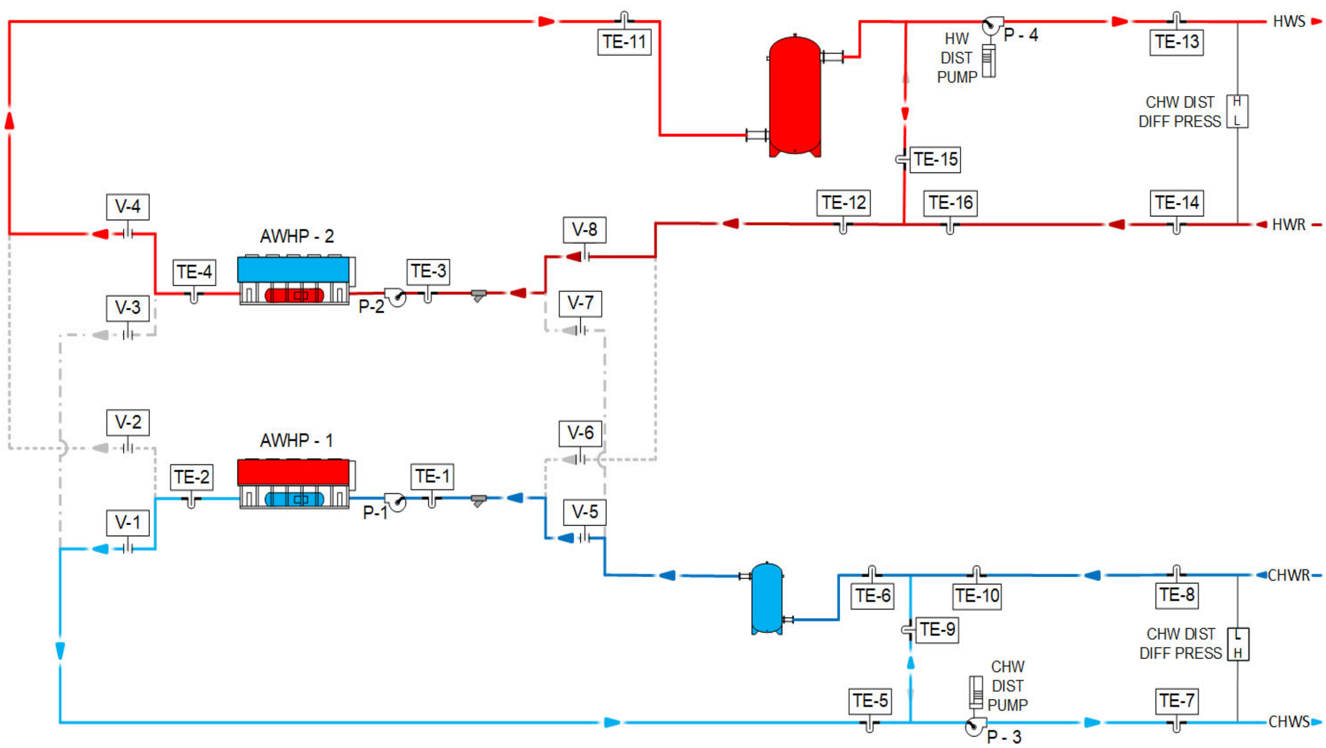
Application Guide

Air-to-Water Heat Pump System with Cascade Option: Part of the Comprehensive Heat Pump Chiller System



⚠SAFETY WARNING

Only qualified personnel should install and service the equipment. The installation, starting up, and servicing of heating, ventilating, and air-conditioning equipment can be hazardous and requires specific knowledge and training. Improperly installed, adjusted or altered equipment by an unqualified person could result in death or serious injury. When working on the equipment, observe all precautions in the literature and on the tags, stickers, and labels that are attached to the equipment.

Preface

As a leading HVAC manufacturer, we deem it our responsibility to serve the building industry by regularly disseminating information that promotes the effective application of building comfort systems. For that reason, we regularly publish educational materials, such as this one, to share information gathered from laboratory research, testing programs, and practical experience.

This publication focuses on air-to-water heat pump hydronic systems for cooling and heating and a water-to-water heat pump cascade option for applications that require higher heating temperatures. This manual discusses system design considerations and options, piping, airside considerations, and system operation and control.

We encourage engineering professionals who design building comfort systems to become familiar with the contents of this guide and to use it as a reference. Architects, building owners, equipment operators, and technicians may also find this publication of interest because it addresses system layout and control.

Trane has a policy of continuous product and product data improvements and reserves the right to change design and specifications without notice. As such all data in this application guide should be considered for reference only, please consult with a Trane® sales associate for current equipment operating range and performance.

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Definitions

The following definitions are used in this discussion of Comprehensive Heat Pump Chiller Systems. Note that these definitions **may or may not** align with their use in other HVAC systems.

Air-to-Water Heat Pump (AWHP): A unit that heats or cools fluid by transferring energy between the fluid and the air via a refrigeration circuit that includes a reversing valve. AWHPs may contain more than one refrigeration circuit and can be configured as a two-pipe or four-pipe unit.

Auxiliary Heat: Heat from an auxiliary source, that operates only when the AWHP is unable to provide sufficient heating capacity due to a machine limitation.

Block Load: A building modeling method that considers the design load profiles and airflows of individual spaces contained in the zone or system to find the collective maximum at any time—also called coincident loads or airflows. For systems with coil or fan sizing displayed as Block, the program will determine fan and/or coil sizes based on this maximum simultaneous load or airflow. “Block” sizing methodology is commonly used for variable volume systems because the airflow can be varied. See also **Sum of the Peaks Load**

Booster: Using a water-to-water heat pump to increase the temperature of partially-heated source fluid to the desired heating application temperature.

Building Automation System (BAS): An energy management system that coordinates overall operation of the building in which it is installed. Example functions include HVAC system control, equipment monitoring, equipment protection from power failure, and building security.

Building Electrification: The process of switching a building’s heating energy source from on-site fossil fuels to electric sources.

Cascade System: Arranging heating machines such that the heat output of one machine is the source heat for the second machine to achieve the desired heating application temperature.

Decarbonization: The process of reducing carbon emissions

Dedicated Heat Recovery (DHR): A unit that can move energy from one distribution stream to the other in only one direction.

Defrost Mode: The operating mode during which the unit is controlled to periodically melt unacceptable accumulation of ice on evaporator tubes (air coil when the AWHP is in heating mode).

Four-Pipe Distribution: A fluid distribution system in which separate piping loops are used to distribute a heating and a cooling fluid. It can deliver heating and cooling to the piping loops simultaneously.

Four-Pipe Unit: A unit that contains connections for four fluid pipes; heating supply and return and cooling supply and return. The unit is capable of heating and cooling the fluids simultaneously. Four-pipe units may also be called multi-pipe units.

Full Heat Recovery Unit: A refrigeration unit with the primary function of providing cooling, where all of its condenser heat is used to satisfy heating loads.

Heat Pump Chiller System: A system that has the flexibility to accommodate a mix of chillers and heat pump units in a common production system.

Heat Recovery: The process of using waste heat from the cooling process for building heating. To be beneficial, this requires a simultaneous demand for cooling and heating in the building.

Intermediate Loop: The fluid loop in a cascaded configuration that connects the low temperature unit's leaving-water temperature with the high-temperature unit's entering-water temperature.

Partial Heat Recovery Unit: A refrigeration unit that primarily provides cooling, but can also satisfy some heating loads using a portion of its condenser heat.

Reversing Valve: A valve that redirects the refrigerant flow such that the evaporator and condenser switch functions in the refrigeration circuit. Heat pumps typically include one reversing valve per refrigeration circuit**.**

Sum of Peaks Load: A building modeling method that determines the fan and/or coil sizes based on the sum of the individual space loads or airflows for spaces contained in the zone or system—also called non-coincident loads or airflows. The individual maximum values do not have to occur at the same time and this method should yield a higher value than the block method. "Sum-of-Peaks" sizing methodology is most commonly used for constant volume systems because the maximum value must be supplied and cannot be varied, this is sometimes called 'peak load'**. See also Block Load**.

Supplemental Heat: Heat from an alternate source that is used to supplement the heat provided by the operating AWHPs.

Two-Pipe Distribution: A fluid distribution system in which the same piping loop is used to distribute either a heating fluid or a cooling fluid. It requires a method of changeover to provide either heating or cooling to the piping loop and cannot provide both simultaneously.

Two-Pipe Unit: A unit that contains connections for two fluid pipes; supply and return. The unit is capable of heating or cooling the fluid, but not both simultaneously.

Water-to-Water Heat Pump (WWHP): A unit that controls to a heating or cooling fluid temperature by transferring heat between two fluid sources.

Introduction to Air-to-Water Heat Pump Systems

EQUIPMENT

An air-to-water heat pump (AWHP) is an air-source refrigeration unit that has the ability to produce either chilled or heated fluid with one refrigerant-to-water heat exchanger (changeover). A refrigerant reversing valve is used to switch between cooling and heating modes.

AWHPs are an emerging equipment category driven by a desire to decarbonize HVAC systems by using electrified heating solutions. Heat pump technology offers a Coefficient of Performance (COP) that far exceeds resistance-based electric heating, enabling a reduced heating energy intensity.

The Trane® ACX air-to-water heat pump is a packaged unit similar to a commercial air-cooled chiller. The ACX is currently available in the range of 140 to 230 tons of nominal cooling capacity, and uses a total of four or six scroll compressors divided into two independent refrigeration circuits. The ACX can deliver up to 140°F fluid temperature at 55°F outdoor air temperature, and is capable of heating operation down to 0°F outdoor air temperature while delivering 90°F fluid temperature.

The Trane® RTWD water-to-water heat pump is available in the range of 80 to 250 tons of nominal cooling capacity, and uses two helical-rotary compressors divided into two independent refrigeration circuits. The RTWD can deliver up to 165°F fluid temperature when using R-515B refrigerant; or up to 140°F fluid temperature when using R-513A refrigerant. This high fluid temperature and precise temperature control makes this unit ideal for boiler replacement and high-temperature applications. The RTWD can operate in different modes to provide heating and cooling simultaneously.

In an air-to-water heat pump system, the RTWD can function as a dedicated heat recovery unit, recovering heat from the cooling load to provide hot water. Or it can be used in a cascade configuration to elevate the ACX leaving-fluid temperature to a hotter fluid temperature.

SYSTEMS

Some design principles from conventional chilled-water systems transfer well to heat pump chiller systems that use AWHP equipment. But many new issues emerge that require consideration and a new way of thinking.

Reliability

A reliable system design is always important, but the consequences of a heating system failure is typically more significant than a cooling system failure. AWHPs have operating limits that become more restrictive as outdoor air temperatures drop. Redundancy and reliable backup heating strategies must be developed. For instance, what happens when that 50-year weather event occurs?

Flexibility

Heat pump equipment must serve two systems with varying expectations. For example, a cooling system may be designed for a 10°F to 12°F ΔT, while a heating system may be designed for a 20°F to 30°F ΔT. That is a substantial difference, and the system must be able to accommodate both needs.

OAT Impact

A heat pump's capacity and maximum supply-fluid temperature are reduced as the outdoor-air temperature (OAT) drops. In addition, the equipment has OAT operating limits. The sizing strategy for the equipment is impacted by the winter design OAT, the dual cooling and heating role of the equipment, and the availability of auxiliary heat sources. Several strategies for sizing system components to deliver reliable, flexible, and cost-effective operation are discussed in this guide.

Defrost

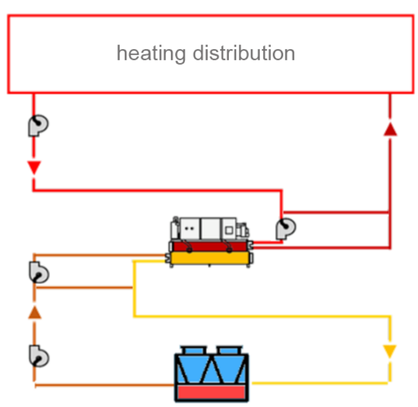
To ensure reliable heat exchange with the ambient air, an AWHP circuit will occasionally need to operate in defrost mode. This results in periodic interruption of sourcing heat from the hydronic system during this defrost cycle. Equipment sizing, buffer tanks, and/or supplemental boilers can all be part of a strategy to mitigate the impact of defrost on the availability of heat.

This guide proposes a system design approach using ACX units and addresses these and other important design considerations to provide a reliable electrified heating and cooling solution.

Air-to-Water Cascade System

An air-to-water cascade system is designed to achieve higher supply-fluid temperatures than traditional AWHPs. This system integrates an ACX air-to- water heat pump with a RTWD water-to-water heat pump, providing supply-fluid temperatures up to 165°F, with future products aiming for >200°F (see Figure 1). The cascade system maintains high performance and stable temperature control, even at ambient temperatures as low as 0°F.

Figure 1. Air-to-water heat pump to water-to-water heat pump cascade arrangement



Understanding Air-to-Water Heat Pump Units

AIR-TO-WATER HEAT PUMP REFRIGERATION SYSTEM

There are various AWHP unit configurations, including two-pipe or four-pipe units, and units with partial or full heat recovery (the later two are also referred to as multi-pipe units).

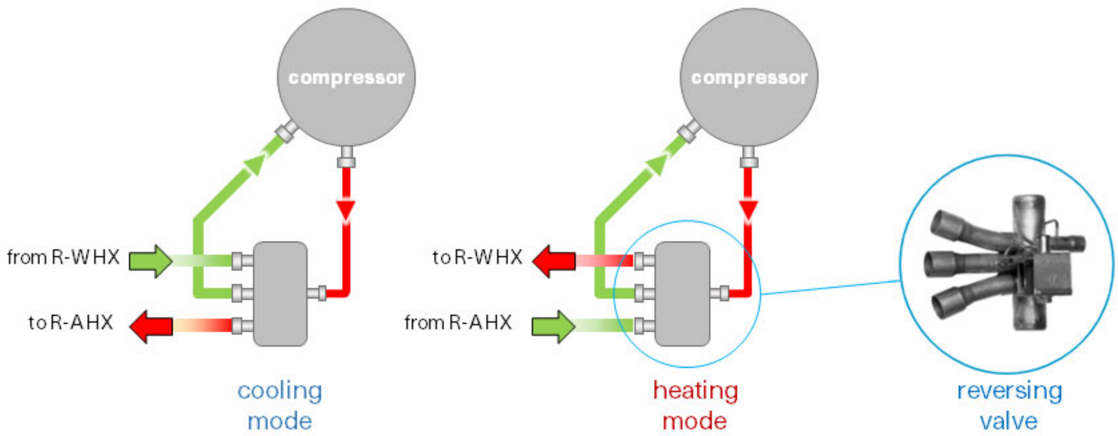
This application guide focuses on the use of two-pipe AWHP units. These units have one refrigerant-to-water heat exchanger that can either cool or heat, but not both simultaneously. Two-pipe units contain two fluid pipes, one for return and one for supply. At any given time, the units operate in either heating or cooling mode, and use outdoor air as the source or sink to provide heating or cooling fluid. “Changeover” between heating and cooling (heat pump) modes is commanded by a BAS or operator via the onboard unit controls.

Figure 2. ACX air-to-water heat pump

It is important to note that the capacity and efficiency of an AWHP can vary significantly depending on the outdoor air temperature and desired leaving fluid temperature. This is particularly true in heating mode, where the operating capacity at a low outdoor temperature (i.e., 0°F) may be 50 percent lower than at a moderate outdoor temperature (i.e., 47°F). This loss of capacity is a normal result of the reduction in refrigerant gas density at low suction temperatures. For that reason, it is critical to select equipment based on actual expected operating conditions, not code-related performance rating points.

The component that enables changeover between cooling and heating modes is a refrigerant reversing valve (i.e., four-way valve). A reversing valve switches the flow of refrigerant through the heat exchangers depending on the operating mode (heating or cooling). As shown in Figure 3, the refrigerant always flows through the compressor in the same direction. The reversing valve controls flow direction through the refrigerant-to-water (R-W) and refrigerant-to-air (R-A) heat exchangers.

Figure 3. Reversing (four-way) valve



AWHP units are applied in systems in two different manners.

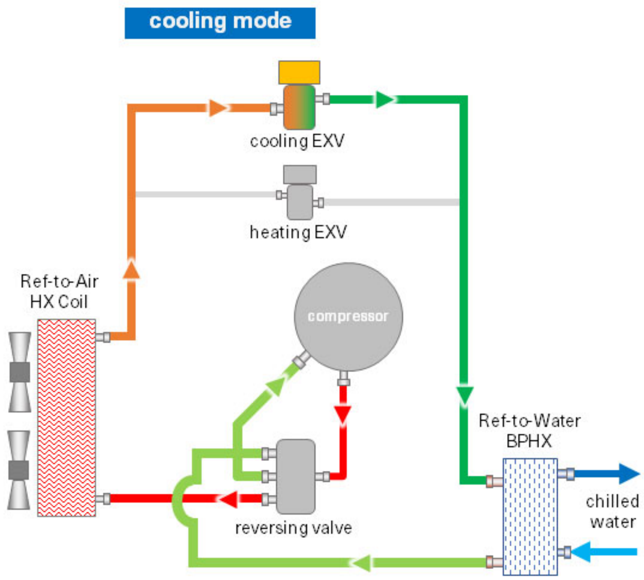
- Changeover cooling/heating (most common): The AWHP unit provides either heating or cooling and changeover between the two modes is commanded by the BAS based on the load demands of the building.
- Dedicated heating only: The AWHP unit operates primarily in heating mode, as cooling mode operation is not required. However, at outdoor air temperatures below 47°F the unit may initiate a *defrost cycle*, which cools the fluid for a short period of time. Again, the BAS commands the unit to operate in heating mode, as required by the heating load demand of the building.

AWHP unit modes of operation

AWHPs can operate in three modes: cooling mode, heating mode, and defrost mode. The figures in this section depict refrigeration circuit operation in each mode, including the respective heat source and heat sink.

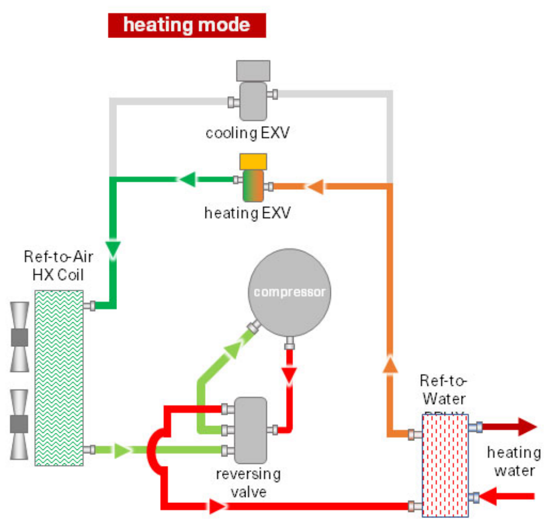
1. Cooling Mode – As shown in Figure 4*,* the refrigerant-to-water heat exchanger is the energy *source* for the refrigeration circuit, absorbing heat from the chilled water. The refrigerant-to-air heat exchanger is the energy *sink,* rejecting heat to ambient air. In this mode, the ACX operation is like that of an air-cooled chiller.

Figure 4. Cooling mode operation: simplified diagram



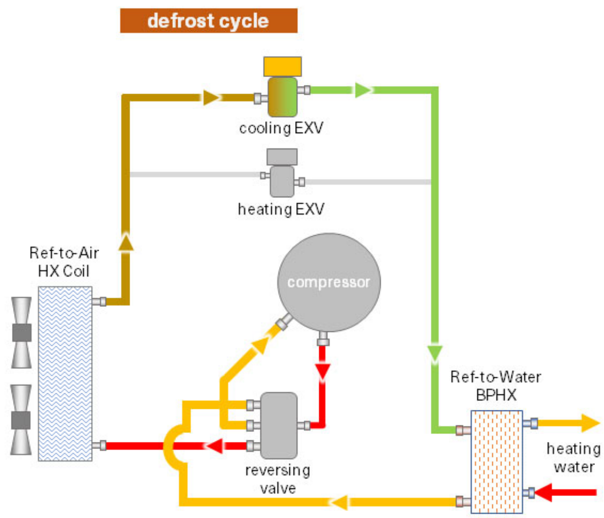
2. Heating Mode – As shown in Figure 5, the refrigerant-to-air heat exchanger is the energy *source* in the circuit, absorbing heat from the ambient air. The refrigerant-to-water heat exchanger is the heat *sink* for the circuit, rejecting heat to the heating water.

Figure 5. Heating mode operation: simplified diagram



3. Defrost Cycle – As shown in Figure 6 the refrigeration circuit is in cooling mode, which melts ice that has built up on the refrigerant-to-air heat exchanger. The return heating water is the heat source needed to defrost the outdoor coil.

Figure 6. Defrost cycle operation: simplified diagram



COOLING/HEATING CHANGEOVER CONTROL

Each AWHP unit mode of operation has a specific permissible range of operation. This includes limits on the minimum and maximum outdoor air temperatures, entering and leaving fluid temperatures, and fluid flow rates. When the system operating conditions are beyond these limits, the unit controls will protect itself by not allowing compressor operation. When designing a system, it is important to understand these operating limits of the specified unit to ensure it can reliably cool and heat as required.

In addition, there may be limits to how frequently the BAS can request the unit changeover between modes, and how long it must operate in each mode before it can be changed back to the previous mode. When switching modes, sufficient time will be needed to allow the system fluid temperatures to moderate enough and allow the unit to start in its new mode of operation. If the BAS rapidly switches the unit from heating mode to cooling mode, or visa-versa, the extreme temperature from the previous mode may cause the unit operational issues in the new mode. For example, if the unit is operating in cooling mode and supplying 42°F chilled water, and its operation is changed to heating mode with a requested setpoint of 120°F, this would initially result in a low condensing pressure which might cause the unit to trip a safety diagnostic to protect itself. In this example, allowing a time duration for the loop to warm up before restarting the heat pump in heating mode could be a way to avoid this trip potential.

The defrost cycle is automatically initiated by the unit controls when frost build-up on the refrigerant-to-air heat exchanger impacts unit performance. The BAS can monitor the unit's mode of operation, or leaving-fluid temperature, to detect defrost mode operation and then initiate operation of an auxiliary heater to mitigate the impact of defrost operation on the supply fluid temperature.

TRANE® MODEL ACX TWO-PIPE AIR-TO-WATER HEAT PUMP (HEAT PUMP CHILLER)

Note: Trane has a policy of continuous product and product data improvements and reserves the right to change design and specifications without notice. As such all data in this application guide should be considered as reference only. Please consult with a Trane sales associate for current equipment operating range and performance.

Trane ACX air-to-water heat pumps are available with the following features:

- Packaged heat pump chiller configuration
- Meets ANSI/ASHRAE/IES 90.1-2022 heating efficiency requirements
- Refrigerant R454B
- Dual refrigeration circuits
- Open-protocol microelectronic controls (Symbio® 800)
- Factory-installed pump package (optional)

Please refer to the Ascend® Air-Cooled Chiller product catalog (AC-PRC002-EN) for more available features and options.

Capacity range:

- Cooling: 140 to 230 tons

Typical ambient operating ranges:

- Cooling mode: 0°F to 125°F OAT
- Heating mode: 0°F to 95°F OAT

Typical fluid temperature range:

- Leaving chilled fluid temperature: 40°F to 68°F
- Leaving heating fluid temperature:
  - Water: 77°F to 140°F
  - With appropriate Glycol: 55°F to 140°F

Typical fluid flow rate range:

- 1.3 to 4 gpm/ton (calculated based on cooling tons at AHRI rated conditions)
  - Corresponding chilled/heated fluid Delta T: 6°F to 18°F

Note: Specific minimum and maximum allowed flow rates vary by unit size. Since AWHP heating capacity varies significantly with outdoor air temperature, please consult the selection program for flow rate and Delta T range.

The above limits are represented in Figure 7 and Figure 8, as operating maps.

Figure 7. ACX operating map, heating mode

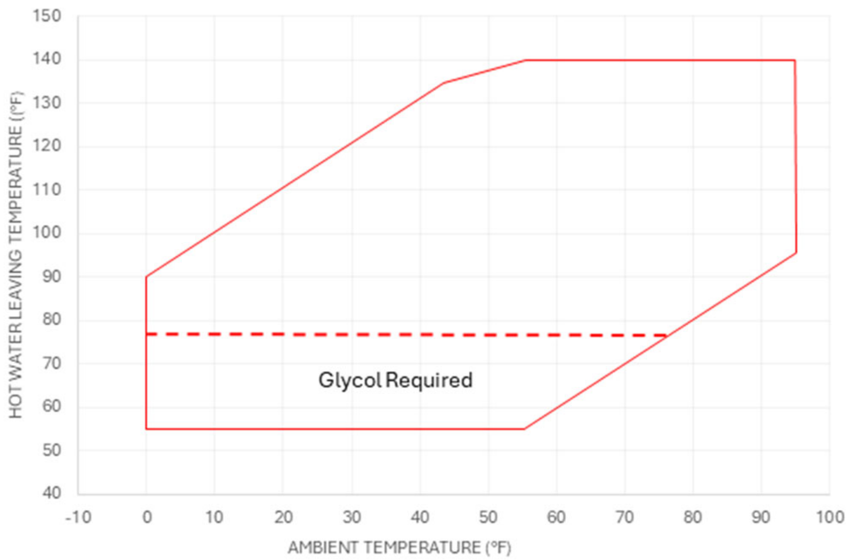
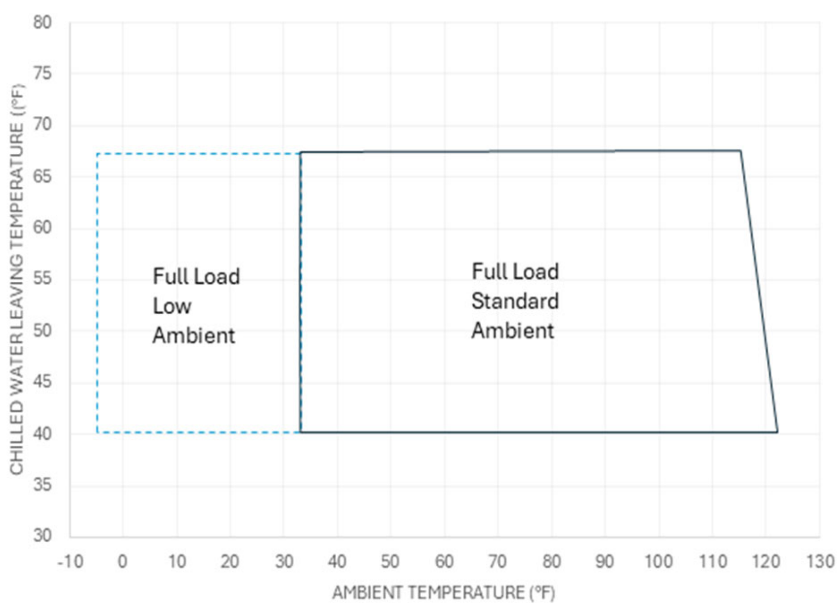


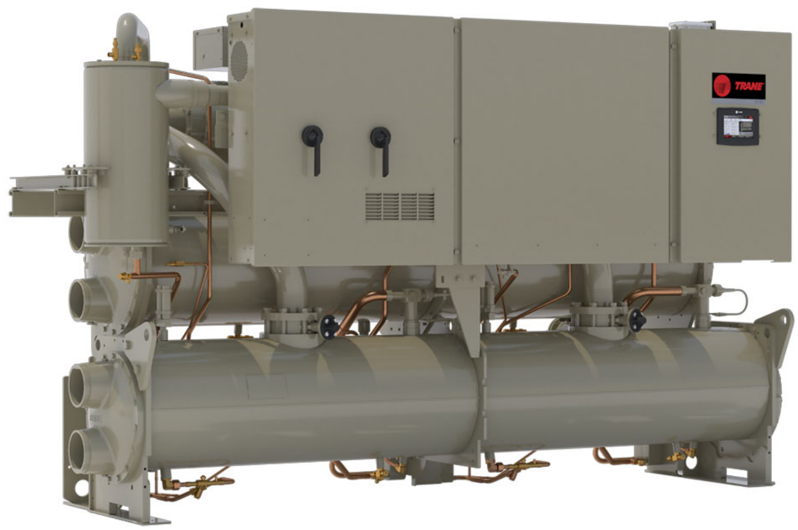
Figure 8. ACX operating map, cooling mode



TRANE® MODEL RTWD WATER-TO-WATER HEAT PUMP CHILLER

The RTWD Series R™ Helical Rotary Screw heat pumps chillers are advanced, reliable, and versatile water-to-water heat pumps designed to meet the evolving needs of building owners (Figure 9). These units offer expanded heating capabilities, precise temperature control, and lower-GWP refrigerants, making them ideal for a wide range of commercial applications. RTWD is engineered to help building owners achieve their sustainability goals by reducing emissions and reliance on natural gas boilers.

Figure 9. RTWD water-to-water heat pump chiller



The RTWD chiller can operate in different modes to provide both heating and cooling needs. In the cascade arrangement, the RTWD is controlled as a heating unit by using the leaving condenser temperature as the control setpoint. It can deliver up to 165°F fluid when using R-515B refrigerant, or up to 140°F when using R-513A refrigerant. These hot fluid temperatures allows this unit to be suitable for boiler replacement and high-temperature applications. The RTWD can also be used as a dedicated heat recovery unit within a heat pump chiller system by recovering heat from the cooling load to provide hot water, thus reducing boiler/hot water heater use when simultaneous heating and cooling loads are present.

Features

- **Reliability:** The Trane helical rotary compressor is a proven design resulting from years of research and thousands of test hours, including extensive testing under extraordinarily severe operating conditions. Trane is the world's largest manufacturer of large helical rotary compressors, with more than 240,000 compressors installed worldwide. The direct drive, low-speed compressors have a simple design with only four moving parts, providing maximum efficiency, high reliability, and low maintenance requirements. The suction gas-cooled motor stays at a uniformly low temperature for long motor life. The electronic expansion valve, with fewer moving parts than alternative valve designs, provides highly reliable operation at high lift conditions.

- **High Performance:** Advanced design enables chilled water temperature control to ±1°F for flow changes up to 10 percent per minute, plus handling of flow changes up to 30 percent per minute for variable flow applications. The two-minute stop-to-start and five-minute start-to-start anti-recycle timer allows tight water temperature control in constant or transient low-load applications. Individual chilled and hot water setpoints for differential to start and stop enable the unit controls to be tailored to a given application.High compressor lift capabilities for use with heat recovery and non-reversible water-to-water heat pump applications allow highly efficient system design with minimal operational concerns.
- **Sound-Reduction Package:** For sound-sensitive applications, an optional acoustical treatment can be factory- or field-installed on the compressors, ensuring quiet operation. The acoustics for a high lift/high temperature water-to-water heat pump are noticeably louder compared to when it operates at standard cooling temperature.

Capacity ranges (R-513A, R-515B):

- **Cooling:** 80-250 tons
- **Heating:** 600-3,000 MBH

Typical fluid temperature ranges:

- **Leaving condenser fluid temperature:** 60°F to 165°F (R-515B); 60°F to 140°F (R-513A)
- **Leaving evaporator fluid temperature:** 10°F to 65°F

Typical fluid flow rate range:

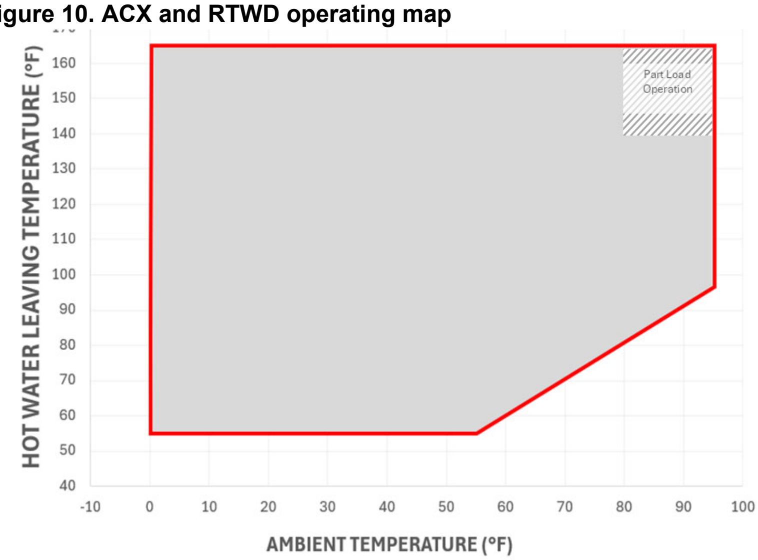
- **Evaporator:** 0.6 to 3.5 gpm/ton
- **Condenser:** 0.7 to 4.5 gpm/ton

Typical compressor lift limits

- **Maximum lift:** 100°F (R-513A); 125°F (R-515B)

When the ACX and the RTWD are configured in a cascade system, the operating map expands significantly, such that the system can produce 165°F hot-water leaving temperatures at all ambient temperatures at 0°F and above (see Figure 10). At higher ambient conditions in the upper right corner of the operation map, the system will need to be operating in part load due to concerns with the intermediate loop getting to warm for the RTWD evaporator temperature limits.

Figure 10. ACX and RTWD operating map



Codes and Standards Considerations

ANSI/ASHRAE/IES Standard 90.1-2022, *Energy Standard for Sites and Buildings Except Low-Rise Residential Buildings* contains requirements that apply to both the products and system designs contained in this application guide.

Minimum equipment efficiencies

Table 6.8.1-16 (Heat Pump and Heat Recovery Water-Chiller Packages-Minimum Efficiency Requirements) defines minimum efficiency requirements for both ACX and RTWD models. In this system, the ACX is considered an “air-source” heat pump unit, and is subject to both heating and cooling efficiency requirements. The RTWD is either considered a liquid source heat recovery unit when applied in a dedicated heat recovery position or a liquid source heat pump boost unit when applied in the cascade position. In both cases, only the heating efficiency ratings apply since the unit controls to the heating setpoint.

Distribution system

ASHRAE 90.1 includes requirements for a two-pipe distribution system, but there are no requirements for a four-pipe distribution system. However, Section 6.5.4.2 essentially requires variable-frequency-driven pump performance if the system has:

- Three or more control valves,
- A combined pump horsepower above a threshold based on climate zone, and
- A pump used for heating or cooling.

In a heat pump chiller system that uses the same pump(s) for both cooling and heating, the more-constraining threshold governs. For this reason, and for better control in all modes of operation, pumps are depicted with variable-frequency drives in all standard configurations.

Note: This “Variable Flow” requirement does not imply that a variable primary flow (VPF) directly-pumped system configuration must be used. This requirement can be satisfied by using a decoupled (primary-secondary) system with variable-flow distribution or a decoupled system with variable-primary/ variable-secondary distribution. The recommended system configuration for the AWHP chiller-heater system is to decouple the production and distribution loops to enhance flexibility of operation and simplify control.

Chilled- and hot-water reset controls

Section 6.5.4.4 requires automatic reset of fluid temperature setpoints if the design capacity of the system exceeds 300,000 Btu/h (25 tons). Systems served by district cooling plants, or that include thermal energy storage, are exempt from this requirement, as are systems that serves a process requiring a specific temperature or systems that use pump pressure optimization based on critical valve position.

Condenser heat recovery exemption from economizer requirement

Exception 5 in Section 6.5.1 allows the design engineer to avoid the requirement for an economizer (either air and fluid) if the system employs condenser heat recovery, as defined in Section 6.5.6.2.2. This can be particularly beneficial for systems with distributed heating and cooling equipment or with a dedicated outdoor-air system, systems with limited space or long distances for outdoor-air ductwork, or other reasons why the engineer wants to avoid an economizer (e.g., data center using air-cooled chillers). The minimum heat recovery capacity needed to invoke this exception is 60 percent of the peak heat-rejection load at design conditions or the capacity required to preheat the peak service hot-water draw to 85°F.

Note: This exception only refers to the minimum capacity requirements listed in Section 6.5.6.2.2. Systems using this exception do not need to meet the thresholds listed in Section 6.5.6.2.1 (24-hour operation, 6,000,000 Btu/h of heat rejection, 1,000,000 Btu/h design service water-heating load).

System and Unit Sizing

Proper design of an electrified system includes understanding required operating conditions, proper sizing of heat pump equipment, providing redundancy for reliable and efficient system operation, and allowing for an affordable installed cost.

Commonly-used high-temperature hot-water supply conditions will not only result in higher system energy use, but might not even be attainable when using commonly-available heat pump technologies. Therefore, past assumptions need to be discarded, and the use of low-temperature hot-water supply concepts embraced.

In addition, historic “rules-of-thumb” for sizing heating system capacity should be avoided. Computerized load analysis for a new building, or accurate load history for an existing building, is essential for proper system sizing and meeting the owner’s environmental and financial goals.

BUILDING LOAD EVALUATION

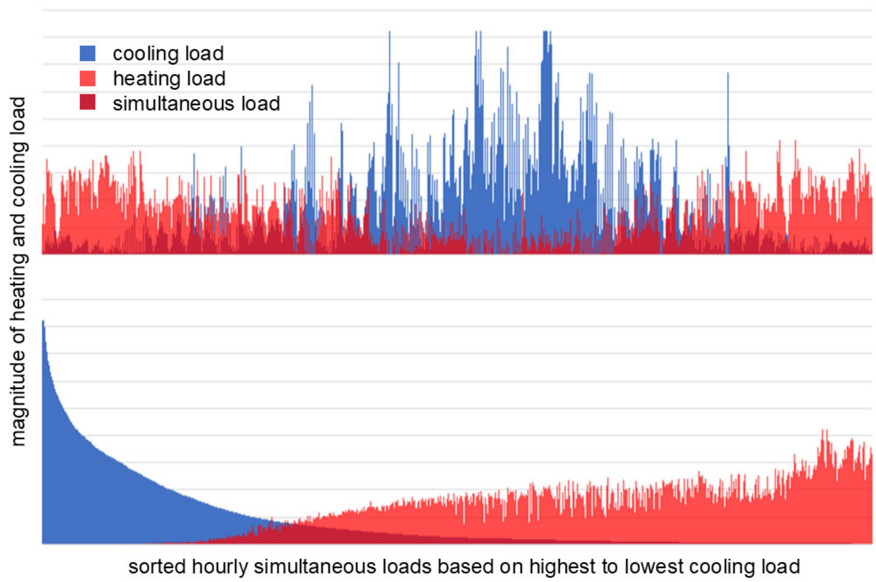
Understanding building loads has always been key to proper equipment sizing, but it is especially important for electrification. Knowing not only the peak loads, but also the simultaneous loads and how many hours are spent at different capacities, allows the system designer to make informed decisions about heat recovery, unloading, redundancy, and resiliency.

Gathering hourly or sub-hourly building load data can be difficult. Early in the design process, using a prototype building model (such as the PNNL EnergyPlus™ models) may be necessary, since there may not yet be enough information to create an accurate building model. As the design progresses, however, energy modeling software can be used to generate an accurate load profile for the proposed building.

In the case of an existing building, it is often best to data log the building to determine actual building loads, rather than relying on a model.

Once a load profile has been generated, one useful way to visualize the data is to sort it by cooling load—largest to smallest—while also showing the simultaneous heating load (Figure 11).

Figure 11. Hourly heating and cooling load plotted chronologically (top chart) and sorted based on highest to lowest cooling load (bottom chart).



Note that ventilation rates often impact cooling and heating loads more than building type or climate. For instance, hotels are characterized by moderate ventilation rates, while large hospitals are characterized by high ventilation rates. Effective use of exhaust-air energy recovery can substantially reduce the ventilation heating load during occupied periods, allowing it to approach the much-lower unoccupied load. The benefit of exhaust-air energy recovery depends on how much exhaust air is available for recovery, and the effectiveness of the energy-recovery technology.See the "References and Resources,"(p. 56) section for additional information.

Additionally, for commercial buildings that use heating and cooling primarily for occupant comfort (no data center or other process loads), there are a limited number of times with simultaneous heating and cooling demands during conditions when economizing is enabled. This reduces the benefit of waterside heat recovery. However, there are other benefits to properly-sized heat recovery, which are discussed in the "System Design" section.

AIR-TO-WATER HEAT PUMP PLANT SIZING

An air-to-water heat pump system needs to be sized to satisfy both the cooling peak load and the heating peak load. That is, the same equipment is expected to satisfy both loads. Not only is the magnitude of these peak loads different, but the capacity of the equipment varies with the outdoor air temperature. The heating capacity of an AWHP, when operating at the design heating ambient temperature, is typically much lower than the unit's "nominal" cooling capacity. This can result in substantially different equipment selections for cooling and heating.

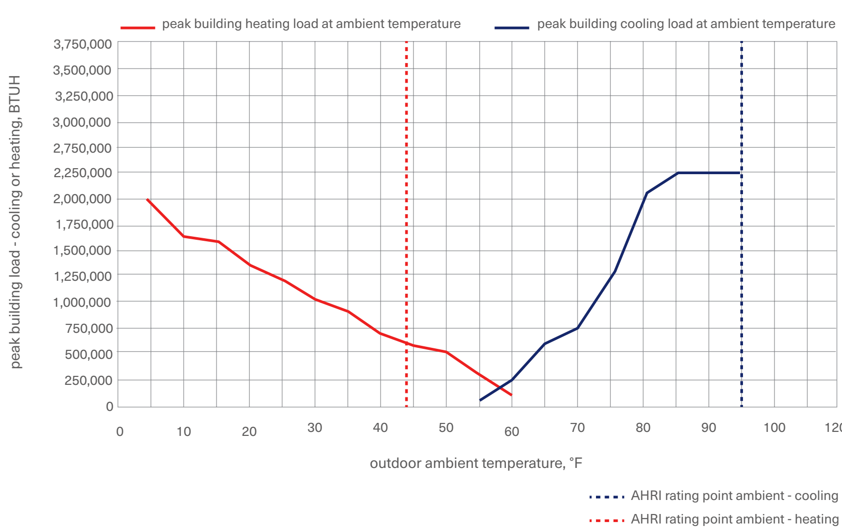
Cooling sizing

Figure 12 graphs the peak loads versus outdoor air temperature for the same school building represented in Figure 11 (p. 17). Not surprisingly, the design operating conditions are not necessarily the same as the standard AHRI rating conditions, so actual performance of the equipment will differ from its rating performance. Design engineers must also remember that the equipment model number's "nominal" capacity often differs from the as-applied capacity, so should not be used for equipment selection.

In this example, the peak heating load is approximately 2,000,000 Btu/h. The ambient temperature for this peak load is 4°F, which is far below the AHRI rating point ambient temperatures (17°F and 47°F). Although the ambient temperature at peak cooling load is the same as the rating point temperature, the design chilled-water supply temperature may be colder (e.g., 40°F to 42°F) than the AHRI rating condition (44°F).

In both cases, the required capacity at the actual (as-applied) design conditions must be determined using the equipment selection software.

Figure 12. Peak building loads versus ambient temperature



Heating sizing

The process for sizing the heating capacity of the heat pump requires the design engineer to consider several factors, some which may be unfamiliar. These factors include, but are not limited to:

- Design heating water supply temperature.
- Design heating outdoor air temperature.
- Equipment cost.
- Operating cost.
- Electrical infrastructure cost to support peak demand.
- Carbon footprint reduction.

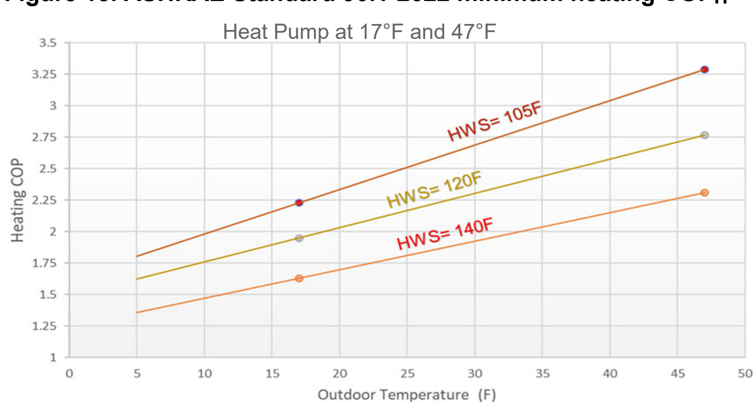
These factors are interrelated, so any trade-off needs to be understood and aligned with the priorities of the building owner.

The design hot-water supply (HWS) temperature is perhaps the most significant factor to consider. Figure 13 plots the ASHRAE Standard 90.1-2022 minimum heating efficiency requirements at common HWS temperatures. The chart linearly extends the required performance across typical heating design outdoor temperatures. This demonstrates the relationship between HWS temperature, outdoor air temperature, and the unit's COP_H (Coefficient of Performance when Heating: a higher COP is better) at full load:

- Lower HWS temperature increases COP_H.
- Colder outdoor air temperature decreases COP_H.

COP impacts operating cost, electrical demand, and carbon footprint. Since the primary driver to electrify a heating system is to reduce carbon footprint, the importance of the HWS temperature is clear. Lower the HWS temperature reduces the negative impact of colder outdoor air temperature and raise the unit's COPH. Note that even lower HWS temperatures than those represented in Figure 13 are feasible in some systems.

Figure 13. ASHRAE Standard 90.1-2022 minimum heating COP_H



Heating temperature impact on carbon emissions

Switching from on-site fossil fuel heating to electrified heating does not guarantee a reduction in atmospheric carbon emissions. In fact, electrified heating can significantly increase carbon emissions. To ensure the system is indeed achieving the goal of decarbonization the design team must ask: What is the electrified heating system efficiency (COP_H) required to reduce atmospheric carbon emissions compared to on-site high-efficiency natural gas (or other fossil fuel) heating?

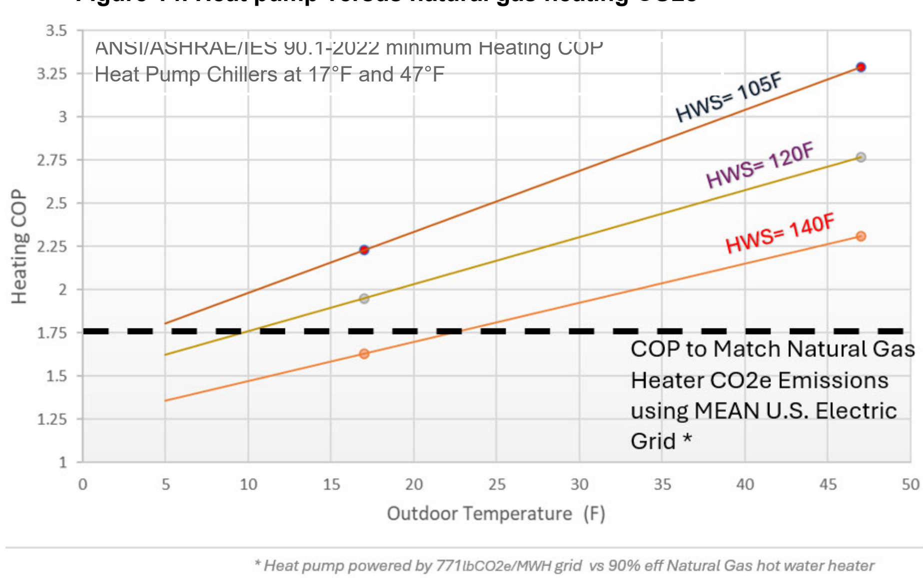
To answer this question, the carbon emission rate of the electric grid must be analyzed. Electrical power production usually comes from a mixture of energy sources, including natural gas, coal, nuclear, hydroelectric, wind turbines, solar arrays, and others. Each has its own carbon emission rate profile. The mixture of energy sources and, therefore, the carbon emission rate of an electric grid varies by location.

Based on the mean carbon emission rate for the U.S. electric grid (771 lb/MWH), the electrified heating system efficiency required to match the emission rate of an on-site natural gas boiler (90 percent efficient) is 1.75 COP_H. This is shown on Figure 14 as a horizontal dashed line. For electrified heating to emit less carbon it must operate above this line.

Note: Electric grid data is sourced from the United States Environmental Protection Agency's (EPA) Emissions & Generation Resource Integrated Database (eGRID).

Depending on the building location, the needed COPH of the electrified heating system could be as high as 3.0 for the most carbon intensive (dirtiest) grids, or as low as 1.5 for the least carbon intensive (cleanest) grids. In general, the higher the system COPH the easier it will be to achieve the goal of reducing carbon emissions.

Figure 14. Heat pump versus natural gas heating CO2e



As explained previously, the lower the HWS temperature the higher the COP_H. As shown in Figure 14, at outdoor temperatures below 10°F with a HWS temperature of 120°F, fossil fuel heating produces less carbon emissions than an air-to-water heat pump (based on the mean U.S. grid carbon intensity and a 90-percent efficient gas boiler: a cleaner or dirtier grid, or a less-efficient boiler, will change this threshold). A higher HWS temperature drives this crossover temperature higher.

To minimize overall carbon emissions, the system should changeover to use gas heat when the outdoor temperature drops below this crossover temperature. As the electric grid becomes "greener" throughout the life of the building, this changeover temperature can be adjusted to maximize environmental performance.

This discussion demonstrates that a single heat source may not provide the lowest carbon emissions at all operating conditions. Therefore, it is important to design and operate the system to minimize carbon emissions rover the course of the year.

SPACE HEATING EQUIPMENT OPTIONS

Of course, the HWS temperature must be considered when selecting heating coils in air-handling equipment. Zone-level (terminal) equipment typically has fewer coil options than central air-handling equipment and likely will drive the final decision for the design HWS temperature. And the average heating coil ΔT must be evaluated to prevent high fluid flow rates from causing excessive pump energy use.

There are numerous options for making use of low-temperature HWS in both centralized and zone-level heating equipment. Many can be applied in existing buildings when considering electrification.

- High-capacity heating coils: For central air-handling units, heating coil options are available that can provide the required heating capacity when using a lower HWS temperature, at an acceptable waterside ΔT (flow) and airside pressure drop.
- Changeover coils: For both terminal equipment and central air-handling units, consider using changeover coil control. That is, using the same coil for both cooling and heating. Cooling coils are typically much larger (more rows) than heating coils. Using the cooling coil for heating also enables the use of a lower HWS temperature, increased waterside ΔT, and low waterside and airside pressure drops. Six-way control valves and high-accuracy pressure independent control (PIC) valves are widely available to enable changeover operation.
- Radiant heaters: Radiant heating systems are commonly designed for a lower HWS temperature, making them a natural complement to AWHP-based systems.
- Hydronic Branch Conductor: This is a valve control unit used in a distributed hydronic heating and cooling system that directs either hot water or chilled water to-and-from an area of the building, based on that area's current need for heating or cooling. It simplifies zone comfort by using the same branch piping

for both cooling and heating, uses dual-purpose (changeover) coils in the terminal units, adapts to varying heating and cooling zone demands, and improves energy efficiency by allowing for the use of a lower HWS temperature. See the “Hydronic Branch Conductor” application guide for more information (APP-APG024*).

What HWS temperature is needed to heat a space? It varies depending on the application. Table 1 summarizes common minimum HWS supply temperatures, and the corresponding ΔT ranges, that can satisfy typical commercial heating applications.

Table 1. Hydronic heating conditions for various airside systems

Equipment	Minimum Heating Water Supply Temperature	Expected System ΔT at Minimum Heating Water Supply Temperature
DOAS air-handling unit	>80°F	20°F to 40°F
central air-handling unit/VAV	80°F to 105°F	18°F to 30°F
single zone VAV air-handling	100°F to 105°F	12°F to 26°F
VAV terminal units (4-row coil)	100°F to 105°F	8°F to 20°F
fan coil units with changeover coil	80°F to 115°F	8°F to 12°F

A dedicated outdoor-air system (DOAS) often includes an energy recovery device for preconditioning the entering outdoor air, and its supply-air temperature during the heating season is typically around 70°F. This allows a DOAS unit to be easily selected with a low HWS temperature and a relatively-high ΔT.

The most challenging application is in-space fan-coil units. This application typically requires the use of a changeover coil. In most cases, the cooling coil in a fan-coil unit, which supplied 55°F air using 45°F chilled water, can changeover to heat the same flow rate of air from 60°F to 95°F using 105°F HWS temperature. Even though the DOAS unit (which is commonly part of a fan-coil system) could use an even lower HWS temperature, this may require significant upsizing of the fan-coils. In general, a HWS temperature of 100°F to 115°F is needed to avoid the need to upsize the fan-coils in an existing building.

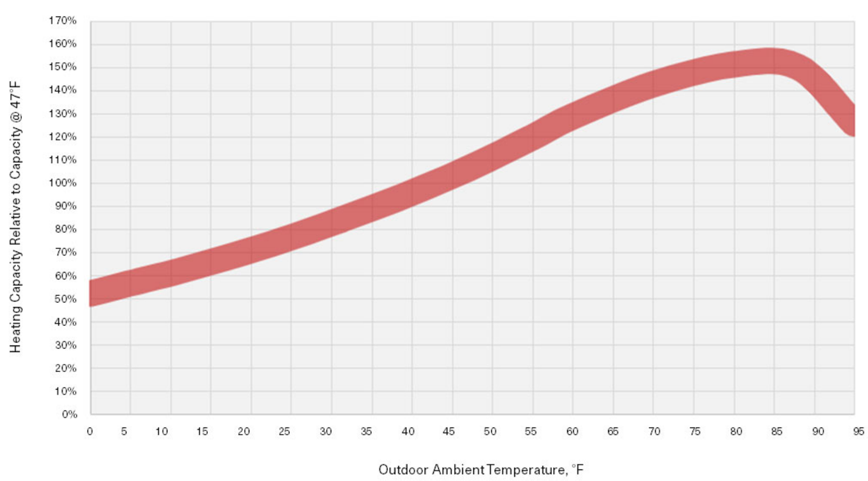
Careful attention to coil selection can allow systems to successfully operate with 100°F to 110°F HWS temperature for any climate. For more information, see the Trane *Engineers Newsletter* titled “Heating with Lower-Temperature Hot Water” (ADM-APN084-EN).

AIR-TO-WATER HEAT PUMP SYSTEM SIZING

Due to the dramatic change in AWHP heating capacity at colder outdoor air temperatures, a careful analysis of unit capacity, sizing, and selection is required.

As noted previously, the outdoor air temperature significantly impacts the full-load heating capacity of an AWHP. The outdoor air temperature used for AHRI rating of heating capacity is 47°F. However, the design outdoor air temperature used for sizing heating equipment is typically much colder than this, resulting in the need for a substantial capacity adjustment. Figure 15 shows the relationship between available AWHP capacity as a function of outdoor air temperature—normalized to the 47°F AHRI rating point. A wide line is used because there is some variation in capacity change between unit sizes. Cold climates can see a 40 to 50 percent reduction in rated heating capacity. The selected hot water supply (HWS) temperature also impacts heating capacity, but in a more limited way.

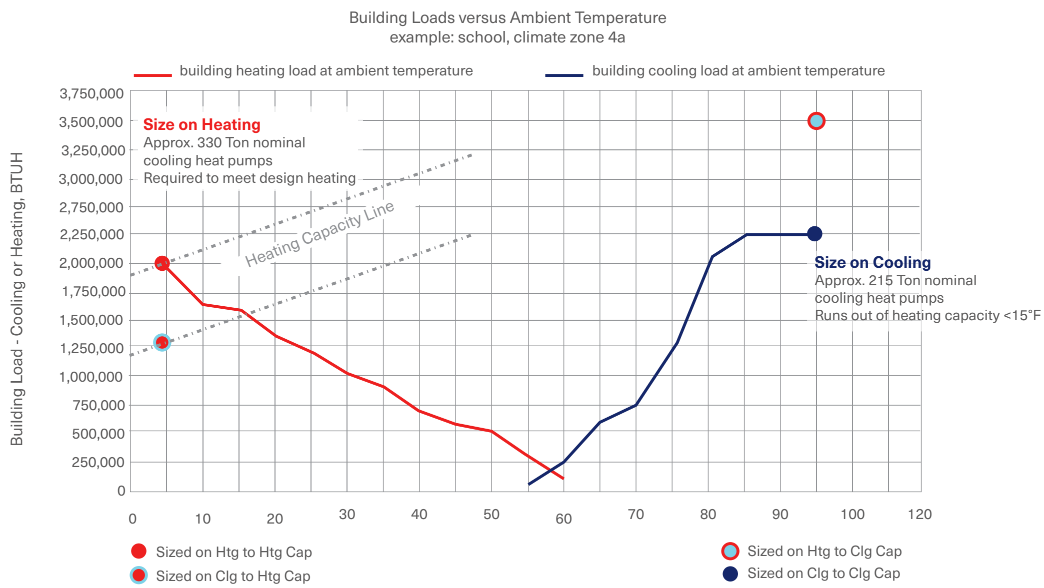
Figure 15. Ambient impact on heat pump capacity



EXAMPLE OF AIR-TO-WATER HEAT PUMP SYSTEM SIZING

Consider the example school building (high ventilation rate) depicted in Figure 11 (p. 17). Figure 15 shows the impact of sizing the AWHP based on cooling design conditions versus heating design conditions..

Figure 16. Sizing AWHP system capacity



- If the system is sized based on design cooling conditions (solid blue dot), system heating capacity (lower dashed line) will fall short of the required heating capacity when it is colder than 15°F outside, due to the impact that cold ambient conditions have on heating capacity. Supplemental heat from an electric or gas boiler can be used to address this capacity shortfall, and may represent the best balance between system installed cost and carbon emission reduction.
- Alternatively, of the system is sized based on design heating conditions (solid red dot), this results in excess (150 percent) cooling capacity (light blue dot with red border), and higher installed cost. When providing an oversized system, it is important to evaluate the system load versus the number of AWHP units operating, to ensure acceptable capacity and flow turndown when operating at lower loads.

Uncertainty regarding the actual building cooling and heating loads, along with potential changing use, often leads design engineers to add safety factors to their design calculations. This tendency seems to be more prevalent as it relates to heating capacity since the consequences of inadequate heat are often more concerning. However, oversizing equipment has consequences with respect to system efficiency, reliable operation, and installed costs. Considering the relatively-high cost of heat pump equipment, it is important to use realistic assumptions and safety factors when estimating building cooling and heating loads.

Use a detailed, computerized load analysis and system “block loads,” not the “sum of peaks” coil loads.

An important design consideration is that the majority of heating operating hours in commercial buildings are at low loads. It is important to evaluate the system for adequate turn-down capability to address numerous hours at lower loads.

AIR-TO-WATER HEAT PUMP SELECTION

Final equipment selection should consider the impact of defrost operation of heating capacity of the AWHP (see defrost implications in Figure 19). For this example, the design outdoor air temperature is 4°F. The defrost derate recommends a 10 percent capacity increase to meet the required design heating capacity.

Packaged equipment is typically available in discrete capacity sizes, which typically leads to slightly oversized equipment. Table 2 shows possible unit sizing options for the example school, assuming two equally-sized ACX heat pumps. The closest unit selection that meets the building block load, including the defrost derate, is two 200-ton models. This results in a selection at 120 percent of design block load, which seems reasonable.

Table 2. Plant equipment sizing choices for mechanical schedule

Equipment		Capacity at 4°F Rating Condition	
Quantity = 2	215 Ton ACXA	2 x 1340 MBH = 2680 MBH	Meet Sum of Peaks Sizing (134% design)
Quantity = 2	200 Ton ACXA	2 x 1197 MBH = 2394 MBH	Match Design Load + Defrost (120% design)
Quantity = 2	180 Ton ACXA	2 x 1080 MBH = 2160 MBH	Match Design Load (108% design)
Quantity = 2	160 Ton ACXA	2 x 938 MBH = 1876 MBH	(94% design)
Sizing for defrost		10% defrost derate at 4°F ambient based on Figure 19	

Note: TRACE® sizing sum of peaks = 2,600 MBH at 4°F (30% oversizing)
Maximum block load at 4°F = 2,000 MBH

WATER-TO-WATER HEAT PUMP SIZING FOR CASCADE SYSTEMS

The water-to-water heat pump (WWHP) should be sized for the “block” load of the connected hot-water system. When selecting WWHPs, both the condenser and evaporator capacity and temperature limits are important. Suitable condenser capacity is needed to meet the building’s hot-water load and hot-water supply (HWS) temperature. Then, the required WWHP evaporator load and temperature limits must be considered when selecting the AWHP. If using sets of WWHPs operating at different temperatures, each set should be sized for the corresponding connected “block” load.

Cascade impact to AWHP sizing

AWHPs that are connected in cascade with WWHPs could be downsized in some scenarios. If the AWHP is sized to meet the design cooling load then its capacity would not change. If the AWHP is sized to meet the design heating load, its capacity may be reduced due to several factors. First, the AWHP can be selected for a lower leaving-fluid temperature to meet the evaporator limitations of the WWHP. At this lower HWS temperature, the AWHP will have increased capacity. Second, since the WWHP evaporator load is equal to the design heating capacity minus the compressor input energy (heat of compression), the AWHP can be downsized to meet the lower evaporator load. Third, the lower HWS temperature allows the AWHP to extend it’s operating range down to lower ambient temperatures while still meeting the required temperature limitations of the WWHP evaporator.

AWHP with cascade sizing example

System design parameters:

- Largest “block” cooling load + safety factor = 300 tons
- Design chilled-water supply temperature = 42°F
- Design chilled-water Delta T = 14°F
- Largest “block” heating load + safety factor = 2800 MBh
- Design hot-water supply temperature = 160°F
- Design hot-water Delta T = 15°F
- ASHRAE 99.6 percent winter design weather = 10°F
- Desired plant turndown (in heating) = 10 percent of design load (280 MBh)

This example starts by sizing the WWHP:

1. The system is primary-secondary distribution, so the Delta T through the heat pump does not need to match the secondary distribution Delta T exactly. For this example, primary loop hot-water Delta T was selected for 10°F.
2. Two, equally-sized 220-ton RTWD units have an actual heating capacity of 1555 MBh/each (3110 MBh in total). While this exceeds the design heating load of

2800 MBh, each individual RTWD can turn down to only 544 MBh, which does not meet the turndown requirement of 280 MBh.

3. Instead of using equally-sized units, selecting one smaller and one larger unit would allow for better turndown, while still only requiring installation of two units. One 90-ton RTWD has an actual heating capacity of 830 MBh and is able to turn down to 249 MBh, which exceeds the turndown requirement. Adding one 250-ton RTWD, which has an actual heating capacity of 2016 MBh, gives the plant a total heating capacity of 2846 MBh, which exceeds the design heating load.

Next, size the AWHP. All selections assume 30 percent propylene glycol for freeze protection.

1. Adding up the WWHP evaporator requirements results in a required heating capacity of 152.5 tons (or 1830 MBh). Applying a defrost derate factor of 0.90, the required AWHP capacity is 2033 MBh (1830 MBh/0.90).
2. Two, equally-sized 160-ton ACX units have an actual heating capacity of 1078 MBh/each (2156 MBh in total). This meets the required capacity of 2033 MBh.
3. The actual cooling capacity of the 160-ton ACX units operating at these conditions, and including the effects of glycol, is 128 tons/each (256 tons in total). This is less than the design cooling load of 300 tons.
4. Two, equally-sized 200-ton ACX units have an actual cooling capacity of 160 tons/each (320 tons in total), which now exceeds the design cooling load. The heating capacity of these larger ACX units totals 2836 MBh, which is well over the required heating capacity of 2033 MBh. In this case, the required cooling capacity dictated the size of the AWHP units.

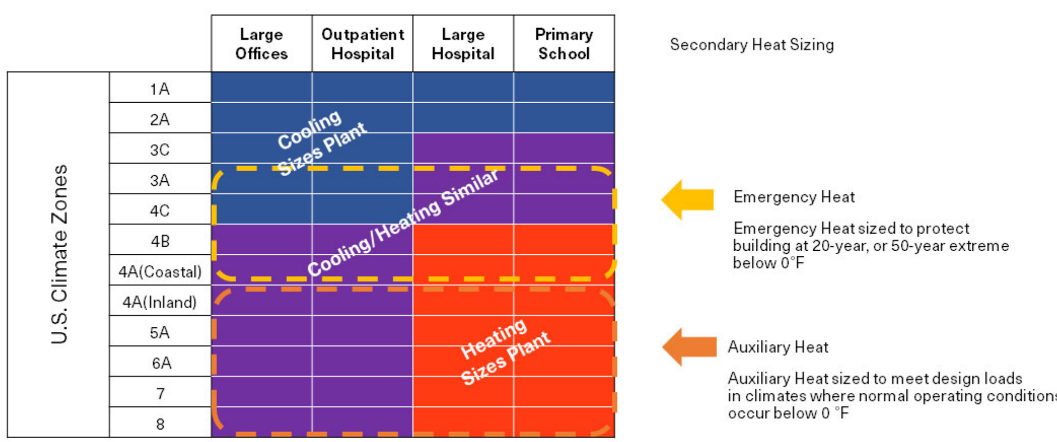
HEAT PUMP CHILLER PLANT SIZING DRIVERS

Building type and climate zone determines the required installed capacity of heat pumps. Using the ASHRAE 90.1-2019 prototype building models, Figure 17 was developed to demonstrate whether the design cooling load or design heating load is expected to dictate the size of a heat pump system. Based on the building type and U.S. Climate Zone:

- Blue indicates that design cooling load is likely to dictate plant capacity.
- Red indicates that design heating load is likely to dictate plant capacity.
- Purple indicates that design heating and cooling loads are relatively balanced.

Figure 17 also demonstrates that systems that serve buildings with higher ventilation rates (such as hospitals and school) are more likely to have the size dictated by design heating loads. This chart also identifies when a secondary heating system (such as a boiler) is likely required for auxiliary heating or emergency heating purposes (climate zone 4A and higher). While not a replacement for specific design work, this is a helpful guide.

Figure 17. Plant sizing basis per climate zones



Trane® Study of ASHRAE® 90.1-2019 Basis Building Models

SIMULTANEOUS HEATING AND COOLING

Dividing the required system capacity between two or more units provides multiple benefits. First, it improves system turndown capability, which is especially beneficial due to many operating hours at lower loads. Second, it provides adequate capacity to meet simultaneous heating and cooling loads without the complexity of heat recovery.

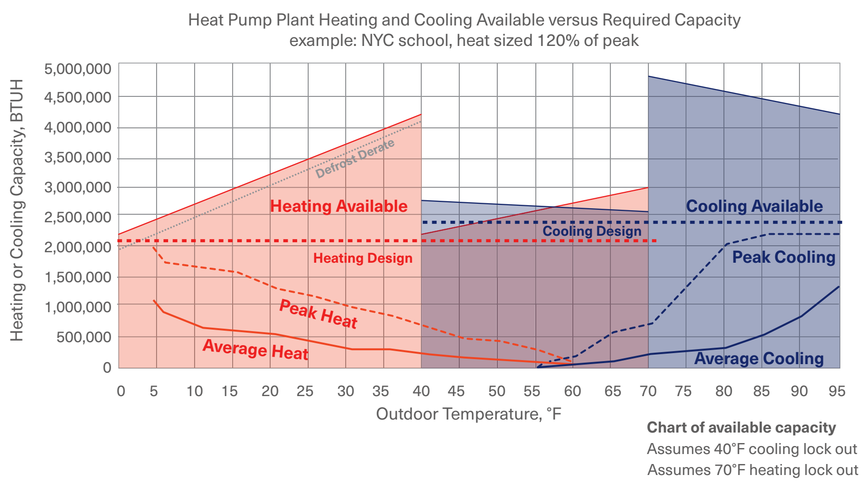
Figure 18 shows the available heating and cooling capacity of a plant with ACX units, at varying ambient temperatures, for the same example school building. The plant is sized for 120 percent of design heating load.

- Cooling is locked out at ambient temperature below 40°F; heating is locked out above 70°F.
- The taller blue region (above 70°F OAT) represents the available cooling capacity with both units operating.
- The taller red region (below 40°F OAT) represents the available heating capacity with both units operating.
- The region between 40°F and 70°F represents the available cooling and heating capacities with just one unit operating. Notice that sufficient capacity is available in this region to satisfy both design heating and cooling loads, with just the one unit operating.

This chart depicts how heat pump capacity changes with outdoor air temperature, represented by the slope at the top of the chart regions. The dashed red and blue lines depict the building heating and cooling loads on a peak day, while the solid lines depict these loads on an average day.

Note that for this example building, simultaneous heating and cooling loads only occur between 55°F and 60°F OAT, and these can both be easily met with one unit operating in cooling mode and the other operating in heating mode. The ACX units operate at a high efficiency in this mild temperature range, so adding a dedicated heat recovery (DHR) unit would provide only a relatively small efficiency benefit. However, a properly-sized (small) DHR unit could eliminate the need to operate two AWHP units, both at low loads, to meet these simultaneous loads. This might extend the life of the AWHP units, as provide some system control and efficiency benefits. Note that the size of the DHR unit would be quite small in this example school: approximately 220,000 Bth/h (18 tons).

Figure 18. Heating/cooling loads and plant capacity vs. outdoor temperature

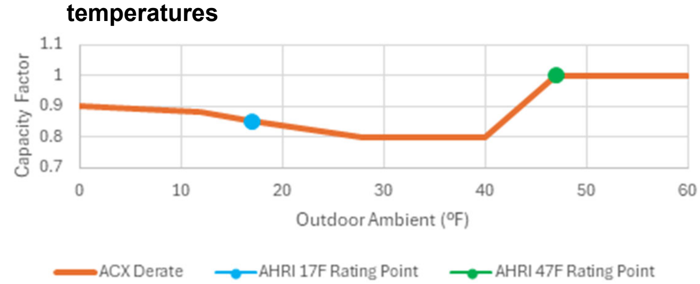


DEFROST IMPLICATIONS TO SIZING

Cold outdoor air temperatures cause the surface temperature of the outdoor coil to drop below freezing, potentially resulting in frost accumulation. Defrost mode typically only occurs at outdoor temperatures below 47°F. An air-to-water heat pump unit will automatically enter defrost mode when it is required. The controller on the Trane® ACX unit has intelligent defrost control to minimize defrost operation, while maximizing unit heating efficiency and capacity. It also limits defrost mode to one circuit at a time, which helps minimize its impact on system temperatures.

Defrost operation results in a weighted performance derate to the equipment heating capacity. Some designer judgment is required since the frequency of defrost operation depends on actual operating conditions. Figure 19 offers a suggested range for the heating derate capacity factor, based on design outdoor air temperature. As discussed in the previous discussion on unit sizing, this derate capacity factor should be applied to the unit's required design heating capacity as shown in the equipment schedule, with notes under the schedule to clarify this has been done. This helps ensure heat pump selections include the proper impact of defrost.

Figure 19. ACX unit heating defrost derate capacity factors at outdoor temperatures



$$\mathit{Scheduled\ Heating\ Capacity} = \frac{\mathit{Building\ Heating\ Block\ Load}}{\mathit{Defrost\ Derate\ Factor}}$$

WEATHER EXTREMES AND AUXILIARY HEAT

Auxiliary heat is defined as heat from an alternate source (other than the AWHPs) that operates only when the AWHPs cannot operate due to outdoor ambient conditions or component failure. If outdoor conditions are cold enough (e.g., below 0°F), an AWHP may not be able to operate. In northern climates, this extreme temperature occurs with normal variations in the weather. In other climate zones, this would be abnormal, but might be expected during a 20- or 50-year extreme weather episode.

Regardless of climate fluctuations, a plan for an auxiliary heating system is required given the operating limitations of heat pump equipment. If full auxiliary heat is to be provided in the design, then sizing for climate extremes is a good design strategy. However, since this auxiliary heating system will be used so infrequently, its impact on annualized carbon emissions is very small; so a high-efficiency, fossil-fuel boiler should not be ruled out.

SUPPLEMENTAL HEAT

Supplemental heat is defined as heat from an alternate source (other than the AWHPs) that operates along with the operating AWHPs. This enables another form of design optimization. As pointed out previously, sizing an AWHP based on the design heating load can lead to substantial oversizing for cooling, higher installed cost, and excessive low-load compressor cycling. One alternative is to use a supplemental heating system to operate alongside “right-sized” AWHP units to satisfy the design heating load. This approach makes particularly good sense if an auxiliary heating system was already required to address 20-, 50- or 100-year extreme weather episodes.

For the same example school building:

- Two nominal 200-ton AWHP units are required to satisfy the design heating load. This results in a 50-percent oversizing of cooling capacity.
- Two nominal 160-ton AWHP unit are required to satisfy the design cooling load, and these units can satisfy all heating loads down to 15°F OAT. Below 15°F, there are only 17 hours/year when the AWHPs would not be able to satisfy the full heating load. For these 17 hours, the supplemental heating system would be energized to satisfy the balance of the building heating load.

Operation of these two heating systems needs to be designed and sequenced so the supplemental heating system does not unintentionally “steal” load from the heat pumps.

REDUNDANCY

Redundancy is a design decision to add equipment that is not required for normal operation, but will be available on standby if other equipment fails or requires service. It is not intended to be a safety factor for capacity because equipment may be down during times when maximum capacity is needed. Redundancy can be provided in several ways:

- If the system design includes an auxiliary heat source, it can provide the required heating redundancy at no additional installed cost.
- One or more (N+1) additional heat pump units can be added to provide redundancy for both heating and cooling.
- Cooling redundancy can be provided by adding a properly-sized, cooling-only chiller. This chiller may even be more efficient than an AWHP (in cooling mode), so it could be used as the lead cooling unit for improved system efficiency. This concept is applicable if the system has an auxiliary heat source or if the AWHP units are sized based on heating load, so there is excess cooling capacity.

Antifreeze

Glycols are used in HVAC systems to prevent damage from corrosion and freezing. Glycol suppliers provide concentration data for freeze protection and burst protection.

Freeze protection indicates the concentration of glycol required to prevent any ice crystals from forming at a given temperature.
Burst protection indicates the concentration required to prevent damage to equipment (e.g. coil tube bursting). Burst protection requires a lower concentration of glycol, which results in less degradation of heat transfer capacity.

BURST PROTECTION

As the temperature drops below the inhibited glycol solution's freezing point, ice crystals will begin to form. Because the water freezes first, the remaining glycol solution is further concentrated and remains fluid. The combination of ice crystals and fluid make up a flowable slush. The system fluid volume increase resulting from the slush formation is absorbed by the expansion tank. The solution never fully freezes and therefore no damage is done to the unit or piping.

Burst protection is usually sufficient in systems that are inactive during winter and have adequate space to accommodate the expansion of an ice/ slush mixture. Given a sufficient concentration of glycol for burst protection, no damage to the system will occur. Burst protection is also appropriate for closed-loop systems which must be protected despite power or pump failure (e.g., an air-cooled chiller that does not need to run during subfreezing weather).

FREEZE PROTECTION

Freeze protection is mandatory in those cases where no ice crystals can be permitted to form or where there is inadequate expansion volume available, for example, a coil runaround loop. Also, HVAC systems that must start-up during cold weather following prolonged winter shutdowns may require freeze protection. However, freeze protection should be specified only when the fluid must remain 100 percent liquid at all times.

For either freeze or burst protection, the required concentration of glycol depends on the operating conditions of the system and the lowest expected ambient or fluid temperature. Often the concentration is selected based on a temperature that is at least 5°F lower than the lowest anticipated design operating temperature. Table 3 is an excerpt from product information bulletins published by The Dow® Chemical Company. It is important that equipment selections are made at the required glycol concentration to ensure proper sizing.

Table 3. Typical antifreeze concentrations by volume

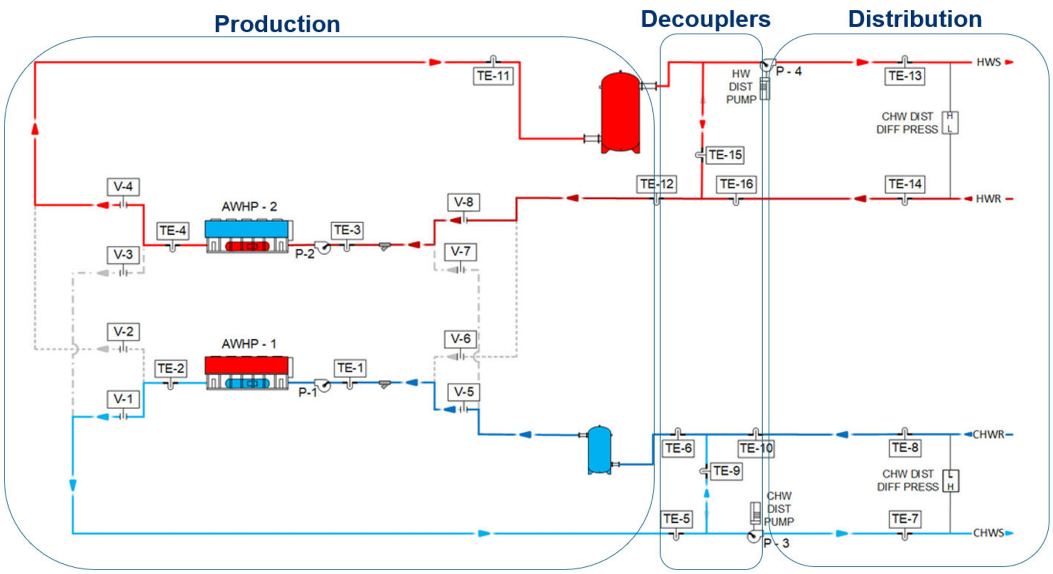
Temperature, °F	DOWTHERM™ SR-1 (ethylene glycol)		DOWFROST™ HD (propylene glycol)	
	Freeze	Burst	Freeze	Burst
20	16.8%	11.5%	18%	12%
10	26.2%	17.8%	29%	20%
0	34.6%	23.1%	36%	24%
-10	40.9%	27.3%	42%	28%
-20	46.1%	31.4%	46%	30%
-30	50.3%	31.4%	50%	33%
-40	54.5%	31.4%	54%	35%
-50	58.7%	31.4%	57%	35%
-60	62.9%	31.4%	60%	35%

Important: Even if a unit is located in a climate which rarely experiences outdoor temperatures below freezing, the unit can still experience freezing conditions due to refrigerant migration.

System Configuration

The base configuration recommended for use with multiple ACX heat pumps—four-pipe distribution with dual-feed chiller-heater system—is represented in Figure 20. While other configurations are possible, this one provides excellent flexibility and simplicity of design and control.

Figure 20. Four-pipe distribution, dual-feed heat pump chiller system



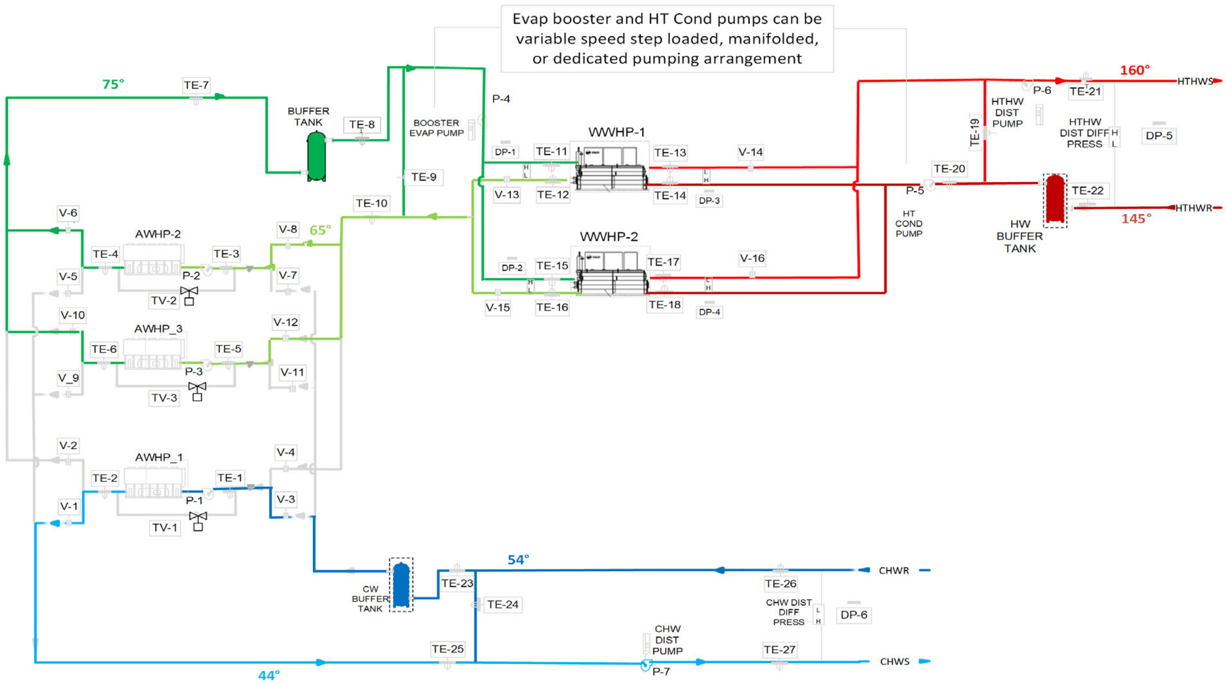
Note: The piping diagrams in this application guide do not include all required hydronic system components. These must be defined and designed into the system by the project's design engineer of record. Examples of possible additional components include expansion tanks, glycol make-up, air separators, strainers, vibration isolators, VFDs, etc.

The The ACX is limited to a maximum HWS temperature of 140°F, but this is only at elevated outdoor temperatures. For example, when it is 0°F outside, the ACX is limited to 90°F HWS temperature. If a higher HWS temperature is required, the base configuration can be altered to include a WWHP booster (Trane RTWD) in a cascade arrangement. This cascade arrangement directs the fluid leaving the AWHP into the evaporator of the WWHP, which then further raises (boosts) the temperature of the supply fluid. Operating with R-515B, the RTWD can achieve a HWS temperature of 165°F; with R-513A, it can achieve 140°F.

Note: New AWHP with vapor-injection technology can extend operation to lower ambient temperatures AND higher HWS temperatures. These units can be included in any of the system configurations discussed. In addition, future WWHP models are expected to achieve HWS temperatures over 200°F. However, the compressor lift is similar to the RTWD, so the evaporator temperature limitations will need to be considered, as this will impact the temperature range of the intermediate loop and possibly the ambient temperature range of the AWHP.

The cascade configuration (Figure 21) maintains many of the features of the base configuration, including the flexibility of using each of the AWHP units in the desired mode.

Figure 21. Cascade configuration



Note: Piping diagrams in this application guide do not include all required hydronic system requirements and these must be defined and designed into the system by the project's design engineer of record. Examples of possible additional components include expansion tanks, glycol make-up, air separators, strainers, vibration isolators, VFDs, etc.

COMMON SYSTEM FLUID

As shown in Figure 20, the system circulates the same fluid through the production loop, and the heating and cooling distribution loops. Oftentimes an antifreeze solution is used in the production circuit due to the AWHPs being located outside, and allows this same solution to circulate through both distribution loops. This is the simplest and safest approach, since it does not rely on any powered or mechanical freeze protection strategy. Fluid isolation heat exchangers can be added between the production loop and either (or both) distribution loops, if the design engineer wishes to avoid antifreeze in the distribution loops. However, realize that fluid at a sub-freezing temperature may enter the heat exchanger from the production loop during a system changeover or a defrost cycle.

In the cascaded arrangement, the hot-water distribution loop is isolated from the production loop via the booster unit, so freeze protection may not be required.

FOUR-PIPE DISTRIBUTION

As the name implies, the system uses four-pipe distribution that consists of separate heating and cooling distribution loops. These loops are typical of many conventional four-pipe systems, and may be optimized for the supply temperatures, flow rates, and Delta Ts as required by the design of the airside system components.

The heating and cooling coils in the airside equipment are controlled using two-way valves, which results in variable flow in both the heating and cooling distribution loops. This provides significant pump energy savings, while also allowing for operational flexibility.

At the heart of the system is the decoupler pipes (Figure 20). These pipes provide hydronic isolation that allows for optimization of flow rates and temperatures in both the distribution and production loops.

DECOUPLING – HYDRONIC ISOLATION

Decoupling greatly simplifies system design and allows for an array of sizes and types of production units to be applied, so as to best match the building load requirements. The principal requirement for the heat pump equipment is that it must produce the required fluid temperature for cooling or heating. The fluid flow rate and pressure drop are of much less of a concern, since the decoupler line allows for natural balancing of flow rates.

The decoupler pipe must be configured and sized to meet the following requirements:

1. Prevent unintended mixing of the return and supply water streams.
2. Provide *adequate* flow and pressure decoupling between the chilled water production and distribution loops.

Figure 22 discusses how to meet these requirements.

Pipe connection tees: The decoupler pipe should be configured so that it enters or exits the side of the return and supply piping via Tee-type connections. This prevents water velocity momentum in the supply or return pipe from inducing flow and/or mixing in the decoupler pipe.

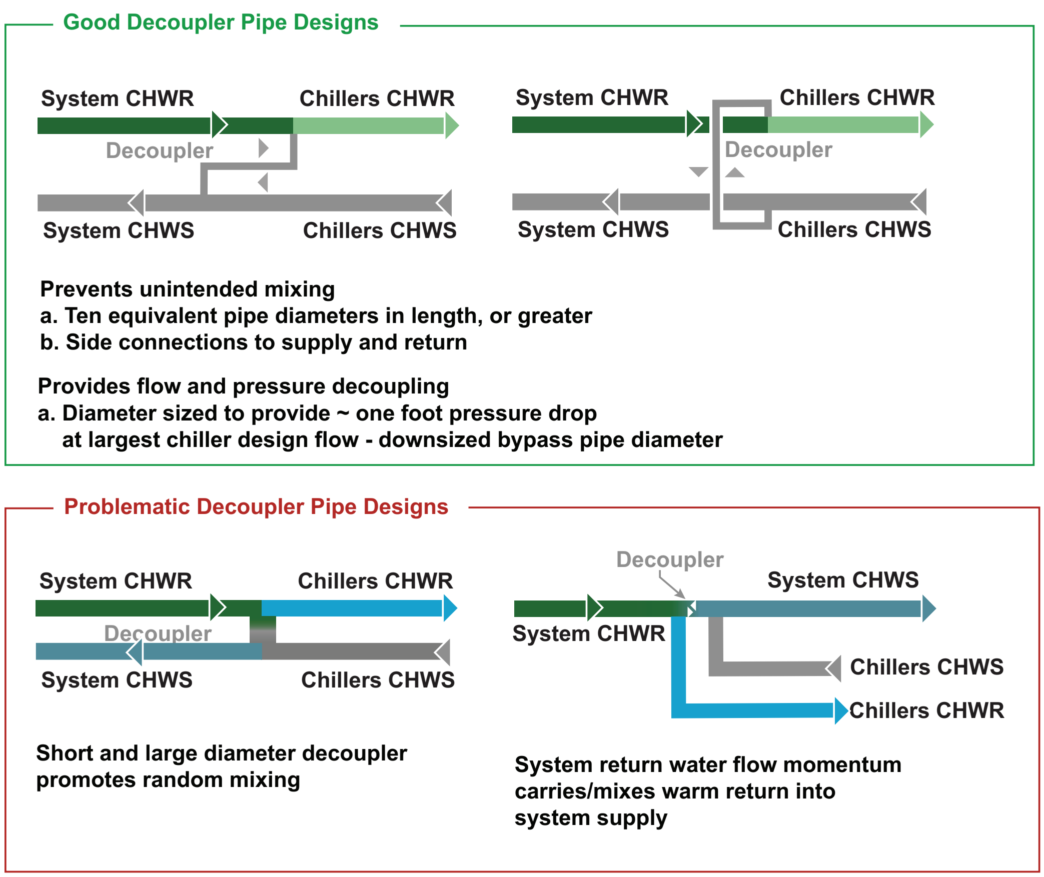
Pipe diameter. The diameter of the decoupler pipe depends on plant operation:

a. For a system with constant flow through the heat pump(s), the decoupler pipe should be sized based on the higher of the cooling or heating design fluid flow rates for the largest unit in the plant. This typically means no larger than— often a size smaller—the diameter of the pipe connected to the largest heat pump.In a system with multiple unit, the decoupler pipe should NOT be sized for the overall **system** design flow rate (i.e., will be smaller that the diameter of the distribution piping).

b. For a variable-primary/variable-secondary system, in which flow varies through the heat pump(s), the decoupler pipe should be sized based on the minimum allowable flow rate for the largest unit in the plant. This typically means one or more pipe sizes smaller than the diameter of pipe connected to the largest heat pump (unless the minimum allowable flow rate for the heat pump is very close to its design flow rate, in which case it may be the same size). A larger diameter is not better, as this increases the likelihood of undesired flow mixing and increases the installed cost.

Pipe length: The length of the decoupler pipe should be at least ten (10) equivalent pipe diameters (elbows counted appropriately). Another rule-of-thumb is for this pipe to have pressure drop of about 1.0 ft. H_2O at the design flow rate through teh decoupler pipe. In large systems, however, a higher pressure drop should not cause operational problems.

Figure 22. Decoupler pipe connection recommendations



Each system is unique, so proper engineering judgment should be used when designing a hydronic system where there is mixing between loops, to avoid potential balancing issues. Pressure differences between heating and cooling distribution loops can lead to water migration from the higher pressure loop to the lower-pressure loop, causing an imbalance in flow rates, loss of efficiency, and potential damage to equipment, such as pumps and valves.

System Fluid Volume

Adequate system fluid volume in the cooling and heating loops is critical to system reliability and comfort.

HEATING LOOP VOLUME

The minimum fluid volume required in the heating loop is typically higher than in the cooling loop, because the heating loop requires more volume to compensate for unit defrost operation. When a heat pump switches into defrost mode, its leaving-fluid temperature drops rapidly. This is most dramatic if only one refrigeration circuit is operating and switches into defrost mode. If the system does not have adequate fluid volume, the following issues can occur:

1. The cold supply fluid can cause the air-handling units or terminal units to deliver cold air into the occupied space, resulting in occupant discomfort.
2. The cold supply fluid can trigger low-temperature alarms or freezestat trips in the air-handling units or terminal units.
3. The temperature of the fluid returning to the heat pump can get cold enough to cause the heat pump to trip on a diagnostic. Mitigating these problems for a unit with two refrigerant circuits, which do not defrost at the same time, requires adequate fluid volume in the heating loop.

The system volume is calculated by summing the piping, coil, and chiller internal volumes. For systems that do not meet the minimum required fluid volume, a volume buffer tank must be installed in the heating supply pipe of the production loop. Note that this differs from typical chilled-water loops, where a buffer tank would typically be installed in the return pipe to the chillers. Locating the buffer tank in the heating supply pipe allows it to moderate both the HWS temperature and AWHP return-water temperature swings that occur during unit defrost cycles.

For the base configuration, which uses only AWHP units, the minimum required fluid volume in the heating production loop is 0.62 gallon/MBh, based on the design heating capacity of the largest AWHP. If additional volume is required, a buffer tank should be installed in the production pipe leaving the AWHP units.

For the cascade configuration, the requirement is different since the AWHP removes heat from the intermediate loop during defrost mode while the WWHP is also removing heat from the same intermediate loop. Therefore, minimum required fluid volume in the intermediate loop is 0.833 gallon/MBh, based on the design heating capacity of the largest AWHP.

In addition, for the cascade configuration, the minimum required fluid volume in the heating production loop (leaving the condenser of the WWHP) is two times the design condenser flow rate of the WWHP, for control stability. If additional volume is required, a buffer tank should be installed in the return pipe to the WWHP condenser.

These values are calculated based on the AHRI heating rating point (120ºF leaving hot-water temperature and 47ºF outdoor air temperature), and assigned to the largest-capacity heat pump in the system. This equates to approximately eight gallons per rated heating ton for the base configuration and approximately ten gallons per rated heating ton for the cascade configuration.

Note: The minimum volume of 0.833 gallon/MBh is based on 10F Delta T at the AHRI rating point. For example, if a larger Delta T is used, the minimum required loop volume can be reduced to ensure the minimum loop time is met with the actual design flow rate.

COOLING LOOP VOLUME

The minimum required fluid volume in the chilled-water loop is two times the design evaporator flow rate (equates to a two-minute circulation time). For a system with a rapidly-changing load profile, this volume should be increased.

If the system volume does not meet this requirement, following are possibly strategies to increase the loop volume and, therefore, reduce the rate at which the return-water temperature changes:

- A buffer tank can be installed in the chilled-water pipe returning to the chiller.
- Increase the diameter of the supply and return header pipes (this also reduces system pressure drop and pump energy use).

Modes of System Operation

Figure 23. Example of base cooling-only system operation

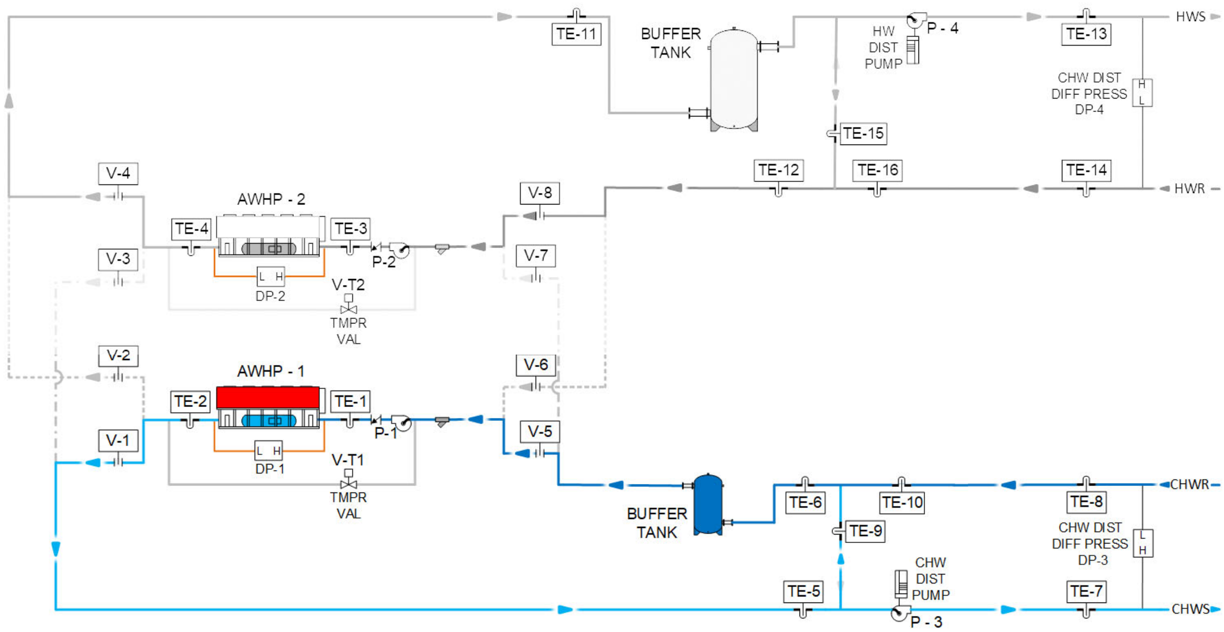


Figure 23 depicts the base system operating in cooling-only mode, with only one unit operating (either unit, or both units, can be operating in this mode to provide cooling). In this mode, the system operates similarly to a conventional chilled-water system.

The chilled-water distribution loop is variable flow, with two-way control valves on the cooling coils and distribution pump (P-3) speed controlled based on the measured fluid differential pressure in the chilled-water distribution loop. Critical-valve, pump-pressure optimization should be implemented, and may be required by many energy codes.

The dual-feed production mode control valves (V-1 and V-2) are positioned to return and supply chilled water through the operating heat pump. AWHP-1 is controlled to supply chilled water at the system's required chilled-water setpoint. Chilled-water temperature reset can be implemented, if appropriate for the chilled-water distribution and airside system design.

The production loop pumps (P-1) can be controlled to a constant flow raterequired for the heat pump's cooling mode, which would be typical if the distribution design flow rate is close to (or less than) the heat pump's minimum allowable flow rate. However, if the heat pump's design flow rate is twenty percent (or greater) higher than its minimum allowable flow rate, consider implementing variable-primary/variable-secondary (VP/VS) flow control logic. VP/VS control will reduce the annual pump energy use, and be easier to control that a variable-primary flow (VPF) configuration.

Figure 24. Example of base heating-only system operation

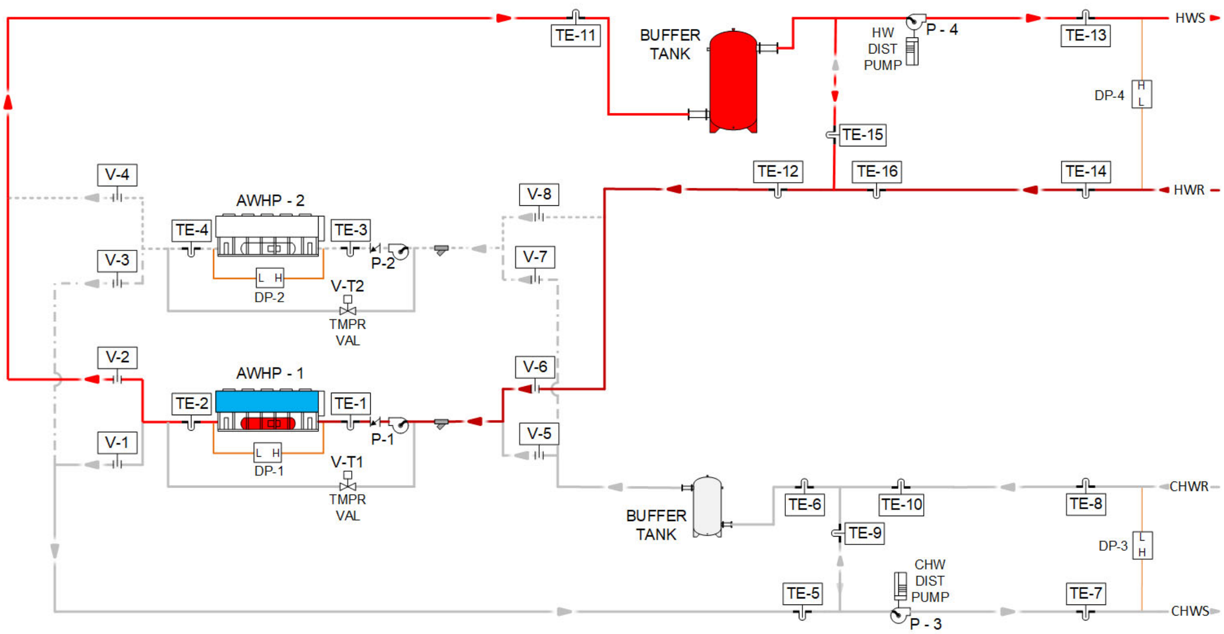


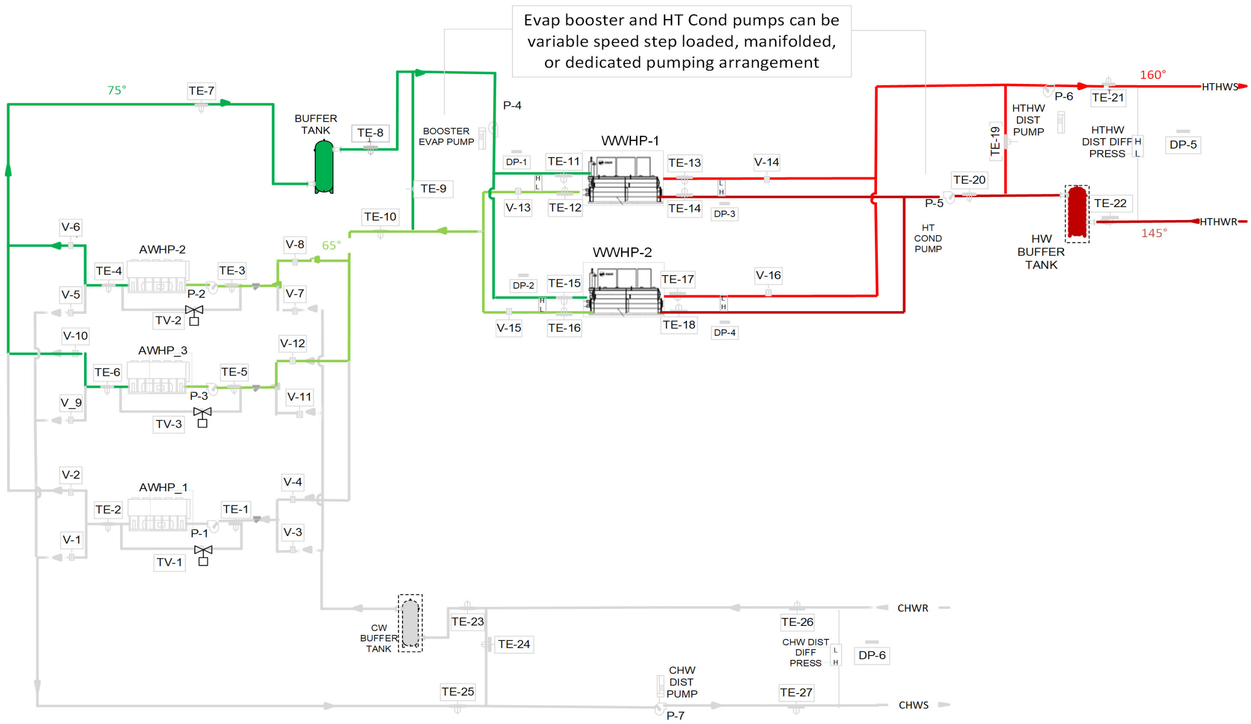
Figure 24 depicts the base system operating in heating-only mode, with the same unit operating. Again, either unit, or both units, can be operating in this mode to provide heating.

The hot-water distribution loop is variable flow, with two-way control valves on the heating coils and distribution pump (P-4) speed controlled based on the measured fluid differential pressure in the hot-water distribution loop. Critical-valve, pump-pressure optimization should be implemented, and may be required by many energy codes.

The dual-feed production mode control valves (V-1 and V-2) are positioned to return and supply hot water through the operating heat pump. AWHP-1 is controlled to supply hot water at the system's required hot-water setpoint. Hot-water temperature reset can be implemented, if appropriate for the hot-water distribution and airside system design.

The production loop pumps (P-1) can be controlled to constant flow rate required for the heat pump's heating mode, which would be typical if the distribution design flow rate is close to (or less than) the heat pump's minimum allowable flow rate (which is likely for most applications). However, if the heat pump's design flow rate is twenty percent (or greater) higher than its minimum allowable flow rate, consider implementing variable-primary/variable-secondary (VP/VS) flow control logic to reduce annual pump energy use. Less than twenty percent flow turndown does not provide for adequate control response.

Figure 25. Example of cascade heating-only system operation



[Figure 25](#) depicts the cascade system operating in heating mode. In this example, AWHP-2 and AWHP-3 are operating in heating mode, but any of the AWHP units can operate to provide heating.

The hot-water distribution loop and the dual-feed production mode control valves are controlled in the same manner as the AWHP-only base system configuration. In this example, with AWHP-2 and AWHP-3 operating in heating mode, valves V-6 and V-8 along with valves V-10 and V-12 are open, while the remaining mode control valves in the production loop are closed.

The pumps in the intermediate loop (P-2, P-3, and P-4) can be either dedicated pumps, manifolded pumps, or single pumps with a VFD.

The AWHP supply temperature is set within a range defined by the minimum heating leaving-water temperature (LWT) for the ACX unit and the maximum evaporator LWT for the RTWD. The actual setpoint within this range can be optimized for efficiency, by balancing the compressor lift between the AWWP and WWHP units. Factors impacting efficiency include the outdoor temperature, the hot-water distribution supply temperature, unit sizing, and load demand. This approach allows the AWHP to be used throughout it's entire operating map, even at cold outdoor temperatures.

Example: The minimum LWT for the ACX in heating mode is 55F. The maximum evaporator LWT for the RTWD is 65F. With a design Delta T of 15F through the RTWD evaporator, the allowable temperature range for the intermediate loop is 55F to 80F.

Figure 26. Example of simultaneous heating and cooling

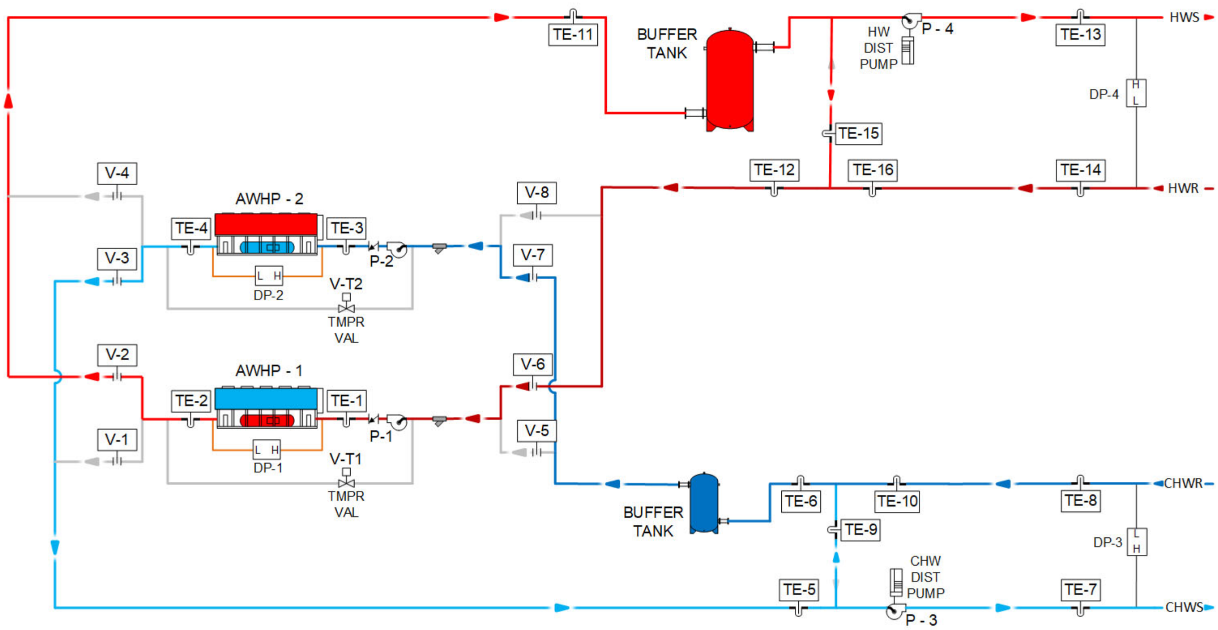


Figure 26 depicts the base system operating with one AWHP unit providing cooling and the other AWHP unit providing heating. This is a common mode of operation during shoulder seasons, depending on the building type. Refer to the "Codes and Standards Considerations,"(p. 14) section for information on adding heat recovery to this system, which reduces simultaneous operation of multiple units and can save energy.

In this example, AWHP-1 is operating in heating mode and AWHP-2 is operating in cooling mode, but the dual-feed module piping allows either unit to serve either load at any time. The chilled-water and hot-water distribution loops are controlled the same as described for the cooling-only and heating-only operating modes.

The dual-feed production mode control valves (V-1, V-2, V-3, and V-4) are positioned to supply chilled water and hot water, as required. The individual AWHPs are controlled to supply fluid at the system's required chilled-water and hot-water setpoints. Chilled-water temperature reset and/or hot-water temperature reset can be implemented, if appropriate for the distribution and airside system designs. The production loop pumps (P-1 and P-2) are controlled as required for the AWHP unit's operating mode, as signaled by the building automation system.

Figure 27. Example of cascade arrangement with simultaneous heating and cooling

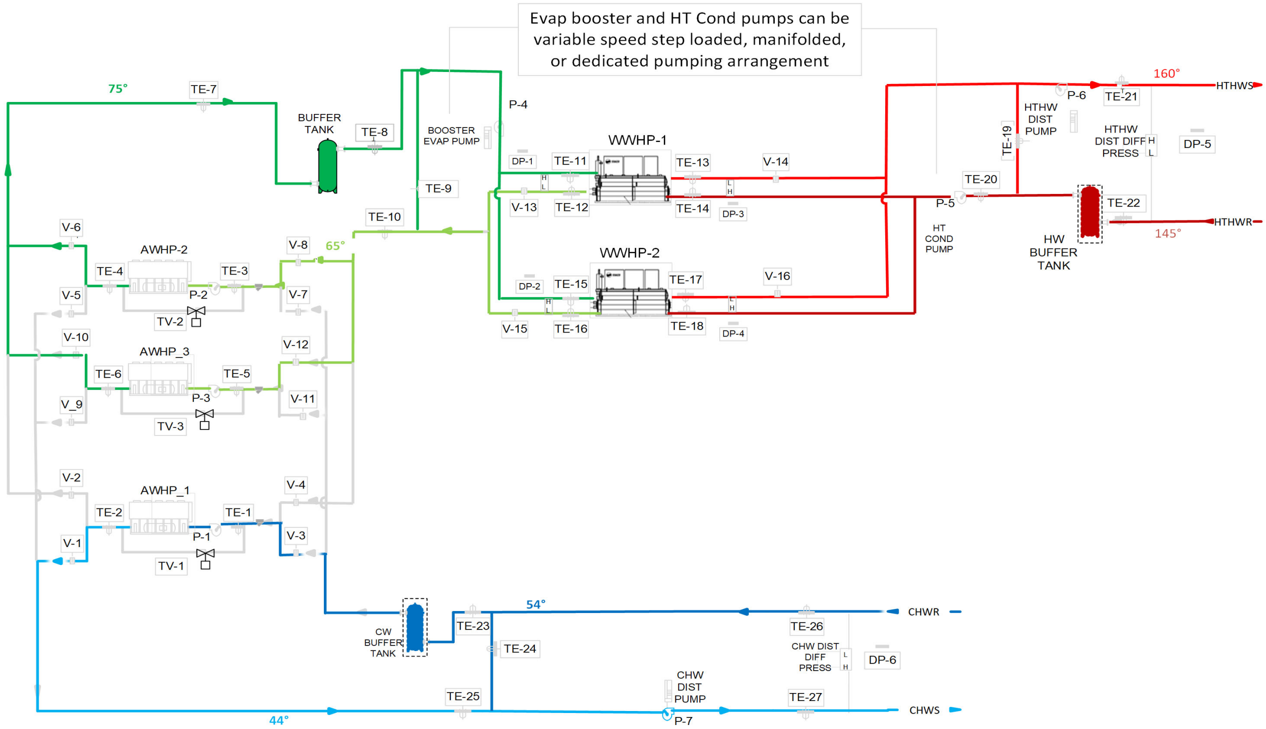


Figure 27 depicts the cascade system operating with simultaneous heating and cooling. Again, the dual-feed valve arrangement provides the flexibility to use any of the AWHP units for heating or cooling.

System Options

There are several options that can be added to the design of a heat pump chiller system. Depending on the specific building requirements, these can increase efficiency, increase accuracy/stability of control, and/or improve redundancy.

AUXILIARY OR BACKUP HEATING

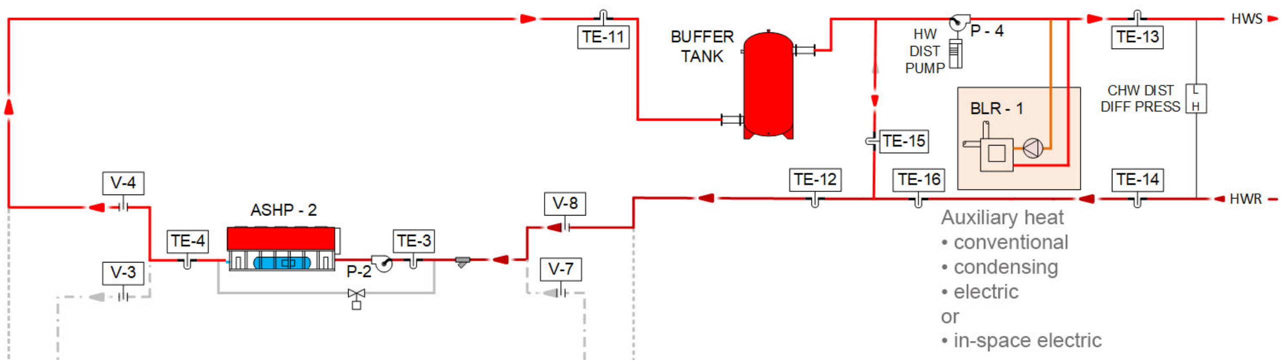
Auxiliary heat will be common in many systems, for one (or more) of the following reasons:

1. AWHPs have a minimum outdoor air temperature operating limit. At the time of this guide's publication, the minimum outdoor air temperature limit for the Trane® ACX unit (heating operation) is 0°F. Below this temperature, the compressors will be locked out of operation. An auxiliary heat source will be required if the equipment is expected to experience temperatures below this limit. The design engineer should evaluate the local weather data for 50- or 100-year extremes to help determine if an auxiliary heating system is required.
2. Resilient buildings often require backup power generation to maintain operation through utility power failures. AWHPs would require the need for large generation capacity and fuel storage. Natural gas, propane, or fuel oil boilers may greatly reduce the required generator capacity, reduces the cost and space required for the backup generation infrastructure.
3. To provide redundancy in the event of an AWHP failure, rather than adding more AWHP units to the plant, evaluate other low-cost auxiliary heating sources, such as electric or natural gas boilers. If they only operate occasionally, due to AWHP failure or maintenance, their impact on the building's carbon footprint is likely to be minimal (see below). Note that electric boilers may require upsizing of the building electrical service and gas boilers require a natural gas supply.
4. The building's life cycle carbon footprint may actually be reduced by using natural gas or propane for auxiliary heat. The current electrical grid is NOT carbon free, and likely will not be for some time. The carbon impact of generating electricity varies throughout the country. As shown in Figure 14 (p. 20) under some conditions, heating with a high-efficiency gas boiler actually results in lower carbon emissions than heating with AWHPs. This is a result of a significant decrease in heat pump's COP at colder outdoor temperatures and the efficiency (carbon impact) of generating and delivering electricity to the building. As the grid becomes cleaner over time, use of the boiler can be reduced (or eliminated) to minimize ongoing carbon emissions.

When auxiliary heat is added to the hydronic system, it is typically best to connect it into the hot-water distribution loop supply pipe, as shown in Figure 28. This location allows the auxiliary heat source to supplement the AWHP capacity when required, or to provide standalone heating (requiring operation of the distribution loop pumps only) when the AWHPs are not able to operate.

As discussed previously, control of the auxiliary heat source will vary depending on building operating conditions. The design engineer must carefully define. The design engineer must carefully define the control sequence to ensure the system design goals are achieved.

Figure 28. Auxiliary heat source cinnected to distribution loop supply pipe



HEATING REDUNDANCY

Redundancy in a heating system is typically a key system requirement. In a heat pump chiller system, redundancy is typically provided by installing additional (redundant) AWHPs and/or installing an auxiliary heating system. Each has its own advantages and drawbacks.

Redundant AWHP units. Installing N+1 AWHP units provides all-electric redundancy. The incremental cost of adding AWHP units, relative to the base system cost, varies depending on the number of base units required. Table 4 demonstrates this concept: as the number of base AWHP units increases, the incremental size and cost of adding a redundant AWHP lessens, relative to the system's design heating load.

To provide redundancy in a system with one AWHP (sizes to meet 100 percent of the design heating load), another full-sized AWHP must be installed. In this case, the total installed capacity of the two AWHPs will be equal to twice the design heating load of the system. However, if the system is designed with two AWHPs (each sized for 50 percent of the design load) to meet the design load, the incremental size and cost of the redundant AWHP (also sized for 50 percent of the design load) is lessened. With this approach, the total installed capacity of the three AWHPs will be equal to 1.5 times the design load. The equipment cost, and infrastructure cost, may be less to install three smaller units rather than two large units. In addition, the system will benefit from the more stable control and greater capacity turndown.

Table 4. Example plant size and cost impact for a redundant AWHP unit

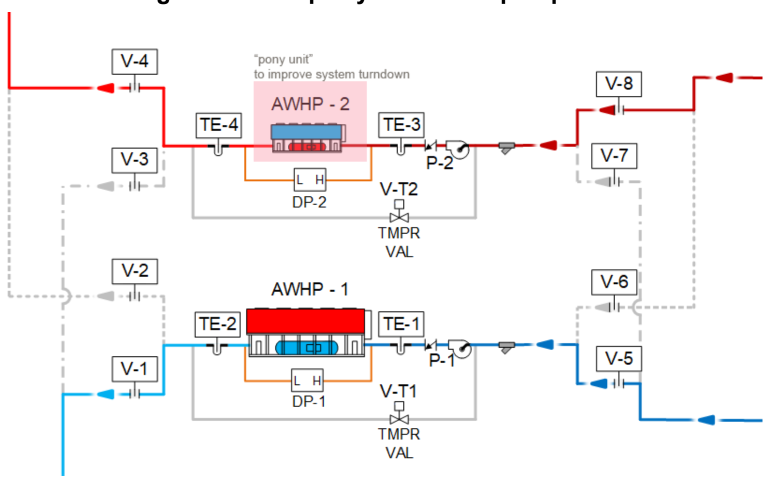
Base Number of HP Units to Meet Load	Redundancy	
	Unit Size	ACX Installed Cap
3 AWHP	33% of system load	= 1.3 x heating load
2 AWHP	50% of system load	= 1.5 x heating load
1 AWHP	100% of system load	= 2 x heating load
1 AWHP	auxiliary heat	= heating load

In this four-pipe distribution, dual-feed chiller-heater system, redundant units are added to the system in parallel with the base units, with piping identical to the base units. The redundant unit can be rotated into operation, to serve either cooling or heating loads. If the system is designed with unequally-sized units, the redundant unit likely needs to be selected for a capacity equal to the largest unit in the system. In a cascade system, to provide redundancy, additional AWHP and WWHP unit(s) would be needed.

UNEQUALLY SIZED UNITS

Figure 29 depicts the production loop with unequally-sized AWHPs, often using the common one-third/two-third sizing. For many buildings, with a significant number of operating hours at low loads, this approach can offer greater turndown of plant capacity and flow, which can result in increased efficiency, more stable operation, and more accurate temperature control.

Figure 29. Unequally-sized heat pump units



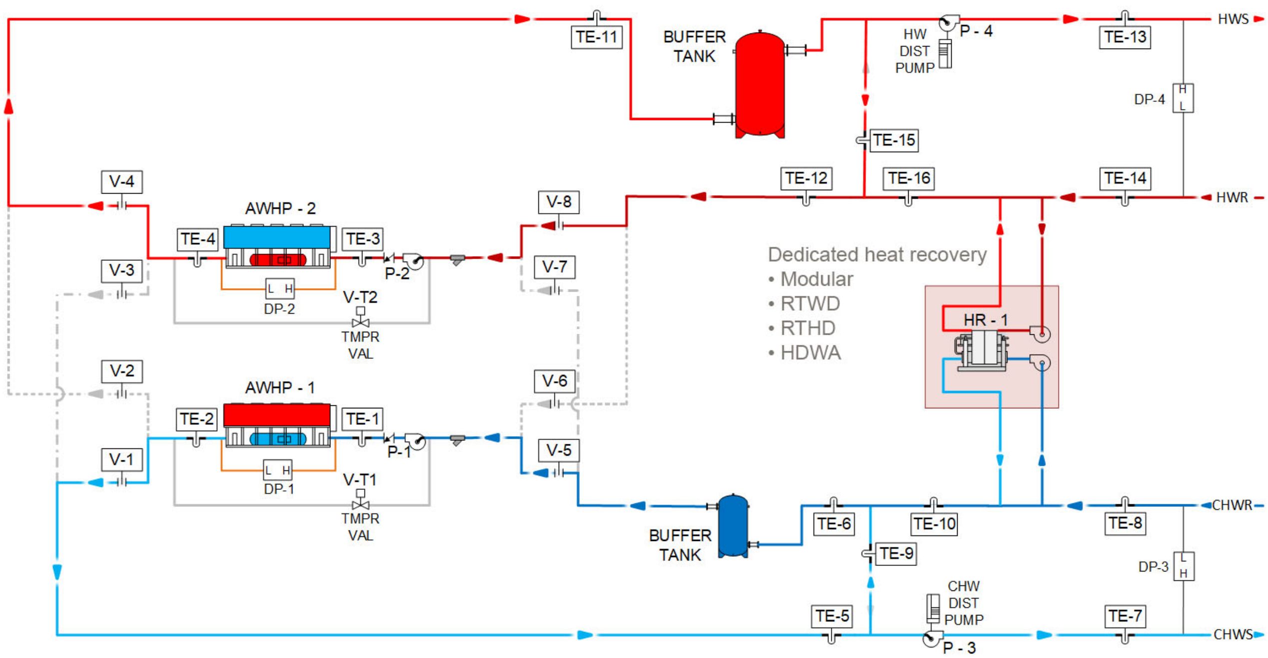
By using the dual-feed configuration, the smaller heat pump (AWHP-2 in this case) can be used to serve either the heating or cooling load, allowing for improved low-load operation for both.

Decoupling both the chilled-water and hot-water distribution loops from the dual-feed production loop enables easier sequencing and control of the unequally-sized units when serving either distribution loop. As mentioned previously, if heating redundancy is to be provided with an additional AWHP unit, it should be sized equal to the larger of the base units. In a cascade system, this same premise applies to the WWHP units.

DEDICATED HEAT RECOVERY

Another popular option in electrified heating systems is the use of a dedicated heat recovery (DHR) unit. A DHR unit is located between the two distribution loop return pipes (Table 30) and is used to move energy from one distribution loop to the other.

Figure 30. Dedicated heat recovery unit option



Proper sizing of this DHR unit is critical to the cost-effective design and selection of the system components. The amount of energy that can be transferred between the distribution loops is limited to the smaller of the loads (heating or cooling) at the specific moment in time. This result in a substantially-smaller design capacity of this DHR unit compared to the AWHP units.

The proper way to size the capacity of this DHR unit is to perform an 8760-hour load analysis of the system heating and cooling loads, and then find the minimum of system heating and cooling loads for each hour of the year when these heating and cooling loads occur simultaneously (coincidently). This hourly minimum-coincident load data can then be evaluated to determine the appropriate capacity of the DHR unit.

For any project, it is worth comparing the potential benefit of applying a DHR unit versus other competing efficiency measures (such as economizer cooling). However, the DHR unit can provide benefits beyond reducing system energy use.

A DHR unit can reduce the number of hours when two AWHP units need to operate, one for cooling and one for heating. Referring back to the example school building, AWHP operating hours were reduced by up to 1600 hours/year (Table 5). By using the DHR unit to transfer energy from one distribution loop to the other, it can fully satisfy one load and partially satisfy the other. A single AWHP unit satisfies the remainder of the load in the dominant distribution loop. This helps stabilize system operation and extends the operating life of the AWHP units.

Table 5. Impact of DHR unit on AWHP operating hours

		Operating hours without DHR	Operating hours with DHR	Reduction in AWHP operating hours
NYC school	cooling	2,596	1,820	776
	heating	3,017	2,193	824

Figure 31. Cascaded system with dual temperature and DHR unit

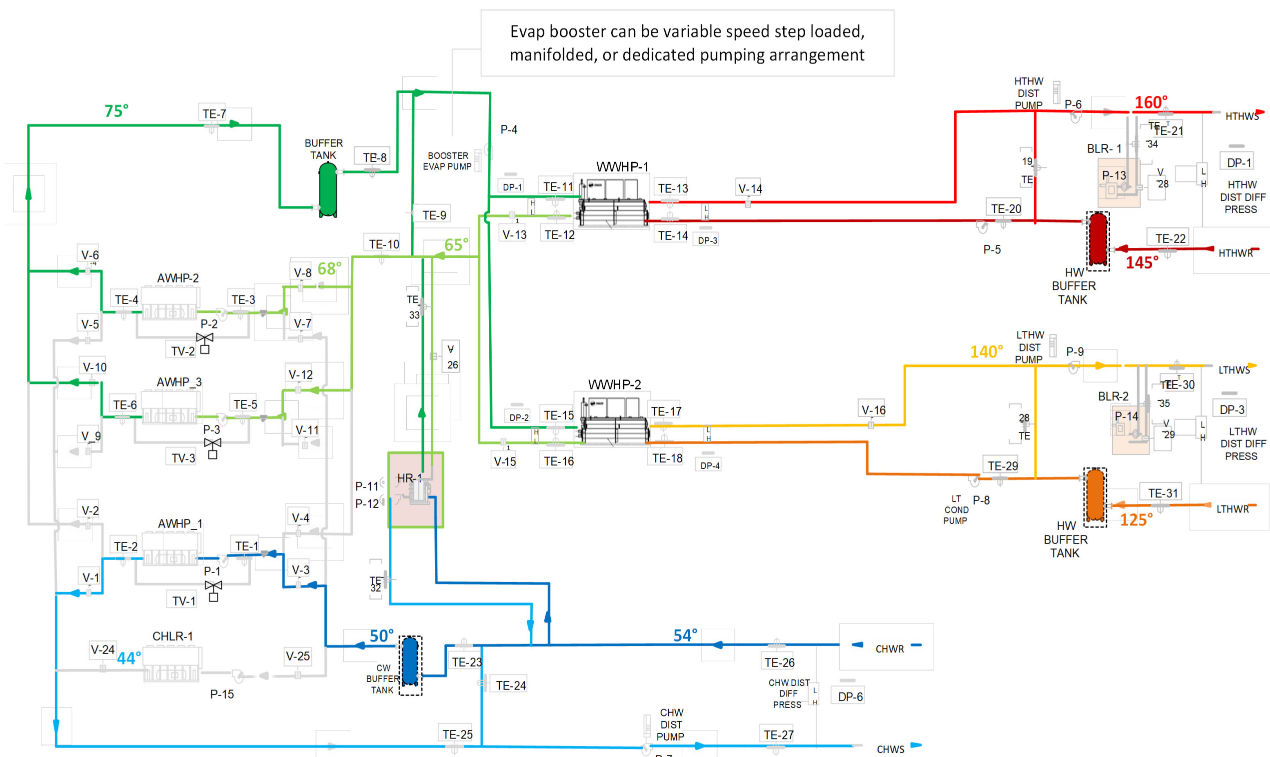


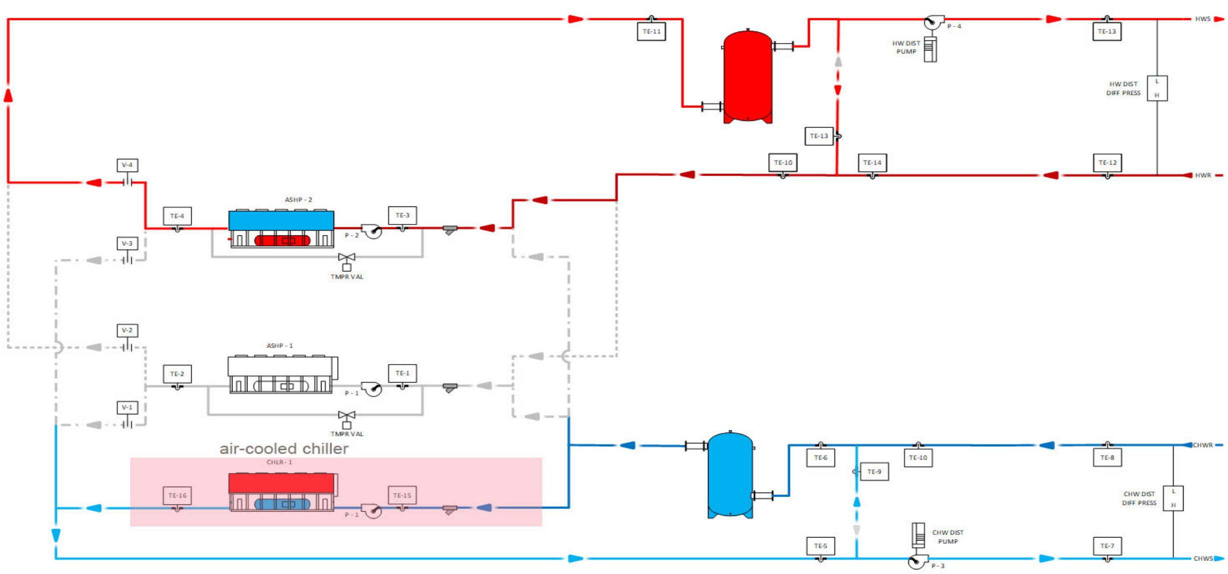
Figure 31 depicts the cascade system configuration with a dual temperature option and a dedicated heat recovery (DHR) unit installed between the cooling distribution loop and intermediate loop. This allows heat recovery to be used regardless of the difference in heating loads between the high-temperature and low-temperature hot-water distribution loops. If this flexibility is not needed, the DHR unit could instead be piped directly to either of the hot-water distribution loops.

DEDICATED CHILLER - COOLING DOMINANT HYBRID OPTION

Many types of buildings have a considerable number of hours requiring either cooling-only or cooling-dominant operation. Air-cooled chillers typically have operating efficiencies that are 5 to 15 percent better than AWHPs in cooling mode. As a result, some building owners choose to invest in hybrid plants, which include some AWHPs and some air-cooled chillers, to increase annualized cooling efficiency (Table 32).

One or more cooling-only, air-cooled chillers can be added to the cooling side of the production loop. Whenever cooling is required, this chiller can operate to efficiently satisfy the cooling load. This cooling-only chiller is typically not sized to satisfy the design cooling load, but rather some smaller capacity. A load design and analysis tool, such as TRACE® 3D Plus, can be used to determine the chiller capacity that provides the best life cycle payback. The AWHPs can then provide the additional cooling needed on peak cooling days, provide redundancy, and also satisfy any building heating loads.

Figure 32. Hybrid plant with a dedicated, cooling-only air-cooled chiller



DUAL-TEMPERATURE CASCADE OPTION

Some applications may require two different hot-water supply (HWS) temperatures. This might be due to comfort heating loads plus a process heating load (e.g., domestic hot water). Or it might be due to construction phasing of a building retrofit, where the retrofit makes use of a lower HWS temperature, but the system needs to also provide a higher HWS temperature to those portions of the building not yet retrofitted.

Figure 33. Dual-temperature cascade configuration

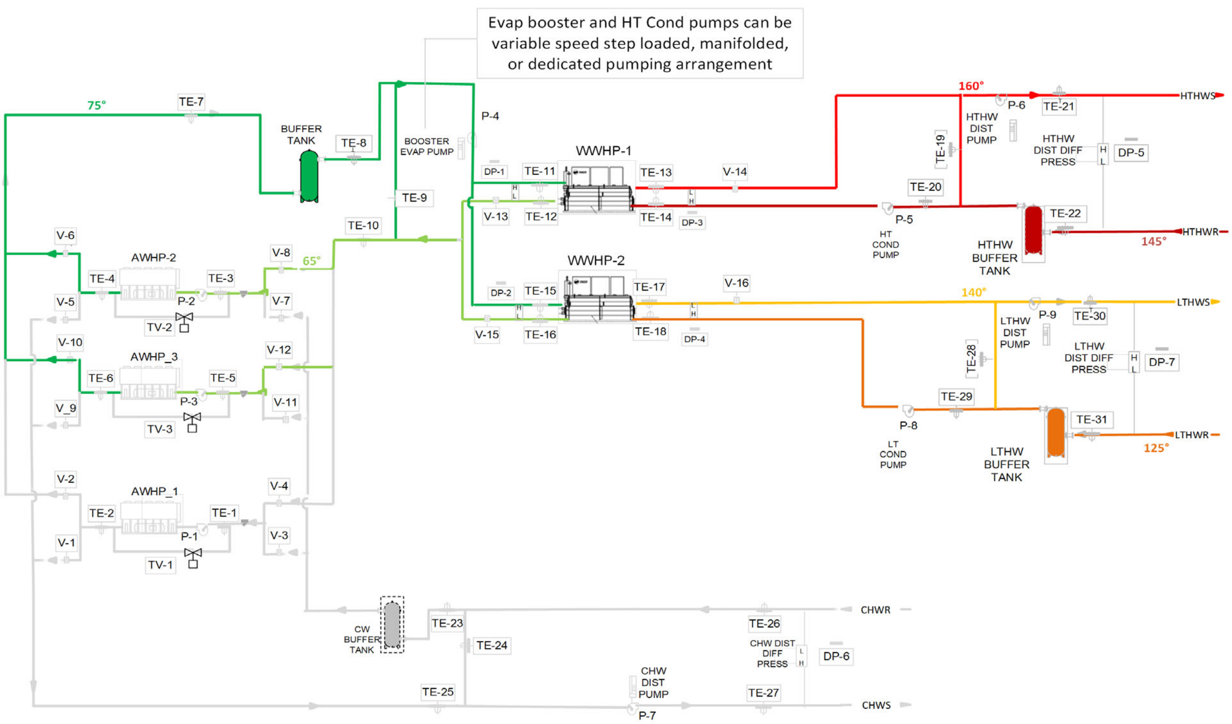


Figure 33 depicts the cascade system configuration with two WWHP units. Each WWHP unit supplies a separate hot-water distribution loop, with different HWS temperatures. The production AWHP units is controlled the same as in a single-temperature cascade system. Each hot-water distribution loop is controlled the same as a single-temperature cascade system, but with different WWHP condenser leaving-water setpoints based on the unique HWS temperature requirements.

This configuration is desirable when more than one WWHP is required due to the magnitude of the heating load. Using each WWHP unit to control to a different HWS temperature improves efficiency compared to both WWHP units being controlled to the higher of the two HWS temperatures and blending to achieve the lower HWS temperature. The capacity of each WWHP unit needs to align with the design heating load of the corresponding hot-water distribution loop, since the two distribution loops and independent in this configuration.

Figure 34. **WWHPs piped in dual feed, dual temperature configuration**

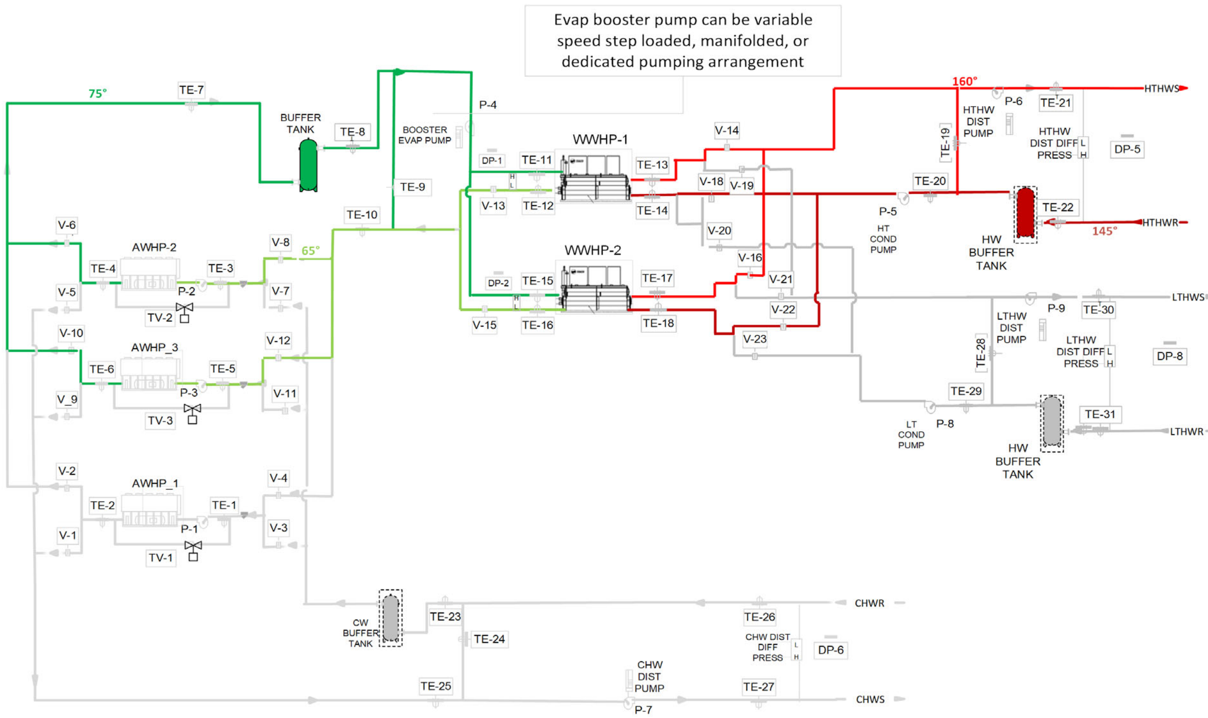


Figure 34 depicts a dual-temperature configuration that adds greater flexibility, as the WWHP units are piped in a dual-feed arrangement, similar to the AWHPs in the production loop. This allows for more-efficient loading of the WWHPs when the high-temperature and low-temperature heating loads do not peak at the same time. An added benefit of this configuration is redundancy, since either WWHP can supply water toeither of the hot-water distribution loops.

Figure 35. Dual temperature configuration via a blending valve

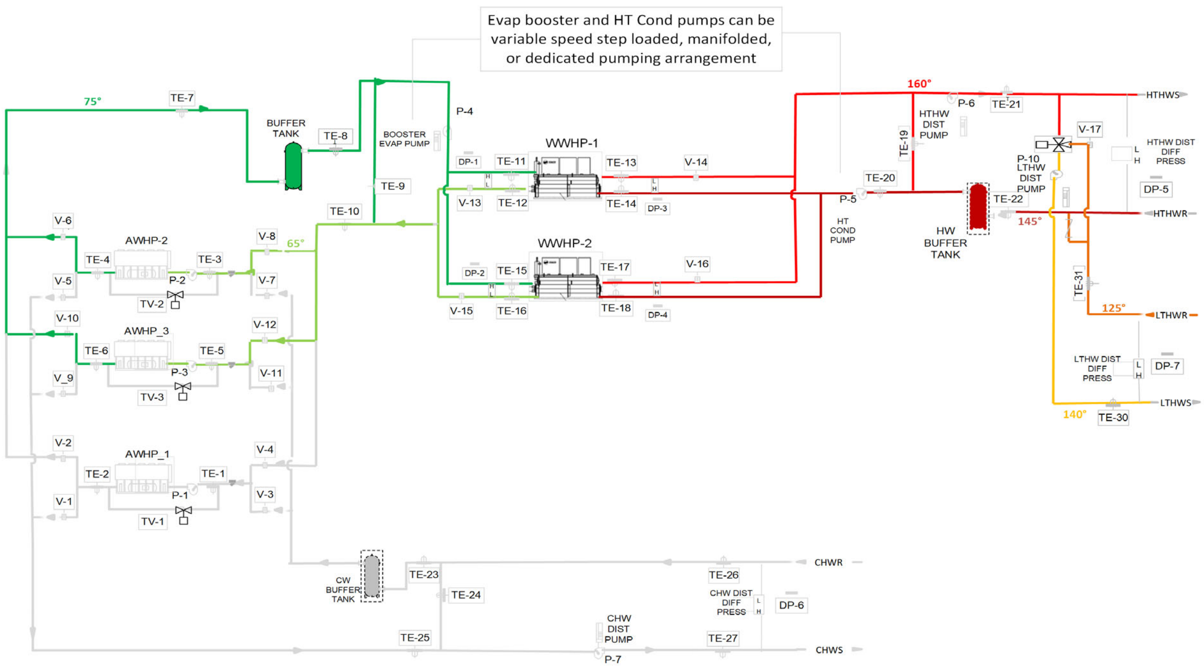


Figure 35 depicts a cascade system configuration that allows for dual-temperature distribution by adding a blending valve (V-7). This approach is not as efficient as using a dedicated WWHP for each HWS temperature (Figure 33), since all the fluid is heated to the higher of the two HWS temperatures and this valve blends this water with return water to achieve the lower HWS temperature. However, this configuration is preferred when only one WWHP is installed, or if the lower HWS load is relatively small.

Figure 36. Dual temperature configuration with auxiliary heat exchanger

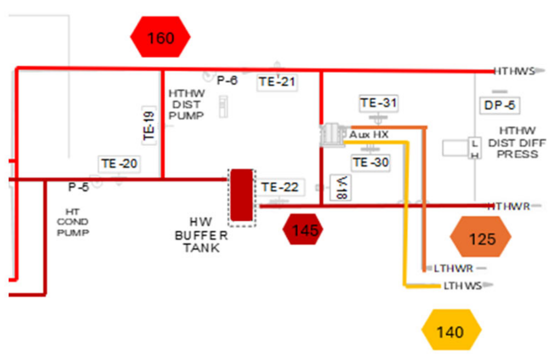


Figure 36 depicts a cascade system configuration that allows for dual-temperature distribution by adding an auxiliary heat exchanger to provide the lower HWS temperature.

Summary

An electrified cooling/heating system using air-to-water heat pump (AWHP) technology does not need to be complex to design or operate. However, careful consideration must be given to a number of unique system and equipment characteristics and operating limitations that differ from traditional chilled-water systems.

As covered in this application guide, the design engineer should be mindful of the following when designing this type of system:

- **Ensure system will meet project decarbonization goals.** Understand the project-specific electrical grid emissions and select equipment efficiencies and system operating temperatures to ensure goals will be met.
- **Auxiliary heat may be required** due to extremes in outdoor air temperature that may be outside the operating range of the heat pump technology.
- **Equipment selection should account for coldest design conditions** since the outdoor air temperature has a significant impact on AWHP unit capacity and maximum available supply hot-water supply (HWS) temperature.
- **Lower design and operating HWS temperatures result in more efficient AWHP operation.** The target for the central air-handling unit or terminal units should be a design HWS temperature of 95°F to 105°F. One method to allow for use of this lower HWS temperature is to use the same changeover coil for both cooling and heating.
- **Cascade systems using water-to-water heat pumps (WWHP)** can be effective in retrofit applications, or applications that require higher HWS temperatures in which consistent temperatures are required at low ambient conditions.
- **Proper system/equipment sizing is key to efficient and reliable operation**. Improper equipment sizing penalizes system efficiency and can shorten the operating life of equipment, reducing decarbonization benefits of the system.
- **Decoupling the production and distribution loops** is the most reliable and efficient method to ensure continued, reliable system operation over the full operating range.
- **Airside design, specification and control is key to achieving maximum decarbonization**. In addition to using lower HWS temperatures, exhaust-air energy recovery and economizers should be considered.
- **When considering the application of a dedicated heat recovery (DHR) unit, a careful analysis of 8760-hour simultaneous cooling and heating loads is required to properly size the DHR equipment and realize the heat energy and cost savings. The products and configurations used to electrify hydronic heating systems are maturing quickly. Contact your Trane Sales Representative for the latest version of this application guide and other decarbonization-related support materials.**

The products and systems available for commercial building electrified hydronic heating systems are maturing quickly. Contact your Trane Sales Representative for the latest version of this application guide and other decarbonization related support materials.

References and Resources

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