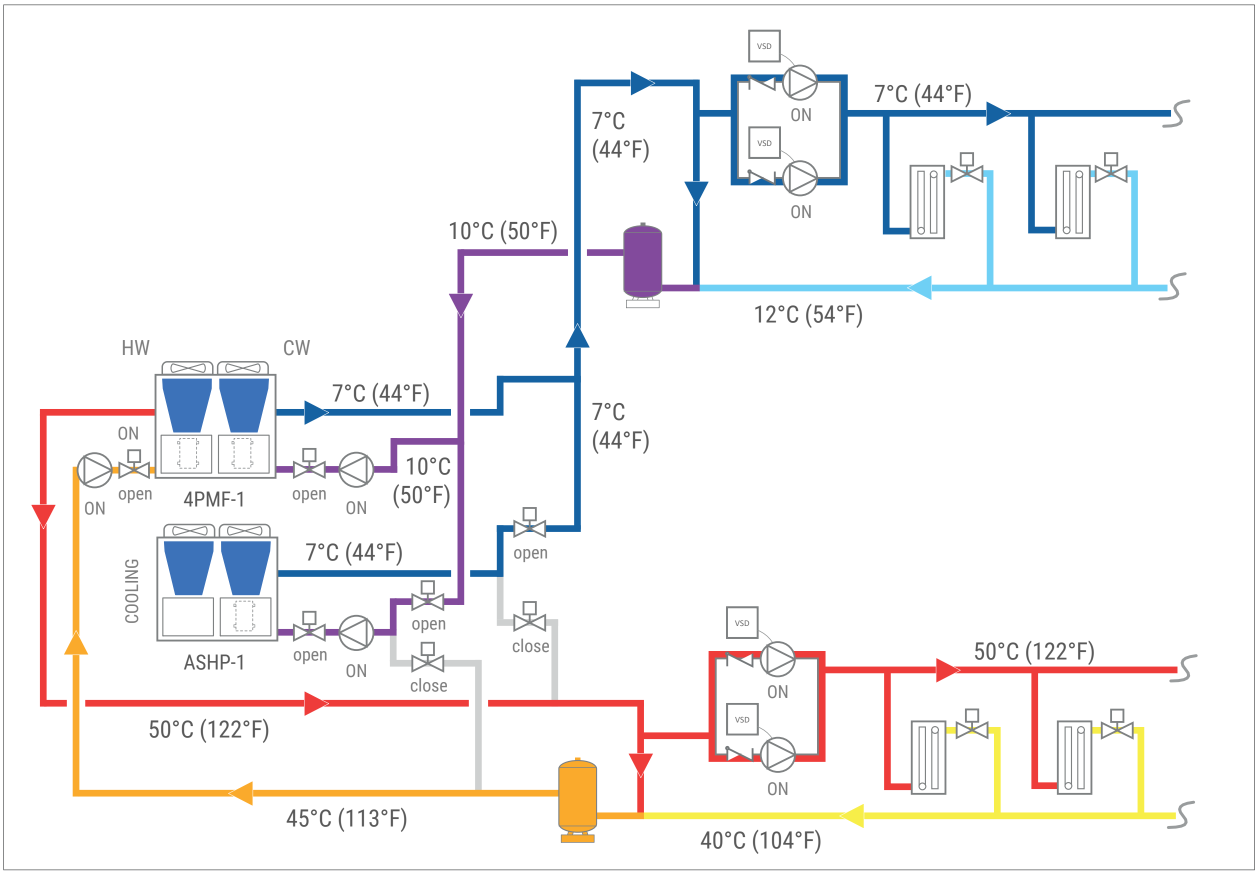

Figure 7.12 | *4-Pipe Primary-Secondary System with ASHP and 4PMF - Cooling Dominant*

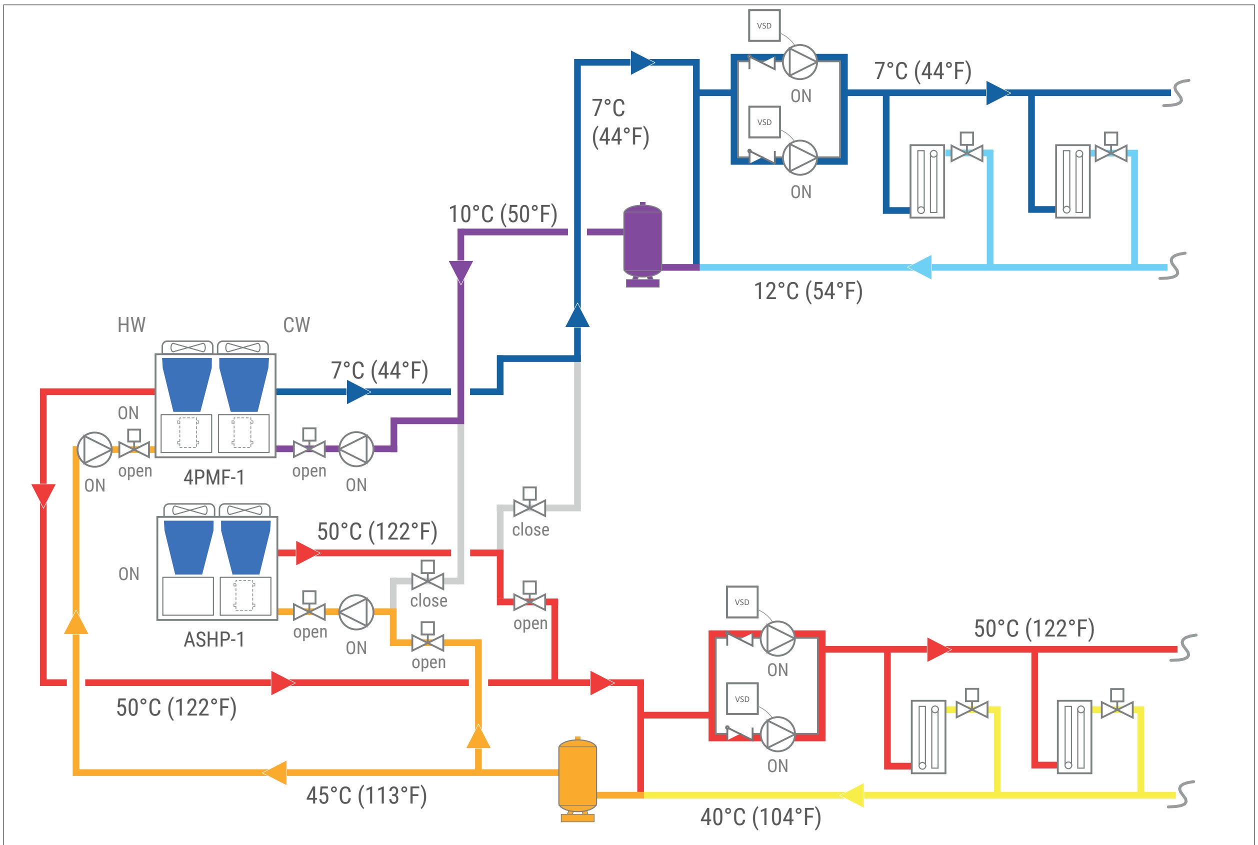

Figure 7.13 | *4-Pipe Primary-Secondary System with ASHP and 4PMF - Heating Dominant*

4-Pipe System with ASHPs — Heating

Figure 7.10 shows heating mode. The ASHPS are valved into the hot water loop and the BAS instructs the units to operate in heating.

4-Pipe System with ASHP and 4PMF — Cooling and Heating

Figure 7.11 shows the same plant as before with one of the ASHPs replaced with a 4-pipe multifunctional Unit (4PMF) (see Chapter 4: 4-Pipe Multifunctional Units.) The 4PMF is connected to both the hot water and chilled water loops. In Figure 7.11, the building is experiencing simultaneous heating and cooling loads. The 4PMF unit provides both hot and chilled water. Moreover, the unit is moving the heat from the chilled water loop to the hot water loop offering significant energy savings. In combination systems with 4PMF and ASHP units, the 4PMF unit is first used to meet the cooling and heating loads. When the load exceeds the 4PMF unit capacity, the ASHPs are connected to the appropriate loop to supply additional capacity.

4-Pipe System with ASHP and 4PMF — Cooling Dominant

Figure 7.12 shows the plant with significant cooling and some heating. The ASHP has been enabled as a chiller and valved to the chilled water loop. The plant can deliver up to 100% cooling and 50% heating.

4-Pipe System with ASHP and 4PMF — Heating Dominant

Figure 7.13 shows the plant with the ASHP enabled in heating and valved to the hot water loop. The plant can deliver up to 100% heating and 50% cooling.

4-Pipe System with 4PMF — Heating and Cooling

Figure 7.14 shows the plant with all 4-pipe multifunctional units. The units are connected to both the hot and chilled water loops. The arrangement can provide up to 100% cooling and 100% heating

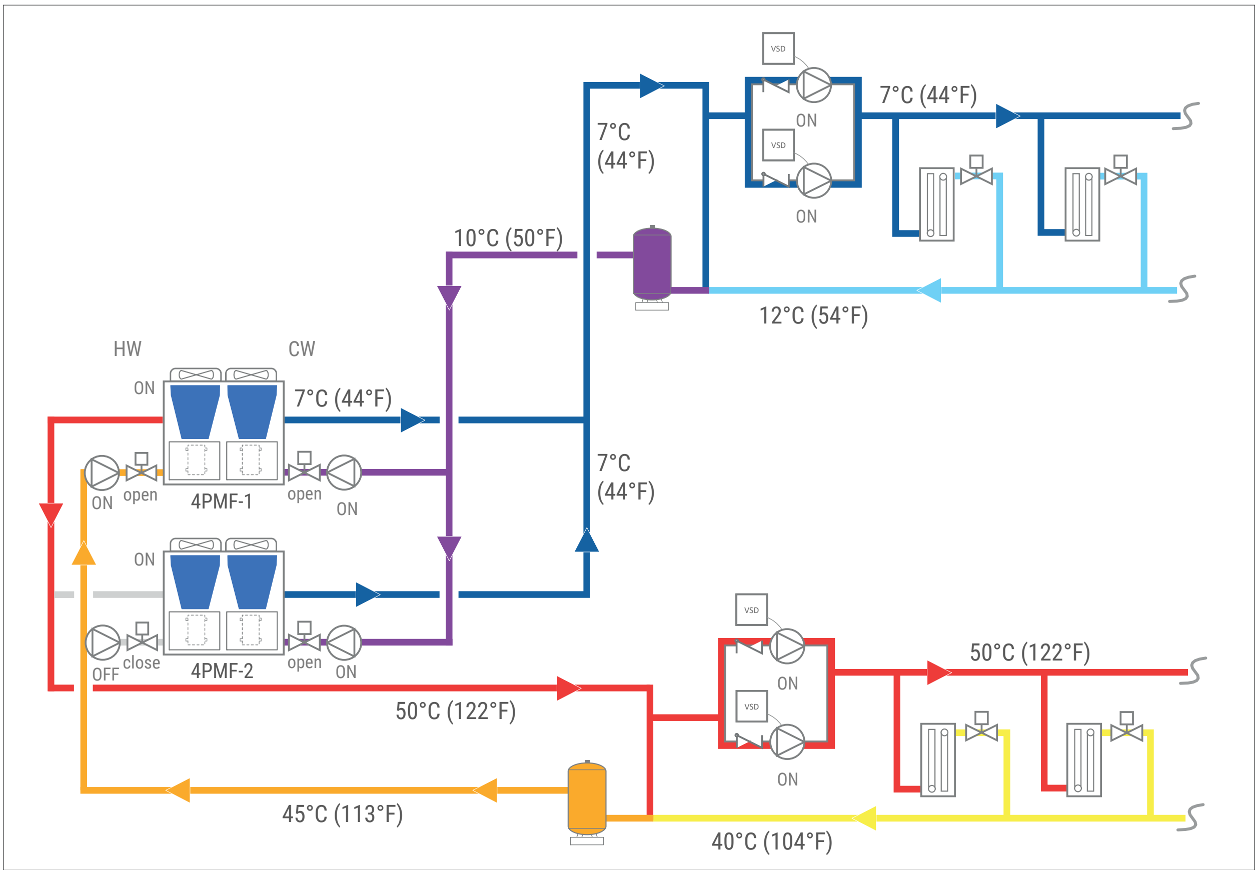

Figure 7.14 | *4-Pipe Primary-Secondary System with 4PMF - Heating and Cooling*

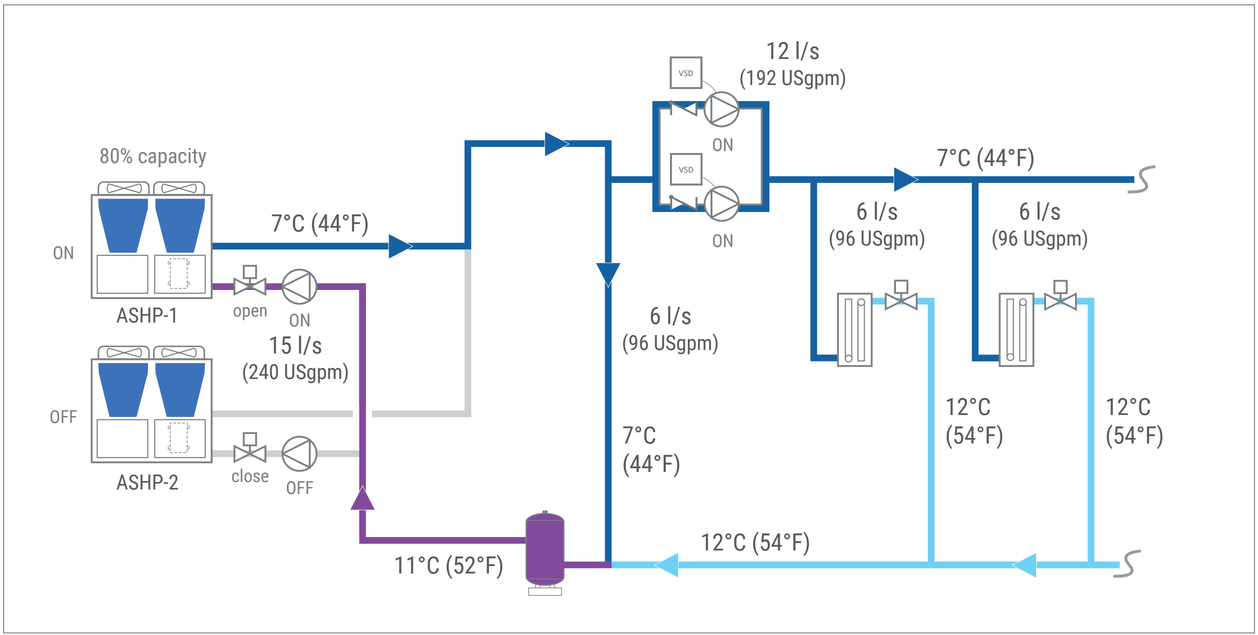

Figure 7.15 | *Primary-Secondary System - Expected Operation*

at any time. There is no need to switch any units from one loop to the other. For smaller systems this is a straightforward design and easy to operate.

For larger systems with more units, it is likely that in summer months, some of the 4PMF units will operate entirely in cooling for extended periods of time. This will require some controls to allow the units to switch to heating for oil management. It means the hot water primary pump will need to be managed by the 4PMF unit.

4-Pipe Systems with Auxiliary Heat

Adding auxiliary heat to 4-pipe systems is similar to 2-pipe systems. If a boiler is required in a 4-pipe ASHP system, the boiler can be used during simultaneous heating and cooling so the ASHPs can stay in chilled water mode. While this may increase the carbon footprint, it will improve temperature control and occupant satisfaction.

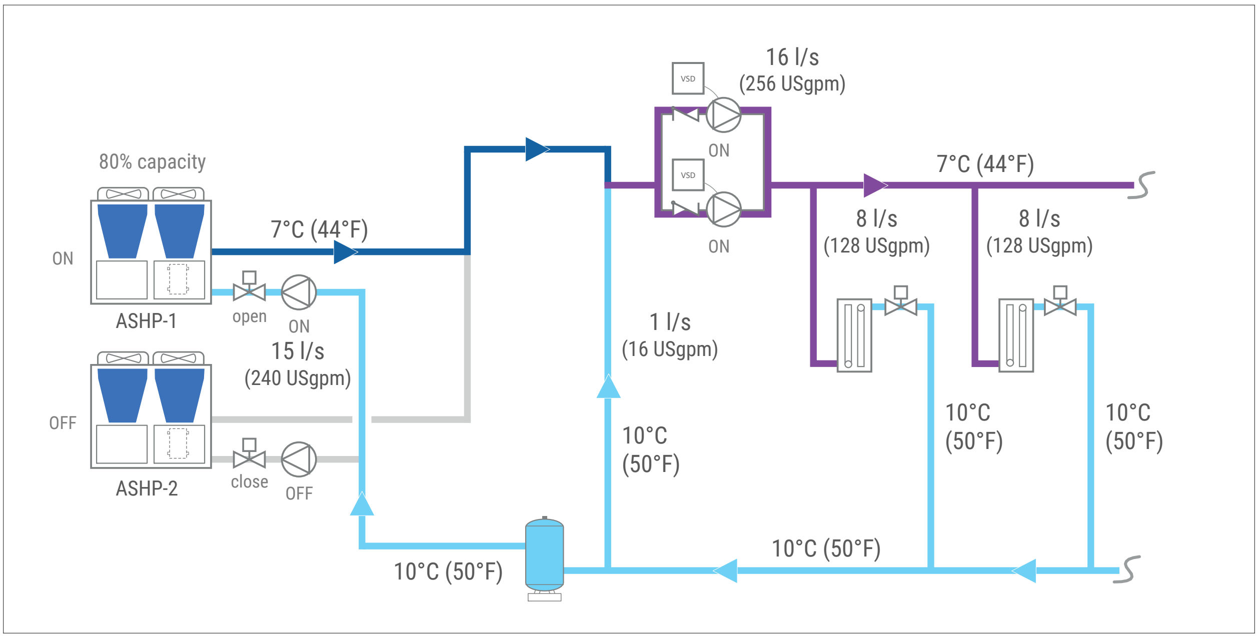

Figure 7.16 | *Primary-Secondary System - Low Delta T*

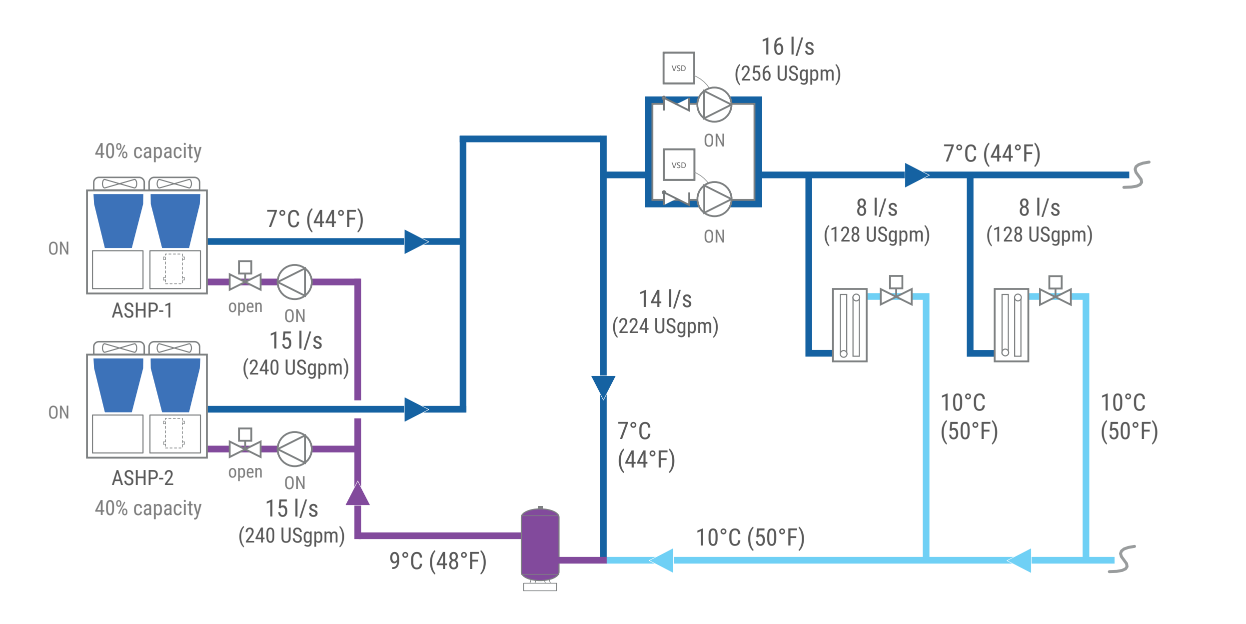

Figure 7.17 | *Primary-Secondary System - Low Delta T Corrected*

Impact of Low Delta T

Primary-secondary systems work on the premise that the secondary water temperature range will remain constant, and the flow will change as the building load varies. In practical applications this is rarely true and the impact on primary-secondary system performance can be significant.

While many things can cause Low Delta T, consider the coils in the terminal units are allowed to become dirty. The dirt will lower the coil heat transfer effectiveness. The terminal unit controller monitoring the supply air temperature, will open the chilled/hot water valve to maintain setpoint. In most cases this will work, and the occupants will not be affected.

However, assume all the terminal unit controls valves double their flow and half their temperature range to meet the zone load requirement. This will cause the secondary pumps to double their flow and half the secondary loop delta T.

In Figure 7.16 a 350 kW (100 Ton) ASHP is operating at 80% capacity. Its primary pump operates at 15 l/s (240 USgpm). If the load in the building is 280 kW (80 tons) of cooling and the range is 6 °C (10 °F) then the secondary flow should be 12 l/s (192 USgpm).

Figure 7.16 shows a Low Delta T situation where the secondary flow has increased to 16 l/s (256 USgpm). On the primary side, one 350 kW (100 ton) ASHP is operating. Its primary pump is constant flow delivering 15 l/s (240 USgpm). While one ASHP can meet the building load, the primary flow is less than the secondary flow. The secondary pump will draw return water backwards though the decoupler mixing 1 l/s (16 USgpm) of 10 °C (50 °F) return water with the 15 l/s (240 USgpm) primary flow. The warmer chilled water flowing to the terminal units will cause the terminal unit controls to open the valves further to maintain setpoint and the whole system operation will spiral into an unworkable situation.

The only option is to start a second ASHP to increase the primary flow above the secondary flow as shown in Figure 7.17. Now two primary pumps will operate delivering 30 l/s (480 USgpm) and two ASHPs will operate at 40% capacity. The system operation is now stable but not energy efficient.

The golden rule for primary-secondary systems is ***the primary flow must always meet or exceed the secondary flow regardless of ASHP loading.***

Other system designs where the primary flow can be variable are less impacted by Low Delta T and will be discussed in the next chapters.

Primary-Secondary System Summary

Primary-secondary systems are traditional systems that support both the need for energy savings from variable flow throughout the building while providing constant flow through the heating and cooling units for reliable operation. With modern digital controls, the need for constant flow through the heating and cooling units has diminished greatly. Primary-Secondary systems are vulnerable to the effects of Low Delta T which can lead to poor energy performance (multiple heating/cooling units plus their primary pumps operating when the load does not require it) and in some cases the system may not function at all.

Using fixed flow primary pumps with ASHPs introduces some challenges. The heating capacity rarely matches the cooling capacity so with fixed primary flow, the unit delta T decreases in heating. It also means that more ASHPs will need to operate to provide enough capacity which also increases the primary flow. While all this can be managed with careful design, it makes variable primary flow systems attractive when using ASHPs.

8 Variable Primary Flow System

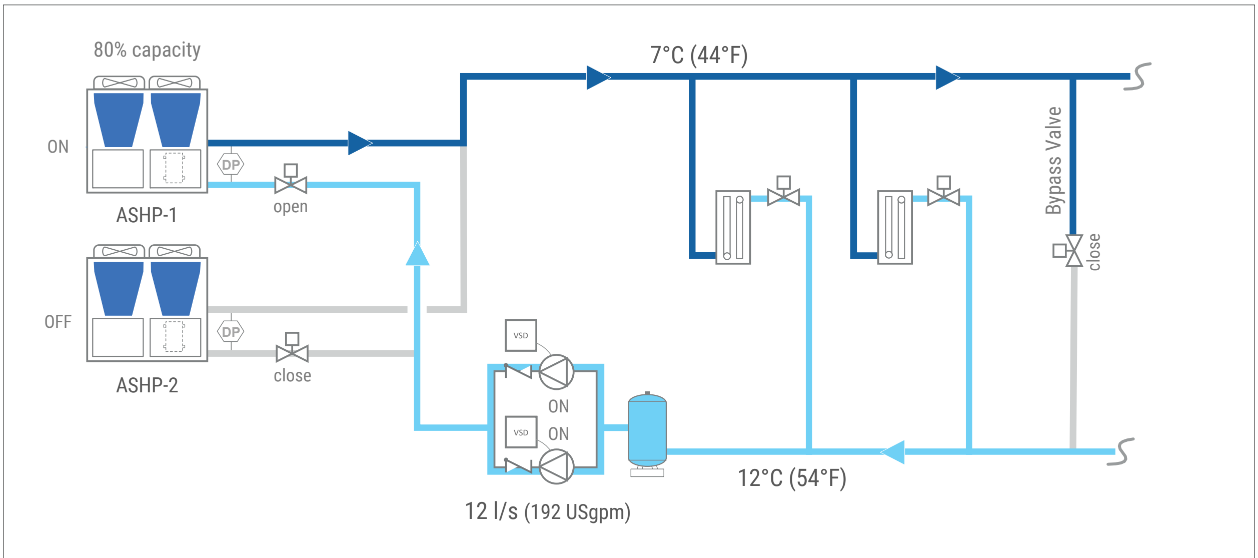

Figure 8.1 | *Basic Variable Primary Flow (VPF) System*

Introduction

Figure 8.1 shows a Variable Primary Flow (VPF) system operating in cooling. Comparing the VPF system to primary-secondary it can be seen there is only one set of (primary) pumps that distribute water through the central plant as well as the entire hydronic system. The pumps are variable flow and sized for the design flow and pressure loss of the entire system. As the load varies, the primary flow will vary in proportion meaning the flow through the ASHP units will vary. In a VPF system the decoupler is replaced with a bypass line with a modulating control valve. Also, there are (slow opening) isolation valves and some form of water flow measurement for each ASHP.

In this example there are two 350 kW (100 ton) ASHPs. The system is operating with 280 kW (80 tons) of plant load which is met by operating one ASHP. The second ASHP is off and the isolation valve is closed so all the flow passes through the ASHP that is operating. The flow required for 280 kW (80 tons) is 12 l/s (192 USgpm). The design flow for the ASHPs is 15 l/s (240 USgpm) so the flow has been reduced to 80%.

Varying the flow through a chiller or heat pump is only possible with modern digital controllers that can respond to load changes caused by variations in water flow and/or return water temperature.

All chillers and heat pumps have a minimum flow below which they cannot operate. This is due to laminar flow in the refrigerant to fluid heat exchanger. Once there is laminar flow, the heat transfer is not sufficient to operate the unit properly. A rough rule of thumb is the flow can be reduced to half of design flow. Many factors can affect the minimum flow such as the introduction of glycol, so it is strongly recommended that the equipment manufacturer provides the minimum flow based on the application and the unit selected.

Figure 8.2 shows the same system operating at 88 kW (25 tons) building load. The flow to the building has been reduced to 3.8 L/s (60 USgpm). This flow rate is less than the minimum flowrate for the ASHP. The flow rate is measured at the ASHP inlet. Once the low flow rate is detected, the control system opens and modulates the valve in the bypass line until the minimum flow through the ASHP unit is achieved while also maintaining correct flow to the building.

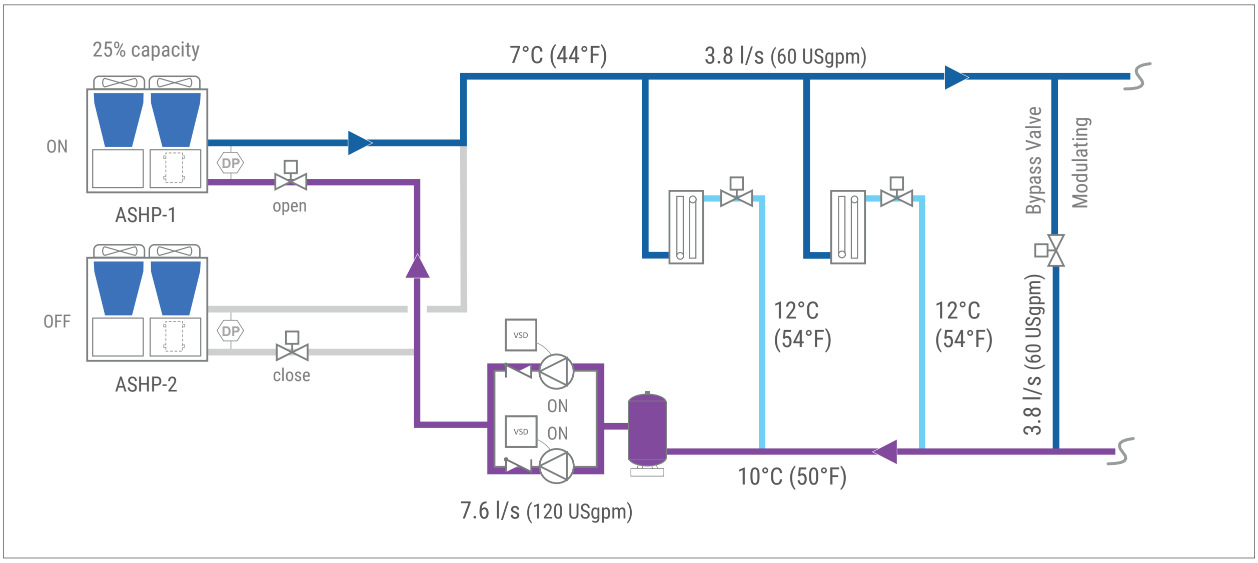

Figure 8.2 | *Basic VPF System operating below Minimum Flow*

Starting Additional ASHPs

As the plant goes from 0 to 100% capacity the first ASHP operating will go from 0 to 100% of its capacity. In the example above, the load on ASHP 1 has increased to 350 kW (100 ton) and 15 L/s (240 USgpm). ASHP 1 is now at 100% capacity. As the building demand increases above 350 kW (100 tons), the second ASHP must be started. The first step is to open the isolation valve on ASHP 2. When this occurs, the current 15 L/s (240 USgpm) will be evenly split between the two ASHPs. ASHP 1 will now be providing 350 kW (100 ton) of cooling to 7.5 L/s (120 USgpm) which will double the temperature range (ΔT). This 'full capacity at half flow' condition persists until the ASHP unit controller recognizes the change and reduces unit output. It can bring the ASHP very close to its low temperature or pressure safety resulting in the unit tripping off.

There are several methods to manage the start of the second ASHP including, slow opening isolation valves, demand limiting the first unit prior to opening the second unit isolation valves and increasing the primary flow before opening the second unit isolating valves. For reliable plant operation, some method of flow/load management for starting additional units must be considered.

VPF System Advantages

There are several advantages in selecting VPF over primary-secondary systems including;

» Capital savings from only one set of pumps

» Overall system pressure drop reduction due to only one set of pump isolation valves, strainers etc. This can reduce the capital cost of the pumps and deliver pump energy savings

» Additional pump energy savings from not having flow through the decoupler. It is true that if the bypass valve is open there will be pump energy cost, but it happens much less of the time than flow through the decoupler.

» An ASHP rarely provides the same heating and cooling capacity. Variable primary flow allows the ASHP delta T to be maintained in both heating and cooling modes by changing the primary flow to match the unit capacity.

» VPF is much more forgiving to Low Delta T issues.

The downside of VPF systems is they can be more difficult to commission and operate.

Design Parameter	Cooling Mode	Heating Mode
Fluid	40% Propylene Glycol	
Ambient Temp. °C (°F)	35 °C (95 °F)	-20 °C (-4 °F)
Supply/Return Water Temp. °C (°F)	6.67/12.2 °C (44-54 °F)	54.4/48.9 °C (130/120 °F)
Capacity Output	430 kW (122 Tons)	266 kW (908 kBtu/h)
Flow Rate l/s (USgpm)	20.4 (324 USgpm)	12.4 L/s (197 USgpm)

Figure 8.3 | *Basic Variable Primary Flow (VPF) System*

Bypass Line Sizing

The bypass line should be sized for the minimum flow of the largest ASHP in the system. The bypass line can be located near the ASHPs or at the end of the hydronic loop but must be located downstream of the pumps so water can be recirculated back to the ASHPs. The bypass control valve should be carefully considered. If the valve hunts, the pump flow may start to fluctuate. In extreme conditions, the pump flow oscillation could cause the ASHP to trip off. The bypass flow control valve should have linear flow curve characteristics as opposed to equal percentage valves to give stable flow control.

ASHP Sizing, System Temperature Range and Pump Sizing

Sizing of the ASHPs is discussed in Chapter 5. Recommended temperature ranges are discussed in Chapter 6. It is very strongly recommended that all the ASHPs be the same model and size. It is extremely difficult to manage flowrates if the units have different minimum flowrates and pressure drop requirements.

Primary pumps are variable flow. It is generally better to have the primary pumps in a remote pump bank than dedicated to a specific unit. One of the advantages of variable primary flow systems is the water flow management and the ASHP capacity management are individually controlled. A remote pump bank can be indoors (as opposed to installed in the ASHP) and gives the option to “over pump” an ASHP during periods when there is a Low Delta T situation. The pump flow will be determined by the building need (typically maintaining delta pressure in the building piping). If there is some issue causing Low Delta T the primary pumps will “over pump” the ASHP assuming the ASHP has capacity to meet the load. This is not possible with primary-secondary systems where the primary pumps are constant flow, matched to the ASHP they serve.

It is important to understand the minimum flow rate for the ASHPs and consider this in the design. It is best to have the equipment manufacturer provide the minimum flow based on the application and the model selected.

The flow range between heating minimum and cooling maximum is usually significant and can have a large impact on pump selection. This is best illustrated in the following example.

Consider pump selection for a VPF 2-pipe changeover system consisting of three 475 kW (135 ton) ASHPs.

Referring to Figure 8.3, the chilled water flow rate is 20.4 (324 USgpm) while the hot water flow rate is 12.4 l/s (197 USgpm) per unit. The total design cooling flow rate is 61.2 L/s (972 USgpm). The total heating flow rate at -20 °C (-4 °F) is 37.2 L/s (591 USgpm), a difference of 40% between design rating points of the plant. This illustrates the difference in heating and cooling capacity of an ASHP and how this ratio is dependent on ambient temperature.

The minimum flow rate of a single unit is 10.2 l/s (162 USgpm). This is a property of the ASHP refrigerant to water heat exchanger, so it is the same in heating or cooling.

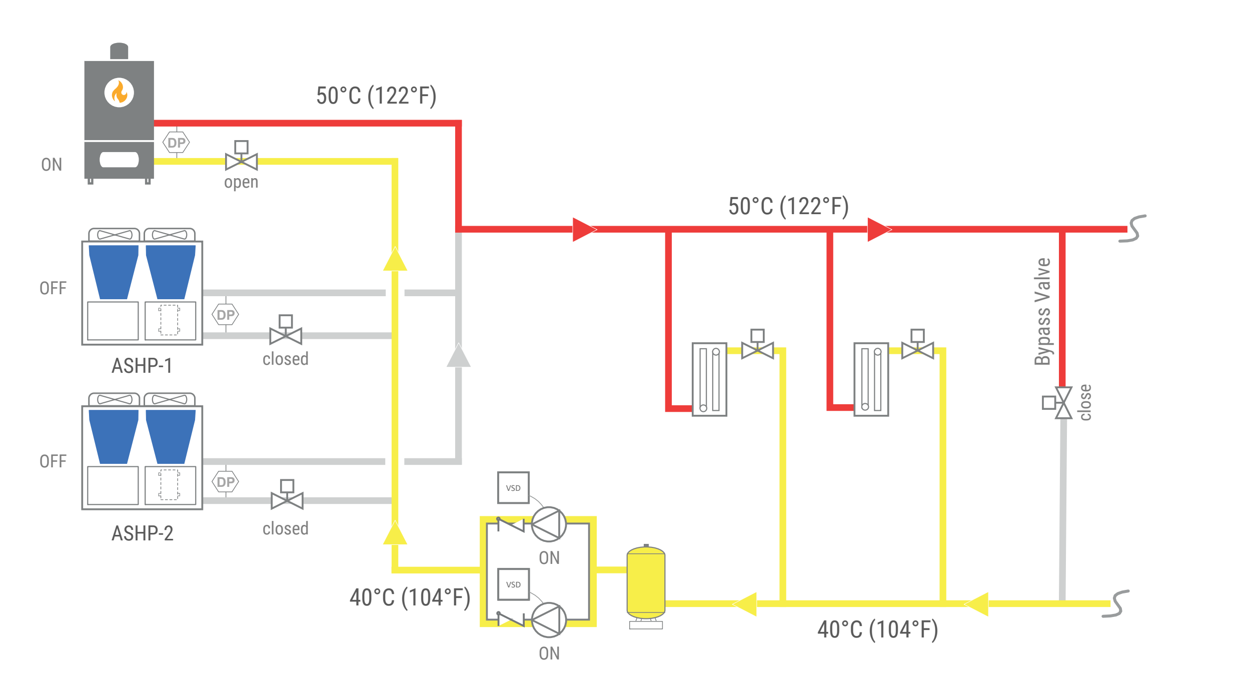

Figure 8.4 | *2-Pipe VPF System with Boiler in Parallel*

The pump flow range will be;

Maximum Flow — 61.2 L/s (972 USgpm)

This is set by the design cooling flow rate.

Minimum Flow — 10.2 l/s (162 USgpm).

This is set by the minimum flow allowed for a single ASHP. If the building flow needs to be less than the unit minimum flow (10.2 l/s (162 USgpm)), the bypass line will need to open to maintain minimum flow through the ASHP heat exchanger.

The primary pump system serving the three ASHPs requires a 6:1 turndown ratio. This will be very difficult to do with a single pump and would not offer any redundancy. It is not a requirement to match the number of pumps to the number of ASHPs, nor do the individual pumps need to match the individual ASHP flow requirement. It is better to select the pumps to efficiently deliver the system flow range and the required system pressure.

Consider the same system shown in Figure 8.3 but now the system is a VPF 4-pipe system. This will have dedicated pump banks for heating and cooling.

Maximum Cooling Flow - 61.2 L/s (972 USgpm)

This is set by the design cooling flow rate.

Minimum Cooling Flow - 10.2 l/s (162 USgpm).

Maximum Heating Flow - 37.2 L/s (591 USgpm)

This is set by the design heating flow rate.

Minimum Heating Flow - 10.2 l/s (162 USgpm).

The minimum flow for cooling and heating is still set by the minimum flow allowed for a single ASHP. If the building flow needs to be less than the unit minimum flow (10.2 l/s (162 USgpm)), the heating or cooling bypass line will need to open to maintain minimum flow through the ASHP heat exchanger.

Primary pump sizing may be impacted in systems with sequential boilers where the boiler is sized for the heat load. If the control strategy is to operate only the boilers or the ASHPs, then there is no issue. If however, the boiler may operate simultaneously with the ASHPs, there may not be enough primary pump flow to meet the minimum flow requirements of both the boiler and the ASHPs. This application effectively has two complete heating plants running at the same time with only enough

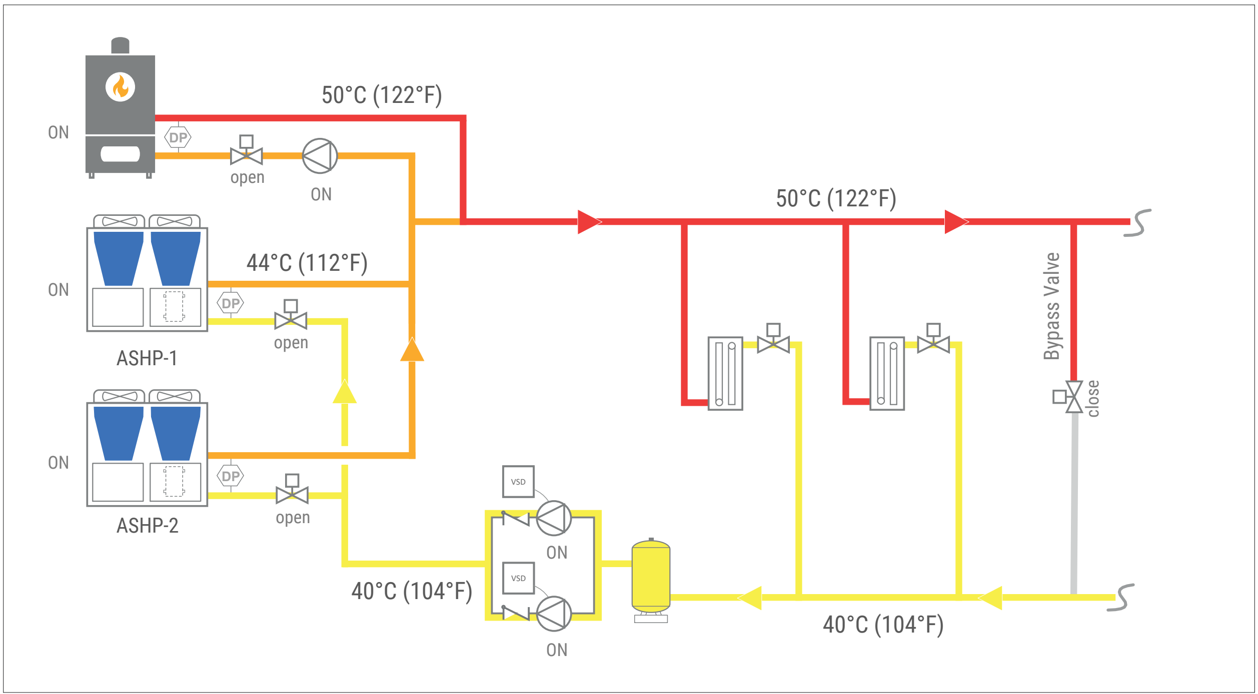

Figure 8.5 | *2-Pipe VPF System with Boiler in Series - Simultaneous Operation*

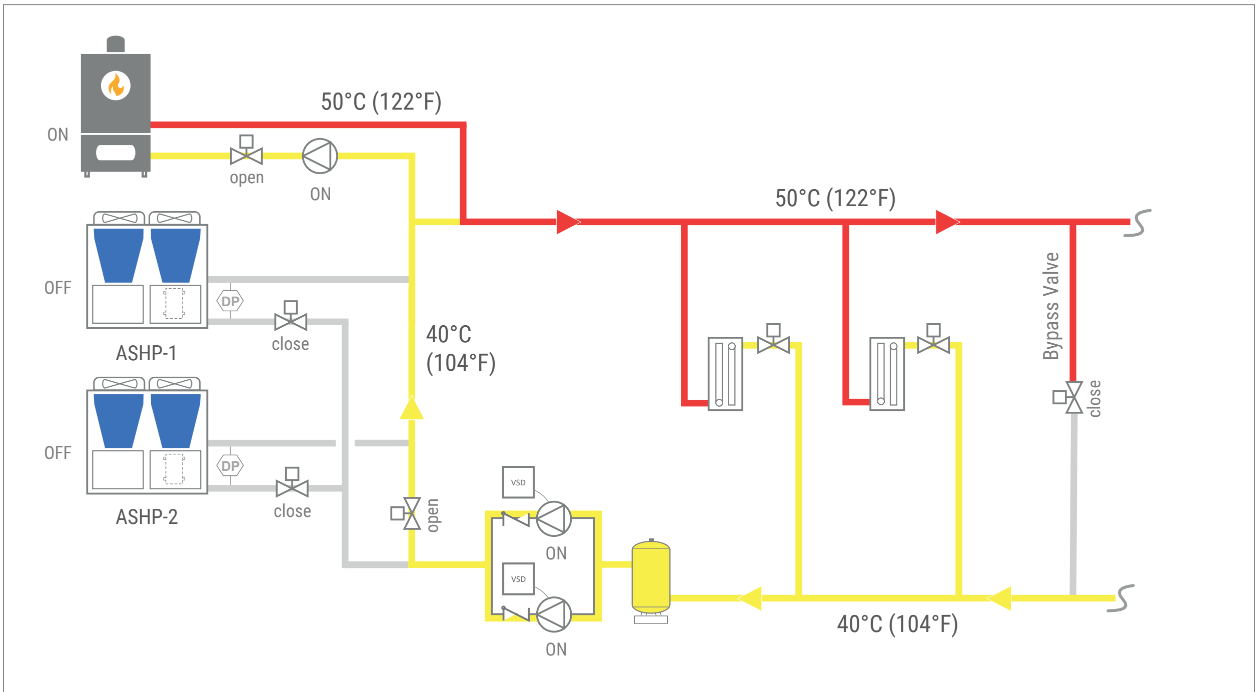

Figure 8.6 | *2-Pipe VPF System with Boiler in Series - Sequential Operation*

flow to feed one system at a time. If this sequence is used, the minimum flow requirements during simultaneous ASHP-boiler operation must be checked. The primary pump size may need to be increased.

2-Pipe Changeover VPF with Boiler

Details on auxiliary heat (i.e. a boiler) were introduced in Chapter 5. Where boilers require constant flow, some care must be taken when applying them in a VPF system. Even if the boilers can operate with variable flow, its minimum flow and pressure drop will be different than the ASHPs so maintain-

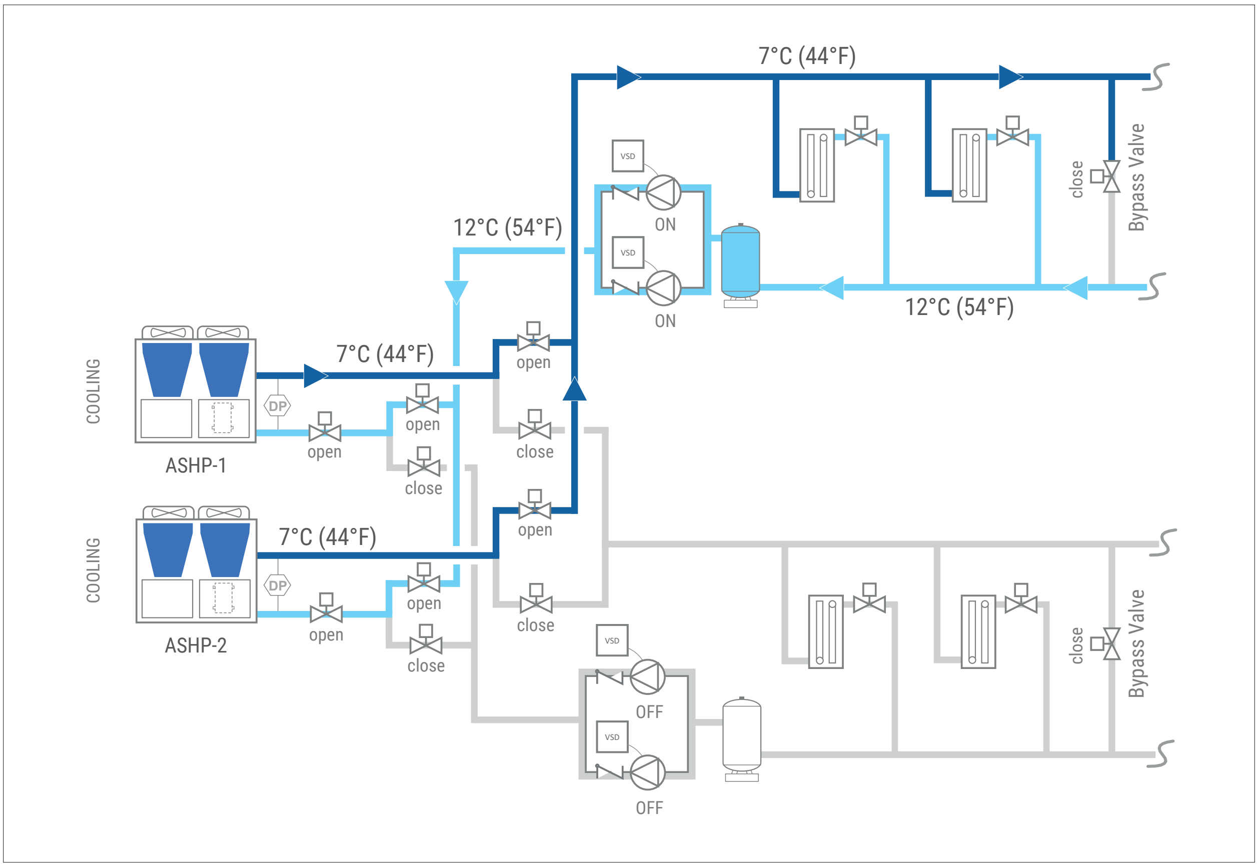

Figure 8.7 | *4-Pipe VPF System - Cooling Only*

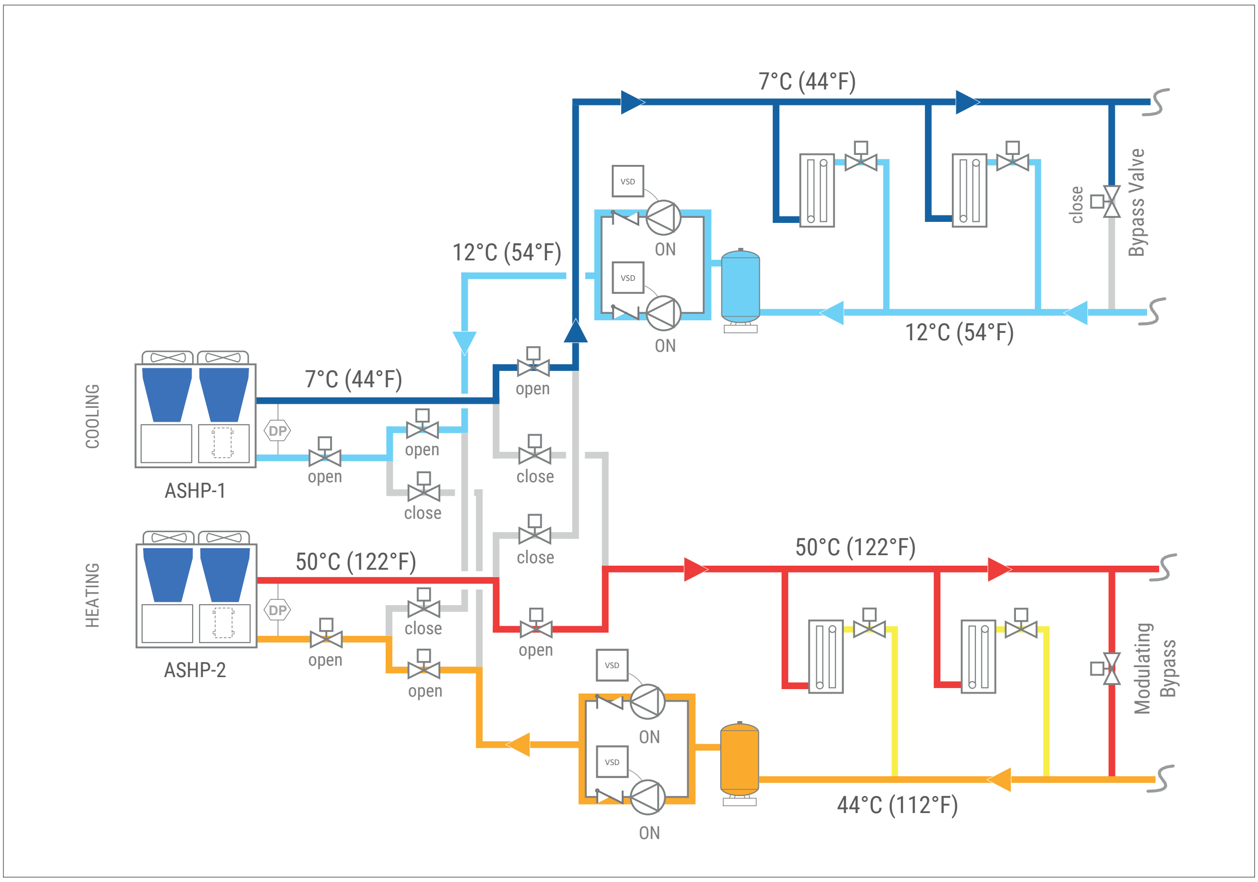

Figure 8.8 | *4-Pipe VPF System - Heating and Cooling*

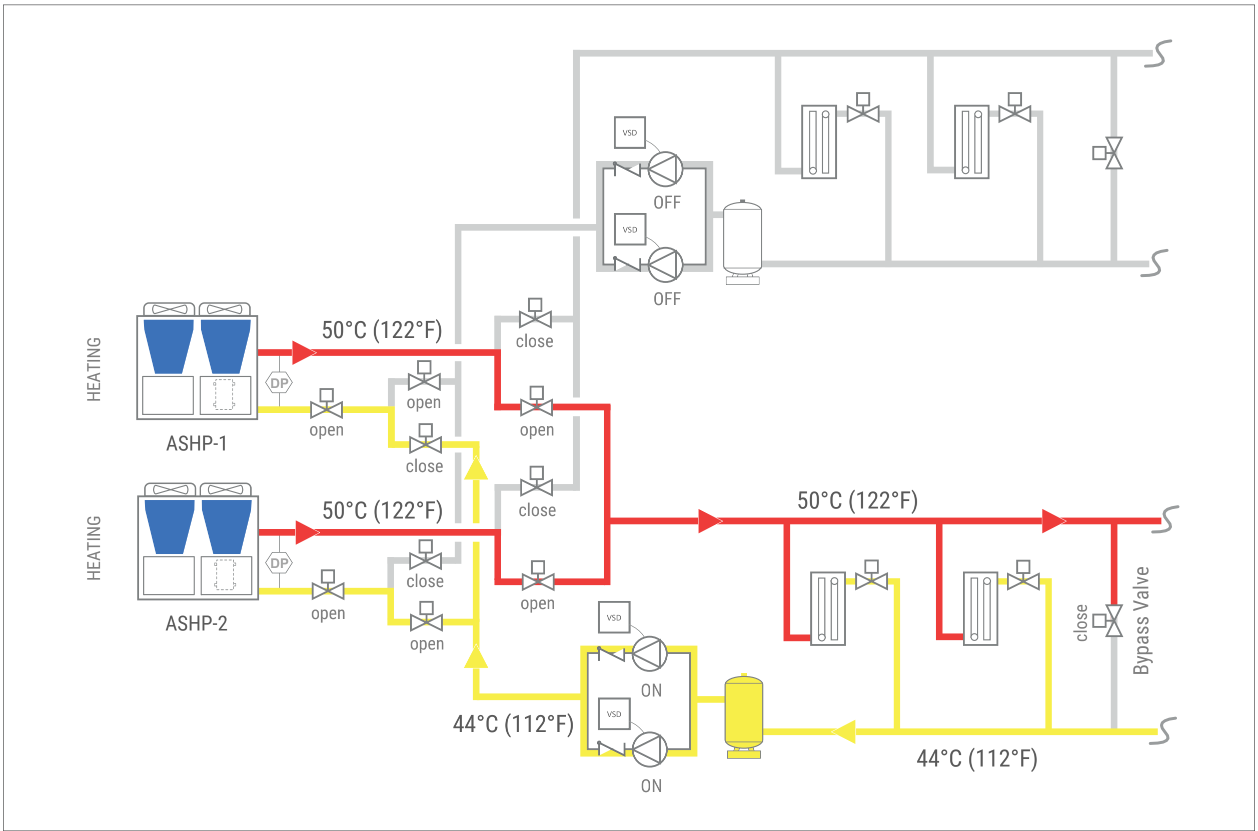

Figure 8.9 | *4-Pipe VPF System - Heating*

ing minimum flow through the boiler must be carefully considered.

Figure 8.4 shows auxiliary heat placed in parallel with ASHPs in a 2 pipe VPF system. The parallel arrangement is problematic if the boiler and ASHPs are to operate simultaneously. It is very difficult to get the correct flows between the boiler and the ASHPs when their design water flow and pressure drops are different.

Parallel boiler arrangements can work well when the boiler and ASHPs work sequentially. In this case either ASHPs or the boiler are operating. When the system switches from ASHPs to boiler, the ASHPs shut down and close their valves. Now the primary flow is directed to the boiler which is enabled and opens its valve. This design should be done with a boiler that accepts variable flow.

Figure 8.5 shows a VPF system with the boiler positioned in series with the ASHPs. When the boiler is operated simultaneously, the heating load is first met with the ASHPs. When the ASHPs cannot meet the load (the bivalence point is reached) the boiler can be used to supplement the ASHPs to meet the heating load. The series arrangement is good when the ASHPs and boiler operate simultaneously.

Using a series arrangement when the ASHPs and boiler operate sequentially requires some form of ASHP bypass line around the ASHPs as shown in Figure 8.6. When the ASHPs are commanded off their valves close and a bypass valve allows flow around the ASHPs.

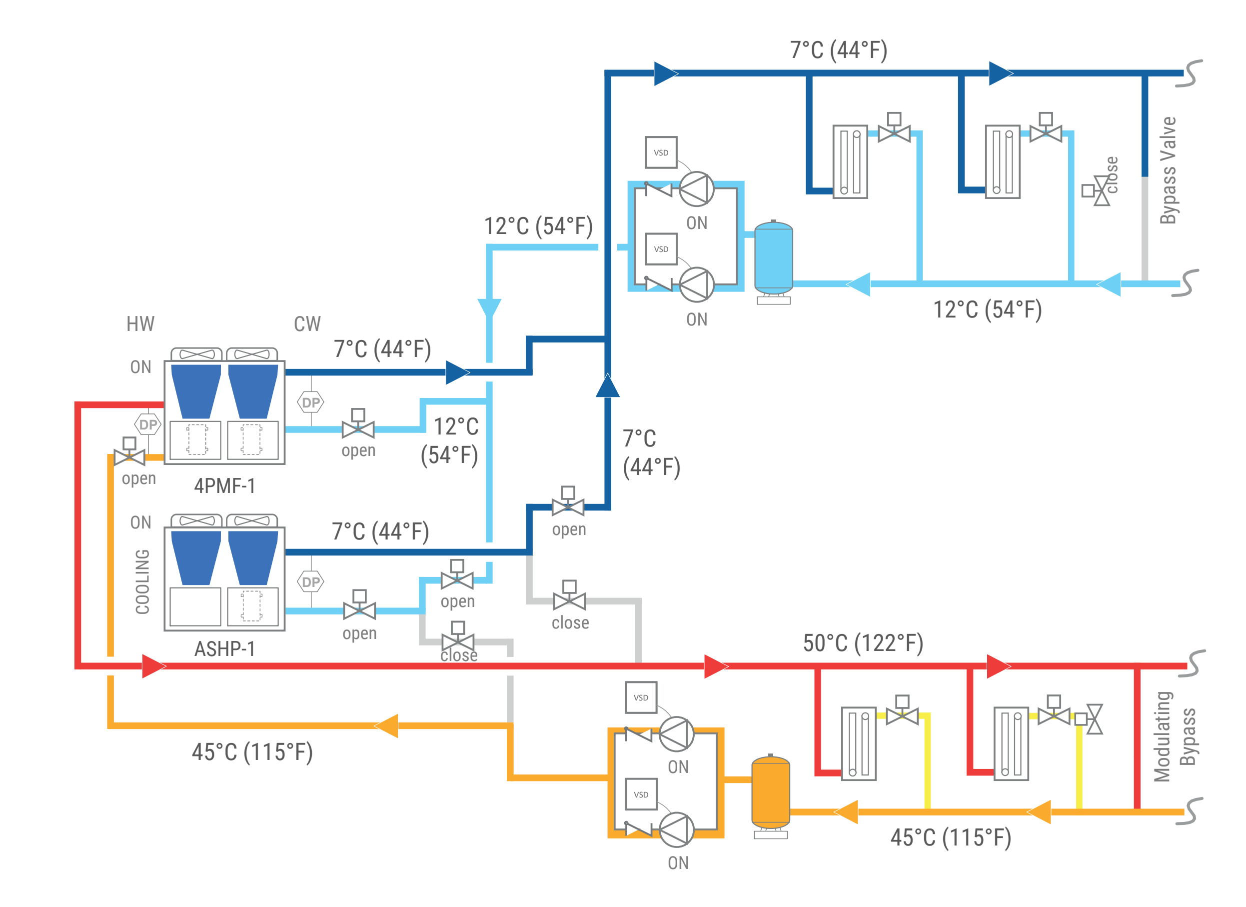

Figure 8.10 | *4-Pipe VPF System with 2-Pipe ASHP and 4PMF - Heating and Cooling*

4-Pipe VPF Design Considerations

4-pipe hydronic systems can simultaneously deliver hot and chilled water to the terminal units for better occupant comfort. When ASHPs are used, the configuration must allow the ASHPs to be connected to the appropriate loop as loads demand. The following figures will show how 4-pipe VPF systems operate.

4-Pipe VPF System with ASHPs — Cooling

Figure 8.7 shows a 4-pipe VPF system with two ASHP units. Changeover valving allows the ASHPs to be connected to either the hot or chilled water loop. In Figure 8.7, the system is operating in cooling only. The system chilled water flow is high enough that the bypass valved is closed.

4-Pipe VPF System with ASHPs — Heating and Cooling

Figure 8.8 shows the system with simultaneous heating and cooling loads. ASHP-2 is operating in heating. The heating load is low enough that the ASHP-2 is below minimum flow, so the heating bypass valve is modulating to maintain minimum flow through the unit.

The hot water temperature range is 6 °C (10 °F). This is possible because the heating system can be designed to meet the heating capacity of the ASHP. The pump and hot water terminal unit coils are sized to match the ASHP heating performance. It is another advantage of a 4-pipe system over a 2-pipe changeover system.

4-Pipe VPF System with ASHPs — Heating

Figure 8.9 shows the system operating in heating. The load is large enough that both ASHPs are connected to the hot water circuit and the flow is high enough that the bypass valve is closed.

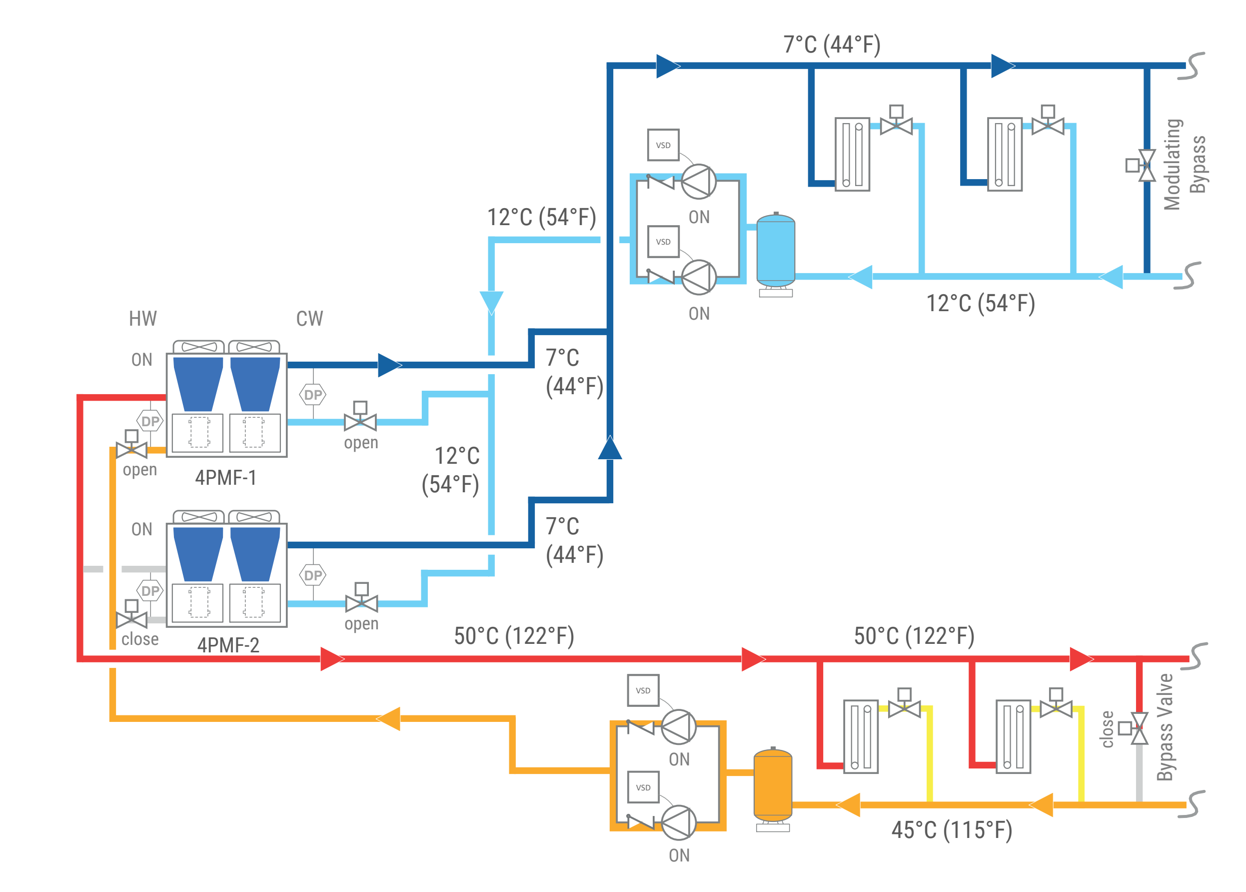

Figure 8.11 | *4-Pipe VPF System with 4PMF Units - Heating and Cooling*

4-Pipe VPF System with ASHP and 4PMF — Heating and Cooling

In Figure 8.10, one of the units has been switched to a 4-pipe multifunctional unit (4PMF). This unit can produce hot and chilled water simultaneously and is permanently connected to both the hot and chilled water loops. The other 2-Pipe Reversible ASHP is valved so it can be switched between the hot and chilled water loops according to the mode of operation of the unit.

In this example, both the 4PMF and ASHP are producing chilled water. The flow is high enough that the bypass valve is closed. The 4PMF unit is also producing hot water. The hot water flow is below the 4PMF minimum flow level so bypass valve modulates to maintain minimum flow. The simultaneous heating and cooling capability of the 4PMF allows the plant capacity to match the heating and cooling loads, Further, the 4PMF unit is moving heat from the chilled water loop to the hot water loop often with the COP around 7 W/W (24 Btu/Wh).

For more information see Chapter 4: 4-Pipe Multi-functional Units.

4-Pipe VPF System with 4PMF — Heating and Cooling

Figure 8.11 shows a 4-pipe system with two 4PMF units. The 4PMF units are permanently connected to both the hot and chilled water loops. The system is operating with a large heating load (both units producing hot water). The hot water flow is high enough that the bypass valve is closed. There is also a small cooling load being met by 4MPF-1. The chilled water flow is low enough that chilled water bypass valve is modulating.

4-Pipe VPF System with 4PMF and Series Boiler — Heating

In applications where ASHPs cannot meet the heating load a boiler can be added. Figure 8.12 shows a boiler added in series to a VPF system. When the system uses the heat pumps below the bivalence point (simultaneous heating), the series boiler can be enabled to meet the heating load.

4-Pipe VPF System with 4PMF and Parallel Boiler — Heating

Figure 8.13 shows a 4-pipe VPF system with 4PMF units and a boiler in parallel. The plant is operating in heating only with the boiler and operating sequentialy (the ambient temperature is below the changeover point).

When the boiler is operated sequentially (the boiler meets 100% of the heating load), and the changeover temperature is reached the 4PMF units are turned off and their isolating valves close. Now all the primary flow will be directed to the boiler. If the boiler minimum flow is reached, the bypass line can be used to maintain minimum flow.

Variable Primary Flow Summary

Variable Primary Flow systems offer many advantages including energy savings and lower installation costs. Using only one set of pumps saves capital cost while removing the pressure drop from the second set of pump pipe fittings (isolation valves, check valves etc.).

For smaller systems the primary pumps, VFDs and controls can sometimes be factory mounted in the units for additional capital savings. Remote primary pumps offer solutions to manage Low Delta T issues by decoupling the flow control from the capacity control. It also allows the pumps to be located indoors, N+1 arrangements and level loading the run hours.

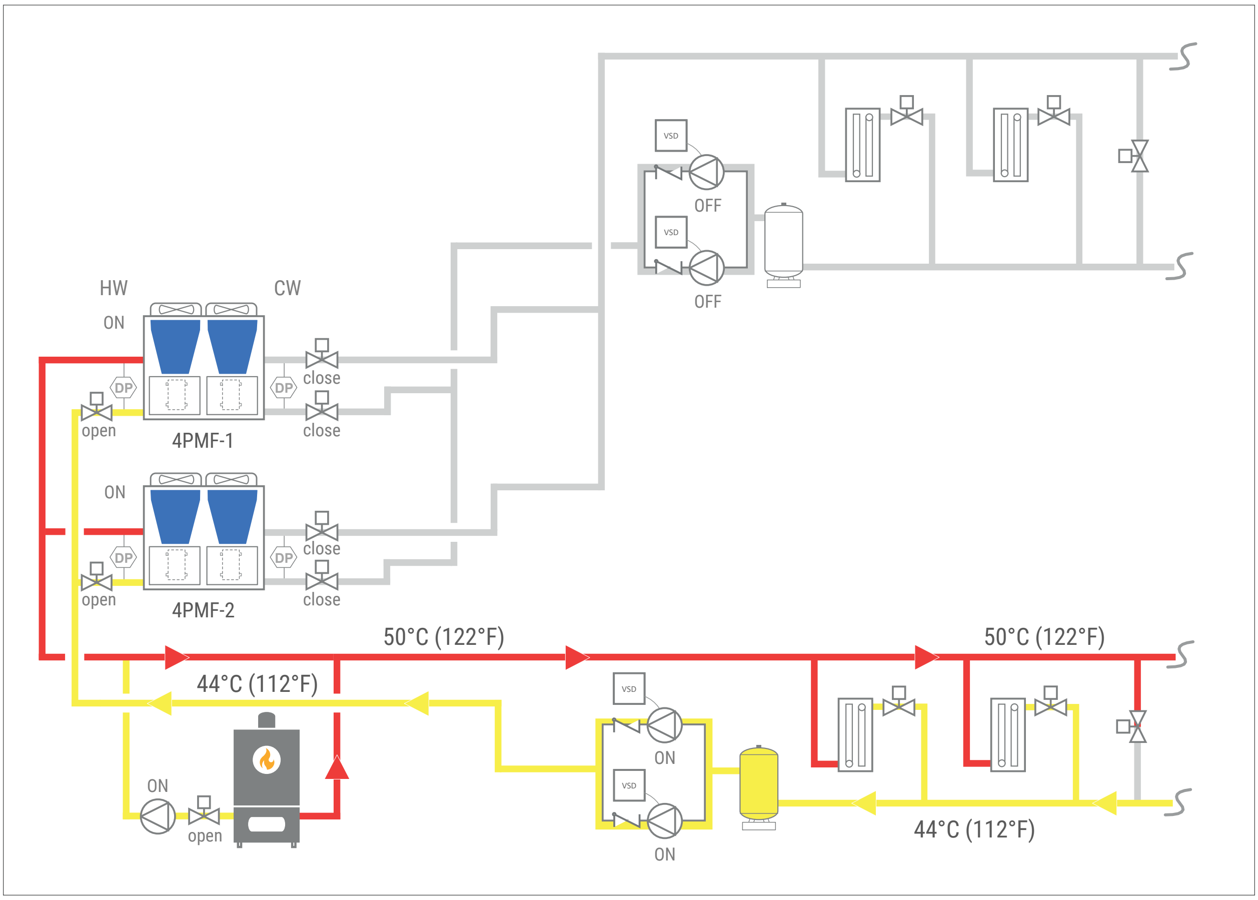

Figure 8.12 | *4-Pipe VPF System with 4PMF Units and Boiler in Series*

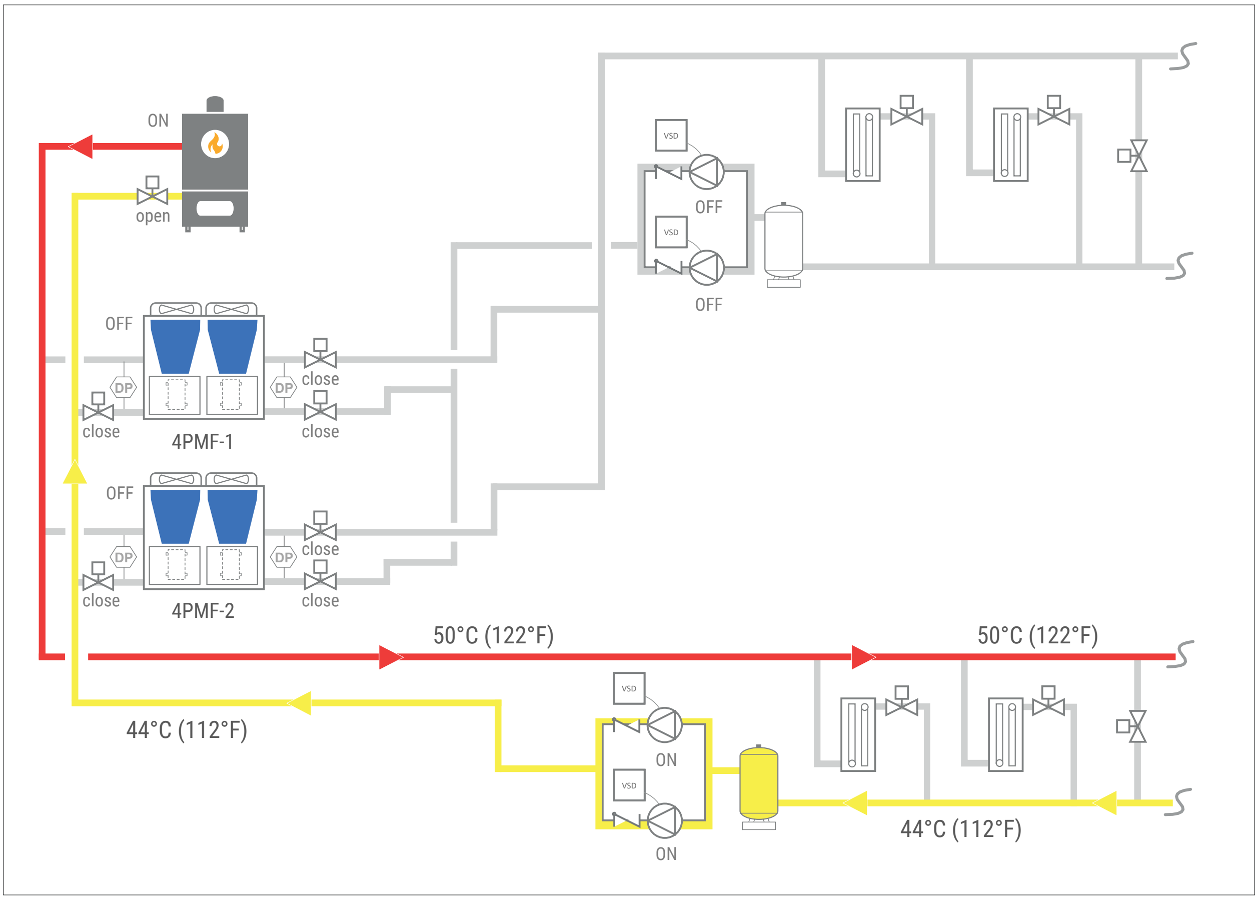

Figure 8.13 | *4-Pipe VPF with 4PMF Units and Boiler in Parallel*

VPF design considerations include;

- » ASHP sizing depends on whether application is summer or winter dominant
- » Primary pumps are sized for design cooling flow and system head
- » Minimum flow is the ASHP (or boiler) minimum flow
- » Have a clear strategy for starting the second ASHP
- » 2-pipe heating flow characteristics are hard to predict but the system will find an energy and flow balance
- » When the boiler and ASHPs are expected to operate simultaneously, a series boiler arrangement is recommended. The different boiler flow rate and pressure drop will not affect the hydronic design. The boiler should be a variable flow type.
- » When the boiler and the ASHPs are expected to operate sequentially, a parallel boiler arrangement is advantageous. The piping supports using the primary pumps to direct all the flow to the boilers. Variable flow boilers are necessary.
- » 4-pipe systems require valving to connect the ASHPs to the appropriate loop
- » 4-pipe systems can take advantage of 4PMF units which are energy efficient and can meet any combinate ion of heating or cooling required

Successful VPF systems start with careful design, installation and commissioning. Integrated controls solutions from the equipment manufacturer can reduce the design and commissioning efforts and provide single point responsibility.

9 Variable Primary, Variable Secondary Systems

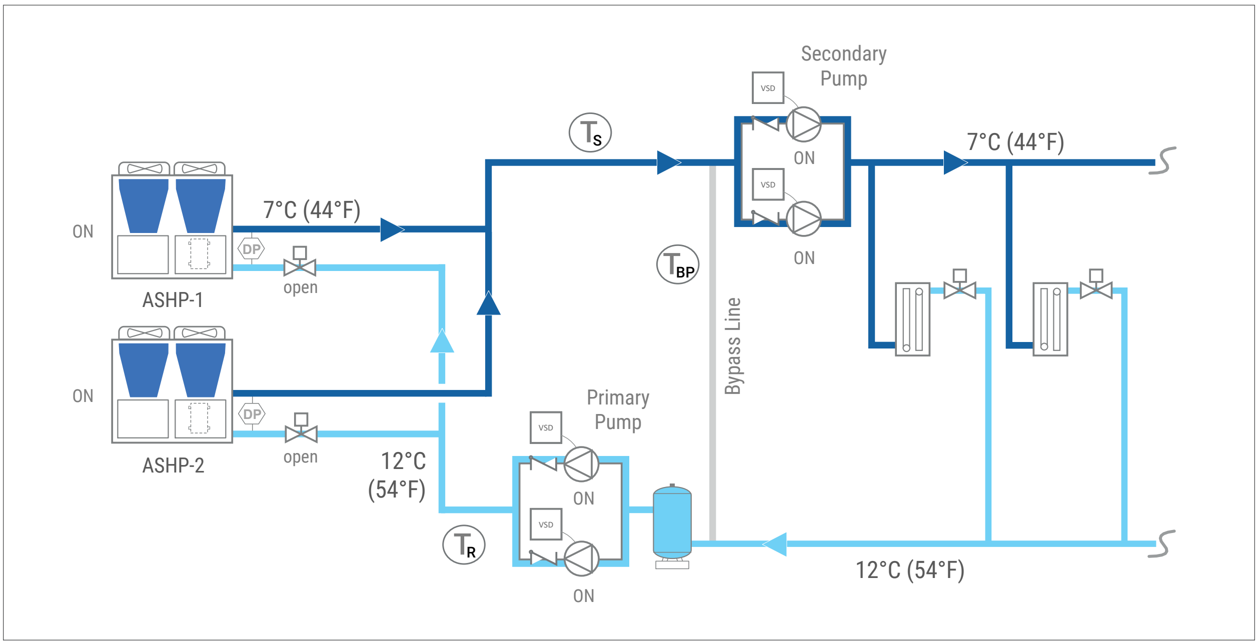

Figure 9.1 | *Variable Primary Variable Secondary System*

Introduction

Figure 9.1 shows a basic Variable Primary, Variable Secondary (VP-VS) system. It is a modification of the variable primary flow covered in chapter 8. In a VP-VS system, there are two sets of pumps, and both are variable flow. The system has a bypass line but no bypass valve. Each ASHP has some form of flow measurement and a two-position isolating valve.

The system in Figure 9.1 is operating in cooling and the system flow is above the minimum flow of the ASHPs. The secondary pumps are controlled to maintain system pressure in the secondary (building) loop. As the terminal unit valves open and close, the loop pressure changes, and the secondary pumps respond ensuring adequate flow is sent to all the terminal units.

The primary pumps operate to match the flow of the secondary pumps. This is typically accomplished by monitoring water temperatures at T_S, T_R and T_{BP}. If T_{BP} starts to approach T_S, the primary pumps have more flow than the secondary pumps, and water is bypassing the secondary side of the system and recirculating back to the inlet of the ASHPs. The response to balance the system is to decrease the pump speed of the primary pumps. The logic is the same when the secondary flow is greater than the primary flow; If T_{BP} starts to approach T_R, the primary pumps have less flow than the secondary pumps, so the primary pump(s) increase flow to balance the system.

Figure 9.2 shows the same system but now the cooling load is low enough that the flow is below the ASHP minimum flow rate. Minimum flow is recognized by the ASHP-1 flow sensor (DP Transducer). When this occurs, the primary pump flow is increased until the minimum flow is achieved. The additional flow from the primary pumps passes through the bypass line, mixing with the return water lowering the return water temperature.

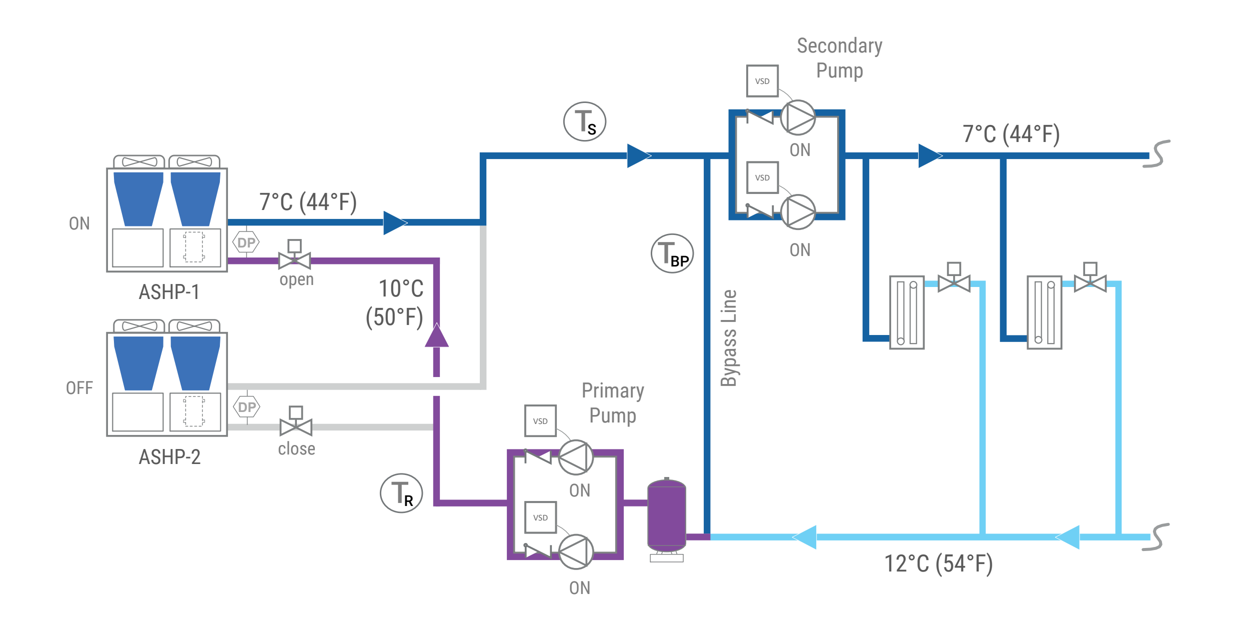

Figure 9.2 | *VP-VS System - Below Minimum Flow*

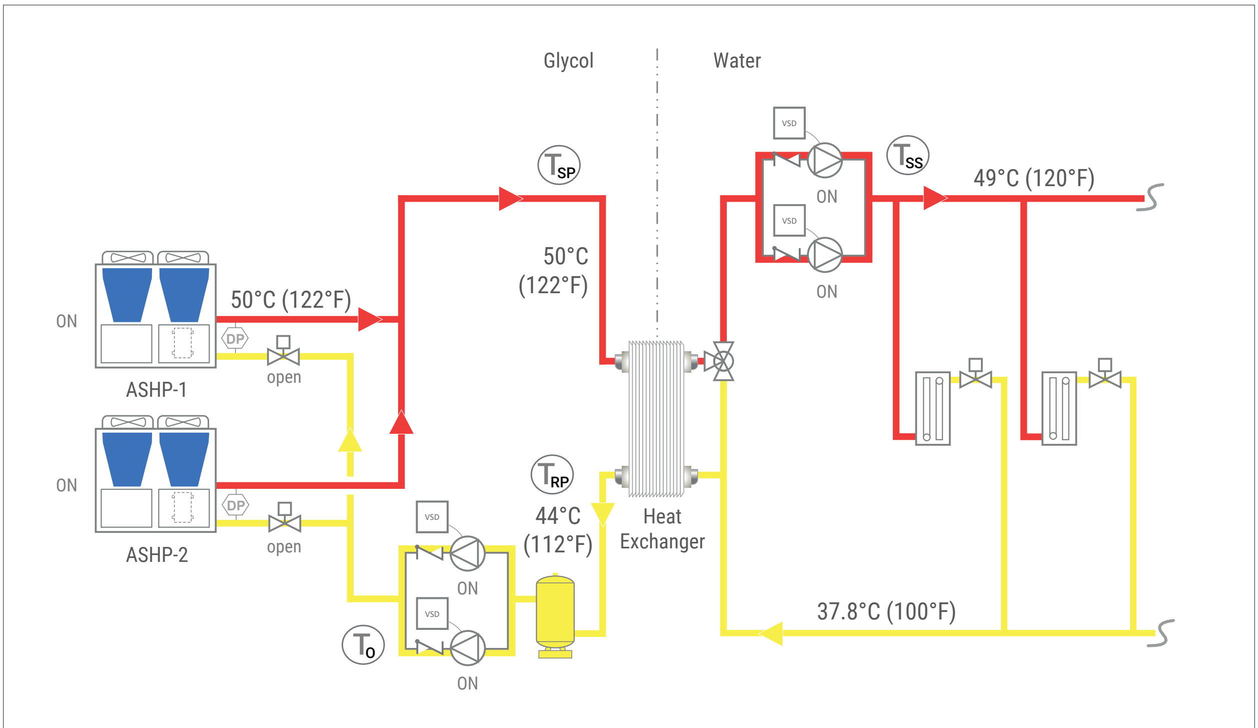

Figure 9.3 | *2-Pipe ASHP VP-VS System with Isolating Heat Exchanger*

VP-VS Advantages

A key advantage of VP-VS is using the variable flow capability of the primary pumps to maintain the minimum flow through the ASHPs rather than modulating the valve in the bypass line. In some applications with a VPS system, modulating the by-pass valve can change the flow out to the building causing the system pressure to fluctuate. This can cause the system to 'hunt' with the primary pump speed oscillating on the bypass valve position. This risk is avoided with VP-VS systems.

Another advantage is the option to introduce either heat exchangers or 4 connection tanks (Buffer/Hy-draulic Separator combination tanks). This will be discussed in the next section.

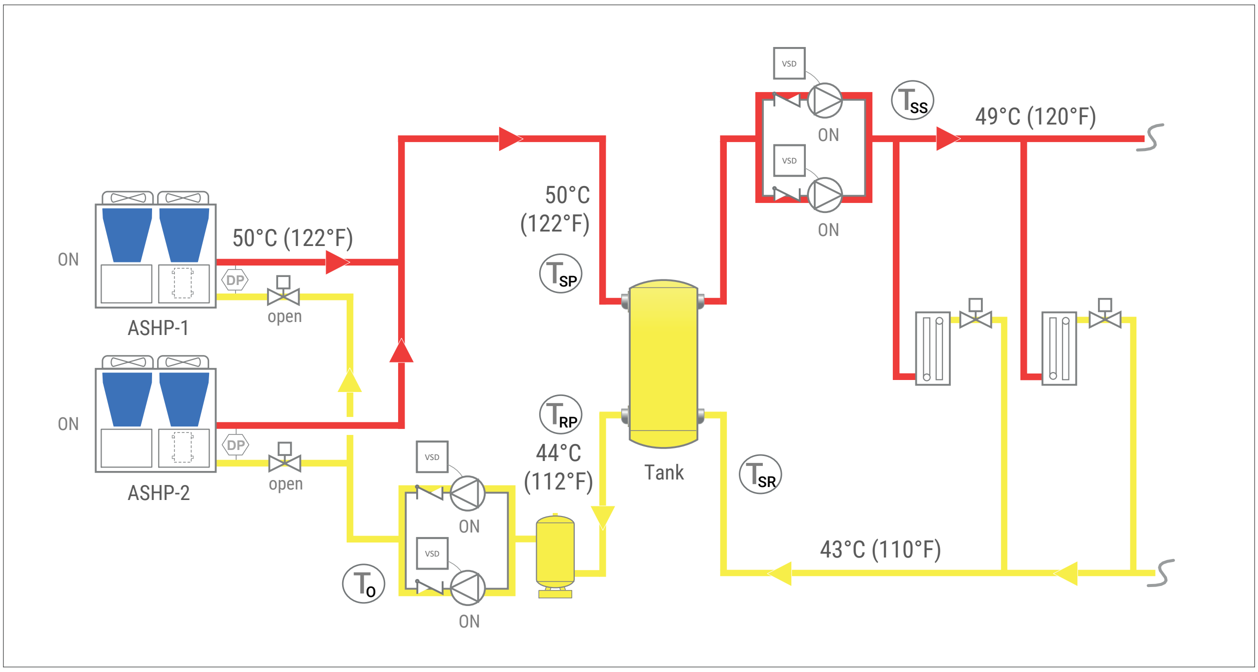

Figure 9.4 | *2-Pipe VP-VS System with Buffer Tank*

The downside of VP-VS is the cost of the second set up pumps and the increase in overall system design pressure drop from the fittings (isolation valves, strainers etc.) associated with the second set up pumps.

Pump Sizing

The primary pumps are variable flow and sized for the design flow and the pressure drop in the primary loop. Additional details about the minimum and maximum flow rates are discussed in Chapter 8. The Secondary pumps are also variable flow sized for the design flow rate and the pressure drop in the secondary loop.

Bypass Line

The bypass line is sized for the minimum flow of the largest ASHP. There is no bypass valve as the primary pumps will control flow through the bypass line. The bypass line location is shown in Figure 9.1.

Isolating the Primary and Secondary Loops

Figure 9.3 shows a VP-VS system with the bypass line replaced by a heat exchanger. The heat exchanger gives the option to have glycol in the primary loop only and distribute water through the building secondary loop without the performance penalties associated with glycol. The primary loop is outside the building so some form of antifreeze is desirable where freezing ambient temperatures are expected. This is particularly true for sequential systems where the ASHPs can be off for extended periods of time.

Another advantage of including the heat exchanger is the primary and secondary loops can be operated with different temperature ranges. In the example, the primary loop is operating at 50 °C – 44 °C (122 °F – 112 °F) while the secondary loop is operating at 49 °C – 37.8 °C (120 °F – 100 °F). Increasing the temperature range on the secondary side can reduce pipe and pump sizing and lower the pump energy use. How much the primary and secondary temperatures can differ will depend on the heat exchanger selection. It is important that the heat exchanger performance be checked at both the design and the minimum flow rates. Note there is a supply temperature difference created by the approach needed by the heat exchanger. The heat

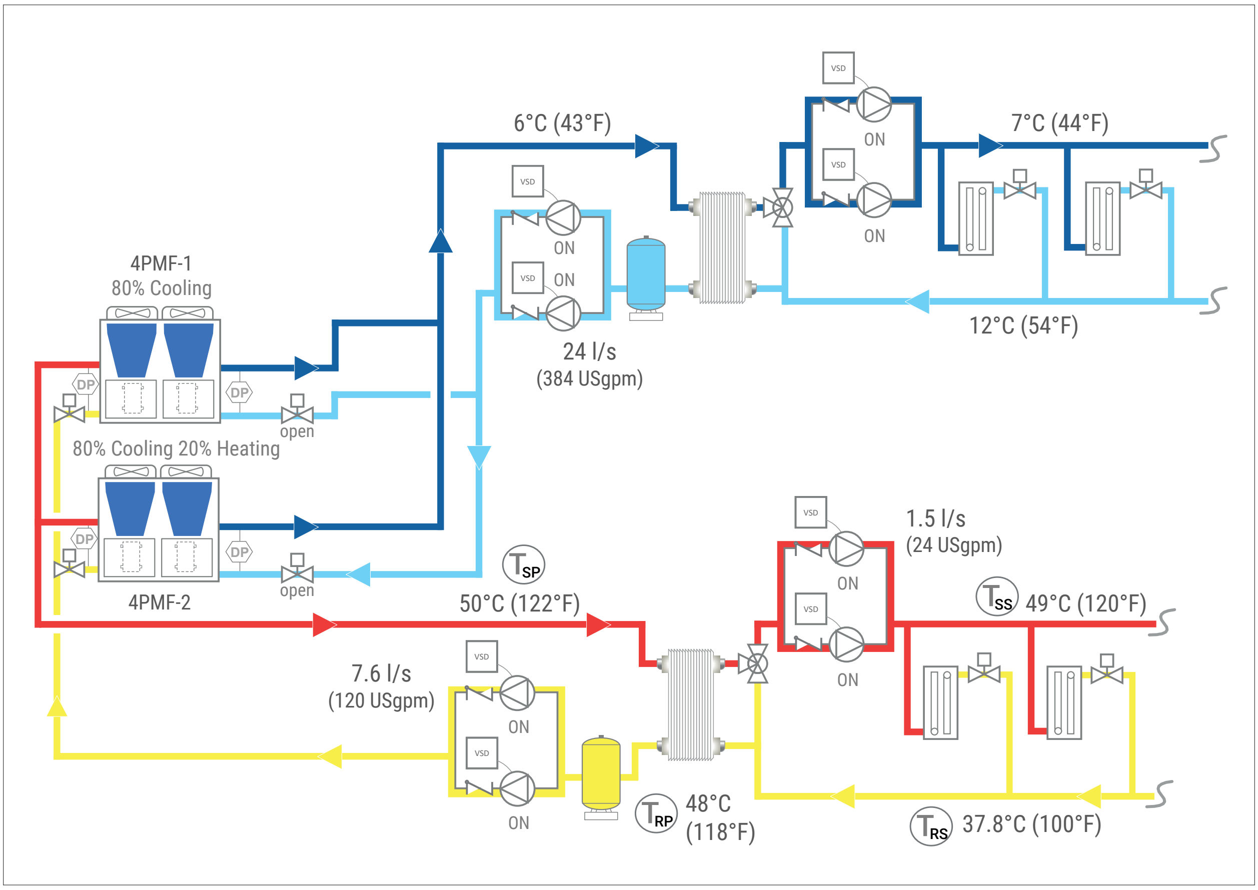

Figure 9.5 | *4-Pipe VP-VS System with Heat Exchangers*

exchanger should be selected to keep the approach temperature to a minimum.

The primary loop is controlled by maintaining a constant temperature range. In this example the range is 50 °C – 44 °C (122 °F – 112 °F) as measured at T_{SP} and T_{RP}. If the temperature range starts to increase, there is not enough flow, and the primary pumps will speed up. If the temperature range starts to decrease, the pumps will slow down. The ASHP capacity changes to meet the supply water temperature setpoint.

Secondary loop flow is controlled by maintaining system pressure. The secondary supply water temperature will be T_{SP} — the heat exchanger approach. Alternatively a three valve can be installed on the secondary side controlled by T_{SS} to maintain a specific supply water temperature.

In Figure 9.4, the bypass line has been replaced with a 4 connection buffer tank. The tank decouples the primary and secondary loops and allows different temperature and flow rates in each loop. The tank must be designed to promote stratification. In heating, hot water from the ASHPs is introduced at the top of the tank filling it with hot water from the top down. The flow rate can be optimized for the ASHPs. The primary flowrate is maintained by controlling the primary loop temperature range. The secondary loop flow maintains system pressure. If the secondary loop exceeds the primary loop capacity, the buffer tank will fill up with cold water from the bottom. This will cause the primary flow to increase.

The 4 connection tank volume will help the system water volume (thermal mass) assessment. It is also possible to have different primary and secondary temperature ranges. However, if glycol is required, it will be throughout the system.

Figure 9.5 shows a 4-pipe VP-VS system with two 350 kW (100 ton) 4PMF units. The system has a 560 kW (160 ton) cooling load. There is a 140kW (240 kBtu/h) heating load (e.g. reheat) being serviced by 4PMF-1. The 4PMF unit is using heat collected from the chilled water loop so it is recovered heat with a very high COP. The heating load is low enough that the hot water flow is below the minimum flow 7.6 l/s (120 USgpm) of the 4PMF unit. The low flow is measured by the DP sensor on the 4PMF hot water loop and the control system increases the primary flow until the minimum flow is met. This results in the return primary water temperature being 48 °C (118 °F).

VP-VS Summary

VP-VS systems offer the best from primary-secondary and variable primary flow systems. The energy savings are close to VPF. The variable flow primary pumps can maintain design delta Ts across the ASHPs in both heating and cooling mode. Using variable flow primary pumps is more reliable during conditions below minimum flow than using a bypass valve.

By adding either, a heat exchanger or a 4 connection tank it is possible to increase the secondary system temperature range, the capital cost and pump work of the secondary loop can be reduced. Using a heat exchanger also allows the primary loop to have glycol while the secondary loop can remain water.

The VP-VS system will likely have a higher capital cost however, if the secondary loop temperature range is increased enough, the pipe savings may cover the additional cost of variable flow primary pumps. The capital cost is only slightly higher than primary-secondary (VFDs for the primary pumps).

VP-VS design considerations include:

» Requires 2 sets of variable flow pumps

» Primary pumps maintain temperature range until minimum flow is reached.

» No bypass valve required.

» Secondary pumps maintain secondary loop pressure.

» Where glycol is required, consider using a heat exchanger to isolate the glycol to just the primary loop.

» Where a larger heating temperature range is desirable, consider using a heat exchanger or 4 connection tank.

» Ensure there is a strategy to start the second ASHP as the load increases to avoid nuisance trips

10 Other Design Considerations

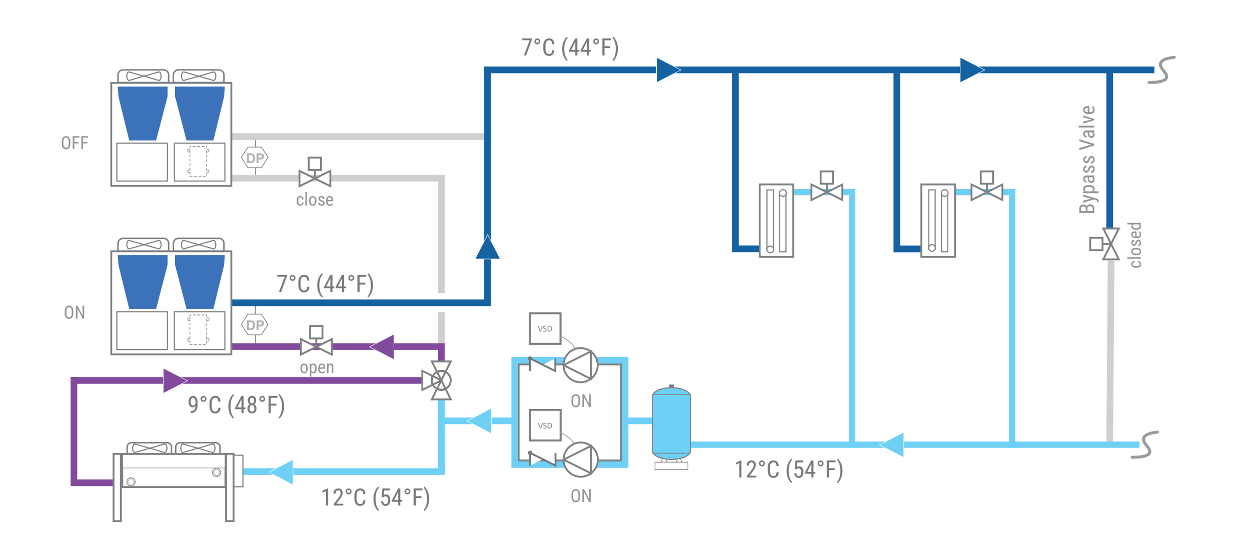

Figure 10.1 | *Integrated Free Cooling*

Introduction

This chapter will cover additional topics that affect the system design regardless of which system type is selected.

Free Cooling

Introducing waterside free cooling to the system design can improve the overall chilled water plant system performance. In some applications it can be an energy code requirement.

Figure 10.1 shows a VPF chilled water system with a dry cooler installed in series with the chillers. Having the dry cooler in series with the chillers makes this an integrated solution meaning the chillers can operate simultaneously. The dry cooler will first cool the chilled water. If the dry cooler cannot meet the load, then the chillers will provide additional cooling.

In Figure 10.1, the dry cooler cannot meet the load, only cooling the chilled water from 12 °C (54 °F) to 9 °C (48 °F). One chiller is operating cooling the chilled water to the 7 °C (44 °F) setpoint.

The typical control sequence is to consider free cooling as the first stage of cooling. When ambient conditions allow (i.e. dry bulb temperature is low enough) the BAS modulates the three way valve to

Figure 10.2 | *Air Cooled Chiller With Integrated Free Cooling*

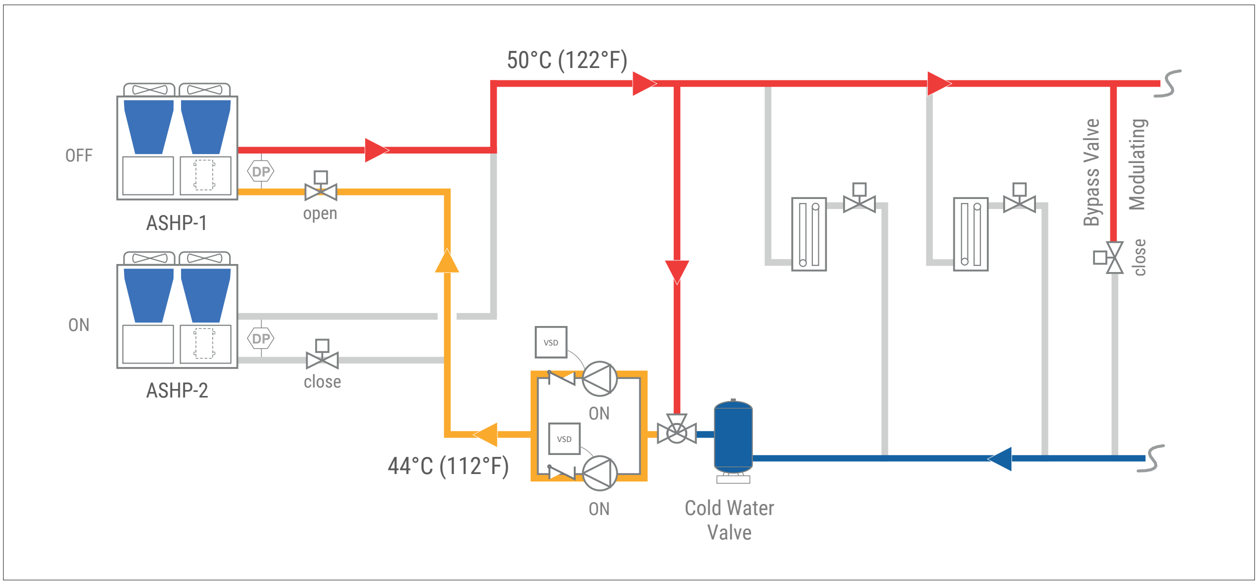

Figure 10.3 | *Cold Water Start Up*

direct some or all the chilled water flow through the dry cooler. If all the flow is directed through the dry cooler and the load cannot be met, then additional mechanical cooling via the chiller is enabled. This control sequence may also be offered as part of the chiller unit controller from the manufacturer. This is typical when the dry cooler is integrated into the air-cooled chiller and sometimes may be a feature that can also be used with a remote dry cooler.

Whether free cooling is the right choice from an energy savings point of view depends on the system design. Applications, whether there is a need for chilled water cooling in the winter lend themselves to free cooling. This is often a process load such as data centers or healthcare applications.

Where the building load profile has simultaneous heating and cooling, using 4-pipe multifunctional units is generally a better choice. Using 4PMF units will allow heat collected in one part of the building (i. e. the building core) to be redistributed to the parts of the building that require heat. If free cooling is used for the core area, then either a boiler or heatpump will be required for the area that needs heat negating any free cooling savings.

Cold Water Start Up

Where winter weather can be cold and the system may be shut down for extended periods such as over a weekend or holiday, consideration must be given to cold water start up. The water in the hot water piping, buffer tank etc. can cool off significantly during periods of inactivity. This cold water can create challenges for the heat pump (chiller if operating in cold temperatures) during start up as the additional load from having to warm the water up can lend to reliability issues.

The solution is to recirculate hot water with a three way valve back to the ASHP to warm the water up quickly. Then the valve is slowly opened to heat the remaining water in the system up to normal operating temperature. The valve is controlled by the ASHP.

Figure 10.3 shows a VPF system with a cold water valve. Note the valve location must allow the pump to circulate water through the ASHP with the least amount of water volume possible. If the pumps are unit mounted in the ASHP, this is straight forward. With remote pumps, the valve location can be more complex.

On a cold water start, the water in the lines could be between -20 and 20 °C (-4 and 68 °F) depending on how much piping is indoors or outdoors. The three way valve will recirculate the water back the ASHP allowing the water to quickly warm up to the hot water setpoint (i.e. 55 °C (130 °F)). Once this small volume of water has reached normal operating temperatures, the three way valve will start to modulate and introduce water from the rest of the system into the ASHP to gradually raise the system

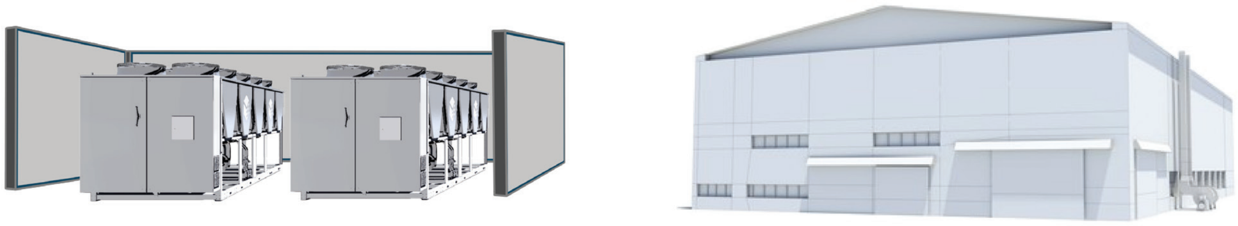

Figure 10.4 | *Installation Considerations*

water temperature until the temperature setpoint has been met.

Access, Clearances and Prevailing Wind

The ASHP location and any enclosures around the ASHP must be carefully considered. The fans on a 350 kW (100 ton) ASHP can move 42,500 l/s (90,000 cfm) to heat or cool the unit. Any restriction on this airflow will impact unit performance. Figure 10.4 shows common applications that can impact airflow performance such as a position close to a building or an enclosure.

The ASHP manufacturer can provide minimum clears around the unit that consider both service access and airflow restrictions. It is important to get approval from the manufacturer if the site conditions make it difficult to meet the manufacturers' requirements and any deviations are required.

There is an additional consideration with ASHPs. The air that passes through the outdoor coil in chiller mode is heated from 35 °C (95 °F) to 49 °C (120 °F) becoming more buoyant rising up from the ASHP and any enclosures that may be around it. However, in heating mode the air can enter at -10 °C (14 °F) and is cooled to -17 °C (1 °F), becoming denser and falling back down around the ASHP (see Figure 10.5). Lowering the entering air temperature to the ASHP will reduce its capacity and could cause the unit to reach its lower ambient limit even though the actual temperature is not that cold.

The cold, dense air situation is compounded if several ASHPs are aligned in a row across the prevailing wind as shown in Figure 10.6 left side. The air from the upstream unit will fall on the downstream units effectively lowering the ambient temperature for those units. Where possible ASHPs should be pointed into the wind so the discharge air does not

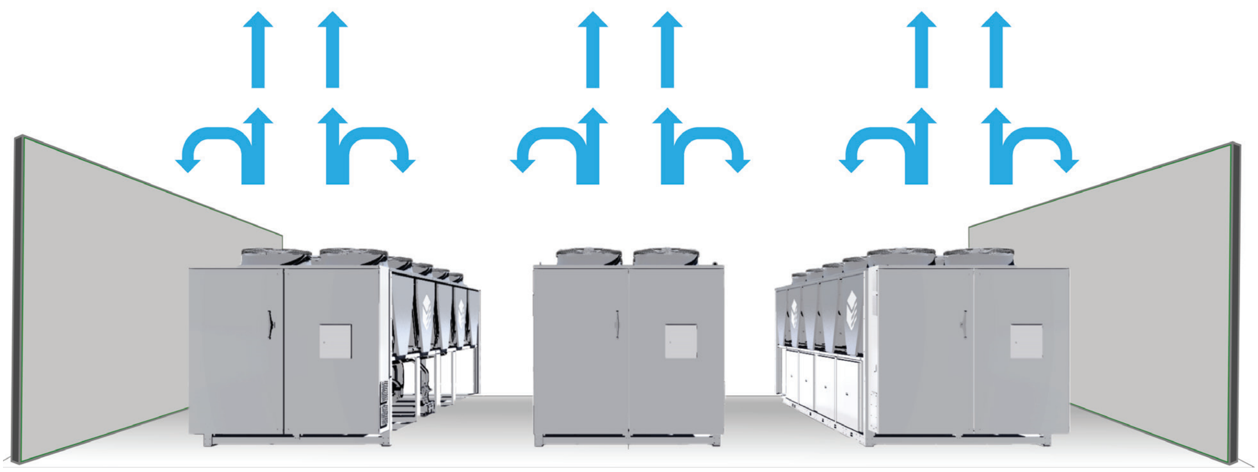

Figure 10.5 | *Air Distribution in Heating Mode*

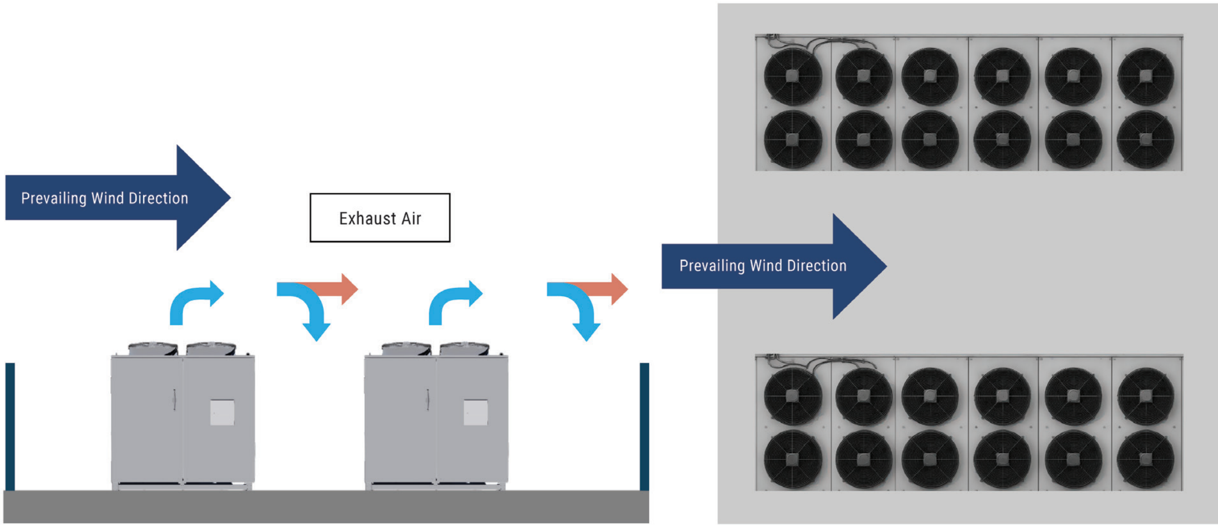

Figure 10.6 | *Prevailing Winds*

interfere with surrounding units. (See Figure 10.6 right side.)

ASHRAE Std 15 and CSA B52

Most jurisdictions adopt a refrigeration safety code which must be considered in the system design. Common safety standards include;

» ASHRAE Standard 15 – Safety Standard for Refrigeration Systems.

» CSA B-52 – Mechanical Refrigeration Code.

For North America, ASHRAE Std 15 is used in the USA while CSA B52 is used in Canada. These two standards are closely aligned with minor differences.

INCREASING FLAMMABILITY

Higher flammability	A3	B3
Flammable	A2	B2
Lower flammability	A2L	B2L
No flame propogation	A1	B1
	Lower toxicity	Higher toxicity

INCREASING TOXICITY

Figure 10.7 | *ASHRAE Std 34 Refrigerant Classification*

The refrigerant choice impacts the requirements. Figure 10.7 shows ASHRAE Std 34 refrigerant classifications. In North America, ASHPs are now using A2L refrigerants such as R-454B and R-32. In Europe, A2Ls are common and the use of A3s (R-290 [Propane]) is growing rapidly. As A2Ls are mildly flammable, additional safety requirements are required according to these safety standards. In addition to the safety standards, UL60335-2-40 (the safety listing used for ASHPs) has been updated to include requirements for the product design and construction when A2Ls are used.

Using ASHPs keeps the refrigeration systems outdoors making compliance with either code straight forward. ASHPS are designated as low probability, indirect closed systems. This means a refrigerant leak has a low chance of entering the occupied space. For comparison, a variable refrigerant flow (VRF) system is a high probability, direct system. A leak in a VRF system can easily impact the occupied space.

Paragraph	Designation	Cooling or Heating Source	Air or Substance to be Cooled or Heated
5.1.1	*Direct system*		
5.1.2.1	*Indirect open spray system*		
5.1.2.2	*Double indirect open spray system*		
5.1.2.3	*Indirect closed system*		
5.1.2.4	*Indirect vented closed system*		

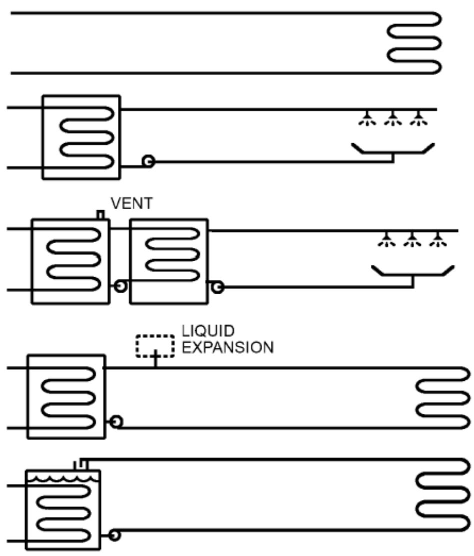

Figure 10.8 | *ASHRAE Std 15 System Classification*

The following is a summary of requirements for ASHPs that are installed outside in low Probability, Indirect systems. It is recommended that the appropriate standard be thoroughly reviewed for compliance.

- ASHP placement should be 6.1 m (20 ft) from windows, building ventilation openings, pedestrian walkways or building exits.
- For refrigerants that are heavier than air, the vent discharge should be 6.1 m (20 ft) horizontally from any below grade walkways, entrances, pits or ramps if a full system release would raise the refrigerant level above the Refrigerant Concentration Limit (RCL).

Appendix A — Glossary

2-Pipe System A hydronic system with a single loop (2-pipe – supply and return) that provides either hot water or chilled water to the terminal units but not at the same time. The system must switch over from heating to cooling.

4-Pipe Multifunctional Units (4PMF) A special version of an Air Source Heat Pump that can simultaneously provide hot and chilled water. When there are simultaneous loads, the unit moves heat from the chilled water loop to the hot water loop at a very high efficiency level. A 4PMF is 4-pipe system that is permanently connected to both the hot and chilled water systems.

4-Pipe System A hydronic system with 2 loops (4-pipe – supply and return hot water and supply and return chilled water) that provides either hot water or chilled water to the terminal units but at the same time.

Actuator An electric or pneumatic motor used to operate valves and dampers. Can be modulating or two position type.

Adiabatic Process A process where no energy is lost to the surroundings. Enthalpy does not change for the fluid. An example is the refrigeration expansion process.

Affinity Laws The laws for a fixed system relate pump speed, impeller diameter, flow pressure and power for centrifugal pumps and fans.

Air Source Heat Pump (ASHP) A refrigeration unit that can produce either hot or chilled water using ambient air as either the energy source or the sink. An ASHP is a 2-pipe unit.

Azeotropic Refrigerants Refrigerant blends that behave as one substance. An example is R-500.

Backflow preventer A specific type of check valve used to prevent non potable water from flowing back into the potable water system.

Balance Valve A valve used to adjust flow by adding a pressure drop in a system. It can be automatic or manual type.

Bernoulli Theorem A theorem that indicates there are three different pressures in a system – static head, velocity head and pressure head and all can be used to move a fluid through a system. It is also possible to convert energy from one form of pressure to another.

Bivalence Point The ambient temperature where the capacity of the ASHP matches the building heating load. If the ambient temperature drops below the bivalence point, the ASHPs can no longer meet the building heating requirements.

Buffer (thermal storage) Tank A storage tank sized to hold enough water in a heating or chilled water system to provide stable operation.

Carnot Cycle The ideal, reversible heat cycle between two infinite heat sinks.

Centrifugal Pump A dynamic pump that converts velocity pressure to static pressure to create fluid flow.

Check Valve A valve that allows flow only in one direction.

Chiller A machine that cools water for air conditioning and process cooling. Can be vapor compression or absorption type.

Circulator pump A small in line centrifugal pump.

Closed Loop System A hydronic system that has only one contact point with atmosphere. E.g. a chilled water loop.

Compressor A component in a refrigeration system that compresses refrigerant vapor to a higher pressure and temperature and consumes power to do so.

Compression Ratio The discharge pressure divided by the suction pressure. Typically used with positive displacement compressors.

Condenser A component in a refrigeration system where refrigerant is condensed from a gas to a liquid and heat is rejected to the surroundings.

Connected Load The sum of all the zone heating or cooling loads.

Cooling Tower A device used to reject heat from water to atmosphere through an evaporative process. The water is exposed to ambient air where a small portion evaporates and cools the remaining water. This process is based on ambient wet-bulb. The cooled water is often used with water cooled chillers.

COP (Coefficient of Performance) The measure of the refrigeration system's efficiency. Defined as refrigeration effect/work input.

Decoupler See Primary-Secondary Systems.

Density (d) The mass of a substance divided by the volume that substance occupies. It is measured in kg/m^3 ($lb/ft\ lbm/ft^3$)

Design (Peak) Load The maximum heating or cooling load experienced by the entire building.

Direct Expansion (D-X) Evaporator An evaporator where the refrigerant evaporates in the coil or heat exchanger to cool air or water. This can be the refrigerant to water heat exchanger in an ASHP or a DX coil in a packaged air handling unit.

Distribution System A piping system that connects loads with sources for either hydronic heating or cooling.

Diversity The design load divided by the connected load.

DWV Abbreviation for Drain – waste - vent.

Dynamic Compressor A compressor that imparts velocity pressure on the fluid to increase the fluid's pressure.

EER (Energy Efficiency Ratio) A means of measuring the efficiency of an air-cooled chiller. In North America (based on AHRI 550/590) the EER is Refrigeration effect in Btu/hr/power input in watts.

Enthalpy (h) The measure of energy content of a fluid. Measured in J/kg (Btu/lbm).

Entropy (s) The measure of molecular disorder of a substance. A process with no change in entropy is considered ideal and reversible. Measured in J/kg-°K (Btu/lbm°R).

Evaporator A component in a refrigeration system where refrigerant is boiled from a liquid to gas and heat is absorbed from the surroundings. It is the component that performs the cooling effect.

Expansion Device A component that reduces the pressure and temperature adiabatically in a refrigeration system. There are several types, but a thermal expansion (T-X) valve is most common for air conditioning applications. The device is in the liquid line as close to the evaporator as possible.

Expansion Tank A component in a closed hydronic system that allows the fluid to expand and contract with temperature without damaging the system.

Falling Film Evaporators A special form of flooded evaporators where refrigerant is dispersed over the tubes by trays wetting all the out-tube surface without having to immerse the tubes in a bath of refrigerant. Offer similar performance to flooded evaporators with less refrigerant charge.

Flooded Evaporator For chiller evaporators, the refrigerant is outside the tubes, and all the tubes are submersed in liquid refrigerant.

Friction Head The head or pressure imposed on the fluid being pumped to overcome the losses in the system.

Glide The change in volumetric composition and saturation temperature experienced by Zeotropic refrigerants during boiling.

Hybrid System A chilled water system that uses chillers with more than one primary energy source. For example, electricity and natural gas.

Lift The change, described as pressure or temperature, the compressor must provide in a refrigeration circuit to meet the operating conditions.

Microchannel A type heat exchanger used for condensers and occasionally evaporators made of extruded aluminum.

Open Loop System A hydronic system that has at least two points in contact with atmosphere. E.g. cooling tower loop.

Polyvalent Unit Another name for 4-pipe multifunctional unit.

Positive Displacement Compressor Compressors that raise the pressure of a fluid by reducing the volume of space around the fluid. A common example is a scroll compressor.

Pressure (p) Force over unit area. This is what can burst a pipe or expand a balloon. It is measured in Pa (PSI)

Primary-Secondary System A closed chilled water system where the main (secondary) loop is decoupled from the chiller (primary) loop. The secondary loop varies the flow based on load while the primary loop is fixed flow based on the number of chillers operating. Any unused primary flow is returned to the chillers through the decoupler.

Proportional – Integral – Differential (PID) A control method used for modulating control.

Pump A device used in a hydronic system to move fluid throughout the distribution system.

Reversing Valve A refrigerant component that is used to redirect the refrigerant flow in an ASHP to switch it from heating to cooling mode.

Reynolds Number (Re) A dimensionless parameter that determines the behavior of viscous flow patterns. Can be used to determine laminar vs. turbulent flow.

Saturated Condensing Temperature The saturation condition that exists in the condenser.

Seasonal Total Efficiency Ratio (STER) A seasonal version TER, similar to how SCOP (seasonal COP and SEER (Seasonal Energy Efficiency Ratio) work with standard heatpumps.

Semi Hermetic Compressor A compressor and motor in a common gas tight enclosure that has service access ports. A common example is a screw compressor.

Sensible Heat Energy that causes a substance's temperature to change.

Saturated Suction Temperature The saturation condition that exists in the evaporator.

Specific Heat (cp) The energy needed to raise the temperature of a unit of mass one degree of temperature. Measured in J/kg-K (Btu/lbm°F).

Static Head The pressure created by the weight of a column of fluid.

Temperature (T) Represents the average motion of the molecules in the fluid. It is measured in °C (°F).

Thermal Conduction Heat transfer where two or more substances are in contact with each other and have different temperatures.

Thermal Expansion (T-X) Valve A pressure-regulating valve used in refrigeration systems to lower the liquid refrigerant pressure from the condensing pressure to the evaporation pressure. Thermal expansion valves modulate refrigerant flow based on superheat in the suction line.

Thermal Convection Heat transfer between a moving fluid and another substance at a different temperature.

Total Efficiency Ratio (TER) A measure of a 4-pipe multifunctional (polyvalent) unit efficiency during simultaneous heating and cooling. It is based on (cooling capacity + heating capacity)/Total Power

Total Heat of Rejection The energy rejected by the condenser including the energy absorbed by the evaporator, the compressor work and any other form of heat that was added to the refrigerant.

Variable Primary Flow System A closed chilled water or hot water system where the water flow is variable based on load through all components including the heating/cooling unit refrigerant to water heat exchangers. There is only one set of pumps for the system.

Variable Primary, Variable Secondary Flow System A closed chilled water or hot water system where the water flow is variable based on load through all components including the heating/cooling unit refrigerant to water heat exchangers. There are variable flow primary pumps and variable flow secondary pumps.

Velocity Head The kinetic energy of water flowing in a pipe equal to $V^2/2g$.

Volume (V) The geometrical space occupied by a fluid. It is measured in Liters (l) or cubic feet or ft^3.

Volumetric Efficiency (ŋva) Measure of compressor performance defined as actual volume flow rate/ideal volume flow rate.

Water Brake Horsepower The theoretical power required by a 100% efficient pump to meet the flow and head requirements of a system.

Zeotropic Refrigerants Refrigerant blends where the components do not behave as one substance. Zeotropic refrigerants experience glide. An example is R-407C.

Zone A portion of the building identified and used to calculate the heating and cooling loads.

Appendix B — Conversion Factors

Volume

Convert To / Convert From	US gallons	Imperial gallons	Cubic inches (in^3)	Cubic feet (ft^3)	Liters (l)	Cubic meters (m^3)	Pound (lb) water 60 °F (15.6 °C)	US tons water 60 °F (15.6 °C)	Kilogram (kg) water 60 °F (15.6 °C)
US gallons	1	0.8327	231	.13368	3.7854	.0037854	8.38	0.00417	3.782
Imperial gallons	1.20094	1	277.39	.16054	4.546	.004546	10.0134	0.005	4.542
Cubic inches (in^3)	0.004329	0.003605	1	0.0005787	0.016387	0.000016387	0.036095	55409	0.016372
Cubic feet (ft^3)	7.48052	6.229	1728	1	28.317	0.02832	62.3714	0.03119	28.291
Liters (l)	0.2642	0.22	61.024	0.035315	1	0.001	2.2029	0.0011	0.1000
Cubic meters (m^3)	264.2	220.0	61024	35.315	1000	1	2202.65	1.10133	1000.00
Pound* (lb)	0.1199	0.09987	27.71	0.016033	0.4539	0.000454	1	0.0005	0.45359
US tons*	239.87	199.7	55409	32.066	907.9	0.908	2000	1	907.2
Kilogram* (kg)	0.2644	0.2202	61.08	0.03534	1.000	0.001	2.205	0.0011	1

Volume Flow Rate

Convert To / Convert From	US gal/min (USgpm)	Imp gal/min	ft^3/s	ft^3/min (cfm)	l/s	m^3/s	m^3/min
US gal/min	1	0.8327	0.00223	0.13368	0.0631	0.00006309	0.003785
Imp gal/min	1.201	1	0.002676	0.16054	0.0758	0.00007577	0.004546
ft^3/s	448.83	373.7	1	60	28.32	0.02832	1.699
ft^3/min (cfm)	7.4805	6.2288	0.01667	1	0.47195	0.00047195	0.02832
l/s	15.85	13.2	0.0353	2.1189	1	0.001	0.06
m^3/s	15852	13200	35.35	2118.9	1000	1	60
m^3/min	264.2	220	0.5886	35.3147	16.667	0.01666	1

Energy

Convert To Convert From	Btu	$ft \cdot lb_f$	Calorie (cal)	Therm	Joule (J) watt-sec (W·s)	Watt-hour (W·h)	Kwh	Quad
Btu	1	778.17	251.9958	0.00001	1055.056	0.293071	0.0002931	1.0×10^{-15}
$ft \cdot lb_f$	0.0012851	1	0.32383	128.51	1.355818	0.000376616	0.000000376616	1.285×10^{-18}
Calorie (cal)	0.0039683	3.08803	1	3.96926×10^{-8}	4.1868	0.001163	0.000001163	3.968×10^{-18}
Therm	100000	77816937	25199580	1	105505600	29307.1	29.3071	1×10^{-10}
Joule (J)	0.00094782	0.73756	0.23885	9.4804×10^{-9}	1	0.00027778	0.00000027778	9.478×10^{-19}
Watt-hour (W·h)	3.41214	2655.22	859.85	0.0000341296	3600	1	0.001	3.412×10^{-15}
Kwh	3412.14	2655220	859850	0.0341296	3600000	1000	1	3.412×10^{-12}
Quad	1×10^{15}	7.7817×10^{17}	2.51996×10^{17}	1×10^{10}	1.05506×10^{18}	2.930722×10^{14}	2.930722×10^{11}	1

Area

Convert To Convert From	in^2	ft^2	yd^2	acre	mm^2	cm^2	m^2	hectare (ha)
in^2	1	0.00694	0.000772	1.5942×10^{-7}	645.16	6.4516	0.00064516	6.4516×10^{-8}
ft^2	144	1	0.11111	0.00002296	92903	929.03	0.092903	0.0000092903
yd^2	1296	9	1	0.0002066	836127	8361.3	0.8361	0.00008361
acre	6272640	43560	4840	1	4046856422	40468564	4046.9	0.4047
mm^2	0.00155	0.00001076	0.0000011959	2.47105×10^{-10}	1	0.01	0.000001	1.0×10^{-10}
cm^2	0.155	0.001076	0.00011959	2.47105×10^{-8}	100	1	0.0001	1.0×10^{-8}
m^2	1550	10.76	1.1959	0.000247105	1000000	1000	1	0.0001
hectare (ha)	15500031	107639	11959.9	2.471	1.0×10^{10}	1.0×10^{8}	10000	1

Pressure

Convert To Convert From	lb/in^2	lb/ft^2	in. of water 68°F (20°F)	ft. of water 68°F (20°F)	in. mercury 32°F (0°C)	atmosphere	mm. mercury 32°F (0°C)	bar	kg/cm^2	Pa	kPa
lb/in^2	1	144	27.7276	2.3106	2.0360	0.068046	51.715	0.068948	0.07030696	6894.8	6.8948
lb/ft^2	0.0069444	1	0.1926	0.01605	0.014139	0.000473	0.35913	0.000479	0.000488	47.9	0.0479
in. of water 68°F (20°F)	0.036092	5.1972	1	0.08333	0.07343	0.002454	1.8651	0.00249	0.00253	249	0.249
ft. of water 68°F (20°F)	0.432781	62.3205	12	1	0.88115	0.029449	22.3813	0.029839	0.03043	298.39	2.9839
in. mercury 32°F (0°C)	0.491154	70.7262	13.6185	1.1349	1	0.033421	25.400	0.033864	0.034532	3386.4	3.3864
atmosphere	14.696	2116.22	407.484	33.957	29.921	1	760.0	1.01325	1.03323	101325	101.325
mm. mercury 32°F (0°C)	0.0193368	2.7845	0.53616	0.04468	0.03937	0.0013158	1	0.001333	0.0013595	133.32	0.13332
bar	14.5038	2088.55	402.156	33.513	29.530	0.98692	750.062	1	1.01972	10000	10
kg/cm^2	14.223	2048.16	394.38	32.865	28.959	0.96784	735.559	0.980665	1	9806.65	9.80665
Pa	0.00145038	0.020886	0.00402156	0.00033513	0.0002953	0.0000098692	0.00750062	0.00001	0.0000101972	1	0.001
kPa	0.145038	20.886	4.02156	0.33513	0.2953	0.0098692	7.50062	0.01	0.010972	1000	1

Length

Convert To Convert From	inches (in)	feet (ft)	Yards (yds)	millimeter (mm)	meter (m)
inches (in)	1	0.0833	0.0278	25.4	0.0254
feet (ft)	12	1	0.3333	304.8	0.3048
Yards (yds)	36	3	1	914.4	0.9144
millimeter (mm)	0.03937	0.0032808	0.0010936	1	0.001
meter (m)	39.37	3.2808	1.0936	1000	1

Power

Convert To / Convert From	Btu/h	ft-lb$_f$ /min	Ton refrigeration	horsepower (hp)	horsepower (boiler)	Watt (W)	kilowatt (kW)	megawatt (mW)
Btu/h	1	12.9695	0.00008333	0.000393	0.000029876	0.29307	0.00029307	2.9307 x 10^{-7}
ft-lb$_f$ /min	0.077104	1	0.0000064253	0.0000303	0.000002304	0.0226	0.0000226	2.26 x 10^{-8}
Ton refrigeration	12000	155633.85	1	4.716177	0.35851	3517	3.517	0.003517
horsepower (hp)	2544.4	33000	0.076165	1	0.0760165	745.7	0.7457	0.0007457
horsepower (boiler)	33475	434107	2.7893	13.155	1	9809.5	9.8095	0.0098095
Watt (W)	3.41214	44.254	0.00028434	0.001341	0.0001019	1	0.001	0.000001
kilowatt (kW)	3412.14	44254	0.28434	1.341	0.1019	1000	1	0.001
megawatt (mW)	3412140	44254000	284.34	1341000	101.938	1000000	1000	1

Mass

Convert To / Convert From	Grain (gr)	Ounce (oz)	Pound (lb)	Short ton (US)	Long ton (UK)	Gram (g)	Kilogram (kg)
Grain (gr)	1	0.002286	0.00014286	0.00000007143	0.00000006378	0.0648	0.0000648
Ounce (oz)	437.5	1	0.06250	0.00003125	0.0000279	28.35	0.028350
Pound (lb)	7000	16	1	0.0005	0.0004464	453.59	0.45359
Short ton	14000000	32000	2000	1	0.89286	907200	907.2
Long ton	15680000	35840	2240	1.12	1	1016000	1016
Gram (g)	15.432	0.03527	0.002205	0.0000011	0.0000009842	1	0.001
Kilogram (kg)	15432	35.274	2.20462	0.001102	0.0009842	1000	1

Mass Flow Rate

Convert To Convert From	lb/min	lb/h	kg/s	kg/h
lb/min	1	60	0.007559	27.216
lb/h	0.016667	1	0.000126	0.453592
kg/s	132.28	7936.64	1	3600
kg/h	0.03674	2.2046	0.0002777	1

Temperature

Convert To Convert From	Kelvin (K)	Celsius (°C)	Rankine (°R)	Fahrenheit (°F)
Kelvin (x K=)	1	x-273.15	1.8x	1.8x - 459.67
Celsius (x °C=)	x + 273.15	1	1.8x + 491.67	1.8x + 32
Rankine (x °R=)	x/1.8	(x-491.67)/1.8	1	x - 459.67
Fahrenheit (x °F=)	(x+459.67)/1.8	(x-32)/1.8	x + 459.67	1

Temperature Interval

Convert To Convert From	Kelvin (K)	Celsius (°C)	Rankine (°R)	Fahrenheit (°F)
Kelvin (x K=)	1	1	9/5 = 1.8	9/5 = 1.8
Celsius (x °C=)	1	1	9/5 = 1.8	9/5 = 1.8
Rankine (x °R=)	5/9	5/9	1	1
Fahrenheit (x °F=)	5/9	5/9	1	1

Density

Convert To Convert From	lb/ft³	lb/USgal	g/cm³	kg/cm³
lb/ft³	1	0.133680	0.016018	16.018463
lb/USgal	7.48055	1	0.119827	119.827
g/cm³	62.4280	8.34538	1	1000
kg/cm³	0.0624280	0.008345	0.001	1

Water Hardness

Convert from	to	multiply by
grains/US gal	grams/liter	0.01712

Acceleration

Convert from	to	multiply by
ft/s²	m/s	0.3048
Free fall	m/s	9.8067

Energy/Area Time

Convert from	to	multiply by
Btu/ft²s	W/m²	11348
Btu/ft²h	W/m²	3.1546

Thermal Conductivity

Convert from	to	multiply by
Btu·in/h·ft²·°F	W/m (Kelvin)	0.1442279
Btu·ft/h·ft²·°F	W/m (Kelvin)	1.730735

Overall Heat Transfer Coefficient (U)

Convert from	to	multiply by
Btu/h·ft²·°F	W/(m²·K)	5.678263

Capacity, Displacement

Convert from	to	multiply by
in³/revolution	l/r	0.01639
in³/revolution	mL/r	16.39

Energy/Area

Convert from	to	multiply by
Btu/ft²	J/m²	11356.53

Energy/Area

Convert from	to	multiply by
Btu/ft²	J/m²	11356.53

Energy/Volume

Convert from	to	multiply by
Btu/ft³	J/m³	37258.951
kw/1000cfm	kJ/m³	2.11888

Thermal Expansion

Convert from	to	multiply by
in/100ft	mm/m	0.833

Efficiency

Convert from	to	multiply by
EER	COP	0.293
kW/ton	COP	

Flow/Capacity

Convert from	to	multiply by
USgpm/ton refrigeration	ml/j	0.0179
USgpm/ton refrigeration	l/kW	

Uniform Load

Convert from	to	multiply by
lb_f/ft	N/m	14.5939

Specific Energy (Latent Heat)

Convert from	to	multiply by
ft·lb_f/lb	joule/kilogram (J/Kg)	2.99
Btu/lb	kilojoule/kilogram (kJ/kg)	2.326

Kinematic Viscosity (v)

Convert from	to	multiply by
centristokes	mm^2/s	1
ft^2/s	mm^2/s	92900

Torque or Moment

Convert from	to	multiply by
$ft·lb_f$	N·m	1.355818
$in·lb_f$	mN·m	113
kg·m	N·w	9.807

Specific Heat (c_p)

Convert from	to	multiply by
Btu/lb·°F	kilojoules/ kilogram-Kelvin (kJ/(Kg·K)	4.1868

Dynamic Viscosity (μ)

Convert from	to	multiply by
centipose	mPa·s	1
lb/ft·h	mPa·s	0.4134
lb/ft·s	mPa·s	1490
$lbf·s/ft^2$	Pa·s	47.88026

Power/Area

Convert from	to	multiply by
$Watts/ft^2$	W/m^2	10.76

Humidity Ratio

Convert from	to	multiply by
gr/lb	g/kg	0.143

Light

Convert from	to	multiply by
foot candle	lx	10.76391

Appendix C — Common Equations

I-P Units

Air Side Equations

$Q_{sensible} = 1.085 \times CFM \times \Delta T\ (°F)\ (Btu/h)$

$Q_{latent} = 0.68 \times CFM \times \Delta GR\ (gr/lb)\ (Btu/h)$

$Q_{total} = 4.5 \times CFM \times \Delta H\ (Btu/lb)\ (Btu/h)$

Humidification Equation

$$\text{Humidification} = \frac{CFM \times \Delta GR\ (gr/lb)}{1555}\ (lb/hr)$$

Fan Power Equation

$$BHP_{fan} = \frac{USgpm \times Head\ (ft)}{6356 \times fan\ eff.}\ (HP)$$

Water Side Equations

$Q = 500 \times USgpm \times \Delta T\ (°F)\ (Btu/h)$

$$Q = \frac{USgpm \times \Delta T\ (°F)}{24}\ (Tons)$$

Pump Power Equation

$$BHP_{pump} = \frac{USgpm \times Head\ (ft)}{3960 \times Pump\ eff.}\ (HP)$$

S-I Units

Air Side Equations

$Q_{sensible} = 1.2 \times l/s \times \Delta T\ (°C)\ (W)$

$Q_{sensible} = 3.0 \times l/s \times \Delta W\ (g/kg)\ (W)$

$Q_{sensible} = 1.2 \times l/s \times \Delta H\ (kJ/kg)\ (W)$

Fan Power Equation

$$BHP_{fan} = \frac{(m3/h \times S.P.(Pa))}{(3600 \times 1000 \times fan\ eff.)}\ (kW)$$

Water Side Equations

$Q_{water} = 4.19 \times l/s \times \Delta T\ (°C)\ (kW)$

Pump Power Equation

$$BHP_{pump} = \frac{(l/s \times S.P.(m))}{(102 \times fan\ eff.)}\ (kW)$$

Appendix D — Properties of Propylene and Ethylene Glycol

Freezing Point of Propylene Glycol Solutions						
%	°C	°F	Capacity Correction Factor	Power Input Correction Factor	Flow Correction Factor	Pressure Drop Correction Factor
10	-3.9	25	0.984	0.988	1.02	1.35
20	-6.7	20	0.975	0.982	1.05	1.43
25	-9.4	15	0.972	0.980	1.07	1.47
30	-12.2	10	0.968	0.978	1.09	1.51
40	-20.6	-5	0.962	0.975	1.30	1.58
50	-31.7	-25	0.958	0.974	1.41	1.66

Figure 14.1 | *Propylene Glycol - Freezing Point (% P.G. by weight)*

Freezing Point of Ethylene Glycol Solutions						
%	°C	°F	Capacity Correction Factor	Power Input Correction Factor	Flow Correction Factor	Pressure Drop Correction Factor
10	-3.9	25	0.990	0.994	1.01	1.06
20	-6.7	20	0.981	0.988	1.04	1.12
25	-9.4	15	0.978	0.986	1.06	1.15
30	-12.2	10	0.974	0.984	1.08	1.18
40	-20.6	-5	0.968	0.981	1.14	1.24
50	-31.7	-25	0.964	0.980	1.20	1.30

Figure 14.2 | *Ethylene Glycol - Freezing Point (% E.G. by weight)*

Example

- A 350 kW (100 ton) ASHP will operate in -10 °C (14 °F) ambient.
- Input power is 106.8 kW
- Design Flow rate is 16.7 l/s (264 Usgpm)
- Design Pressure Drop 35 kPa (11.6 ft w.c.)
- Select correct propylene glycol and correct performance

Solution

- The correct amount of propylene glycol for 10 °C (14 °F) ambient is 30%.
- Capacity Correction:
  350 kW x 0.968 = 339 kW
  (100 tons x 0.968 = 97 tons)
- Input Power Correction;
- 106.7 kW x 0.978 = 104.4 kW
- Note while the input power decreased, the capacity decreased more so the COP decreased.
- Pressure Drop Correction:
  35 kPa x 1.51 = 53 kPa
  (11.6 ft. w.c. x 1.51 = 17.5 ft. w.c.)
- To maintain the same water temperature range the flow needs to be increased by;
- 16.7 l/s x 1.09 = 18.2 l/s
  (264 USgpm x 1.09 = 288 USgpm)
- Note that if the flow is increased to maintain the temperature range, then the pressure drop needs to be adjusted for the increased flow.

Appendix E — Recommended Piping Accessories

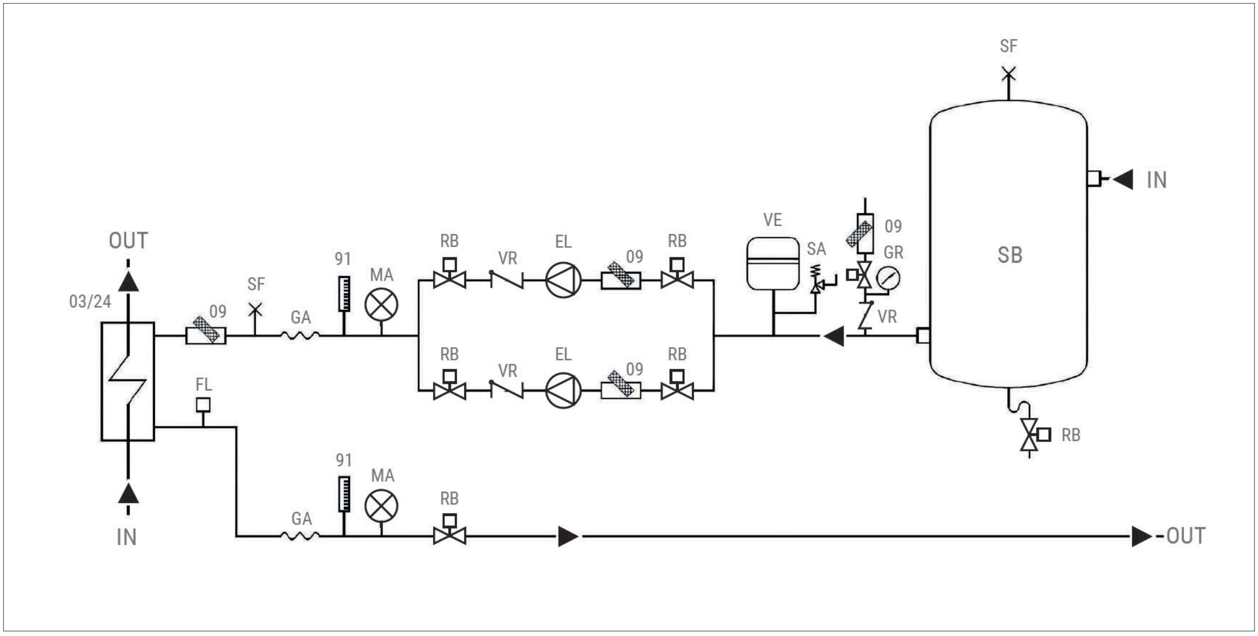

Figure 15.1 | *Recommended Hydraulic Circuit*

	Hydraulic Circuit Piping Accessories[1]
03	Cold-circuit heat exchanger
24	Hot-circuit heat exchanger
09	Water filter strainer
91	Thermometer
EL	Motor-driven pump
FL	Flow switch
GA	Flexible coupling
GR	System filling unit
MA	Water pressure gauge
RB	Manual isolation valve
SA	Temperature / pressure relay
SB	Buffer tank (may be placed on supply or return side)
SF	Air vent
VE	Expansion tank
VR	Check valve

[1] Codes correspond to labeled components in Figure 15.1

Feel good **inside**