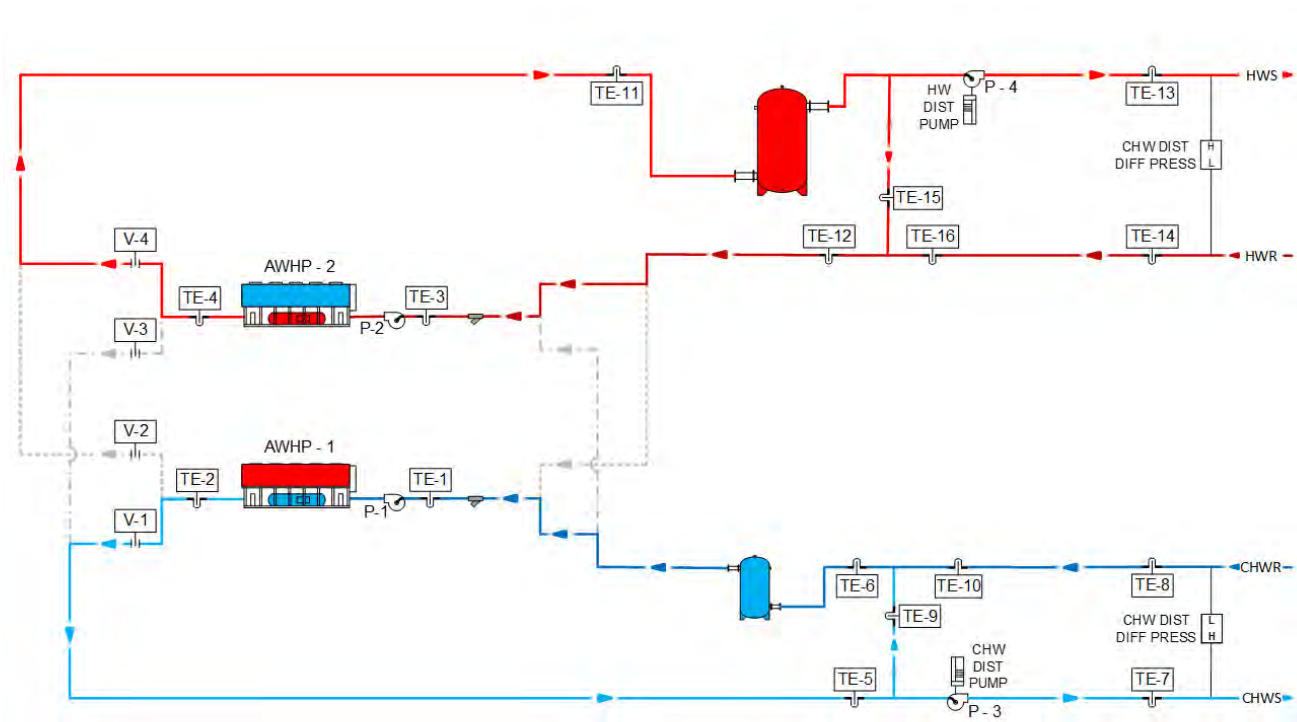
Application Guide

ACX Comprehensive Chiller-Heater 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.

April 2022

SYS-APG003A-EN

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. 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 apply for terms as used in Comprehensive Chiller-Heater Systems. Please note that these definitions **may or may not** align with their use in other HVAC systems.

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

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

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

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

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 AWHPs cannot operate to meet the full heating requirement 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**

Building Automation System (BAS): A multiple capability energy management system that relates to the overall operation of the building in which it is installed. Some examples are equipment monitoring, equipment protection from power failure, and building security.

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

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

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 operational mode controlling the unit to periodically melt the unacceptable accumulation of ice on evaporator tubes.

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

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

Partial Heat Recovery Unit: A refrigeration unit that primarily provides cooling but can also satisfy some heating loads, utilizing 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 in addition to the heat provided by the operating AWHPs.

Introduction to Air-to-Water Heat Pump Systems

EQUIPMENT

Air-to-water heat pump units are air-source refrigeration units with the ability to produce 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.

Air-to-water heat pumps (AWHPs) are an emerging equipment category driven out of a desire to decarbonize HVAC systems through electrified heating solutions. Heat pump technology offers coefficient of performance (COP) far exceeding resistance-based heating thus enabling a reduced heating energy intensity.

Trane® offers the ACX air-to-water heat pump, a packaged unit similar to a commercial air-cooled chiller. The ACX is available in six sizes in a range of 140 to 230 tons of nominal cooling capacity and it uses a total of four or six scroll compressors divided into two circuits. The ACX can deliver 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 100°F fluid temperature.

SYSTEMS

Some design principles from chilled water systems transfer well to chiller-heater systems using 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 are more significant than cooling system failure. AWHPs have operating limits that become more restrictive as outdoor air temperatures drop. Redundancy and reliable back up heating strategies must be developed. What happens when that 50-year weather event occurs?

Flexibility

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

OAT Impact

Heat pump capacity and maximum supply fluid temperature are reduced as the outdoor air temperature (OAT) drops. The equipment has outdoor air temperature operating limits. The sizing strategy for the equipment is impacted by outdoor air temperature design conditions, 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, a given AWHP circuit will occasionally operate in defrost mode. This results in periodic heating interruption with sourcing of heat from the hydronic system during the defrost cycle. Equipment sizing, buffer tank and/or supplemental boiler use can all be part of a strategy to mitigate the impact of defrost on the heating system.

This application guide proposes a system design approach using ACX units and will address these and other important design considerations to provide a reliable electrified heating and cooling solution for common building heating and cooling needs.

Understanding Air-to-Water Heat Pump Units

AIR-TO-WATER HEAT PUMP REFRIGERATION SYSTEM

There are various AWHP unit configurations including: two- and four-pipe units and units with partial or full heat recovery—the later two unit classifications can also be referred to as multi-pipe units.

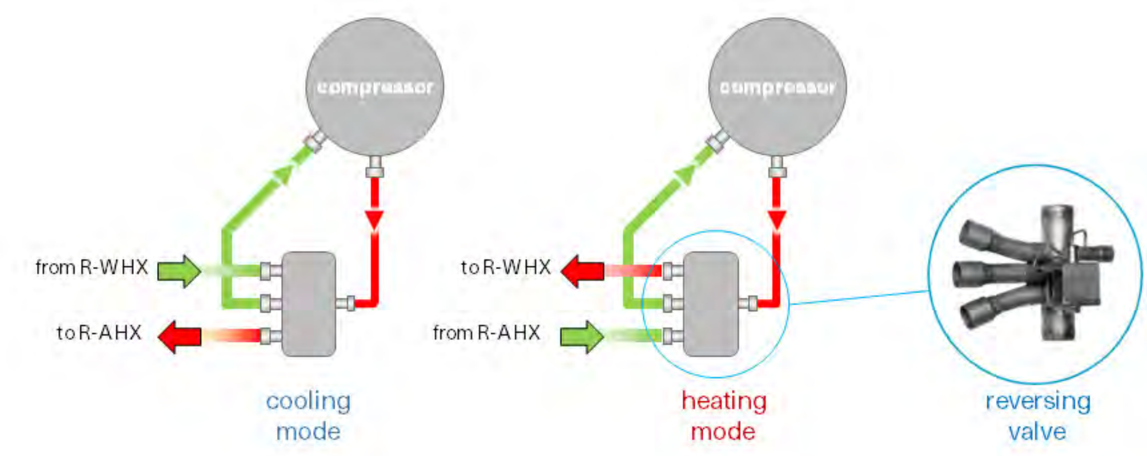
This application guide focuses on the use of two-pipe AWHP units. These units have one refrigerant-to-water heat exchanger that can cool or heat, but not both simultaneously. Two-pipe units contain two fluid pipes, one for return and one for supply. At any given time, the units operate in only one mode—either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

Figure 1. Air-to-water heat pump unit

It is important to note that an AWHP unit capacity and efficiency can vary significantly depending on the outdoor air temperature and desired leaving fluid temperature. This is particularly true in the 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). The 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 system capacity based on actual operating conditions, not code related performance rating points.

The refrigeration component that enables changeover between the cooling and heating modes of operation is a refrigerant reversing valve (four-way valve). A reversing valve switches the flow of refrigerant through the heat exchangers depending on the unit mode of operation (heating or cooling). As shown in Figure 2, the refrigerant always flows through the compressor in the same direction. The reversing valve coordinates flow direction through the Refrigerant-to-Water (R-W) and Refrigerant-to-Air (R-A) heat exchangers in the unit.

Figure 2. Reversing (four-way) valve



AWHP units are applied in systems in two different manners.

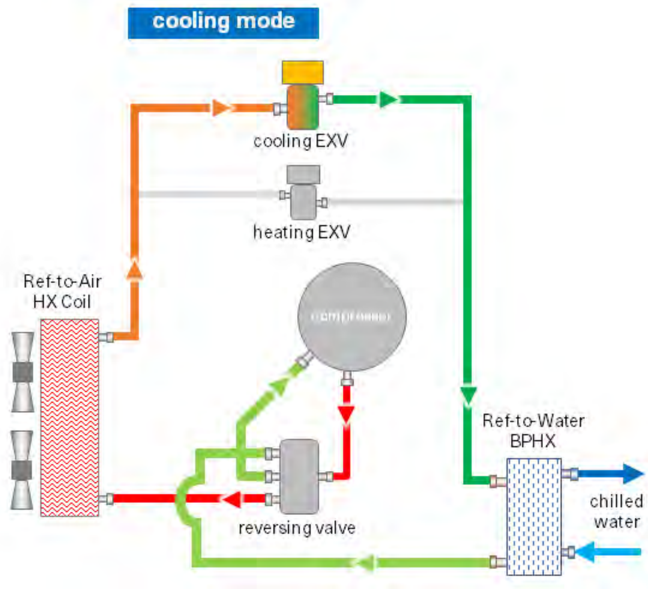
- Changeover cooling/heating: This is the most common application. The AWHP units provide either heating or cooling and changeover between the two modes depending on the load demands of the building and as commanded by the building automation system.
- Dedicated heating only: The unit only runs primarily in the heat mode as cooling mode operation is not required. However, at outdoor air temperatures below 47°F the unit may initiate *defrost cycles* which cool the water stream for short periods of time. Again, the building automation system commands the unit to operate in heating as required by the system.

AWHP unit modes of operation

AWHPs can operate in three modes: cooling mode, heating mode, and defrost mode. Figures 3, 4, and 5 show the refrigeration circuit operation in each mode including the respective heat source and sink in each mode.

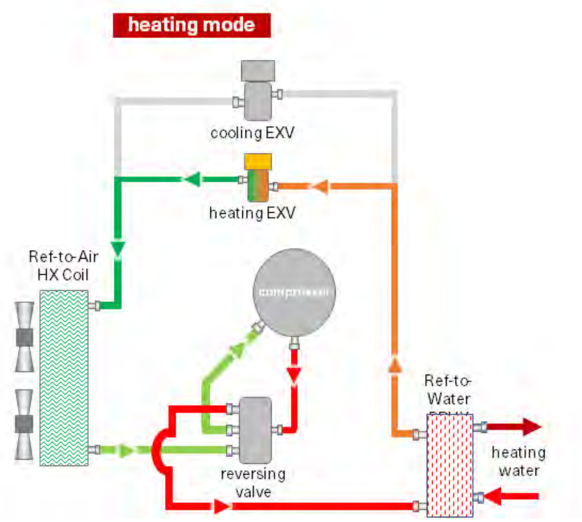
1. Cooling Mode – As shown in Figure 3, the refrigerant-to-water heat exchanger is the energy *source* for the refrigeration circuit, absorbing heat from the chilled water. The refrigerant-to-air coil is the energy *sink,* rejecting heat to ambient air. In this mode, the ACX has the same operation as an air-cooled chiller.

Figure 3. Cooling mode operation: simplified diagram



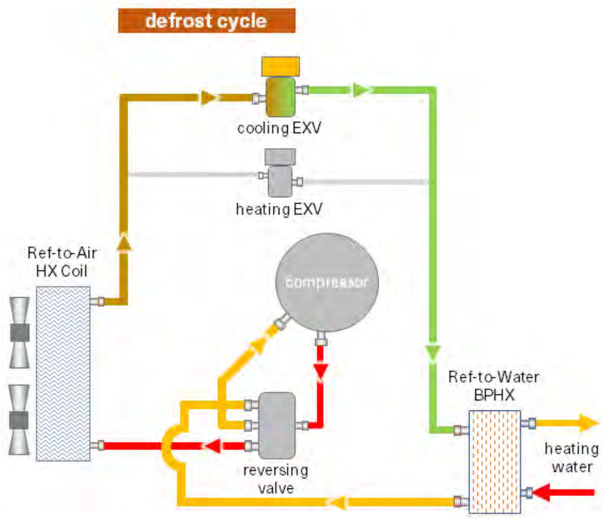
2. Heating Mode – As shown in Figure 4, the refrigerant-to-air heat exchanger is the energy *source* in the circuit, absorbing heat from the outdoor air, while the refrigerant-to-water is the energy sink in the circuit, rejecting heat to the heating water circuit.

Figure 4. Heating mode operation: simplified diagram



3. Defrost Cycle – Figure 5 shows the refrigeration circuit is in the cooling mode supplying heat to melt ice built up on the refrigerant-to-air. The return heating water is the source for the heat energy needed to defrost the outdoor air coil. The heat energy for defrost is extracted from the heating water loop.

Figure 5. 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 minimum and maximum outdoor air temperatures, entering and leaving heat exchanger fluid temperatures, and fluid flow rates. When the system operating conditions are beyond the operating limits of the unit it will protect itself by not allowing compressor operation. When designing a system, it is important to understand the specified unit's operating limits to ensure the unit can cool and heat as required.

In addition, there may be limits to how frequently the BAS can request the unit switch between modes and how long it must operate in each mode before it can be switched back to the previous mode. When switching modes time will need to be allowed so that the system temperatures can moderate enough for the unit to start in the new mode of operation. If the BAS rapidly switches the unit from heating to cooling or visa-versa the extreme temperature from the previous mode may cause the unit operational issues in the newly commanded mode. For example, if the unit is operating in cooling mode producing 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 may 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 starting the heat pump could be a way to mitigate this trip potential.

The defrost cycle is automatically initiated by the unit control when frost build-up on the air-source coil impacts unit performance. The BAS can monitor the unit mode of operation or leaving fluid temperature to detect defrost mode operation. If required, the BAS can initiate operation of an auxiliary heater to mitigate the impact of defrost operation on the system heating supply fluid temperature.

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

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ACX units are available with the following features:

- Packaged chiller-heater configuration
- Capacity range of 140 to 230 nominal cooling tons
- Meets ANSI/ASHRAE/IES 90.1-2019 heating efficiency requirements
- 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.

The Trane ACX air-to-water heat pump unit has a broad operating range. Below are examples of its typical acceptable operating range data.

ACX Ambient Operating Range:

- Cooling Mode: 0°F to 125°F OAT (-17.8°C to 51.7°C)
- Heating Mode: 0°F to 95°F OAT (-17.8°C to 35°C)

ACX Supply Fluid Temperature Range:

- Leaving Chilled Fluid Temperature: 40°F to 68°F (4.5°C to 20°C)
- Leaving Heating Fluid Temperature:
  - Water: 77°F to 140°F (25°Cto 60°C)
  - With appropriate Glycol: 68°F to 140°F (20°C to 60°C)

ACX Typical Operation Flow/Delta T Range:

- Flow: 1.3 to 4 gpm/ton (calculated based on cooling tons
  - Corresponding Chilled/Heated Fluid Temperature Delta T: 6°F to 18°F (3.4°F to 10°C)

Note: As AWHP heating capacity varies significantly with outdoor temperature, please consult the selection program for selection point flow and Delta T range.

Specific minimum and maximum allowed flow rates vary by unit size and must be obtained via the selection software.

The above limits are represented in Figure 6, ACX operating maps.

Figure 6. ACX operating map, heating mode

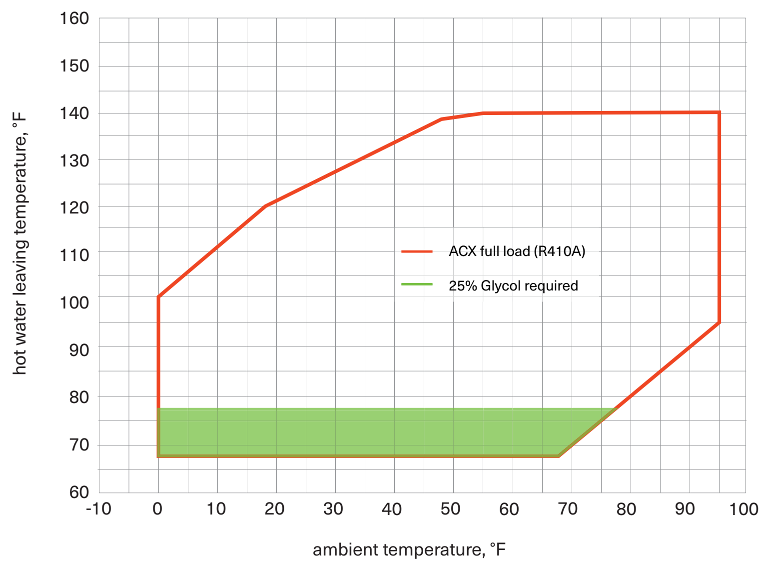
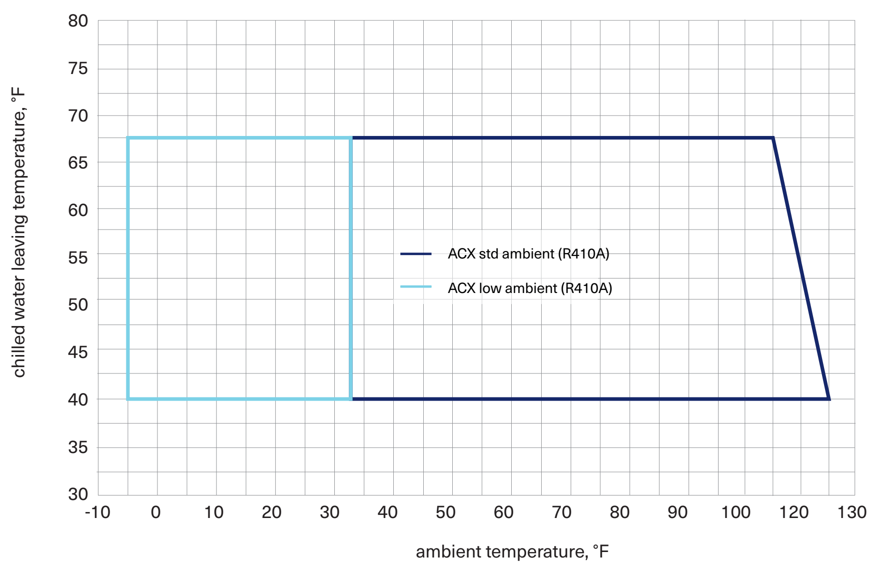


Figure 7. ACX operating map, cooling mode



Chiller-Heater System Codes and Standards Considerations

DECARBONIZATION REGULATIONS AND LEGISLATION

As electric utility producers convert to non-fossil-based renewable energy, their electric consumers have lower environmental emissions. This is sometimes referred to as decarbonization. It is noteworthy that on some present electric grids, predominantly those relying heavily on coal, natural gas, or petroleum to produce power, heating with electricity results in higher total environmental emissions than heating at the site with a fossil fuel such as natural gas. In such cases the future "greening" of the grid is expected to reduce building lifetime environmental emissions even though present emissions are higher.

At the time this document was created, the Sierra Club® reports that:

> *"Over 180 cities, more than ten counties, and eight states across the U.S. have goals to power their communities with 100% clean, renewable energy. These commitments—formalized in resolutions, climate action plans, renewable portfolio standards, and other policies—are the product of leadership from coalitions of civic champions, frontline activists, and Ready For 100 organizers nationwide. In total, over 100 million people now live in a community with an official 100% renewable electricity target."*

Transforming power generation is just one key part of the decarbonization crusade. "Building Electrification"—removing fossil fuel consumption at the building—is necessary for this current or future green grid to reduce overall carbon emissions.

As noted in the Trane® applications manual "*Heating with Compressors in HVAC Systems* (SYS-APM005-EN)", many jurisdictions are backing up their pledges to reduce carbon emissions with legislation and/or regulations that penalize buildings that continue to burn fossil fuels. The mechanisms vary: carbon offsets, tradeable/monetized agreements, limits on infrastructure permitting, and utility surcharges are just a few methods being implemented or considered.

Reasons for implementing decarbonization

Some building owners use heating with compressors (which may include heat recovery) to reduce site emissions to the environment. Reducing emissions may be beneficial for a variety of reasons, including:

Regulatory. Burning fossil fuels, such as natural gas in a boiler, increases site emissions. If the site has already reached its maximum emissions level, heat from electric compressor-bearing units can satisfy heating loads.

Environmental stewardship. All electric or hybrid electric systems are perceived as a "green technology." This may benefit not only the environment but the building owner both monetarily and from a public-relations perspective.

Technical limits. Limited natural gas line infrastructure and imposed fines or fees for site environmental (e.g., carbon) emissions. "Electrification" of building heating (heating using compressors) becomes more beneficial.

Financial incentives. If emissions trading becomes common, the credits earned by electrified buildings may be extremely valuable.

ELECTRIFICATION OF HEATING CODE REQUIREMENTS

For simplicity—because it is adopted directly or via reference by most U.S. states—we'll concentrate on *ANSI/ ASHRAE/IES 90.1–2019 - Energy Standard for Buildings Except Low-Rise Residential Buildings* for code related considerations.

Particular jurisdictions may have modified or eliminated these requirements and it is up to the design engineer to check with local code officials or visit www.energycodes.gov to find out which requirements apply.

EQUIPMENT EFFICIENCY REQUIREMENTS

Beginning in the 2019 version, ANSI/ASHRAE/IES 90.-2019 included a new class of equipment, heat-pump and heat-recovery chiller packages with minimum efficiency requirements. Details are included in Table 6.8.1-16 with performance requirements updated in Addendum y.

Table 1 details efficiency targets not only for the cooling mode of operation (heat pump chillers typically lose a bit of efficiency) but also for the heating mode of operation. Several classes of heating water temperature are included at varying outdoor air temperatures. By interpretation and addendum, ANSI/ASHRAE/IES 90.1-2019 clarified that air-to-water heat pump equipment must meet the Path A or Path B cooling efficiency requirements and heating efficiency requirement(s) at **both** of the air-source temperatures and for at least **one** of the heating liquid temperatures. If the unit is never used for cooling (e.g., piped in where a boiler might have been) it can also be exempted from the cooling requirements.

ANSI/ASHRAE/IES 90.1–2019 Chiller/Heat Pump Efficiency Requirements

Table 1. Air-source heat pump: minimum efficiency requirements (Source: ANSI/ASHRAE/IES Standard 90.1-2019, Table 6.8.1-16 and Addendum Y, Table 6.8.1-16)

Equipment type	Size category refrigerating capacity (ton_R)	Cooling-operation efficiency air-source (EER, FL/IPLV), Btu/W-hr		Heating source conditions OAT (db/wb) °F	Heat pump heating full load efficiency (COP_H), W/W			Test procedure
					Entering/Leaving heating liquid temperature			
		Path A	Path B		Low 95°F/105°F	Medium 105F/120°F	High 120°F/140°F	
per ANSI/ASHRAE/IES Standard 90.1-2019 as originally published								
air-source	all sizes	≥9.595 FL ≥13.02 IPLV.IP	≥9.215 FL ≥15.01 IPLV.IP	47 db 43 wb	≥3.290	≥2.770	≥2.310	AHRI 550/590
		≥9.595 FL ≥13.30 IPLV.IP	≥9.215 FL ≥15.30 IPLV.IP	17 db 15 wb	≥2.230	≥1.950	≥1.630	
per ANSI/ASHRAE/IES Standard 90.1-2019 Addendum Y (approved December 9, 2021)								
air-source	<150.0	≥9.595 FL ≥13.02 IPLV.IP	≥9.215 FL ≥15.01 IPLV.IP	47 db 43 wb	≥3.290	≥2.770	≥2.310	AHRI 550/590
				17 db 15 wb	≥2.029	≥1.775	≥1.483	
	>150.0	≥9.595 FL ≥13.30 IPLV.IP	≥9.215 FL ≥15.30 IPLV.IP	47 db 43 wb	≥3.290	≥2.770	≥2.310	
				17 db 15 wb	≥2.029	≥1.775	≥1.483	

Note: See ANSI/ASHRAE/IES Standard 90.1-2019 and Addendum y for details and footnotes related to the date shown above.

HYDRONIC SYSTEM CODE REQUIREMENTS

Four-pipe distribution systems. There is no special call-out for four-pipe systems within the hydronic system controls section. However, since most terminals are required to have VAV airflow (down to ¼ hp motors), the single-zone controller will handle the coordination between fan speed and valve position. The section below on variable flow pump control applies for minimizing pump energy.

Two-pipe distribution systems. In two-pipe distribution systems, the same pipes are used to distribute both heating and cooling fluids, but not at the same time. The fluid loop must be changed over.

Section 6.5.2.2 of ANSI/ASHRAE/IES 90.1-2019, requires three things that are geared toward limiting energy waste associated with changeover:

1. System controls include a dead band between heating and cooling of at least 15°F outdoor air temperature. This is to limit the system from switching back and forth too often.
2. System controls require at least four hours of operation in one mode before changing to the other. This is a preventative limit to reduce frequent switching of modes within the system, thereby cooling or heating fluids that were previously heated or cooled.
3. System controls reset the heating and cooling supply temperatures so that they are no farther than 30°F apart at changeover. This limits the pull-down loads after the system is changed over.

Variable flow. Section 6.5.4.2 essentially requires variable-frequency-driven pump performance once a system has:

- three or more control valves,
- the collective pump horsepower rises above a threshold based on climate zone, and
- whether the pump is used for heating or cooling.

In a chiller-heater system that reuses pumps for cooling and heating, the more constraining threshold governs. For this reason, and for better control in all modes of operation, pumps are equipped with variable-frequency drives in all standard configurations.

Note: The above "Variable Flow" requirement does not imply a requirement for a Variable Primary Flow (VPF) directly pumped system configuration. This requirement can be satisfied by a decoupled 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 includes decoupling of the production and distribution to enable the flexibility of operation and simplified control.

Chilled- and hot-water reset controls. Section 6.5.4.4 requires automatic resetting of the fluid temperature setpoints once the system gets larger than 300,000 Btu/h (25 tons). Systems served by district energy or thermal storage systems are exempt from this requirement, as are systems that use pump-pressure optimization based on critical valve position or that are serving processes that require a specific temperature.

Hot-water temperature for gas-boiler served space heating. A new section in the 2019 version essentially prescribes the use of condensing boilers for medium-sized systems in new buildings. While chiller-heater systems are not necessarily going to have gas boilers, lower temperature hot-water systems are going to become the design standard. Another potential consequence affecting chiller-heater systems is that if higher heating-water temperatures are necessary, a designer may be obliged to use a different system design that doesn't rely solely on gas boilers.

Section 6.5.4.8 "*Buildings with High-Capacity Space-Heating Gas Boiler Systems*" requires three things for new buildings with a total system input of at least 1,000,000 Btu/h but not more than 10,000,000 Btu/h:

1. The building must have increased boiler efficiency E_t of 90 percent or better. This can be achieved at each boiler or with a weighted average efficiency of multiple boilers.
2. The design and selection of hot-water distribution system coils and heat exchangers must allow hot water return temperatures of 120°F or lower.
3. All operating conditions must use 120° F or lower boiler entering-water temperatures, or bypass no more than 20 percent of the design water flow back to the boilers, either by three-way valves or minimum-flow-bypass controls.

Auxiliary heat rejection or heating. Section 6.5.2.2.3 normally was thought of as a section for water-source heat pump systems. In some chiller-heater configurations the description seems to apply, which would mean the requirements would as well.

"Hydronic heat pumps connected to a common heat pump water loop with central devices for heat rejection (e.g. cooling tower) and heat addition (e.g. boiler)"

1. The first requirement is that the controls have a deadband of at least 20°F between heat rejection and heat addition by central devices, unless the system uses a temperature optimization that determines the most efficient temperature based on real-time conditions of demand and capacity.
2. The second requirement relates to isolating the heat rejection cooling tower except for minimal flow for freeze protection.

HEAT RECOVERY REQUIREMENTS

Many standards and building codes directly or indirectly require condenser heat recovery.

Heat recovery for service-water heating. ANSI/ASHRAE/IES 90.1–2019, Section 6.5.6.2 "*Heat Recovery for Service Water Heating*", **requires** heat recovery for service-water heating when:

- The facility operates 24 hours a day.
- The total installed heat-rejection capacity of the water-cooled systems exceeds 6,000,000 Btu/h of heat rejection (about 400 tons (1,400 kW_t) of cooling).
- The design service water heating load exceeds 1,000,000 Btu/h.

After all three thresholds are met, the required heat recovery is the smaller of:

- 60 percent of the peak heat-rejection load at design conditions, or
- Preheat of the peak service hot water draw to 85°F.

Other sections of ANSI/ASHRAE/IES 90.1-2019 can trigger, if not directly require, condenser heat recovery. The sections outlawing simultaneous heating and cooling don't essentially outlaw, but instead serve as a limitation: the base requirement (and the simplest way to comply) is to not allow *any* simultaneous heating and cooling, but then goes on to provide exceptions that serve as limits on the energy that can be used for certain purposes, namely comfort and precise humidity control.

Simultaneous heating and cooling for comfort control. Section 6.5.2.1 "*Zone Controls*", triggers the use of heat recovery if the system is reheating more than the allowed amount of previously cooled air. For VAV systems, the amount of air that can be reheated (when the system is in cooling mode) is one of three limits:

- The amount of air for ventilation,
- 30 percent of the peak cooling airflow, or
- Some other value that uses less overall energy.

That is, if the system opens a few critical VAV boxes more and reheats to meet space temperature, so that overall ventilation can be reduced while still properly ventilating all rooms. If the VAV uses 20 percent of cooling design airflow for a minimum airflow, at peak heating the VAV damper may be opened to 50 percent before using the maximum temperature setpoint

(20°F higher than the space temperature setpoint except during preoccupancy warm-up and setback). This allows for lower temperature hot water to be used because there is more airflow allowed. Trane® VAV box controllers include both options for control.

For distributed hydronic systems such as fan-coils, or for the heating and cooling fluids in air-handling systems, the ANSI/ASHRAE/IES 90.1-2019Standard 90.1 limits simultaneous heating and cooling by prescribing a maximum supply temperature changeover deadband in the control logic. During changeover from heating to cooling, a two-pipe system must have a maximum 30°F deadband between the heating and cooling supply water temperature setpoints. See the previous section on hydronic system controls for other requirements.

Simultaneous heating and cooling for humidity control. Section 6.5.2.3 "*Dehumidification*", triggers the use of recovered energy for reheating for precise humidity control in certain applications. At least 75 percent of the reheating energy must come from site-sourced or site-recovered energy. Additional requirements in the mandatory section 6.4.3.6 limit the amount of fossil fuel or electricity for relative humidity control, as well as software, switches, or mechanical methods for preventing the simultaneous use of dehumidification and humidification equipment. The warmest zone can't be humidified above 30 percent RH and the coldest zone can't be dehumidified below 60 percent RH. There are a few exceptions, directed at the same applications (museums, hospitals) that are targeted by the prescriptive section limits in 6.5.2.3 previously mentioned.

Condenser heat recovery as an exit from economizer requirements. Section 6.5.1 "*Economizers exception 5",* allows a designer to avoid an economizer (both air and water/fluid types) if the system employs sufficient condenser heat recovery, as defined in Section 6.5.6.2.2. This applies particularly to systems with distributed heating and cooling and dedicated outdoor air systems, those with duct limitations or long outdoor air paths to air-handlers, and applications that don't wish to include them (e.g., data centers using air-cooled chillers). The capacity referenced is the same as the one included in service water heating: 60 percent of the peak heat-rejection load at design conditions or preheat of the peak service hot-water draw to 85°F.

Note: Systems using this exception do not need to meet the service water heating size thresholds (24-hour operation, 6 MMBtu of heat rejection, 1 MMBtu of peak service water heating load). The economizer exception purposely points intentionally and specifically to the capacity portion of that section: the 60 percent or 85°F preheat capacity requirements.

Energy modeling for code compliance. When trying to comply with an energy cost or energy performance path using a simulation model, recovering condenser heat when it's not required is one way to offset the energy impact of other requirements that the proposed building design cannot or doesn't wish to meet. For example, a designer has elected to include more glazing than the 30 percent window-to-wall ratio allowed by the prescriptive path. In that case, an energy model is the only way to demonstrate compliance. The modeler will be looking for design elements that can be added to the proposed design to make up for the energy impact of design choices or limitations.

Building owners may decide to mandate its use. For example, the U.S. Army Corps of Engineers publication "*Humidity Control for Barracks and Dormitories in Humid Areas*"[1] states:

> *"Army shall use condenser heat recovery in accordance with ASHRAE® 90.1."*

System and Unit Sizing

Proper design of an electrified system including operating conditions, system and heat pump equipment sizing, as well as redundancy is critical to reliable and efficient system operation, as well as affordable first cost. Commonly used high temperature hot water conditions will not only result in high system energy consumption but are not even attainable with commonly available heat pump technologies. Past operating assumptions must be discarded, and low-temperature supply heating concepts embraced.

Historic “rules-of-thumb” for system capacity sizing dare not be used. Computerized load analysis for new buildings and accurate load history for existing buildings is essential to proper system design and meeting owners environmental and financial goals. This section discusses many of the important points of system design relative to electrified system design.

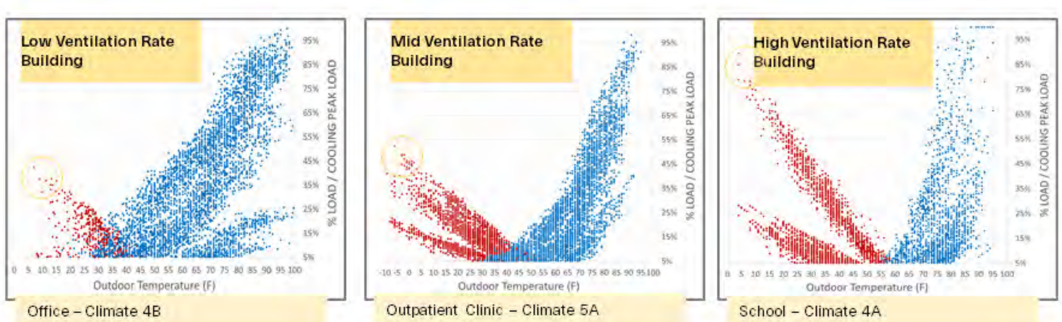
BUILDING LOAD EVALUATION

This section considers building loads from several perspectives. Figure 8 is an example plot of load verses outdoor air temperature for three building types with varying ventilation requirements and in three different climate zones.

The buildings represented are the ASHRAE® building models used as the basis for the ANSI/ASHRAE/IES Standard 90.1-2019, “*Energy Efficiency Standard for Buildings Except Low-Rise Residential Buildings”*. Hourly load data was extracted from TRACE® 3D Plus, which uses the EnergyPlus® calculation engine, and plotted.

Each point on the plot represents an hourly load point (8760 hours) as a function of outdoor air temperature over a typical calendar year. The cloud of points in blue and to the right of the plot inflection represents cooling loads and in red and to the left are heating loads. The higher grouping to the left is occupied heating loads and the lower grouping to the left is unoccupied heating loads. The difference is hard to discern in the office plot as a result of the lower ventilation characteristics which drives the occupied/unoccupied heat load separation.

Figure 8. Hourly heating and cooling load plotted versus ambient temperature



Notice the variation in the difference between the cooling and heating peaks on the plots. The magnitude of the differences varies primarily by ventilation rate and secondarily by building type and climate.

It is well understood that 100 percent cooling or heating capacity is not often needed. This is even more true for heating. Also, notice the point density near the peak heating hours (yellow circles). Hours near peak load are very few and even less than cooling.

It is important to note that ventilation rates impact loads more so than building type and climate. In addition to the buildings shown, hotels are characterized by moderate ventilation rates whereas large hospitals are characterized by high ventilation rates. Effective air-to-air energy recovery can substantially reduce the ventilation driven occupied heating load allowing them to approach the much lower unoccupied loads. Air-to-air energy recovery effectiveness will depend on how much exhaust air is available for recovery and the efficiency of the chosen heat recovery technology. See the “References and Resources,”(p. 47) section for additional information.

Another observation from Figure 8 is that commercial buildings with primarily comfort heating and cooling, no data center or other process loads, have limited simultaneous heating and cooling demand during prime economizing times. This also occurs at lower loads and reduces the energy benefit of waterside heat recovery. However, there are other benefits to properly sized heat recovery that are discussed in the system design section.

AIR-TO-WATER HEAT PUMP PLANT SIZING

The air-to-water heat pump system needs to be sized to handle both the cooling and the heating peak loads. The same equipment is expected to satisfy both loads. Not only is the magnitude of the peaks different but the capacity of the equipment varies with the outdoor air temperature. The AWHP heating capacity at the design heating ambient temperature is typically much less than the unit “nominal” cooling capacity. This may result in substantially different equipment selections for design cooling and heating.

Cooling Sizing

Figure 9 graphs the peak loads versus outdoor air temperature for the school represented in Figure 8 (p. 18). Not surprisingly the building design points are not necessarily the same as standard AHRI rating points so actual applied performance differs from the standard rating performance point. Designers must also remember that the equipment model number “nominal” capacity often differs from the unit applied capacity and therefore should not be used for unit selection. In this example, the peak heating load is ≈2,000,000 BTUH or 167 tons. The ambient temperature for this load peak is 4°F which is far below the AHRI rating point temperatures. Although the design cooling ambient is coincident with a rating point temperature, the design chilled water supply temperature may be colder (e.g. 40-42°F) than the AHRI rating condition (44°F).

In both cases the actual capacity, at actual applied design conditions, must be determined using the equipment selection software.

Figure 9. Peak building loads versus ambient temperature

Heating Water Supply Temperature

The heating sizing process for heat pumps requires the designer 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.

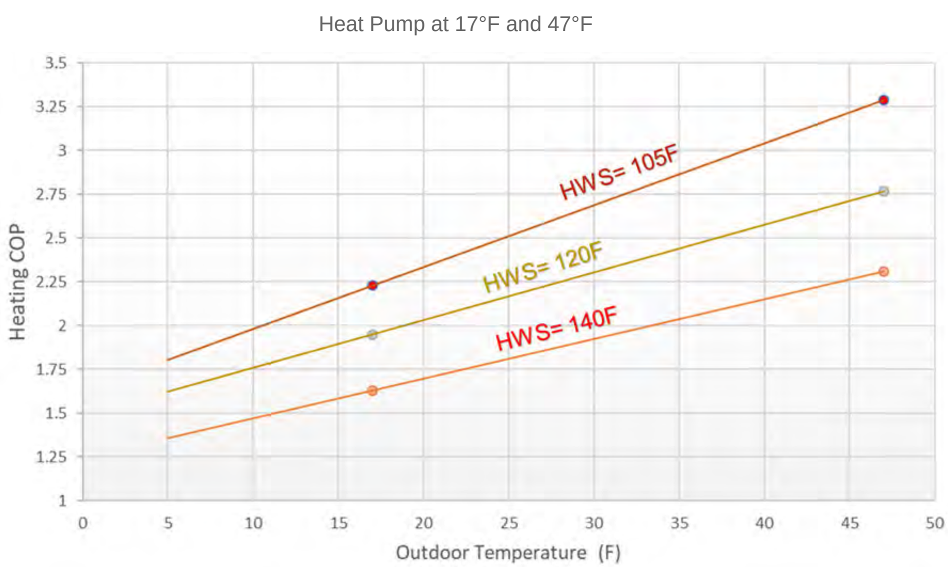
These factors are interrelated, so the trade-off needs to be understood and aligned with the priorities of the building owner.

The design heating water supply temperature is perhaps the most significant factor to consider. Figure 10 is a plot of the ANSI/ASHRAE/IES 90.1-2019 minimum heating efficiency points at common heating water supply temperature points. The chart linearly extends the required performance across typical heating design outdoor temperatures. This shows the relationship between heating water supply temperature (HWS), outdoor air temperature, and unit COP_H (Coefficient of Performance-Heating) at full load. COP impacts operating cost, electrical demand, and carbon footprint. A higher COP is better.

- Lower heating water supply temperature increases COP_H.
- Colder outdoor air temperatures reduce COP_H.

Since the primary driver to electrify heating is reduced carbon footprint, the importance of the heating water supply temperature is clear. Lower heating supply water temperature is a primary way to reduce the negative impact of cold outdoor air conditions and raise the system COP_H. Even lower heating water supply temperatures than those represented in Figure 10 are feasible in some systems.

Figure 10. ANSI/ASHRAE/IES 90.1-2019 minimum heating COP_H



Carbon Emissions and COP

Switching from on-site fossil fuel heating to electrified heating will not guarantee a reduction in atmospheric carbon emission. In fact electrified heating can significantly increase carbon emissions. To be sure the system is achieving the goal of decarbonization we 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?

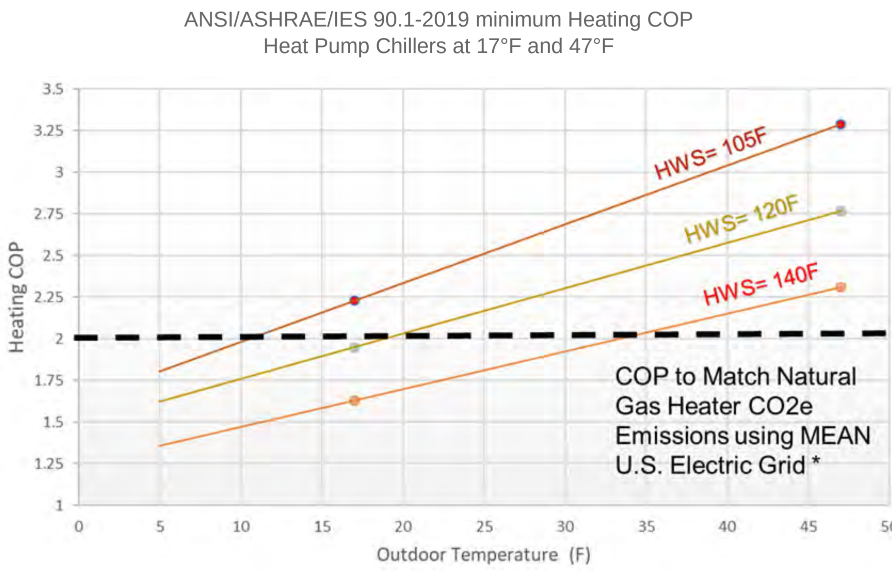
To answer that question the carbon emission rate of the electric grid must be analyzed. Electrical power production comes from a mix of energy sources including natural gas, coal, nuclear, hydro, wind turbines, solar arrays, and other sources. Each has its own carbon emission rate profile. The combination of energy sources and therefore carbon emission rate of the electric grid varies by location.

Based on the mean carbon emission rate for the U.S. electric grid (*884 lb/MWH), the electrified heating system efficiency, stated in terms of Coefficient of Performance (COP_H) required to match an on-site natural gas boiler (90 percent efficient) emission rate is determined to be 2.0 COP_H. This is shown on Figure 11 as a horizontal dotted line. For electrified heating to emit less carbon it must operate *above the line*.

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

Depending on the location, the required electrified heating system COP_H could be as high as 3.0 for the most carbon intensive grids (dirtiest) or as low as 1.5 for the lowest carbon intensive grids (cleanest), to be better than on-site fossil fuel heating. In general the higher the system COP the easier it will be to realize the goal of reducing carbon emissions.

Figure 11. Heat pump versus natural gas heating CO2e



It is important to note that, the lower the design heating water temperature the higher the system COP. Figure 11 indicates an outdoor ambient temperatures below 10°F with a supply heating water temperature of 105°F, fossil fuel heating produces less carbon emissions than an air-to-water heat pump. The 2.0 COP_H carbon emissions crossover point is based on the MEAN U.S. grid carbon intensity and a 90 percent efficient gas boiler. Less carbon intensive grids or less efficient boilers will result in a lower crossover temperature. Higher hot water supply temperatures drive the crossover temperature higher. Whatever the specific crossover point may be, the system should changeover to gas heat at outdoor air temperatures below that point for best carbon emission performance. As the grid becomes "greener", through the life of the building, the changeover temperature can easily be adjusted for the best ongoing environmental performance. A single heat source system may not provide the lowest carbon emissions at all operating conditions, it is important to design for the annualized impact to provide carbon emissions reduction. This view of the data brings a critical perspective on the importance of making good system design decisions.

SPACE HEATING EQUIPMENT OPTIONS

Of course, the heating water supply temperature must be considered when selecting coils for the air handling equipment. Zone control equipment typically has fewer coil options than central air handling equipment and likely will drive the final heating water supply temperature design decision. Average system heating coil ΔT must be evaluated to prevent high water flow rates from causing excessive pump energy consumption.

There are numerous low temperature heating supply options available for central and zone heating. Many can be applied to existing buildings when considering electrification.

- High capacity heating coils – For central air-handers heating coil options are available that can provide required heating with low supply water temperature, acceptable waterside ΔT (flow) and airside pressure drop.
- Changeover coils – For both terminal equipment and central air-handlers consideration should be given to applying changeover coil control. That is, using the same coils for both cooling and heating. Cooling coils are typically much larger than heating coils and when provided with heated water they become high-capacity heating coils. This enables low heating water supply temperatures, increased system ΔT and low coil pressure drop on both the waterside and airside. Six-way valves and high accuracy pressure independent control (PIC) valves are widely available enabling changeover operation. This is an excellent way to retrofit an existing central AHU or terminal unit to enable its use of lower temperature heating water.
- Radiant heaters – Radiant heating systems are commonly designed with lower supply water temperatures making them a natural complement to AWHP based systems.

So what heating water supply temperature is needed to heat the space? It varies some depending on the air handling system application. Table 2 summarizes common minimum heating water supply temperatures and the corresponding ΔT ranges that can satisfy typical commercial heating applications.

Table 2. 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 often has an energy recovery device for preconditioning the air and its supply air temperature in the heating season is commonly 70°F. This allows a DOAS unit to be selected at a low heating water supply temperature with a substantial ΔT.

The most challenging application is in-space fan coil units. This application typically requires a changeover coil configuration. Although a DOAS unit that may be part of a fan coil system could use 95°F, trying to use 95°F supply water for the fan coils may require significant upsizing. In general, 100°F to 115°F heating water is needed to prevent fan coil upsizing. In most cases a terminal unit cooling coil supplying 55°F mixed-air with 45°F chilled water can changeover to heat the air from 60°F to a 100°F with 105°F leaving air temperature with a 105°F supply heating water temperature.

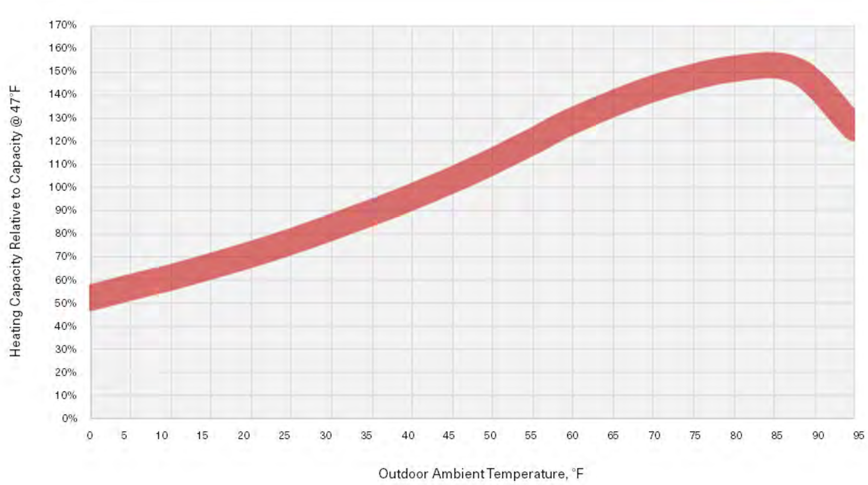
Careful attention to coil selection can allow systems to successfully operate with 100°F to 110°F heating supply water temperatures for any climate.

AIR-TO-WATER HEAT PUMP SYSTEM SIZING

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

The outdoor air temperature significantly impact the equipment full load heating capacity. A commonly used AHRI *rating point* for heating capacity is at 47°F OAT. However, the typical **outdoor air design** heating temperature is much colder resulting in a substantial capacity adjustment. Figure 12 shows the relationship between available heat pump 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 will see a 40 to 50 percent reduction in rated capacity. The selected heating water supply temperature impacts capacity as well, but in a more limited way.

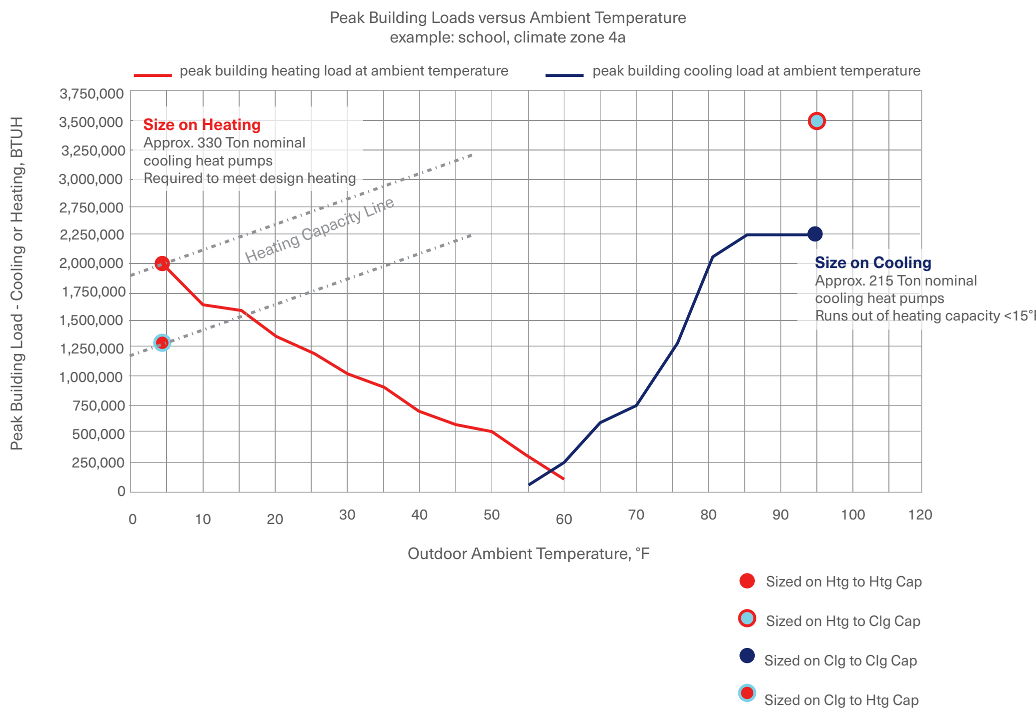
Figure 12. Ambient impact on heat pump capacity



AIR-TO-WATER HEAT PUMP SYSTEM CAPACITY SIZING EXAMPLE

Consider the high ventilation rate school heating load shown in Figure 8 (p. 18). Figure 13 shows the impact of sizing the equipment for cooling design versus heating design conditions.

Figure 13. Sizing AWHP system capacity



- If the system is sized for cooling, the system will fall short of the design heating capacity below 15°F OAT because of the impact cold ambient conditions have on heating capacity.
  Supplemental heat from an electric or gas boiler can be used to address the capacity shortfall and may represent the best balance between system first cost and carbon emissions reduction.
- Alternatively, sizing the system for heating results in excess system cooling capacity (+50 percent) and increased first cost.
  When providing an oversized solution, it is important to evaluate the system load versus the number of AWHP units applied for both acceptable capacity and AWHP flow and capacity turndown to effectively meet the many operating hours at relatively low loads.

Uncertainty regarding actual building loads and its varied use will cause many designers to add significant safety factors to their design assumptions. This tendency may be more prevalent as it relates to building heat because the consequences of inadequate heat capacity are particularly concerning. However, oversizing equipment has consequences to system efficiency, operation and particularly system first cost. Considering the relatively high-cost of heat pump equipment, it is important to use realistic load design assumptions and safety factors.

Make sure the building design load is based on a detailed computerized load analysis and system "Block Loads" and not "Sum of The Peaks" of coil loads.

An important design consideration is that the majority of heating operating hours are at low loads. A careful examination of Figure 8 (p. 18), reveals that for all the building examples shown, 50 percent of the operating hours are less than 25 percent of the design heating load. And less than 20 percent of the heating hours require more than 50 percent of the design load. This reality is common for many commercial buildings. It is important to evaluate the system for adequate turn-down capability to address numerous hours at lower loads.

AIR-TO-WATER HEAT PUMP UNIT SIZING

The final equipment selection should consider the heating capacity impact of the AWHP units' defrost operation. See the defrost implications in Table 3. For this example, the design outdoor air temperature is 4°F. The defrost derate table recommends a 15 percent capacity increase to meet the intended design capacity requirement.

Packaged equipment with its discrete capacity unit sizes typically leads to slightly oversized selections. The closest unit selection that meets the building block load including the defrost derate is two nominal 200 ton units. This selection results in a reasonable 120 percent of design block load selection.

Table 3 shows possible unit sizing options for the school example using two equally sized ACX units.

Table 3. 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		Design Day = 0°F to 5°F (+15%) Design Day = 5°F to 20°F (+10%)	

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

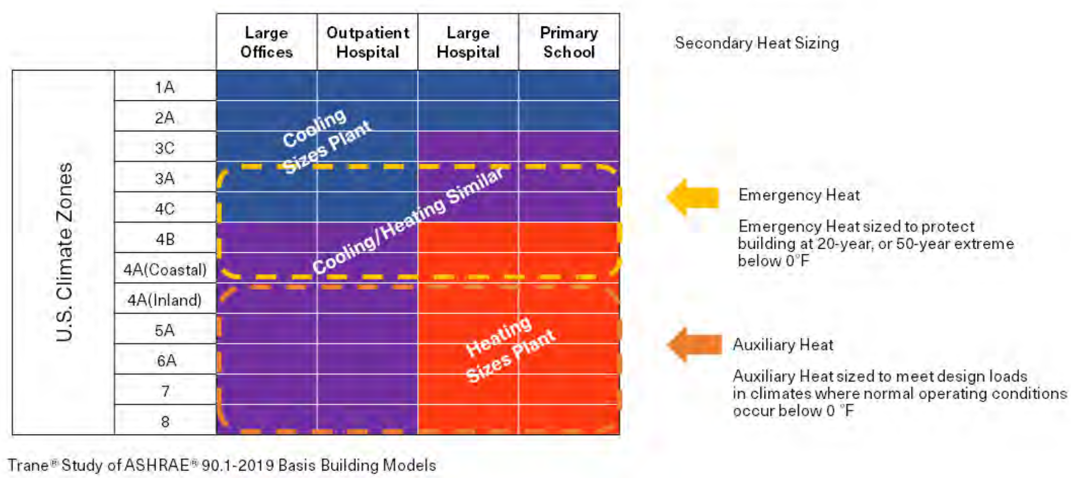
CHILLER-HEATER PLANT SIZING DRIVERS

Building type and climate zone determines the required installed capacity of heat pumps. Using the ANSI/ASHRAE/IES 90.1-2019 building models, Figure 14 was developed as a demonstration of whether design cooling load or design heating load is expected to drive the heat pump system size. Based on the building type and U.S. Climate Zone:

- The dark blue indicates cooling design load likely driving the plant capacity.
- The red area indicates heating design load likely driving the plant capacity.
- The purple area indicates that heating and cooling loads are balanced relative to plant capacity.

Figure 14 also demonstrates higher ventilation rate buildings are more likely to be heating design driven (hospitals and schools). It also identifies when a secondary heating system (e.g., boilers) is required for auxiliary heating or emergency heating purposes (Climate Zone 4A and higher). While certainly not a replacement for specific design work it is a helpful guide.

Figure 14. Plant sizing basis per climate zones



SIMULTANEOUS HEATING AND COOLING

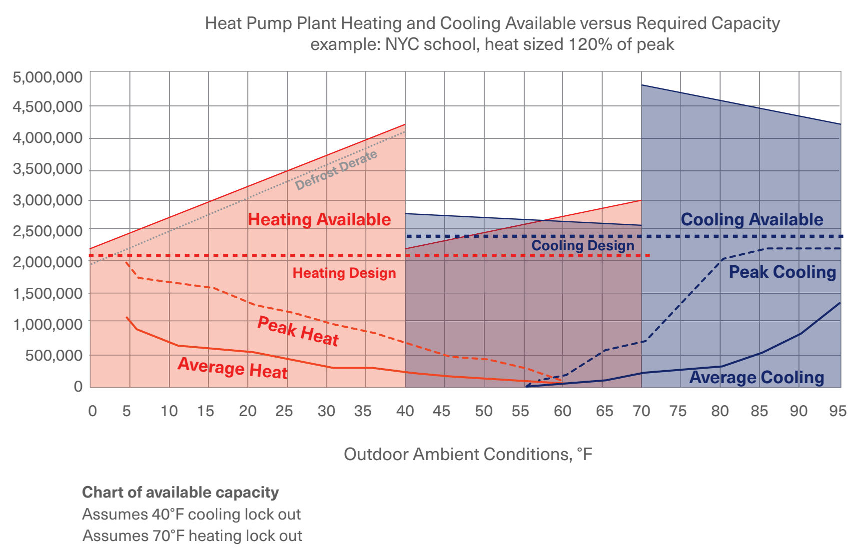
Dividing the system capacity between two or more units provides multiple benefits. First, it improves system turn-down capability which is especially beneficial to the many operating hours at lower loads. Second, it provides adequate capacity to meet simultaneous heating and cooling loads and efficiently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

Figure 15 shows the available heating and cooling capacity at varying ambient temperatures for the school example using the ACX units chosen to deliver 120 percent of design heating capacity at design heating conditions.

- Cooling and heating lockout assumptions are 40°F and 70°F respectively.
- 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.
- Between 40°F and 70°F OAT the available cooling and heating capacity with one unit operating is represented.
  - Notice that the full design heating and cooling capacity is available between 40°F and 70°F with only one unit.

It is easy to see how the equipment capacity changes with outdoor air temperature as represented by the slope of the top of the unit capacity curves. Also, the simultaneous heating and cooling loads that exist between 55°F and 60°F OAT can easily be met with one unit in cooling and the other in heating mode. The ACX units perform at high efficiency in this mild temperature range which means that a dedicated heat recovery unit (DHR) will have a relatively small efficiency benefit. However, a properly sized (small) DHR unit can eliminate the need to operate two AWHP units, both at low loads to meet the simultaneous loads. This can extend the life of the AWHP units as well as provide some system control and efficiency benefit. Note that the size of the DHR unit to provide this benefit is quite small, approximately 220,000 BTUH (18 tons) in this school example.

Figure 15. Heating and cooling loads and plant capacity versus ambient temperature



DEFROST IMPLICATIONS TO SIZING

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

Defrost operation results in a weighted performance derate to the equipment heating capacity. Some designer intuition is involved because the frequency, and therefore impact of defrost operation, is dependent on actual operating conditions. Table 4 offers suggested heating capacity derate ranges based on design outdoor air temperature. As discussed in the previous unit sizing exercise, the derate factor should be applied to the required unit design capacity for the specified equipment capacity in the equipment schedules and schedule notes should clarify this has been done. This will help ensure equipment selections include the proper defrost impact.

Table 4. Unit defrost capacity derate factors

Outdoor Air Temperature, °F	Capacity Derate Factors to Selected Heating Capacity
>47	1 (no derate)
35 to 47	0.95 to 0.98
20 to 34	0.90 to 0.95
5 to 20	0.85 to 0.90
0 to 5	0.80 to 0.85

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

WEATHER EXTREMES AND AUXILIARY HEAT

Auxiliary Heat is defined as heat from an alternate source from the AWHPs that operates only when the AWHPs **cannot operate** because of outdoor ambient conditions or AWHP system component failure. If outdoor conditions are cold enough (such as below 0°F) AWHP units will not be able to operate. In northern climates this will occur with normal variations in weather. In other climate zones, it may be abnormal, but can be part of the 20- or 50-year extreme conditions. Regardless of climate fluctuations, a plan for an alternate heat source 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 fluctuations (and extremes) would be a good design strategy. Since it is used infrequently, its impact on annualized carbon emissions is limited, so the use of high efficiency fossil fuel boilers should not be ruled out.

SUPPLEMENTAL HEAT

Supplemental heat is defined as heat from an alternate source from the AWHPs **in addition** to the heat provided by the operating AWHPs. The concept of supplemental heat enables another form of design optimization. As it's been pointed out, sizing for design heating capacity can lead to substantial oversizing for cooling, higher first cost, and excessive low load compressor cycling. One alternative is to use the auxiliary heat system to supplement "right sized" AWHP units heating capacity during peak heating loads. This approach makes particularly good sense if an auxiliary heat system will already be required to address 20-, 50- or 100-year extreme weather episodes.

In the school example:

- Two nominal 200 ton AWHP units are required to meet the design heating load.
  - This results in a 50 percent oversizing of cooling capacity and the associated costs.
- Two nominal 160 ton AWHP unit are required to meet the design cooling load.
  - These units can meet all heating loads down to 15°F OAT
  - Below 15°F there are a total of **17 hours per year** that the AWHPs would not be able to meet the full heating load. For these 17 hours the auxiliary heating source would be energized to meet the balance of the building heating load.

The operation of the two systems needs to be designed and sequenced so the auxiliary heat system does not unintentionally steal load from the heat pump system.

REDUNDANCY

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

- If the system design includes an auxiliary heat source that can provide the required heating redundancy at no additional system cost.
- One or more (N+1) additional heat pump units can be added to provide both heating and cooling redundancy.
- Cooling redundancy can be provided with a properly sized cooling only chiller. In fact, the cooling only chiller will prove to be more efficient at cooling than an AWHP and could be used as the lead cooling unit for better annualized system efficiency. This concept is applicable:
  - If the system has Auxiliary Heat.
    OR
  - If the system/units are sized for heating and 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. One example is an air-cooled chilled-water system 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 5 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 5. Typical antifreeze concentrations by volume

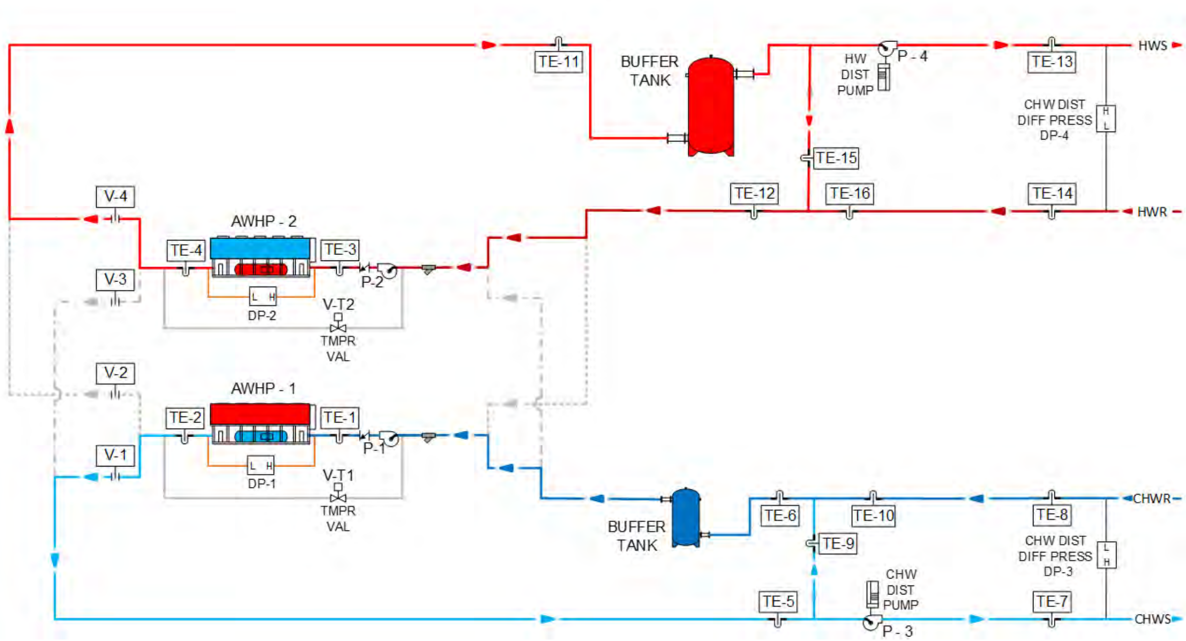
Temperature, °F	DOWTHERM™ SR-1 (ethylene glycol)		DOWFROST™ HD (propylene glycol)	
	Freeze	Burst	Freeze	Burst
20	16.8%	11.5%	18%	12%
10	26.2%	17.8%	29%	20%
0	34.6%	23.1%	36%	24%
-10	40.9%	27.3%	42%	28%
-20	46.1%	31.4%	46%	30%
-30	50.3%	31.4%	50%	33%
-40	54.5%	31.4%	54%	35%
-50	58.7%	31.4%	57%	35%
-60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

Freeze Avoidance: Important: Even if a unit is located in a climate which rarely sees freezing ambient temperatures, the unit can still experience freezing conditions due to refrigerant migration.

System Configuration

The base configuration—four-pipe distribution with dual-feed chiller-heater system—recommended for use with multiple ACX heat pumps is represented in Figure 16. While other system configurations are possible, and will not be unusual, this one provides excellent flexibility and simplicity of design and control.

Figure 16. Four-pipe distribution, dual-feed chiller-heater system



Note: *The 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 job design/consulting 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, the system example circulates the same fluid throughout the production circuit and heating and cooling distribution loops. Often an anti-freeze solution is desired in the outdoor production circuit, allowing the solution to circulate throughout both distribution loops. This is the simplest and safest application since it does not rely on any powered or mechanical freeze protection strategy. Fluid isolation heat exchanger(s) can be properly applied between the production and either or both distribution loops if the designer calls for differing fluids in the distribution loop(s). The designer must take into consideration that below freezing temperature may enter a heat exchanger from the production loop during system changeover or a defrost cycle and take the appropriate design precautions.

FOUR-PIPE DISTRIBUTION

As the name implies the system has four-pipe distribution including separate heating and cooling distribution loops. These are standard distribution loops and may be optimized for the supply temperatures, flows, and temperature changes as required by the air-side design. The air-side coil capacity may be controlled with two-way valves causing widely variable distribution pumping flow for both the chilled water and heating water distribution. This provides for significant operational flexibility and opportunities for pumping energy savings.

At the heart of the system are the decoupler lines. These lines provide the hydronic isolation that allows for optimization of flows and temperatures in both the distribution and production loops.

DECOUPLING – HYDRONIC ISOLATION

Decoupling greatly simplifies system design and allows an array of sizes and types of production units that can be applied to best match building load requirements. The principal requirement for the heat pump selection is that it can produce the supply water TEMPERATURE required for cooling or heating. Unit flow and pressure drop requirements are of much less concern as the decoupler allows for natural balancing of required flows.

The primary/secondary chilled water system decoupler pipe must be configured and sized to meet two primary 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.

Achieving these requirements typically means observing several principles. Refer to Figure 17 for more details.

Pipe connection tees: The decoupler pipe should be configured so that it enters and exits in and out the side of the return and supply system piping with tee type connections. This is to prevent water velocity momentum in the supply or return pipe from inducing flow and/or mixing in the decoupler pipe. See Figure 17.

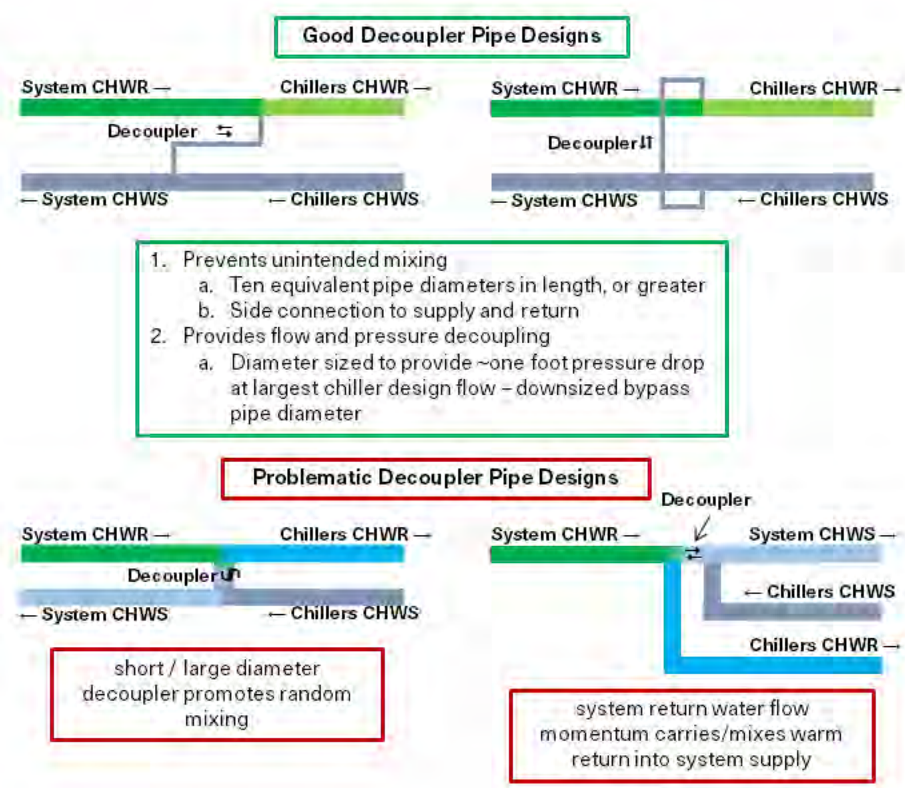
Pipe diameter: The decoupler pipe diameter sizing differs depending on the plant operating intent.

a. For systems with constant heat pump flow it should be sized based on the higher of the cooling or heating design flow of the largest unit in the plant. This typically means no larger— often a size smaller—than the diameter of piping connecting to the largest unit. In multiple unit systems it should NOT be sized for **system** design flow, such as equal to the distribution piping diameter.

b. For Variable Primary / Variable Secondary systems, with varying flow through the operating heat pumps, it should be sized based on the MINIMUM recommended flow of the largest unit in the plant. This is typically one or more pipe sizes smaller than the diameter of piping connecting to the largest heat pump. Although if the heat pump's selected design flows are close to the minimum allowed heat exchanger flow it may be the same size as the unit piping connection. Larger is not better. Larger increases the likelihood of undesired flow mixing and increases installation cost.

Pipe length: The decoupler pipe length should be approximately ten (10) equivalent pipe diameters long or greater (elbows are counted appropriately). Another rule of thumb is for the pipe to have about one (1) foot of pressure drop at the decoupler design flow. In large chilled water/ hot water systems a somewhat higher pressure drop will not cause operational problems.

Figure 17. Decoupler pipe connection examples



System Fluid Volume

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

HEATING LOOP VOLUME

The heating loop minimum fluid volume requirement is typically higher than chilled water loops. The key reason heating loops require greater volume is to compensate for unit defrost operation. When a unit goes into defrost the unit leaving fluid temperature drops rapidly. This is particularly so if only one refrigeration circuit is operating and goes into defrost. In a system with inadequate fluid volume at least three issues can occur:

1. The cold fluid can cause the air-handers or terminal units to dump cold air into the space causing occupant discomfort.
2. The cold fluid can cause low temperature alarms or freezestat trips in AHUs.
3. The return fluid temperature to the heat pump can get cold enough to cause a heat pump unit diagnostic.

Mitigating these problems for units with two refrigeration circuits that do not defrost at the same time requires a heating loop minimum fluid volume of 0.62 gals/per Mbh at the AHRI 120°F LHWT/47°F OAT heating rating point. This is approximately eight gallons per rated heating ton at those conditions, for the largest capacity heat pump in the system.

The system is calculated based on the piping, coil, and chiller internal volumes. For systems which do not meet the recommended fluid volume a volume buffer tank must be installed in the production plant heating supply line. Note that this is different than chilled water loops where a buffer tank, if needed, is typically installed in the return line to the chillers. Locating the buffer tank in the heating supply line allows it to moderate both the heating system supply water temperature and AWHP return water temperature swings that occur during unit defrost cycles.

COOLING LOOP VOLUME

For the chilled water section of the system the typical minimum fluid volume requirements of providing for a two-minute circulation time is recommended. That translates to a volume of water in the chilled water loop equal to or exceeding two times the cooling mode design evaporator flow rate. For systems with a rapidly changing load profile the amount of volume should be increased.

If the installed system volume does not meet the above recommendations, the following items should be given careful consideration to increase the volume of water in the system and, therefore, reduce the rate of change of the return water temperature:

- A volume buffer tank located in the plant return chilled water piping to the chiller.
- Larger system supply and return header piping (which also reduces system pressure drop and pump energy use).

Modes of System Operation

Figure 18. Example of cooling only system operation

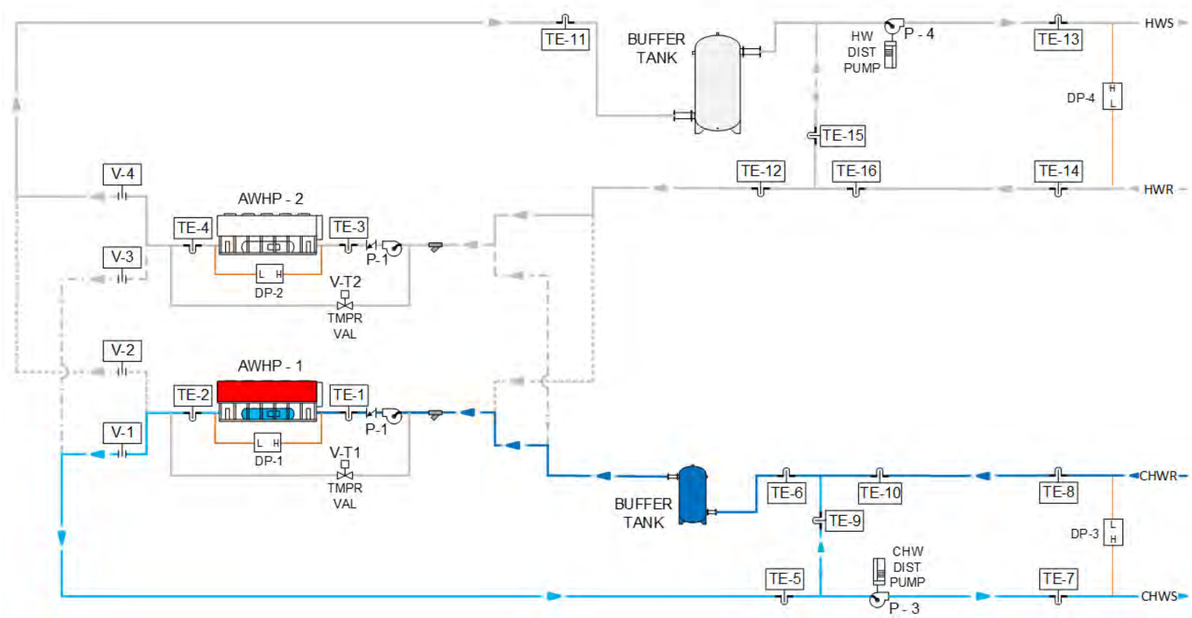


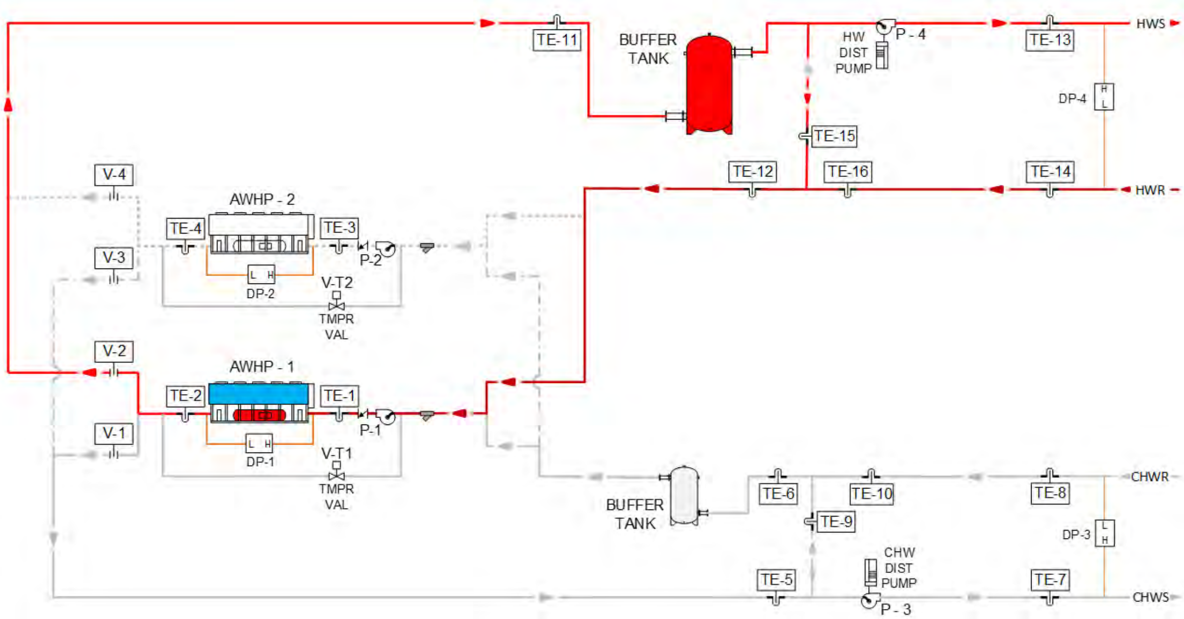
Figure 18 shows the system in a cooling only mode with only one unit running and it operates similar to any chilled-water system. This is shown with the lower unit operating in cooling, but either or both units can provide cooling.

The chilled water distribution loop can be variable flow with two-way control valves on unit coils and the pump(s) speed control based on a water differential pressure in the system. Optimization strategies such as critical-valve pump pressure optimization should be applied per required codes.

The dual-feed production module mode control valves (V-1 and V-2) are controlled to supply chilled water as required. The heat pump operates at the system required chilled-water setpoint. Optimization strategies such as chilled water temperature reset can be applied if appropriate for the chilled water distribution and air-side system design.

The production loop/unit pumps are controlled to constant heat pump cooling mode flow, which would be typical if the distribution design flow rate is close to or less than the chillers' minimum allowed flow. If the chillers' design flow is twenty percent or greater than its minimum allowed flow, implementation of variable primary/variable secondary flow control logic should be considered. VP/VS control would reduce the annualized pumping energy consumption and cost, equal to or below what a VPF configuration might provide, and its control is easier and more sustainable.

Figure 19. Example of heating only system mode



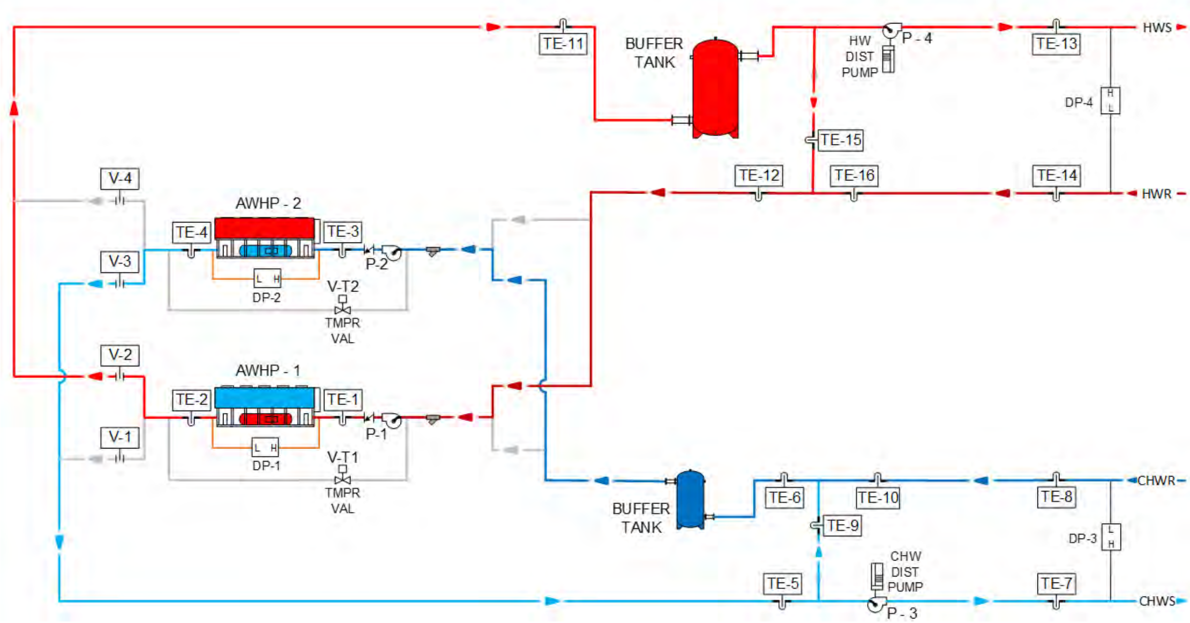
The example in Figure 19 shows the system operating in the heating only mode with the same unit running. This is shown with the lower unit operating in heating, but either or both units can provide heating.

The heating water distribution loop can be variable flow with two-way control valves on unit coils and the pump(s) speed control based on a fluid differential pressure in the system. Optimization strategies such as critical-valve pump pressure optimization should be applied per required codes.

The dual-feed production module mode control valves (V-1 and V-2) are controlled to supply hot water as required. The heat pump operates at the system required heating water setpoint. Optimization strategies such as heating water temperature reset can be applied if appropriate for the heating water distribution and air-side system design.

The production loop/unit pumps can be controlled to constant heat pump heating mode flow, which would be typical if the distribution design flow rate is close to or less than the heat pumps' minimum allowed flow. It's likely the heating distribution system flow is less than the minimum allowed flow for the air-to-water heat pump fluid heat exchanger. However, if the heat pumps' **design** flow is twenty percent or greater than its **minimum** allowed flow implementation of variable primary/variable secondary flow control logic should be considered. Less than twenty percent of allowable flow turndown does not provide for adequate control response. However, if sufficient flow turndown is allowable meaningful pumping energy savings is possible.

Figure 20. Example of simultaneous heating and cooling



The example in Figure 20 shows one heat pump unit providing cooling and the other heat pump is providing heating. This is a common mode of operation during shoulder seasons depending on the building type. The “Heat Recovery Requirements,”(p. 15) section of this application guide discusses adding heat recovery to the system which reduces the simultaneous operation of multiple units and saves energy.

Figure 20 is shown with the lower unit heating and the upper unit cooling to demonstrate how the dual-feed module piping allows any unit to serve either load at any time. The chilled water and heating water distribution loops are controlled as in the cooling only and heating only modes of operation.

The dual-feed production modules mode control valves (V-1, V-2, V-3, and V-4) are controlled to supply chilled and hot water as required. The individual heat pumps operate at the system required chilled water and heating water setpoints. Optimization strategies such as chilled water temperature setpoint reset and/or heating water temperature setpoint reset can be applied if appropriate for the heating water distribution and air-side system design. The production loop/unit pumps are controlled as appropriate for the heat pump unit operating mode as signaled by the BAS system.

System Options

There are several options for the chiller-heater system. Depending on the specific building requirements there are options to increase efficiency, increase accuracy of control and/or improved redundancy.

AUXILIARY OR BACKUP HEATING

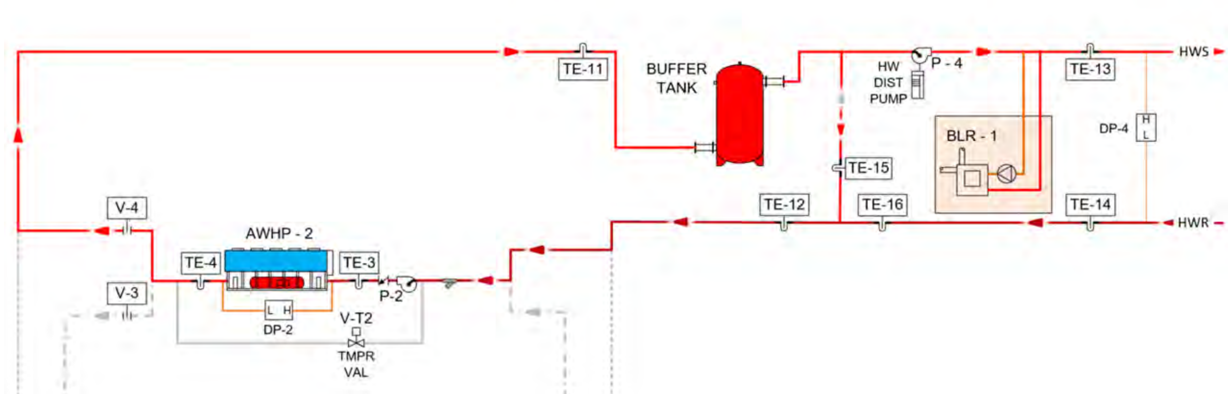
Auxiliary heat will be common on many systems for one or more of the following reasons:

1. Air-source heat pumps have a minimum outdoor air temperature operating limit. The Trane® ACX unit's heating operation minimum outdoor air temperature is 0°F at the time of this application guide's publication. Below that temperature the unit compressors will be locked out of operation. An alternate heat source will be required if the building is in a location that may experience temperatures below the unit limit.
   It is the responsibility of the design engineer to look closely at the long term weather data for 50- or 100-year extremes to help decide if an alternate heating source is required.
2. Resilient buildings often require backup power generation to maintain building operation through utility power failures. Air-source heat pumps drive the need for large generation capacity and fuel storage. Natural gas, propane or fuel oil boilers may greatly reduce the required generator capacity. The building temperature can be maintained by running only the heating system distribution pumps and the boiler. This reduces the cost and space required for the back-up generation infrastructure.
3. For heat pump unit failure recovery, a low first cost alternate heating source, rather than an additional air-to-water heat pump unit should be evaluated. Alternate heating sources such as electric or natural gas boilers may have a much lower first cost than an additional heat pump. If they are only run occasionally, for failure recovery or heat pump unit maintenance, their potential impact on the building's carbon footprint may be minimal. See point four below. Of course, the full first cost to the building infrastructure must be considered. Electric boilers may require upsizing of the building electrical service and gas boilers require a gas supply infrastructure.
4. A building's life cycle carbon footprint may be reduced by using natural gas or propane for auxiliary heat. The current grid is **NOT** carbon free and will likely not be for some time. The carbon impact of generating electricity varies depending upon location throughout the country. As shown in Figure 11 (p. 22), under some conditions heating with a high-efficiency gas boiler actually results in lower carbon emissions than heating with air-source heat pumps. This is a result of the combination of the significant decrease in air-to-water heat pump COP at low ambient temperatures and the efficiency (carbon impact) of generating and delivering energy to buildings. As the grid becomes cleaner and heat pump technology improves throughout the life of the building the boiler use can reduced or eliminated to provide ongoing carbon emission minimization.

When auxiliary heat is applied in the hydronic system it is best connected into the heating distribution loop supply line as shown in *Figure 21*. This position has the advantage of allowing the auxiliary heat to supplement the heat pump capacity when required or provide stand-alone heating with operation of the distribution loop pumps only and auxiliary heat with the heat pump production plant shut down. The auxiliary heat source can take practically any form, from low-cost conventional natural gas boilers to electric heat.

The previous sections on auxiliary and supplemental heat discussed that the control of auxiliary heat will vary depending on the installation need and the building operating conditions. The design engineer must carefully define the control sequence to ensure the system design goals are achieved.

Figure 21. Auxiliary heat source in distribution loop supply line



HEATING REDUNDANCY

Redundancy in a heating system is a core requirement. There are fundamentally two basic ways to provide heating redundancy:

1. Additional air-to-water heat pump units.
2. An alternate auxiliary heat source.

Each has its own set of pros and cons.

Redundant Heat Pump Unit(s). Installing N+1 AWHP units provides all electric redundancy. The incremental cost of the additional AWHP relative to the base system cost varies greatly depending on the number of base units required. Table 6 demonstrates this concept by showing that as the number of base heat pump units increases there is a reduction in the incremental capacity addition as relative to the design system load.

To provide redundancy in a system with a single unit, double the design capacity must be installed. The first cost of the heat pump units and infrastructure may be less by installing three smaller units rather than two large ones. And the system will benefit from the more stable control and greater capacity turndown than a single large base unit would provide.

Table 6. Redundant unit sizing examples

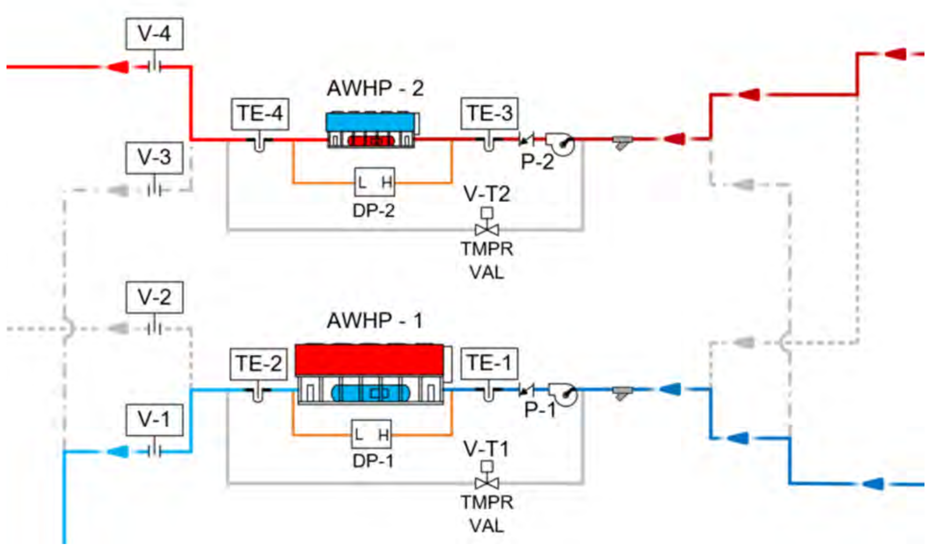
Base Number of HP Units to Meet Load	Redundancy	
	Unit Size	ACX Installed Cap
3 AWHP	33% of system load	= 1.3 x heating load
2 AWHP	50% of system load	= 1.5 x heating load
1 AWHP	100% of system load	= 2 x heating load
1 AWHP	auxiliary heat	= heating load

In the four-pipe distribution, dual-feed chiller-heater system, redundant units are added to the system in parallel with the base units and piping being identical to the base units. The redundant unit can operate similar to any other system unit and can be rotated into operation to serve the cooling or heating loads. Its runtime can be equalized with the other system units if desired. If the system is designed with unequally sized heat pump units, the redundant unit will likely need to be selected with a capacity equal to the larger unit in the system.

UNEQUALLY SIZED UNITS

Figure 22 represents a production system with unequally sized units, perhaps with the classic one-third/two-thirds sizing. For many buildings, with significant low load hours, this can provide a number of benefits. These benefits include greater plant turndown of capacity and flow by providing improved efficiency, more stable operation, and more accurate temperature control.

Figure 22. Unequally sized heat pump units



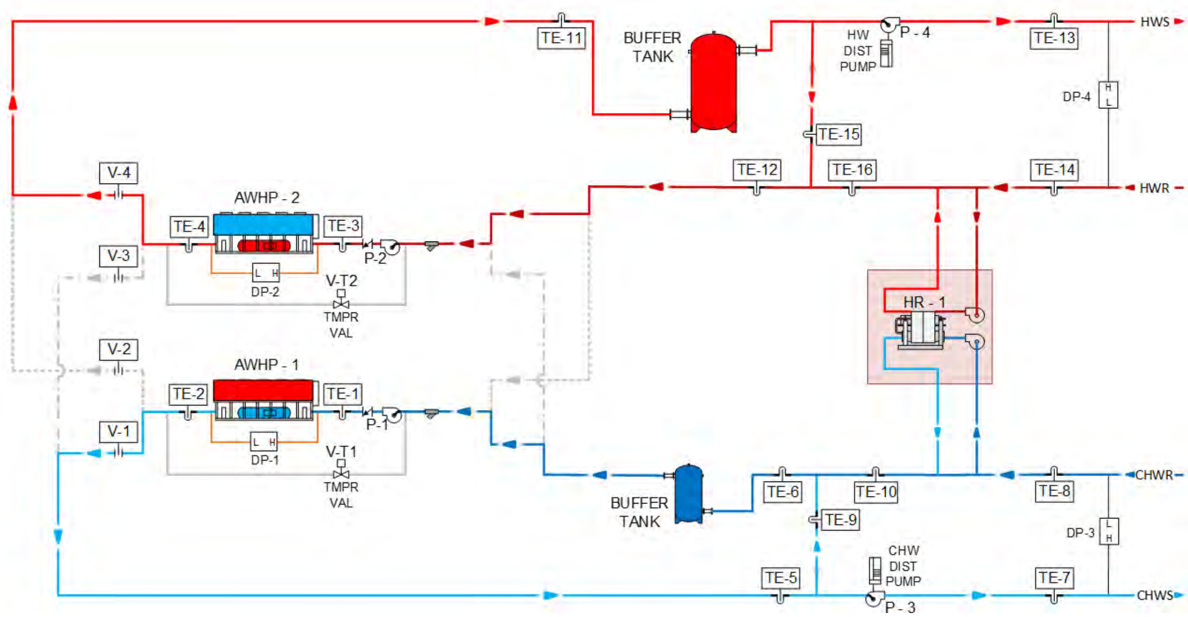
The dual-feed configuration of the smaller unit can serve either the heating or cooling load benefiting the low load operation of both.

The decoupling of both the chilled water and heating water from the dual-feed production modules enables sequencing and control of unequally sized units to either distribution loop. As mentioned previously, if heating redundancy is to be provided with an additional heat pump unit it should be sized equal to the larger unit.

DEDICATED HEAT RECOVERY

Another popular option in electrified systems is dedicated heat recovery (DHR). A DHR unit is simple to apply between the two distribution loop return pipes. A DHR unit can only move energy from one distribution stream to the other.

Figure 23. Dedicated heat recovery unit option



Proper sizing of the DHR is critical to the cost effective design and selection of the system. The amount of energy that can be transferred between the loops is limited to the lower of the two loops' loads at any moment in time. The result is a substantially less design capacity of a DHR unit than the air-to-water heat pump units.

The proper way to size the DHR capacity is by performing an 8760 hour load analysis for the heating and cooling loads and then extracting the minimum of the two loops' loads for each hour of the year where heating and cooling loads occur coincidently. This hourly minimum coincident data can then be evaluated to determine the appropriate DHR sizing.

For any installation, it's worth comparing the potential benefit of applying a DHR to investing in other competing efficiency measures such as economizer cooling. However, there are benefits beyond system efficiency that heat recovery provides. The application of DHR in the system provides all the benefits of a system with an air-source multi-pipe unit at less cost and better annualized efficiency.

DHR can reduce the time when two air-to-water heat pump units need to operate, one for cooling and one for heating. Referring back to the school load example, heat pump operating hours are reduced by a total of up to 1600 hours per year as shown in Table 7. By recovering energy from one loop to the other, the DHR can fully meet one required load and partially meet the other. A single air-to-water heat pump unit meets the remainder of the dominant loop's load. This can help stabilize system operation and extend the heat pump units' operating life.

Table 7. DHR impact on heat pump operation

Fraction of Design Capacity		Operating Time without DHR	Operating Time with DHR	Operating Time Reduction
annual time		hours	hours	hours
NYC school	cooling	2,596	1,820	776
	heating	3,017	2,193	824

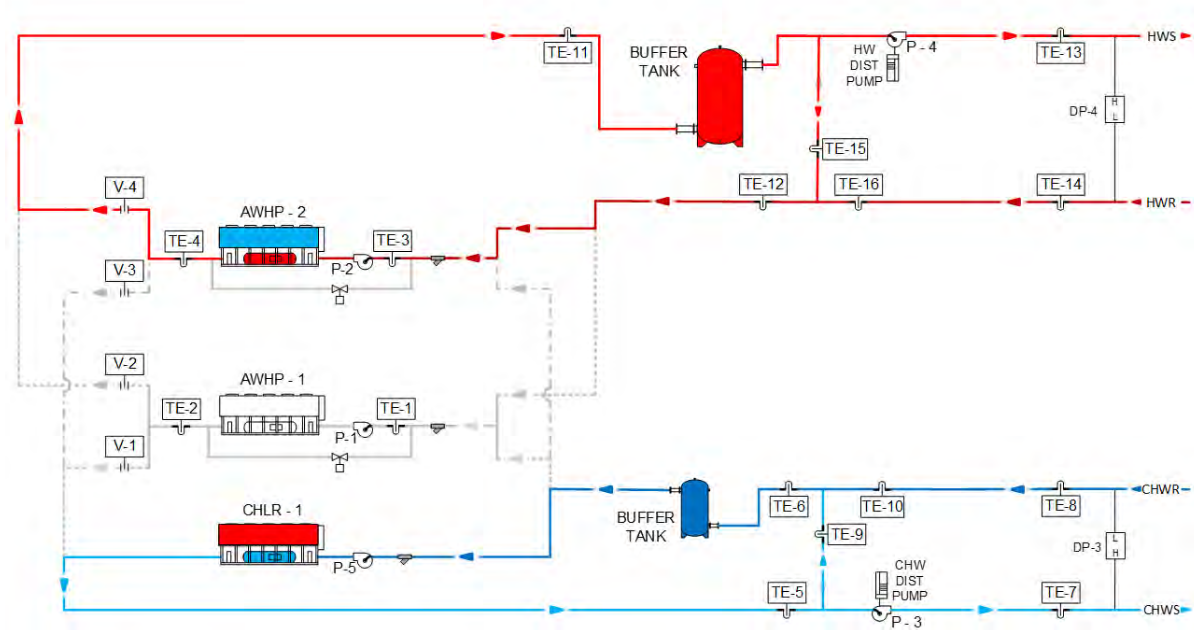
DEDICATED CHILLER - COOLING DOMINANT HYBRID OPTION

Many building types have considerable cooling-only or cooling-dominant operating hours. Air-cooled chillers typically have operating efficiencies 5 to 15 percent better than air-source heat pumps in the cooling mode. As a result, some owners will choose to invest in hybrid plants, with some air-source heat pumps and some air-cooled chillers, for better annualized cooling efficiency.

Figure 24 shows a representation of a chiller in that system application.

One or more cooling-only units can easily be added to the cooling loop side of the production plant. Whenever cooling is required, the chiller can run to efficiently meet it. The chiller is typically not sized to meet the design day cooling load but rather something less. A design and analysis tool such as TRACE® 3D Plus can be used to determine the chiller size that provides the best life cycle payback. Heat pumps can provide additional cooling on peak cooling days and cooling redundancy as well as meet building heating loads.

Figure 24. Hybrid plant with dedicated cooling only chiller



ENERGY STORAGE USING THERMAL BATTERIES

Trane® Thermal Battery™ Systems can be used to recover non-simultaneous waste heat and substantially increase the recovered heat as compared to simultaneous heat recovery approaches. It also allows storage of heat with a high COP heat pump during warmer outdoor periods. Stored heat can be used during colder outdoor periods, even below outdoor air temperatures where AWHPs cannot operate, thus reducing system dependence on auxiliary heat. Trane offers a storage source heat pump solution with these benefits. For more information contact your local Trane representative.

Summary

An electrified cooling/heating system based on air-to-water heat pump 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 are different from those in chilled water systems. These must be addressed in order to have an effective and reliable decarbonization system.

As covered in this application guide, there are several key points that the design engineer should be mindful of when designing the system:

- **Ensure system will meet project decarbonization goals.** Understand the applicable grid emissions and select equipment efficiencies and system operating temperatures that ensure goals will be met.
- **Auxiliary heat may be required** due to extremes in outdoor air temperature that can often exceed the operating range of the heat pump technology.
- **Equipment selection should account for coldest design conditions** as the outdoor air temperature has a significant effect on heat pump unit capacity and maximum available supply hot water temperature.
- **Lower design and operating hot water supply temperatures results in more efficient heat pump unit operation.** The target for the central air handler and terminal equipment design heating water temperature should be 95°F to 105°F. One method to enable this temperature range is to changeover the unit cooling coil for use in heating.
- **Proper system/equipment sizing is key to efficient and reliable operation.** Improper equipment sizing penalizes system efficiency and shortens equipment operating life, therefore reducing the benefits of decarbonization of the system.
- **Decoupling of the production and distribution systems' pumping** is the most reliable and efficient method to ensure continued reliable system operation over its full operating range.
- **Air side design, specification and control is key to achieving maximum decarbonization.** In addition to heating fluid temperature considerations, features such as air side heat recovery, and economizers should be considered.
- **When considering the application of waterside heat recovery a careful analysis of the full year (8760 hours) simultaneous load curves is required for proper equipment sizing and heat recovery energy and operating cost benefit estimation.**

When taking these key points into account during the design process, the design engineer will have greater success in providing a highly flexible, efficient, and reliable electrified heating system.

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.

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