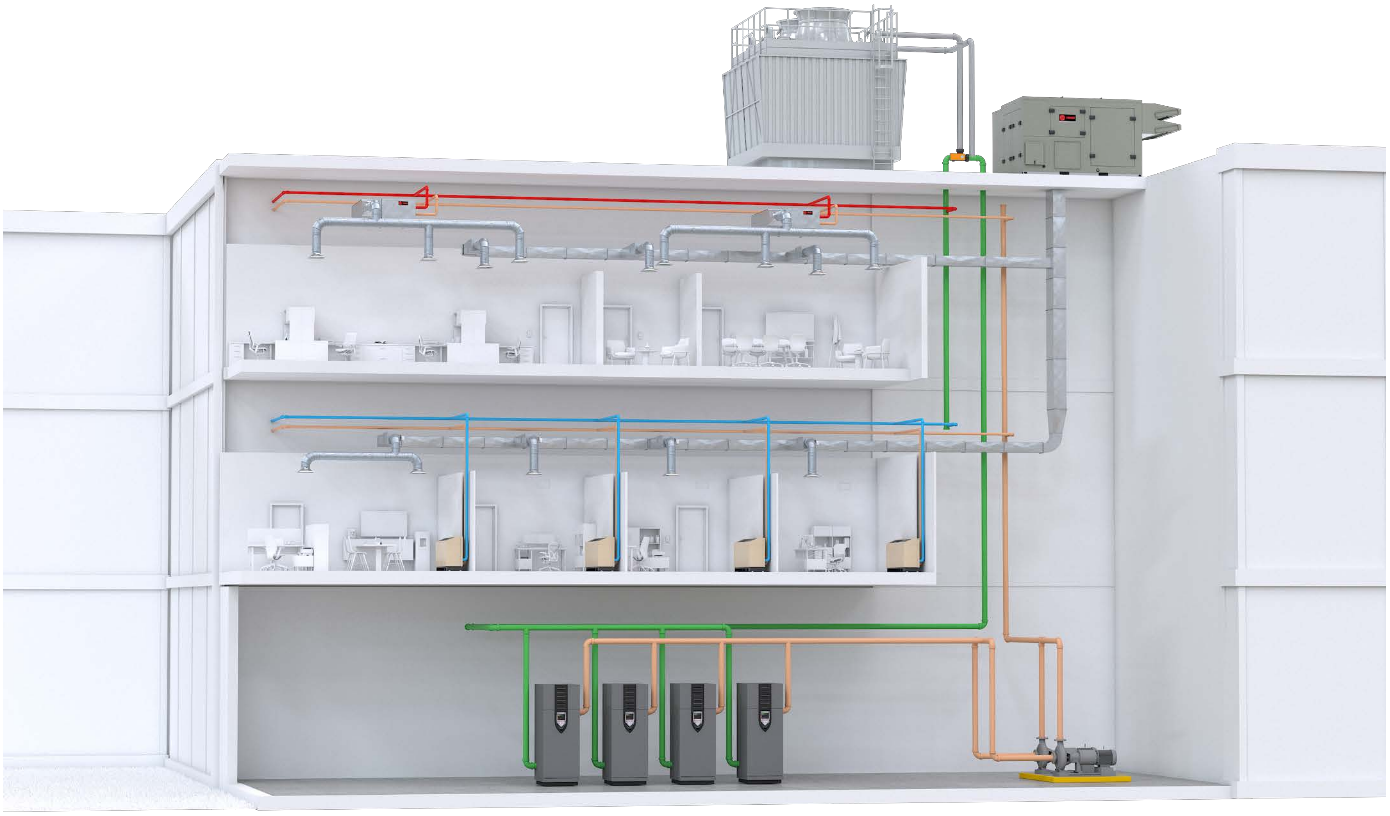
Application Manual

Water-Source and Ground-Source Heat Pump Systems



⚠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.

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Water-Source and Ground-Source Heat Pump Systems

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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 water-source heat pump (WSHP) and ground-source heat pump (GSHP) systems, including boiler/tower, ground-coupled, ground-water, and surface-water systems. These systems are used to provide comfort in a wide range of building types and climates. To encourage proper design and application of a these systems, this guide discusses the advantages and drawbacks of the system, reviews the various components that make up the system, proposes solutions to common design challenges, explores several system variations, and discusses system-level control.

We encourage engineering professionals who design building comfort systems to become familiar with the contents of this manual 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®, in proposing these system design and application concepts, assumes no responsibility for the performance or desirability of any resulting system design. Design of the HVAC system is the prerogative and responsibility of the engineering professional.

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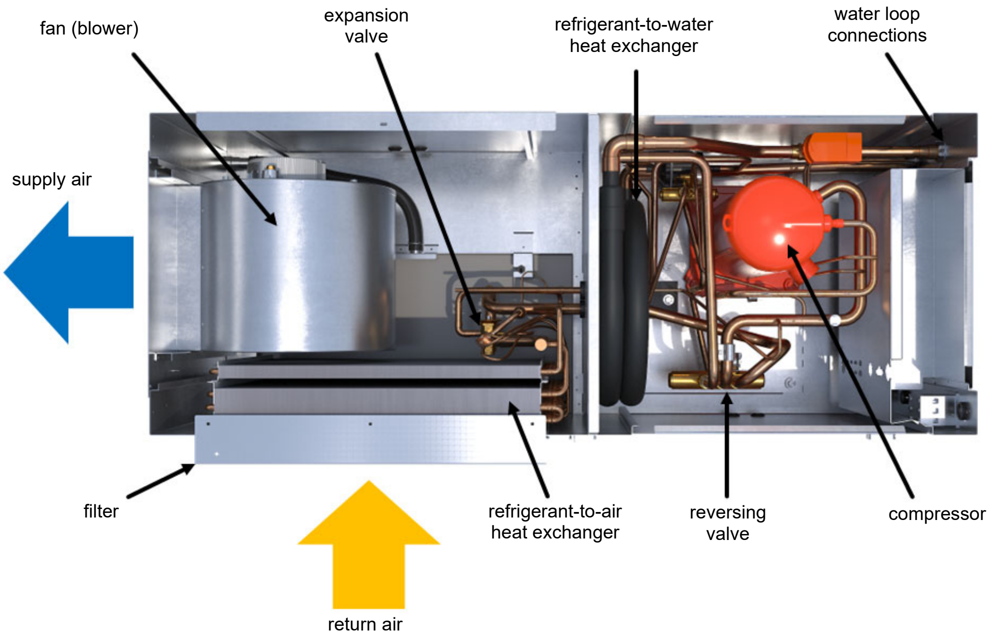
Overview of a Water-Source Heat Pump System

In a typical water-source heat pump (WSHP) system, each zone is served by a dedicated WSHP that cools or heats air to maintain the desired temperature in that zone. A WSHP contains all the components of a refrigeration circuit, including one or more compressors, a refrigerant-to-air heat exchanger, a refrigerant-to-water heat exchanger, and an expansion device (Figure 1). In addition, a reversing valve allows the WSHP to reverse the direction of refrigerant flow, and change the operation of the refrigeration circuit to provide either cooling or heating.

Figure 1. Components of a typical WSHP

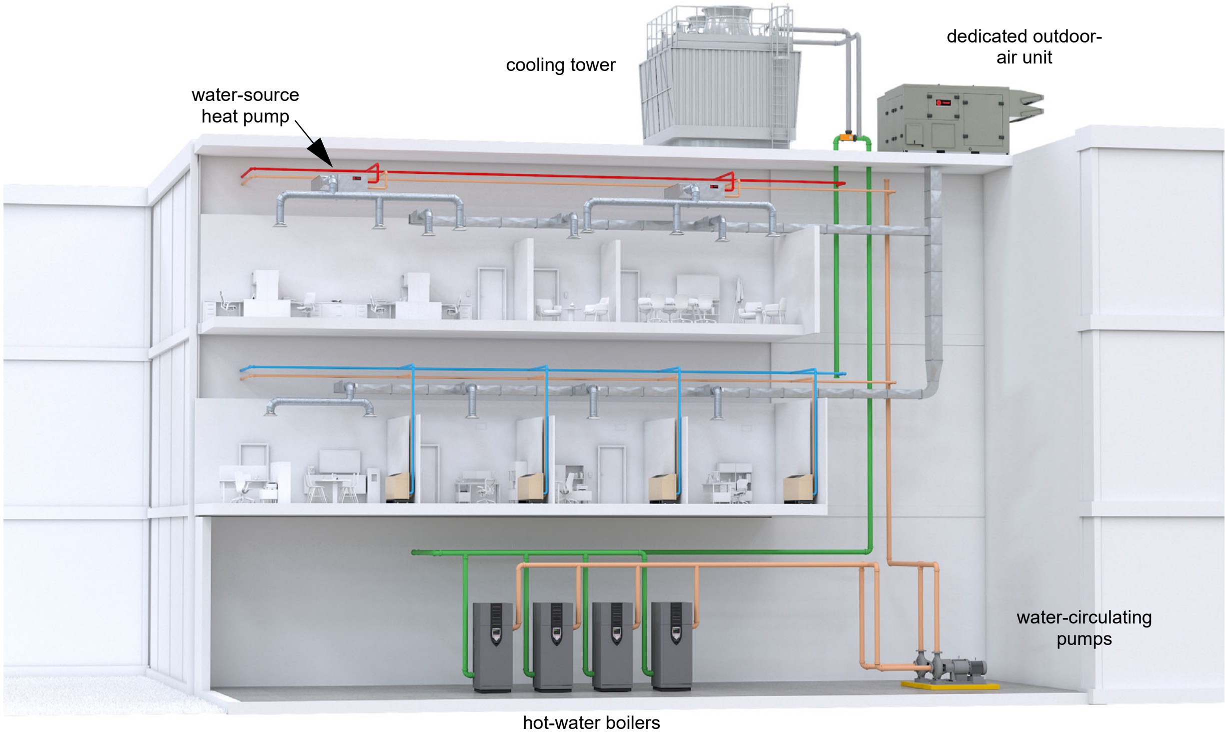


Depending on the style of equipment used, a WSHP may be installed along the wall within the occupied space, in the ceiling plenum above the space, in a closet or mechanical room near the space, or on the roof of the building.

Return air from the zone is drawn into the WSHP through the intake (Figure 1). This air passes through a filter and refrigerant-to-air heat exchanger, before the fan discharges it either directly into the zone or through supply ductwork and supply-air diffusers.

All the heat pumps are connected to a common water loop (Figure 2). Also connected to this loop are a “heat rejecter” (such as a cooling tower, fluid cooler, or ground heat exchanger), a “heat adder” (such as a hot-water boiler or ground heat exchanger), and water-circulating pumps.

Figure 2. Primary components of a water-source heat pump system



Typically, outdoor air required for ventilation is conditioned and delivered by a separate, dedicated outdoor-air system.

Each WSHP is equipped with a unit controller that regulates cooling and heating for the zone it serves. A system-level controller coordinates the operation of the individual WSHP unit controllers so they operate together as an efficient system.

Basic System Operation

The following section describes, in a very simple manner, how a WSHP system operates. For a more detailed discussion, see “System Controls,” p. 166.

Zone is occupied and requires cooling

A sensor in each zone compares the dry-bulb temperature in the zone to a setpoint, and the controller in the WSHP cycles (or varies the speed of) one or more compressors to match the changing cooling load in the zone. As the cooling load decreases, the compressor operates for a shorter period of time between cycles.

Inside the WSHP, the refrigeration circuit extracts heat from the recirculated air and rejects heat to the water loop.

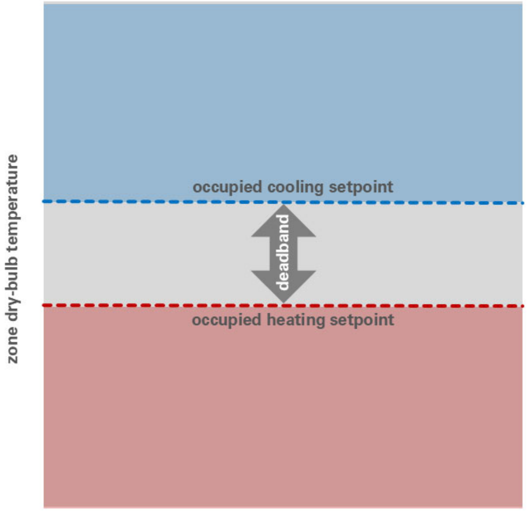
The dedicated outdoor-air system operates to provide the required amount of outdoor air to the zone for ventilation.

Zone is occupied, but requires no cooling or heating

Section 6.4.3.1.2 of ASHRAE Standard 90.1 requires a deadband of at least 5°F (3°C) between the occupied cooling and heating setpoints.

As the cooling load in the zone decreases, eventually the dry-bulb temperature in the zone drops below the cooling setpoint. If the temperature falls below the cooling setpoint, but remains above the heating setpoint, the WSHP compressor remains off. The temperature range between the cooling and heating setpoints is called the deadband (Figure 3).

Figure 3. Occupied zone temperature setpoints



The dedicated outdoor-air system continues to operate, providing the required amount of outdoor air to the zone for ventilation.

Zone is occupied and requires heating

When the temperature in the zone reaches the heating setpoint, the controller in the WSHP activates the reversing valve to switch operation of the refrigeration circuit to the heating mode, and cycles (or varies the speed of) one or more compressors to match the changing heating load in the zone. As the heating load increases, the compressor operates for a longer period of time between cycles.

Inside the WSHP, the refrigeration circuit extracts heat from the water loop and rejects heat to the recirculated air.

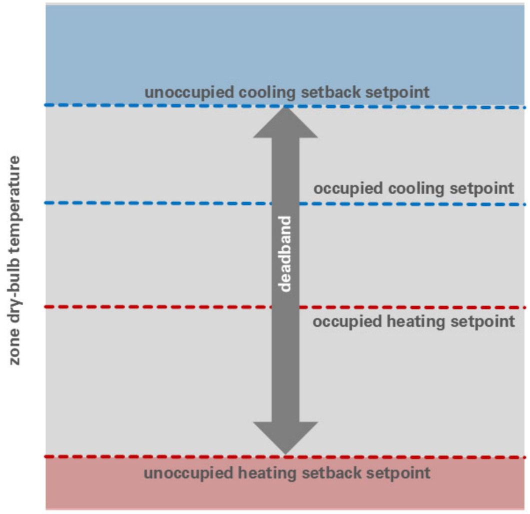
The dedicated outdoor-air system operates to provide the required amount of outdoor air to the zone for ventilation.

Zone is unoccupied

Section 6.4.3.3.2 of ASHRAE Standard 90.1 requires the unoccupied heating setback temperature to be at least 10°F (5.6°C) below the occupied heating setpoint, and the unoccupied cooling setback temperature to be at least 5°F (2.8°C) above the occupied cooling setpoint.

Zone setpoints are typically relaxed when the zone is scheduled to be unoccupied, allowing the temperature in the zone to either increase or decrease. These new setpoints are often called setback temperatures, and the result is a much wider deadband (Figure 4).

Figure 4. Unoccupied zone setback temperatures



During unoccupied periods, as long as the temperature in the zone is within this wider deadband, the controller in the WSHP shuts off the fan and compressor.

If all zones served by the WSHP system are unoccupied and the zone temperatures are within the deadband, the water-circulation pumps can be shut off. Because the building is unoccupied, no ventilation is required and the fan in the dedicated outdoor-air unit is also shut off.

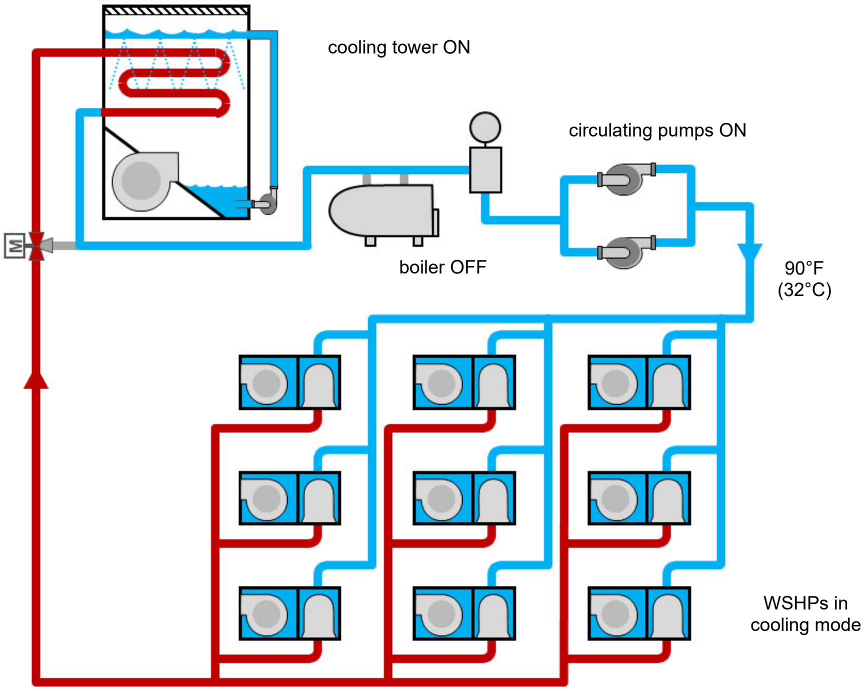
Some systems incorporate a **"timed override"** button on the zone temperature sensor that allows the occupant to temporarily switch the zone into the occupied mode, even though it is scheduled to be unoccupied. After a fixed period of time (three hours, for example), the zone automatically returns the zone to the unoccupied mode.

In addition, an occupancy sensor can be used to indicate that a zone is actually unoccupied, even though it is scheduled to be occupied. This signal can be used to switch the zone to an **"occupied standby" mode**, in which all or some of the lights can be shut off, the temperature setpoints can be raised or lowered slightly, and the outdoor air delivered to that zone can be reduced. When the occupancy sensor indicates that the zone is again occupied, the zone is switched back to the normal occupied mode.

Seasonal operation of the water loop

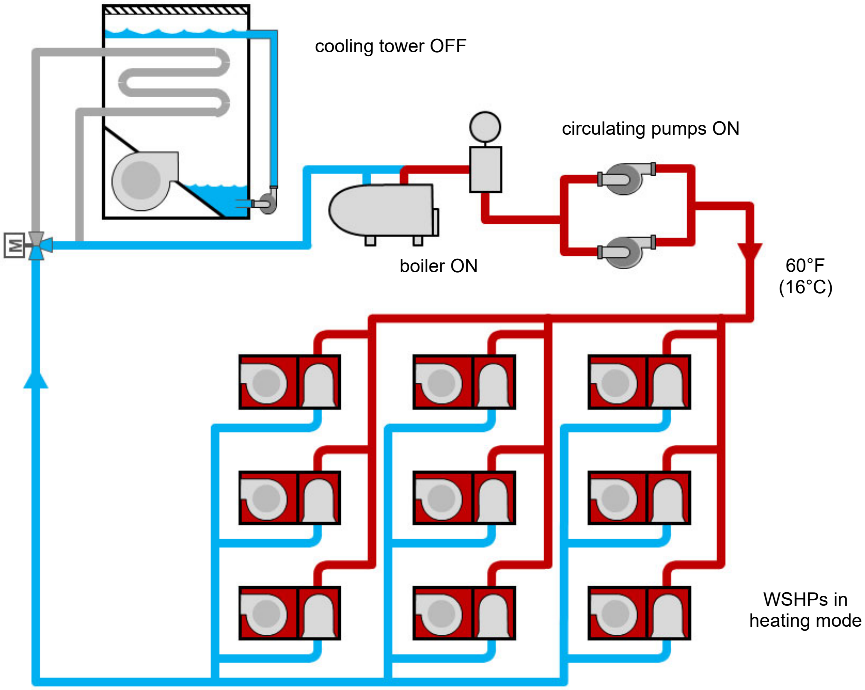
During **warm weather**, when most (or all) of the heat pumps are operating in cooling mode, heat removed from the air is transferred to the water loop. This causes the temperature of the water in the loop to increase, making it necessary to remove heat from the water (Figure 5). A “heat rejecter” (such as a closed-circuit cooling tower or fluid cooler) is used to reject heat from the loop, maintaining a leaving-water temperature of approximately 90ºF (32ºC).

Figure 5. System operation during warm weather (summer)



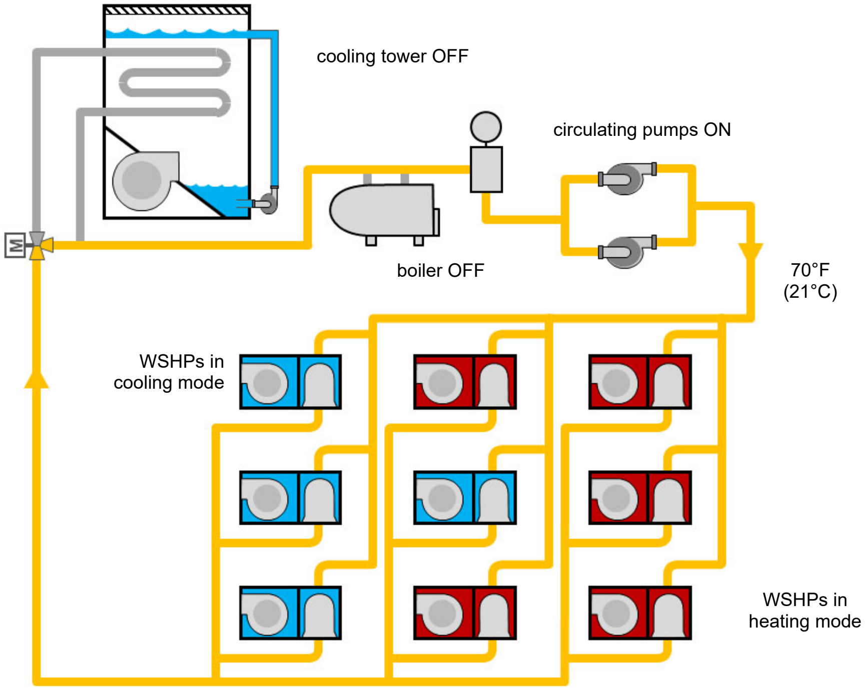
During **cold weather**, when most (or all) of the heat pumps are operating in heating mode, heat is extracted from the water loop and transferred to the air. This causes the temperature of the water in the loop to decrease, making it necessary to add heat to the water (Figure 6). A “heat adder” (such as a hot-water boiler) is used to add heat to the loop, maintaining a leaving-water temperature of approximately 60ºF (16ºC).

Figure 6. System operation during cold weather (winter)



During **mild weather**, such as spring or fall, the heat pumps serving the sunny perimeter and interior of the building may operate in cooling mode and reject heat to the water loop. At the same time, the heat pumps serving the shady perimeter of the building may operate in heating mode and extract heat from the water loop. Heat rejected by units operating in cooling mode raises the loop temperature while heat extracted by units operating in heating mode lowers the loop temperature. If the water temperature stays between about 60ºF (16ºC) and 90ºF (32ºC), for example, neither the boiler nor the cooling tower needs to operate (Figure 7). In this manner, a WSHP system provides a form of heat recovery.

Figure 7. System operation during mild weather (spring and fall)



In applications such as office buildings, heat generated by lights, people, and office equipment may require year-round cooling in the interior zones of the building. In these applications, the benefit of this heat recovery further reduces boiler operation during the winter months.

Benefits of WSHP Systems

The following section discusses some of the primary benefits of using a WSHP system.

Provides multiple zones of comfort control

Water-source heat pump systems are capable of controlling the temperature in many zones with dissimilar cooling and heating requirements. This is accomplished by providing a separate WSHP and temperature sensor for each independently-controlled zone.

When the sun is shining against the west side of the building in the late afternoon, the WSHPs serving the zones on that side of the building can operate in the cooling mode, while the WSHPs serving the zones along the east exposure can cycle their compressors to avoid overcooling or operate in the heating mode, if necessary.

Note: In the heating mode, a water-source heat pump is typically capable of supplying air up to about 100°F (38°C), depending on airflow and operating conditions. This is typically warmer air than can be supplied by an air-source heat pump when operating during cold weather.

Opportunity to save energy

Several characteristics of a WSHP system make it an energy-efficient system choice.

First, as explained previously, during some parts of the year, a WSHP system provides heat recovery because the heat rejected by WSHPs serving zones that require cooling can be used to provide heat for those zones that require heating (see “All heat recovery is not equal!,” p. 9). This saves energy by reducing the need to operate the cooling tower or boiler.

Second, using a heat pump for heating is typically more efficient than using electric resistance heat or a gas-fired burner. When operating in the heating mode, the coefficient of performance (COP) of a water-source heat pump might be somewhere between 3.0 and 6.0—depending on model and operating conditions. This is significantly higher than a COP of 1.0 for electric resistance heat.

Third, when operating in the cooling mode, the refrigeration cycle of a WSHP is water-cooled. Water-cooled condensing is typically more efficient than air-cooled condensing.

Finally, water-source heat pumps can easily be applied in a ground-coupled system. Ground-coupled systems use the earth as the heat rejecter and heat adder to reduce (or eliminate) the need to operate a cooling tower or boiler (see “Ground-Source Heat Pump Systems,” p. 135).

All heat recovery is not equal!

It is important to understand that while several types of systems can incorporate heat recovery, the energy-saving benefits are not the same for all systems. To illustrate this point, let's compare a WSHP system to a variable-refrigerant-flow (VRF) system.

In a typical WSHP system, all the heat pumps in the building are connected to a common water loop. Therefore, heat can be recovered from any zone operating in cooling mode and provided to any zone operating in heating mode, regardless of its location in the building.

In a typical VRF system, multiple indoor terminals are connected to an outdoor, air-cooled condensing unit via refrigerant piping. However, in most buildings, all VRF terminals are not connected to a single refrigeration circuit, so heat can only be transferred between zones that are served by the same outdoor unit.

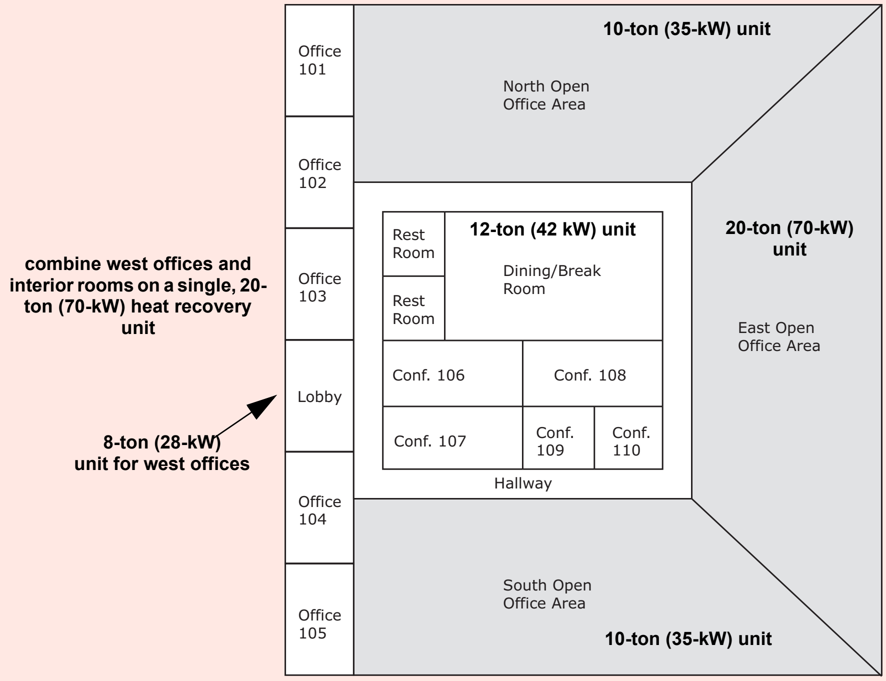
To demonstrate, consider the example office building shown in Figure 8. The "East Open Office Area" requires 20 tons (70 kW) of cooling, both the "North Open Office Area" and "South Open Office Area" require 10 tons (35 kW) each, the grouping of private offices along the west side of the building require a total of 8 tons (28 kW), and the grouping of rooms in the core of the building require a total of 12 tons (42 kW).

For this example, the maximum available VRF "system" (refrigerant circuit) size is 20 tons (70 kW). Since this building requires more cooling than this, multiple "systems" must be used. The "East Open Office Area" would be served by a dedicated system (refrigerant circuit). While heat can be transferred between the many indoor VRF terminals connected to that 20-ton (70-kW) outdoor unit, it cannot be transferred to zones that are not served by that same outdoor unit.

In this example, the most likely implementation of heat recovery would be to connect the indoor VRF terminals serving the group of offices on the west side of the building and those serving the rooms in the core of the building to a common, heat-recovery outdoor unit. When the VRF terminals serving the interior zones operate in the cooling mode, the heat rejected can be used for heating the offices along the west perimeter. While this would provide some energy savings, heat recovery is only applied to 43 percent of total floor area.

The difference between a WSHP system and a VRF system is that a WSHP system can recover heat from any zone in the building and use it in any other zone in the building. A typical VRF system, however, can only transfer heat between zones that are served by a given outdoor unit, not throughout the entire building.

Figure 8. Heat recovery in a typical VRF system



Helps achieve decarbonization goals

Reducing the carbon dioxide equivalent footprint of a building, often referred to as decarbonization, typically involves:

- Improving the energy efficiency of the overall system to reduce emissions from fuel combustion (either on site or at the power plant),
- Using refrigerants with a low Global Warming Potential (GWP) and minimizing leakage of these refrigerants, and
- Reducing the use of fossil fuels by installing electrified HVAC equipment (often referred to as **electrification**) served by an electrical grid that relies more heavily on carbon-free energy sources (e.g., solar, wind, and other renewables).

As explained previously, WSHP systems offer significant opportunities to improve energy efficiency. And in addition to the heat pumps using electricity for heating, the system can be designed to further reduce on-site fossil fuel use by either:

- Employing an electric boiler or electric resistance heat (see “Electric resistance heat for a “boiler-less” system,” p. 56),
- Adding a ground heat exchanger to eliminate the need for a boiler (see “Ground-Source Heat Pump Systems,” p. 135), or
- Using an air-to-water heat pump in place of both the cooling tower and boiler (see “Electrified WSHP System,” p. 155).

Limits impact of equipment failure

Because each zone is served by a separate WSHP that can provide either cooling or heating, if one WSHP fails and needs to be replaced, it does not impact the remaining zones within the building. Some building operators even keep a few spare units on site, so if a unit fails, it can quickly be replaced.

For this same reason, the system can typically adapt easily if the use of a zone changes (due to a change in space layout or a new tenant, for example). As long as the water-distribution loop and dedicated outdoor-air system have sufficient capacity, the WSHP serving the affected zone can be replaced with a larger (or smaller) model without needing to modify the rest of the system.

Ease of installation

The fact that the same piece of equipment is used to provide cooling and heating to the zone, and that all the components of the WSHP are pre-assembled in a factory, makes installation fairly simple. And, even though a separate cooling tower and boiler may be included in the system, only one set of water pipes is required.

In addition, using pre-engineered, factory-installed, and factory-commissioned controls simplifies installation and commissioning, and can result in faster system start-up.

Scalable capacity … add it as needed

For speculative buildings, such as commercial office buildings, retail strip malls, or high-rise condominiums, individual water-source heat pumps can be installed and connected to the water distribution loop as areas of the building are fit-out for lease to a tenant. Of course, the water-distribution loop (including cooling tower and boiler) and dedicated OA system must be installed early in the project and must have adequate capacity to serve the entire system. This not only improves cash flow for the developer, but also provides flexibility to accommodate the specific needs of future tenants.

Opportunity for individual tenant metering

Because each zone is served by a separate WSHP, this system offers the opportunity to meter individual units and bill the tenant for the operating costs for their space only. This often makes a WSHP system an attractive option for condominiums, apartment buildings, retail strip malls, and leased office space. If a dedicated outdoor-air system is used, however, individual metering of energy used to condition outdoor air for ventilation is more difficult.

Drawbacks/Challenges of WSHP Systems

The following section discusses some of the primary drawbacks (or challenges) of using a WSHP system, along with some potential ways to address those challenges.

Equipment is located in or near the occupied spaces

Typically, WSHPs are located in, or very close to, the occupied space. This requires the use of floor space or ceiling space throughout the building. And because this equipment contains both a compressor and a fan, achieving acceptable noise levels in the space can sometimes be challenging and needs to be considered during system design (see “Acoustics,” p. 125).

Distributed maintenance

Because the WSHPs are distributed throughout the building, maintenance must be performed within the occupied spaces, or in the ceiling or closet near the occupied spaces. This can be disruptive to occupants or may lead to neglecting proper equipment maintenance. Proper maintenance of WSHPs requires that they be located in accessible areas. In a new building, this requires close coordination with the architect.

Dedicated outdoor-air system is typically required

Because this system uses distributed equipment to provide cooling and heating, the introduction of outdoor air for ventilation may bring a few challenges, including insufficient dehumidification and/or heating capacity of the WSHPs. While the requirement for ventilation can be handled in various ways, most WSHP systems use a dedicated outdoor-air system to separately condition all the outdoor air required for ventilation and deliver it to the individual zones.

Common Building Types That Use WSHP Systems

Water-source heat pump systems are used in many building types, but the most common applications are:

- Commercial, medical and government office buildings
- Schools (both K-12 and higher education)
- Hotels and motels
- Apartment buildings and high-rise condominiums
- Dormitories and military barracks
- Extended-care facilities
- Retail stores (standalone, strip malls, and large malls)

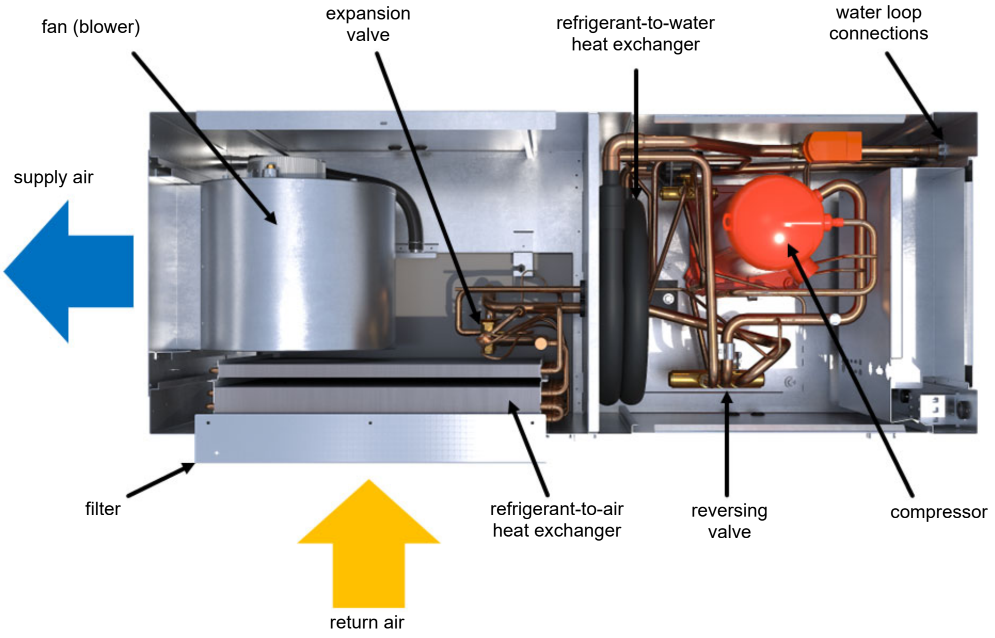
Primary System Components

This chapter discusses the primary components of a typical water-source heat pump system in greater detail. For details on specific pieces of equipment, consult the manufacturer.

Water-Source Heat Pumps

Typically, each zone of the building is served by a separate water-source heat pump (WSHP). A WSHP is a packaged, heating-and-cooling unit with a reversible refrigeration cycle. Return air from the zone is drawn into the WSHP through the intake (Figure 9). This air passes through a filter and refrigerant-to-air heat exchanger, before the fan discharges it directly into the zone or through supply ductwork and supply-air diffusers.

Figure 9. Typical horizontal-style water-source heat pump



Reversible, direct-expansion (DX) refrigeration circuit

The direct-expansion (DX), vapor-compression refrigeration circuit of a WSHP is comprised of one or more compressors, a refrigerant-to-air heat exchanger, a refrigerant-to-water heat exchanger, an expansion device, and a reversing valve (Figure 9). This refrigeration circuit is pre-engineered and assembled in a factory, so no field-installed refrigerant piping is required.

Note: A variation of the water-source heat pump, called a water-to-water heat pump (see Figure 20, p. 28), contains two refrigerant-to-water heat exchangers and no refrigerant-to-air heat exchanger.

Components of the DX refrigeration circuit

Being a refrigeration system, a WSHP must comply with applicable safety codes related to refrigerants, such as **ASHRAE Standard 15**. For information on this standard, and how its requirements apply to WSHP equipment, refer to the Trane application manual, *Refrigeration Systems and Machinery Rooms Application Considerations for Compliance with ASHRAE® Standard 15-2022* (APP-APM001*-EN).

Each WSHP includes at least one compressor; some models may use multiple compressors. Depending on its size and manufacturer, this compressor may be rotary (rolling piston), reciprocating, or scroll type (Figure 10). The selection of the type of compressor generally depends on the capacity and electrical requirements of the WSHP.

Figure 10. Types of compressors typically used in a WSHP

rotary (rolling piston) reciprocating scroll

Methods of compressor capacity modulation

Traditionally, smaller-capacity heat pumps have contained a single compressor that cycles on and off. Larger-capacity heat pumps often have multiple compressors, allowing for multiple stages of capacity control. Recently, some heat pumps have been equipped with either a two-stage, variable-capacity (Digital™) or variable-speed compressor.

Compared to the on/off compressor historically used in this type of equipment, these other methods of compressor capacity modulation are better able to match cooling or heating capacity with the changing load in the zone. This typically improves comfort and, in some cases, also results in reduced energy use during part-load conditions.

Figure 11 compares the part-load performance of these different methods of compressor capacity modulation in an example 4-ton (14-kW) water-source heat pump.

- At part-load conditions, a **single on/off compressor** will operate for a period of time, cycle off for a period of time, and then cycle back on again. **Hot gas bypass** (HGBP) could be used to divert some of the hot refrigerant vapor to the low-pressure side of the system, allowing the compressor to continue operating at part-load conditions. This is inefficient, however, because the compressor pumps refrigerant that never is used for the cooling.
- A **two-stage scroll compressor** can operate at either 100 percent capacity or some reduced capacity (67 percent, for example). This ability to operate at partial capacity results in improved efficiency (EER) at part-load conditions.
- At part-load conditions, a **Digital™ scroll compressor** continues to operate at a constant speed, but the two scrolls are periodically separated to release the compressor refrigerant vapor and reduce the capacity of the compressor to better match the changing load. While this approach is able to more closely match capacity to the load, it does not improve efficiency at part-load conditions.
- When **two compressors are manifolded together** on the same circuit, either both compressors operate together, one compressor operates while the other is off, or both compressors are off. Because the individual compressors are selected at optimal points, their motors are often operating at a high efficiency and part-load performance is pretty good.
- Finally, a **variable-speed compressor** changes the rotational speed of the compressor to vary its capacity as the load changes. This approach results in the greatest efficiency improvement at part-load conditions.

Compressor technologies used in heat pumps will continue to evolve. For the most recent information, contact the equipment manufacturer.

The decision on which type of compressor capacity modulation is best suited for a given project depends on available budget, range of heat pump capacities needed, maintenance capabilities, and energy use targets.

Figure 11. Part-load compressor performance for an example 4-ton (14-kW) water-source heat pump

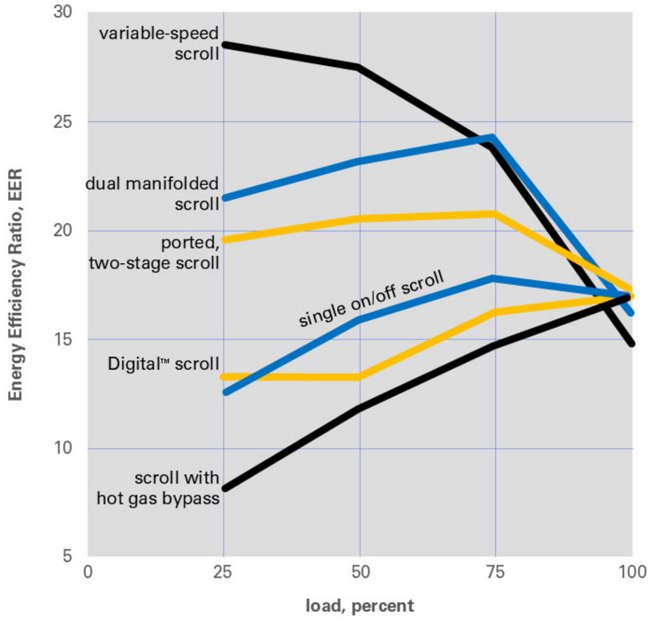
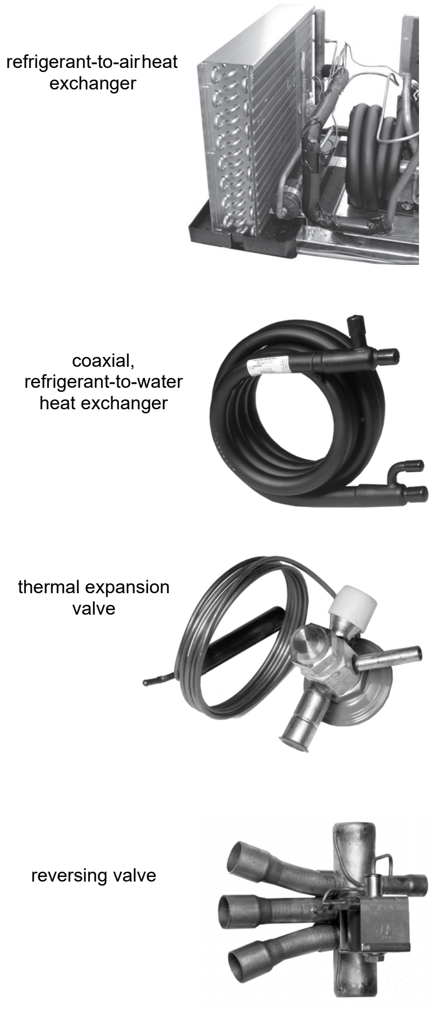


Figure 12. Components of the DX refrigeration circuit



The **refrigerant-to-air heat exchanger** is a finned-tube coil similar to a direct-expansion (DX) refrigerant coil found in a packaged rooftop unit (Figure 12). In the cooling mode, this refrigerant-to-air heat exchanger acts as the evaporator, and the air is cooled as heat is transferred from the air to the refrigerant inside the tubes. In the heating mode, it acts as the condenser, and heat is transferred from the refrigerant to the air.

The **refrigerant-to-water heat exchanger** may be a tube-in-tube, tube-in-shell, or brazed-plate design. The example shown in Figure 12 is a tube-in-tube, or coaxial, heat exchanger. It is constructed as a small tube running inside a larger tube. The water flows through the inner tube and refrigerant flows through the outer tube. In the cooling mode, this refrigerant-to-water heat exchanger acts as the condenser, and water flowing through the inner tube extracts heat from the refrigerant flowing through the outer tube. In the heating mode, it acts as the evaporator, and the refrigerant extracts heat from the water.

- When used in a conventional boiler/tower WSHP system, the refrigerant-to-water heat exchanger is typically made of copper.
- When used in a ground-source or seawater (marine) application, this heat exchanger is often made of cupronickel (CuNi), which has a higher tolerance for chlorides. However, the ASHRAE *Design of Ground-Source Heat Pump Systems* manual cautions that cupronickel does not prevent all water quality problems: “The alloy (CuNi) is ineffective (or only marginally better than copper) at addressing many problems commonly encountered in ground-water applications, such as hydrogen sulfide, low-pH corrosion, or iron, manganese, and carbonate scale or fouling.”

The most common type of **expansion device** used in water-source heat pumps is the thermal expansion valve (TXV). Some models, however, use an electronic expansion valve (EEV), short orifice, or capillary tube. All of these devices reduce the pressure and temperature of the refrigerant within the cycle. Expansion valves, such as the TXV shown in Figure 12, have the added capability of metering the quantity of refrigerant flowing through the cycle to match the load. This enhances the efficiency of the refrigeration cycle.

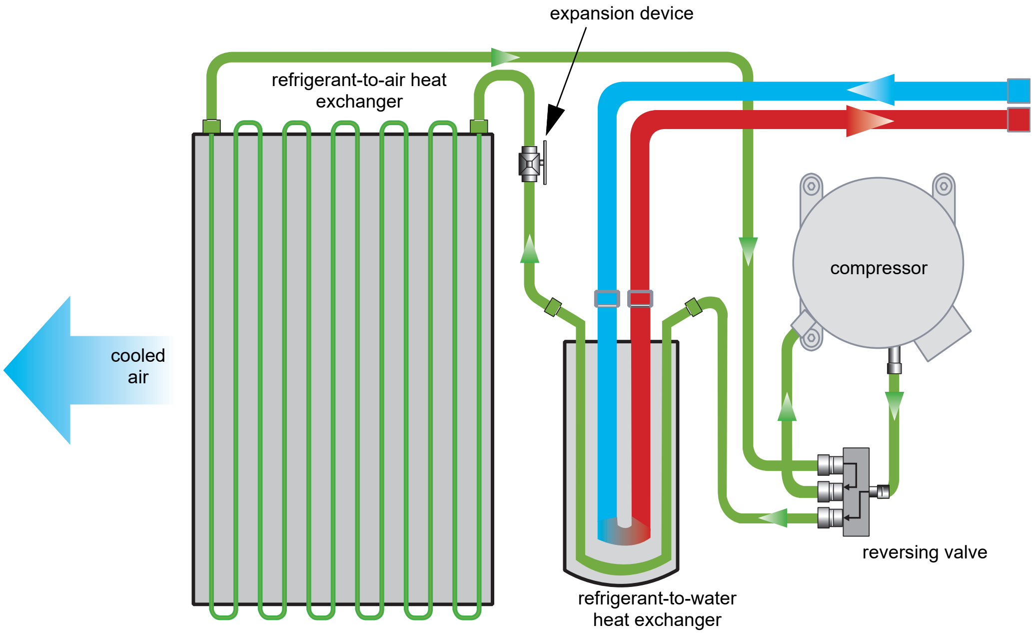
TXVs used in heat pumps may be bi-directional, meaning that the refrigerant flows in one direction during the cooling mode and in the opposite direction during the heating mode. The alternative would be to design the refrigerant piping inside the heat pump to ensure that refrigerant flow through the valve is in the same direction in either mode.

The **reversing valve**, sometimes called a **four-way valve**, reverses the direction of the refrigerant flow through the cycle, as described in the next section. This allows the WSHP to change operation of the refrigeration circuit to provide either cooling or heating (Figure 12).

WSHP operation during cooling mode (Figure 13)

Hot, high-pressure refrigerant vapor is pumped from the compressor to the refrigerant-to-water heat exchanger that, in the cooling mode, functions as the condenser. Inside this heat exchanger, heat is transferred from the hot refrigerant vapor to the lower-temperature water, warming the water and cooling the refrigerant, causing it to condense into a liquid.

Figure 13. Operation of a WSHP in the cooling mode



This liquid refrigerant then flows through an expansion device that reduces the pressure of the refrigerant. At this lower pressure, a small portion of the refrigerant boils (or flashes), cooling the remaining liquid refrigerant to the desired evaporator temperature. The resulting mixture of cool liquid and vapor travels to the refrigerant-to-air heat exchanger that, in the cooling mode, functions as the evaporator. Inside this heat exchanger, the refrigerant extracts heat from the relatively warm air passing over the outer surfaces of the tubes, cooling the air and causing the liquid refrigerant to boil (or evaporate). The resulting refrigerant vapor is then pumped back to the compressor, which increases its pressure and temperature to repeat the cycle.

Table 1 shows the example performance of a water-source heat pump. In the cooling mode, this heat pump provides 54 MBh (16 kW) of cooling capacity while supplying 1700 cfm (0.80 m^3/s) of cooled air. The heat removed from the air stream, plus the heat of compression, is rejected to the 15 gpm (0.95 L/s) of water flowing through the refrigerant-to-water heat exchanger. This **heat rejected to the loop**—69 MBh (20 kW), in this example—raises the water temperature from 90°F (32°C) entering the heat exchanger, to 99°F (37°C) leaving.

heat rejected to loop = heat removed from air + heat of compression

Table 1. Example WSHP performance*

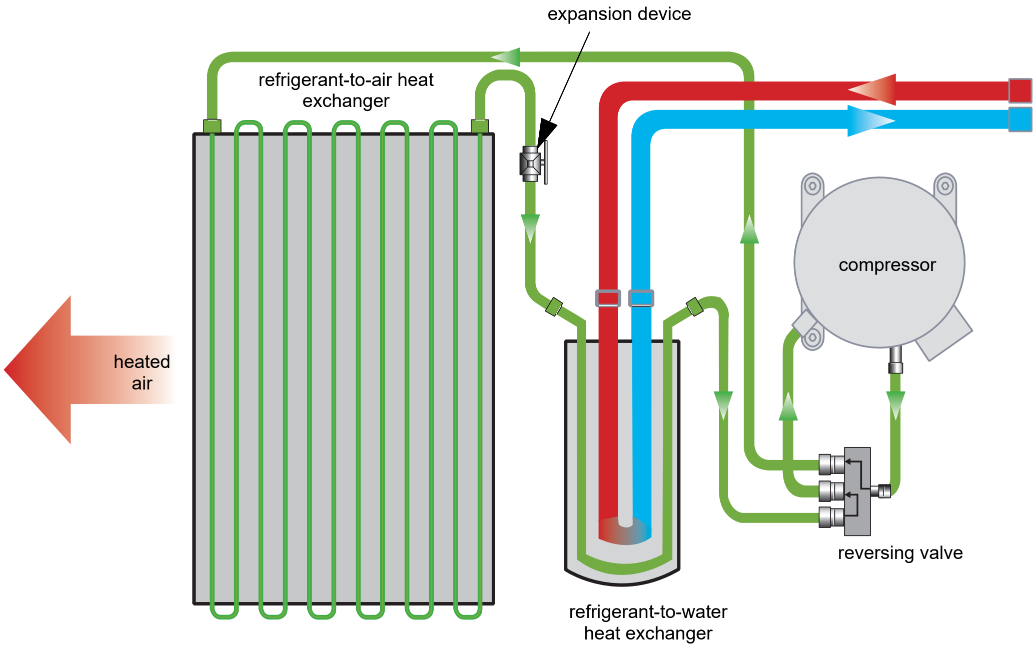
	Cooling mode	Heating mode
Airflow	1700 cfm (0.80 m^3/s)	1700 cfm (0.80 m^3/s)
Water flow rate	15 gpm (0.95 L/s)	15 gpm (0.95 L/s)
Entering water temperature	90°F (32°C)	60°F (16°C)
Leaving water temperature	99°F (37°C)	53°F (11°C)
Capacity	54 MBh (16 kW)	71 MBh (21 kW)
Heat rejected to the loop	69 MBh (20 kW)	
Heat extracted from the loop		54 MBh (16 kW)

* *Assumes entering-air conditions of 77°F (25°C) dry bulb and 63°F (17°C) wet bulb during cooling mode and 68°F (20°C) during heating mode, and 0.5 in.* H_2O *(125 Pa) of external static pressure.*

WSHP operation during heating mode (Figure 14)

Hot, high-pressure refrigerant vapor is pumped from the compressor and diverted by the reversing valve to the refrigerant-to-air heat exchanger. In the heating mode, this heat exchanger functions as the condenser, and heat is transferred from the refrigerant vapor to the lower-temperature air passing over the outer surfaces of the tubes. The air is heated and the refrigerant condenses into a liquid.

Figure 14. Operation of a WSHP in the heating mode



This liquid refrigerant then flows through the expansion device and travels to the refrigerant-to-water heat exchanger that, in the heating mode, now functions as the evaporator. Inside this heat exchanger, the refrigerant extracts heat from the relatively warm water, cooling the water and causing the liquid refrigerant to boil.

The refrigerant vapor travels back through the reversing valve to the compressor to repeat the cycle.

Returning to the same example (Table 1, p. 18), in the heating mode this heat pump provides 71 MBh (21 kW) of heating capacity while supplying 1700 cfm (0.80 m^3/s) of warm air. The heat added to the air stream is comprised of heat extracted from the 15 gpm (0.95 L/s) of water flowing through the refrigerant-to-water heat exchanger plus the heat of compression. This **heat extracted from the loop**—54 MBh (16 kW), in this example—lowers the water temperature from 60°F (16°C) entering the heat exchanger, to 53°F (11°C) leaving.

heat extracted from loop = heat added to air – heat of compression

Water-regulating valves

ASHRAE Standard 90.1-2022 requires that each WSHP be equipped with a motorized, two-position, isolation valve. This isolation valve, when combined with an automatic flow-control device, typically makes a water-regulating valve unnecessary. An isolation valve is less expensive, opens quickly when the compressor turns on, closes slowly when the compressor turns off to avoid water hammer, and is a positive shutoff valve that ensures the valve is closed to prevent water flow. (See "Isolation valves and flow-control devices," p. 38).

While a rare few applications may require the use of a water-regulating valve, the need for them has all but disappeared due to the common use of TXVs and less-expensive isolation valves.

Historically, water-regulating valves were used to modulate water flow through the refrigerant-to-water heat exchanger to maintain the condensing (head) pressure high enough for the refrigeration system to operate properly as water temperature in the loop decreased. This was necessary because older water-source heat pumps typically used a capillary tube or fixed orifice as the refrigerant expansion device. To operate properly, these types of expansion devices required that the entering water temperature be maintained between 60°F (16°C) and 90°F (32°C), in both the cooling and heating modes.

Today, most water-source heat pumps use a thermal expansion valve (TXV, see Figure 12, p. 16), rather than a capillary tube or fixed orifice. The operating range for a typical TXV is between 45°F (7°C) and 120°F (49°C) in the cooling mode, and between 25°F (-4°C) and 86°F (30°C) in the heating mode. For this reason, very few applications require the use of a water-regulating valve. In fact, the use of water-regulating valves is discouraged because it "competes" with the TXV to ensure proper compressor operation.

If the loop temperature is expected to drop below 45°F (7°C) when some heat pumps need to operate in the cooling mode, a waterside economizer can be used to completely eliminate the need to operate the compressor(s) at such conditions (see "Economizer control," p. 170). This also avoids the need to install a water-regulating valve.

Water flow rate

The water flow rate through the refrigerant-to-water heat exchanger impacts the capacity and energy use of the heat pump compressor, cooling tower, and water-circulating pumps. Therefore, it is important to assess the impact of water flow rate on overall system performance.

The water flow rate through a WSHP is typically between 2.2 and 3.4 gpm/ton (0.039 to 0.061 L/s/kW)—with 3.0 gpm/ton (0.054 L/s/kW) being a good rule-of-thumb that often achieves a good balance between compressor efficiency and pump energy use.

Increasing the water flow rate improves the efficiency of the heat pump compressor, but it increases pump energy use. Decreasing the water flow rate reduces pump energy use, but it decreases both the capacity and efficiency of the WSHP. In climates that experience more hours of high outdoor wet-bulb temperatures, higher flow rates may be beneficial because of the warmer water temperatures leaving the cooling tower. In climates that experience fewer hours of high outdoor wet-bulb temperatures, lower flow rates may be beneficial because of the cooler water temperatures leaving the cooling tower.

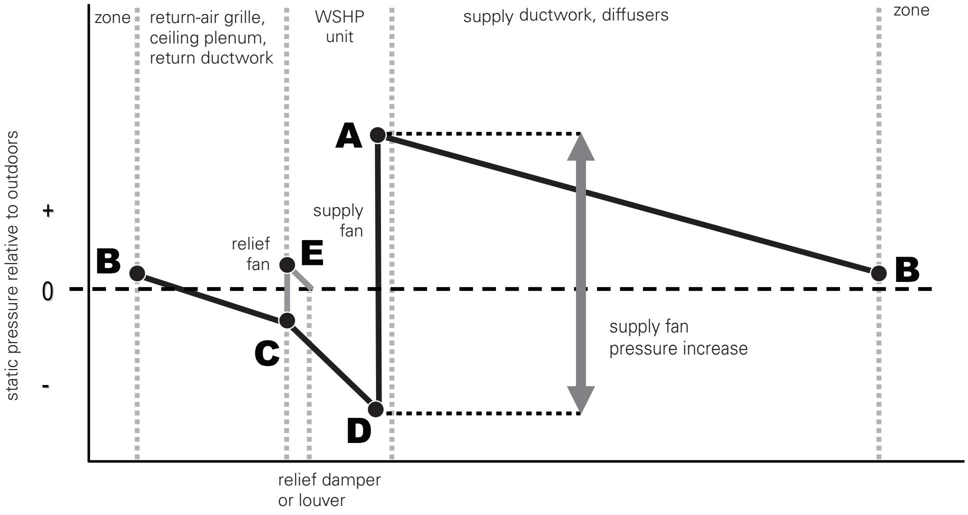
Do not use less than the manufacturer's minimum recommended flow rate as this may increase the risk of the compressor tripping off on a safety limit: high-pressure cutout during the cooling mode or low-temperature cutout during the heating mode.

Fan

Each water-source heat pump contains a fan to draw return air from the zone and through the components of the heat pump, before discharging it into the zone.

This fan must create a high enough pressure at its outlet (A) to overcome the pressure losses associated with pushing the air through the supply ductwork and supply-air diffusers (A to B)—or through the supply-air grille, in the case of a console-type or vertical-stack WSHP (Figure 15). In addition, the fan must create a low enough pressure at its inlet (D) to overcome the pressure losses associated with drawing the air through the return-air grille and possibly the open ceiling plenum and/or return ductwork (B to C), and then through the filter and refrigerant-to-air heat exchanger inside the WSHP (C to D).

Figure 15. WSHP with supply ductwork



Due to the pressure drop through the return-air path, the static pressure in this path (C) might be lower than the ambient pressure. A central relief fan can be used to raise the pressure of the air to be exhausted (from C to E) so that it is high enough to overcome the pressure loss associated with the relief damper, and force the excess air out of the building. Adding the relief fan allows the system to exhaust air that is to be replaced by fresh, outdoor air, and does so without increasing the pressure inside the building (see "Building pressure control," p. 192).

Electronically commutated motor

Figure 16. Electronically commutated motor (ECM) on fan

Some water-source heat pumps use an electronically commutated motor (ECM) for the fan. An ECM is a brushless DC motor that combines a permanent-magnet rotor, wound-field stator, and an electronic commutation assembly to eliminate the brushes (Figure 16).

Some benefits of using an ECM on the heat pump fan include:

- *Energy savings*

  ECMs are typically more efficient than the single-speed, fractional-horsepower motor technologies that have traditionally been used in smaller water-source heat pumps. This efficiency difference often allows ECMs to offer substantial energy savings compared to conventional motor technologies.

  When the heat pump includes a two-stage, variable-capacity, or variable-speed compressor, or if it includes more than one compressor, the ECM can be used to change fan speeds at part-load conditions. This provides another opportunity for fan energy savings.

- *Gradually changing sound levels*

  The “soft-start” nature of the ECM allows the fan to ramp up slowly when activated and ramp down slowly when turned off. This minimizes the distraction of the fan cycling on and off, especially when the heat pumps are located in spaces where people are sleeping—such as hotels, motels, apartments, condominiums, dormitories, barracks, and extended care facilities.

Potential drawbacks include:

- *Higher installed cost*

  An ECM requires a power transistor to drive the stator windings at a specified motor current and voltage level. This addition, coupled with electronic commutation controls, currently make them more expensive to purchase than their AC counterparts.

- *Potential for disruptive harmonic currents*

  Harmonic currents are created when AC power is converted to DC power. In some cases, these currents can overheat conductors and connectors, interfere with the operation of sensitive equipment, and in severe cases, burn out transformers and motors.

  Determining whether harmonic currents will cause a problem in a particular building requires review of the electrical system before it is installed so that appropriate steps can be taken. When necessary, it is possible to alter the design of the system (by oversizing the neutral wire, for example) and/or reduce motor-generated harmonics (by adding a harmonic filter, for example).

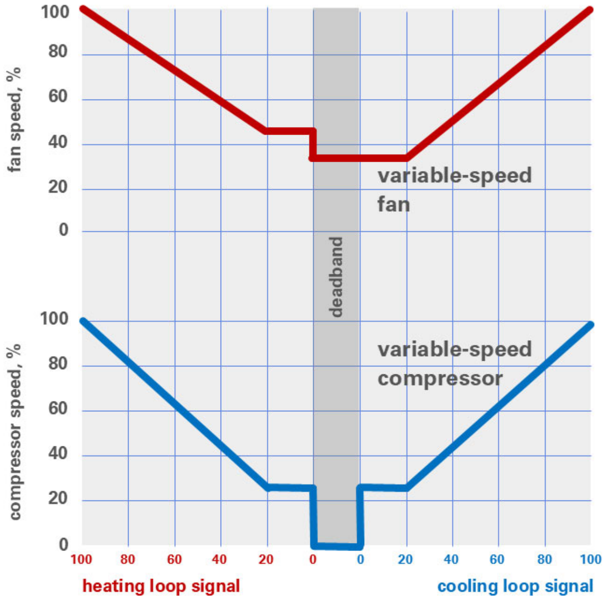
Multiple-speed fan operation

When the heat pump includes a two-stage, variable-capacity, or variable-speed compressor, or if it includes more than one compressor, the opportunity exists to use the ECM to change fan speeds at part-load conditions.

For example, when a two-stage compressor (or a heat pump with two compressors) operates at 100 percent capacity, the controller operates the fan at high speed, but when the compressor operates at reduced capacity—67 percent, for example—the controller operates the fan at a reduced speed.

However, in a heat pump equipped with a variable-speed compressor, as the load decreases, the controller reduces fan speed at the same time that compressor capacity (rotational speed) is reduced (Figure 17).

Figure 17. Variable-speed fan with a variable-speed compressor



Note: Assumes continuous (not cycled) fan operation.

In addition to reducing fan energy use, operating the fan at a reduced speed can provide acoustic and dehumidification benefits at part-load conditions. During the cooling mode, reducing airflow when the compressor unloads results in a colder discharge-air temperature at part load. It also allows the compressor to operate for longer continuous periods of time, without needing to cycle off quickly to avoid over-cooling the zone. The colder, drier air and lengthened compressor run-time typically improves part-load dehumidification performance (see “Dehumidification impact of compressor cycling and constant-speed fan,” p. 97).

During the heating mode, reducing airflow when the compressor unloads results in a warmer discharge-air temperature at part load. The warmer air can sometimes improve occupant perception of comfort.

When a multiple-speed or variable-speed fan is used, take care to ensure that the zone receives the proper quantity of outdoor air required for ventilation requirements, regardless of fan speed (see “Dedicated OA system configurations,” p. 59).

Filters

For more information on various types of particulate filters, or other air cleaning technologies, refer to the Trane application manual, *Air Cleaning Systems* (APP-APM002*-EN).

Another requirement of the HVAC system is to ensure that the air delivered to the conditioned space is relatively clean. This improves system performance (by keeping the refrigerant-to-air heat exchanger cleaner, for example) and keeps the air distribution system relatively clean.

Airborne particulates vary in size, ranging from submicron to 100 microns (μm) and larger. Many types of particulate media filters are available. Some are designed to remove only large particles, while others—high-efficiency particulate air (HEPA) filters, for example—also remove particles with diameters less than one micron.

The "minimum efficiency reporting value" (MERV), defined by ASHRAE Standard 52.2, relates to how efficiently a filter removes particles of various sizes, from 0.3 to 1.0 micron. Table 2 identifies common types of particulate filters and their typical applications.

Key factors to consider when selecting particulate filter types for a specific application include:

- *Target particle size and degree of cleanliness required (collection efficiency)*

  The "collection efficiency" of a particulate filter is a function of how well it removes particles of various sizes. Filters with higher efficiencies remove a higher percentage of particles, and smaller particles, than filters with lower efficiencies. Since particulate contaminants vary in size, it is important to define the contaminants of concern for a given facility when selecting the type of filter to be used (see Table 2).

- *Allowable airside pressure drop*

  A direct correlation usually exists between collection efficiency and airside pressure drop. Generally, a filter with a higher efficiency will cause a higher pressure drop in the passing air stream, increasing fan energy use. The number of pleats in a media filter determines the surface area of the media. In general, the more surface area, the lower the airside pressure drop. Pressure drop is also related to air velocity: higher air velocity through a media filter results in a higher static pressure drop.

- *Dirt-holding capacity*

  Dirt-holding capacity is an indication of how much dirt the filter will hold at the "dirty" (or final) pressure drop. This indicates how often the filter will need to be replaced. In general, a filter with more media surface area will hold more dirt and will need to be replaced less frequently. (This varies with the brand of filter.)

- *Available space*

  In general, filters with a higher collection efficiency, lower airside pressure drop, and/or greater dirt-holding capacity, require more space than filters that perform more poorly in these categories.

- *Available budget*

  Filters with more media surface area generally cost more than filters with less surface area. This impacts both the installed cost and maintenance (replacement) cost of the filter system.

Table 2. Application guidelines for various filter types

MERV[1]	Examples of contaminants controlled	Example applications	Example air filter/cleaner types
MERV 16 MERV 15 MERV 14 MERV 13	**0.3 to 1.0 µm size range** • Bacteria • Smoke (ETS) • Paint pigments • Face powder • Some virus • Droplet nuclei • Insecticide dusts • Soldering fumes	• Hospital general ventilation • Day surgery, general surgery • LEED commercial buildings • Smoking lounges • Turbo equipment, compressors • Welding, soldering • Prefiltration for HEPA filters	Box-style wet-laid or lofted fiberglass, box-style synthetic media, mini-pleated synthetic or fiberglass paper, depths from 2 in. to 12 in. Pocket filters of fiberglass or synthetic media, depths from 12 in. to 36 in.
MERV 12 MERV 11 MERV 10 MERV 9	**1.0 to 3.0 µm size range** • Milled flour • Lead dust • Combustion soot • Legionella • Coal dust • Some bacteria • Process grinding dust	• Commercial buildings • Schools • Gymnasiums • Better residential buildings • Industrial air cleaning • Food processing facilities • Air separation plants • Prefiltration for higher-efficiency filters	Box-style wet-laid or lofted fiberglass, box-style synthetic media, mini-pleated synthetic or fiberglass paper, depths from 2 in. to 12 in. Pocket filters either rigid or flexible in synthetic or fiberglass, depths from 12 in. to 36 in.
MERV 8 MERV 7 MERV 6 MERV 5	**3.0 to 10.0 µm size range** • Pollens • Earth-origin dust • Mold spores • Cement dust • Powdered milk • Snuff • Hair spray mist	• General HVAC filtration • Commercial property • Schools • Paint booth intakes • Industrial equipment filtration • Electrical, phone equipment protection • Prefiltration for higher-efficiency filters	Wide range of pleated media, ring panels, cubes, pockets in synthetic or fiberglass, disposable panels, depths from 1 in. to 24 in.
MERV 4 MERV 3 MERV 2 MERV 1	**> 10.0 µm size range**	• Protection from blowing large particle dirt and debris • Industrial environment ventilation air	Inertial separators

[1] *Minimum Efficiency Reporting Value (MERV) is defined by ANSI/ASHRAE Standard 52.2-2017, Method of Testing General Ventilation Air-Cleaning Devices for Removal Efficiency by Particle Size.*

Source: 2024 *ASHRAE® Handbook—HVAC Systems and Equipment*, Chapter 29, Table 3. ASHRAE, Inc. www.ashrae.org.

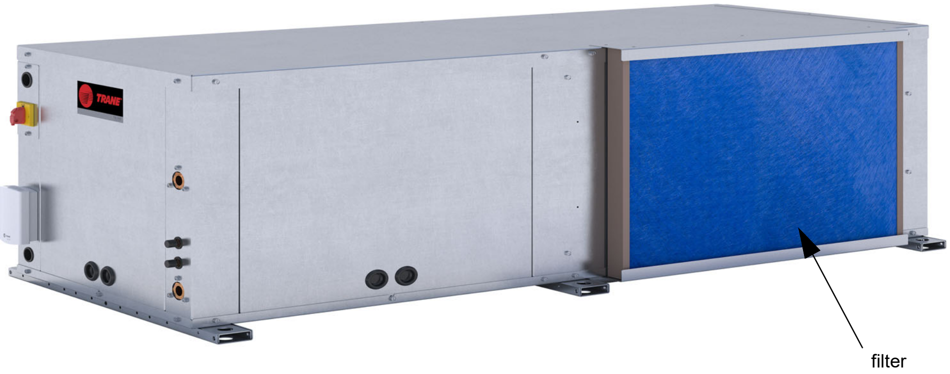
It is important to maintain and replace filters as recommended by the manufacturer. The replacement filters should have similar performance characteristics as the filters originally specified by the design engineer. Critical characteristics include collection efficiency (MERV rating), airside pressure drop at the desired operating airflow, and physical size.

In addition, air bypass can reduce the effectiveness of the filtration system. During replacement, the filter assembly should be carefully inspected to identify any areas that can allow air to bypass around the filters. These areas should be sealed (with gasketing, for example) to minimize airflow through the space between adjacent filters.

Filter located in each water-source heat pump

In most WSHP configurations, recirculated return air passes through a particulate filter to remove airborne particulate contaminants. Locating these filters upstream of the refrigerant-to-air heat exchanger (Figure 18) helps keep it cleaner for a longer period of time, and allows the system to operate more efficiently.

Figure 18. Particulate filter installed in a horizontal WSHP



ASHRAE Standard 62.1 (Section 5.5) requires that a filter with a MERV rating of not less than 8 be installed upstream of all wetted surfaces, including cooling coils. In general, this requirement can be met with standard "throwaway" or "pleated" filters (see Table 2, p. 24).

Earning LEED credits

One of the available strategies for earning the "Enhanced Indoor Air Quality Strategies" credit of LEED v4.1 is to install a MERV 13 (or higher) filter to clean the **outdoor air** prior to its introduction to the occupied spaces. Another one of the available strategies is to install a MERV 13 (or higher) filter to clean any **recirculation air** also.

To maintain the desired level of cleanliness and minimize system energy use, never operate a WSHP without the filter in place, especially during construction or renovation. Filters used during construction should be replaced prior to building occupancy.

Filter located in the dedicated outdoor-air unit

When the WSHP system uses a dedicated outdoor-air system to condition all the outdoor air (OA) required for ventilation, the dedicated OA unit typically includes a particulate filter to remove airborne particulate contaminants entering the building.

If the dedicated OA unit contains a cooling coil, Section 5.5 of ASHRAE 62.1 requires a filter with a MERV rating of mot less than 8 be installed upstream of the coil. In some applications, high-efficiency filtration of the outdoor air may be desired or required (see Table 2, p. 24).

For example, if the building is located in an area of the country that exceeds the U.S. EPA limit for airborne particles with a diameter of 10 microns or less (PM10), ASHRAE 62.1 (Section 6.1.4.1) requires a MERV 8 filter be used to clean the outdoor air before it is introduced to any occupied space. And if the building is located in an area that exceeds the EPA limit for airborne particles with a diameter of 2.5 microns or less (PM2.5), the standard (Section 6.1.4.2) requires a MERV 11 filter be used to clean the outdoor air.

Some high-efficiency filtration systems incorporate a lower-efficiency pre-filter upstream to capture larger particles, and thus extend the useful life of the higher-efficiency filter downstream. The benefit of this longer life, however, must be carefully weighed against the additional cost and pressure drop of the upstream pre-filters, as well as the labor required to periodically replace them.

Condensate management

For more information on water management in buildings, including proper condensate trap design, refer to the Trane application manual, *Managing Building Moisture* (SYS-AM-15).

Preventing moisture problems in buildings is a shared responsibility among all parties involved in the design, construction, maintenance, and use of the building. To prevent water-related problems within the WSHP itself, follow these basic practices:

- *Provide access suitable for regular inspection and cleaning.*

  Ensure that the WSHP includes removable panels to allow regular inspection and cleaning. Poor location of the WSHP or limited service clearance can also discourage inspection and cleaning.

- *Use sloped drain pans and clean them regularly.*

  A flat drain pan retains water, and stagnant water can provide a habitat for microbial growth. A sloped pan improves drainage considerably and eliminates standing water. Be sure that the drain connection is located at the lowest point in the pan, and install the WSHP within the manufacturer’s tolerance for levelness.

- *Properly install condensate traps and periodically clean them out.*

  If the refrigerant-to-air heat exchanger and its associated drain pan are located *upstream* of the supply fan (*draw-thru* configuration), the pressure inside the WSHP unit casing at the location of the drain pan is *less than* the pressure outside, so air can be drawn in through the condensate drain line from outside. This results in the wetting of the interior of the unit, and may even allow water to leak into the building.

  If the refrigerant-to-air heat exchanger and drain pan are located *downstream* of the supply fan (*blow-thru* configuration), the pressure inside the casing is *greater than* the pressure outside, and air and condensate are pushed out through the condensate drain line. This eliminates concerns for wetting the interior of the unit, but results in conditioned air leaking out of the unit (wasted energy).

  In either a draw-thru or blow-thru configuration, the condensate drain line must include a properly designed drain seal to allow condensate to properly flow out of the drain pan, and maintain the air seal. Although other sealing devices are sometimes used, a simple P-trap is used in the majority of installations. Follow the manufacturer’s recommendations for the design and installation of this condensate trap. Note that the design of the trap differs depending on whether the refrigerant-to-air heat exchanger is a draw-thru or blow-thru configuration.

  Remember, even a well-designed trap, if plugged, causes the drain pan to overflow. Inspect traps regularly for blockage. Clean and prime the trap, if necessary, especially just prior to the cooling season.

- *Include a condensate overflow float switch in each heat pump.*

  This switch turns off the compressor (and closes the OA damper, if equipped) to prevent the drain pan from overflowing if the condensate drain line is plugged. A unit controller can simultaneously send an alarm or diagnostic message to the building automation system, indicating the need to service.

WSHP configurations

Water-source heat pumps are available in several configurations to suit various building types (Figure 19).

A **horizontal** WSHP is designed for installation in a ceiling plenum, with supply air ducted to the zone. They are typically used in buildings where floor space is at a premium, such as office buildings, schools, and retail strip malls.

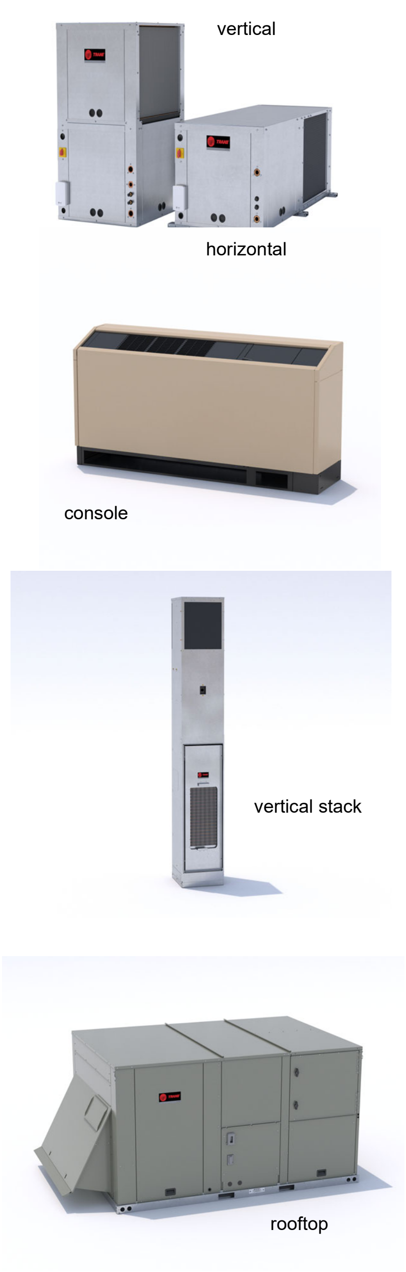
A **vertical** WSHP is designed for installation in a closet or maintenance room, with supply air ducted to the zone. Common applications for smaller vertical units include schools, apartments, condominiums, dormitories, and extended care facilities. Larger vertical units are generally used for very large zones, such as cafeterias and large meeting rooms.

A **console** WSHP is designed for installation within the occupied space (often under windows in perimeter zones), hallways, or entryways. They may also be used when ducted units are not feasible. Typical applications include office buildings, apartment buildings, dormitories, hotels, and extended care facilities.

A **vertical stack** WSHP is designed for installation in the corner of a room in a multi-story building, with supply air typically delivered directly into the zone. They are designed to be stacked above each other to minimize the cost of installing piping and electrical service. Common applications include hotels, apartment buildings, high-rise condominiums, and dormitories.

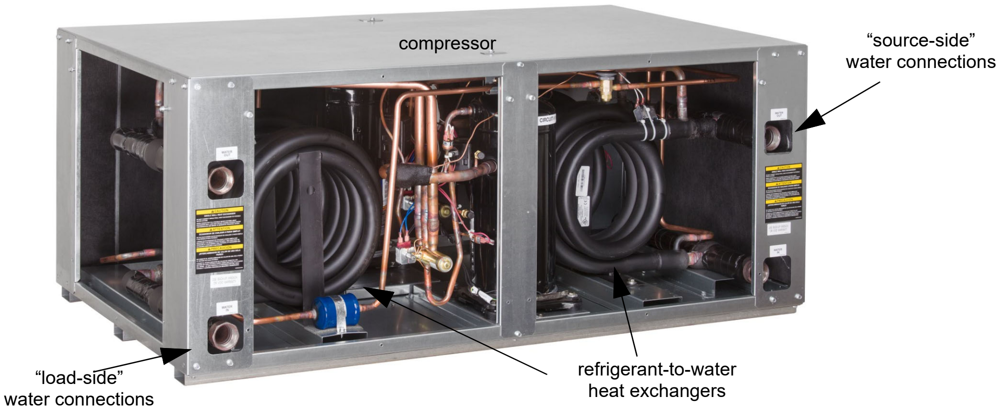
A **rooftop** WSHP is designed for installation outside, typically on the roof of the building, with supply air ducted to the zone. They are commonly used for very large zones, such as cafeterias and gymnasiums, or to replace existing packaged rooftop equipment in a renovation.

Figure 19. Typical water-source heat pump configurations



A variation of the water-source heat pump, called a **water-to-water heat pump**, contains one or more compressors, two refrigerant-to-water heat exchangers (no refrigerant-to-air heat exchanger), an expansion device, and a reversing valve (Figure 20). The refrigeration circuit is pre-engineered and assembled in a factory, so no field-installed refrigerant piping is required.

Figure 20. Water-to-water heat pump

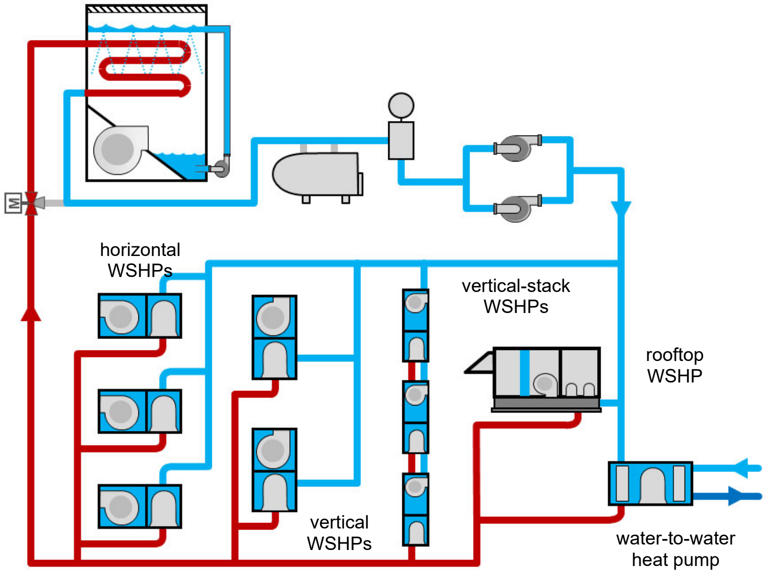


Like a WSHP, a water-to-water heat pump is connected to the common water loop, but rather than cooling or heating air, this type of heat pump is used to cool or heat water in a separate loop. In the cooling mode, the refrigeration circuit transfers heat from the water flowing through the "load-side" heat exchanger to the water flowing through the "source-side" heat exchanger. In the heating mode, the reversing valve changes the operation of the refrigeration circuit to transfer heat from the "source-side" to the "load-side" heat exchanger. In this manner, a water-to-water heat pump can provide either cold or warm water.

Water-to-water heat pumps are commonly used to serve radiant heating (and/or cooling) systems, snow or ice melting systems, and air-handling units as part of a dedicated OA system (see "Dedicated OA equipment types," p. 73). They are also used for heating service water for restrooms, showers, laundry, or kitchens.

The availability of these multiple configurations allows a single WSHP system to be used in a building that has various types of zones. For example, in a hotel building, vertical-stack units might be used in the guest rooms, horizontal units might be used in the smaller meeting rooms, vertical units might be used in the larger meeting rooms and banquet halls, rooftop units might be used for the lobby and restaurant, and water-to-water units might be used to serve the dedicated OA units. All of these different WSHP configurations can be connected to the same water loop (Figure 21).

Figure 21. Example WSHP system serving a hotel, using various unit configurations



AHRI/ISO rating standards

The Air Conditioning, Heating, and Refrigeration Institute (AHRI) establishes rating standards for many types of HVAC equipment. The International Organization for Standardization (ISO) facilitates the international coordination and unification of standards.

The objective of ANSI/AHRI/ASHRAE/ISO Standards 13256–1, *Water-to-Air and Brine-to-Air Heat Pumps—Testing and Rating for Performance*, and 13256–2, *Water-to-Water and Brine-to-Water Heat Pumps—Testing and Rating for Performance*, is to promote the consistent rating of various types and sizes of water-source heat pumps. They cover equipment that is designed for use in either water-source, ground-source, or ground-water heat pump systems.

Table 3 and Table 4 include a summary of these standard rating conditions.

Table 3. ANSI/AHRI/ASHRAE/ISO 13256–1 (1998) Standard Rating Conditions for Water-to-Air Heat Pumps

	Water-source heat pumps	Ground-source heat pumps	Ground-water heat pumps
Cooling mode			
Airflow rate		specified by the manufacturer	
Entering air dry-bulb temperature	80.6°F (27°C)	80.6°F (27°C)	80.6°F (27°C)
Entering air wet-bulb temperature	66.2°F (19°C)	66.2°F (19°C)	66.2°F (19°C)
External static pressure	0 in. H_2O (0 Pa)	0 in. H2O (0 Pa)	0 in. H2O (0 Pa)
Liquid flow rate		specified by the manufacturer	
Entering liquid temperature	86°F (30°C)	77°F (25°C)	59°F (15°C)
Heating mode			
Airflow rate		specified by the manufacturer	
Entering air dry-bulb temperature	68°F (20°C)	68°F (20°C)	68°F (20°C)
Entering air wet-bulb temperature	59°F (15°C)	59°F (15°C)	59°F (15°C)
External static pressure	0 in. H2O (0 Pa)	0 in. H2O (0 Pa)	0 in. H2O (0 Pa)
Liquid flow rate		specified by the manufacturer	
Entering liquid temperature	68°F (20°C)	32°F (0°C)	50°F (10°C)

Notice that neither the airflow rate nor the liquid flow rate is specified by the standard. Rather, they are left to the discretion of the manufacturer. Since these factors can significantly impact the performance of a WSHP, use caution when comparing the performance of one manufacturer to another.

As an example, one manufacturer's 3-ton (11-kW) heat pump may be rated at a liquid flow rate of 8.4 gpm (0.53 L/s), while another manufacturer may rate the same-size unit using 9 gpm (0.57 L/s). The heat pump will operate more efficiently with the higher flow rate, but system pumping energy will also increase. While the 0.6 gpm (0.04 L/s) difference in this example may seem small, this 7 percent increase in flow adds up when you consider that the system may be comprised of 50 or 100 heat pumps.

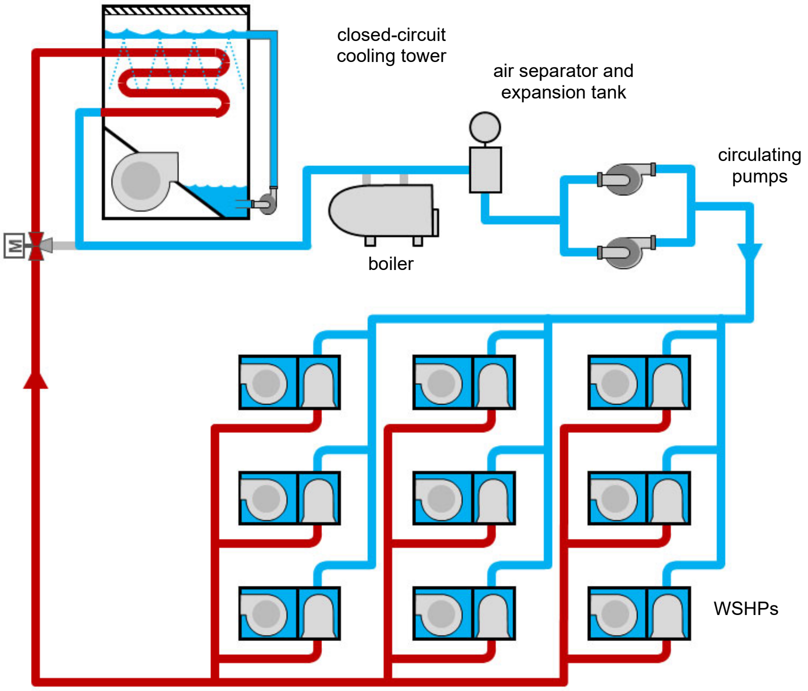
Table 4. ANSI/AHRI/ASHRAE/ISO 13256–2 (1998) Standard Rating Conditions for Water-to-Water Heat Pumps

	Water-source heat pumps	Ground-source heat pumps	Ground-water heat pumps
Cooling mode			
"Load-side" liquid flow rate		specified by the manufacturer	
"Load-side" entering liquid temperature	53.6°F (12°C)	53.6°F (12°C)	53.6°F (12°C)
"Source-side" liquid flow rate		specified by the manufacturer	
"Source-side" entering liquid temperature	86°F (30°C)	77°F (25°C)	59°F (15°C)
Heating mode			
"Load-side" liquid flow rate		specified by the manufacturer	
"Load-side" entering liquid temperature	104°F (40°C)	104°F (40°C)	104°F (40°C)
"Source-side" liquid flow rate		specified by the manufacturer	
"Source-side" entering liquid temperature	68°F (20°C)	32°F (0°C)	50°F (10°C)

Water Distribution Loop

The individual water-source heat pumps are connected to a common water distribution loop. This loop consists of piping, pumps, valves, an air separator and expansion tank, and other accessories. It also connects to the heat rejecter and heat adder. In the example shown in Figure 22, a closed-circuit cooling tower is used as the heat rejecter and a hot-water boiler is used as the heat adder.

Figure 22. Components of water distribution loop



Water-circulating pumps

Because a WSHP can only extract or reject heat while water flows through the refrigerant-to-water heat exchanger, the water-circulating pumps play a critical role in the operation of the system.

Centralized versus distributed pumping

In a WSHP system that employs centralized pumping, the water-circulating pumps are typically installed downstream of the cooling tower and boiler, and upstream of the heat pumps (Figure 22). This helps to ensure positive water pressure throughout the system.

When centralized pumping is used, the most common configuration is to use two pumps manifolded together (see Figure 22), with each pump sized to meet the flow requirements of the entire system. Only one of the pumps operates at any given time, with the second available as a “standby” pump in case the operating pump was to fail.

In some systems, operation of these two pumps is switched periodically to equalize the runtimes and starts (sometimes called “lead/lag”). For example, Pump 1 will be operated for the entire week, with Pump 2 acting as the standby pump; and during the following week, Pump 2 will operate with Pump 1 acting as the standby pump.

In a WSHP system that employs distributed pumping, each heat pump contains a small, single (or dual) pump module sized to meet the flow requirement of just that heat pump. When the heat pump compressor turns off, the pump serving that heat pump also turns off.

Distributed pumping is most commonly used in single-pipe systems (see Figure 25, p. 36) and in ground-coupled systems where each heat pump is connected to a dedicated ground heat exchanger (Figure 91, p. 143).

Advantages of centralized pumping include:

- Fewer pumps to install; fewer connections reduce risk of leaks
- Centralized pump maintenance and fewer strainers to clean
- Pumps can typically be selected to better match the application, which can result in higher pump efficiency

Advantages of distributed pumping include:

- Eliminates the need to install a motorized isolation valve at each heat pump, although a check valve may be required instead
- Often eliminates the need for pump capacity control to achieve variable flow, since each small pump turns off when the heat pump compressor turns off
- Pump failure only impacts one heat pump rather than the entire system (although using a standby centralized pump minimizes this concern)

Constant- versus variable-flow pumping

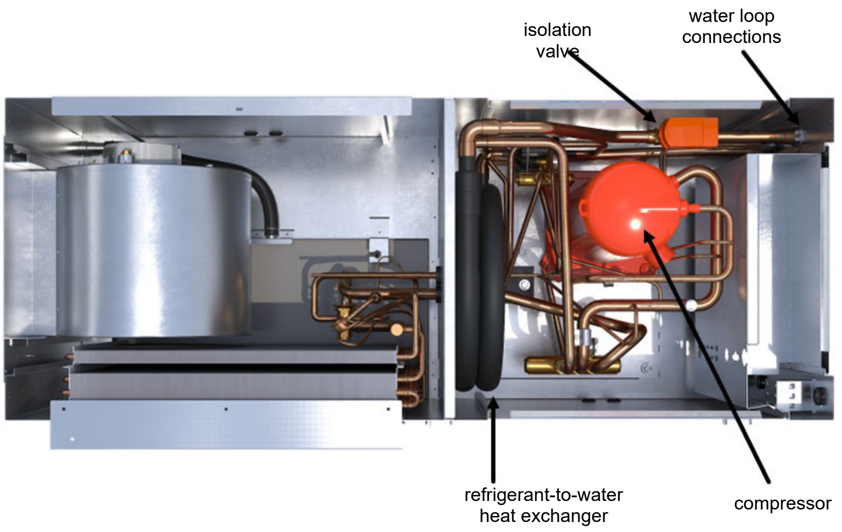
The water-circulating pumps can be either constant- or variable-flow pumps.

Constant-flow pumps operate whenever the system is on, delivering a constant flow of water throughout the loop. This approach is simple and inexpensive to install because no method of pump capacity control is used. However, a constant-flow pump consumes a constant amount of pump energy, regardless of building load.

For many WSHP systems, variable-flow pumping is a prescriptive requirement of ASHRAE Standard 90.1 (see “WSHP distribution loop design and control,” p. 120).

Variable-flow pumps take advantage of the fact that not all of the WSHP compressors in the system are operating at the same time. For example, when a zone needs neither cooling nor heating, the compressor turns off. When the compressor turns off, a motorized, two-position valve (Figure 23) can be used to shut off water flow to that heat pump, so less total water flow is required in the loop. A variable-frequency drive (VFD) on the circulating pump allows the pump to unload, saving energy by delivering only the amount of water required by the operating heat pumps. While variable-flow pumping reduces system energy use, it requires some method to control pump capacity [see “Water-circulating pump(s),” p. 175].

Figure 23. Motorized isolation valve for variable-flow pumping



In a variable-flow system, consider installing an automatic-balancing, flow-control valve for each heat pump. This device helps ensure proper water flow through the heat pump (when the compressor is operating) as the overall system flow rate and pressure change (see "Isolation valves and flow-control devices," p. 38).

Note: In a ground-source heat pump system, consider installing a bypass valve and pipe to avoid pumping water through the ground heat exchanger whenever the temperature of the loop is within the desired range (Figure 124, p. 191). This lowers the pressure drop that the pump must overcome, and reduces pump energy use.

Because it is unlikely that all compressors will need to operate simultaneously, water does not need to flow through all heat pumps simultaneously. This zone-by-zone load variation throughout the day (called "system load diversity," see sidebar on p. 45) affords the opportunity to downsize the water-circulating pumps.

Table 6, p. 44 shows an example eight-zone WSHP system serving a small office building. This system is comprised of 15 water-source heat pumps—totaling 72 tons (250 kW) of installed cooling capacity—connected to a common water-distribution loop. For this example system, the load calculation software indicates the "block" cooling load to be 61 tons (210 kW), so system load diversity is 0.85 (D = 61/72 tons or 210/250 kW).

If constant-flow pumping is used, the water-circulating pumps must be sized to deliver the sum of the individual heat pump water flow rates—216 gpm (13.7 L/s), in this example. However, if variable-flow pumping is used, the pumps might be able to be sized to deliver 15 percent less flow—184 gpm (11.6 L/s), in this example—because the sum of the individual heat pump water flow rates is multiplied by system load diversity [216 gpm × 0.85 (13.7 L/s × 0.85)].

Some HVAC design engineers choose NOT to account for system load diversity when sizing the water-circulating pumps. This allows the pumps to deliver the design water flow rate through each heat pump in case all the heat pumps need to operate simultaneously—during morning warm-up or cool-down modes, for example. This avoids the risk of a nuisance trip due to low water flow, and because the pumps are equipped with a VFD, there is no energy penalty associated with the higher system design water flow rate.

However, other design engineers DO account for system load diversity when sizing the pumps because it allows for the installation of smaller pumps, smaller VFDs and electrical service, and smaller main header piping. Their argument is that it is highly unlikely that all of the heat pumps will ever need to operate simultaneously, and even if they did, the use of automatic flow-control devices and preheating (or precooling) the water loop will minimize the risk of nuisance trips.

A variation on the variable-flow system is to have a separate pump serving different sections of the building. As an example, a “finger-style” elementary school may use a smaller central pump plus a separate pump serving each wing of the school building. With this approach, an entire wing of the building can be shut off when not in use, reducing pumping energy use.

The distributed pumping concept discussed previously uses this same concept, but in that case a separate pump is used for each heat pump. When the heat pump compressor turns off, the pump serving that heat pump also turns off to reduce overall pump energy use.

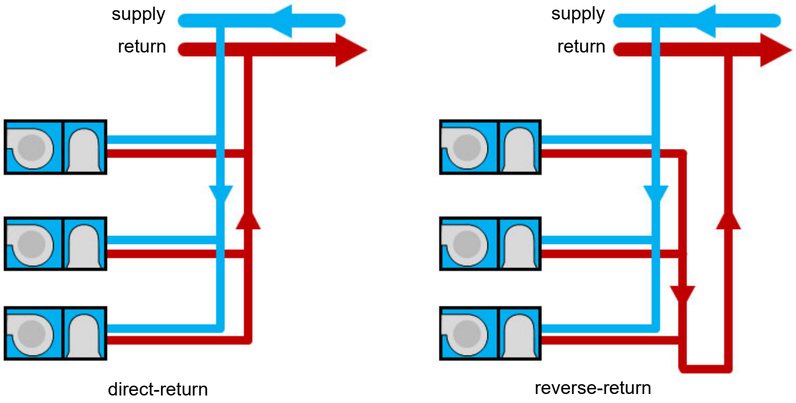
Piping layout

The water distribution loop should be designed to deliver the required water flow to each heat pump while minimizing pump energy use and noise problems. Because piping can account for a large percentage of the total system installation cost, careful planning of the piping arrangement is important.

Direct- versus reverse-return arrangement

A **direct-return piping arrangement** minimizes the amount of piping by returning the water along the same path as it was supplied (Figure 24). In other words, the supply and return pipes for a particular heat pump will be similar in length. Their combined length, however, will be different from that of other heat pumps. The direct-return arrangement, while less costly, requires strict attention to piping layout. Flow-control devices must be used to balance the individual heat pump flow rates to ensure proper water distribution throughout the system.

Figure 24. Direct-return versus reverse-return piping arrangement



Though initially more expensive because of the additional length of return pipe, a **reverse-return piping arrangement** can reduce design layout time and system balancing requirements after installation. With the reverse-return arrangement, the water supplied to each heat pump travels through essentially the same combined length of supply and return pipe (Figure 24).

In multiple-story applications, the risers are typically piped in a direct-return arrangement while a reverse-return arrangement is used to connect the heat pumps on each floor. This arrangement avoids the expense of adding a third, full-sized riser. Balancing valves and proper pipe sizing are used to ensure proper water flow to each floor. Consider installing a drain valve at the base of each supply and return riser to permit system flushing during start-up and maintenance.

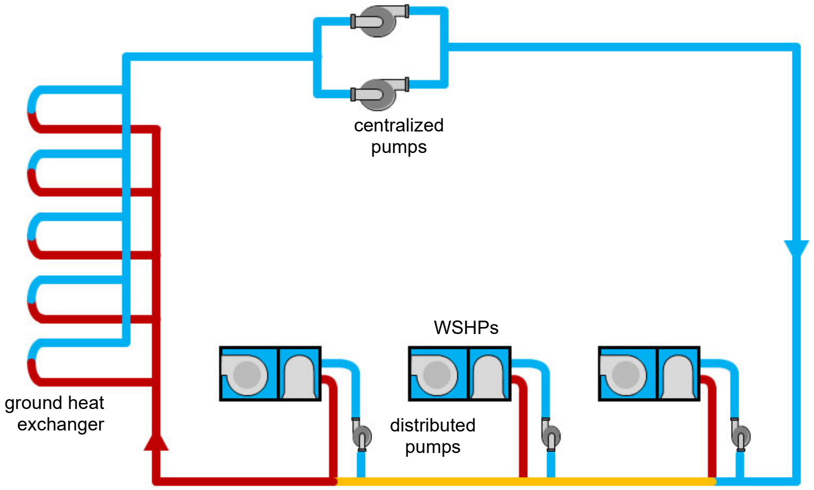
For more information on single-pipe, ground-source heat pump systems, refer to the October 2009 *ASHRAE Journal* article, “One-Pipe Geothermal Design: Simplified GCHP System.”

Single-pipe system

In a single-pipe system, each WSHP is connected to a single pipe that serves as both supply and return. In other words, a portion of the water from the pipe passes through the refrigerant-to-water heat exchanger to either extract heat from the refrigerant or reject heat to the refrigerant, and then returns into the same pipe.

This piping configuration typically uses distributed pumping, in that each heat pump contains a small pump that is sized to meet the flow and pressure drop requirements of only that heat pump (Figure 25). In some cases, a centralized pump may be used to circulate water through the main piping loop and through the heat rejection and heat addition equipment.

Figure 25. Single-pipe system



The primary advantage of this approach is reduced installed cost due to less piping. However, when many heat pumps are connected to the loop, the heat pumps at the end of the pipe can receive water that is significantly warmer or colder than the heat pumps at the beginning of the pipe. This impacts the efficiency of the heat pumps. In addition, flow rates and possibly pipe sizes must be larger because the fluid ∆Ts near the end of the pipe will be smaller.

Pipe sizing

For more information on sizing water piping, refer to Chapter 22, "Pipe Design," in the 2025 *ASHRAE Handbook—*

In addition to the pipe sizing methods described by the *ASHRAE Handbook—Fundamentals*, ASHRAE Standard 90.1 defines a maximum fluid flow rate for various pipe sizes (Table 5). The maximum allowable flow rate depends on the annual system operating hours and whether constant- or variable-flow pumping is used. In order to use the "variable flow/variable speed" column, the system must include a two-position isolation valve at each WSHP and a VFD on the water-circulating pump.

Since the water-circulating pumps in a WSHP system operate during both cooling and heating seasons, the combined number of pump operating hours should be used.

Table 5. Maximum design flow rate for various pipe sizes, gpm (L/s)

	Operating hours/year					
	≤ 2000 hrs/yr		**> 2000 and ≤ 4400 hrs/yr**		**> 4400 hrs/yr**	
Nominal pipe size, in. (mm)	Other	Variable flow/ variable speed	Other	Variable flow/ variable speed	Other	Variable flow/ variable speed
2 ½ (75)	120 (8)	180 (11)	85 (5)	130 (8)	68 (4)	110 (7)
3 (90)	180 (11)	270 (17)	140 (9)	210 (13)	110 (7)	170 (11)
4 (110)	350 (22)	530 (33)	260 (16)	400 (25)	210 (13)	320 (20)
5 (140)	410 (26)	620 (39)	310 (20)	470 (30)	250 (16)	370 (23)
6 (160)	740 (47)	1100 (69)	570 (36)	860 (54)	440 (28)	680 (43)
8 (225)	1200 (76)	1800 (114)	900 (57)	1400 (88)	700 (44)	1100 (69)
10 (280)	1800 (114)	2700 (170)	1300 (82)	2000 (126)	1000 (63)	1600 (101)
12 (315)	2500 (158)	3800 (240)	1900 (120)	2900 (183)	1500 (95)	2300 (145)
For pipe sizes larger than 12 in. (315 mm), the standard specifies a maximum velocity:						
>12 (315)	8.5 ft/s (2.6 m/s)	13.0 ft/s (4.0 m/s)	6.5 ft/s (2.0 m/s)	9.5 ft/s (2.9 m/s)	5.0 ft/s (1.5 m/s)	7.5 ft/s (2.3 m/s)

Source: Table 6.5.4.6 from ASHRAE Standard 90.1-2022. © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org

Pipe insulation

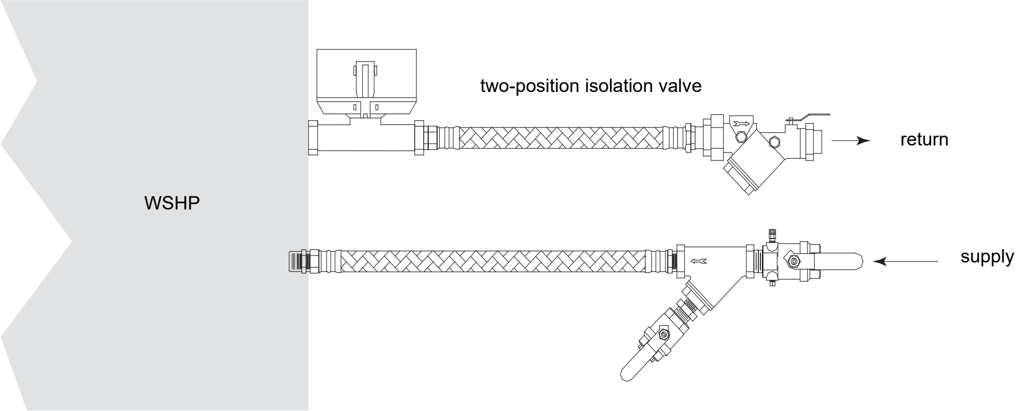
For a conventional boiler/tower WSHP system, typical loop water temperature ranges from about 60ºF (16ºC) to 90ºF (32ºC). At 60ºF (16ºC), the outer surfaces of the piping is typically not cold enough for condensation to form, and at 90ºF (32ºC), the amount of heat loss from the water piping is relatively small. Therefore, in most applications, insulation is added only to those sections of the water loop piping that pass through unheated areas or outside of the building.

This practice is consistent with the requirements of ASHRAE 90.1. Section 6.4.4.1.3 requires piping to be thermally insulated, and includes a table defining minimum insulation thickness. However, exception 2 in this section excludes "Piping that conveys fluids having a design operating temperature range between 60°F (16°C) and 105°F (41°C), inclusive." If the loop water temperature in a WSHP system is within this range, Standard 90.1 does not require insulation.

Isolation valves and flow-control devices

An **isolation valve** is a motorized, two-position, spring-return water valve that is installed in the water pipe leaving the WSHP (Figure 23 and Figure 26). The isolation valve opens whenever the WSHP compressor turns on, allowing water to flow through the refrigerant-to-water heat exchanger. When the compressor turns off, the valve closes slowly to shut off water flow. This provides the opportunity to reduce pump energy use because the pump will only need to move the amount of water required by the operating WSHPs.

Figure 26. Isolation valve installed in the return water pipe



An isolation valve should open quickly to avoid the compressor tripping off on a safety, and should close slowly to avoid water hammer. The electric solenoid in an isolation valve ensures it remains closed to prevent water flow when not desired.

If every heat pump is equipped with an isolation valve, ensure that some isolation valves in the system are open before starting the pump or include a bypass pipe with a pressure-actuated valve in the piping system.

Historically, **water-regulating valves** were used to modulate water flow through the refrigerant-to-water heat exchanger to maintain proper condensing (head) pressure. While a rare few applications may still require their use, the need for water-regulating valves has all but disappeared due to the commonplace use of TXVs and less-expensive isolation valves (see "Water-regulating valves," p. 19).

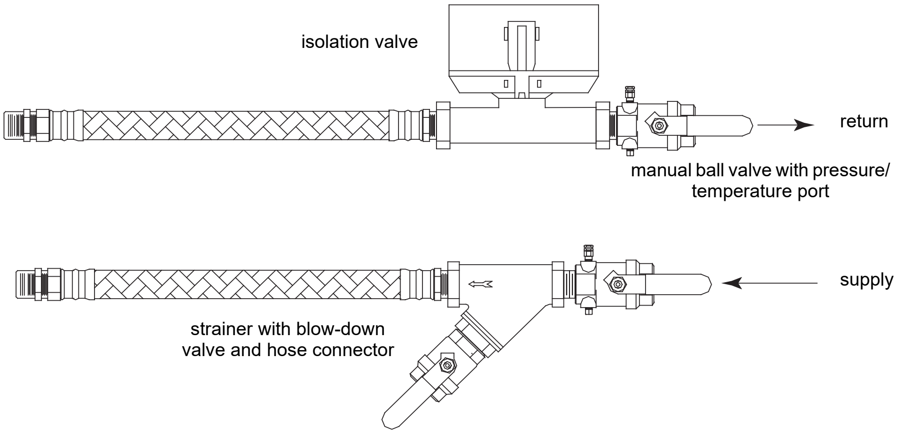
A **flow-control valve** is used to facilitate water balancing at system start-up and to optimize WSHP performance by ensuring the proper water flow rate when the compressor operates. Too little flow can increase compressor energy use and may even damage equipment or shorten equipment life, while too much flow can result in excessive pump energy use.

Some common approaches used to provide for flow control include:

- *Manual ball valves*

  In simple, constant-flow pumping systems, manual, ball-type balancing valves can be installed in the water pipes for each WSHP (Figure 27). During system balancing, pressure and temperature measurements via the ports on the ball valves are used to manually adjust one of the valves to allow more or less water flow through the WSHP. (While only one valve is needed for balancing, valves are typically installed in both the entering and leaving pipes to allow for easy removal or service of the WSHP.)

Figure 27. Manual ball valves for water balancing

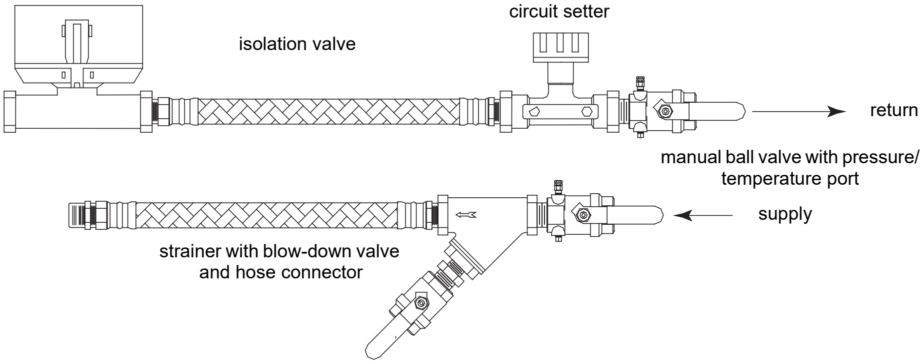


System balancing can be an arduous task because adjusting any ball valve changes the system pressures, so the other valves may need to be re-adjusted to ensure the proper flow rate.

- *Manual flow-control device (circuit setter)*

  Manual balancing using an in-line flow measurement device (such as a circuit setter) offers a simplified approach compared to the conventional technique of measuring the water temperature change across the unit and calculating the corresponding flow rate. The circuit setter flow controller, installed in the leaving water pipe for each WSHP (Figure 28), combines the readout and the adjustment feature in one device.

Figure 28. Manual flow-control device (circuit setter) for water balancing



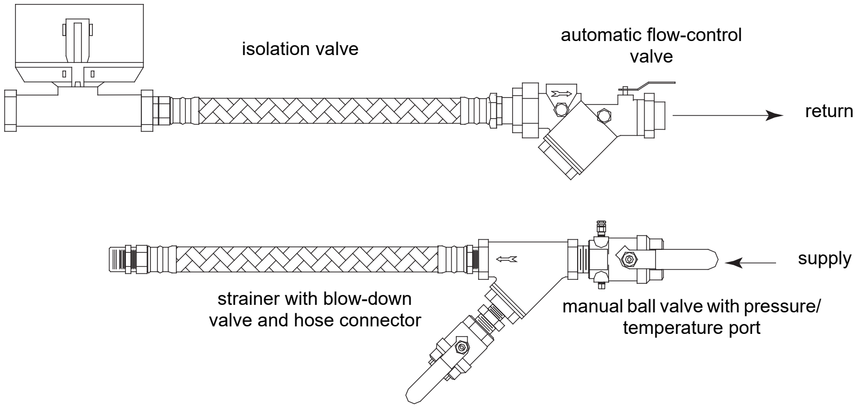
While this simplifies the task of water balancing at each WSHP, the other valves may still need to be re-adjusted since adjusting any circuit setter changes the system pressures.

- *Automatic flow-control device*

  In larger systems, manual balancing may be impractical. And in systems that use variable-flow pumping, manual balancing does not adjust for the variations in system pressure that occur as isolation valves open and close.

An automatic balancing (or self-balancing) flow-control valve, installed in the leaving water pipe for each WSHP (Figure 29), automatically adjusts to provide a constant water flow rate over a wide range of system pressures. It also eliminates the need for the iterative re-adjustments associated with manual flow-control devices.

Figure 29. Automatic flow-control device for water balancing



Manual **shut-off valves** are often installed in both the entering and leaving water pipes for each WSHP to allow for easy removal or service. Sometimes the flow-control valve can be used for this purpose.

Because a WSHP system typically contains many WSHPs that need to be connected to the water distribution piping, factory-provided hose kits are often used as a convenient means for connecting individual heat pumps to the loop piping (Figure 30).

Figure 30. Examples of hose kits used with WSHPs

Other hydronic accessories

There are typically several other accessories included in the water distribution loop.

- A **strainer** is typically installed upstream of each water-circulating pump to protect it from debris flowing inside the water distribution loop. Strainers must be inspected and cleaned periodically to avoid pump cavitation or wasted pump energy use. In some systems—for example, when each WSHP is equipped with an automatic (or self-balancing) flow-control valve—it may also be desirable to install a strainer in the entering water pipe for each WSHP (Figure 29) to protect the flow-control valve from debris.
- Air entrained in the loop water can separate and become "pocketed" inside the system during pump shutdown. While proper piping design and venting can minimize air entrainment, an **air separator** is typically installed upstream of the water-circulating pump(s) to remove any air that does become entrained in the loop (Figure 22, p. 31).
- An **expansion tank** accommodates the expansion and contraction of the water as temperature and, therefore, density changes. While closed or bladder-type tanks can be located anywhere in the system, they are typically installed upstream of the water-circulating pump(s), where the water pressure is lowest (Figure 22, p. 31).

For more information on sizing the expansion tank, refer to Chapter 13, "Hydronic Heating and Cooling," in the 2024 *ASHRAE Handbook—HVAC Systems and Equipment* (www.ashrae.org).

Heat Rejection

A heat rejecter is used to maintain the temperature of the water in the loop below a pre-determined upper limit, such as 90ºF (32ºC) for a boiler/tower WSHP system.

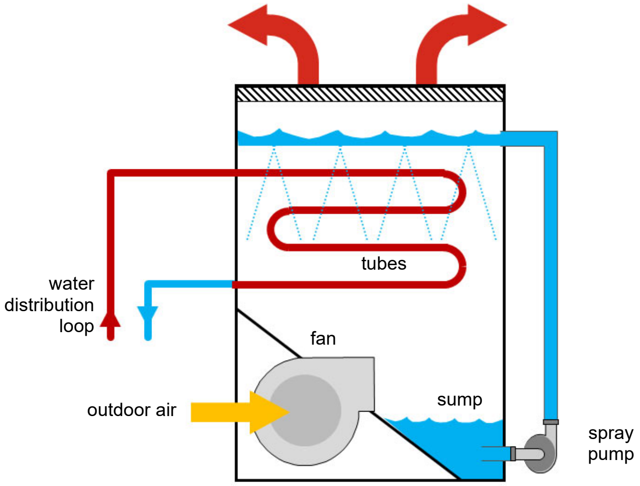
Cooling tower

For some projects, an air-to-water heat pump is used in place of the cooling tower to eliminate water use (see “Electrified WSHP System,” p. 155).

In a boiler/tower WSHP system, the heat rejecter is typically either a closed-circuit cooling tower or an open cooling tower with an intermediate heat exchanger. Either method helps prevent the WSHP heat exchangers from getting clogged with debris.

In a **closed-circuit cooling tower** (sometimes called a **fluid cooler**), warm water from the water distribution loop travels through closed tubes inside the tower, while air is drawn or blown over these tubes by a fan (Figure 31). Water is pumped from the tower sump and sprayed over the tube surfaces. As the air passes over the tubes, it causes a small portion of the water on the outer tube surfaces to evaporate. This evaporation process extracts heat, cooling the water inside the tubes.

Figure 31. Closed-circuit cooling tower (fluid cooler)



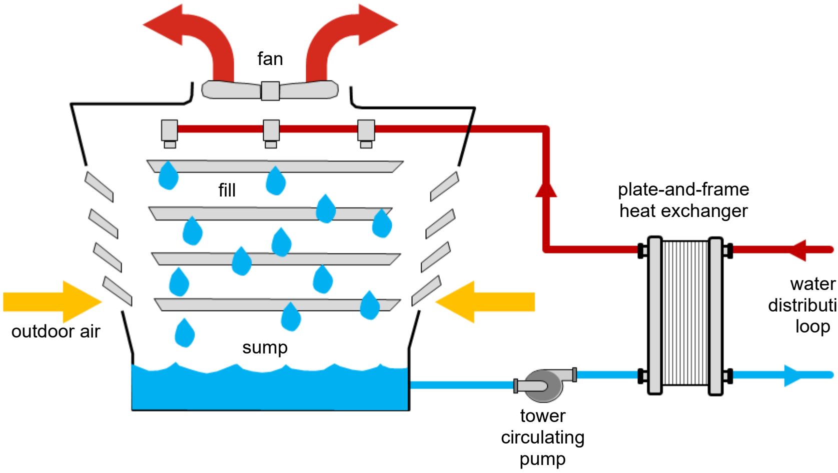
In a closed-circuit cooling tower, the water that is sprayed over the tubes is kept separate from the water that flows inside the tubes and through the refrigerant-to-water heat exchangers of the individual heat pumps. This prevents clogging, corrosion, and scaling inside the WSHP heat exchangers.

A primary advantage of this configuration is that the tower is a single, factory-assembled package. Unlike an open cooling tower, the heat exchanger and circulating pump are an integral part of the tower, so there is no need to design and install an intermediate heat exchanger, or separate pump and piping to pump water through the intermediate heat exchanger and cooling tower (see Figure 32).

Another advantage is the fan in a closed-circuit cooling tower is typically capable of generating higher static pressures than open towers. This provides the opportunity to install the tower indoors, reducing the risk of freezing and minimizing heat loss during cold weather (see “Freeze protection,” p. 46).

In an **open cooling tower**, relatively warm water is sprayed over the fill inside the tower while a fan draws outdoor air upward through the fill (Figure 32). The movement of air through the spray causes some of the water to evaporate, a process that cools the remaining water before it falls into the tower sump. This cooled water is pumped from the sump through a separate heat exchanger, where heat is transferred from the warm water returning from the heat pumps to the cooler water coming from the cooling tower. This intermediate heat exchanger is used to keep the two water loops separate, preventing clogging, corrosion, and scaling inside the refrigerant-to-water heat exchangers of the individual heat pumps.

Figure 32. Open cooling tower with intermediate heat exchanger



A primary advantage of this configuration is that the cooling tower can be located outside while the heat exchanger is located inside. This still reduces the risk of freezing and minimizes heat loss during cold weather, because only the tower sump must be protected from freezing (see “Freeze protection,” p. 46). Less space inside the building is required to house the heat exchanger than to accommodate a closed-circuit cooling tower.

Another advantage is an open cooling tower typically uses less fan energy than a closed-circuit tower. However, this configuration requires design and installation of an intermediate heat exchanger and a separate pump and piping to pump water through the intermediate heat exchanger and open cooling tower.

Sizing the cooling tower

Because all heat pumps on the loop are not likely to be operating in the cooling mode at the exact same time, the cooling tower should be sized to account for system load diversity. The 2024 *ASHRAE Handbook—HVAC Systems and Equipment* (p. 9.21) defines system load diversity as “the maximum instantaneous cooling load of the [system] divided by the installed cooling capacity.” For a WSHP system, the installed cooling capacity is the sum of all the individual heat pump cooling capacities.

Table 6 shows an example eight-zone WSHP system serving a small office building. This system is comprised of 15 heat pumps connected to a common water distribution loop.

Table 6. Sizing the cooling tower for a WSHP system (example office building)

	Nominal cooling capacity, tons (kW)	Water flow rate, gpm (L/s)	Heat rejected to loop, tons (kW)*
South offices	5 (18)	15 (0.95)	5.8 (20)
West offices	5 (18)	15 (0.95)	5.8 (20)
South conf room	(2) 4-ton (14-kW)	24 (1.5)	9.1 (32)
East offices	5 (18-kW)	15 (0.95)	5.8 (20)
South interior offices	(4) 5-ton (18-kW)	60 (3.8)	23 (81)
North interior offices	(4) 5-ton (18-kW)	60 (3.8)	23 (81)
North offices	4 (14)	12 (0.76)	4.5 (16)
North conf room	5 (18)	15 (0.95)	5.8 (20)
Sum	72 (250)	216 (13.7)	83 (290)

** Assumes 90°F (32°C) entering water temperature, 1520 cfm (0.72 m^3/s) of airflow for each 4-ton (14-kW) unit and 1700 cfm (0.80 m^3/s) for each 5-ton (18-kW) unit, entering-air conditions of 77°F (25°C) dry bulb and 63°F (17°C) wet bulb, and 0.5 in. H_2O (125 Pa) of external static pressure loss.*

Following is a process for sizing the cooling tower:

1. Sum the heat rejection from all heat pumps connected to the loop.
2. Apply system load diversity to estimate the maximum, instantaneous heat rejection required of the cooling tower.
3. Calculate the cooling tower range.
4. Select the cooling tower using the design ambient wet-bulb temperature, design system water flow rate, design leaving-water temperature, and range.

The **first step** is to sum the heat rejected from all heat pumps that are connected to the loop. The most accurate approach is to use a manufacturer's catalog or selection software to determine the performance of each heat pump—including heat rejected to the loop—at the project-specific operating conditions (such as airflow, entering air conditions, water flow rate, and entering water temperature).

The eight-zone system in this example is comprised of 15 water-source heat pumps (Table 6). The sum of the heat rejected from each heat pump is 83 tons (290 kW).

An alternate approach is to sum the nominal cooling capacities of all the heat pumps, and then estimate total heat rejected to the loop using an estimated value for cooling COP (Coefficient of Performance). The total heat rejected to the loop ($Q_{rejected}$) equals the cooling capacity ($Q_{cooling}$) plus the heat of compression ($Q_{compressor}$), which can be determined using the estimated cooling COP.

$$Q_{rejected} = Q_{cooling} + Q_{compressor} = Q_{cooling} \times (1 + 1/COP_{cooling})$$

For this example, the sum of the nominal cooling capacities for all 15 heat pumps is 72 tons (250 kW). From the manufacturer's catalog, the rated cooling EER for the 5-ton (18-kW) heat pump is 14.8, which equates to a COP of 4.34 (COP = EER x 0.293). Therefore, using this alternate approach, the heat rejection for all heat pumps is estimated to be 89 tons (310 kW).

$$Q_{rejection} = 72 \text{ tons} \times (1 + 1/4.34) = 89 \text{ tons}$$

$$[Q_{rejection} = 250 \text{ kW} \times (1 + 1/4.34) = 310 \text{ kW}]$$

Failing to account for system load diversity when selecting the cooling tower will likely result in a larger-than-necessary tower. This increases installed cost and energy use, while providing little added benefit to system performance. If load calculation software has not been used to determine the "block" cooling load, consider using the following conservative estimates:

For a system water flow rate less than 200 gpm (13 L/s), assume 90 percent system load diversity.

For a system water flow rate between 200 gpm (13 L/s) and 300 gpm (19 L/s), assume 85 percent system load diversity.

For a system water flow rate greater than 300 gpm (19 L/s), assume 80 percent system load diversity.

The **second step** is to apply system load diversity (D) to estimate the maximum, instantaneous heat rejection required of the cooling tower. As explained earlier, system load diversity is the maximum instantaneous (or "block") cooling load of the system—which is typically determined with building load calculation software—divided by the total installed cooling capacity.

For this example system, the load calculation software indicates the "block" cooling load to be 61 tons (210 kW), so the system load diversity is 0.85 (D = 61/72 tons or 210/250 kW). Using the sum of heat rejected to the loop, as estimated by a manufacturer's selection software (Table 6), the maximum instantaneous heat rejection required of the cooling tower is calculated to be 71 tons (250 kW).

$$Q_{rejection} = 83 \text{ tons} \times 0.85 = 71 \text{ tons}$$

$$(Q_{rejection} = 290 \text{ kW} \times 0.85 = 250 \text{ kW})$$

The **third step** is to calculate the range of the cooling tower. The range is the difference (ΔT) between water temperatures entering and leaving the cooling tower.

$$\text{Range} = 24 \times Q_{rejection} \text{ (tons) / system water flow rate (gpm)}$$

$$[\text{Range} = 0.24 \times Q_{rejection} \text{ (kW) / system water flow rate (L/s)}]$$

In the equations for Range, 24 and 0.24 are not constants, but are derived from properties of water:

Density = 8.33 gallons/ft^3 (998 kg/m^3)

Specific heat = 1.0 Btu/lb•°F (4.18 kJ/kg•°K)

12000 Btu/hr/ton / (8.33 gallons/ft^3 × 1.0 Btu/lb•°F × 60 min/hr) = 24

[1000 L/m^3 / (998 kg/m^3 × 4.18 kJ/kg•°K) = 0.24]

If constant-flow pumping is used in this example, the system water flow rate is 216 gpm (13.7 L/s) and the cooling tower range is 7.9°F (4.4°C).

$$\text{Range} = 24 \times 71 \text{ tons} / 216 \text{ gpm} = 7.9°\text{F}$$

$$[\text{Range} = 0.24 \times 250 \text{ kW} / 13.7 \text{ L/s} = 4.4°\text{C}]$$

If variable-flow pumping is used, the sum of the individual heat pump water flow rates is multiplied by system load diversity (D), and range is calculated using 184 gpm (11.6 L/s) as the system water flow rate (216 gpm x 0.85 or 13.7 L/s x 0.85). In this case, the resulting cooling tower range is 9.3°F (5.2°C).

$$\text{Range} = 24 \times 71 \text{ tons} / 184 \text{ gpm} = 9.3°\text{F}$$

$$[\text{Range} = 0.24 \times 250 \text{ kW} / 11.6 \text{ L/s} = 5.2°\text{C}]$$

Note: Even if system load diversity has not been used to downsize the water-circulating pumps and piping in a variable-flow system, it is important to apply *diversity to the water flow rate used to select the cooling tower. This avoids selecting a larger-than-necessary tower.*

The **fourth step** is to use the manufacturer's catalog or selection software to select the cooling tower, using the design outdoor wet-bulb temperature, the system water flow rate, the design leaving-water temperature, and the calculated range.

For this example system, assuming variable-flow pumping is used, the cooling tower should be selected based on:

- Design ambient wet-bulb temperature = 78°F (26°C)
- Design system water flow rate = 184 gpm (11.6 L/s)
- Design leaving-water temperature = 90°F (32°C)
- Range = 9.3°F (5.2°C)

Freeze protection

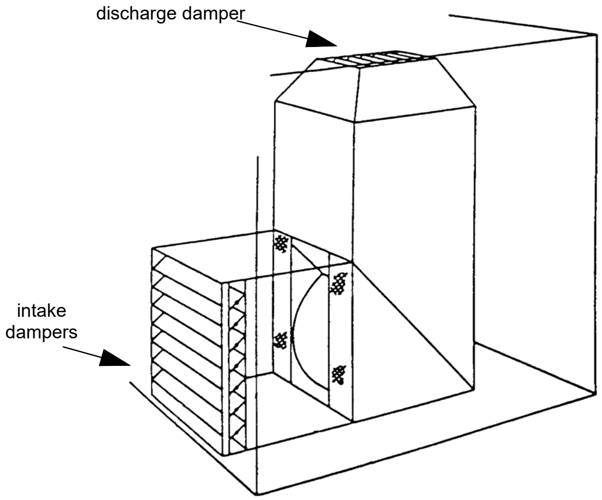
Because the water loop operates all year long, freeze protection during winter months is important. In a boiler/tower WSHP system, freeze protection depends on the type of cooling tower used.

If a *closed-circuit cooling tower* is used, one of the following approaches is typically used for freeze protection:

- **Locate the cooling tower inside the building**

  With this approach, dampers are typically installed in the tower discharge and in either the intake ductwork or behind the louvers of an intake plenum (Figure 33). Both sets of dampers close whenever the tower is off, minimizing heat loss and preventing water inside the tower sump and heat exchanger tubes from freezing.

Figure 33. Closed-circuit cooling tower with intake and discharge dampers



  Using a modulating damper in the tower discharge provides more precise temperature control during cold weather, which might eliminate the need to drain the tower sump for winter operation. If a two-position (rather than a modulating) damper is used, the tower sump should be drained when the outside temperature is below freezing conditions. The tower is still able to operate, but it will operate as a “dry cooler,” with no water flowing over the outside surfaces of the tubes.

Of course, locating the tower inside the building requires extra floor space, and the associated dampers and ductwork increase the system installed cost.

- **Use an antifreeze solution**

Ground-coupled heat pump systems can also experience temperatures below freezing. These systems are generally protected by adding antifreeze to the loop water (see “Ground-coupled heat pump systems,” p. 137).

Mixing in antifreeze (such as glycol) with the water in the closed distribution loop lowers the temperature at which the solution will freeze. Given a sufficient concentration of glycol, no damage will occur to the closed distribution loop. The tower sump, however, must still be protected from freezing.

As the temperature drops below the glycol solution freeze point, ice crystals begin to form. Because the water freezes first, the remaining glycol solution is further concentrated and remains a fluid. The combination of ice crystals and fluid makes up a flowable slush. The fluid volume increases as this slush forms and flows into available expansion volume.

“Freeze protection” indicates the concentration of antifreeze required to prevent ice crystals from forming at the given temperature (Table 7). “Burst protection” indicates the concentration required to prevent damage to equipment (e.g., coil tubes bursting). Burst protection requires a lower concentration of glycol, which results in less degradation of heat transfer.

Table 7. Concentration required for freeze protection vs. burst protection

	Ethylene Glycol		Propylene Glycol	
	Concentration (% volume)		Concentration (% volume)	
Temperature	Freeze protection	Burst protection	Freeze protection	Burst protection
20°F (-7°C)	16	11	18	12
10°F (-12°C)	25	17	29	20
0°F (-18°C)	33	22	36	24
-10°F (-23°C)	39	26	42	28
-20°F (-29°C)	44	30	46	30
-30°F (-34°C)	48	30	50	33
-40°F (-40°C)	52	30	54	35
-50°F (-46°C)	56	30	57	35
-60°F (-51°C)	60	30	60	35

Source: Dow Chemical Company. 2008. *HVAC Application Guide: Heat Transfer Fluids for HVAC and Refrigeration Systems*. www.dow.com/heattrans.

If the closed-circuit cooling tower is bypassed during sub-freezing weather—meaning that fluid does not flow out through the tower (see Figure 115, p. 178)—a concentration that provides “burst protection” is usually sufficient. A concentration that provides “freeze protection” is only needed in those cases where no ice crystals can be permitted to form (which would be the case if the tower was not bypassed during sub-freezing weather) or where there is inadequate expansion volume available.

The advantage of this approach is that it is predictable and relatively easy to maintain. However, adding antifreeze to the loop degrades the capacity and efficiency of the heat pumps, possibly increasing the size and cost of these components. In addition, it increases the fluid pressure drop through the system, which increases pump energy use. Therefore, ensure that the selection of individual heat pumps and water-circulating pumps reflect the effect of the antifreeze solution.

Metal pipes and heat exchangers that are exposed to antifreeze solutions are vulnerable to corrosion, so appropriate inhibitors must be added to the solution to prevent corrosion. Because these inhibitors can degrade over time, it is important to conduct a periodic chemical analysis to maintain proper antifreeze and inhibitor concentrations. In addition, a minimum percentage of antifreeze may be required to minimize the potential for microbial growth. Consult the fluid provider.

- **Winterize the cooling tower**

  With this approach, a closed-circuit cooling tower installed outside is "winterized" by insulating the casing of the tower, adding an ice-proof damper in the tower discharge, adding insulation and heat tape to all exposed water piping (including the make-up water line and spray pumps), and adding a sump heater, if the sump will not be drained during the winter.

  The addition of insulation and discharge dampers significantly decreases the amount of heat loss from the tower during cold weather.

If an *open cooling tower* is used, the intermediate heat exchanger should be located inside the building. This usually eliminates the need to protect the closed water distribution loop from freezing. One of the following approaches is typically used to protect the tower sump from freezing:

- **Drain the tower sump during winter**

  In some systems, an automatic drain valve opens to drain the water from the tower sump when the outdoor temperature drops below a pre-determined limit, such as 35°F (2°C). In other systems, building maintenance personnel manually drain the tower sump when the weather begins to get cold.

  In either case, the sump should be inspected periodically to ensure that leaves and debris do not clog the sump drain. If some zones require cooling when the tower sump is empty, there needs to be enough heat pumps operating in the heating mode, extracting heat from loop, to keep the temperature of the loop from getting too warm.

- **Locate the tower sump inside the building**

  With this approach, after the water drops through the tower fill, it drains by gravity into the sump, which is located inside the building.

  Of course, locating the sump inside requires extra floor space, and the associated piping increases the system installed cost.

Natural heat sink

Some WSHP systems use a natural heat sink, such as the ground or a lake, as the heat rejecter. For more information on ground-coupled, surface-water, and groundwater systems, see "System Design Variations," p. 135.

Heat Addition

A heat adder is used to maintain the temperature of the water in the distribution loop above a pre-determined lower limit, such as 60ºF (16ºC) for a boiler/tower WSHP system.

Hot-water boiler

For some projects, an air-to-water heat pump is used in place of the boiler to avoid the use of fossil fuels (see "Electrified WSHP System," p. 155).

In a boiler/tower WSHP system, the heat adder is typically a hot-water boiler operated either by electricity or a fossil fuel, such as natural gas or heating oil. Alternatively, buildings that contain separate hot-water or steam heating systems may use a heat exchanger to add heat to the water distribution loop. This isolates the water in the distribution loop from the water in the other system.

A hot-water boiler is a pressure vessel that typically consists of a water tank (or tubes with water flowing through them), a heat exchanger, fuel burners, exhaust vents, and controls. It transfers the heat generated by burning fuel to either water or steam. The majority of boilers used in WSHP systems are low-pressure (<160 psig [1100 kPa] and <250°F [120°C]), hot-water boilers.

Non-condensing versus condensing boilers

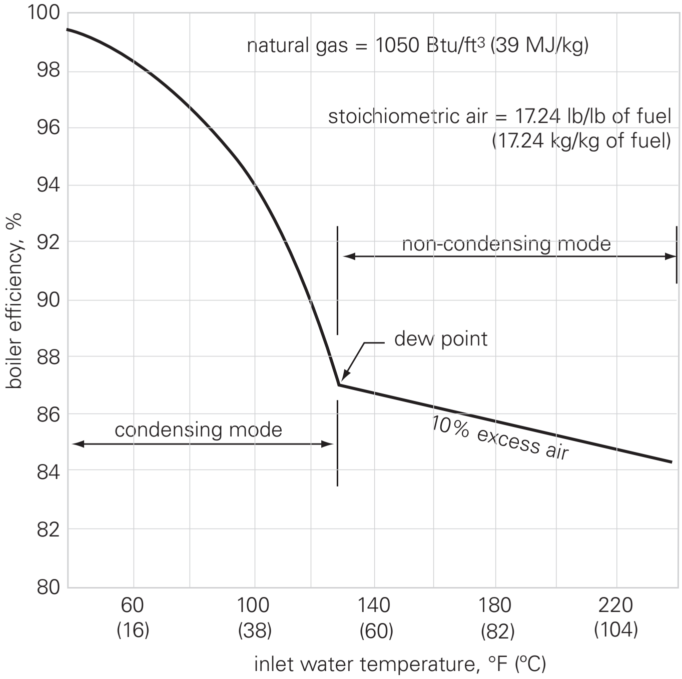
For more information on the various types of boilers, refer to Chapter 32 (Boilers) of the 2024 *ASHRAE Handbook—HVAC Systems and Equipment* (www.ashrae.org).

Hot-water boilers are classified by whether they are condensing or non-condensing. A conventional, non-condensing boiler is designed to operate without condensing the flue gases inside the boiler. Only the sensible heat value of the fuel is used to heat the hot water. All of the latent heat value of the fuel is lost up the exhaust stack. This avoids corrosion of cast-iron or steel parts. Hot-water systems with non-condensing boilers are often operated to ensure that the return-water temperature is no lower than 140ºF (60ºC) to prevent condensing.

A condensing boiler, on the other hand, uses a high-efficiency heat exchanger that is designed to capture nearly all of the available sensible heat from the fuel, as well as some of the latent heat of vaporization. The result is a significant improvement in boiler efficiency. Condensing, gas-fired boilers have combustion efficiencies that range from 88 percent to over 95 percent, while non-condensing boilers have combustion efficiencies that range from 80 percent to 86 percent.

Condensing of the flue gases also allows for a lower return-water temperature, much lower than the 140ºF (60ºC) limit that is common with non-condensing boilers. In fact, the efficiency of a condensing boiler *increases* as the return-water temperature decreases (Figure 34). To maximize the efficiency of a condensing boiler, therefore, it is important that the rest of the heating system be designed to operate at these lower return-water temperatures.

Figure 34. Impact of return-water temperature on efficiency of condensing boilers



Source: 2024 *ASHRAE Handbook—HVAC Systems and Equipment*, Chapter 32, Figure 6. © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org.

Because of the potential for corrosion, a condensing boiler must be constructed of special materials that will resist the corrosive effects of the condensing flue gases. This typically results in a higher first cost.

Finally, condensing boilers must be vented with a corrosion-resistant stack. However, since most of the heat has been removed from the combustion gases, the stack for a condensing boiler is usually smaller than for a non-condensing boiler. In addition, it can often be constructed out of PVC pipe (although stainless steel may be required in some cases) and can often be directly vented through an exterior wall of the building.

Sizing the boiler in a system with night setback

Zone setpoints are typically relaxed when the zone is scheduled to be unoccupied, allowing the temperature in the zone to either increase or decrease (see "Zone is unoccupied," p. 4)—a practice often called "night setback." While this strategy reduces energy use during unoccupied periods, night setback can impose an additional heating load on the system during morning warm-up mode.

When night setback is used, all heat pumps on the loop might need to operate in the heating mode at the exact same time during the morning warm-up mode. In this case, the boiler must be sized to offset the heat extracted by all the units connected to the loop.

One way to reduce the boiler capacity needed to satisfy morning warm-up mode is to add a storage tank (see "Hot-water (thermal) storage," p. 53).

Alternatively, the building automation system could be used to stagger the morning warm-up mode for different parts of the building, thus warming only part of the building at a time. This can also reduce required boiler capacity. A potential secondary benefit is lowering the electrical demand during this staggered startup.

Table 8 includes an example eight-zone WSHP system serving a small office building. This system is comprised of 15 heat pumps connected to a common water distribution loop.

Table 8. Sizing the boiler for a WSHP system (example office building)

	Nominal cooling capacity, tons (kW)	Water flow rate, gpm (L/s)	Heating capacity, MBh (kW)	Heat extracted from loop, MBh (kW)*
South offices	5 (18)	15 (0.95)	71 (21)	55 (16)
West offices	5 (18)	15 (0.95)	71 (21)	55 (16)
South conf room	(2) 4-ton (14-kW)	24 (1.5)	108 (32)	86 (25)
East offices	5 (18)	15 (0.95)	71 (21)	55 (16)
South interior offices	(4) 5-ton (18-kW)	60 (3.8)	284 (84)	220 (64)
North interior offices	(4) 5-ton (18-kW)	60 (3.8)	284 (84)	220 (64)
North offices	4 (14)	12 (0.76)	54 (16)	43 (13)
North conf room	5 (18)	15 (0.95)	71 (21)	55 (16)
Sum	72 (250)	216 (13.7)	1014 (300)	790 (230)

* *Assumes 60°F (16°C) entering water temperature, 1520 cfm (0.72 m^3/s) of airflow for each 4-ton (14-kW) unit and 1700 cfm (0.80 m^3/s) for each 5-ton (18-kW) unit, 68°F (20°C) entering-air temperature, and 0.5 in. H_2O (125 Pa) of external static pressure loss.*

The most accurate approach for determining the heat extracted by the heat pumps is to use a manufacturer's catalog or selection software to determine the performance of each heat pump—including heat extracted from the loop—at the project-specific operating conditions (such as airflow, entering air conditions, water flow rate, and entering water temperature). For this example, the sum of the heat extracted by all the heat pumps is 790 MBh (230 kW).

An alternate approach is to sum the heating capacities of all the heat pumps, and then estimate total heat extracted from the loop using an estimated value for heating COP. The total heat extracted from the loop ($Q_{extracted}$) equals the heating capacity ($Q_{heating}$) minus the heat of compression ($Q_{compressor}$), which can be determined using the estimated heating COP.

$$Q_{extracted} = Q_{heating} - Q_{compressor} = Q_{heating} \times (1 - 1/COP_{heating})$$

For this example, the sum of the heating capacities for all 15 heat pumps is 1014 MBh (300 kW). From the manufacturer's catalog, the rated heating COP for the 5-ton (18-kW) heat pump is 5.0. Therefore, using this alternate approach, the heat extracted by all heat pumps is estimated to be 810 MBh (240 kW).

$$Q_{extracted} = 1014 \text{ MBh} \times (1 - 1/5.0) = 810 \text{ MBh}$$

$$[Q_{extracted} = 300 \text{ kW} \times (1 - 1/5.0) = 240 \text{ kW}]$$

In this case, the boiler should be selected with 810 MBh (240 kW) of capacity.

Note: If other cooling equipment—such as water-cooled, computer room air conditioning equipment—is connected to the loop, and operates during the morning warm-up mode, the heat rejected by this equipment can be used to reduce the size of the boiler.

Sizing the boiler in a system without night setback

In some applications—particularly those buildings that operate 24 hours a day, 7 days a week—night setback is not used, so there is no morning warm-up mode where all heat pumps will operate in the heating mode simultaneously. In this case, the total heat extracted from the loop ($Q_{extracted}$) equals the building's worst-case heating load ($Q_{heating}$) minus the heat of compression ($Q_{compressor}$), which can be determined using the estimated heating COP.

$$Q_{extracted} = Q_{heating} - Q_{compressor} = Q_{heating} \times (1 - 1/COP_{heating})$$

For this same example system (Table 8), load calculation software indicates the design heating load to be 890 MBh (260 kW). From the manufacturer's catalog, the rated heating COP for the 5-ton (18-kW) heat pump is 5.0. Therefore, the heat extracted is estimated to be 710 MBh (210 kW).

$$Q_{extracted} = 890 \text{ MBh} \times (1 - 1/5.0) = 710 \text{ MBh}$$

$$[Q_{extracted} = 260 \text{ kW} \times (1 - 1/5.0) = 210 \text{ kW}]$$

In this case, the boiler should be selected with 710 MBh (210 kW) of capacity.

Hot-water (thermal) storage

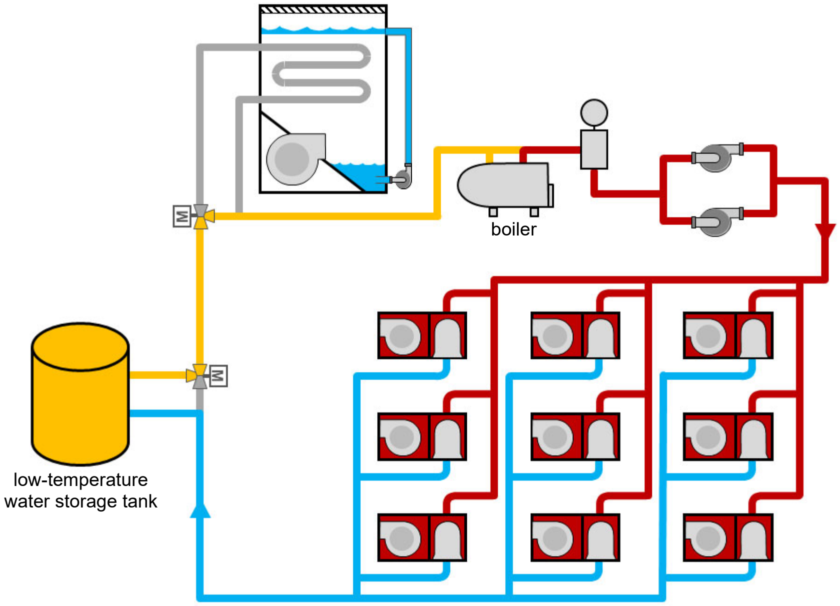
As mentioned previously, the use of night setback might require the selection of a larger-capacity boiler because all the heat pumps on the loop might need to operate in the heating mode at the exact same time during the morning warm-up mode.

One approach to reducing the required boiler capacity is to install a hot-water storage tank in the water distribution loop. During the unoccupied mode, the boiler is operated for a brief period of time to increase the temperature of the water inside the loop and storage tank. When morning warm-up mode begins, the heat stored in the water is extracted to offset some of the heating loads, allowing for the installation of a smaller boiler.

Low-temperature storage

The most common approach for adding hot-water storage to a WSHP system is to simply store heat at temperatures that correspond to the typical limits of loop operation—between 60°F (16°C) to 90°F (32°C), for example. A "low-temperature" storage tank is typically installed upstream of the boiler, and downstream of the heat pumps, to allow the boiler to quickly add heat to the loop during normal operation (Figure 35).

Figure 35. Low-temperature storage tank added to the loop



With this "low-temperature" storage approach, no additional heating equipment is needed, and the tank can also be used for other purposes:

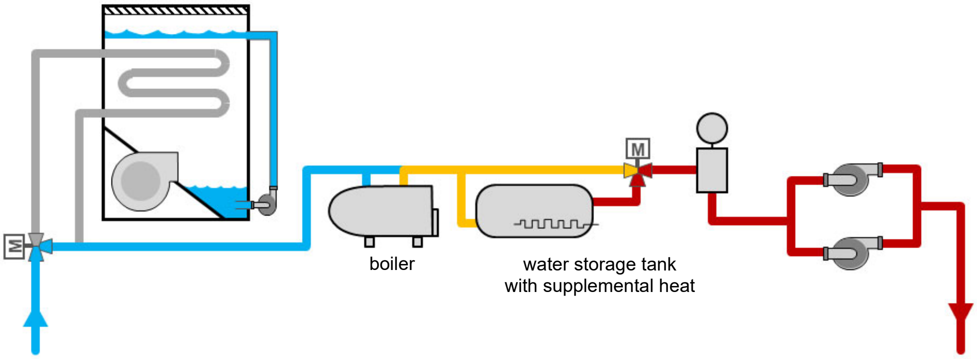
- If the electric utility has a time-of-day rate structure, during cold weather the boiler could be operated during off-peak hours (when the cost of electricity is lower) to increase the temperature of the water inside the loop and storage tank. This would allow the heat pump compressors—those operating in the heating mode—to operate more efficiently during on-peak hours (when the cost of electricity is higher) and use less electricity. This is especially valuable if the system uses an electric boiler, because it shifts operation of the boiler to the off-peak period.

- If the electric utility has a time-of-day rate structure, during the cooling season, the cooling tower could be operated during off-peak hours to decrease the temperature of the water inside the loop and storage tank. This would allow the heat pump compressors—those operating in the cooling mode—to operate more efficiently during on-peak hours and use less electricity. This strategy can also delay, or avoid, the need to operate the cooling tower during the on-peak period, at which time the outdoor wet-bulb temperatures would likely be higher.
- During those times in the year when it is warm during the day and cool at night, the addition of a storage tank allows the loop to store more heat that is rejected from heat pumps operating in the cooling mode during daytime hours. This delays, and may even avoid, the need to operate the cooling tower. The heat stored in the water can then be used to offset the nighttime heating loads.

High-temperature storage

If the electric utility has a time-of-day utility rate with a high, on-peak electrical demand charge, the use of "high-temperature" storage might be a more attractive approach (Figure 36). In this case, the storage tank is installed downstream of the boiler. During off-peak hours (when the cost of electricity is lower), a heating element in the storage tank is used to increase the temperature of the water inside the tank—typically up to 180°F (82°C) or higher. By storing the water at a much warmer temperature, a given tank size can store significantly more heat, or a much smaller size can be used to store the same amount of heat as a larger "low-temperature" storage tank.

Figure 36. High-temperature storage tank added to the loop



During morning warm-up mode—or whenever heat must be added to the loop during the day—hot water from the storage tank is mixed into the loop, maintaining the loop temperature at either the lower limit, or at some higher temperature that would allow the heat pump compressors—those operating in the heating mode—to operate more efficiently during on-peak hours.

Sizing the storage tank

As mentioned, the most common use of hot-water storage in a WSHP system is to reduce or eliminate the need to operate the hot-water boiler during the morning warm-up mode. This often allows for the installation of a smaller boiler, and can reduce peak electrical demand if an electric boiler is used.

1. **Calculate the total amount of heat that must be extracted from the loop during morning warm-up.** The total amount of heat extracted during morning warm-up mode depends on the difference between the unoccupied and occupied heating setpoints, the thermal capacitance of the building, and the COP of the heat pumps.

   First, calculate the total amount of heat ($Q_{warm\text{-}up}$, in Btu [kJ]) needed to raise the indoor temperature from the unoccupied heating setpoint to the occupied heating setpoint. This calculation should assume that the dedicated outdoor-air system is shut off, and should take credit for heat generated by lights or any other heat-producing equipment that will be operating during the morning warm-up period.

   Then, calculate the amount of heat that must be extracted from the loop ($Q_{extracted}$). This equals the total amount of heat needed to warm-up the building ($Q_{warm\text{-}up}$) minus the heat of compression, which can be approximated using the average heating COP of the heat pumps.

   $Q_{extracted} = Q_{warm\text{-}up} \times (1 - 1/COP_{heating})$

   As an example, consider a building where the total amount of heat needed for morning warm-up ($Q_{warm\text{-}up}$) is 500,000 Btu (527,000 kJ). Assuming an average heating COP of 4.0, the total amount of heat that must be extracted from the loop ($Q_{extracted}$) is 375,000 Btu (395,000 kJ).

2. **Calculate the amount of heat stored in the loop water (without storage).** The amount of heat stored in the water loop depends on the volume of fluid in the loop, properties of that fluid, and the difference between the temperature in the loop at the beginning of the morning warm-up period and the lower temperature limit at which the boiler will be activated.

   First, calculate the volume of water—in gallons (Liters)—inside the loop. Then, assuming that the loop is preheated prior to the start of the morning warm-up period, calculate the heat stored by the water in the loop (Q_{loop}).

   $Q_{loop} = V \times \rho \times C_p \times \Delta T$

   where,

   V = volume of fluid inside the loop, gal (L)

   ρ = density of the fluid, lb/gal (kg/L)

   C_p = specific heat of the fluid, Btu/lb-°F (kJ/kg-°K)

   ΔT = initial loop temperature minus lower limit, °F (°K)

   Using the same example, consider that the WSHP system serving the building has a water loop volume of 600 gal (2300 L), which is preheated to 90°F (32°C) prior to the start of the morning warm-up period. The density of water is 8.33 lb/gal (1.0 Kg/L) and the specific heat of water is 1.0 Btu/lb-°F (4.18 kJ/kg-°K). The loop lower limit, at which the boiler will be activated, is 60°F (16°C).

   Q_{loop} = 600 gal × 8.33 lb/gal × 1.0 Btu/lb-°F × (90°F - 60°F) = 150,000 Btu

   [Q_{loop} = 2300 L × 1.0 kg/L × 4.18 kJ/kg-°K × (32°C - 16°C) = 155,000 kJ]

   In this example, the loop by itself can provide a total of 150,000 Btu (155,000 kJ) of heat to the heat pumps.

3. **Calculate the size of the storage tank.** The difference between the total amount of heat that must be extracted from the loop during morning warm-up mode and the heat stored by the water loop by itself determines the amount of heat that must be added to the loop by the hot-water boiler and/or the storage tank.

   If the intent is to avoid the need to operate the boiler during the morning warm-up period—as might be the case when an electric boiler is being used and there is a benefit in avoiding a higher electrical demand when all the heat pumps start in the morning—then the storage tank should be large enough to provide any heat that the loop by itself is unable to provide.

   $Q_{tank} = Q_{extracted} - Q_{loop}$

   For the same example, a storage tank sized for 225,000 Btu (240,000 kJ) plus the 150,000 Btu (155,000 kJ) provided from the loop itself would be sufficient to provide the total heat required for the morning warm-up period. Using the equation above, this corresponds to a 900-gallon (3400-Liter) tank.

   However, if the intent is to simply avoid the need to oversize the boiler to be able to provide enough heat for morning warm-up, then the storage tank could be smaller since the boiler will also be operating during the morning warm-up period.

   $Q_{tank} = Q_{extracted} - Q_{loop} - Q_{boiler}$

   For this example, assuming that the boiler has a capacity of 100,000 Btu/h (107,000 kJ/h) and the morning warm-up period lasts one hour, a storage tank sized for 125,000 Btu (133,000 kJ) would be needed. This size of tank plus the 100,000 Btu/h (107,000 kJ/h) from the boiler plus the 150,000 Btu (155,000 kJ) provided from the loop itself would be sufficient to provide the total heat required for the morning warm-up period. This corresponds to a 500-gallon (1900-Liter) tank.

Alternatively, the tank could be sized to store excess heat during the day for use later at night. Many newer buildings, designed and constructed in accordance with current building and energy codes, experience a net cooling load during daytime operation, even at cold ambient temperatures. In such a case, the heat rejected to the loop during daytime hours (or during a mild winter day) could be stored for use when the building requires heat during the cold nighttime hours (or during a colder winter day).

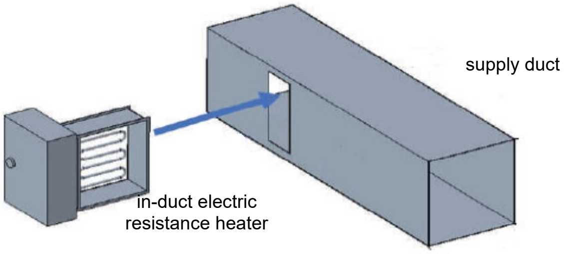
The process for sizing the tank is the same as above, except that Step 1 is used to calculate the total amount of heat extracted from the loop during the nighttime (unoccupied) hours. While the storage tank can be sized to provide all of the nighttime heat required, it should also be verified that there is sufficient excess heat rejected to the loop during the daytime hours to “re-charge” the loop and tank.

Of course, budgetary constraints, available space, and ability to support the weight of the tank are also factors that can dictate the maximum storage tank size that can be installed in a given building. After calculating the required tank size, the design engineer can evaluate the installed cost versus the operating cost savings to determine the optimal tank size.

Electric resistance heat for a “boiler-less” system

For applications where the heat pumps are not expected to operate in the heating mode for many hours during the year, an alternative to using a centralized boiler as the “heat adder” is to install an electric resistance heater (Figure 37) in the downstream ductwork. In this configuration, the heat pump operates the compressor in normal heating mode until the temperature of the water loop drops below a pre-determined low limit—55ºF (13ºC) for example. At that time, the compressor is disabled and the electric resistance heater is energized to provide heat to the zone.

Figure 37. Electric resistance heater for a “boiler-less” system



In a cooling-dominated application, it is likely that many heat pumps in the system will still be operating in the cooling mode, adding heat to the water loop. When the loop temperature rises again—to 60ºF (16ºC) for example—the electric resistance heater is disabled and the heat pump compressor is again allowed to operate in the normal heating mode.

Table 9 summarizes the advantages and disadvantages of this “boiler-less” approach.

Table 9. Advantages and disadvantages of a “boiler-less” WSHP system

Advantages:	Disadvantages:
• Electric resistance heaters in the WSHP ductwork typically result in a lower installed cost than a centralized boiler • Avoids space required to install a centralized boiler • Affords the opportunity to bill each tenant for all electricity used for heating, making it attractive for a building where each tenant is billed for the electricity used to cool and heat their space only • With an electric resistance heater installed, the zone can still be supplied with heat if the compressor fails, or if the water pump fails and cannot circulate water through the heat pump • Potential improvement in perceived comfort due to warmer supply-air temperatures that are possible from electric heat during heating mode	• May increase the size of electrical wiring to the individual heat pumps, which impacts installed cost • Unable to use a hot-water storage tank to limit electrical demand during morning warm-up or to shift energy use to an off-peak period (this drawback may be minimal in locations where the electric utility does not use a time-of-day rate structure or does not include a demand charge) • Not able to use a fossil fuel (such as natural gas, propane, or fuel oil) as the heat source which, depending on local utility rates, may result in higher utility bills • Uses more energy since the COP of an electric resistance heater is 1.0, while the heating COP of a WSHP might be between 3.0 and 6.0 (depending on model and operating conditions)

For a “boiler-less” WSHP system in a cold climate, where most of the heat pumps may need to operate in the heating mode simultaneously, very little heat will be added to the loop. During such a situation, heat loss through the tower and any exposed or underground piping may cause the loop water temperature to decrease below the low-temperature safety limit. This could trip the WSHP controller and prevent the compressor from starting until the diagnostic is cleared manually. To prevent this from occurring, consider installing a small boiler or water heater, sized only to offset any expected heat loss through the tower and exposed or underground piping.

Natural heat source

Some WSHP systems use a natural heat source, such as the ground or a lake, as the heat adder. For more information on ground-coupled, surface-water, and ground-water systems, see “System Design Variations,” p. 135.

Dedicated Outdoor-Air System

Most building codes require a minimum quantity of outdoor air (OA) be provided to each zone for ventilation (see “Ventilation,” p. 90). This outdoor air can be brought into the building locally or centrally.

A rooftop-style WSHP includes an outdoor-air damper that allows outdoor air to be brought into, and conditioned by, each heat pump. Similarly, a console-type WSHP may include an optional OA damper that allows outdoor air to be brought into the heat pump through an opening in the perimeter wall of the building.

However, in most WSHP systems—particularly those that use horizontal, vertical, console, or vertical stack models—the outdoor air required for ventilation is typically conditioned and delivered by a dedicated outdoor-air system (DOAS).

Conditioning the outdoor air with a dedicated system allows the local heat pumps to handle only the zone loads and not the ventilation load. This can result in more stable comfort control, lower indoor humidity levels (see “Methods for improving dehumidification performance,” p. 101), and possibly smaller heat pumps. In facilities that require special filtering of the outdoor air or humidification during the winter, a dedicated OA system allows these processes to be handled in a centralized piece of equipment.

Dedicated OA system configurations

Following are four example dedicated OA system configurations that are commonly used to deliver the conditioned outdoor air in a WSHP system. Table 10 summarizes the advantages and drawbacks of each configuration.

Conditioned OA delivered directly to each zone

The example configuration shown in Figure 38 delivers the conditioned outdoor air (CA) directly to each zone through a separate duct system and diffuser(s). The WSHP conditions only recirculated air (RA).

Figure 38. Conditioned OA delivered directly to each zone

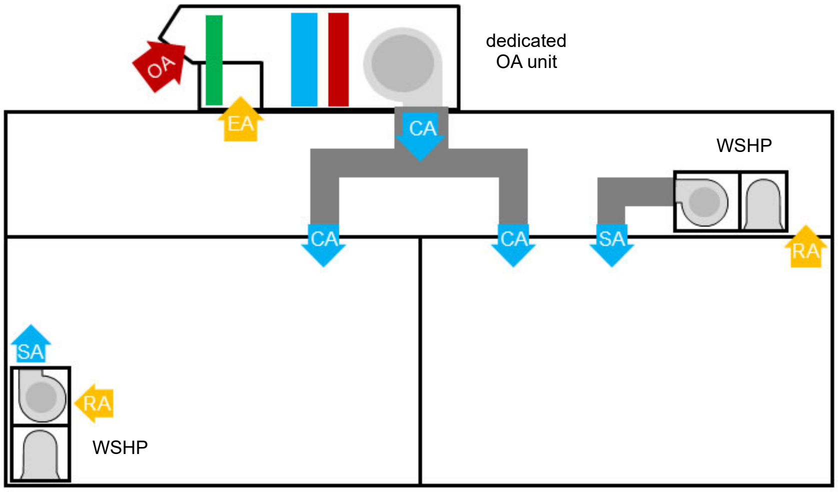
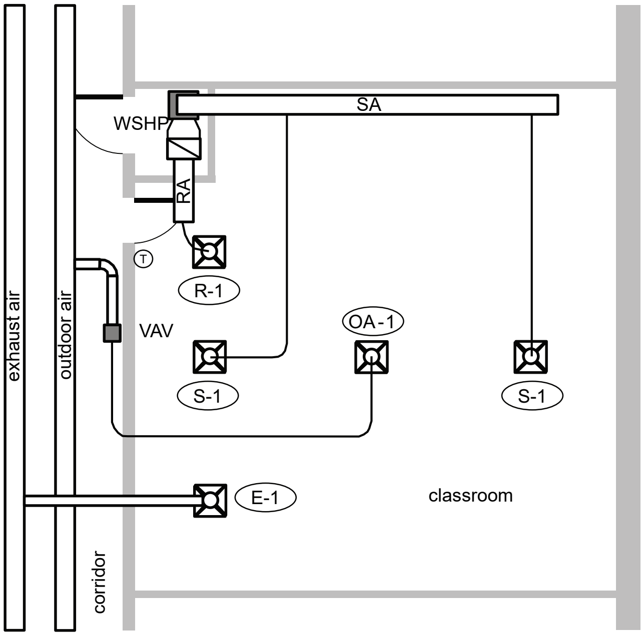


Figure 39 depicts another example of delivering conditioned OA directly to the zone. In this case, a vertical WSHP installed in an adjacent closet conditions recirculated air (RA).

Figure 39. Conditioned OA ducted directly to a classroom, with a vertical WSHP installed in the adjacent closet



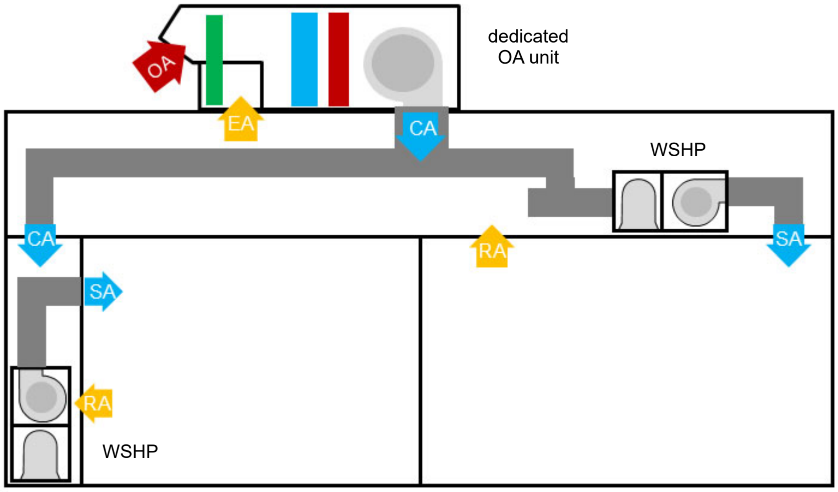
In this example layout, there is an added cost to install extra ductwork and diffuser(s), versus simply dumping the conditioned outdoor air into the closet where the WSHP is located. However, if the conditioned OA is delivered at a cold (rather than "neutral") temperature, most of the heat pumps can typically be downsized, since they deliver less airflow with less cooling capacity (see "Neutral- versus cold-air delivery," p. 64). This reduces not only the cost of the heat pumps, but less supply airflow means that the supply (SA) and return (RA) ductwork can be smaller. And smaller-capacity heat pumps require less water flow, which results in smaller piping, valves, and pumps, and smaller electrical service.

Finally, delivering the conditioned OA at a cold temperature directly to each space provides cooling and fan energy savings and allows the local fans to cycle or reduce speed without impacting the delivery of outdoor air for ventilation. Be sure to consider all the cost impacts, and energy savings, when considering the benefit of this approach.

Conditioned OA delivered directly to the intake of each WSHP

The example configuration shown in Figure 40 delivers the conditioned outdoor air (CA) directly to the intake of each WSHP, where it mixes with recirculated air (RA) from the zone. The WSHP conditions this mixture of conditioned outdoor air and recirculated air, and delivers it to the zone through a shared supply duct system and diffuser(s).

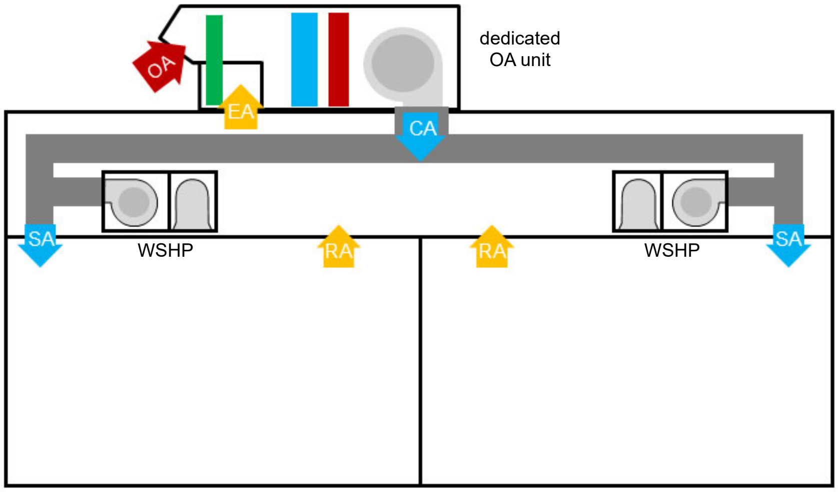
Figure 40. Conditioned OA delivered to the intake of each WSHP



Conditioned OA delivered to the supply-side of each WSHP

The example configuration shown in Figure 41 delivers the conditioned outdoor air (CA) directly to the supply-side of each WSHP, where it mixes with supply air from the heat pump before being delivered to the zone. The WSHP conditions only recirculated air (RA).

Figure 41. Conditioned OA delivered to the supply-side of each WSHP



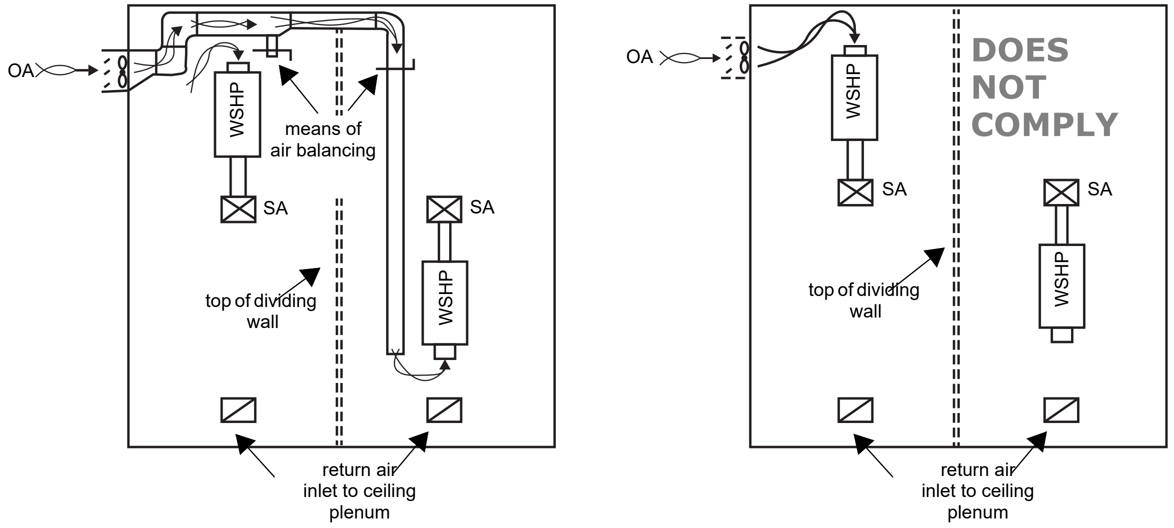
Conditioned OA delivered to the open ceiling plenum, near each WSHP

The example configuration shown in Figure 42 delivers the conditioned outdoor air (CA) to the ceiling plenum, near the intake of each WSHP. The conditioned outdoor air mixes with recirculated air (RA) in the plenum before being drawn in through the WSHP intake. The WSHP conditions this mixture and delivers it to the zone through a shared supply duct system and diffuser(s).

ASHRAE Standard 62.1 addresses this configuration in Section 5.10.2, with additional explanation found in the accompanying *Standard 62.1 User's Manual* (Figure 42):

"**5.10.2 Plenum Systems.** When the ceiling or floor plenum is used both to recirculate return air and to distribute ventilation air to ceiling-mounted or floor-mounted terminal units, the system shall be engineered such that each space is provided with its required minimum ventilation airflow. **Informative Note:** Systems with direct connection of ventilation air ducts to terminal units, for example, comply with this requirement."

Figure 42. Conditioned OA delivered to the open ceiling plenum, near each WSHP



Correct plan of plenum system with discharge near terminal ends
Though the ducts are not connected to the terminal units, they discharge near them, with balancing means available to provide correct airflow to each.

Incorrect plan of plenum system
In this case, outdoor air ventilation is provided to one ventilation zone, but not the other. This could only meet the requirement if it could be shown that sufficient air gets to the remote system, perhaps by mixing between the zones.

Source: ASHRAE *62.1-2019 User's Manual,* Figures 5-D and 5-E ©American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc, www.ashrae.org.

Table 10. Comparison of different dedicated OA system configurations

Conditioned OA delivered directly to each zone	
Advantages:	**Disadvantages:**
• Makes it easier to ensure the required amount of outdoor air reaches each zone, because separate ventilation diffusers allow easy airflow measurement and balancing • Affords opportunity to cycle off the fan inside the heat pump (reducing fan energy use) when the compressor cycles off, because outdoor air is not distributed to the zone by the WSHP fan • Allows the dedicated OA system to operate during unoccupied periods (for after-hours humidity control or preoccupancy purge, for example) without needing to operate the fans inside the heat pumps • Affords the opportunity to downsize local heat pumps (reducing installed cost and energy use) if the conditioned outdoor air is delivered at a cold temperature (rather than reheated to “neutral”)	• Requires installation of additional ductwork and separate diffusers • May require multiple diffusers to ensure that outdoor air is adequately dispersed throughout the zone
Conditioned OA delivered to the intake of each WSHP	
Advantages:	**Disadvantages:**
• Helps ensure the required amount of outdoor air reaches each unit, because the OA is ducted directly to each intake • Avoids the cost and space needed to install additional ductwork and separate diffusers • Easier to ensure that outdoor air is adequately dispersed throughout the zone, because outdoor air is distributed by the WSHP fan	• Measurement and balancing is more difficult than if the OA was delivered directly to the zone via separate diffusers • May need to increase zone outdoor airflow to account for $E_Z < 1.0$ during heating mode • Typically requires a field-fabricated plenum or section of duct to connect the outdoor-air duct and mix it with recirculated air prior to entering the heat pump • Fans inside the heat pumps must operate continuously to provide ventilation during scheduled occupancy, rather than cycling off with the compressor • If the dedicated OA system operates during unoccupied periods (e.g., for after-hours humidity control or preoccupancy purge), the fans inside the heat pumps typically must operate also
Conditioned OA delivered to the supply-side of each WSHP	
Advantages:	**Disadvantages:**
• Helps ensure the required amount of outdoor air reaches each unit, because the OA is ducted directly to the supply-side of each heat pump • Avoids the cost and space needed to install additional ductwork and separate diffusers • Affords the opportunity to downsize local heat pumps (reducing installed cost and energy use) if the conditioned outdoor air is delivered at a cold temperature (rather than reheated to “neutral”) • Easier to ensure that outdoor air is adequately dispersed throughout the zone, because outdoor air is distributed with the WSHP supply air	• Measurement and balancing is more difficult than if the OA was delivered directly to the zone via separate diffusers • May need to increase zone outdoor airflow to account for $E_Z < 1.0$ during heating mode • Fans inside the heat pumps typically must operate continuously to provide ventilation during scheduled occupancy, rather than cycling off with the compressor (unless a pressure-independent VAV terminal is used to maintain outdoor airflow)
Conditioned OA delivered to the open ceiling plenum, near each WSHP	
Advantages:	**Disadvantages:**
• Avoids the cost and space needed to install additional ductwork, separate diffusers, or field-fabricated mixing plenums	• More difficult to ensure the required amount of outdoor air reaches each unit, since the OA is not ducted directly to each zone or heat pump • May need to increase zone outdoor airflow to account for $E_Z < 1.0$ during heating mode • Conditioned outdoor air may not be able to be delivered at a cold temperature, due to concerns over condensation within the ceiling plenum (rather, it is typically reheated to a “neutral” temperature) • Fans inside the heat pumps must operate continuously to provide ventilation during scheduled occupancy, rather than cycling off with the compressor • If the dedicated OA system operates during unoccupied periods (e.g., for after-hours humidity control or preoccupancy purge), the fans inside the heat pumps typically must operate also

Neutral- versus cold-air delivery

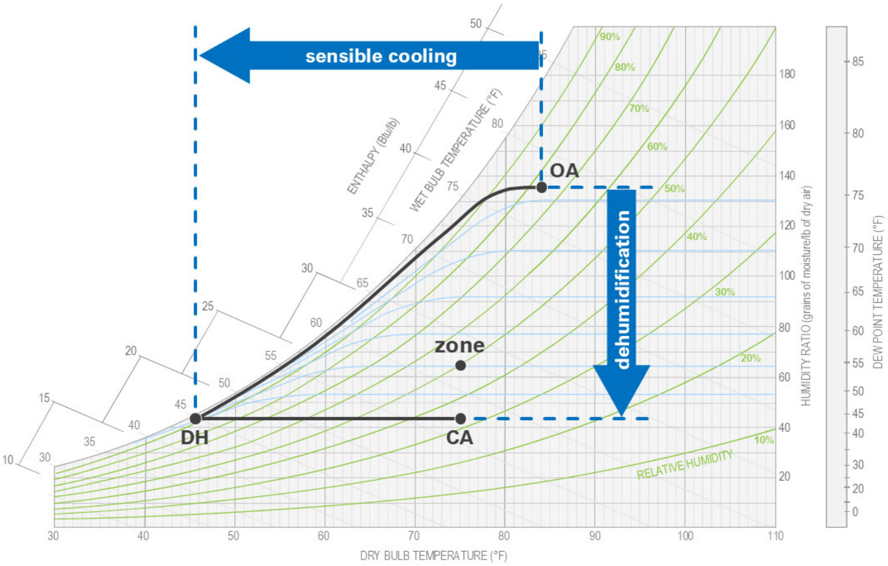
Regardless of where the conditioned outdoor air is delivered, the dedicated OA unit should dehumidify the outdoor air so that it is drier than the zone. This offsets the latent load associated with ventilation and, if the dew-point temperature of the conditioned outdoor air is lower than the dew point in the zone (Figure 43), also offsets some (or all) of the zone latent loads. This approach can adequately limit indoor humidity levels, at both full- and part-load conditions, without the need for additional dehumidification enhancements in the local heat pumps (see “Methods for improving dehumidification performance,” p. 101).

For a detailed discussion on how to determine the **required supply-air dew-point temperature** from a dedicated OA unit, refer to the Trane application guide, *Dedicated Outdoor Air Systems* (SYS-APG001*-EN), or the Trane *Engineers Newsletter*, titled “Impact of DOAS Supply-Air Dew-Point Temperature on Space Humidity” (ADM-APN073-EN).

Many dedicated OA systems are designed to dehumidify the outdoor air and then reheat it to approximately zone temperature (neutral). Delivering the dehumidified outdoor air at a neutral dry-bulb temperature can simplify control because it has no impact on the zone sensible cooling or heating loads.

However, when a chilled-water or DX cooling coil is used for dehumidification, a by-product of that process is that the dry-bulb temperature of the air leaving the coil is colder than the zone (Figure 43). If the dehumidified outdoor air (DH) is reheated to neutral (CA), most of the sensible cooling performed by the dedicated OA unit is wasted.

Figure 43. Sensible cooling is a by-product of 'cold-coil' dehumidification



If the dedicated OA system delivers air directly to each zone (see Figure 38, p. 59) or to the supply-side of each WSHP (see Figure 41, p. 61), the dehumidified outdoor air (DH) can be delivered “cold,” rather than reheated to neutral. The low dry-bulb temperature of the conditioned OA offsets part of the sensible cooling load in the zone, reducing the energy used by the local heat pump. At design conditions, this means that the heat pump can be sized for less airflow and less cooling capacity than in a neutral-air system.

As an example, consider a four-classroom wing of an elementary school served by a dedicated OA system (Table 11). In the “neutral-air delivery” scenario, the dedicated OA unit dehumidifies the outdoor air to 45°F (7.2°C) dew point, then reheats the air to 71°F (22°C) dry bulb before delivering it directly into the classrooms. The dew point of this conditioned OA is low enough to offset the latent load in the classrooms, but because it is delivered at a dry-bulb temperature that is near the desired zone temperature—which is 74°F (23°C) in this example—it offsets only a small portion of the sensible cooling load in the classrooms.

For Classroom 101, the design sensible cooling load in the zone is 29,800 Btu/hr (8.7 kW). When the 450 cfm (0.21 m^3/s) of outdoor air required for that zone is delivered at a neutral dry-bulb temperature—71°F (22°C) in this example—the conditioned OA offsets only 1,470 Btu/hr (0.43 kW) of the zone sensible cooling load. The local WSHP must be sized to offset the remaining 28,330 Btu/hr (8.3 kW) of sensible load, with a supply airflow of 1380 cfm (0.65 m^3/s)—assuming the WSHP cools the air to 55°F (13°C) DBT.

Table 11. Example of cold-air versus neutral-air delivery

	Classroom 101		**Classroom 102**	
Zone outdoor airflow (V_{oz})	450 cfm (0.21 m^3/s)		450 cfm (0.21 m^3/s)	
Zone sensible cooling load ($Q_{zone,sensible}$)	29,800 Btu/hr (8.7 kW)		26,800 Btu/hr (7.9 kW)	
Zone cooling setpoint (DBT_{zone})	74°F (23°C)		74°F (23°C)	
	Neutral-air delivery DBT_{ca} = 71°F (22°C)	**Cold-air delivery DBT_{ca} = 55°F (13°C)**	**Neutral-air delivery DBT_{ca} = 71°F (22°C)**	**Cold-air delivery DBT_{ca} = 55°F (13°C)**
Sensible cooling offset by the DOAS: $Q_{ca} = 1.085 \times V_{oz} \times (DBT_{zone} - DBT_{ca})$	1,470 Btu/hr (0.43 kW)	9,280 Btu/hr (2.7 kW)	1,470 Btu/hr (0.43 kW)	9,280 Btu/hr (2.7 kW)
Sensible cooling required by local WSHP: $Q_{wshp} = Q_{zone,sensible} - Q_{ca}$	28,330 Btu/hr (8.3 kW)	20,520 Btu/hr (6.0 kW)	25,330 Btu/hr (7.4 kW)	17,520 Btu/hr (5.1 kW)
Airflow required by local WSHP, assuming DBT_{sa} = 55°F (13°C): $Q_{wshp} = 1.085 \times V_{sa} \times (DBT_{zone} - DBT_{sa})$	1380 cfm (0.65 m^3/s)	1000 cfm (0.47 m^3/s)	1230 cfm (0.58 m^3/s)	850 cfm (0.40 m^3/s)
	Classroom 103		**Classroom 104**	
Zone outdoor airflow (V_{oz})	480 cfm (0.23 m^3/s)		440 cfm (0.21 m^3/s)	
Zone sensible cooling load ($Q_{zone,sensible}$)	26,900 Btu/hr (7.9 kW)		28,300 Btu/hr (8.3 kW)	
Zone cooling setpoint (DBT_{zone})	74°F (23°C)		74°F (23°C)	
	Neutral-air delivery DBT_{ca} = 71°F (22°C)	**Cold-air delivery DBT_{ca} = 55°F (13°C)**	**Neutral-air delivery DBT_{ca} = 71°F (22°C)**	**Cold-air delivery DBT_{ca} = 55°F (13°C)**
Sensible cooling offset by the DOAS: $Q_{ca} = 1.085 \times V_{oz} \times (DBT_{zone} - DBT_{ca})$	1,560 Btu/hr (0.46 kW)	9,900 Btu/hr (2.9 kW)	1,430 Btu/hr (0.42 kW)	9,070 Btu/hr (2.7 kW)
Sensible cooling required by local WSHP: $Q_{wshp} = Q_{zone,sensible} - Q_{ca}$	25,340 Btu/hr (7.4 kW)	17,000 Btu/hr (5.0 kW)	26,870 Btu/hr (7.9 kW)	19,230 Btu/hr (5.6 kW)
Airflow required by local WSHP, assuming DBT_{sa} = 55°F (13°C): $Q_{wshp} = 1.085 \times V_{sa} \times (DBT_{zone} - DBT_{sa})$	1230 cfm (0.58 m^3/s)	830 cfm (0.39 m^3/s)	1300 cfm (0.61 m^3/s)	930 cfm (0.44 m^3/s)

In the “cold-air delivery” scenario, however, the same dedicated OA unit dehumidifies the outdoor air to the same 45°F (7.2°C) dew point, but then reheats it to only 55°F (13°C) so that it is delivered cold (not neutral). The conditioned outdoor air is still dry enough to offset the latent load in the classrooms, but because it is delivered at a dry-bulb temperature that is much cooler than the zone temperature—55°F (13°C) versus 74°F (23°C)—it offsets a significant portion of the sensible cooling load in the classrooms. This reduces the sensible load that must be offset by the local heat pumps, allowing them to be sized for less airflow and less cooling capacity than in a neutral-air system.

For Classroom 101, when the 450 cfm (0.21 m^3/s) of outdoor air is delivered at a cold dry-bulb temperature—55°F (13°C) in this example—the conditioned OA offsets 9,280 Btu/hr (2.7 kW) of the zone sensible cooling load. The local WSHP need only be sized to offset the remaining 20,520 Btu/hr (6.0 kW) of sensible load, which corresponds to a supply airflow of only 1000 cfm (0.47 m^3/s) (Figure 44 and Figure 45).

Figure 44. Conditioned OA delivered directly to each zone, like shown in Figure 38 (Classroom 101)

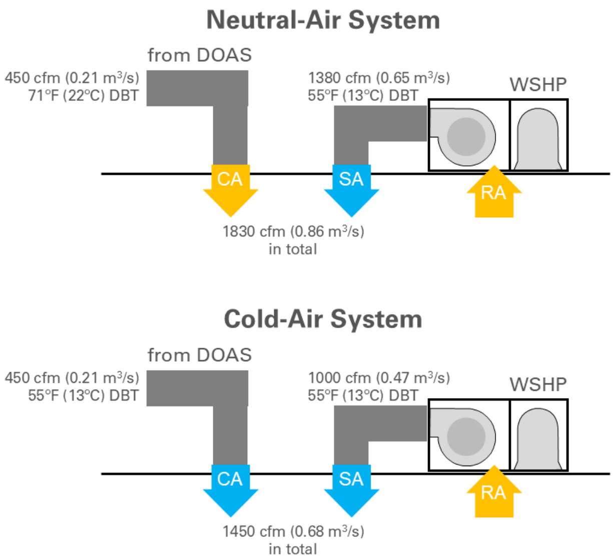
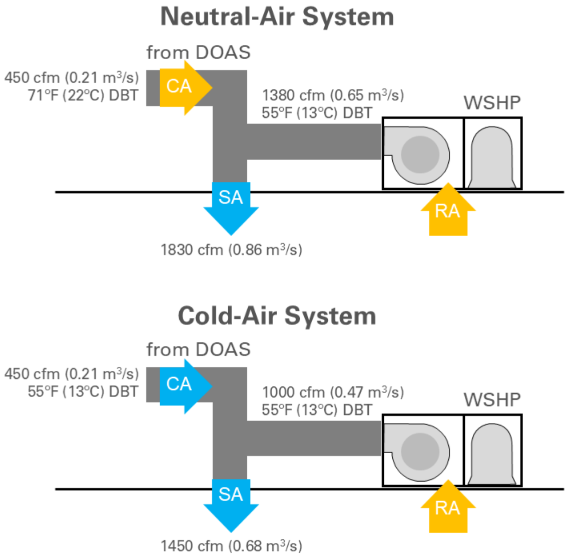


Figure 45. Conditioned OA delivered directly to the supply-side of each WSHP, like shown in Figure 41 (Classroom 101)



Section 6.5.2.6 of ASHRAE Standard 90.1 prohibits the DOAS from heating or reheating the conditioned outdoor air to any warmer than 60°F (16°C) when most of the zones require cooling:

“Units that provide ventilation air to multiple zones and operate in conjunction with zone heating and cooling systems shall not use heating or heat recovery to warm supply air above 60°F (16°C) when representative building loads or outdoor air temperature indicate that the majority of zones require cooling.”

Since both dedicated OA units dehumidify the same quantity of outdoor air to the same leaving-air dew point, the required capacity and airflow of the dedicated OA unit are the same whether the outdoor air is delivered neutral or cold. The impact on the local WSHPs, however, is significant (see Table 11). Compared to a neutral-air system, a dedicated OA system that delivers cold air directly to each zone (or to the supply-side of each WSHP):

- **Requires less overall cooling capacity**

  The required cooling/dehumidification capacity of the dedicated OA unit is the same for both configurations, but the required cooling capacity of each heat pump is less in a cold-air system than in a neutral-air system.

- **Requires less overall cooling energy for much of the year**

  By taking advantage of the sensible cooling already done by the dedicated OA unit, the cold-air system requires less cooling energy at each heat pump. The neutral-air system throws away this sensible cooling benefit by reheating the air to approximately zone temperature (see Figure 43, p. 64).

- **Requires less overall fan airflow and, therefore, less fan energy**

  The airflow delivered by the dedicated OA unit is the same for both configurations, but for those zones that require seasonal cooling and heating, the supply airflow delivered by the heat pump is less in a cold-air system than in a neutral-air system (Figure 44, and Figure 45). For zones that require year-round cooling, the WSHP may need to be sized based on the warmest temperature expected to be delivered by the dedicated OA unit.

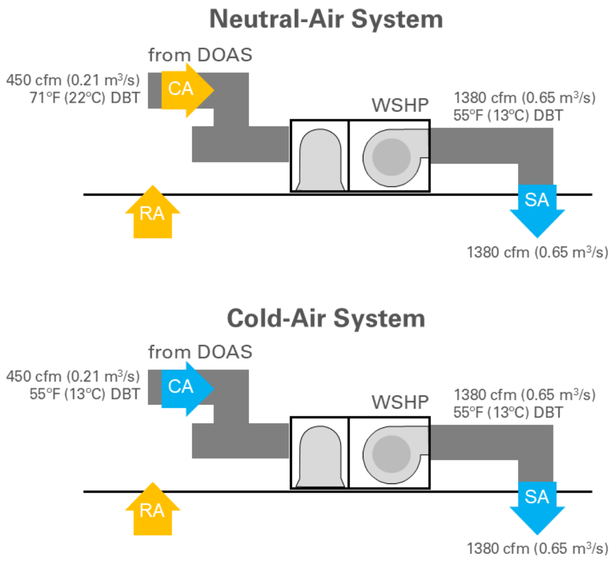
Less supply airflow and less cooling capacity allows for the selection of smaller heat pumps, which can lower the system installed cost and requires less space. Alternatively, selecting the same-sized WSHP (cabinet) and operating the fan at a lower speed can provide an acoustical benefit.

Finally, if the outdoor air is delivered directly to each zone (see Figure 38, p. 59), the fan inside the heat pump is no longer required to operate to ventilate the zone. This affords the opportunity to cycle the local fan on and off along with the compressor, reducing fan energy use. If the outdoor air is delivered to the supply-side of each heat pump (see Figure 41, p. 61), the fan inside the heat pump could cycle on and off if a pressure-independent VAV terminal is used to maintain required outdoor airflow.

What happens if conditioned outdoor air is delivered to the intake of each WSHP, rather than directly to each zone?

If the conditioned OA is delivered to the intake of each WSHP, it mixes with recirculated air from the zone before entering the heat pump. Because the conditioned outdoor air (CA) is not delivered directly to the zone, the sensible cooling load in the zone is unchanged, so supply airflow is unaffected (Figure 46) and the WSHP cannot be downsized. Also, since the local fan is tasked with delivering the outdoor air into the zone, it cannot cycle off without interrupting ventilation.

Figure 46. Conditioned OA delivered directly to the intake of each WSHP, like shown in Figure 40 (Classroom 101)



If the conditioned OA is delivered at a cold temperature, it lowers the enthalpy of the mixed air entering the WSHP, thereby reducing the cooling load that must be offset by the heat pump compressor.

However, the packaged nature of water-source heat pumps means that they are typically only available with pre-matched components (fans, compressors, heat exchangers, etc.). This limits their selection to a finite cfm/ton ($m^3/s/kW$) range of application.

When cold, conditioned outdoor air is delivered to the intake of a heat pump, supply airflow (cfm [m^3/s]) is unaffected, but the lowered mixed-air enthalpy reduces the required cooling capacity (tons [kW]). This raises the cfm/ton ($m^3/s/kW$) required of the heat pump, possibly above the maximum allowed for the equipment.

In this case, it may be necessary to reheat the dehumidified outdoor air to allow proper selection of the heat pumps—but may not require reheating the air all the way to neutral.

When should the conditioned OA be reheated?

For many applications, another approach to avoid overcooling is to implement demand-controlled ventilation (see "Demand-controlled ventilation," p. 195). This control strategy reduces the quantity of outdoor air delivered to a zone when there are fewer people in that zone. This often avoids overcooling altogether, and reduces the energy used by the DOAS to condition and deliver that air.

While the conditioned outdoor air should be delivered cold whenever possible, there are situations when the dedicated OA unit should reheat the dehumidified outdoor air.

- **To avoid overcooling at part-load conditions**

  As explained earlier, delivering the conditioned OA at a dry-bulb temperature colder than the zone temperature offsets part of the sensible cooling load in the zone. As the zone sensible cooling load decreases—due to changes in outdoor conditions, solar heat gain, and/or internal loads—it is possible that the cold, conditioned OA may provide more sensible cooling than the zone requires. As a result, the temperature in the zone begins to drop and the WSHP eventually switches to operate in the heating mode.

  While this may appear strange to the building operator, if only a few heat pumps are operating in the heating mode, this situation likely improves system efficiency. As those heat pumps operating in the heating mode extract heat from the water loop, they reduce the amount of heat that must be rejected by the cooling tower. In addition, the rest of the zones (in which the heat pumps are still operating in the cooling mode) continue to benefit from the sensible cooling provided by the cold, conditioned outdoor air.

  However, if enough heat pumps are operating in the heating mode that the temperature of the water loop approaches the lower setpoint—60°F (16°C), for example—and the source of reheat energy in the dedicated OA unit is recovered from another part of the system (hot gas reheat or an air-to-air heat exchanger, for example), it may be more efficient to reheat the dehumidified outdoor air to avoid needing to activate the boiler.

  During cold weather, it may be desirable to heat the outdoor air to a temperature near the desired zone temperature before delivering it directly to the zones.

- **In applications where zone sensible cooling loads differ greatly at any given time**

  In hotel guest rooms or dormitories, the sensible cooling loads can be drastically different from zone to zone. The result is that, if the conditioned OA is delivered cold, it may be more likely that some zones will experience overcooling. For these applications, it may be simpler to deliver the conditioned OA at a neutral dry-bulb temperature because the benefit of delivering the air cold occurs less frequently.

  In classrooms or offices, however, sensible cooling loads in the zones are relatively high during daytime hours. In fact, for some climates, classrooms may never reach the point when overcooling occurs during occupied hours, especially if demand-controlled ventilation is used to reduce outdoor airflow when zone population decreases. These applications are typically well-suited for delivering the conditioned OA at a cold temperature.

- **In applications that require lower-than-normal dew points**

  If an application has very high indoor latent loads or requires a lower-than-normal dew point, the outdoor air may need to be dehumidified to a very low dew point. In this case, the corresponding dry-bulb temperature of the air leaving the cooling coil may be colder than the HVAC design engineer is willing to discharge directly into an occupied zone—below 45°F (7°C), for example. In this case, the dehumidified OA could be reheated to a more traditional supply-air temperature—55°F (13°C), for example—but still not reheated all the way to neutral.

- **To avoid condensation when conditioned OA is delivered to the ceiling plenum**

  In some applications, the dedicated OA system delivers the conditioned outdoor air (CA) to the ceiling plenum (or closet), near the intake of each WSHP (see Figure 42, p. 62). The outdoor air mixes with recirculated air (RA) in the plenum before being drawn in through the WSHP intake.

  In this configuration, the dedicated OA unit should reheat the dehumidified OA to a dry-bulb temperature that is above the expected dew-point temperature of the air within the ceiling plenum. If cold air is dumped into the ceiling plenum, it could cool surfaces (structural beams, electrical conduit, ceiling framework). At night, when the dedicated OA unit is off, wind or operating exhaust fans may cause humid outdoor air to leak into the plenum, which may lead to condensation on these cold surfaces.

Exhaust-air energy recovery

For more information on exhaust-air energy recovery, including application and control in a dedicated outdoor-air system, refer to the Trane application manual, *Air-to-Air Energy Recovery in HVAC Systems* (SYS-APM003*-EN).

A dedicated outdoor-air system often makes it more feasible to implement exhaust-air energy recovery, if exhaust air from the building can be routed back to the dedicated OA unit. The energy-recovery device transfers sensible heat, or sensible heat and water vapor, between the outdoor air (OA) and exhaust air (EA) streams.

As an example, Figure 47 shows a total-energy wheel used to precondition the entering outdoor air. During the cooling season, this desiccant-coated wheel revolves between the outdoor and exhaust air streams, removing both sensible heat and water vapor from the entering outdoor air and rejecting it to the exhaust air. During the heating season, the wheel recovers both sensible heat and water vapor from the exhaust air, and transfers it to the outdoor air being brought into the building for ventilation.

Figure 47. Total-energy wheel inside a dedicated outdoor-air unit

Sensible-energy recovery devices transfer only sensible heat. Common examples include coil loops, fixed-plate heat exchangers, heat pipes, and sensible-energy rotary heat exchangers (also known as sensible-energy wheels or heat wheels). *Total-energy recovery* systems not only transfer sensible heat, but also water vapor (or latent heat). Common examples include total-energy rotary heat exchangers (also known as total-energy wheels or enthalpy wheels) and membrane exchangers.

In many climates and building types, exhaust-air energy recovery is an effective means of reducing the energy required to cool, dehumidify, heat, or humidify the entering outdoor air. It also reduces the required cooling and heating capacity of the dedicated OA unit.

However, adding an air-to-air energy-recovery device increases the static pressure drop in both the outdoor- and exhaust-air paths, which impacts fan energy use. The energy saved by preconditioning the outdoor air must exceed any increase in fan energy use.

In addition, routing most of the exhaust air back to the energy-recovery device often requires the installation of additional ductwork.

Note: An advantage of a coil loop is that it can be used to transfer heat between air streams that are physically separated by some distance, making it particularly advantageous in retrofit situations. Also, a networked coil loop can be used to recover heat from multiple, separate exhaust air streams (using multiple exhaust-side coils).

When using air-to-air energy recovery to precondition the entering outdoor air, consider the following general recommendations:

- *Sensible- or total-energy recovery?*

  In most climates, a total-energy recovery device allows for greater downsizing of the cooling and heating equipment (and usually provides the best payback) because it recovers both sensible heat and water vapor (latent heat). The most notable exception is in warm, dry climates, where it is often unnecessary to mechanically dehumidify the outdoor air. In this case, a sensible-energy recovery device likely provides the best value.

  There is a common misperception that only hot, humid climates justify the need for total-energy recovery. When compared with sensible-energy recovery, however, total-energy recovery devices can provide advantages in climates where heating operation prevails:

- Frost forms on a total-energy recovery device at a much colder outdoor temperature than it does on a sensible-energy recovery device. This allows total-energy recovery to recover more heat during cold weather and lessens (and may even eliminate) the need for frost prevention.
- Total-energy recovery devices generally have a higher effectiveness than most sensible-energy recovery devices, so they save more heating energy and may permit greater downsizing of the heating equipment.
- Water vapor transferred by a total-energy recovery device humidifies the entering outdoor air during the heating season, which helps keep the space from becoming too dry. “Free” humidification also reduces the energy used by the mechanical humidification system (if installed) and may allow this equipment to be downsized.
- Most heating climates also include a cooling season. Applying a total-energy recovery device enables a larger reduction in cooling capacity than a sensible-energy recovery device, which can offset some of the first-cost premium for energy recovery.

- *Strive for balanced airflows*

  Duct as much of the exhaust airflow to the energy-recovery device as possible. The less disparity between the outdoor and exhaust airflows, the more energy can be recovered.

  If demand-controlled ventilation (DCV) is being used, the amount of outdoor air being brought into the building is reduced for many hours during the year. The energy-recovery device provides less benefit because there is less outdoor air to precondition and, with less air entering the building, less air is exhausted. Air exhausted by local exhaust fans and exfiltration due to building pressurization are relatively constant, so when DCV reduces intake airflow, less centralized exhaust air is available for energy recovery.

- *Provide a means to properly control the device at part load*

  During mild weather, the energy-recovery device should be shut off to avoid transferring unwanted heat from the exhaust air to the outdoor air. For example, when the enthalpy of the outdoor air is less than the enthalpy of the exhaust air, a total-energy wheel should be turned off to prevent increasing the cooling energy use.

  In addition, when it is cool outside and the dedicated OA system needs to add heat to the entering air, many dedicated OA systems will require a means to modulate the capacity of the energy-recovery to avoid overheating the air. Unnecessarily operating the device at full capacity may require recooling and wastes energy.

  The method used for capacity control depends on the device. Coil loops either vary the speed of the circulation pump or use a three-way mixing valve to bypass some of the fluid around the exhaust-side coil. Fixed-plate and membrane exchangers often use a modulating damper to bypass some of the exhaust air. Heat pipes may use bypass dampers or a series of solenoid valves to shut off refrigerant flow for individual heat pipes. Wheels use a modulating damper to bypass air around the exhaust-side of the wheel or vary the rotational speed of the wheel.

For more information on methods used for capacity control and frost prevention with various air-to-air energy recovery devices, refer to the Trane application manual, *Air-to-Air Energy Recovery in HVAC Systems* (SYS-APM003*-EN).

- *Provide a method for frost prevention in cold climates*

  Any exhaust-air energy-recovery device that preconditions outdoor air is subject to frost buildup during very cold weather. If the surface temperature of the device falls below the dew point of the exhaust air, water vapor will condense on the exhaust-side of the device. If the exhaust-side surface temperature falls below 32°F (0°C), this water freezes, eventually blocking airflow. The method used for frost prevention depends on the device. Typically, one of the following two approaches is used: 1) reduce the heat-transfer capacity of the energy-recovery device (which results in a warmer exhaust-side surface temperature); or 2) preheat either the outdoor or exhaust air before it enters the device (which also raises the surface temperature of the device to prevent frosting).

  For most applications and most climates, reducing heat-transfer capacity (by modulating an OA bypass damper, for example) is sufficient for frost prevention. However, for applications with extremely cold OA and/or higher indoor humidity levels during cold weather, preheating may be desirable.

- *Decide what amount of cross-leakage is acceptable*

  Many types of air-to-air energy-recovery devices permit some degree of cross-leakage. Through fan configuration and properly adjusted seals, the amount of leakage is usually less than 5 percent (even for wheels) in most applications.

Dedicated OA equipment types

Depending on the climate, the dedicated OA equipment may be used to cool, dehumidify, heat, and/or humidify the entering outdoor air. This approach allows the heat pumps to handle only the zone cooling and heating loads, not the ventilation load.

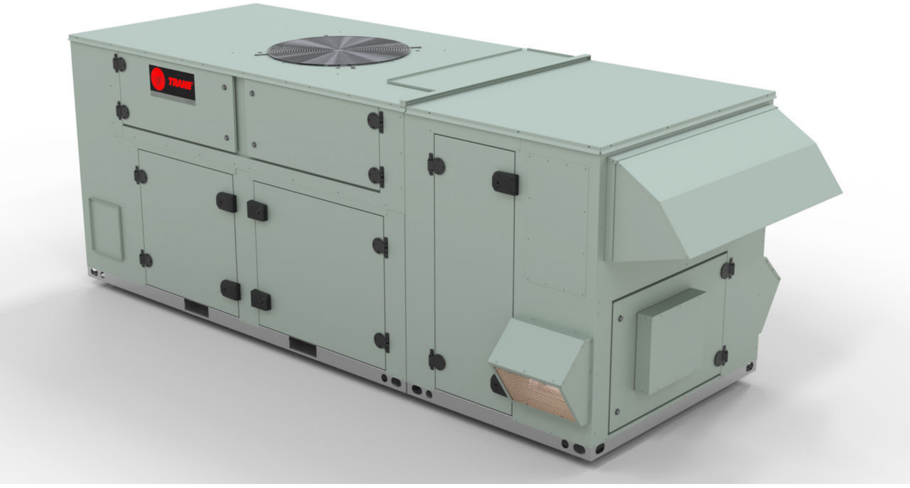
While there are many types and combinations of equipment that can be used, following are a few examples of dedicated OA equipment types commonly used with WSHP systems.

Standalone air-cooled DX unit (packaged or split)

One of the most common types of dedicated OA equipment is a standalone, air-cooled, direct-expansion (DX) unit. This might be a packaged unit or a split system (comprised of two separate sections connected by field-installed refrigerant piping).

A packaged DX unit is typically installed on the roof of the building, and contains a fan, filter, a gas-fired burner or heating coil, and all the components of a DX refrigeration system—an evaporator (cooling) coil, one or more compressors, an air-cooled condenser complete with propeller-type fans, and expansion devices (Figure 48). In addition, it may contain an air-to-air heat exchanger for exhaust-air energy recovery (see “Exhaust-air energy recovery,” p. 71).

Figure 48. Packaged, air-cooled DX dedicated OA unit



A split DX system is comprised of a condensing unit—which contains one or more compressors and an air-cooled condenser with propeller-type fans—and an air-handling unit (AHU)—which contains the evaporator coil and expansion devices, along with a fan, filter, heating coil or gas-fired burner, and possibly an air-to-air heat exchanger. The condensing unit is typically installed on the roof or on the ground next to the building, while the AHU can be installed indoors or outdoors. The two components are connected by field-installed refrigerant piping to complete the refrigeration circuit.

A packaged unit typically offers less flexibility in selection and fewer options, while a split DX system increases flexibility since the AHU typically has more options for fans, air cleaning devices, air-to-air energy recovery devices, and sound attenuation.

The primary advantages of using air-cooled DX equipment for the dedicated OA unit are lower installed cost (typically) and simplicity in design and installation, especially when a packaged unit is used. The selection and performance of the standalone unit is independent of the water-source heat pumps and other components of the water distribution loop.

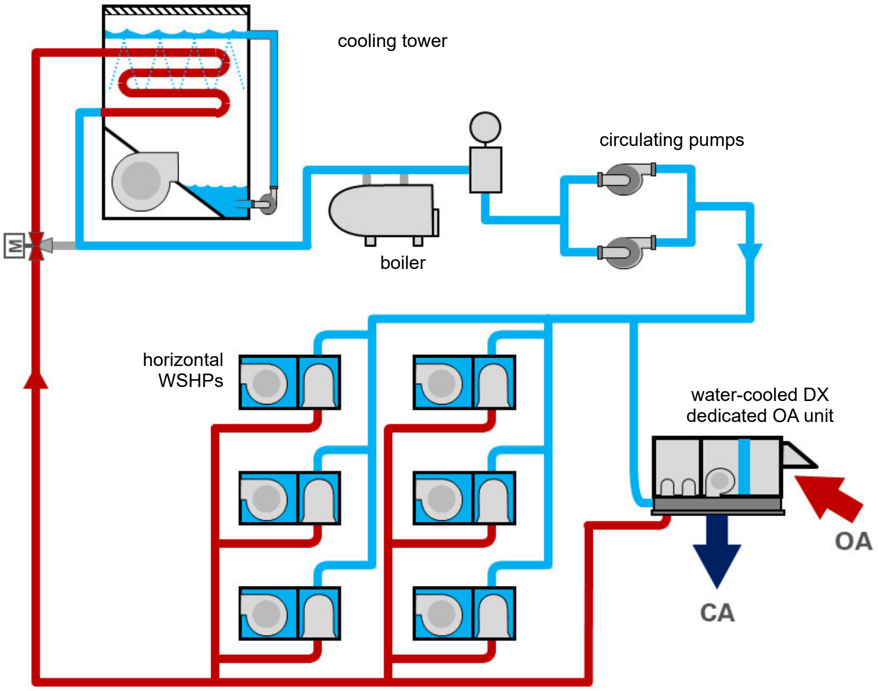
Additionally, the dedicated OA unit can be operated (to control indoor humidity after hours, for example) without requiring the pumps, and possibly the boiler or cooling tower, to operate.

The drawbacks of this approach include limited flexibility (especially when a packaged unit is used), lower efficiency, and installation of some or all of the equipment outside. Regarding efficiency, air-cooled condensing is typically not as efficient as water-cooled condensing. And, while an air-source heat pump could be used in this application, heat pump operation is typically less effective when it is very cold outside.

Water-cooled DX dedicated OA unit connected to the water loop

Because of the presence of the water distribution loop, some WSHP systems use a water-cooled DX unit for the dedicated OA equipment. Instead of an air-cooled condenser, this type of equipment uses a water-cooled condenser, allowing it to be connected to the same water distribution loop that serves the WSHPs (Figure 49).

Figure 49. Packaged, water-cooled DX dedicated OA unit connected to the loop



A water-cooled DX dedicated OA unit could be installed outdoors, or even indoors since it does not require outdoor air for condensing.

The water-cooled DX refrigeration circuit is reversible, allowing the unit to operate in either the cooling or heating (heat pump) mode. This allows for improved heating efficiency due to the higher COP of the heat pump. Note, however, that the dedicated OA unit will likely be operating in the heating mode whenever the outdoor temperature is below 50°F or 60°F (10°C or 16°C), which will cause the loop temperature to decrease, and may require the boiler to operate more often than in a system where the dedicated OA unit is not connected to the loop. However, even though the boiler will likely need to operate more often, the “overall” heating COP will still likely be higher than using a gas-fired burner or electric heater in the dedicated OA unit.

In a “boiler-less” system [see “Electric resistance heat for a “boiler-less” system,” p. 56], a cooling-only DX unit—containing a separate gas-fired burner or electric heater—would likely be preferred, to prevent overcooling the loop during cool weather, and increasing the need to use the electric resistance heat in the individual heat pumps.

The primary advantage of using a water-cooled DX dedicated OA unit that is connected to the loop is improved efficiency. Water-cooled condensing is typically more efficient than air-cooled condensing equipment, and for cooling-dominated buildings, the excess heat rejected to the loop by the heat pumps can be extracted by the dedicated OA unit for use in heating the entering outdoor air during cool weather.

In addition, the dedicated OA unit and individual heat pumps share the same cooling tower for heat rejection, avoiding the space needed outside to install two separate pieces of heat rejection equipment.

However, selection of a loop-connected unit is more complicated, since its performance is inter-related to the performance and selection of the water-source heat pumps and other components of the water distribution loop. Both the cooling tower and boiler will typically need to be larger than if the dedicated OA unit was standalone.

Air-handling unit connected to a water chiller or water-to-water heat pump (standalone or connected to the water loop)

Probably the most flexible type of dedicated OA equipment is an air-handling unit connected to a water chiller or water-to-water heat pump. The water chiller may be a standalone air-cooled chiller or a water-cooled chiller that is connected to the water distribution loop.

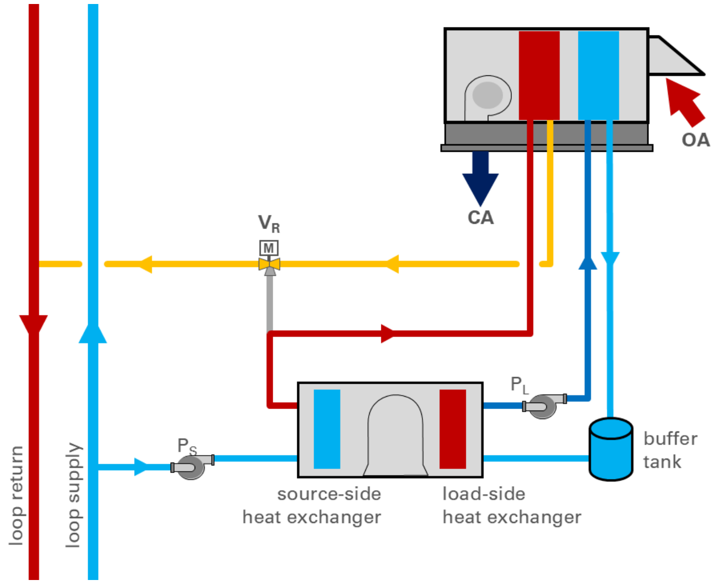
One advantage of using a standalone chiller is that the selection and performance of the chiller is independent of the water-source heat pumps and other components of the water distribution loop. Also, the dedicated OA system can be operated—to control indoor humidity after hours, for example—without requiring the water distribution loop (pumps and possibly the boiler or cooling tower) to operate.

For more information on piping a water-cooled chiller so that it can be used for either cooling or heating, refer to Trane application manual, *Central Geothermal Systems* (SYS-APM009*-EN).

Alternatively, a water-cooled chiller or water-to-water heat pump can be connected to the water distribution loop. The water-cooled chiller may be used for cooling only or it might be piped into the water distribution loop to allow it to be used for either cooling or heating.

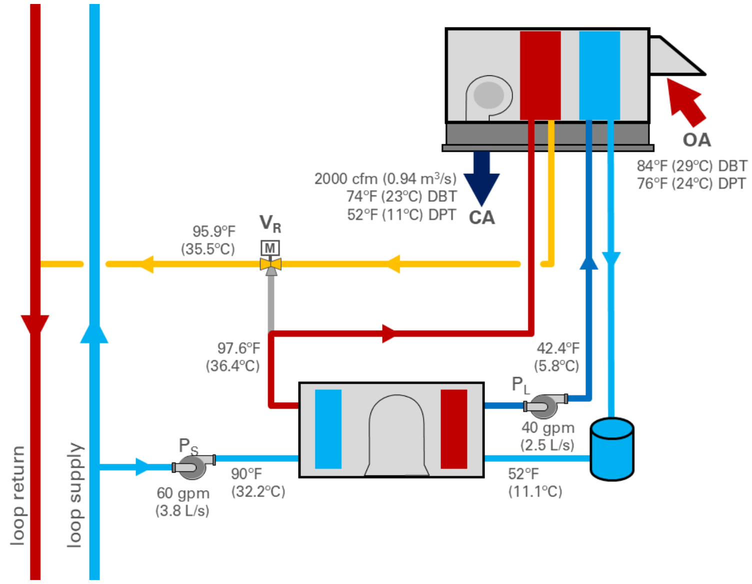
A water-to-water heat pump is a small, reversible-cycle water chiller that contains one or more compressors, a thermal expansion valve, a reversing valve, and two refrigerant-to-water heat exchangers (see Figure 20, p. 28). Figure 50 depicts a water-to-water heat pump, which is connected to the water distribution loop, serving an air-handling unit that conditions all of the outdoor air required for ventilation. This system uses a dedicated water-circulating pump (P_S) to draw water from the main loop.

Figure 50. Water-to-water heat pump connected to loop



When it is hot or humid outside, cold water leaving the “load-side” refrigerant-to-water heat exchanger is pumped through the cooling coil in the AHU to cool and/or dehumidify the entering outdoor air (Figure 51 and Table 12, p. 78). Water from the loop flows through the “source-side” heat exchanger to extract heat rejected from the heat pump.

Figure 51. Operation of water-to-water heat pump in cooling mode



If desired, the warm water leaving the “source-side” heat exchanger—at 97.6°F (36.4°C), in this example—can be diverted through a reheat coil in the AHU to reheat the dehumidified air. After leaving the reheat coil, the water is returned to the main loop.

When it is cold outside, the reversing valve changes the operation of the refrigeration circuit of the water-to-water heat pump to the heating mode. Hot water leaving the “load-side” heat exchanger is pumped through the dual-purpose coil in the AHU to heat the entering outdoor air (Figure 52 and Table 12). The heat pump extracts heat from the loop water (thereby cooling the water) flowing through the “source-side” heat exchanger.

Figure 52. Operation of water-to-water heat pump in heating mode

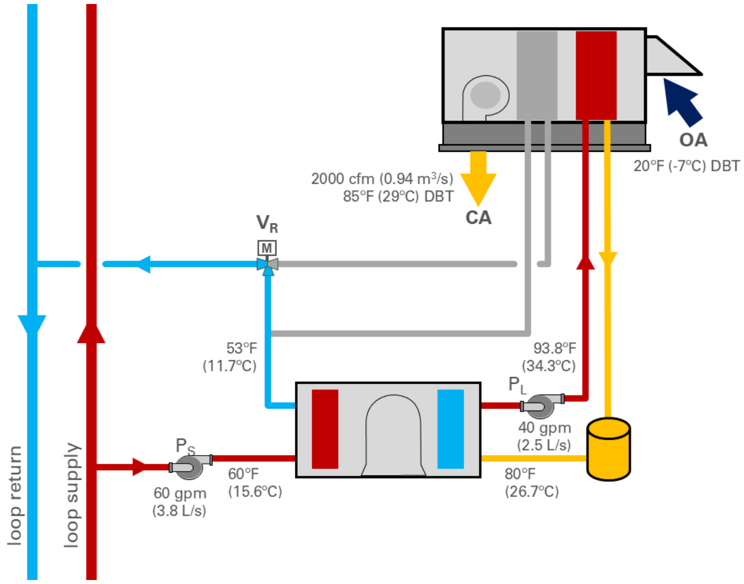


Table 12. Water-to-water heat pump operating modes

Operating mode	Heat pump operation	Source-side pump (P_S)	Load-side pump (P_L)	Reheat valve (V_R)
cooling	cooling mode	on	on	modulate to reheat as needed
heating	heating mode	on	on	full bypass

The primary advantages of using a water-to-water heat pump connected to the loop are piping simplicity and the fact that the cooling/heating equipment can be distributed throughout the building, often located very close to the dedicated OA unit that it is connected to.

With this equipment configuration, the air-handling unit may be installed indoors or outdoors, and contains a fan, filter, chilled-water cooling coil, heating coil or gas-fired burner, and possibly one or more air-to-air heat exchangers. An air-handling unit typically provides more flexibility and has more options—for fans, air cleaning devices, air-to-air energy recovery devices, and sound attenuation, for example—than packaged equipment.

For example, the AHU depicted in Figure 53 includes a total-energy wheel to precondition the entering outdoor air (see “Exhaust-air energy recovery,” p. 71) and a fixed-plate heat exchanger to recover heat for reheating the dehumidified air, when necessary.

Loop Volume and Buffer Tanks

An important characteristic of a hydronic loop is to ensure sufficient volume in the loop such that temperature control is stable enough to maintain the loop temperature within an acceptable range. When a loop has more volume, any change in supply temperature (due to compressors cycling, for instance) has less of an impact on the overall loop temperature. Consult the product manufacturer for recommendations on minimum loop volume.

Minimum loop volume is commonly expressed as "minimum loop time" (in minutes)—minimum total volume of fluid in the loop, in gallons (Liters), is equal to this minimum loop time (in minutes) multiplied by the design fluid flow rate, in gallons per minute (Liters per minute). For small water-to-water heat pumps, general guidance is to ensure a minimum loop time of 5 minutes, since the controllers typically include a 5-minute timer between compressor starts and stops. A larger loop volume would result in even more-stable temperature control and less compressor cycling.

If multiple water-to-water heat pumps are operating in parallel, use the sum of the design flow rates to calculate the minimum loop volume. But do not include redundant heat pumps that will never be operating at the same time.

If loop volume is insufficient, a **buffer tank** can be added to increase the loop volume. This tank is typically installed in the return piping for the water-to-water heat pump (as shown in Figure 50). This buffer tank should be designed to provide adequate mixing of the incoming fluid with the fluid inside the tank; accomplished by forcing the fluid to make several turns as it passes though the tank.

Figure 53. Dedicated OA unit with total-energy recovery and a fixed-plate HX

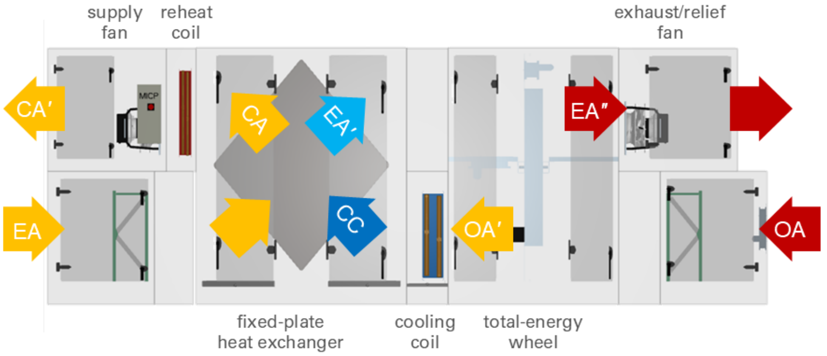
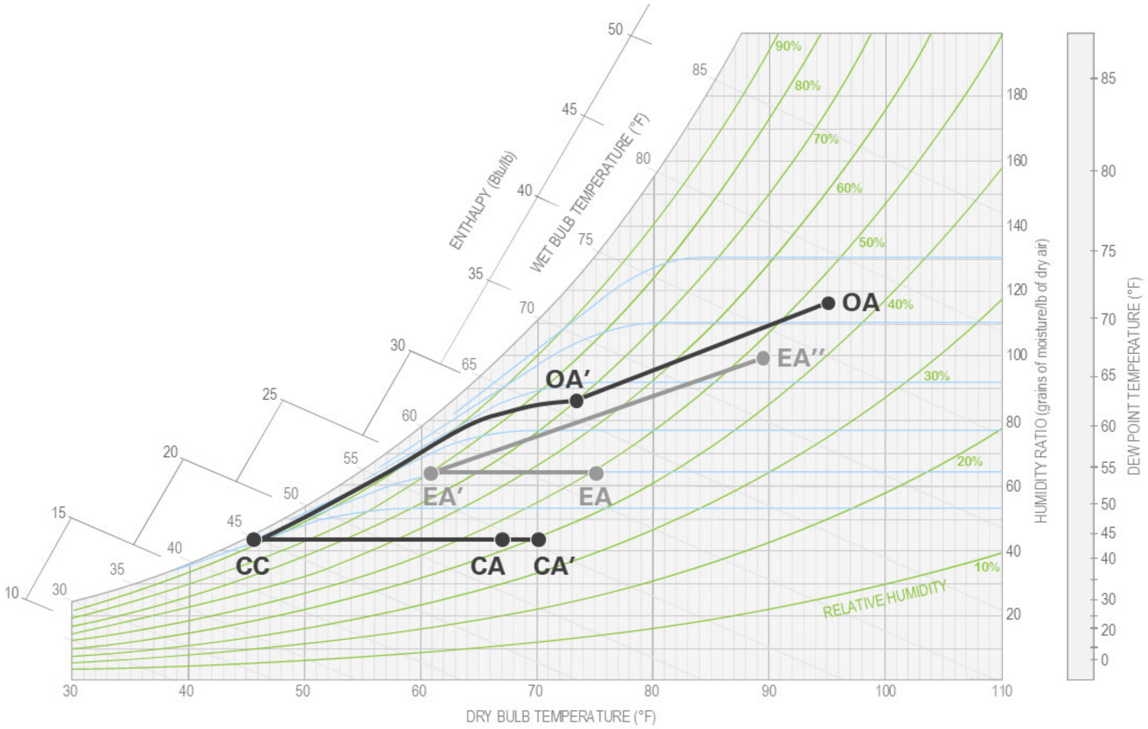


Figure 54 shows example performance of this AHU configuration during the cooling season. The total-energy wheel cools and dehumidifies the entering outdoor air—to 73°F dry bulb and 63°F dew point (23°C DBT and 17°C DPT)—by transferring both sensible heat and water vapor to the exhaust air stream. The cooling coil then dehumidifies the air to the desired leaving-air dew point of 45°F (7.2°C). When necessary, the fixed-plate heat exchanger transfers sensible heat recovered from the exhaust air stream to reheat the dehumidified air to 70°F dry bulb and 45°F dew point (21°C DBT and 7.2°C DPT).

Figure 54. Example performance of a dedicated OA unit with total-energy recovery and a fixed-plate HX



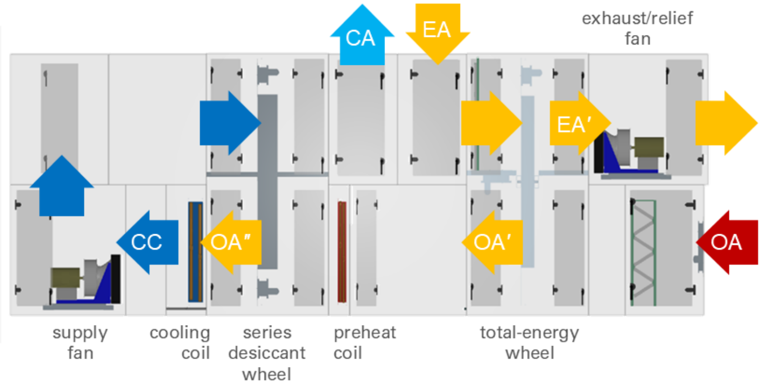
By using heat recovered from the exhaust air stream, the fixed-plate heat exchanger pre-cools the air entering the exhaust-side of the wheel (EA to EA').

For more information on using a series desiccant wheel in a dedicated OA system, refer to the Trane *Engineers Newsletter*, “Advances in Desiccant-Based Dehumidification” (ADM-APN016-EN), and the Trane engineering bulletin, “Trane CDQ™ Desiccant Dehumidification” (CLCH-PRB020-EN).

This increases the heat-recovery capacity of the wheel, resulting in more pre-cooling of the entering outdoor air (OA to OA’) and less cooling energy required.

As another example, the air-handling unit depicted in Figure 55 includes a total-energy wheel and a series desiccant wheel used for improved dehumidification.

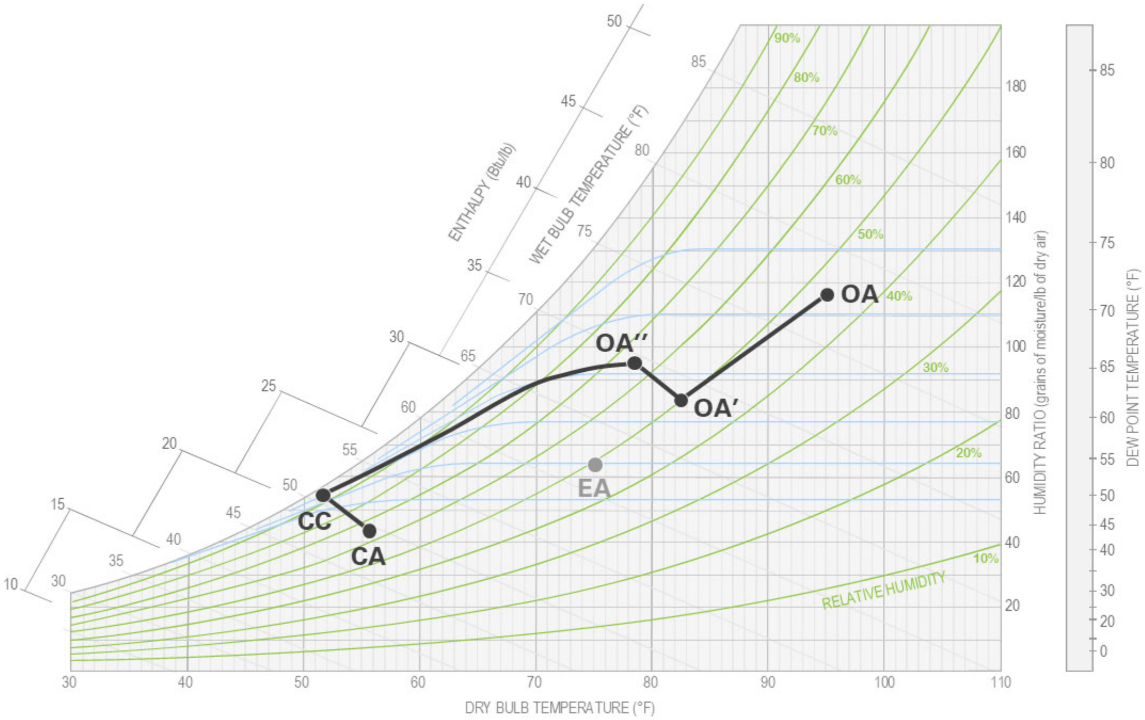
Figure 55. Dual-wheel dedicated OA unit



In this series configuration, the desiccant dehumidification wheel adsorbs water vapor from the process air downstream of the cooling coil and then releases the collected moisture upstream of that coil, enabling the AHU to deliver drier supply air (at a lower dew point) without lowering the coil temperature. In addition, the moisture transfer occurs within a single air stream; a separate, regeneration air stream is not needed.

Figure 56 shows the performance of this AHU configuration. Air leaves the cooling coil (CC) at a dry-bulb temperature of 52°F (11°C) and a dew point of 51°F (10.7°C). The series desiccant wheel adsorbs water vapor, drying the conditioned outdoor air (CA) to a dew point of 45°F (7.2°C). Sensible heat added by the adsorption process raises the dry-bulb temperature of this air to about 56°F (13°C). The wheel rotates into the air upstream of the cooling coil (OA’), where water vapor released from the wheel passes into the air (OA”) and then condenses on the cold coil surface. The series desiccant wheel allows the AHU to deliver drier air—at 45°F (7.2°C) dew point, in this example—without requiring a lower leaving-coil temperature.

Figure 56. Example performance of a series desiccant wheel in a dedicated OA unit



When a series desiccant wheel is used in a dedicated OA unit, it may be necessary to preheat the entering outdoor air (OA) when the relative humidity is high (on a mild rainy day, for example). Using a preheat coil to raise the dry-bulb temperature slightly—5°F to 20°F (3°C to 11°C), for example—lowers the relative humidity of the air. Lowering the relative humidity of the air entering the regeneration (upstream) side of the wheel allows the desiccant to reject more water vapor to the regeneration air, thus enabling it to adsorb more water vapor from the process air downstream of the cooling coil.

Typically, the amount of heat added by the preheat coil is small and it may be required for only a small number of hours throughout the year. Therefore, it may be practical to recover the needed heat from the condenser of a water chiller. A small, inexpensive electric heater is another option.

Alternatively, a total-energy wheel can be added to the system (Figure 55 and Figure 56). When high RH conditions occur, the total-energy wheel will transfer water vapor from the entering outdoor air (OA) to the exhaust air (EA), thus lowering the relative humidity of the air before it enters the regeneration side of the series desiccant wheel (OA'). In such cases, adding a total-energy wheel reduces (and often eliminates) the need to add regenerative preheat. Of course, this requires exhaust air to be ducted back to the dedicated OA unit.

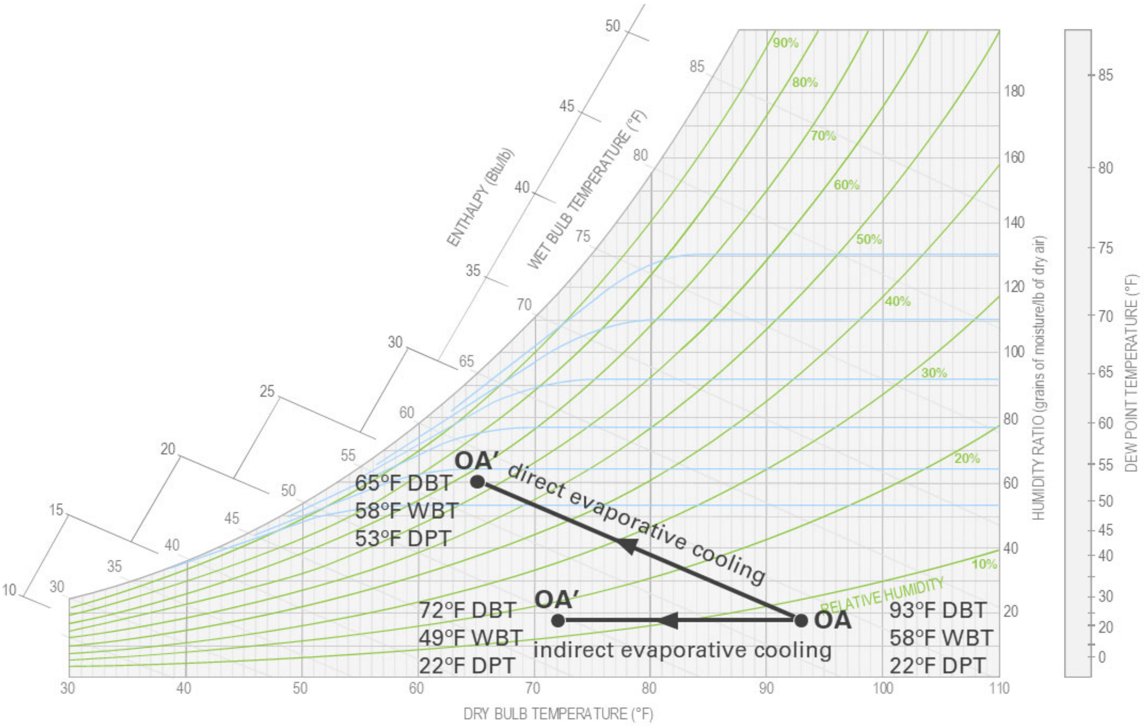
Evaluative cooling

For more information on evaporative cooling, refer to Chapter 41, “Evaporative Air-Cooling Equipment,” in the 2024 *ASHRAE Handbook—HVAC Systems and Equipment* (www.ashrae.org).

Using an evaporative process to cool the air can reduce the energy used by mechanical cooling equipment. Any cooling energy saved is offset somewhat by the increased fan energy use, as the evaporative media increases the airside pressure drop that the fan must overcome.

Direct evaporative cooling introduces water directly into the air stream, usually with a spray or wetted media. The water evaporates as it extracts heat from the passing air stream, which lowers the dry-bulb temperature of the air. Evaporation of the water, however, also raises the dew point of the air (Figure 57).

Figure 57. Direct versus indirect evaporative cooling



The leaving-air temperature depends on how much the dry-bulb temperature of the entering air exceeds its wet-bulb temperature. For example, if the condition of the entering outdoor air (OA) is 93°F dry bulb and 58°F wet bulb (34°C DBT, 14°C WBT), and the direct evaporative process is 80 percent effective, the condition of the leaving air (OA’) will be 65°F DBT and 58°F WBT (18°C DBT, 14°C WBT).

$$\text{DBT}_{\text{leaving}} = \text{DBT}_{\text{entering}} - \text{effectiveness} \times (\text{DBT}_{\text{entering}} - \text{WBT}_{\text{entering}})$$

$$\text{DBT}_{\text{leaving}} = 93°\text{F} - 0.80 \times (93°\text{F} - 58°\text{F}) = 65°\text{F}$$

$$(\text{DBT}_{\text{leaving}} = 34°\text{C} - 0.80 \times [34°\text{C} - 14°\text{C}] = 18°\text{C})$$

For a dedicated OA system, direct evaporative cooling is most applicable in dry climates, where the outdoor dew point is well below the desired dew point indoors. A conventional cooling coil may be required to supplement the evaporative cooling process during times of the year when the outdoor dew point is higher.

Indirect evaporative cooling typically uses an evaporative cooling tower to cool water, and then pumps this water through a conventional cooling coil to cool the air. This approach does not involve the evaporation of water into the air stream, so it does not increase the dew point of the air (Figure 57).

In some applications, indirect evaporative cooling is implemented using a standalone cooling tower (or similar device) and a separate coil located upstream of the conventional cooling coil. However, in a water-source heat pump system, because a water distribution system is already part of the system, another approach may be to route the cool condenser water through a separate coil to pre-cool the entering outdoor air before it passes through the conventional cooling coil. This configuration is similar to a waterside economizer used in a water-source heat pump (see Figure 109, p. 170).

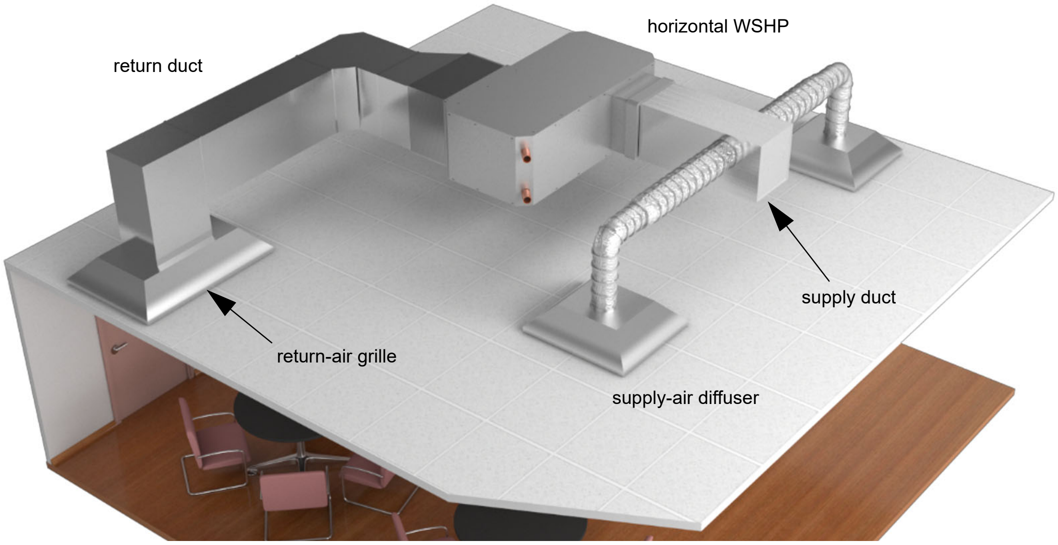
Evaporative cooling requires careful attention to water treatment, periodic cleaning, and routine maintenance to ensure safe and efficient operation. Finally, it consumes water, which may be in limited supply in the arid climates where evaporative cooling provides the greatest energy-saving benefit.

Air Distribution

For water-source heat pumps that are installed within, or directly adjacent to, the occupied zone (such as console and vertical-stack models), air is typically supplied directly to, and returned directly from, the zone without the use of ductwork.

However, for heat pumps that are installed outside of the occupied zone (such as horizontal, vertical, and rooftop models), a supply duct system is typically used to transport air from the WSHP to supply-air diffusers for delivery to the zone. Figure 58 shows an example of supply- and return-air ductwork connected to a horizontal water-source heat pump that is installed in the ceiling plenum.

Figure 58. Air distribution components of a WSHP system



Supply duct system

For more information on best practices for the design and layout of duct systems, refer to the Sheet Metal and Air Conditioning Contractors National Association (SMACNA) manual, *HVAC Systems Duct Design*.

Supply ductwork is typically routed through the ceiling plenum above the occupied zone. A successful design of the supply duct system achieves the following:

- Supplies the required quantity of air to each supply-air diffuser without excessive noise
- Minimizes the static pressure and associated power requirements of the fan
- Minimizes the installed cost without great sacrifices in system efficiency
- Accommodates space limitations without excessive pressure drop

Other publications contain more complete details related to duct design, but following are a few general recommendations:

- *Keep the duct layout as simple and symmetrical as possible.*

  Use low-pressure-drop duct fittings and follow the best practices published by the Sheet Metal and Air Conditioning Contractors National Association (SMACNA) for designing and installing duct systems.
- *Use at least three diameters of straight duct for the first section downstream of the discharge from the WSHP.*

  Satisfactory fan performance and distribution of air throughout the system requires unrestricted and relatively uniform airflow from the discharge of the WSHP. This first section of supply ductwork should be straight for at least three duct diameters to allow a uniform velocity profile to develop (conversion of fan energy from velocity pressure to static pressure).

  If site conditions dictate that an elbow be installed near the WSHP discharge, the loss of fan capacity and static pressure can be somewhat minimized by installing turning vanes within the elbow.
- *When possible, locate ceiling-mounted WSHPs above a hallway or other unoccupied area of the building.*

  This will typically simplify installation and maintenance, and help minimize sound radiated to the occupied space.
- *Limit the use of flexible ductwork.*

  Flexible ductwork is sometimes used to connect the sheet metal duct to the inlet of each supply-air diffuser, allowing for some flexibility when the ductwork and ceiling are not installed at the same time. However, it is best to limit the use of flexible duct to no longer than 6-ft (2-m) sections to minimize the turbulence and high pressure drop associated with flexible duct. If the overall length of duct between the WSHP and diffuser is greater than this, sheet metal should be used for the initial sections of ductwork, while limiting the use of flexible duct to no more than the last 6 ft (2 m) needed to connect to the supply-air diffusers.
- *Add a balancing damper in the runout duct for each supply-air diffuser.*

  This allows adjustment to deliver the desired airflow to different spaces, or portions of a space, served by the WSHP. Many types of diffusers are available with an integral balancing damper to simplify installation.

Supply-air diffusers

For more information on space air diffusion, refer to:

- 2025 *ASHRAE Handbook—Fundamentals*, Chapter 20 (www.ashrae.org)
- ASHRAE's *Designer's Guide to Ceiling-Based Air Diffusion* (www.ashrae.org)

Proper selection and placement of supply-air diffusers generates air movement throughout the occupied zone, eliminating areas of stagnant and stratified air, increasing air circulation, and preventing cold air "dumping."

Other publications contain more complete details related to sizing and locating supply-air diffusers, but following are a few general recommendations:

- *Select and lay out supply-air diffusers to achieve at least an 80 percent ADPI at cooling design airflow.*

  Air Diffusion Performance Index (ADPI) is a measure of diffuser performance when delivering cool air to the zone.
- *When ceiling-mounted diffusers will deliver warm air to the zone, try to limit the difference between the supply-air temperature and the zone temperature to not more than 15°F (8°C).*

  Limiting the supply-air temperature during heating avoids excessive temperature stratification when supplying warm air from overhead diffusers. This may also increase the zone air-distribution effectiveness used to determine outdoor airflow required for ventilation (see "Impact of zone air-distribution effectiveness," p. 91).
- *In perimeter zones with high heat loss through the building envelope, position diffusers to "blanket" the perimeter wall or window area.*

  This helps prevent downdraft problems that can occur when large volumes of heated air are distributed through ceiling-mounted diffusers.

Return-air path

When a WSHP is installed outside of the occupied zone, return air typically leaves the zone through a ceiling- or wall-mounted return-air grille and travels through the open ceiling plenum back to the intake of the WSHP. This minimizes installed cost and lowers the pressure loss through the return-air path.

Alternatively, some applications use sheet metal ductwork for all or part of the return-air path. This increases installed cost and adds more pressure loss that the fan needs to overcome. So why do it? Sometimes it is required to meet a local building code. Sometimes it is done to allow easier cleaning of the return-air path.

When designing the return-air path for a WSHP, consider the following general recommendations:

- *Avoid undersizing return-air grilles.*

  If the return-air grilles are too small, they will create too high a pressure drop, and result in a significant pressure difference between the occupied space and the ceiling plenum. A space-to-plenum pressure difference of no more than 0.02 to 0.03 in. H_2O (5.0 to 7.5 Pa) is acceptable under most conditions.

  When a suspended T-bar ceiling is used, a high pressure difference between the occupied space and the ceiling plenum will typically cause some of the return air to be forced around the edges of the ceiling tiles. This causes soiling of the tiles, which increases the frequency of cleaning or replacement.
- *Avoid undersizing return-air openings within the ceiling plenum.*

  If the return-air path must pass through an interior partition wall that extends from floor-to-floor, make sure the opening through the wall is large enough to avoid an excessive pressure drop. In addition, the opening into the return air ductwork must be large enough to avoid an excessive pressure drop.

- *Use an open ceiling plenum, rather than a ducted return, whenever possible*.

  Using an open ceiling plenum for the return-air path reduces installed cost and lowers airside pressure drop, which results in less fan energy used. However, open plenum returns should not be used when prohibited by local codes.

Controls

The control of a water-source heat pump system is often grouped into unit-level and system-level control functions. Unit-level control refers to the functions required to control and protect each individual piece of equipment. System-level control refers to the intelligent coordination of the individual pieces of equipment so they operate together as an efficient system. A common analogy is to view the individual unit-level controllers as members of an orchestra, and the system-level controller as the conductor.

In a typical system, each WSHP is equipped with a unit-level controller, and the water-circulating pumps, cooling tower, and boiler are each equipped with separate unit-level controllers.

In some applications, these unit-level controllers operate with no system-level coordination. In other applications, the unit-level controllers are connected to a centralized, system-level controller. In the latter configuration, each unit-level controller is capable of performing its functions, even if communication with the system-level controller is lost.

Common unit-level and system-level control functions for a WSHP system are discussed in detail in “System Controls,” p. 166. Specific details should be obtained from the equipment or controller manufacturer.

System Design Issues and Challenges

This chapter proposes solutions to several common challenges when designing a water-source heat pump system. This is not an exhaustive list of all challenges or all solutions, but is meant to cover the most common.

Thermal Zoning

Definition of a thermal zone

A space or group of spaces within a building with similar heating and cooling requirements served by a single temperature sensor to maintain the desired temperature.

In a WSHP system, each thermal zone has a heat pump that is controlled to maintain the temperature in the zone it serves. Defining the zones in a WSHP system is often more of an art than a science, and requires judgment by the system design engineer. An individual "zone" might be either of the following:

- *A single room separated by physical boundaries (walls, windows, doors, floor and ceiling)*

  The private offices in an office building, individual classrooms in a school, or individual hotel guest rooms could each be a separate zone. In this case, each office, classroom, opr guest room would be served by an individual WSHP and zone temperature sensor.

- *A group of several rooms*

  Several of the offices or classrooms along the west-facing perimeter of the building could be grouped together as one zone. In this case, one WSHP could be used to serve the entire group of rooms, and a zone temperature sensor would typically be installed in only one of the rooms.

- *A subsection of a large, open area*

  An office building might include a large open area that is divided into cubicles. The interior portion of this open area might be separated into several zones to provide better temperature control. If the area is bounded by a perimeter wall, the outer 15 ft (4.6 m) of this area might be its own zone, due to the impact of heat gain and loss through the building envelope.

In all cases, rooms that are grouped together as a single thermal zone should have similar heating and cooling requirements and operating schedules. Whenever possible, a zone should have definite physical boundaries (walls, windows, doors, floor and ceiling). Temperature control may suffer if air can be supplied to the zone by a WSHP other than the one connected to the zone temperature sensor.

Perimeter versus interior zones

For simplicity, a typical building can be described as having two types of zones: perimeter and interior. As mentioned earlier, each zone is typically served by an individual WSHP, allowing independent temperature control.

In many climates, **perimeter** zones with walls and windows exposed to the outdoors require seasonal cooling or heating. Such zones require cooling in the summer because it is warm outside, the sun is shining through the windows, people are occupying the zone, and the lights are turned on. In the winter, these zones can require heating to offset the heat loss through the exterior walls and windows, even though some heat is generated in the zone by people, lights, and equipment.

Interior zones are typically surrounded by other zones at the same temperature, so they do not experience the same heat gain and heat loss fluctuations as a perimeter zone. Therefore, many interior zones require year-round cooling due to the relatively constant amount of heat generated by people, lights, and equipment, and the absence of heat loss through the building envelope. Interior zones on the top floor of a building might need to be treated as a perimeter zone if they experience a significant amount of heat loss through the roof.

Optimizing the number of zones

If a WSHP system is designed with too few thermal zones, it may result in undesirable temperature variations in the areas that are farther away from the zone temperature sensor. A smaller zone is typically better able to closely control temperature, which contributes to better occupant comfort. However, increasing the number of independently controlled zones also raises the installed cost of the system.

Therefore, the optimum number of zones balances occupant comfort requirements with the budgetary limits of the project. The first step is to determine the maximum number of potential zones, ignoring cost. Each room separated by physical boundaries should be a separate zone. Larger open areas should be divided into several, smaller zones.

The next step is to determine how many of these zones can be easily combined, using the following criteria.

For perimeter zones (or interior zones on the top floor of the building):

- Are there adjacent zones in which the perimeter wall and/or roof have the same exposure (east-facing, west-facing, and so on)?
- If so, do these zones have the same percentage and type of glass?
- If so, do these zones have approximately the same density of occupants, lighting, and equipment, and are the time-of-use schedules similar?
- If so, will the occupants accept the temperature varying slightly?

For interior zones (not on the top floor of the building):

- Are there adjacent zones that have approximately the same density of occupants, lighting, and equipment, and are the time-of-use schedules similar?
- If so, will the occupants accept the temperature varying slightly?

If adjacent zones meet these criteria, they likely can be grouped together into a single zone without much sacrifice in occupant comfort.

In a WSHP system that uses horizontal-style heat pumps installed in a ceiling plenum, zone size is sometimes limited by plenum height. That is, the distance between the top of the ceiling structure and the bottom of the roof or floor above may limit the size (capacity) of the heat pumps used for the project, and this might dictate the maximum zone size.

Locating the zone sensor

The zone temperature sensor should be installed in a representative location within the zone. If the zone consists of more than one room, place the sensor in the room where tighter temperature control is most important. The temperature in the other rooms may vary more than in the room with the temperature sensor.

Follow these general guidelines when locating the zone sensor:

- Do not place the zone temperature sensor where it will be affected by air discharged from a supply-air diffuser.
- Make certain that only the WSHP that is connected to the zone sensor can influence the temperature being measured by that sensor.
- Do not place the zone sensor directly on a wall with a large amount of heat gain or loss, or where solar radiation will create a false reading (generally, this means placing the sensor on an interior wall).
- Do not place the zone sensor directly above heat-generating equipment, such as a copy machine, computer terminal, or coffee maker.

Using wireless technology

In the HVAC industry, the use of wireless technology can often result in an overall lower installed cost when compared to traditional wired sensors, especially in historical or difficult-to-wire buildings, renovations, or in locations with high labor rates. It can also improve reliability by eliminating possible failure modes of hard-wired systems.

Figure 59. Example wireless zone sensor

By eliminating the wires between a zone temperature sensor and a water-source heat pump, the sensor (Figure 59) can be easily placed in the best location to accurately measure the zone temperature. This might be on a cubicle wall, a concrete or brick wall, or some other difficult-to-wire location. A wireless zone sensor is easy and inexpensive to move when the layout or use of the zone changes, or if the initial placement of the sensor turns out to be a poor location.

Another benefit of wireless technology is the ease of averaging temperature readings from multiple zone sensors. For a large or diverse zone, several wireless zone sensors can be located around the room. Then the unit controller averages the readings from those sensors, and uses this average temperature to control the water-source heat pump.

Furthermore, wireless communication eliminates the wires between the controllers on each water-source heat pump and the centralized, system-level controller. Benefits include faster project completion and easier relocation when space layout or use changes in the future. Wireless communication also makes it easier to upgrade an older system to reap the benefits of networked unit controls (see "System Controls," p. 166).

To ensure reliable operation, make sure the wireless technology adheres to the Institute of Electrical and Electronics Engineers (IEEE) Standard 802.15.4. This standard was created to minimize the risks of interference with other wireless devices. In addition, ensure that the wireless sensor has a long battery life and a visible "low battery" indicator to minimize ongoing maintenance.

Ventilation

Ventilation refers to the introduction of an adequate amount of fresh outdoor air to dilute contaminants that are generated inside the building (by people, equipment, processes, or furnishings). This requires the removal of an equal quantity of air from the building.

For a discussion of the “IAQ Procedure” (Section 6.3) in ASHRAE Standard 62.1, refer to the Trane white paper titled “Compliance with the Indoor Air Quality Procedure of ASHRAE Standard 62.1” (IAQ-WPR001*-EN).

The “Ventilation Rate Procedure” (Section 6.2) in ASHRAE Standard 62.1, *Ventilation for Acceptable Indoor Air Quality,* prescribes the quantity of outdoor air that must be delivered to each zone, based on the expected use of that zone, and then prescribes how to calculate the outdoor airflow that must be brought through the system-level intake.

In addition, Section 5 of this standard includes several requirements related to the design of the ventilation equipment and distribution system. The requirements related to ventilation system controls, particulate filtration, and humidity control are each discussed in other sections of this manual.

Note: *Because ASHRAE 62.1 is under continuous maintenance, it can change frequently. This manual is based on the 2022 published version of the standard. Refer to the most current version for specific requirements.*

Zone-level ventilation requirements

ASHRAE 62.1 requires the following three-step procedure to determine the outdoor airflow required for each ventilation zone:

1. Calculate the outdoor airflow that must be delivered to the breathing zone (V_{bz}), using the prescribed rates in Table 6-1 of the standard.
2. Determine the zone air-distribution effectiveness (E_z), which depends on the location of supply-air diffusers and return-air grilles, using the default values in Table 6-4 of the standard.
3. Calculate the outdoor airflow required for the zone (typically at the supply-air diffusers) by dividing the breathing-zone outdoor airflow by the zone air-distribution effectiveness ($V_{oz} = V_{bz}/E_z$).

Minimum ventilation required in breathing zone (V_{bz})

Table 6-1 of ASHRAE 62.1 prescribes two ventilation rates for each occupancy category: one for people-related sources of contaminants and another for building-related sources.

For step 1, determine the occupancy category for the zone using Table 6-1 and identify the corresponding ventilation rates. The people-related ventilation rate (R_p) is quantified in terms of cfm/person (L/s/person) and the building-related ventilation rate (R_a) is quantified in terms of cfm/ft^2 (L/s/m^2). Then determine the number of people expected to occupy the zone during typical usage (P_z) and the occupiable floor area (A_z). Finally, solve the following equation to find the minimum outdoor airflow required for the breathing zone (V_{bz}).

$$V_{bz} = R_p \times P_z + R_a \times A_z$$

Note: *Occupant load, or exit population, is often determined for use in designing egress paths that comply with the fire code. However, this population is typically much larger than the expected zone population (P_z) used for* *designing the ventilation system and for calculating design cooling loads. Using occupant load, rather than expected zone population, to calculate ventilation requirements will often result in oversized HVAC equipment and excessive energy use.*

Impact of zone air-distribution effectiveness

In addition to defining the breathing-zone outdoor airflow (V_{bz}), ASHRAE 62.1 also prescribes zone air-distribution effectiveness (E_Z) that accounts for how well the ventilation air, which is delivered to the zone through supply-air diffusers, actually gets into the breathing zone (Figure 60). The breathing-zone outdoor airflow (V_{bz}) is divided by this effectiveness (E_Z) to determine the outdoor airflow that must be delivered through the supply-air diffusers (V_{oz}).

Figure 60. Impact of zone air-distribution effectiveness

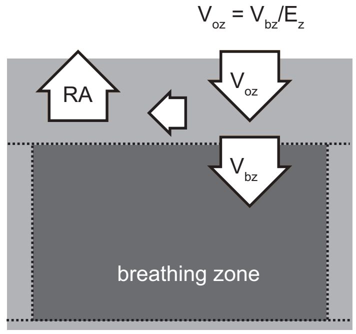


Table 13 is an excerpt from ASHRAE 62.1, and provides default values of E_Z for those air distribution configurations commonly used in WSHP systems. It is based on the placement of supply-air diffusers and return-air grilles, and the dry-bulb temperature at which the air is supplied to zone (DBT_{SA}).

Table 13. Zone air-distribution effectiveness (E_Z)

Location of supply-air diffusers	Location of return-air grilles	Supply-air temperature (DBT_{SA})	E_Z
ceiling	ceiling	cooler than zone	1.0
ceiling	floor	cooler than zone	1.0
ceiling	ceiling	warmer than zone ≥ DBT_{zone} + 15°F (8°C)	0.8
ceiling	ceiling	warmer than zone < DBT_{zone} + 15°F (8°C)[1]	1.0
ceiling	floor	warmer than zone	1.0
floor	ceiling	cooler than zone low vertical throw[2]	1.2
floor	ceiling	cooler than zone high vertical throw[3]	1.05
floor	ceiling	warmer than zone	0.7
outdoor air enters the room at a location more than half the length of the space from the exhaust, return, or both			0.8
outdoor air enters the room at a location less than half the length of the space from the exhaust, return, or both			0.5

Source: Table 6-4 of ASHRAE Standard 62.1-2022

1 Provided that the 150 fpm (0.8 m/s) supply air jet reaches to within 4.5 ft (1.4 m) of floor level. For lower velocity supply air, use E_z = 0.8.

2 Provided that vertical throw < 60 fpm (0.25 m/s) at a height of 4.5 ft (1.4 m) above the floor, and ceiling return is at a height <= 18 ft (5.5 m) above the floor.

3 Provided that vertical throw >= 60 fpm (0.25 m/s) at a height of 4.5 ft (1.4 m) above the floor, and ceiling return is at a height <= 18 ft (5.5 m) above the floor.

In many WSHP systems, the supply-air diffusers are located in or near the ceiling. When cool air ($DBT_{SA} < DBT_{zone}$) is delivered to the zone through these ceiling-mounted diffusers, the system is 100 percent effective at getting the outdoor air into the actual breathing zone (that is, $E_Z = 1.0$). This is the case whether the return-air grilles are located in (or near) the ceiling or in (or near) the floor.

However, when hot air ($DBT_{SA} \geq DBT_{zone}$ + 15°F [8°C]) is delivered to the zone through the same ceiling-mounted diffusers, and then leaves the zone through ceiling-mounted return-air grilles, the zone air-distribution effectiveness is only 0.8.

When supplied and returned overhead, the buoyancy of this hot air tends to cause some of the air to bypass from the supply-air diffusers to the return-air grilles, without reaching the actual breathing zone. Therefore, this configuration is less than 100 percent effective at delivering outdoor air from the diffusers into the breathing zone.

For zones that require heating, employ one of the following strategies:

- If $DBT_{SA} \geq DBT_{zone}$ + 15°F (8°C), increase the outdoor airflow delivered through the diffusers ($V_{oz} = V_{bz}/0.8$) during the heating mode to compensate for the zone air-distribution effectiveness ($E_Z = 0.8$) of using ceiling-mounted return-air grilles. Alternatively, locate the return-air grilles in the floor or at the base of a side wall, in which case E_Z would equal 1.0 during both cooling and heating modes.
- Design the system so that the supply-air dry-bulb temperature (DBT_{SA}) during heating mode is less than 15°F (8°C) above the zone temperature (DBT_{zone}), and select the supply-air diffusers to achieve a velocity of at least 150 fpm (0.8 m/s) within 4.5 ft (1.4 m) of the floor. With this design, a zone air-distribution effectiveness (E_Z) of 1.0 can be achieved, even with overhead supply of warm air and overhead return.
- If a dedicated OA system is being used, consider delivering the outdoor air directly to each zone through separate “ventilation” diffusers, at either a cold or “neutral” dry-bulb temperature. Since outdoor air is not delivered to the zone by the WSHP (which is used to provide heating for the zone), the outdoor air does not need to be delivered at a temperature warmer than the zone (DBT_{zone}), and zone air-distribution effectiveness (E_Z) can be 1.0.

Note: *The only configurations that have a zone air-distribution effectiveness greater than 1.0 is when cool air is delivered to the zone using low-velocity, thermal displacement ventilation (TDV) or properly-designed underfloor air distribution (UFAD) diffusers. However, if either of these systems is used to deliver warm air through its floor-mounted diffusers, and return air through ceiling-mounted grilles, zone air-distribution effectiveness is only 0.7.*

System-level ventilation requirement

ASHRAE 62.1 also defines procedures for calculating the outdoor airflow that must be brought in through the system-level intake (V_{ot}) to ensure the required quantity of outdoor air is delivered to each zone (V_{oz}). Which procedure to use depends on the configuration of the ventilation system.

For more information on ASHRAE Standard 62.1 and its procedures for calculating zone-level and system-level outdoor airflow requirements for a WSHP system, refer to the Trane *Engineers Newsletter Live* video titled “ASHRAE Standard 62.1-2019” (APP-CMC077-EN).

Many WSHP systems use a dedicated outdoor-air system to deliver only outdoor air to the individual zones—defined by ASHRAE 62.1 as a “100 percent outdoor-air system.” In some WSHP configurations, such as rooftop or some vertical or console models, outdoor air might be brought in through the WSHP itself—defined by ASHRAE 62.1 as a “single-zone system,” assuming that the WSHP serves only a single ventilation zone.

Note: *If outdoor air is delivered through a WSHP that serves more than one ventilation zone, it might be classified by ASHRAE 62.1 as a “multiple-zone recirculating system.” In that case, the outdoor airflow brought in through the system-level outdoor-air intake at the WSHP would need to be determined using Section 6.2.4 of ASHRAE 62.1.*

Calculating system intake airflow (V_{ot}) for a dedicated outdoor-air system

Most WSHP systems use a dedicated (100 percent) outdoor-air system to condition all the outdoor air for the system. As described in "Dedicated OA system configurations," p. 59, conditioned outdoor air (CA) is then either:

1. Ducted directly to each zone
2. Ducted to the supply-side of each WSHP, where it mixes with supply air from the heat pump before being delivered to the zone
3. Ducted to the intake of each WSHP, where it mixes with recirculated air from the zone before entering the WSHP
4. Ducted to the ceiling plenum, near the intake of each WSHP, where it mixes with recirculated air from the zone before entering the WSHP

In any of these configurations, since the outdoor air is delivered to each zone or to each zone-level WSHP, this is not considered a multiple-zone *recirculating* ventilation system. Rather, it is considered a 100 percent outdoor-air system because one unit delivers only outdoor air to one or more ventilation zones. Accordingly, per Section 6.2.3 of ASHRAE 62.1, the system-level intake airflow (V_{ot}) delivered by the dedicated OA unit should be the sum of the calculated zone outdoor airflows (V_{oz}):

$$V_{ot} = \Sigma V_{oz}$$

Figure 61. Example office building with a WSHP system (E_z)

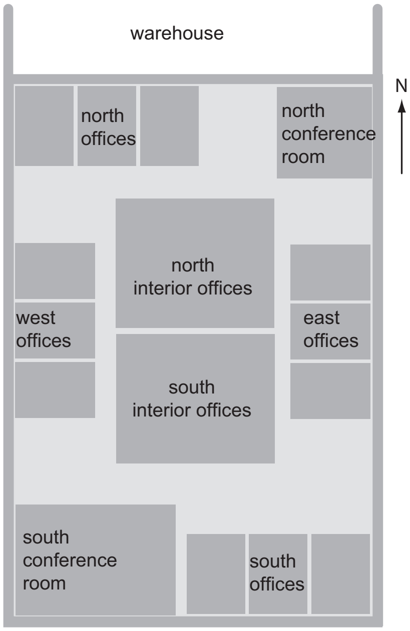


Figure 61 shows an example eight-zone office building. The three-step, zone-level ventilation calculations have already been completed (Table 14); see "Zone-level ventilation requirements," p. 90.

In this case, each zone is served by a water-source heat pump and outdoor air is delivered directly to the intake of each ceiling-mounted heat pump by a dedicated OA unit. The preconditioned outdoor air mixes with locally recirculated air from the zone, and this mixture is then either cooled or heated by the WSHP before it is supplied to the zone. Assuming negligible duct leakage, all intake air reaches all supply diffusers, so the system intake airflow at the dedicated OA unit must equal the sum of the zone outdoor airflows ($V_{ot} = \Sigma V_{oz}$). For this example, at the cooling design condition, the required system outdoor-air intake flow (V_{ot}) is 3100 cfm (1.46 m^3/s).

Table 14. 100% OA system ventilation calculations for example office building (cooling design condition)

	R_p (cfm/p)	P_z (qty)	R_a (cfm/ft^2)	A_z (ft^2)	V_{bz} (cfm) /	E_z =	V_{oz} (cfm)
South offices	5	18	0.06	2000	210	1.0	210
West offices	5	20	0.06	2000	220	1.0	220
South conf room	5	30	0.06	3000	330	1.0	330
East offices	5	20	0.06	2000	220	1.0	220
South interior offices	5	50	0.06	10000	850	1.0	850
North interior offices	5	50	0.06	10000	850	1.0	850
North offices	5	16	0.06	2000	200	1.0	200
North conf room	5	20	0.06	2000	220	1.0	220
System totals							3100 ($V_{ot} = \Sigma V_{oz}$)

When a WSHP is operating in the heating mode, it will likely be supplying air to the zone at a dry-bulb temperature (DBT_{SA}) that is warmer than the zone. As mentioned earlier, supplying warm air to the zone through ceiling-mounted diffusers may result in a zone air-distribution effectiveness (E_Z) that is less than 1.0. How does this impact system intake airflow? Table 15 includes calculations for this same example system at the heating design condition, with several zones having a zone air-distribution effectiveness of 0.8. For this example, at the heating design condition, the higher required zone outdoor airflows (V_{oz}) increase the required system outdoor-air intake flow (V_{ot}) to 3450 cfm (1.63 m^3/s).

Table 15. 100% OA system ventilation calculations for example office building (heating design condition)

	R_p (cfm/p)	P_z (qty)	R_a (cfm/ft^2)	A_z (ft^2)	V_{bz} (cfm) /	E_z =	V_{oz} (cfm)
South offices	5	18	0.06	2000	210	0.8	263
West offices	5	20	0.06	2000	220	0.8	275
South conf room	5	30	0.06	3000	330	0.8	413
East offices	5	20	0.06	2000	220	0.8	275
South interior offices	5	50	0.06	10000	850	1.0	850
North interior offices	5	50	0.06	10000	850	1.0	850
North offices	5	16	0.06	2000	200	0.8	250
North conf room	5	20	0.06	2000	220	0.8	275
System totals							3450 ($V_{ot} = \Sigma V_{oz}$)

Consider, however, if the dedicated OA system was designed to deliver conditioned OA directly to each zone through separate "ventilation" diffusers, at either a cold or "neutral" dry-bulb temperature. Since outdoor air is not delivered to the zone by the WSHP (which is used to provide heating for the zone), the outdoor air does not need to be delivered at a temperature warmer than the zone (DBT_{zone}), and zone air-distribution effectiveness (E_Z) can be 1.0. For this example, this would result in the required system outdoor-air intake flow (V_{ot}) being 3100 cfm (1.46 m^3/s) at both the cooling and heating design conditions.

Calculating system intake airflow (V_{ot}) for a dedicated outdoor-air system when conditioned OA is supplied to a central corridor

In some buildings, a dedicated OA unit delivers conditioned OA to a central corridor. This OA is then drawn into each zone underneath the door or through a grille in the door (Figure 62). This configuration is most commonly used in dormitories, barracks, apartment buildings, high-rise condominiums, extended care facilities, and hotels.

As explained in example 6-H in the *Standard 62.1-2019 User's Manual*, two rows in the table of default values for zone air-distribution effectiveness (E_Z) were included for this specific configuration (see Table 13, p. 91). When outdoor air is drawn into the breathing zone and enters the zone at a location that is near (rather than on the opposite side of the room from) the location of the exhaust outlet, ASHRAE 62.1 prescribes the default value for E_Z as 0.5. This suggests that a substantial amount of the outdoor air that is drawn in from the corridor will bypass the breathing zone and leave through the exhaust.

Figure 62. Hotel guest room with outdoor air drawn in from a corridor

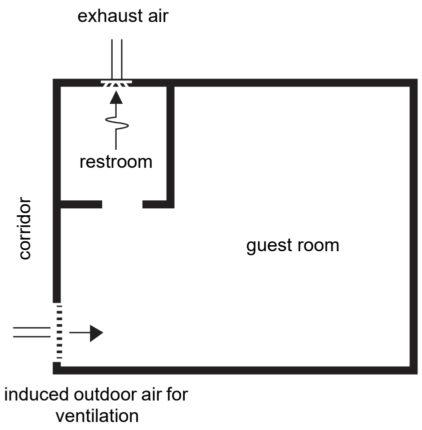


Table 16 shows the zone-level ventilation calculations for an example 12-room wing of a hotel. While the required breathing-zone outdoor airflow (V_{bz}) is only 25 cfm (0.012 m^3/s) for each guest room, the zone air-distribution effectiveness (E_Z) of 0.5 increases the required zone outdoor airflow (V_{oz}) to 50 cfm (0.024 m^3/s). For this example, the required system outdoor-air intake flow (V_{ot}) is 624 cfm (0.29 m^3/s).

Note: Although Table 6-2 of ASHRAE 62.1 only requires a minimum exhaust rate of 25 cfm (0.012 m^3*/s) for a private restroom (assuming continuous exhaust fan operation), for this example, the restroom exhaust airflow would need to be increased to 50 cfm (0.024* m^3*/s) to draw 50 cfm (0.024* m^3*/s) of air into the zone from the corridor.*

Table 16. 100% OA system ventilation calculations for example hotel building with OA delivered to a central corridor

	R_p (cfm/p)	P_z (qty)	R_a (cfm/ft^2)	A_z (ft^2)	V_{bz} (cfm) /	E_z =	V_{oz} (cfm)
Guest room 101	5	2	0.06	250	25	0.5	50
Guest room 102	5	2	0.06	250	25	0.5	50
Guest room 103	5	2	0.06	250	25	0.5	50
Guest room 104	5	2	0.06	250	25	0.5	50
Guest room 105	5	2	0.06	250	25	0.5	50
Guest room 106	5	2	0.06	250	25	0.5	50
Guest room 107	5	2	0.06	250	25	0.5	50
Guest room 108	5	2	0.06	250	25	0.5	50
Guest room 109	5	2	0.06	250	25	0.5	50
Guest room 110	5	2	0.06	250	25	0.5	50
Guest room 111	5	2	0.06	250	25	0.5	50
Guest room 112	5	2	0.06	250	25	0.5	50
Corridor			0.06	400	24	1.0	24
System totals							624 ($V_{ot} = \Sigma V_{oz}$)

Consider, however, if the dedicated OA system ducted the outdoor air directly to each guest room, rather than requiring it to be drawn into each room from the corridor. Assuming the outdoor air is never delivered at a dry-bulb temperature warmer than the zone, the zone air-distribution effectiveness (E_Z) would be 1.0, and the required zone outdoor airflow (V_{OZ}) would be only 25 cfm (0.012 m^3/s) for each guest room. For the example shown in Table 16, this would reduce the required system outdoor-air intake flow (V_{ot}) to 324 cfm (0.15 m^3/s).

Calculating system intake airflow (V_{ot}) for a single-zone system

In some WSHP configurations (such as rooftop, console, or larger vertical models), outdoor air might be brought in through the WSHP itself and then delivered to the zone. When one WSHP delivers a mixture of outdoor air and recirculated air to only one ventilation zone, ASHRAE 62.1 defines this as a "single-zone system."

For this type of ventilation system, Section 6.2.2 of ASHRAE 62.1 requires that the system-level intake (V_{ot}) needs to equal the calculated zone outdoor airflow (V_{oz}):

$$V_{ot} = V_{oz}$$

Returning to the example office building in Figure 61, p. 93, assume that the South Conference Room is served by a rooftop-style WSHP that brings in outdoor air, mixes it with recirculated air from that zone, and then delivers the cooled or heated mixture to the zone through ceiling-mounted supply-air diffusers. Air returns from the zone through ceiling-mounted return-air grilles.

Assuming negligible duct leakage, all intake air reaches the supply-air diffusers, so the system intake airflow for this single-zone WSHP must equal the calculated zone outdoor airflow ($V_{ot} = V_{oz}$). For this example, at the cooling design condition (Table 17), the required system outdoor-air intake flow (V_{ot}) is 330 cfm (0.16 m^3/s).

Table 17. Single-zone system ventilation calculations for example office building (cooling design condition)

	R_p (cfm/p)	P_z (qty)	R_a (cfm/ft^2)	A_z (ft^2)	V_{bz} (cfm) /	E_z =	V_{oz} (cfm)
South conf room	5	30	0.06	3000	330	1.0	330
System total							330 ($V_{ot} = V_{oz}$)

When the WSHP is operating in the heating mode, it will likely be delivering air to the zone at a dry-bulb temperature that is warmer than the zone. As mentioned earlier, supplying warm air to the zone through ceiling-mounted diffusers may result in a zone air-distribution effectiveness (E_Z) that is less than 1.0.

For this example, at the heating design condition (Table 18), a zone air-distribution effectiveness of 0.8 results in a higher zone outdoor airflow (V_{oz}), increasing the required system outdoor-air intake flow (V_{ot}) to 413 cfm (0.19 m^3/s).

Table 18. Single-zone system ventilation calculations for example office building (heating design condition)

	R_p (cfm/p)	P_z (qty)	R_a (cfm/ft^2)	A_z (ft^2)	V_{bz} (cfm) /	E_z =	V_{oz} (cfm)
South conf room	5	30	0.06	3000	330	0.8	413
System total							413 ($V_{ot} = V_{oz}$)

Dynamic reset of intake airflow

Section 6.2.6 of ASHRAE 62.1 explicitly permits dynamic reset of intake (outdoor) airflow as operating conditions change, as long as the system provides at least the required breathing-zone outdoor airflow whenever a zone is occupied. The standard specifically mentions "resetting intake airflow in response to variations in zone population."

As the number of people occupying a zone varies, the quantity of outdoor air required to properly ventilate that zone varies. This strategy, commonly referred to as "demand-controlled ventilation" (DCV), attempts to dynamically reset the outdoor airflow delivered to a zone based on the changing population in that zone. Some of the commonly used methods for assessing zone population include time-of-day occupancy schedules in the building automation system (BAS), occupancy sensors, and carbon dioxide (CO_2) sensors.

For more information, see "Demand-controlled ventilation," p. 195.

Humidity Control

While "humidity control" is apt to imply special applications, such as museums or printing plants, managing humidity should be a key design consideration in any HVAC application.

Dehumidification

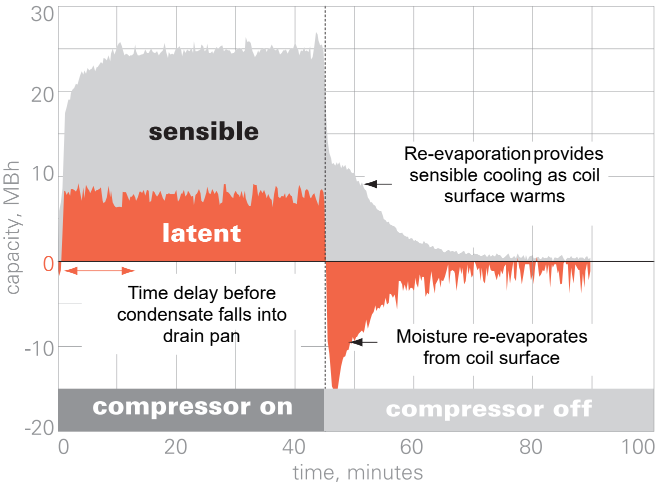
A conventional water-source heat pump typically supplies a constant airflow at all load conditions, and the compressor cycles on and off as needed to maintain zone temperature at setpoint. When the zone cooling load is higher, the compressor operates for a greater portion of the time. When the zone cooling load is lower, the compressor operates for a shorter portion of the time.

For more information on the dehumidification performance of a constant-volume system with cycling compressors, refer to the Trane application manual, *Dehumidification in HVAC Systems* (SYS-APM004-EN), and the Trane *Engineers Newsletter,* "Better Part-Load Dehumidification" (ADM-APN011-EN).

Dehumidification impact of compressor cycling and constant-speed fan

Recent research has demonstrated how compressor cycling affects part-load dehumidification performance when the fan operates at a constant speed. In Figure 63, the X axis depicts time and the Y axis depicts capacity. When the compressor starts, the coil surface quickly becomes cold enough to provide both sensible cooling and latent (dehumidification) capacity.

Figure 63. Part-load dehumidification (cycling compressor, constant-speed fan)



Source: Shirey, D., H. Henderson, and R. Raustad. 2003. DOE/NETL Project DE-FC26-01NT41253

Notice that several minutes elapse after the compressor starts before the water vapor that condenses on the coil surface actually falls into the drain pan below.

Fan cycling

If the fan in the heat pump is controlled to cycle on and off along with the compressor, rather than continue to operate at a constant speed when the compressor is off, part-load dehumidification will likely be improved. If the fan turns off, it minimizes the re-evaporation of condensate from the coil surface.

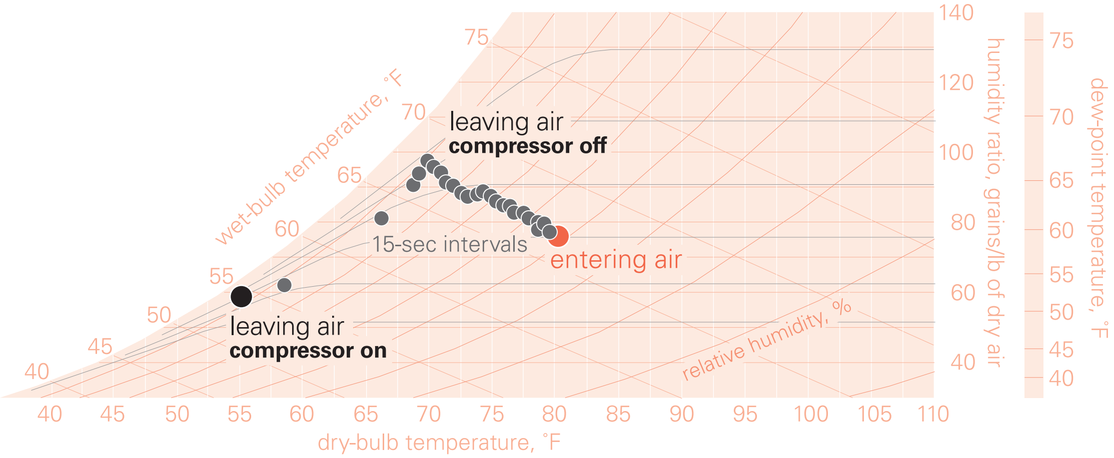
If the heat pump includes a two-stage, variable-capacity, or variable-speed compressor, or if it includes more than one compressor, there may be an opportunity to automatically change fan speeds at part-load conditions (see "Adjustable fan speed," p. 101).

The droplets of condensation on the coil fins must accumulate enough mass for gravity to overcome surface tension and fall into the drain pan.

When the compressor stops, sensible cooling drops off dramatically; meanwhile, latent (dehumidification) capacity not only falls to zero but actually becomes negative. As the fan continues to operate, condensate on the coil surface re-evaporates into the supply air stream—water vapor is being added to the air stream, not removed. This evaporation takes some time, during which evaporation has the effect of sensibly cooling the air.

Figure 64 shows this same data plotted on a psychrometric chart, with the dots indicating the condition of the air leaving the evaporator in 15-second intervals. When the compressor is on, the leaving air is cold and dry. But when the compressor stops, the leaving air quickly warms up, and condensate from the coil surface begins to evaporate back into the air stream. The result is that the humidity ratio (or dew point) of the air leaving the coil is higher than when it entered. As water on the coil surface continues to evaporate, the condition of the leaving air travels along a constant wet-bulb line until all the water has evaporated.

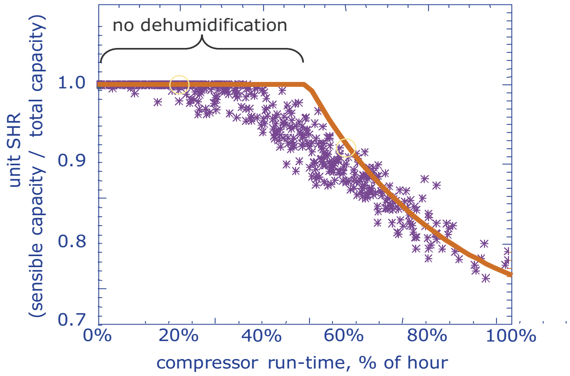
Figure 64. Effect of compressor cycling on condition of the air leaving the evaporator



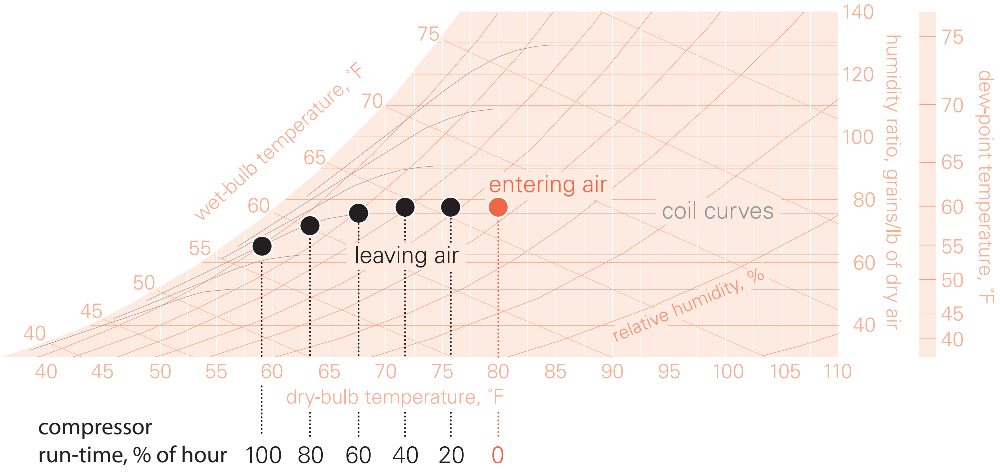
Bear in mind that the test data depicted in Figure 63 and Figure 64 was based on 45-minute on/off cycles. In actual operation, compressor off-time depends on the zone sensible cooling load and may last only a few minutes—too briefly to permit the coil to dry completely between compressor starts.

To better mimic real-world operation, the same researchers varied compressor runtime to determine the effect on net dehumidification capacity (top chart in Figure 65). Net dehumidification capacity equals all of the water vapor that condenses on the coil while the compressor is on, minus the water that re-evaporates after the compressor turns off. Notice that as the compressor operates for a smaller percentage of the hour, this system (constant-speed fan, cycling compressor) provides little or no net dehumidification benefit. The compressor is not operating for a long enough period of time to allow condensate to build up and fall into the drain pan. And the compressor remains off for longer periods of time, which allows more water to re-evaporate from the coil.

Figure 65. Net dehumidification as a function of compressor run-time

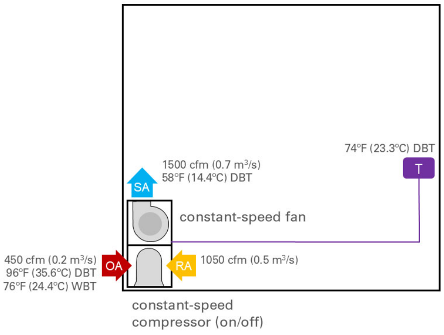


Source: Henderson, H. 1998. Proceedings of ASHRAE Energy and IAQ Conference.



Note: This chart can also demonstrate what happens when constant-volume DX equipment is oversized. If the equipment is oversized, the compressor operates for a smaller percentage of the hour, and less dehumidification occurs.

Figure 66. Conventional, constant-speed WSHP at full load (unconditioned OA)



The bottom chart in Figure 65 shows this same data plotted on a psychrometric chart. Note that the “coil curves” on the Trane psychrometric chart provide a good depiction of the part-load dehumidification performance of a cycling DX system with a constant-speed fan.

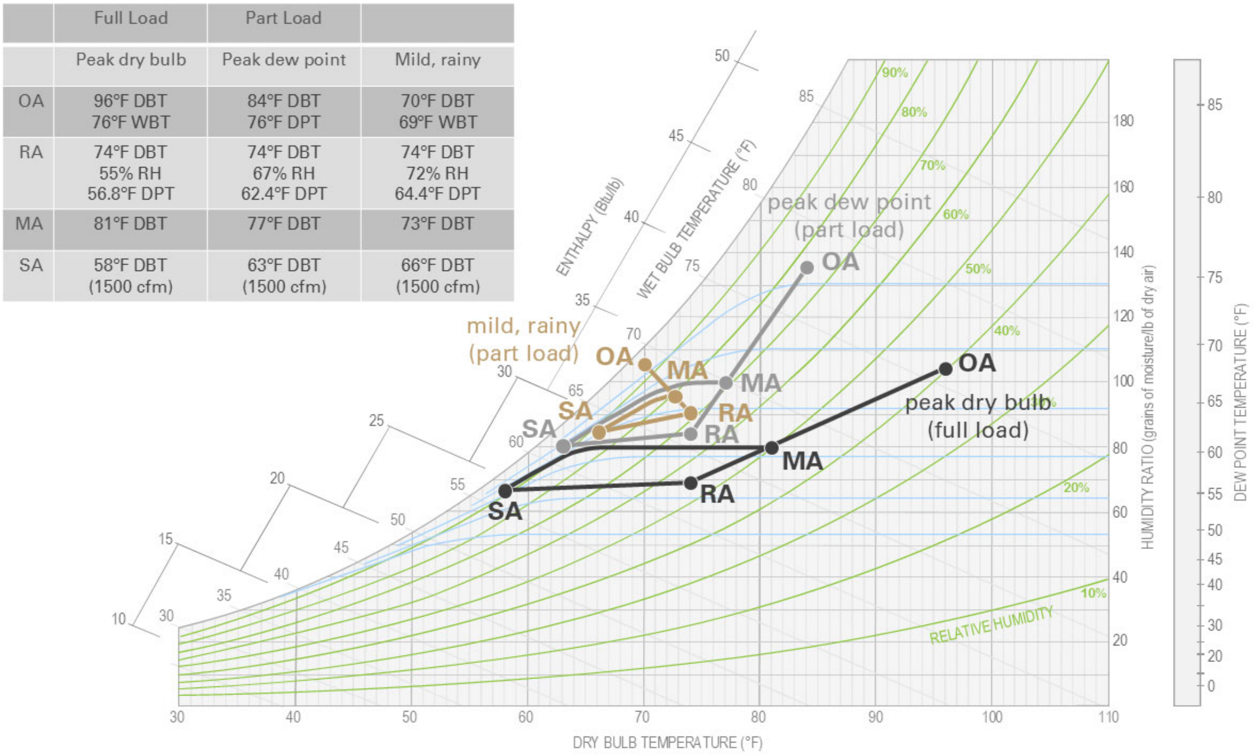
Full-load versus part-load dehumidification performance of a conventional, constant-speed WSHP (unconditioned OA)

First, consider if the outdoor air enters the WSHP directly, and is not preconditioned by a dedicated outdoor-air system (Figure 66). In this basic, constant-volume, mixed-air configuration, the heat pump mixes outdoor air (OA) with recirculated air (RA), and supplies a constant volume of air to a single thermal zone. The mixture passes through the refrigerant-to-air heat exchanger to be cooled and dehumidified. When the supply air (SA) reaches the zone, it extracts sensible heat and moisture (latent heat).

To demonstrate the dehumidification performance of a WSHP in this basic, constant-volume, mixed-air configuration, consider a 10,000 ft^3 (283 m^3) classroom in Jacksonville, Florida, that accommodates 30 people. During cooling mode, the zone temperature setpoint is 74°F (23.3°C) dry bulb. At the traditional design condition (peak outdoor dry-bulb temperature), this example system mixes 450 cfm (0.2 m^3/s) of outdoor air (OA) required for ventilation with 1050 cfm (0.5 m^3/s) of air recirculated (RA) from the zone, and then delivers the 1,500 cfm (0.7 m^3/s) of supply air (SA) at 58°F (14.4°C) to offset the sensible cooling load in the zone and maintain the zone temperature at setpoint.

Plotting this system on a psychrometric chart, the resulting dew-point temperature in the zone is 56.8°F (13.8°C) at this design condition, and relative humidity is 55 percent (black lines and labels in Figure 67).

Figure 67. Full- versus part-load dehumidification performance of a basic, constant-volume, mixed-air system (conventional WSHP)



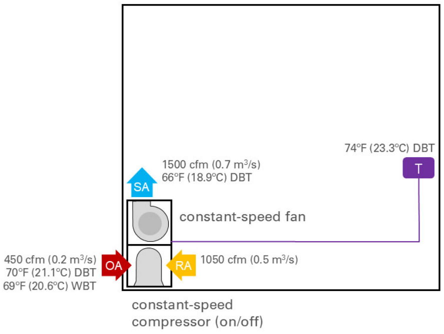
	Full Load	Part Load	
	Peak dry bulb	Peak dew point	Mild, rainy
OA	96°F DBT 76°F WBT	84°F DBT 76°F DPT	70°F DBT 69°F WBT
RA	74°F DBT 55% RH 56.8°F DPT	74°F DBT 67% RH 62.4°F DPT	74°F DBT 72% RH 64.4°F DPT
MA	81°F DBT	77°F DBT	73°F DBT
SA	58°F DBT (1500 cfm)	63°F DBT (1500 cfm)	66°F DBT (1500 cfm)

At part-load conditions, when the zone sensible cooling load decreases, this conventional WSHP continues to supply a constant flow rate of air to the zone. Meanwhile, the compressor cycles on and off, which results in a warmer average temperature delivered to the zone. Although this control action successfully maintains the zone dry-bulb temperature at setpoint, the cycling compressor reduces the amount of moisture removed, and zone humidity rises.

At the example part-load (peak outdoor dew point) condition, the compressor runtime is shortened, resulting in less dehumidification, and zone dew point rises to 62.4°F (16.9°C) (gray lines and labels in Figure 67).

An even more challenging part-load condition is a mild, rainy day. At this example condition, the only sensible load in the classroom is from people and lights, so the compressor cycles off for an even greater portion of the hour. The conventional WSHP continues to supply a constant flow rate of air to the zone, and zone dew point rises to 64.4°F (18.0°C) (brown lines and labels in Figure 67 and Figure 68).

Figure 68. Conventional, constant-speed WSHP at part load (unconditioned OA)



Section 5.12 of ASHRAE 62.1-2022 requires that systems be designed to limit the indoor dew-point temperature to 60°F (15°C) or less during both occupied or unoccupied hours, whenever the outdoor dew point is above 60°F (15°C).

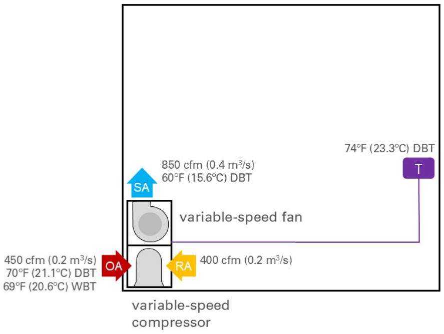
This basic, constant-volume system matches sensible capacity to the sensible cooling load, while dehumidification capacity is coincidental. As the zone sensible cooling load decreases, the compressor operates for less of the hour. Some dehumidification may occur, but only if the zone sensible cooling load is high enough to keep the compressor operating long enough (higher runtime).

Methods for improving dehumidification performance

If the dehumidification performance of a conventional heat pump in this basic, constant-volume, mixed-air configuration is not acceptable, the system can be altered to enhance dehumidification performance:

Adjustable fan speed. A WSHP that combines a multiple-speed fan with a two-stage, variable-capacity, or variable-speed compressor, or more than one compressor, reduces supply airflow at part load (see "Multiple-speed fan operation," p. 21). This results in a lower supply-air temperature and improves dehumidification.

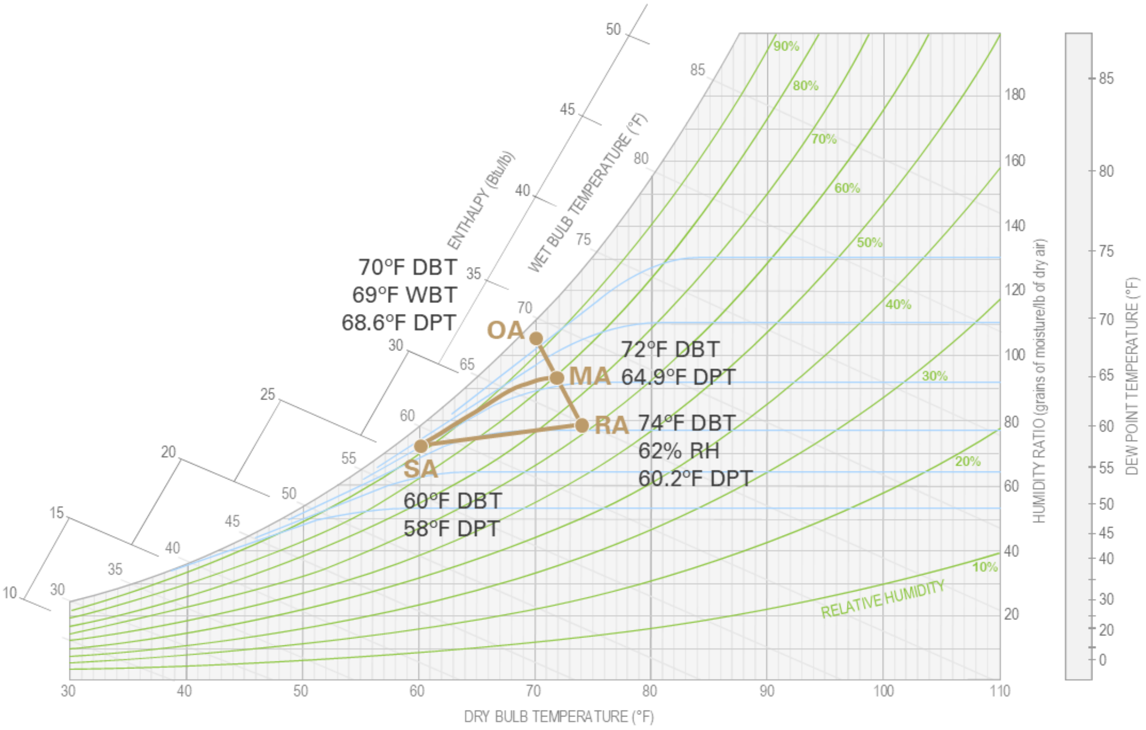
Figure 69. Variable-speed WSHP at part load (unconditioned OA)



Using the previous classroom example, the WSHP delivers 1,500 cfm (0.7 m³/s) of supply air to offset the design space sensible cooling load. However, as the sensible-cooling load in the space decreases, this variable-speed WSHP responds by simultaneously reducing fan speed and compressor capacity.

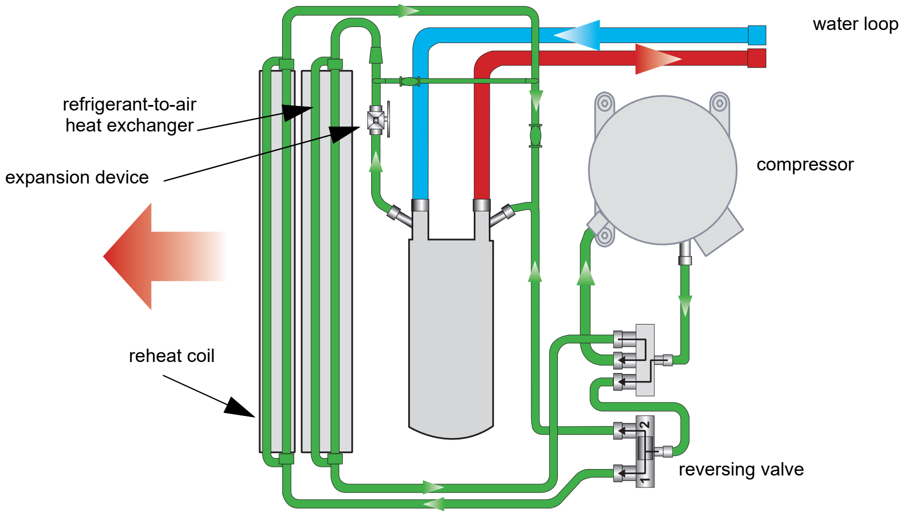
At the same example part-load (mild, rainy day) condition, the reduced supply airflow—from 1,500 cfm (0.7 m³/s) to 850 cfm (0.4 m³/s)—results in a lower supply-air (SA) temperature than if a constant-speed fan was used (Figure 69). Reducing the airflow allows the coil to remove more moisture and lengthens the compressor runtime, both of which improve the dehumidification performance of the system. At this condition, the zone dew-point temperature improves from 64.4°F (18.0°C) with the constant-speed fan to 60.2°F (15.7°C) with the multiple-speed fan and variable-speed compressor (Figure 70).

Figure 70. Dehumidification performance of a multiple-speed supply fan at mild, rainy condition



Hot-gas reheat in the WSHP unit. Another option for controlling humidity is to reheat the dehumidified supply air with heat recovered from the refrigeration circuit in the WSHP. This is sometimes referred to as hot-gas reheat. In this configuration, the air is first cooled and dehumidified by the refrigerant-to-air heat exchanger, then reheated by the reheat coil to control not only the dry-bulb temperature, but also the humidity level in the zone (Figure 71).

Figure 71. Hot gas reheat for humidity control



Doesn't ASHRAE Standard 90.1 prohibit the use of reheat for humidity control?

While Section 6.5.2.3 of ASHRAE 90.1 limits the use of reheat for the purpose of humidity control, it lists several exceptions for which reheat is allowed. Exception 5 allows for the unlimited use of reheat for humidity control if at least 90 percent of the energy for reheating is site-recovered energy (such as heat recovered from the refrigeration circuit, or hot-gas reheat).

As long as the zone humidity level is less than the desired upper limit—60°F (15°C) dew point, for example—the heat pump operates in the standard cooling mode and the compressor cycles on and off to maintain zone temperature. When the humidity sensor indicates that zone humidity is too high, and the zone temperature is at setpoint, the compressor remains on to continue dehumidifying the air, and the reheat valve diverts hot refrigerant vapor from the compressor through the reheat coil (Figure 71), warming the supply air to avoid overcooling the zone.

Using the previous classroom example, the WSHP delivers a constant 1,500 cfm (0.7 m^3/s) of air to the zone. For comparison purposes, assume the controls are set to keep the zone humidity at or below 60°F (15°C) dew point. At the same example part-load condition (mild, rainy day), the compressor operates for a longer portion of the hour to dehumidify the air (CC) and maintain zone humidity at 59.3°F (15.2°C) dew point, and the hot gas reheat coil warms the supply air (SA) to avoid overcooling the zone (Figure 72).

Figure 72. Constant-speed WSHP with HGRH at part load (unconditioned OA)

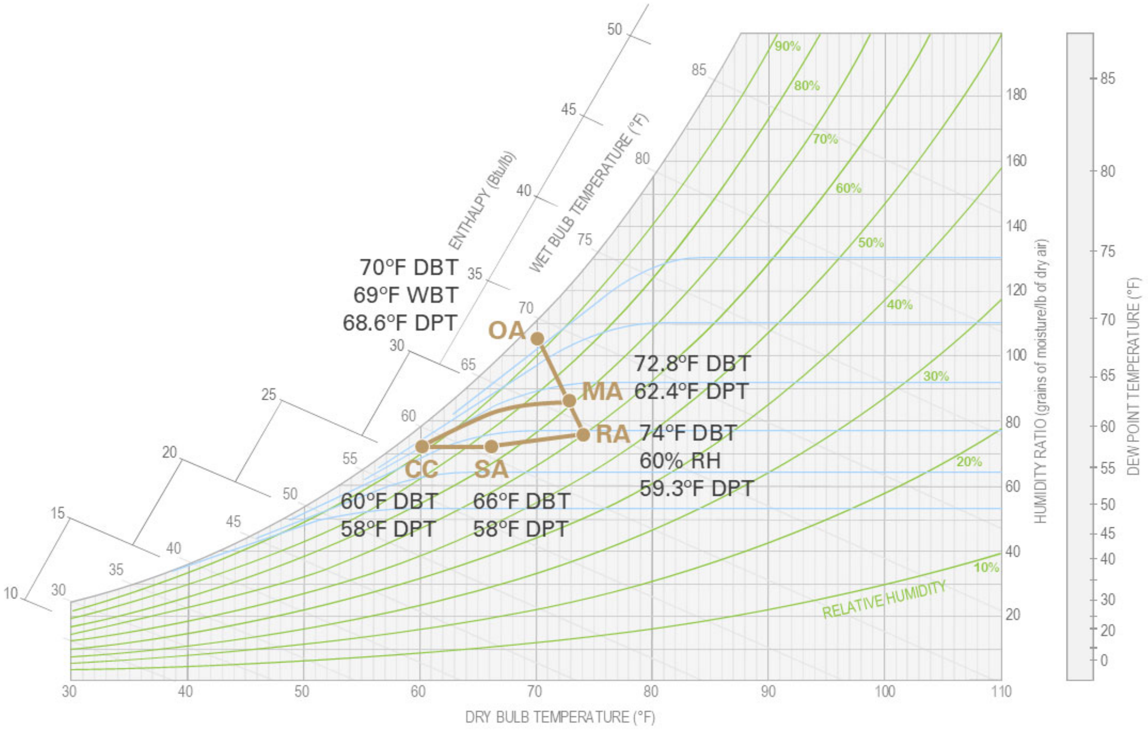


Table 19 compares performance at the example mild, rainy condition. The constant-speed WSHP with hot-gas reheat provides 2.8 tons (9.9 kW) of cooling and the fan moves 1,500 cfm (0.7 m^3/s), while the variable-speed WSHP provides 2.1 tons (7.4 kW) of cooling and the fan moves only 850 cfm (0.4 m^3/s). This results in less compressor energy, and less fan energy.

Table 19. Comparing example part-load performance (mild, rainy day)

	Constant-speed WSHP	Variable-speed WSHP	Constant-speed WSHP with HGRH
Zone dew point	64.4°F (18.0°C)	60.2°F (15.7°C)	59.3°F (15.2°C)
Cooling load	1.8 tons (6.3 kW)	2.1 tons (7.4 kW)	2.8 tons (9.9 kW)
Fan airflow	1500 cfm (0.7 m^3/s)	850 cfm (0.4 m^3/s)	1500 cfm (0.7 m^3/s)

In many applications, the variable-speed WSHP may do a good enough job of limiting indoor humidity levels, avoiding the need to equip the heat pump with hot-gas reheat.

In this example, the WSHP with hot-gas reheat could be controlled to maintain a lower zone humidity level, but that would result in even more compressor energy use. While the heat used for reheat is recovered energy, it is not free; it comes at the price of increased compressor energy. If a project requires lower space humidity levels—50 percent RH or 55°F (13°C) dew point for example—a more efficient approach would be to use a dedicated outdoor-air system to dehumidify the outdoor air centrally (see below). This would result in lower indoor humidity levels, and is more efficient than equipping the WSHPs with hot-gas reheat.

For a detailed discussion on how to determine the **required supply-air dew-point temperature** from a dedicated OA unit, refer to the Trane application guide, *Dedicated Outdoor Air Systems* (SYS-APG001*-EN), or the Trane *Engineers Newsletter*, titled "Impact of DOAS Supply-Air Dew-Point Temperature on Space Humidity" (ADM-APN073-EN).

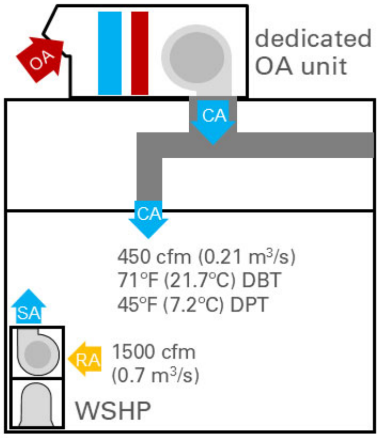
Condition the outdoor air with a separate, dedicated unit. The most common way to improve the dehumidification performance of a WSHP system is to use a dedicated outdoor-air system to separately dehumidify all of the outdoor air to a dew point that is drier than the zone. This conditioned outdoor air (CA) is then either:

1. Ducted directly to each zone
2. Ducted to the supply-side of each WSHP, where it mixes with supply air from the heat pump before being delivered to the zone
3. Ducted to the intake of each WSHP, where it mixes with recirculated air from the zone before entering the WSHP
4. Ducted to the ceiling plenum, near the intake of each WSHP, where it mixes with recirculated air from the zone before entering the WSHP

For more discussion, see "Dedicated OA system configurations," p. 59. In any of these configurations, the dedicated OA unit should dehumidify the outdoor air to a dew point that is drier than the zone, so that it will also offset the zone latent loads. The heat pumps then only need to offset the zone sensible cooling loads.

Returning to the previous classroom example, Figure 73 depicts a dedicated OA system that delivers 450 cfm (0.21 m³/s) of outdoor air (OA) directly to this classroom. The dedicated OA unit dehumidifies the entering outdoor air to a low dew point—45°F DPT (7.2°C DPT) in this example—and then reheats it to a "neutral" dry-bulb temperature (see "Neutral- versus cold-air delivery," p. 64)—71°F DBT (21.7°C DBT). Meanwhile, a console-style WSHP in the classroom cools 1,500 cfm (0.7 m³/s) of recirculated air (RA) from the zone to offset the zone sensible cooling load.

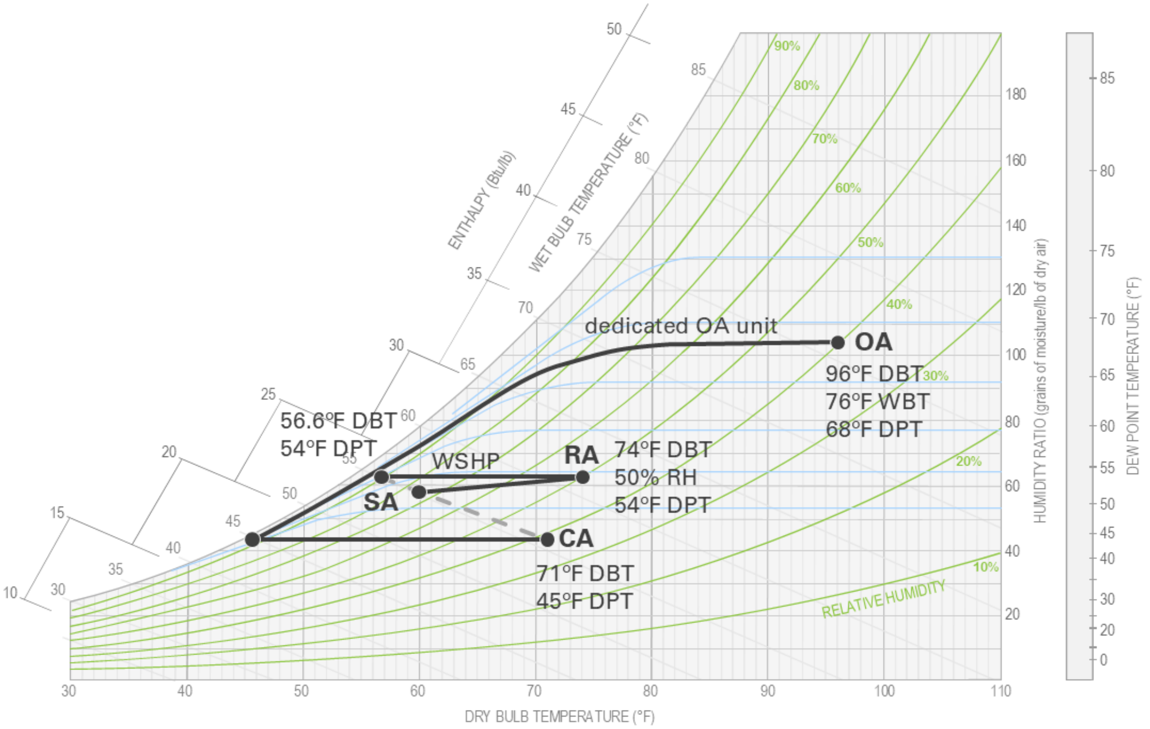
Figure 73. Example dedicated outdoor-air system with a console-style WSHP



Whether the conditioned outdoor air (CA) is delivered at a cold temperature or reheated to neutral (see "Neutral- versus cold-air delivery," p. 64), the resulting indoor humidity level will typically be the same. In either case, the outdoor air is dehumidified to the same leaving-air dew point, 45°F (7.2°C) in this example.

At the peak dry-bulb condition (Figure 74), the WSHP cools the supply air (SA) to 56.6°F DBT (13.5°C DBT). Together with the conditioned outdoor air (CA) delivered to the zone by the dedicated OA unit, the two air streams maintain the zone at the desired temperature of 74°F (23.3°C) and the resulting zone dew point is 54°F (12.2°C).

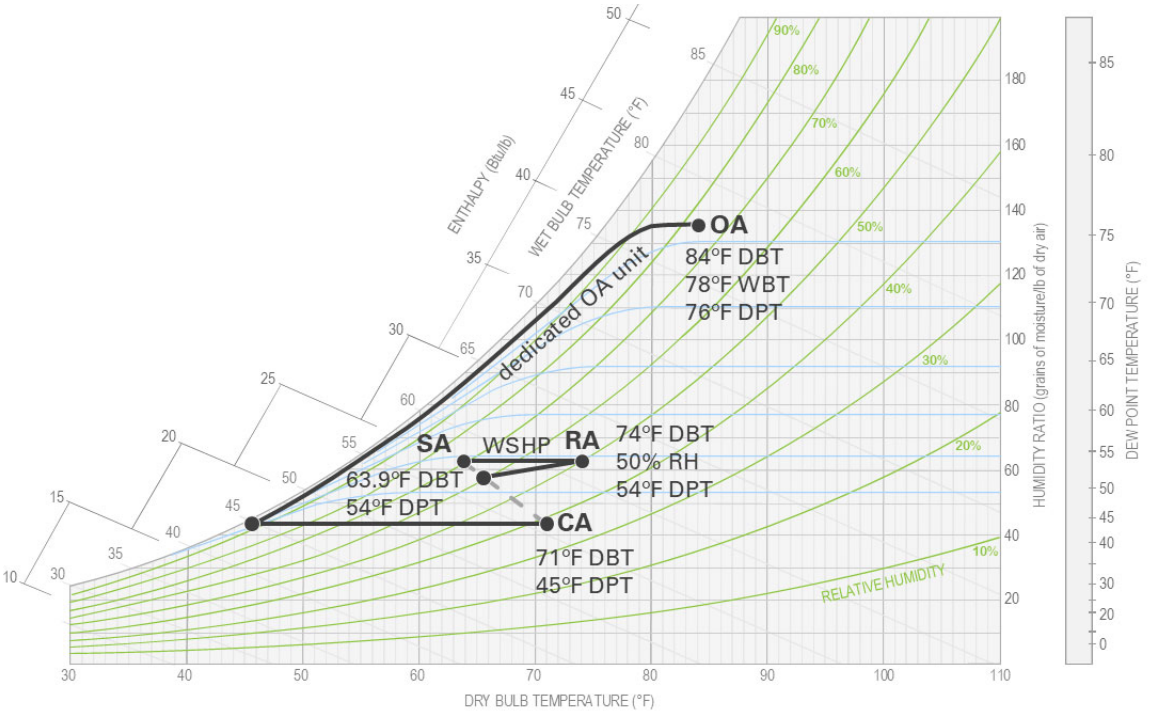
Figure 74. Dehumidification performance of a dedicated OA system at peak dry-bulb condition



At the part-load, peak dew-point condition (Figure 75), the dedicated OA unit continues to deliver the outdoor air at the same conditions, 71°F DBT (21.7°C DBT) and 45°F DPT (7.2°C DPT). Because the sensible cooling load in the zone is lower, the compressor in the WSHP cycles to maintain zone temperature at setpoint. (This example assumes the WSHP includes a single, on/off compressor and a constant-speed fan.)

Even though the cycling compressor results in less dehumidification provided by the WSHP, the 450 cfm (0.21 m³/s) of outdoor air delivered by the dedicated OA unit is dry enough to offset the zone latent load and maintain the indoor dew point at 54°F (12.2°C).

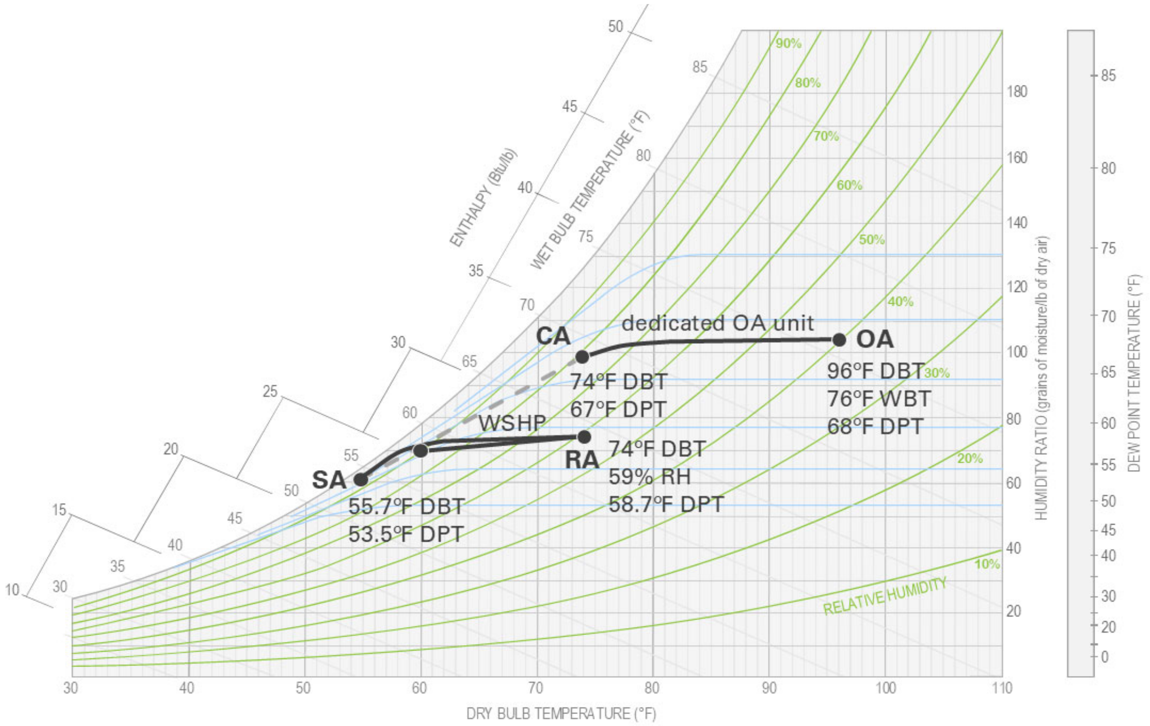
Figure 75. Dehumidification performance of a dedicated OA system at peak dew-point condition



Why not just cool the outdoor air to a neutral dry-bulb temperature without overcooling it to dehumidify to a low dew point? To demonstrate the effect of this design choice, consider if the same example classroom was served by a dedicated OA unit that cools the outdoor air to 74°F (23.3°C), without overcooling it to dehumidify to a low dew point. Although the dedicated OA unit eliminates the sensible cooling load associated with ventilation, it only offsets part of the latent ventilation load because the dew point of the conditioned OA is still much higher than the dew point in the zone. The remaining moisture in the conditioned OA must be removed from the classroom by the WSHP.

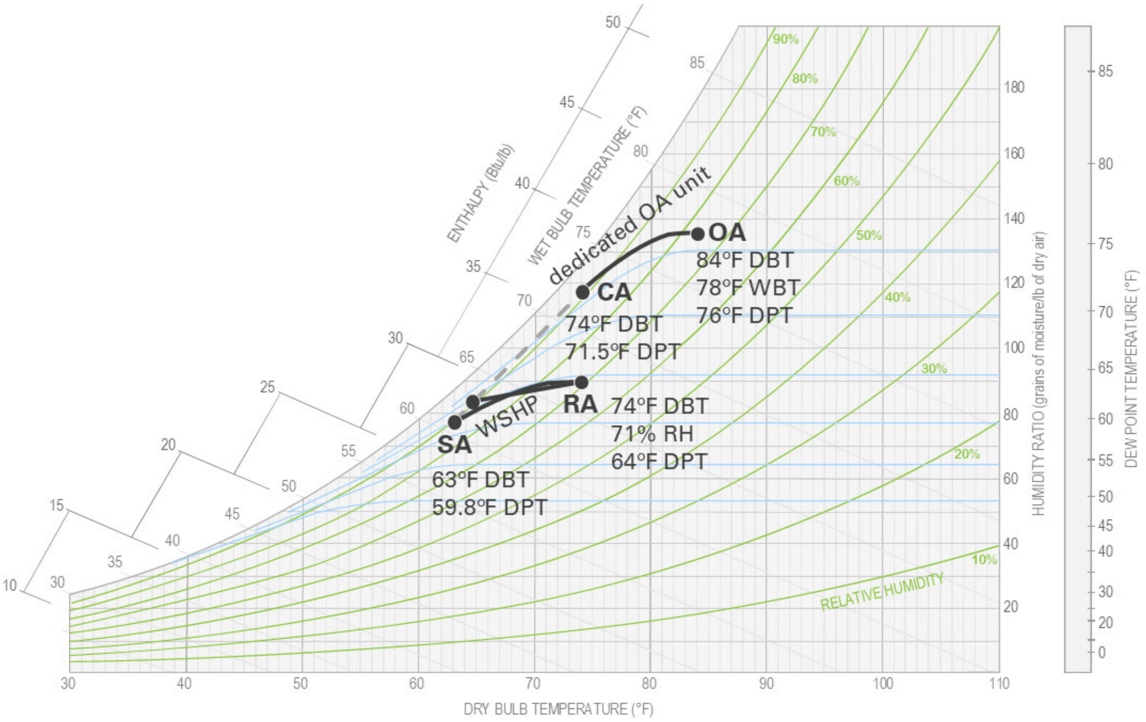
At the full-load, peak dry-bulb condition, the dedicated OA unit delivers conditioned OA (CA) that is at the same dry-bulb temperature as the zone, but at a much higher dew point (Figure 76). This adds a significant latent load to the classroom: moisture which must be removed by the WSHP. The WSHP still needs to cool the recirculated air to 55.7°F DBT (13.1°C DBT) to offset the zone sensible cooling load. At this condition, the resulting zone dew point is 58.7°F (14.8°C).

Figure 76. Dehumidification performance of neutral-temperature conditioned air, without overcooling to dehumidify, at peak dry-bulb condition



At the part-load, peak dew-point condition, the dedicated OA unit continues to cool the outdoor air to the same 74°F DBT (23.3°C DBT) temperature, still adding a significant latent load to the classroom. Because the sensible cooling load in the zone is lower, however, the compressor in the WSHP cycles to maintain zone temperature, and provides less dehumidification. The resulting zone dew point is 64°F (17.8°C) (Figure 77).

Figure 77. Dehumidification performance of neutral-temperature conditioned air, without overcooling to dehumidify, at peak dew-point condition



The preceding example demonstrates that supplying conditioned OA at a neutral dry-bulb temperature, without subcooling it to reduce its moisture content, provides less dehumidification than a system without a dedicated OA unit to begin with (see Figure 67, p. 100).

After-hours dehumidification

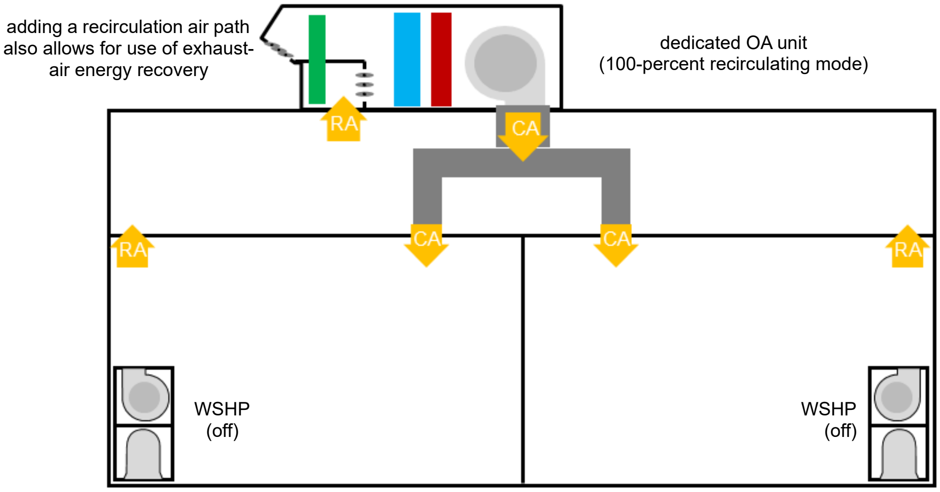
Controlling humidity is not only a priority when the building is occupied. When indoor humidity rises too high during unoccupied periods (after hours), one option could be to turn on the WSHP serving the affected zone, and dehumidify recirculated air. However, there is typically very little sensible cooling load in the zone during unoccupied periods, so the result might be overcooling the zone—unless the WSHP is equipped with hot-gas reheat. Also, this requires turning on the water-circulating pump and possibly the cooling tower.

If a dedicated OA system is used, when after-hours dehumidification is needed, the dedicated OA unit could be turned on to deliver dehumidified air to the zone(s). Because there is typically very little sensible cooling load in the zone, the dehumidified air may need to be reheated to avoid overcooling the zone.

If the dedicated OA system delivers the conditioned outdoor air directly to each zone (Figure 38, p. 59), it could be used for after-hours humidity control without needing to operate the local water-source heat pump(s). However, if the conditioned OA is ducted directly to the intake of each WSHP, then the fan in the WSHP may need to operate in conjunction with the dedicated OA unit.

If the dedicated OA unit includes a recirculating-air path, the outdoor-air damper could be closed and the return-air damper opened (Figure 78). This avoids the energy penalty associated with conditioning unneeded outdoor air during after-hours dehumidification. If the dedicated OA unit includes exhaust-air energy recovery, it likely already has a return-air path, and may only require the addition of a return-air damper to allow for after-hours humidity control.

Figure 78. Dedicated OA system used for after-hours dehumidification



If the dedicated OA unit is connected to the water distribution loop—as would be the case for a water-cooled DX unit or a water-to-water heat pump, for example—the water-circulating pump, and possibly the cooling tower, would also need to operate to provide after-hours dehumidification.

Humidification

Some buildings, or specific areas within a building, require maintaining *minimum* humidity levels for comfort or process requirements. If a building with a WSHP system requires humidification, consider the following:

- *Location of the humidifier*

  If humidification is needed for comfort, consider locating the humidifier in the dedicated outdoor-air system, downstream of the heat source. During cold (and dry) weather, outdoor air brought in for ventilation does not have much capacity to hold additional moisture. After the outdoor air has been warmed by a gas-fired burner or heating coil, it has a much greater capacity to absorb moisture.

  If humidification is needed for only a few specific zones within the building, consider locating a humidifier in ductwork downstream of each WSHP serving those zones that require humidification. This avoids the energy needed to humidify all the zones, but does require space and maintenance near the occupied zones.

- *Avoid oversizing the humidification equipment*

  An oversized humidifier typically results in unstable control, with large swings in humidity levels. In an application where humidification is provided for comfort, avoid the use of overly conservative assumptions or safety factors. During cold weather, adding too much moisture also increases the likelihood of moisture-related problems in the building envelope, where there are likely to be surface temperatures that are below the indoor dew point.

  Sizing the humidifier can be particularly challenging if the WSHP includes an airside economizer, as might be the case with a rooftop-style WSHP. Typically, when the outdoor air is driest (at the winter design condition, for example), the OA damper is closed to its minimum position. But, at other times during the year, the airside economizer may open the OA damper further when it is still cold (and dry) outdoors. While the outdoor air may not be as dry as it is at the winter design condition, the system is introducing a larger quantity of outdoor air. The design engineer should estimate the humidification load at both conditions.

- *Follow the manufacturer's recommendations for downstream absorption distance and maximum relative humidity*

  If the moisture is not fully absorbed by the air stream, it can cause downstream surfaces to get wet. The humidifier should be far enough upstream of elbows, junction, sensors, or dampers to allow for sufficient absorption. Absorption distances are shorter with lower air velocities.

For more information on the various types of humidification equipment, including sizing and application, refer to the ASHRAE *Humidity Control Design Guide for Commercial and Institutional Buildings* and Chapter 22, "Humidifiers," in the 2024 *ASHRAE Handbook—HVAC Systems and Equipment* (www.ashrae.org).

Energy Efficiency

Decisions made solely, or primarily, based on installed (first) cost often ignore such factors as energy use, maintenance requirements, or expected life of the equipment. Life-cycle cost includes the total cost of owning and operating the HVAC system over a given period of time. This includes installed cost, energy cost, maintenance cost, replacement cost, and any other known and expected costs.

As mentioned in other parts of this manual, WSHP systems are, in many ways, inherently energy efficient. A heat pump is an efficient method of heating, and when some zones require cooling at the same time other zones require heating, the heat recovery nature of the system saves energy by reducing the operating time of the cooling tower and boiler. In addition, various control strategies and design options (including ground coupling) provide the opportunity to further reduce the energy use of this type of system.

Minimum efficiency requirements

For more information, refer to ANSI/ASHRAE/IESNA Standard 90.1, *Energy Standard for Buildings, Except Low-Rise Residential Buildings*, and the *Standard 90.1 User's Manual*, both available for purchase at www.ashrae.org.

Many state and local building codes include requirements for minimum levels of energy efficiency. Some of these requirements relate to the efficiency of specific equipment (such as water-source heat pumps, cooling towers, boilers, and dedicated outdoor-air units), while others relate to the design and control of the overall HVAC system.

ANSI/ASHRAE/IESNA Standard 90.1, *Energy Standard for Buildings, Except Low-Rise Residential Buildings*, is the basis for many of these local codes. Its purpose is "to establish the minimum energy efficiency requirements for buildings" and, as such, it addresses the entire building. The HVAC section of ASHRAE 90.1 includes a large number of requirements related to system design, control, and construction. However, this section of the manual will focus on only a few of the HVAC-related requirements that are of specific interest to designers of typical WSHP systems.

Note: Because ASHRAE 90.1 is under continuous maintenance, it can change frequently. This manual is based on the 2022 published version of the standard. Refer to the most current version for specific requirements.

Minimum equipment efficiencies

Section 6.4.1 of ASHRAE 90.1 contains minimum efficiency requirements for various types of HVAC equipment, including water-source heat pumps, boilers, cooling towers, and dedicated outdoor-air units.

Table 20 includes the minimum efficiency requirements for water-source heat pumps, based on operating conditions defined by AHRI/ASHRAE/ISO Standard 13256-1, *Water-Source Heat Pumps—Testing and Rating for Performance—Part 1: Water-to-Air and Brine-to-Air Heat Pumps*. ASHRAE 90.1 includes minimum efficiency requirements for heat pumps used in conventional boiler/tower systems ("water loop"), as well as for ground-coupled ("ground loop") and ground-water systems.

For heat pumps, meeting both the cooling and heating efficiencies is mandatory, whether the prescriptive or Energy Cost Budget (ECB) method of compliance is used. For example, a water-source heat pump ("water-to-air, water loop") with a cooling capacity less than 17,000 Btu/hr (5 kW), or about 1.4 tons, must have a cooling efficiency of 12.2 EER (3.58 COP_C) or higher. In addition, the heating efficiency must be 4.3 COP_H or higher.

Table 20. Minimum equipment efficiencies for water-source heat pumps

Equipment type	Size category	Entering fluid temperature	Minimum efficiency
water-to-air, water loop (cooling mode)	< 17,000 Btu/hr (5 kW)	86°F (30°C)	12.2 EER (3.58 COP_C)
	≥ 17,000 Btu/hr (5 kW) and < 65,000 Btu/hr (19 kW)	86°F (30°C)	13.0 EER (3.81 COP_C)
	≥ 65,000 Btu/hr (19 kW) and < 135,000 Btu/hr (40 kW)	86°F (30°C)	13.0 EER (3.81 COP_C)
water-to-air, water loop (heating mode)	< 135,000 Btu/hr (40 kW) in terms of cooling capacity	68°F (20°C)	4.3 COP_H
brine-to-air, ground loop (cooling mode)	< 135,000 Btu/hr (40 kW)	77°F (25°C)	14.1 EER (4.13 COP_C)
brine-to-air, ground loop (heating mode)	< 135,000 Btu/hr (40 kW) in terms of cooling capacity	32°F (0°C)	3.2 COP_H
water-to-air, groundwater (cooling mode)	< 135,000 Btu/hr (40 kW)	59°F (15°C)	18.0 EER (5.28 COP_C)
water-to-air, groundwater (heating mode)	< 135,000 Btu/hr (40 kW) in terms of cooling capacity	50°F (10°C)	3.7 COP_H

Source: Table 6.8.1-15 from ASHRAE Standard 90.1-2022. © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org

For more information on AHRI Standard 920 and the ISMRE2 and ISCOP2 metrics, refer to the Trane *Engineers Newsletter* titled "AHRI 920: Rating Standard for DX Dedicated Outdoor-Air Units" (ADM-APN060-EN).

ASHRAE 90.1 also includes minimum efficiency requirements for DX dedicated outdoor-air units, based on the Integrated Seasonal Moisture Removal Efficiency (ISMRE2) and Integrated Seasonal Coefficient of Performance (ISCOP2) defined by AHRI Standard 920, *Performance Rating of DX Dedicated Outdoor Air System Units.* ASHRAE 90.1 includes separate efficiency requirements for units equipped with exhaust-air energy recovery versus those without.

Maximum allowable fan system power

ASHRAE 90.1 defines the **fan system design conditions** as the “operating conditions that can be expected to occur during normal system operation that result in the highest supply airflow rate to conditioned spaces served by the system, other than during air economizer operation.”

Because fan energy use depends heavily on the design of the air distribution system, it is difficult to prescribe a minimum efficiency requirement for a fan. Section 6.5.3.1.1 of ASHRAE 90.1 prescribes a limit to the allowable fan system power, but this limit only applies if the fan system has a total motor nameplate power exceeding 5 hp (4 kW).

As defined by ASHRAE 90.1, fan system power is the sum of the power demand for “all fans that are required to operate at design conditions, to supply air from the heating or cooling source (e.g., coils) to the conditioned space(s) and return it to the source or exhaust it to the outdoors.”

According to this definition, each WSHP is considered a separate “fan system” because each has a heating and cooling source: the refrigerant-to-air heat exchanger. (This interpretation of the definition of “fan system” is confirmed by example 6-FFF in the *Standard 90.1-2019 User’s Manual*.) However, the “fan system” includes not only the fan inside the WSHP, but also the fan inside the dedicated OA unit and any central relief (exhaust) fans. (Individual exhaust fans with nameplate motor power of 1 hp [0.75 kW] or less are exempt and do not need to be included in any calculations.)

Example 6-FFF in the *Standard 90.1-2019 User’s Manual* clarifies that the fan in the dedicated OA unit and any central relief (exhaust) fans must be allocated to each WSHP on an airflow-weighted basis. In other words, if the outdoor airflow delivered to a given zone is 5 percent of the total airflow delivered by the dedicated OA unit, 5 percent of the dedicated OA unit fan motor power must be added to the fan motor power of the WSHP serving that zone.

For example, consider a wing of an elementary school building that contains eight classrooms. Each classroom is served by a separate WSHP, each equipped with a 3/4 hp (0.56 kW) fan motor. The dedicated OA system delivers 500 cfm (0.24 m^3/s) of conditioned outdoor air—for a total of 4000 cfm (1.9 m^3/s)—directly to each classroom. The dedicated OA unit is equipped with a 5 hp (3.7 kW) supply fan motor and a 1 hp (0.75 kW) exhaust fan motor.

As explained previously, each WSHP is considered a separate “fan system” because each has a separate cooling and heating source. The power of the two fans in the dedicated OA unit must be allocated to each heat pump on an airflow-weighted basis. For each classroom, 12.5 percent [500/4000 cfm (0.24/1.9 m^3/s)] of the dedicated OA unit fan power must be added to the fan power for each WSHP:

$$0.75 \text{ hp} + (0.125 \times 5 \text{ hp}) + (0.125 \times 1 \text{ hp}) = 1.5 \text{ hp}$$

$$[0.56 \text{ kW} + (0.125 \times 3.7 \text{ kW}) + (0.125 \times 0.75 \text{ kW}) = 1.1 \text{ kW})]$$

For this example system, even with the dedicated OA unit fans allocated, the total fan motor nameplate power for each WSHP “fan system” is 1.5 hp (1.1 kW), which is less than the 5 hp (4 kW) threshold in Section 6.5.3.1.1. Therefore, this system does not need to comply with the maximum allowable fan power defined by Section 6.5.3.1.1.

For most horizontal, vertical, console, vertical stack, and smaller rooftop WSHP configurations, the total fan system motor nameplate power will likely be smaller than this 5 hp (4 kW) threshold, making them exempt from the current ASHRAE

90.1 limit on fan power. For larger heat pumps, however, the fan power limitation may apply.

When the maximum allowable fan power limit does apply, ASHRAE 90.1 includes two options for compliance (Table 21), depending on whether the fan system is constant volume or variable volume. Historically, most WSHPs have been equipped with a constant-speed fan, so they would be classified as constant-volume fan systems. However, today some models are equipped with an electronically commutated motor that can be used to vary airflow delivered to the zone (see "Electronically commutated motor," p. 21). These would be considered variable-volume fan systems.

Table 21. Fan system power limitation

	Constant volume	Variable volume
Option 1: Allowable motor nameplate power	$hp \le CFM_{supply} \times 0.0011$ $(kW \le L/s_{supply} \times 0.0017)$	$hp \le CFM_{supply} \times 0.0015$ $(kW \le L/s_{supply} \times 0.0024)$
Option 2: Allowable fan input (brake) power	$bhp \le CFM_{supply} \times 0.00094 + A$ $(kW_i \le L/s_{supply} \times 0.0015 + A)$	$bhp \le CFM_{supply} \times 0.0013 + A$ $(kW_i \le L/s_{supply} \times 0.0021 + A)$

Source: Table 6.5.3.1-1 of ASHRAE Standard 90.1-2022. © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org

Option 1 is based on motor nameplate power. It is easier to apply, but not as flexible. To comply using Option 1, the sum of the motor nameplate powers for all fans that operate at peak design (cooling) conditions must be no greater than the value listed in Table 21.

For example, if the design supply airflow for a rooftop-style WSHP is 7600 cfm (3.6 m^3/s or 3600 L/s), the total allowable nameplate motor power for the fan system is 8.4 hp (6.1 kW).

Allowable Nameplate Motor Power = 7600 cfm x 0.0011 = 8.4 hp

(Allowable Nameplate Motor Power = 3600 L/s x 0.0017 = 6.1 kW)

This limit applies to the sum of all fans that operate at peak design (cooling) conditions for this particular system.

Option 2 is based on input power to the fan shaft (brake horsepower). To comply using Option 2, the sum of the fan input powers for all fans that operate at peak design conditions must be no greater than the value listed in Table 21. This fan power limitation contains the following adjustment to account for special filters and other devices.

$$A = \Sigma\,(PD \times CFM_{device} / 4131)$$

$$[A = \Sigma\,(PD \times L/s_{device} / 650{,}000)]$$

where,
PD = pressure drop adjustment for each device (Table 19), in. H_2O (Pa)
CFM_{device} (L/s_{device}) = design airflow through each device (Table 19), cfm (L/s)

Table 22. Fan power limitation pressure drop adjustments

Device	Adjustment (PD)
Fully ducted return and/or exhaust air systems	0.5 in. H_2O (125 Pa) or 2.15 in. H_2O (535 Pa) for laboratory or vivarium systems
Airflow control devices in the return and/or exhaust air path	0.5 in. H_2O (125 Pa)
Exhaust filters, scrubbers, or other exhaust air treatment	Pressure drop through device at fan system design condition
MERV 9 through 12 particulate filtration	0.5 in. H_2O (125 Pa)
MERV 13 through 15 particulate filtration	0.9 in. H_2O (225 Pa)
MERV 16 and higher particulate filtration, or electronically enhanced filters	2 x clean filter pressure drop at fan system design condition
Carbon and other gas-phase air cleaners	Clean filter pressure drop at fan system design condition
Biosafety cabinet	Pressure drop through device at fan system design condition
Exhaust-air energy recovery device (e.g. wheel, heat pipe, fixed-plate heat exchanger), other than coil loop	(2.2 x Enthalpy Recovery Ratio) - 0.5 in. H_20 [(550 x Enthalpy Recovery Ratio) - 125 Pa] for each air stream
Coil loop	0.6 in. H_20 (150 Pa) for each air stream
Evaporative humidifier/cooler in series with another cooling coil	Pressure drop through device at fan system design condition
Sound attenuation section for fans serving spaces < NC35	0.15 in. H_2O (38 Pa)
Exhaust system serving fume hoods	0.35 in. H_20 (85 Pa)
Laboratory and vivarium exhaust systems in high-rise buildings	0.25 in. H_2O (60 Pa) for each 100 ft (30 m) of vertical duct exceeding 75 ft (25 m)

Source: Table 6.5.3.1-2 of ASHRAE Standard 90.1-2022. © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org

Consider if the same 7600-cfm (3600-L/s) rooftop-style WSHP includes a total-energy wheel with an Enthalpy Recovery Ratio of 0.70. At design cooling conditions, the outdoor airflow through the supply-side of the wheel is 3000 cfm (1400 L/s) and the airflow through the exhaust-side of the wheel is 2400 cfm (1100 L/s). For this example device, the pressure drop adjustment (PD) is 1.04 in. H_2O (260 Pa).

$$PD = (2.2 \times 0.70) - 0.5 \text{ in. } H_2O = 1.04 \text{ in. } H_2O$$

$$(PD = [550 \times 0.70] - 125 \text{ Pa} = 260 \text{ Pa})$$

The factor (A), which sums adjustments for both the supply-side and exhaust-side air streams, is 1.36 bhp (1.0 kW).

$$A = (1.04 \text{ in. } H_2O \times 3000 \text{ cfm} / 4131) + (1.04 \text{ in. } H_2O \times 2400 \text{ cfm} / 4131) = 1.36 \text{ bhp}$$

$$(A = [260 \text{ Pa} \times 1400 \text{ L/s} / 650{,}000] + [260 \text{ Pa} \times 1100 \text{ L/s} / 650{,}000] = 1.0 \text{ kW})$$

Therefore, the total allowable fan input power (brake horsepower) for the fan system is adjusted to 8.5 bhp (6.4 kW):.

Allowable Fan Input Power = 7600 cfm x 0.00094 + 1.36 = 8.5 bhp

(Allowable Fan Input Power = 3600 L/s x 0.0015 + 1.0 = 6.4 kW)

Economizers

Section 6.5.1 of ASHRAE 90.1 states that either an air or fluid economizer is required on “each cooling system.” For all climate zones (except 0A, 0B, 1A and 1B), this section requires an economizer if the rated cooling capacity of the WSHP is 54,000 Btu/hr (16 kW), which equates to 4.5 tons, or larger. Most horizontal, vertical, vertical-stack, and console heat pumps are smaller than this limit, in which case an economizer would not be required. (This interpretation is confirmed by example 6-KK in the *Standard 90.1-2019 User’s Manual*.) Note that if the WSHP is installed outside the building, such as a rooftop-style WSHP, this threshold is lowered to 33,000 Btu/hr (9.7 kW), or 2.75 tons.

Like other requirements in the standard, there are several exceptions provided. For instance, an economizer can be avoided (per exception 10) if the heat pump cooling efficiency meets or exceeds the requirements in Table 6.5.1-2 of the standard. For example, in Climate Zone 2A, if the rated cooling efficiency (EER or COP_C) of the WSHP is at least 17 percent higher than the minimum efficiency listed in Table 6.8.1-15 of the standard, the economizer can be eliminated.

When required, either an air or fluid (waterside) economizer can be used, provided that it meets the requirements stated in Section 6.5.1 of the standard. The design team might consider one of the following potential solutions:

- If the WSHP has an outdoor-air intake (as might be the case with a rooftop-style WSHP), an air economizer could be implemented.
- If ventilation is provided by a dedicated OA system, that system could be oversized to “provide up to 100 percent of design supply air quantity as outdoor air for cooling,” as required by Section 6.5.1.1.1. For most applications, this is probably not desirable since it would require much larger ductwork and larger fans.
- A “pre-cooling” fluid (waterside) economizer coil could be included in the WSHP itself (see discussion below).
- Or the design team could choose to comply using the Energy Cost Budget (Section 10) method, rather than following the prescriptive requirements of ASHRAE 90.1.

For a WSHP system, some engineers are concerned that Section 6.5.1.4 (“Economizer Heating System Impact”) may not allow for the use of a “pre-cooling” waterside economizer coil to meet this requirement for an economizer. Section 6.5.1.4 states that:

> “HVAC system design and economizer controls shall be such that economizer operation does not increase the building heating energy use during normal operation.”

In order for a pre-cooling waterside economizer coil (see Figure 109, p. 170) to provide enough capacity to meet the requirements of Section 6.5.1.2, the loop water temperature may need to be allowed to drift colder than normal—below 60°F (16°C), for example. While this colder loop temperature will allow for waterside economizing in those zones that require cooling, the colder loop water will decrease the efficiency of any WSHP compressors that are operating in the heating mode.

In June 2012, the ASHRAE Standard 90.1 committee issued an official interpretation (IC 90.1-2010-15) to clarify that “pre-cooling” waterside economizer coils *can* be used in WSHP systems to meet the requirements of Section 6.5.1.

But Section 6.5.1.4 explicitly refers to the overall “building heating energy use,” not just energy used by individual heat pump compressors operating in heating mode. For a WSHP system, the building heating energy use is the sum of the energy used by compressors operating in heating mode plus any energy used by the boiler that is connected to the water distribution loop.

When a zone is cooled by a water-source heat pump that is equipped with a pre-cooling waterside economizer, the heat removed from the zone is rejected to the water distribution loop. This heat rejected to the loop reduces the amount of heat that the boiler must add to the loop. This corresponding reduction in boiler energy use must be considered when evaluating the impact of a waterside economizer on the overall building heating energy use.

Table 23 illustrates the impact of a pre-cooling waterside economizer on overall building heating energy use, using example 5-ton (18-kW) water-source heat pumps.

Table 23. Example impact of a pre-cooling fluid (waterside) economizer on overall building heating energy use

	Entering water temperature from loop	
	45°F (7.2°C)	**55°F (12.8°C)**
Heating load in Zone 1 (heating mode)	61,000 Btu/hr (17.9 kW)	61,000 Btu/hr (17.9 kW)
WSHP heating efficiency, COP_H	3.70	4.04
WSHP compressor energy use (heating load / COP_H)	4.83 kW	4.43 kW
WSHP heat of compression (heating load / COP_H)	16,500 Btu/hr (4.8 kW)	15,100 Btu/hr (4.4 kW)
Heat extracted from loop (heating load – heat of compression)	44,500 Btu/hr (13.0 kW)	45,900 Btu/hr (13.4 kW)
Cooling load in Zone 2 (cooling mode)	5,000 Btu/hr (1.5 kW)	5,000 Btu/hr (1.5 kW)
Heat rejected to loop (by waterside economizer coil[1])	5,000 Btu/hr (1.5 kW)	5,000 Btu/hr (1.5 kW)
Heat added to loop by boiler (heat extracted from loop – heat rejected to loop)	39,500 Btu/hr (11.6 kW)	40,900 Btu/hr (12.0 kW)
Boiler energy use (heat added to loop by boiler / 80 percent efficiency)[2]	49,400 Btu/hr (14.5 kW)	51,100 Btu/hr (15.0 kW)
Building heating energy use (WSHP compressor energy use + boiler energy use)	65,900 Btu/hr (19.3 kW)	66,200 Btu/hr (19.4 kW)

[1] *This analysis assumes the waterside economizer coil is capable of offsetting the 5,000 Btu/hr (1.5 kW) cooling load in Zone 2 with either entering water temperature—45°F (7.2°C) or 55°F (12.8°C). The waterside economizer coil may need to be larger to offset the entire cooling load with warmer water.*

[2] *This example analysis assumes a gas-fired boiler with 80 percent efficiency. If, however, an electric boiler (with 100 percent efficiency) is used, the overall building energy use would be equal at these two loop temperatures.*

The colder loop water temperature decreases compressor efficiency (COP_H) for those heat pumps operating in heating mode. However, the heat of compression increases (a less efficient compressor requires more power and generates more heat) so that less heat must be extracted from the loop to offset the same zone heating load. This means that less heat must be added to the loop by the boiler. In addition, those heat pumps operating in cooling mode using a waterside economizer coil are rejecting heat to the loop, which further reduces the amount of heat that must be added by the boiler.

In this example, allowing the loop water temperature to drop from 55°F (12.8°C) to 45°F (7.2°C) has the following results:

- WSHP heating efficiency decreases from 4.04 COP_H to 3.70 COP_H
- WSHP compressor power increases from 4.43 kW to 4.83 kW, to offset the heating load in Zone 1
- Heat of compression increases from 15,100 Btu/hr (4.4 kW) to 16,500 Btu/hr (4.8 kW), due to the compressor operating less efficiently
- Amount of heat that must be extracted from the loop (zone heating load – heat of compression) decreases from 45,900 Btu/hr (13.4 kW) to 44,500 Btu/hr (13.0 kW)

In addition, the heat pump serving Zone 2 is operating in the cooling mode, using a waterside economizer coil. The 5,000 Btu/hr (1.5 kW) cooling load is rejected to the loop, which further reduces the amount of heat that must be added to the loop by the boiler.

Operating the loop at a colder water temperature does increase the energy used by those WSHP compressors operating in the heating mode, but it decreases the energy used by the boiler. In this example, the overall building heating energy use is less when the system operates at 45°F (7.2°C) with a waterside economizer, than when it operates at 55°F (12.8°C). Therefore, a pre-cooling waterside economizer coil may be used to meet the requirements stated in Section 6.5.1 of ASHRAE 90.1, without violating Section 6.5.1.4.

Demand-controlled ventilation

If the system is equipped with an air economizer (or a modulating outdoor-air damper) or if the design system-level outdoor airflow (V_{ot}) is greater than 3000 cfm (1.4 m^3/s), then Section 6.4.3.8 of ASHRAE 90.1 requires some method of demand-controlled ventilation (DCV) for any zone larger than the threshold listed in Table 24.

Table 24. DCV threshold requirement

	Occupant Component of Outdoor Airflow from Standard 62.1, cfm/1000 ft^2 (L/s/100 m^2) [1]					
	100 to 199 (50 to 99)	200 to 399 (100 to 199)	>= 400 (>= 200)	100 to 199 (50 to 99)	200 to 399 (100 to 199)	>= 400 (>= 200)
	Minimum Floor Area in ft^2 (m^2) for which DCV is Required					
Climate Zone	**Areas without Exhaust-Air Energy Recovery**			**Areas with Exhaust-Air Energy Recovery [2]**		
7, 8	400 (40)	200 (20)	150 (15)	800 (80)	400 (40)	250 (25)
5A, 6A, 6B	600 (60)	250 (25)	150 (15)	1400 (140)	900 (90)	400 (40)
0A, 0B, 1B, 3A, 4A, 5B, 5C	800 (80)	400 (40)	250 (25)	2000 (200)	1000 (100)	500 (50)
2A, 2B, 4C	1100 (110)	600 (60)	300 (30)	2300 (230)	1100 (110)	600 (60)
3B, 4B	1500 (150)	700 (70)	400 (40)	5200 (520)	2350 (235)	1250 (125)
1A	2400 (240)	1100 (110)	600 (60)	5800 (580)	2600 (260)	1400 (140)
3C	7000 (700)	3000 (300)	1700 (170)	12000 (1200)	6000 (600)	3000 (300)

Source: Table 6.4.3.8 of ASHRAE Standard 90.1-2022. © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org

1. This is calculated by multiplying the Default Occupant Density by the People Outdoor Air Rate (R_p) from Table 6-1 in ASHRAE Standard 62.1-2022.

3. Where exhaust-air energy recovery is required by Section 6.5.6.1 in ASHRAE Standard 90.1-2022.

For a K-12 classroom (age 9+), ASHRAE Standard 62.1 lists the People Outdoor Air Rate (R_p) as 10 cfm/person (5 L/s/person) and the Default Occupant Density as 35 people per 1000 ft^2 (100 m^2). Multiplying these values together yields 350 cfm/1000 ft^2 (175 L/s/100 m^2). For climate zone 4A, DCV is required if the floor area of this classroom is 400 ft^2 (40 m^2) or larger. But if the system requires exhaust-air energy recovery (per Section 6.5.6.1 of ASHRAE 90.1), then this threshold increases to 1000 ft^2 (100 m^2).

If outdoor air is brought directly into a WSHP—as might be the case with a rooftop-style unit—the heat pump may already be equipped with an air economizer (or modulating outdoor-air damper). In this case, it may be required to employ some method of demand-controlled ventilation if it serves a zone that meets the minimum floor area criteria mentioned above.

In most WSHP systems, however, the outdoor air required for ventilation is typically conditioned and delivered by a dedicated OA system. The dedicated OA unit is probably not equipped with an air economizer (or modulating outdoor-air damper). Therefore, the system-level intake airflow (V_{ot}) would need to be greater than 3000 cfm (1.4 m^3/s) for DCV to be required. And then, DCV would only need to be implemented in those zones that meet the minimum floor area criteria mentioned above.

However, like other requirements in the standard, there are several exceptions provided. The following exceptions are the most likely to apply to a typical WSHP system:

- **Exception 3**

  Correctional cells, daycare sickrooms, science labs, barbers, beauty and nail salons, and bowling alley seating areas are all exempt from this DCV requirement.

- **Exception 4**

  Spaces where the requirements of ASHRAE/ASHE Standard 170, *Ventilation of Health Care Facilities*, or other code or accreditation standard, do not allow for the reduction of outdoor airflow.

WSHP distribution loop design and control

ASHRAE 90.1 contains several requirements that impact the design and control of the water distribution loop.

When the water loop of a WSHP system contains both a heat rejecter (such as a cooling tower) and a heat adder (such as a hot-water boiler), Section 6.5.2.2.3 requires that the temperature deadband used for loop temperature control must be at least 20°F (12°C). For example, if the boiler is activated whenever the loop water temperature drops below 60°F (15.6°C), the cooling tower cannot be activated until the loop water temperature rises to at least 80°F (26.6°C). A smaller temperature deadband is allowed, however, if a system-level controller is used to optimize loop temperature control to minimize overall system energy use based on real-time operating conditions (see “Loop temperature optimization,” p. 199).

In climates that experience cold outdoor temperatures (climate zones 3 through 8), Section 6.5.2.2.3 requires the system be designed and controlled to minimize heat loss through the heat rejecter during cold weather:

- If a closed-circuit cooling tower is used, the cooling tower must be equipped with low-leakage dampers to prevent airflow when heat rejection is not needed. Alternatively, a valve can be installed to bypass water around the tower when heat rejection is not needed.
- If an open cooling tower is used in conjunction with an intermediate heat exchanger, the pump in the separate cooling tower loop must be turned off when heat rejection is not needed.
- If an open cooling tower is used without an intermediate heat exchanger (that is, the loop water passes directly through the open tower), a valve must be installed to bypass water around the tower when heat rejection is not needed.

Section 6.5.4.5.1 requires that each WSHP be equipped with two-position valve that automatically closes to shut off water flow whenever the compressor turns off (see “Isolation valves and flow-control devices,” p. 38). This means that the pump(s) must be capable of handling variable flow. (Heat pumps that are equipped with a waterside economizer coil are exempt from this requirement.)

If total pump system power exceeds 5 hp (3.7 kW), Section 6.5.4.5.2 requires the pumps be equipped with a variable-speed drive, or some other device that provides a comparable reduction in pump power at reduced water flow rates.

Cooling tower control

If a cooling tower is equipped with a motor of 5 hp (3.7 kW) or larger, Section 6.5.5.2 requires the fan to have controls that automatically modulate the fan speed to control the leaving fluid temperature. In many cases, this is accomplished by equipping the cooling tower fan with a variable-speed drive.

Exhaust-air energy recovery

For systems serving non-transient dwelling units, Section 6.5.6.1.1 requires exhaust-air energy recovery in all climate zones, except for 3C. However, in certain climate zones (0, 1, 2, 3, 4C, and 5C), dwelling units with gross conditioned floor area of 500 ft^2 (50 m^2) or smaller are exempt.

For systems serving spaces other than non-transient dwelling units, Section 6.5.6.1.2 states that exhaust-air energy recovery is required on “each fan system” in which the system design airflow exceeds the value listed in Table 25. For example, in climate zone 4A, if the system brings in 30 percent outdoor air, exhaust-air energy recovery is required if the design supply airflow rate for the system is 5500 cfm (2596 L/s) or higher.

Table 25. Exhaust-air energy recovery requirement for systems operating < 8000 hours/year*

Climate zone	Percent outdoor air at design supply airflow							
	≥10% and <20%	≥20% and <30%	≥30% and <40%	≥40% and <50%	≥50% and <60%	≥60% and <70%	≥70% and <80%	≥80%
	Design supply airflow for which exhaust-air energy recovery is required							
3B, 3C, 4B, 4C, 5B	not required	not required	not required	not required	not required	not required	not required	not required
0B, 1B, 2B, 5C	not required	not required	not required	not required	≥26,000 cfm (12271 L/s)	≥12,000 cfm (5663 L/s)	≥5000 cfm (2360 L/s)	≥4000 cfm (1888 L/s)
6B	≥28,000 cfm (13215 L/s)	≥26,500 cfm (12507 L/s)	≥11,000 cfm (5191 L/s)	≥5500 cfm (2596 L/s)	≥4500 cfm (2124 L/s)	≥3500 cfm (1652 L/s)	≥2500 cfm (1180 L/s)	≥1500 cfm (708 L/s)
0A, 1A, 2A, 3A, 4A, 5A, 6A	≥26,000 cfm (12271 L/s)	≥16,000 cfm (7551 L/s)	≥5500 cfm (2596 L/s)	≥4500 cfm (2124 L/s)	≥3500 cfm (1652 L/s)	≥2000 cfm (944 L/s)	≥1000 cfm (472 L/s)	≥120 cfm (60 L/s)
7, 8	≥4500 cfm (2124 L/s)	≥4000 cfm (1888 L/s)	≥2500 cfm (1180 L/s)	≥1000 cfm (472 L/s)	≥140 cfm (70 L/s)	≥120 cfm (60 L/s)	≥100 cfm (50 L/s)	≥80 cfm (40 L/s)

Source: Table 6.5.6.1.2-1 of ASHRAE Standard 90.1-2022. ©American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org
** There is a separate table in the standard (Table 6.5.6.1.2-2) for systems that operate for 8000 or more hours per year.*

If outdoor air is brought directly into the WSHP, as might be the case with a rooftop-style model, exhaust-air energy recovery might be required, depending on the climate zone, percent outdoor air, and design supply airflow.

In most WSHP systems, however, the outdoor air required for ventilation is typically conditioned and delivered by a dedicated (100 percent) outdoor-air system. In this case, for most climate zones, exhaust-air energy recovery is likely to be required, except for very small systems. The impact is that most dedicated OA units will be required to use exhaust-air energy recovery, unless they meet one of the exceptions listed in the standard.

Like other requirements in ASHRAE 90.1, there are several exceptions provided. For spaces other than non-transient dwelling units, the following exceptions are the most likely to apply to a typical WSHP system:

- **Exception 6**

  If the building has been designed with multiple exhaust/relief locations, such that the sum of the airflow rates exhausted or relieved within 20 ft (6 m) of each other is less than
  75 percent of the design outdoor airflow rate, then exhaust-air energy recovery is not required. Excluded from this sum is air used for another energy recovery system, exhaust air that ASHRAE/ASHE Standard 170 prohibits recovering energy from, or Class 4 exhaust air (as defined by ASHRAE Standard 62.1). This exception acknowledges the impracticality of recovering heat from multiple exhaust sources for a single intake.

- **Exception 7**

  If the dedicated OA system is needed for dehumidification in climate zone 0, 1, 2, 3, or 4, and the dedicated OA unit includes an air-to-air heat exchanger that is configured in series with (wrapped around) the cooling coil, then the system is not required to also provide exhaust-air energy recovery. The series configuration relies on the warm, entering outdoor air to be the source of heat that is transferred to reheat the cold, dehumidified air downstream of the cooling coil. If a separate air-to-air heat exchanger is used for exhaust-air energy recovery, during warm weather it will pre-cool the entering outdoor air, reducing the amount of heat that can be transferred by the series air-to-air heat exchanger. To qualify for this exception, the series air-to-air heat exchanger must result in a series energy recovery ratio (SERR) of 0.40 or higher.

Auxiliary heat control

If a WSHP is equipped with an internal electric resistance heater (see “Electric resistance heat for a “boiler-less” system,” p. 56), Section 6.4.3.5 requires that the unit-level controller must prevent the electric heater from operating whenever the heat pump compressor is capable of offsetting the heating load.

Examples 6-Z and 6-AA in the *Standard 90.1-2019 User’s Manual* discuss this requirement in more detail.

Opportunities to further reduce system energy use

While local building codes might include requirements for *minimum* levels of energy efficiency, many building owners desire even higher efficiency levels for their systems. In addition, programs like ENERGY STAR® (administered by the U.S. Environmental Protection Agency and Department of Energy [DOE]) and LEED® (Leadership in Energy and Environmental Design, created by the U.S. Green Building Council, a building industry coalition) have encouraged higher levels of energy efficiency in buildings.

Table 26 contains a list of several system design options and control strategies that can help further reduce the energy use of a WSHP system. This list is not intended to be all-encompassing, but focuses on those energy-savings strategies that are of specific interest to designers of typical WSHP systems.

Table 26. Potential energy-savings strategies for water-source heat pump systems

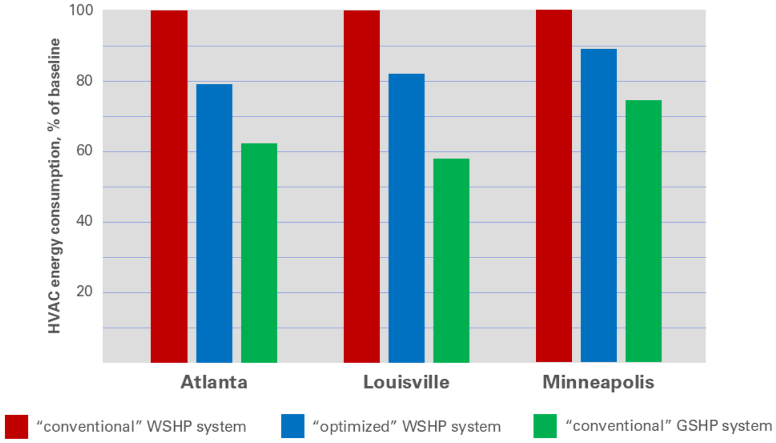
High-efficiency WSHPs	p. 15
Multiple-speed fan operation	p. 21
Cycle WSHP fan with compressor	p. 67 and p. 169
Waterside (or airside) economizer	p. 170
Water distribution loop	
Variable-flow pumping	p. 32
Increase pipe sizes to reduce pressure loss	p. 37
Heat rejection/heat addition	
Condensing boiler	p. 49
VFD on cooling tower fan(s)	p. 178 and p. 179
Thermal energy storage	p. 53
Ground-source system	p. 135
Dedicated outdoor-air system	
Deliver conditioned OA cold (rather than “neutral”) directly to spaces	p. 64
Precondition outdoor air with exhaust-air energy recovery	p. 71
System-level controls	
Night setback	p. 184
Occupancy sensor to enable occupied standby mode	p. 184
Optimal start**Water-source heat pumps**	p. 194
Demand-controlled ventilation	p. 195
Loop temperature optimization	p. 199

For more information on the Trane's TRACE® 3D Plus building analysis software, visit www.tranecds.com.

The impact of any energy-saving strategy on the operating cost of a specific system depends on climate, building usage, and utility costs. Building analysis software tools can be used to analyze these strategies and convert energy savings to operating cost dollars that can be used to make financial decisions.

Figure 79 shows the potential energy savings of using various strategies in an example office building with a WSHP system. The "conventional" WSHP system includes night setback control, variable-flow pumping with a VFD, and a total-energy wheel on the dedicated outdoor-air unit.

Figure 79. Example energy savings versus a baseline WSHP system



The "optimized" WSHP system adds optimal start and loop temperature optimization to the system-level controls, and the conditioned outdoor air is ducted directly to each zone (rather than to the inlet of each WSHP). Delivering the OA directly to each zone allows it to be delivered "cold" (rather than reheated to neutral) during the cooling season, and allows the fan in each heat pump to cycle off with the compressor when that zone does not require either cooling or heating.

For this example, the optimized WSHP system reduced the overall HVAC energy use by 20 percent for the building in Atlanta, by 15 percent in Louisville, and by 8 percent in Minneapolis. If the system is converted to a ground-coupled heat pump system, the energy savings increases to 37 percent in Atlanta, 40 percent in Louisville, and 24 percent in Minneapolis.

There is a real potential to save energy in WSHP systems through optimized system design and control strategies. This savings reduces operating costs for the building owner and can help in achieving points toward LEED® certification.

Acoustics

For more information on acoustical analysis, and the topic of HVAC acoustics in general, refer to the Trane application manual, *Acoustics in Air Conditioning* (ISS-APM001*-EN).

HVAC equipment creates sound and, in a well-designed application, that sound provides a positive effect on occupant comfort. That is, it provides an appropriate level of background sound for speech isolation or permits clear communication in a classroom. However, it is also possible for the sound from HVAC equipment to be considered noise because it disrupts the intended function of the space.

Equipment sound levels play a role in proper room sound levels, but a larger role is played by how the equipment is applied. One common approach to addressing HVAC acoustics is to use a fixed set of design practices on every project. With sufficient experience, this may be all a design engineer needs to create an installation that is free of noise problems. However, this may also unnecessarily inflate the installed cost of some projects, and may not provide sufficient attenuation on others.

On projects where acoustics is critical, or prior experience is lacking, the proper approach is to conduct an acoustical analysis early in the design process. Even a simple acoustical analysis can help achieve occupant satisfaction, while minimizing installed cost.

Defining an acoustical model

A simple acoustical model consists of a source, receiver, and path.

Source

The source is where the sound originates. The primary sound source in a WSHP system is the heat pump itself. However, secondary sources include the dedicated outdoor-air unit, water-circulating pumps, hot-water boilers, and cooling towers. This section will focus on the heat pumps; however, a quick review of the other equipment is recommended. Each source has a unique sound quality and level, and all of them play a role in determining the sound the receiver hears.

The foundation of an acoustical analysis is the equipment sound data. An accurate analysis depends on accurate sound data for the equipment. Indoor sound data for air moving equipment should be measured in accordance with AHRI Standard 260, *Sound Rating of Ducted Air Moving and Conditioning Equipment*. This will ensure that the sound data accurately reflects the contributions of all the sound sources, and accounts for the effects of the cabinetry.

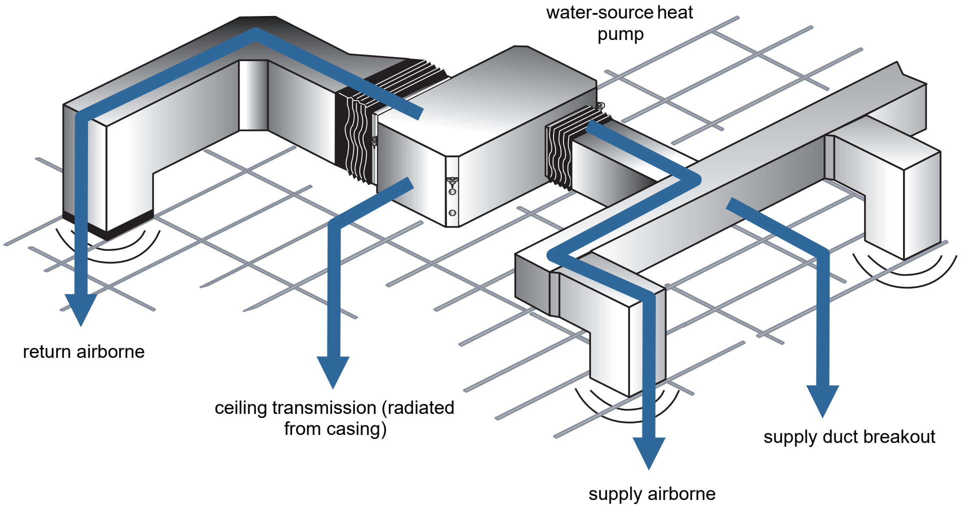
Receiver

The receiver is simply the location where the sound will be heard and judged against some defined criteria. This could be a private office, a conference room, an open office area, a classroom, a hotel guest or dormitory room, and so on. Typically a building will have many indoor receiver locations but often these can be grouped by similarity; e.g., a school may have many classrooms that use the same equipment and are dimensionally similar.

Path

The path is the route the sound travels from the source to the receiver. Sound from a single source may follow more than one path to the receiver location (Figure 80). For example, sound from the fan inside the WSHP follows the supply ductwork and enters the occupied space through the supply-air diffuser. That fan sound also travels back out through the WSHP inlet (against the direction of airflow), and then through the return air path into the space.

Figure 80. Typical sound paths from a horizontal WSHP



For each sound source it is necessary to determine the paths that the sound travels from the equipment to the receiver location. These sound paths are dependent on the type and location of the equipment, but generally fall into the following categories:

- *Airborne*

  Sound follows the airflow path. Supply airborne sound travels in the same direction as the supply air. Return airborne sound travels against the direction of airflow back through the return air path.

- *Duct breakout*

  Sound passes through walls of the ductwork, into the ceiling plenum, and then through the ceiling into the occupied space.

- *Radiated*

  Sound radiated from the casing of the equipment travels through whatever is between the equipment and the receiver location.

- *Structure-borne*

  This path differs from the others in that energy is transmitted through the framework of the building. This energy may come directly from the vibration of the sound source (a fan or compressor, for example), or may be airborne sound that is transferred to the structure.

An acoustical analysis consists of five basic steps:

Step 1: Set acoustical goals for the finished space.

It is critical to establish realistic acoustical goals for the occupied space at the outset of any project. There are always implicit (often subjective) expectations, and it is much easier if you understand these expectations before designing the HVAC system.

Sound goals will vary depending on how the space is expected to be used. Once the sound goals are understood, they can be stated using an appropriate descriptor, such as Noise Criteria (NC) or Room Criteria (RC) for indoor environments or dBA for classrooms or outdoor environments. Remember the following when defining the desired sound levels:

- As a general rule, lower sound levels cost more to achieve.
- All spaces in the building do not have the same acoustical requirements. Restrooms and hallways do not need to be as quiet as executive offices and conference rooms. A low-cost, quiet installation takes advantage of this fact.
- Successful acoustics requires a team effort, including the owner, HVAC design engineer, architect, equipment manufacturer, and installing contractor.

Step 2: Identify each sound path and its elements.

Paths are defined by the end points: the source location and the receiver location. There may be many receiver locations, depending on the installation, but the number can be reduced by determining the critical receiver locations.

In general, sound diminishes with distance, so the space closest to the unit will typically be the loudest. If adjacent spaces have sound targets that are considerably below the level required in the space closest to the unit, these spaces should also be analyzed. Common examples include conference rooms, executive offices, hotel guest or dormitory rooms, and classrooms.

After the critical receiver locations are defined, the sound paths from the source to each receiver can be identified.

Step 3: Perform a path-by-path analysis.

For more information on the Trane Acoustics Program (TAP™) acoustical analysis software, visit www.tranecds.com.

Once each path has been identified, individual elements can be analyzed for their contribution. For example, the supply airborne path includes various duct elements (elbows, straight duct, junctions, diffusers, and so on) and a room-correction factor. Algorithms available from ASHRAE can be used to calculate the acoustical effect of each duct element. The effect of changing an element, such as removing the lining from a section of ductwork, can also be calculated. Software tools make these algorithms easier to use.

Step 4: Sum the results to determine the acoustical performance of the installation.

Once the contributions of the individual paths for a particular receiver location are calculated, they must be added together to determine the total sound at the receiver. A unique sum is required for each “critical” receiver location.

Step 5: Compare the summations with the acoustical goals in the context of the project budget.

The sum of the sound paths for a particular receiver location is a prediction of the sound level at that location. If the sum is lower than the sound target for that location, the design does not need to be changed, although it may be reviewed for potential cost reductions.

If the estimate exceeds the sound target, the paths are reviewed to determine which paths are dominant. Alterations to the source and/or the path elements are then made to reduce the sound at the receiver location. This is typically an iterative process, comparing the acoustical effect of various alterations.

Once a design meets the acoustical goals for the project, everyone on the team must understand the work and cost required to implement the design. It may also be prudent to review the cost of meeting the acoustical goals and reconsider system layout alternatives or equipment options that were initially rejected due to cost.

Specific acoustical recommendations

For more acoustic- and vibration-related recommendations for water-source heat pump systems, refer to the ASHRAE manual, *A Practical Guide to Noise and Vibration Control*.

It is challenging to put together a list of specific acoustical practices that should be used on every project. Nearly everything on the list increases the installed cost—cost that may or may not be justified by the acoustical requirements. For this reason, an acoustical analysis is preferred to meet the acoustical goals at the lowest cost.

The following sections should be used to identify potential problems in WSHP systems. Consider both source attenuation and path attenuation to determine the most cost effective way to achieve the acoustical goals.

- Use flexible conduit and wiring connections to minimize vibration transmission to the building structure.
- For ducted applications, use canvas duct connectors to prevent vibration transmission to steel ducts.
- To avoid vibration transmission from ducts to the ceiling, do not attach ceiling wires to or through ducts.
- Finally, to minimize vibration transmission to the floor, install floor-mounted WSHPs on rubber (or cork) pads or on rubber-backed carpeting, with padding thickness of 3/8 to 1/2 in. (9.5 to 13 mm).

WSHP: console (or unit ventilator) models

From an acoustical perspective, console-style water-source heat pumps are simple to model but difficult to attenuate. Sound data provided by AHRI Standard 350, *Sound Rating of Non-Ducted Indoor Air-Conditioning Equipment*, is for the entire unit (discharge, inlet, and casing sound are combined). The sound “path” is simply the room correction that accounts for the size and absorptivity of the room where the WSHP is located.

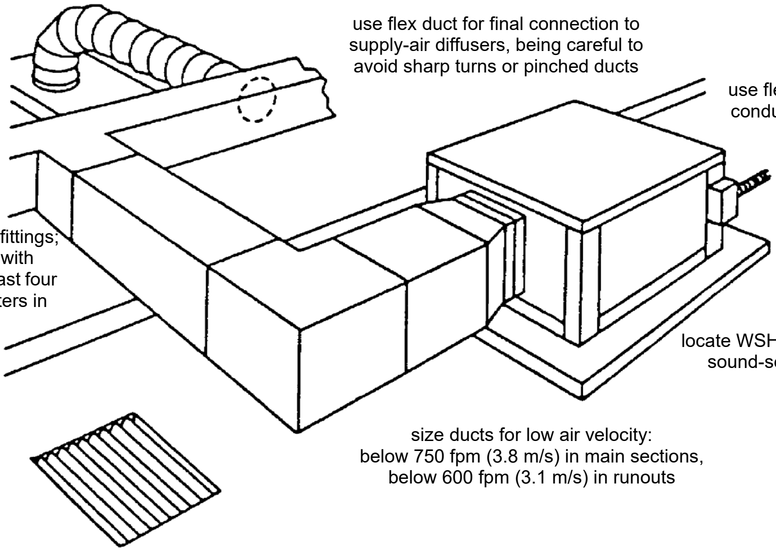
Units placed in “hard” rooms (i.e., rooms with an absorptivity factor less than 0.20) may benefit from adding absorptive materials on walls and ceilings. A typical “hard” (or live) room is characterized by a tile floor and masonry or gypsum board walls and ceiling.

Since very little is typically done to change the acoustical character of the room, the most effective ways to reduce sound in the occupied area are to choose a WSHP with a lower sound level—sometimes oversizing the unit and operating the fan at a slower speed helps—or choosing a different style WSHP that allows for greater flexibility in sound attenuation. If more than one heat pump serves the same area, moving them further away from each other may also help. However, avoid placing a WSHP in the corner of the room, since this would reduce the area for the sound to radiate.

WSHP: horizontal models

Horizontal water-source heat pumps are typically installed in a ceiling plenum with supply air ducted to diffusers and air returning from the space through a return-air grille and the open ceiling plenum (Figure 81). Sound data provided by AHRI Standard 260, *Sound Rating of Ducted Air Moving and Conditioning Equipment*, is separated by sound path (discharge, inlet, and casing radiated).

Figure 81. Recommendations for horizontal models



use flex duct for final connection to supply-air diffusers, being careful to avoid sharp turns or pinched ducts

use flexible duct and conduit connections

avoid close-coupled fittings; separate fittings with straight duct of at least four to five duct diameters in length

locate WSHP above a non-sound-sensitive area

size ducts for low air velocity:
below 750 fpm (3.8 m/s) in main sections,
below 600 fpm (3.1 m/s) in runouts

locate WSHP inlet at least 6 ft. (1.8 m) away from return-air grille

Sound paths and recommendations for this configuration are:

1) Casing radiated. Sound radiates from the casing of the heat pump (and out through the return-air inlet, if un-ducted) into the ceiling plenum. If the WSHP is placed directly over the occupied space, the sound will travel through the ceiling and return-air grille(s) into the occupied space.

The best way to reduce casing radiated sound is to place the WSHP over a non-sound sensitive area (a corridor, for example). Other options include adding a lined return duct to the inlet and/or placing an acoustical barrier under the WSHP. This barrier should be approximately twice the size of the WSHP footprint and have sufficient transmission loss to reduce the transmitted sound to acceptable levels. Placing a layer of absorptive material on top of the barrier will also help.

2) Supply airborne. Sound leaving the discharge of the WSHP travels down the supply ductwork, through the supply-air diffusers, and into the occupied space.

Supply airborne sound can be reduced by adding an acoustical liner to the supply ductwork. Acoustical analysis can be used to determine the required lining thickness and length of ductwork that needs to be lined. Another convenient way to add absorptive duct liner is to use lined flex duct as the final section that connects to the diffuser (Figure 81). Pay careful attention to the attachment of the flex to the diffuser; avoid sharp turns and pinched ducts at, and near, the diffuser.

Sizing ductwork for low velocity and low static pressure loss will reduce the sound produced by the fan. As a general rule, maintain the velocity in main duct sections below 750 fpm (3.8 m/s) and below 600 fpm (3.1 m/s) in runout duct sections.

Avoid close-coupled fittings in the ductwork, as these create high pressure loss and turbulence that generates sound. When possible, separate turns or fittings with straight duct sections that are at least four to five duct diameters in length (Figure 81). Check the static pressure drop against the static capability of the fan.

Finally, remember that supply-air diffusers generate sound that will be added to the sound coming from the WSHP. Select diffusers for at least 10 NC points below the desired NC level for the space. Avoid turbulence at the diffuser by placing the balancing damper near the duct take-off rather than near the diffuser.

3) Supply breakout. Sound traveling down the ductwork can also travel though the duct walls into the ceiling plenum, and then through the ceiling into the occupied space. This is generally only a problem on the main supply duct near the WSHP.

When breakout sound is a problem, it can be reduced by routing the main duct over a non-sound sensitive area (a corridor, for example), splitting the main duct into multiple, smaller ducts that are routed in different directions, or switching to rigid metal round duct.

4) Return airborne. The inlet of a horizontal-style WSHP can be either ducted or un-ducted. In an un-ducted system, the return air enters the ceiling plenum through the return-air grille and travels through an open ceiling plenum to the inlet of the WSHP. A fully ducted return system uses ductwork to connect the return-air grille directly to the WSHP inlet. In a partially ducted system, a section of return ductwork is connected to the WSHP inlet, but it does not connect to the return-air grille.

Avoid placing the return-air grille near the WSHP inlet (Figure 81). Moving the grille at least 6 ft. (1.8 m) away from the inlet will reduce the return airborne sound that enters the space. All ducted returns will benefit from acoustical lining. Size all return duct components for low velocity. Partially-ducted returns will benefit if the open end of the duct has no obstruction (such as a grille) and includes at least four equivalent duct diameters of straight duct prior to the WSHP inlet. An un-ducted return may benefit by connecting an acoustically lined elbow to the return-air grille with the opening facing away from the WSHP inlet.

WSHP: vertical units

Vertical water-source heat pumps are typically installed in a closet adjacent to the occupied space with supply air ducted to diffusers. Air returns from the space through a return-air grille or louver in the closet door (un-ducted) or through a ceiling-mounted return-air grille and the open ceiling plenum (ducted or un-ducted).

Sound data provided by AHRI Standard 260, *Sound Rating of Ducted Air Moving and Conditioning Equipment*, is separated by sound path (discharge, inlet, and casing radiated).

Sound paths and recommendations for this configuration are:

1) Casing radiated. Sound radiates from the casing of the heat pump (and out through the return-air inlet, if un-ducted) into the closet. In an un-ducted return system, the sound then travels through the door and return-air grille into the occupied space.

For an application with a grille in the access door, locate the WSHP in a non-sound sensitive portion of the occupied space. Some attenuation may be provided by placing a "line of sight" barrier just inside the louver (Figure 82) and adding absorptive materials to the inside of the closet.

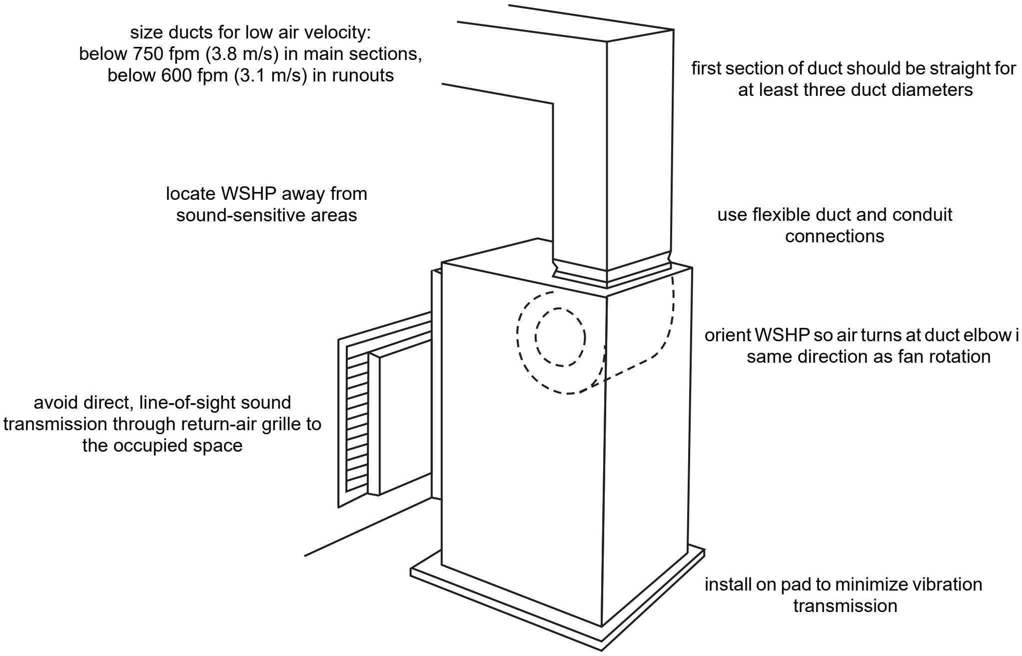
For an application where return air is ducted into the closet, either directly to the inlet or just into the closet space, install a door with sufficient transmission loss to reduce casing radiated sound entering the occupied space. It is also critical to use a gasketed, tight-fitting door to avoid the flanking paths associated with gaps under and around the door.

2) Supply airborne. Sound leaving the discharge of the WSHP travels down the supply ductwork, through the supply-air diffusers, and into the occupied space.

Supply airborne sound can be reduced by adding an acoustical liner to the supply ductwork. Acoustical analysis can be used to determine the required lining thickness and length of ductwork that needs to be lined. Another convenient way to add absorptive duct liner is to use lined flex duct as the final section that connects to the diffuser. Careful attention should be paid to the attachment of the flex to the diffuser; avoid sharp turns and pinched ducts at, and near, the diffuser.

Sizing ductwork for low velocity and low static pressure loss will reduce the sound produced by the fan. As a general rule, maintain the velocity in main duct sections below 750 fpm (3.8 m/s) and below 600 fpm (3.1 m/s) in runout duct sections.

Figure 82. Recommendations for vertical models



When an elbow is located near the discharge of the WSHP, orient the elbow so that the turn of the elbow is in the same direction as the rotation of the fan (Figure 82). Avoid close-coupled fittings in the ductwork, as these create high pressure loss and turbulence that generates sound. When possible, separate turns or fittings with straight duct sections that are at least four to five duct diameters in length.

Finally, remember that supply-air diffusers generate sound that will be added to the sound coming from the WSHP. Select diffusers for at least 10 NC points below the desired NC level for the space. Avoid turbulence at the diffuser by placing the balancing damper near the duct take-off rather than near the diffuser.

3) Supply breakout. Sound traveling down the ductwork can also travel though the duct walls into the ceiling plenum, and then through the ceiling into the occupied space. This is generally only a problem on the main supply duct near the WSHP.

When breakout sound is a problem, it can be reduced by routing the main duct over a non-sound sensitive area (a corridor, for example), splitting the main duct into multiple, smaller ducts that are routed in different directions, or switching to rigid metal round duct.

4) Return airborne. As described previously, the inlet of a vertical-style WSHP can be either ducted or un-ducted. For an un-ducted application, see the recommendations related to the casing radiated sound path.

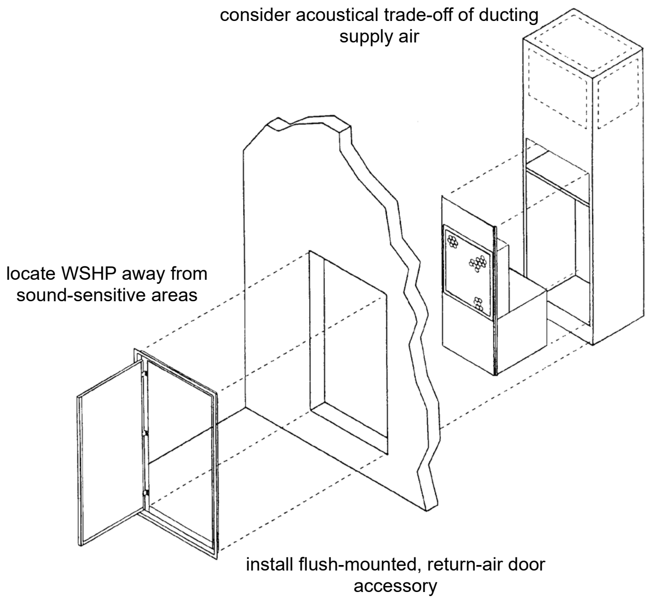
When return air is ducted to the WSHP inlet, sound will travel opposite the direction of airflow from the WSHP to the occupied space. A fully ducted return system uses ductwork to connect the return-air grille directly to the WSHP inlet. In a partially ducted system, a section of return ductwork is connected to the WSHP inlet, but it does not connect to the return-air grille.

components for low velocity. Partially ducted returns will benefit if the open end of the duct has no obstruction (such as a grille) and includes at least four equivalent duct diameters of straight duct prior to the WSHP inlet. A partially ducted return may benefit by connecting an acoustically lined elbow to the return-air grille with the opening facing away from the WSHP inlet.

WSHP: vertical-stack models

Vertical-stack water-source heat pumps are typically installed in the occupied space, so refer to the recommendations for console-style heat pumps (p. 128). Return airborne sound can be reduced by installing a flush-mounted return-air door (typically available as an optional accessory) that recesses into the wall (Figure 83). But, check the dimensions carefully—this door may be wider than the cabinet and require different clearances.

Figure 83. Recommendations for vertical-stack models (exploded view)



Ducting the supply air, rather than discharging it directly into the occupied space, provides an opportunity to provide some attenuation (refer to the discussion of the supply airborne sound path for vertical heat pumps, p. 131). The use of ductwork, however, requires the fan to operate at a higher speed to overcome the added pressure loss, and this results in more noise generated by the fan. To minimize sound, use ductwork for either all or none of the supply-air outlets.

Sound data is provided by either AHRI Standard 350, *Sound Rating of Non-Ducted Indoor Air-Conditioning Equipment*, or AHRI Standard 260, *Sound Rating of Ducted Air Moving and Conditioning Equipment*, depending on whether the vertical-stack heat pump discharges directly into the occupied space or if the supply air is ducted.

Cooling towers and other ancillary equipment

Sound is an important consideration when selecting and locating outdoor equipment, such as cooling towers or dedicated outdoor-air units. Some communities have enacted legislation that limits allowable sound levels for outdoor equipment. Even if legislation does not exist, people who live and work near a tower installation may object if the sound intrudes on their environment.

To assess the acceptability of cooling towers, or other ancillary equipment, follow the five-step acoustical analysis process outlined at the beginning of this section (see “Defining an acoustical model ,” p. 125). Using the acoustical modeling process to identify potential problem areas so they can be addressed prior to construction will save considerable time, cost, and the aggravation of addressing problems after the project is installed.

System Design Variations

This chapter explores several variations to the typical water-source heat pump system design, including ground-source heat pump systems, electrified WSHP systems, and several hybrid WSHP system configurations.

Ground-Source Heat Pump Systems

While much of the system is the same as in a conventional boiler/tower WSHP system, a ground-source heat pump (GSHP) system uses the relatively constant temperature of the earth for heat rejection and heat addition.

GSHP systems offer the potential for saving energy because they can reduce (or eliminate) the energy needed to operate a cooling tower and/or boiler. Eliminating the cooling tower and/or boiler also has architectural (no cooling tower outside) and maintenance advantages, and may free up floor space in the building. In addition, by removing the need for a boiler, a GSHP system reduces (or eliminates) on-site fossil fuel use, helping the building achieve decarbonization goals (see “Helps achieve decarbonization goals,” p. 10.).

This section discusses three types of ground-source systems, including:

- A **ground-coupled heat pump system**, which is a closed system that uses special, high-density polyethylene pipes that are buried in the ground as a heat source and heat sink.
- A **surface-water heat pump system**, which is similar to a ground-coupled heat pump system, except that the pipes are submerged in a pond or lake.
- A **ground-water heat pump system**, in which water is pumped from a well and then either returns to the source through a separate well, or is directed to a drain field or sewer system. Typically, an intermediate heat exchanger separates the well water from the water circulated throughout the building.

While this section focuses primarily on closed-loop, ground-coupled systems—because they are the most common—the other two types of systems certainly should be considered, when feasible.

Table 27 summarizes the advantages and drawbacks of these three types of GSHP systems.

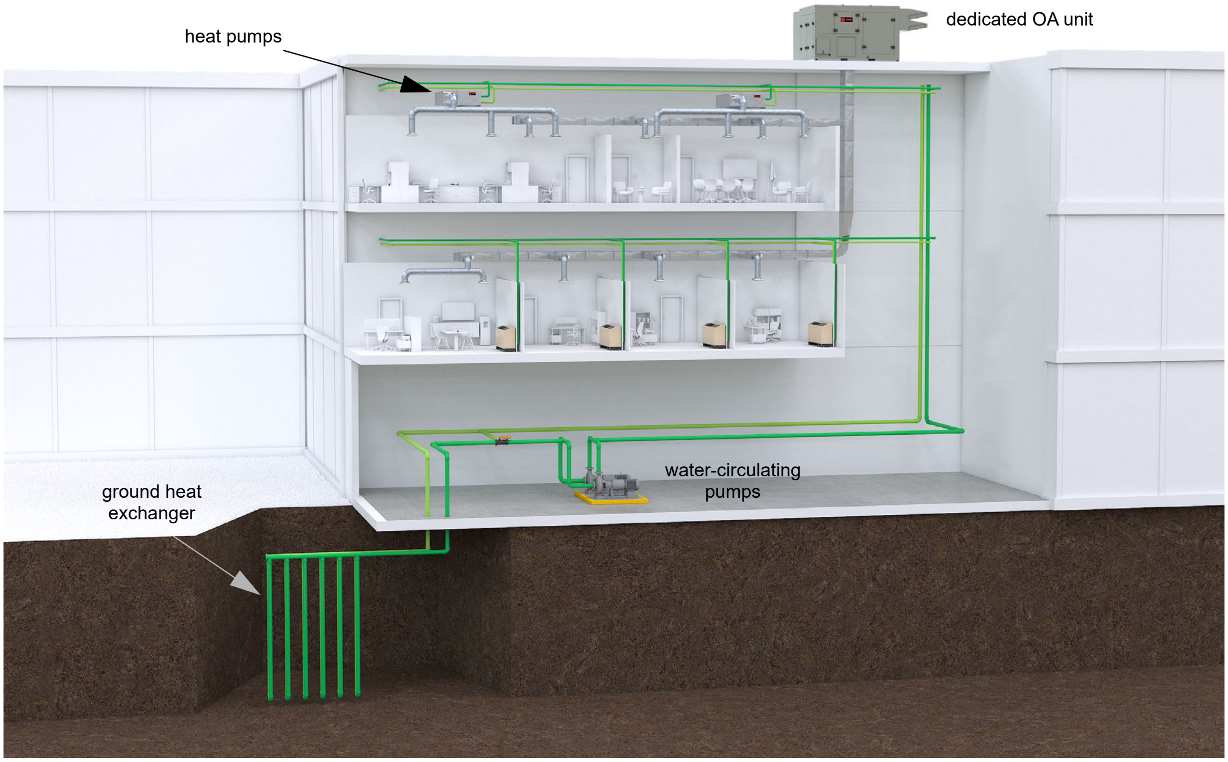
Table 27. Comparison of ground-source heat pump systems

Ground-coupled heat pump system	
Advantages:	**Disadvantages:**
• Simpler to design than the other types of ground-source systems • Closed system means that no water treatment is required and is not reliant on a constant supply of water from a well • Typically requires the least amount of pumping energy	• Typically has a higher installed cost than the other types of ground-source systems because of the amount of drilling or trenching required and the limited availability of certified loop contractors in some regions
Surface-water heat pump system	
Advantages:	**Disadvantages:**
• High thermal conductivity makes a body of water a good heat rejection/heat absorption medium • Closed system means that no water treatment is required and is not reliant on a constant supply of water from a well • Typically requires less total length of pipe than a closed-loop, ground-coupled heat exchanger • Less expensive because no drilling and less trenching is required	• Requires a large body of water for submerging the heat exchanger • Typically experiences lower heat pump efficiencies due to wider temperature variation than a ground heat exchanger or ground water • Risk of damage to submerged heat exchanger
Ground-water heat pump system	
Advantages:	**Disadvantages:**
• Higher heat pump efficiencies due to better thermodynamic performance than closed-loop systems • Typically provides highest return on investment because cost to drill wells does not rise linearly with capacity • Land required for drilling wells is much smaller than required for ground-coupled systems • Well-drilling contractors are widely available	• Sufficient quantity of water is needed • Requires high quality water to minimize fouling or scaling of heat pump heat exchangers (adding an intermediate, plate-and-frame heat exchanger offsets some or all of the performance advantage over a closed-loop system) • Requires a method to re-inject water into the ground or dispose of in a river or sewer system • Typically subject to various local, state, and federal clean water and surface water codes and regulations

Ground-coupled heat pump systems

A ground-coupled heat pump (GCHP) system uses a closed system of special, high-density polyethylene pipes that are buried in the ground at a depth that takes advantage of the earth's relatively constant temperature, using the ground as the heat rejecter and heat adder (Figure 84).

Figure 84. Ground-coupled heat pump system



Most GCHP systems do not actually get rid of heat, they store it in the ground for use at a different time. During the cooling season, heat rejected by the heat pumps causes the loop temperature to increase. As the fluid flows through the buried pipes, heat is transferred from the warm fluid to the cooler ground. In a sense, the heat is stored in the earth for use at a later time. Conversely, during the heating season, heat extracted by the heat pumps causes the loop temperature to decrease. The cool fluid flowing through the buried pipes extracts the stored heat from warmer ground.

GCHP systems offer the potential for reduced energy use when compared to a traditional boiler/tower WSHP system because they can reduce (or eliminate) the energy needed to operate a cooling tower and/or boiler. Eliminating the cooling tower and boiler also has architectural and maintenance advantages, and may free up floor space in the building. In addition, by removing the need for a boiler, a GSHP system reduces (or eliminates) on-site fossil fuel use, helping the building achieve decarbonization goals (see "Helps achieve decarbonization goals," p. 10.). Finally, the loop may operate at cooler temperatures during the cooling season than in a conventional boiler/tower system. This results in the heat pump compressors operating more efficiently.

The installation costs associated with this system, however, must be considered to determine the economic viability. In general, the largest portion of the installation cost is due to the ground heat exchanger. Installation requires excavation, trenching, or boring, and in some locales there may be few qualified contractors for installing the ground heat exchanger.

In a perfectly balanced system, the amount of heat rejected to the ground over the year would equal the amount of heat extracted, eliminating the need for a cooling tower and boiler. In most applications, however, there is an imbalance between heat rejected to the ground and heat extracted. This imbalance requires the ground heat exchanger to be larger to prevent the ground temperature from changing over time.

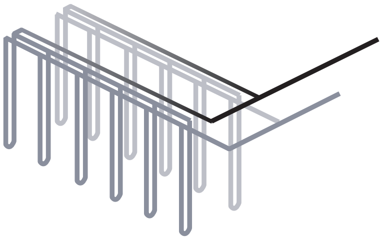
While eliminating both the cooling tower and boiler likely results in the greatest overall energy savings, for many applications it requires a larger (and more expensive) ground heat exchanger. Adding a small cooling tower to the loop for a cooling-dominated application, or adding a small boiler for a heating-dominated application, can reduce the size of the ground heat exchanger, making a GCHP system more economically feasible (see "Hybrid ground-coupled heat pump systems," p. 149).

Ground heat exchanger configurations

The pipes that make up the ground heat exchanger are typically oriented in either a vertical, horizontal, or spiral pattern. Any of these patterns can be designed to provide the same fluid temperatures under a given set of conditions. The choice depends on available land, soil conditions, and excavation costs.

Vertical loops

Figure 85. Vertical heat exchanger



Vertical loops are the most common in commercial applications due to the limited land that is generally available to bury the heat exchanger (Figure 85). Vertical boreholes—with a diameter of 4 to 6 in. (10 to 15 cm) each—are drilled to depths of 200 to 500 ft (60 to 150 m), typically about 10 to 20 ft (3 to 6 m) apart. A closed piping loop (two pipes with a U-bend at the bottom) is inserted into each borehole, after which the hole is filled with grout and backfilled .The grout seals the bore to prevent surface contaminations from getting into the ground or aquifer, and transfers heat from loops to the ground. High-performance grouts are available with additives to improve thermal conductivity.

The HVAC design engineer should be familiar with federal, state, and local codes for drilling of water wells or boreholes for ground-coupled systems, since there can be differences. Some contractors have **low-profile drilling rigs** that can be used in a basement. Or **directional drilling** equipment can allow a large number of bores to be drilled from within a smaller surface footprint, from which they then fan out underground.

Advantages of vertical loops include:

- They typically require the least amount of land of the three configurations. Vertical loops typically require anywhere from 60 to 275 ft^2 of ground surface per "block" cooling ton (1.6 to 7.3 m^2/kW).
- They typically require less total piping than the other two configurations because the ground temperature is more constant at greater depths.

- When piped in a parallel reverse-return configuration, this pattern typically requires the least amount of pumping energy of the three configurations.

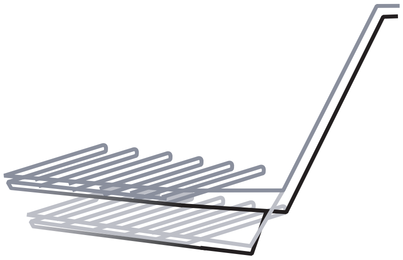
Drawbacks include:

- Drilling costs are frequently higher than the trenching costs associated with horizontal or spiral loops.
- Grouting and backfilling of the boreholes require special attention to fill material and to ensuring that the pipes and surrounding earth remain in contact.
- If the boreholes are spaced too close together, there is a potential for long-term heat build-up in the ground that may be undesirable for a cooling-dominated application.
- Installation requires the knowledge and availability of a certified loop contractor with proper drilling equipment.

Horizontal loops

Horizontal loops are often considered when adequate land is available (Figure 86). Historically, horizontal loops often consisted of a single layer of pipe buried in the ground using a trenching machine. However, land requirements have been reduced with the advent of multiple-layer horizontal loops. While less land and trenching is required, a multiple-layer loop requires more total length of piping than a single-layer loop.

Figure 86. Horizontal heat exchanger



Each closed-loop pipe is placed in a trench, which is typically 6 ft (1.8 m) deep and spaced 6 to 15 ft (1.8 to 4.6 m) apart. Trench length can range from 100 to 400 ft per “block” cooling ton (8.7 to 34.7 m/kW) for a single-layer loop.

One option to reduce excavation needed is to use **directional drilling** equipment, in which the drilling head can be steered up and down and side to side, achieving precision placement. This may allow the horizontal loops to be installed under a parking lot or an athletic field while that space is still in use. Loops can layered at different depths and routed to a precise location, such as an equipment room wall or underground access vault.

Advantages of horizontal loops include:

- Trenching costs are typically lower than the drilling costs associated with vertical loop installation.
- In cooler climates, horizontal loops may not build up as much heat over time as vertical loops, because the pipes are closer to the surface.

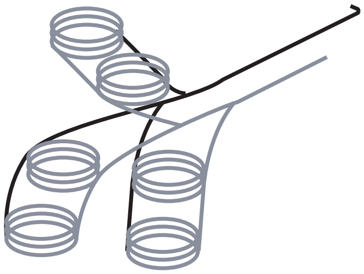
Drawbacks include:

- Horizontal loops require a larger area of land than vertical loops., and the excavation process can be disruptive in that the land cannot be used for other purposes at the time.
- At this shallower depth, ground temperatures are subject to seasonal temperature variations, rainfall, and snow melting. Obtaining the same loop temperatures as a vertical loop requires a more complicated design with longer pipe lengths.

- The longer pipe lengths also require more antifreeze solution (when necessary) and more pumping energy than vertical loops.
- The pipe is at greater risk of damage during backfilling of the trenches.

Spiral loops

Figure 87. Spiral heat exchanger



A variation of the multiple-layer, horizontal loop is the spiral loop (Figure 87). The spiral loop includes a roll of pipe that is unraveled into circular loops, tied together, and then placed either vertically in a trench or horizontally in an open pit.

The spiral loop generally requires more total piping—typically between 500 and 1000 ft per “block” cooling ton (43 to 86 m/kW)—but less trenching than multiple-layer, horizontal loops. Both horizontal and spiral loop systems are generally associated with small commercial or residential buildings where land requirements are less of a factor.

Advantages of spiral loops include:

- Less land area, and less trenching, is typically required for spiral loops than for traditional horizontal loops.
- Installation costs are typically lower than for traditional horizontal loops, because less trenching is required.
- Trenching costs are typically lower than the drilling costs associated with vertical loop installation.
- Spiral loops may not build up as much heat over time as vertical loops, because the pipes are closer to the surface, where heat can be dissipated to the atmosphere.

Drawbacks include:

- Spiral loops require a larger area of land than vertical loops.
- Spiral loops require more total length of piping than either vertical or horizontal loops, which increases pump energy use.
- At the shallower depth, ground temperatures are subject to seasonal temperature variations, rainfall, and snow melting. Obtaining the same loop temperatures as a vertical loop requires a more complicated design with longer pipe lengths.
- The longer pipe lengths also require more antifreeze solution (when necessary) than vertical loops.
- The pipe is at greater risk of damage during backfilling of the trenches.

Design of the ground heat exchanger

Ground heat exchangers should be designed by a certified professional:

Certified Geo-Exchange Designer (CGD)

GSHP Commercial System Designer (GCSD)

A directory of certified professionals is available on the International Ground Source Heat Pump Association (IGSHPA) web site:

www.igshpa.org/business-directory

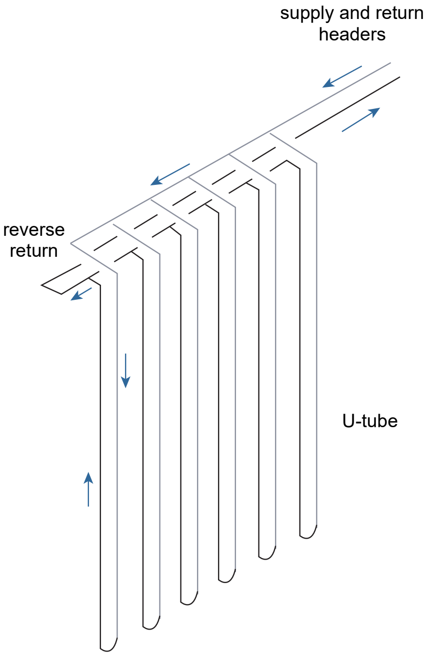
The ground heat exchanger must be sized to maintain the loop temperature within the minimum and maximum limits for which the heat pumps have been selected. And it must be sized to maintain those temperatures over the expected life of the system. If the heat exchanger is too small, the ground temperature may increase over time, degrading the performance of the system.

This section focuses primarily on vertical, closed-loop ground heat exchangers. Due to limited land availability, this is the most common type of ground heat exchanger used in commercial or institutional buildings.

The primary components of the vertical ground heat exchanger include (Figure 88):

- *Supply and return headers.* Pipes used to convey the total system flow from the loop inside the building to the individual, parallel U-tubes. Headers are typically constructed of larger-diameter pipe to minimize pressure drop, and are typically installed in a reverse-return configuration to better equalize pressure drops and balance flows through the individual U-tubes.
- *U-tubes.* Pipes that convey fluid from the supply header, down into a borehole (or trench), and then returned back up the same borehole (or trench) to the return header. The pipe includes a 180-degree fitting, or U-bend, at the bottom of a borehole (or at the end of a trench). The heat exchanger typically consists of multiple U-tubes connected to the supply and return headers. The U-tubes are typically installed in a parallel configuration so that only a portion of the total system flow rate travels through a single U-tube, minimizing overall pressure drop.
- *Grout.* After the U-tube is inserted, the hole is filled with grout. This seals the bore to prevent surface contaminations from getting into the ground or aquifer. High-performance grouts are available with additives to improve thermal conductivity and increase heat transfer between the U-tube and the ground.

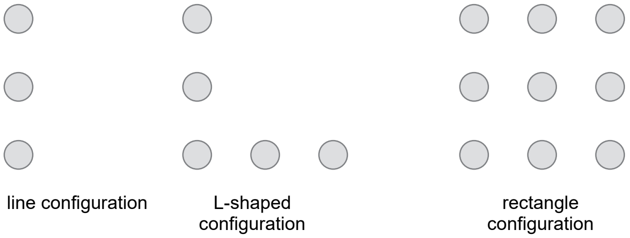
Figure 88. Components of a vertical ground heat exchanger



The piping used for the ground heat exchanger is typically high-density polyethylene (HDPE) with thermally fused joints. Pipe diameter for the U-tubes ranges from 0.75 in. (20 mm) to 1.25 in. (60 mm), depending on the diameter of the borehole.

A borefield is typically laid out to ensure proper separation of the individual U-tubes. Common configurations include arranging the boreholes in a straight line, L-shaped pattern, or rectangle (Figure 89), but other configurations are possible.

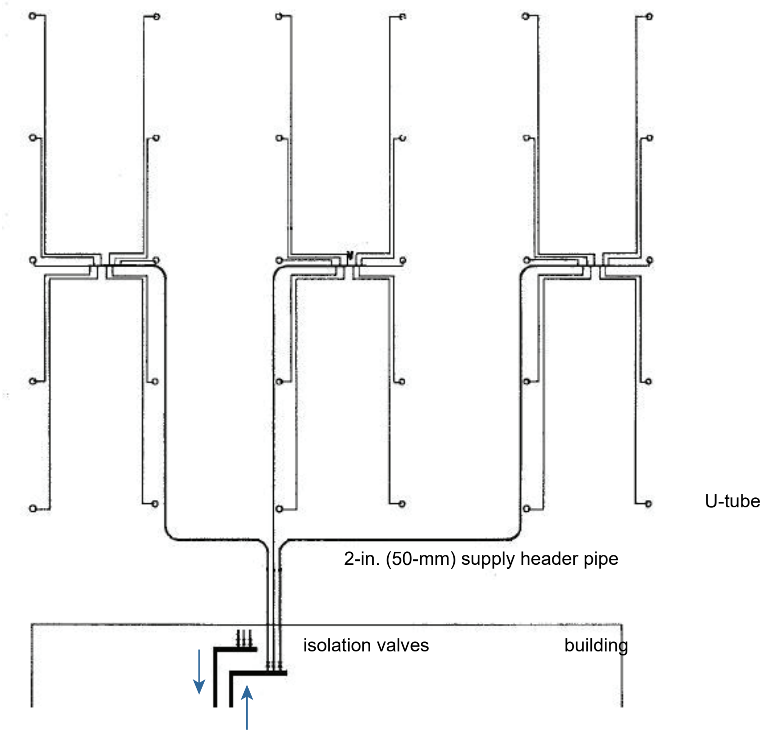
Figure 89. Examples of vertical heat exchanger configurations



It is typically recommended to group 10 to 12 U-tubes on a single header, and isolate each group with valves (Figure 90). This allows for easier flushing of the U-tubes to purge air and debris after installation. And if a leak occurs, the affected section of the borefield can be shut off to fix the leak, while the remaining sections of the field remain in operation.

The final benefit of this approach is simpler installation. Smaller header pipes can often be purchased in a roll, which simplifies installation by greatly reducing the number of field-fabricated, thermally-fused joints. In contrast, larger pipe sizes usually need to be purchased in straight sections, increasing the number of joints fabricated in the field.

Figure 90. Example layout of well field (only supply-side piping shown)



Source: *Ground-Source Heat Pumps: Design of Geothermal Systems for Commercial and Institutional Buildings*, Figure 5.6 © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org.

As an example, a 2-in. (50-mm) HDPE (SDR 11) header pipe will convey 30 gpm (1.9 L/s) at an acceptable pressure drop—approximately 2.3 ft of H_2O per 100 ft of piping (0.23 kPa/m). For a 30-ton (106-kW) system, with a total system flow rate of 90 gpm (5.7 L/s), dividing the U-bends into three separate groups allows each group to be served by a 2-in. (50-mm) header pipe (Figure 90).

Some design engineers or contractors prefer to locate the isolation valves inside the building, routing the separate header pipes from that location to each group of U-tubes in the borefield (as shown in Figure 90). Others prefer to locate the isolation valves in a vault (or pit) that is located near the borefield, and then install a single, larger set of pipes from this vault to connect to the building loop.

Centralized versus dedicated ground heat exchangers

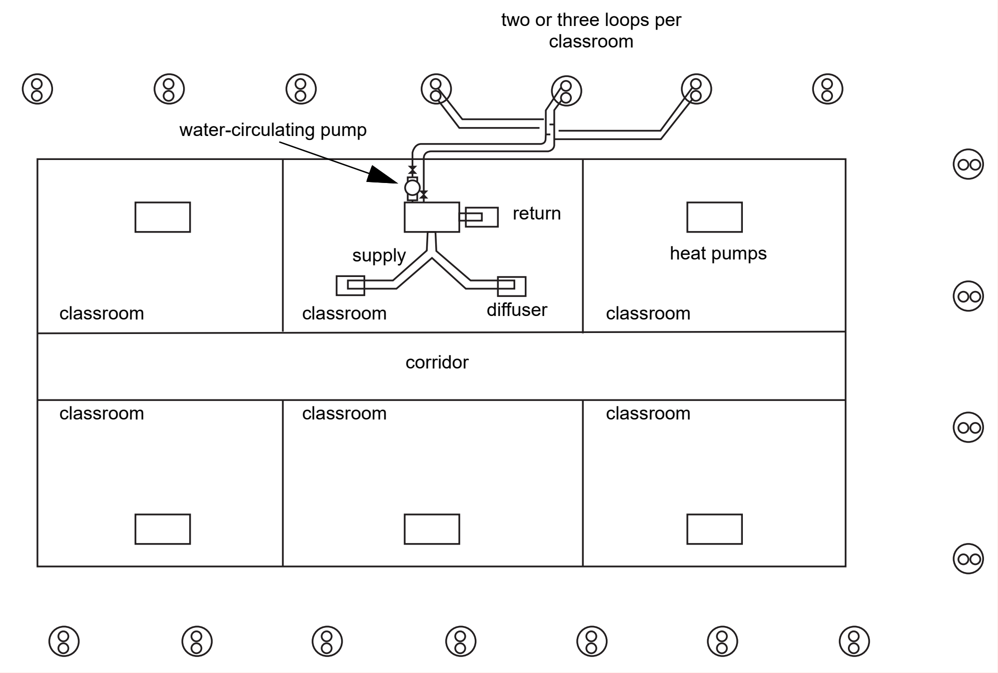
Many ground-coupled heat pump systems are designed to connect all the heat pumps to a common water distribution loop, which is then connected to a centralized (or shared) ground heat exchanger (Figure 84, p. 137 and Figure 90, p. 142).

However, in one- or two-story buildings with large footprints, an alternate approach could be to connect each heat pump to a dedicated ground heat exchanger (Figure 91). In this case, a small water-circulating pump serves each heat pump and heat exchanger, and turns on and off along with the compressor.

Using dedicated heat exchangers typically requires more overall length of pipe for the ground heat exchanger, because this approach is not able to benefit from load diversity. And it is more difficult to add supplemental heat rejection (or heat addition) equipment for a "hybrid" approach (see "Hybrid ground-coupled heat pump systems," p. 149).

However, using dedicated ground heat exchangers requires less piping inside the building and reduces the need for headers, isolation valves, and valve vaults.

Figure 91. Example of a dedicated ground heat exchanger for each pump



Source: 2023 *ASHRAE Handbook—HVAC Applications*, Chapter 35, Figure 19. © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org

Avoid using rules-of-thumb

For a residential building, it is common to size of the ground heat exchanger based on a "rule-of-thumb," typically in terms of feet (m) of heat exchanger per installed ton (kW) of capacity. These estimates are typically based on the rated, or nominal, capacity of the heat pump—3 tons (10.6 kW), for example. (Sometimes the term "connected load" is used, which has the same meaning in this context.)

But in a residential application, only one heat pump is typically connected to the loop. And since the building cooling and heating loads are primarily dictated by ambient weather conditions—meaning that heat gain or loss through the building envelope is typically the largest component of the load—the unit is operating in either heating or cooling mode for a season, rather than cycling between heating and cooling modes often. Because of these characteristics, and the fact that residences typically have similar occupancy patterns, the use of rules-of-thumb for sizing the ground heat exchanger has been somewhat successful for residential applications in similar geographical areas (with similar weather conditions and soil types).

Complicating the matter further, in a commercial or institutional application, "per ton" (per kW) could refer to peak cooling load, block cooling load, connected load (or installed capacity), or some type of "net" operational cooling load. Depending on which load is being used as the basis for the rule-of-thumb, the result may lead to significant oversizing or undersizing of the ground heat exchanger.

In a commercial or institutional application, however, multiple heat pumps are typically connected to the same loop. And since the building cooling and heating loads are heavily influenced by internal heat gains (people, lights, heat-generating equipment)—and less influenced by heat gain or loss through the building envelope—it is common for some units to operate in the cooling mode at the same time other units operate in the heating mode. Because of these characteristics, and the fact that these types of buildings can have drastically different occupancy patterns, the **use of rules-of-thumb should be avoided** for commercial or institutional buildings, as they can lead to severe oversizing or undersizing of the ground heat exchanger.

While some HVAC design engineers have developed their own "rules-of-thumb" for sizing the ground heat exchanger, they are typically only applicable to a limited geographical area (with similar weather conditions and soil types) and to a specific type of building (with similar occupancy patterns and distribution of loads).

Test project-specific soil conditions

Prior to finalizing the system design, a test bore should be drilled and conductivity test completed to determine the actual soil thermal properties and drilling conditions at the project site. This is often the greatest cost-saving task that can be performed.

A soil thermal conductivity test involves the installation of one complete bore at the project site, using the expected materials and construction practices. Water is then pumped through the bore for several days, while a heat load bank adds a constant amount of heat to the bore. Temperature sensors record the entering and leaving fluid temperatures, and the resulting temperature profile is used to calculate the ground's thermal conductivity.

First, the actual drilling or trenching conditions can be described in the bid documents, aiding the contractor in developing the most accurate bid. Unknown soil conditions typically result in the contractor increasing the bid price as a safety margin. Visiting the project site to drill a test bore also helps to determine how difficult it will be for heavy excavation or drilling equipment to access the site.

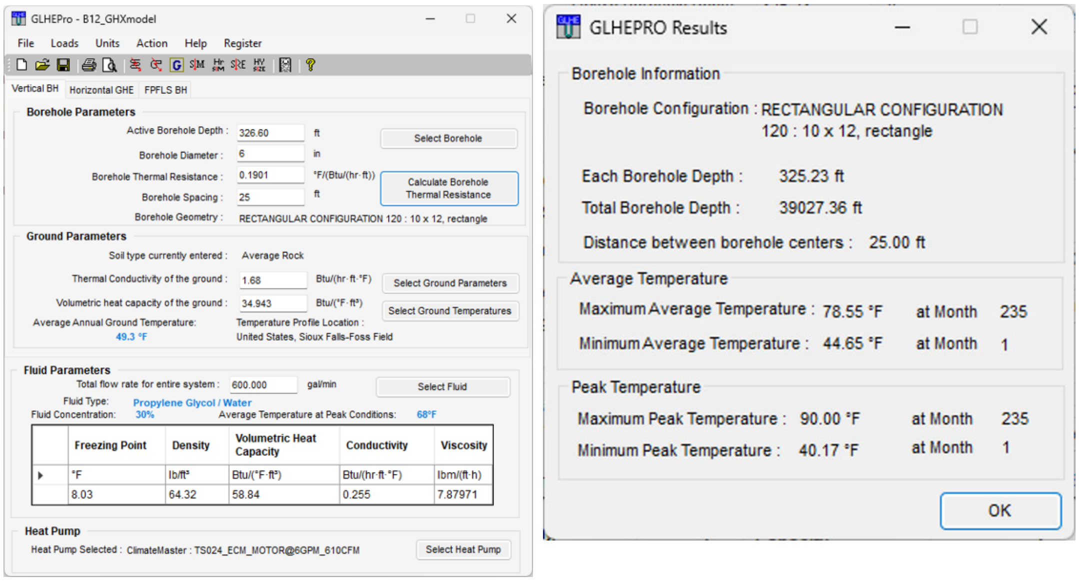
Second, the thermal properties of the soil influence the necessary length of the ground heat exchanger. Using actual soil properties—specifically, the thermal conductivity, thermal diffusivity, and undisturbed ground temperature—helps avoid oversizing or undersizing the ground heat exchanger. The test report should indicate the type of soil or rock found at different depths.

Finally, the site evaluation can also reveal potential problems that could occur during installation, where coordination between trades is critical. On smaller construction sites, waiting for building materials to be moved so that the drilling or trenching area can be accessed may result in project delays or cost overruns.

Sizing the ground heat exchanger

Typically, the ground heat exchanger is designed using computer software that was developed specifically for this purpose (Figure 92). These software programs can typically model various borefield configurations and account for soil properties, grout conductivity, fluid properties (if not pure water), borehole diameter and spacing, and size and type of piping used.

Figure 92. Example of ground heat exchanger design software



GLHEPro - B12_GHXmodel

Borehole Parameters

Active Borehole Depth : 326.60 ft

Borehole Diameter : 6 in

Borehole Thermal Resistance : 0.1901 °F/(Btu/(hr·ft))

Borehole Spacing : 25 ft

Borehole Geometry : RECTANGULAR CONFIGURATION 120 : 10 x 12, rectangle

Ground Parameters

Soil type currently entered : Average Rock

Thermal Conductivity of the ground : 1.68 Btu/(hr·ft·°F)

Volumetric heat capacity of the ground : 34.943 Btu/(°F·ft³)

Average Annual Ground Temperature: 49.3 °F

Temperature Profile Location : United States, Sioux Falls-Foss Field

Fluid Parameters

Total flow rate for entire system : 600.000 gal/min

Fluid Type: Propylene Glycol / Water

Fluid Concentration: 30%

Average Temperature at Peak Conditions: 68°F

Freezing Point	Density	Volumetric Heat Capacity	Conductivity	Viscosity
°F	lb/ft³	Btu/(°F·ft³)	Btu/(hr·ft·°F)	lbm/(ft·h)
8.03	64.32	58.84	0.255	7.87971

Heat Pump

Heat Pump Selected : ClimateMaster : TS024_ECM_MOTOR@6GPM_610CFM

GLHEPRO Results

Borehole Information

Borehole Configuration : RECTANGULAR CONFIGURATION
120 : 10 x 12, rectangle

Each Borehole Depth : 325.23 ft

Total Borehole Depth : 39027.36 ft

Distance between borehole centers : 25.00 ft

Average Temperature

Maximum Average Temperature : 78.55 °F at Month 235

Minimum Average Temperature : 44.65 °F at Month 1

Peak Temperature

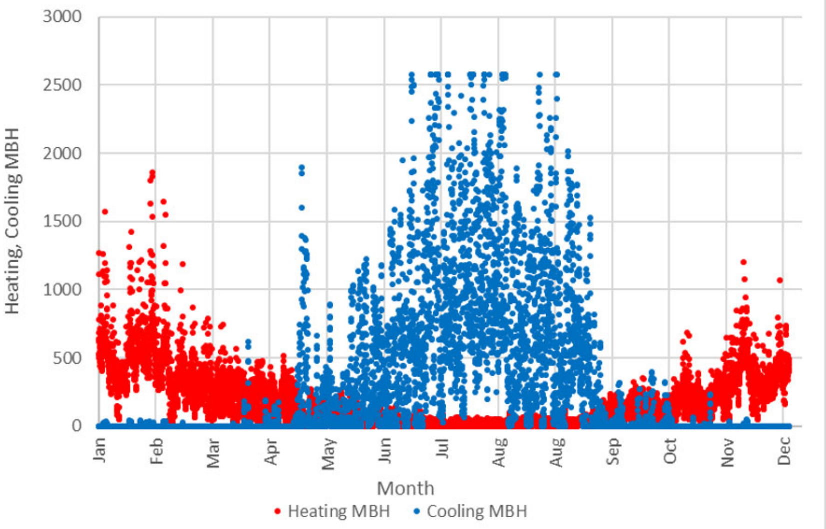
Maximum Peak Temperature : 90.00 °F at Month 235

Minimum Peak Temperature : 40.17 °F at Month 1

The Trane TRACE® 3D Plus software has the capability to export both peak and monthly loads, which can then be imported into GLHEPRO, a ground heat exchanger design software program. After running GLHEPRO, specific characteristics about the ground heat exchanger design can be imported back into TRACE for a more accurate simulation of building energy use.

In addition, the software program requires the peak heating and cooling load —in Btu/hr (kW), for example—for each month of the year, as well as the total quantity of heat—in Btu (kWh), for example—rejected to, and extracted from, the ground during each month of the year (Figure 93). Some software programs have the capability to import this data from HVAC load calculation software.

Figure 93. Monthly heating and cooling loads



Typically, the design engineer uses this type of software to design the ground heat exchanger so that it will maintain the loop temperature (entering the building) between the minimum and maximum temperatures for which the heat pump can operate.

To briefly demonstrate the utility of such design software, consider a small office building located in St. Louis, Missouri. The building is operated for ten hours a day, five days a week. The design block cooling load is 66 tons (230 kW).

For this example, the borefield is arranged as an 8-by-8 rectangle of U-tubes (64 boreholes in total) that are constructed of 0.75-in. (19-mm) HDPE pipe (SDR 11) and spaced 15 ft (4.6 m) apart. The design software determines the required depth of each borehole to maintain the loop temperature (entering the building) between the user-specified minimum and maximum temperatures. For this example, to maintain loop temperatures between 20°F (-7°C) and 90°F (32°C), each borehole must be 240 ft (73 m) deep. The software also simulates the operation of the ground heat exchanger to estimate the average loop temperature for each month.

To demonstrate the impact of the building operation on the size of the ground heat exchanger, consider if this same building is operated for 24 hours a day, seven days a week. The design block cooling load remains unchanged—since it occurs during the daytime—but the total monthly heating and cooling loads will be significantly higher. With this extended operation, there are more months where both cooling and heating are needed, and there is a much larger demand for cooling in each month.

Using the same 8-by-8 rectangular borefield, each borehole must now be 600 ft (180 m) deep. The primary cause of this drastic increase in length is the increased heat rejected to the ground due to cooling operation.

To reduce the required depth, a larger borefield with more boreholes and/or greater spacing between boreholes could be used. Or a small fluid cooler could be added to the loop to help reject the excess heat (see “Hybrid ground-coupled heat pump systems,” p. 149).

This example also demonstrates the risk associated with using a rule-of-thumb to determine borehole depth. The same building, located on the same plot of land, with the same block cooling load required drastically different borehole depths depending on how that building is operated or used. For a typical office building schedule of 10 hours/day and 5 days/week, the required borehole depth is 233 ft per ton of block cooling load (20.2 m/kW). But if operating for 24 hours/day and 7 days/week, the required borehole depth increased to 582 ft/ton (50.4 m/kW).

If the borefield was designed using a rule-of-thumb like 300 ft per ton of block cooling load (26 m/kW), the ground heat exchanger would be oversized by about 25 percent for the typical operating schedule or undersized by almost 50 percent for the 24/7 operating schedule.

General recommendations for designing ground heat exchangers

For more information on the design and layout of ground heat exchangers, refer to Chapter 35, "Ground-Source Heat Pumps and Geothermal Energy," in the 2023 *ASHRAE Handbook—HVAC Applications*; the ASHRAE manual, *Geothermal Heating and Cooling: Design of Ground-Source Heat Pump Systems;* and the International Ground Source Heat Pump Association (IGSHPA) manual, *Closed-Loop, Ground-Source Heat Pump Systems: Installation Guide*.

Other publications contain more complete details related to designing the ground heat exchanger, but following are some general recommendations:

- *Drill a test borehole and perform a thermal conductivity test prior to finalizing system design.*

  As mentioned previously, this helps determine the actual soil thermal properties, as well as drilling (or trenching) conditions, at the site.

- *Size the ground heat exchanger based on the "block" cooling and heating loads, rather than summing installed heat pump capacities.*

  In a system with multiple heat pumps, zone-by-zone load variation throughout the day (called "system load diversity," see sidebar on p. 45) results in an instantaneous load that is less than summing the peak loads of all zones. Sizing the ground heat exchanger based on the "block" load typically results in a smaller heat exchanger.

- *Separate boreholes by at least 20 ft (6 m) to help avoid long-term changes in ground temperature.*

  Spacing the boreholes further apart also allows for a reduced overall length of the heat exchanger, which can result in shallower, or fewer, boreholes.

- *Size boreholes with as small a diameter as possible, and use a grout with high thermal conductivity.*

  This minimizes the use of grout, which often has poorer thermal conductivity than the surrounding ground. When possible, use thermally-enhanced grouts with higher thermal conductivity.

- *The ASHRAE* Design of Ground-Source Heat Pump Systems *manual recommends a total system flow rate of 2.5 to 3.0 gpm per ton of "block" cooling load (0.045 to 0.054 L/s per kW). For systems with higher-efficiency heat pumps and/or higher pump head, an even lower flow rate may be optimal.*

  Due to the pressure drop through the ground heat exchanger, excessive pump energy use can drastically reduce the energy-savings benefit of a ground-coupled system. To avoid over-pumping, the overall system flow rate should account for system load diversity and be based on the "block" load, rather than by summing installed heat pump capacities.

- *Minimize the use of antifreeze.*

  Because the header piping and U-tubes are installed well below grade, the fluid inside the heat exchanger is not exposed to ambient temperatures. The ground heat exchanger design software will typically estimate the minimum expected loop temperature, which would then indicate whether or not antifreeze should be added to the water inside the loop.

  In many climates, ground heat exchangers for commercial or institutional buildings likely require little or no antifreeze because annual cooling loads are greater than annual heating loads, and because the quantity of heat rejected to the loop during the cooling mode is typically higher than the quantity of heat extracted from the loop during the heating mode.

- *Lay out piping and headers to simplify field fabrication and flushing.*

  Beyond drilling boreholes or digging trenches, fabricating the heat exchanger at the project site involves fusing each U-tube to the header and connecting the header pipes to the building loop. This part of the project is labor-intensive and typically occurs in a deep trench. The use of pre-assembled headers and simple layouts can reduce installation cost and minimize risk. It is beneficial for the design engineer to consult with a drilling contractor to find ways to minimize labor cost and time.

- *Hire contractors that are experienced with the installation of ground heat exchangers.*

  Experienced contractors have developed their own proven methods of installing U-tubes and headers. In addition, they may have drilled or trenched in soil conditions similar to those of the project site.

System design variables can be changed to reduce the installed cost of a ground-coupled heat pump system (Table 28). However, these changes are not without side effects. Changing a system design variable often impacts the energy use or some other aspect of system operation.

Table 28. Impact of various ground heat exchanger design decisions

System design variable	Impact on installed cost	Other impacts
Increase the upper temperature limit for water entering the GSHP during cooling mode	Reduces the required length of the ground heat exchanger	GSHP will be less efficient during cooling mode
Decrease diameter of U-tube piping	Reduces the cost of the ground heat exchanger and simplifies installation	Increases the required length of the ground heat exchanger
Increase the separation distance between boreholes	Reduces the required length of the ground heat exchanger	Increases the amount of land required and increases length of header piping
Install a cooling tower to supplement heat rejection ("hybrid" ground-coupled system)	Reduces the required length of the ground heat exchanger	Requires seasonal maintenance and operation of the cooling tower uses energy
Assume movement of groundwater in design of ground heat exchanger	Reduces the required length of the ground heat exchanger	Requires hydrological survey to confirm or risks under-performance of the heat exchanger

Source: *Ground-Source Heat Pumps: Design of Geothermal Systems for Commercial and Institutional Buildings*, Table 4.12 © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org.

Hybrid ground-coupled heat pump systems

For more information on the design and control of hybrid ground-coupled heat pump systems, refer to the Energy Center of Wisconsin publication, *Hybrid Ground-Source Heat Pump Installations: Experiences, Improvements and Tools*.

While eliminating both the cooling tower and boiler likely results in the greatest overall energy savings, for many applications it requires a larger (and more expensive) ground heat exchanger to account for the imbalance between heat rejected to the ground and heat extracted from the ground.

For example, in a cooling-dominated application, a large amount of heat must be rejected to the ground during the cooling season, but a much smaller amount of heat is extracted from the ground during the heating season. This imbalance can cause the temperature of the ground surrounding the heat exchanger to increase over time.

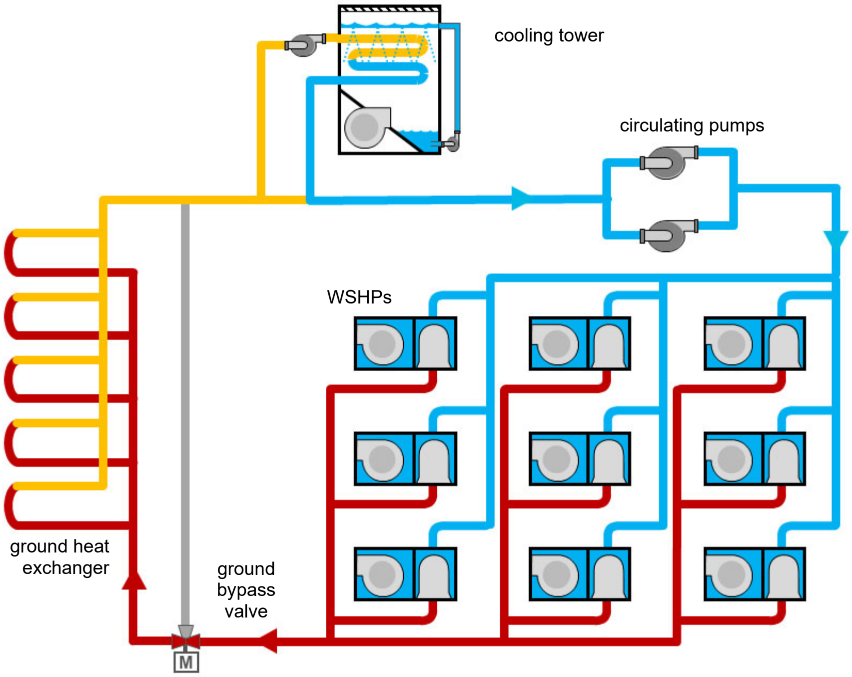
Conversely, in a heating-dominated application, a smaller amount of heat is rejected to the ground during the cooling season, but a larger amount of heat must be extracted from the ground during the heating season. In this case, the temperature of the ground can decrease over time.

In either case, future operation and efficiency of the heat pumps is compromised by this change in ground temperature. This imbalance often requires the ground heat exchanger to be larger to prevent the ground temperature from changing over time. The cost to install such a large heat exchanger often dissuades people from considering a GCHP system.

In a “hybrid” GCHP system, however, either a small cooling tower (for a cooling-dominated application) or a small boiler (for a heating-dominated application) is installed on the loop to supplement the heat rejection or heat addition capacity of the ground heat exchanger. This approach reduces the required size and cost of the heat exchanger by avoiding the imbalance described previously. While the overall energy use may not be as low as in a system with a larger heat exchanger, this approach often results in a more acceptable return on investment. In a hybrid GCHP system, neither of the separate pieces needs to be sized for design capacity, since they complement each another.

In a cooling-dominated application, a small cooling tower—or possibly even a dry cooler—is connected to the loop (Figure 94). In this case, the ground heat exchanger is sized based on the total heat to be extracted from the ground during the heating season. Then the cooling tower is sized to reject the excess heat during the cooling season. If the fluid temperature returning from the ground heat exchanger rises above a preset upper limit—90°F (32°C), for example—the cooling tower is activated to reject the excess heat.

Figure 94. Hybrid GCHP system with supplemental heat rejection for a cooling-dominated application

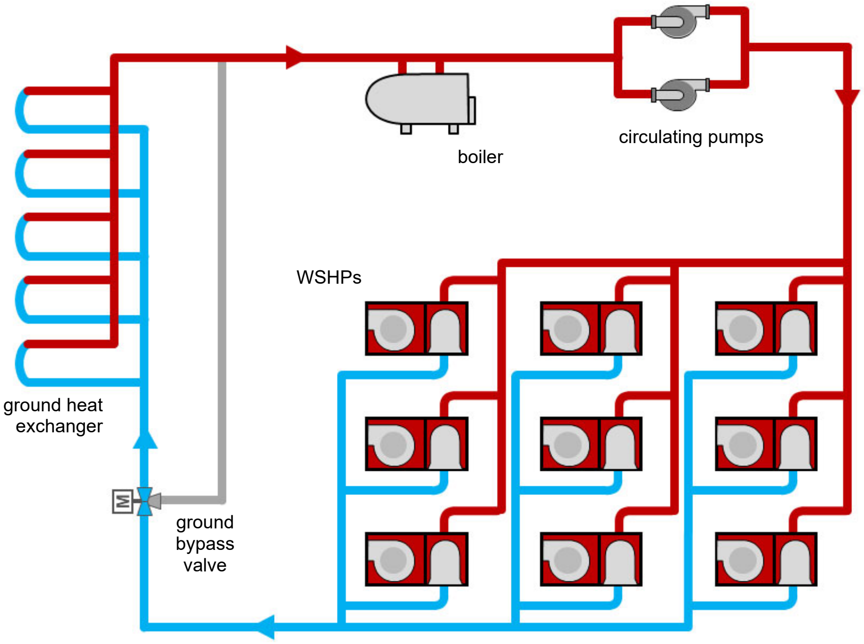


This hybrid concept can also be used to correct an imbalance in an **existing system**. If the building cooling loads have increased over time, it may cause an imbalance between heat rejected to the ground and heat extracted from it, even though the system may have been balanced when originally designed. After years of operating with such an imbalance, the result of the increased cooling loads will likely be a warmer fluid temperature returning from the ground heat exchanger during the cooling season. By installing a small heat rejection device, and operating it at night or in the spring or fall, the excess cooling load can be rejected, restoring the balance between heat rejected to the ground and heat extracted from it.

In a heating-dominated application, a small boiler could be connected to the loop (Figure 95). In this case, the ground heat exchanger is sized based on the total heat to be rejected to the ground during the cooling season. Then the boiler is sized to provide the supplemental heat during the heating season. If the fluid temperature returning from the ground heat exchanger drops below a preset lower limit [25°F (-4°C), for example, the boiler is activated to provide supplemental heat. Use caution to avoid sending too hot of fluid to the ground heat exchanger, which could cause damage to the HDPE piping.

Rather than installing a boiler, it may be more economical to add electric resistance heaters to some of the heat pumps (see “Electric resistance heat for a “boiler-less” system,” p. 56).

Figure 95. Hybrid GCHP system with supplemental heat addition for a heating-dominated application

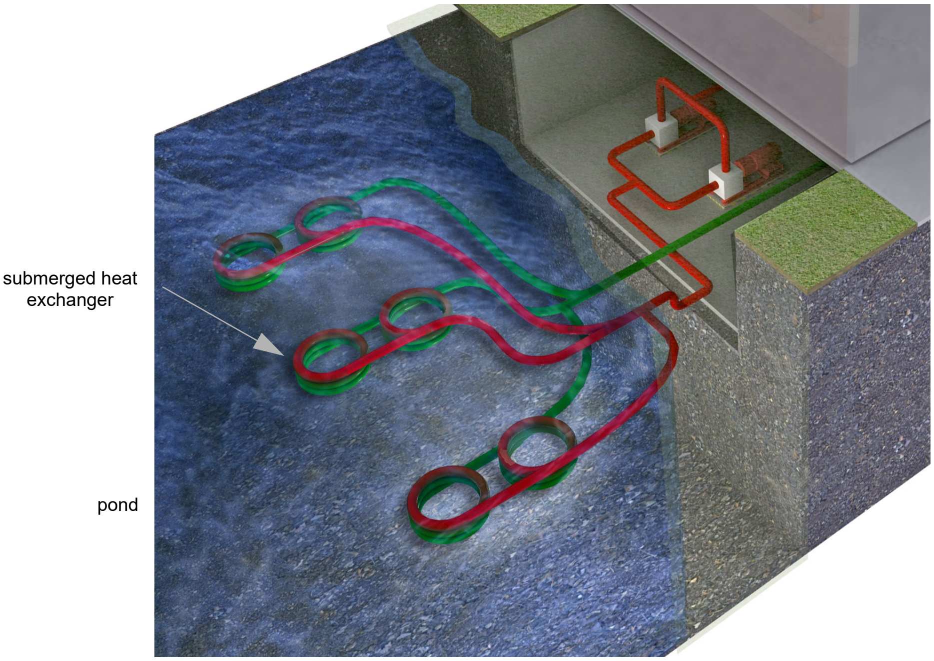


Surface-water heat pump systems

For more information on the design and layout of surface-water heat pump systems, refer to the ASHRAE manual, *Geothermal Heating and Cooling: Design of Ground-Source Heat Pump Systems*.

Rather than burying the heat exchanger underground, a surface-water heat pump (SWHP) system submerges the heat exchanger in a pond or lake (Figure 96). In a cooling-dominated application, a moderately sized pond or lake can provide enough heat rejection and heat addition to maintain proper temperatures in the water loop, without the need for a cooling tower or boiler.

Figure 96. Surface-water heat pump system



For a building site that already contains a pond, or if local building codes require water-retention ponds for short-term storage of surface run-off, a SWHP system can be very cost effective.

Surface-water systems typically use a series of closed loops of piping, similar to the spiral loop pattern used with some ground-coupled heat pump systems. The pipes are submerged in a pond or lake and secured to concrete anchors so they float 9 to 18 in. (23 to 46 cm) above the bottom of the pond, allowing for sufficient flow around the heat exchanger . The pipes should be submerged at least 6 to 8 ft (2 to 2.5 m) below the surface of the pond (preferably deeper), maintaining adequate thermal mass even during low-water levels.

Typical installations require 300 to 500 ft of pipe per “block” cooling ton (26 to 43 m/kW) and approximately 3000 ft^2 of surface water per cooling ton (79 m^2/kW). The recommended minimum total surface water area is about 20,000 ft^2 (1900 m^2).

This type of system will likely experience greater loop temperature variations than a ground-coupled system, which result in lower heat pump efficiencies. However, the lower installed cost may compensate for the reduction in efficiency.

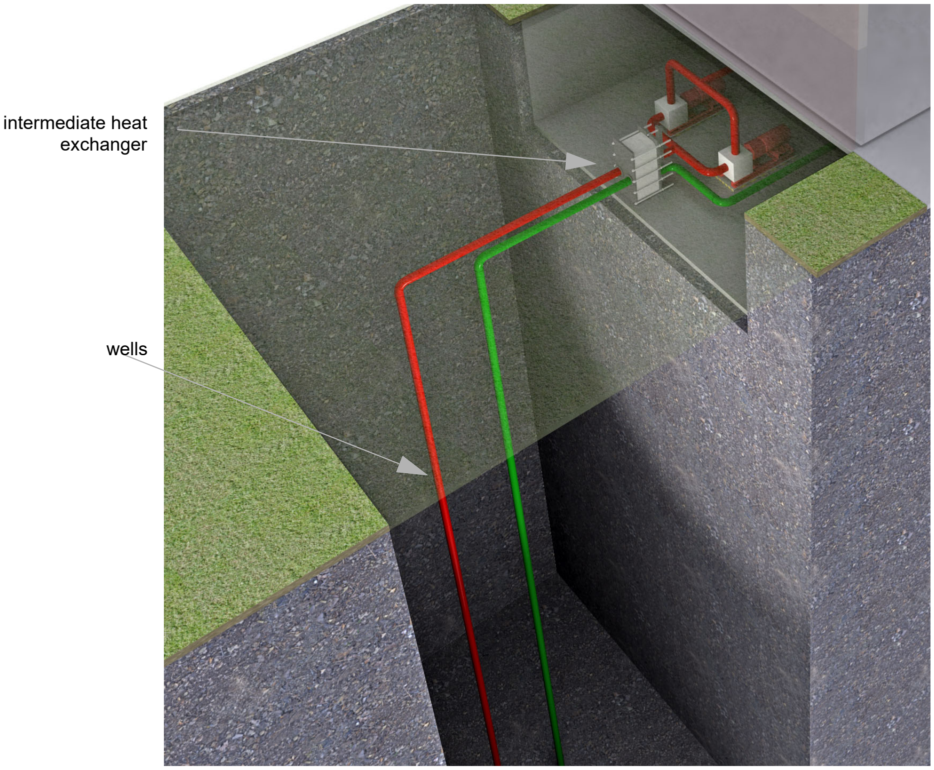
Note: Rivers, or other bodies of water that have appreciable currents, should be avoided. The currents have the potential to damage the heat exchanger.

Ground-water heat pump systems

For more information on the design and layout of ground-water heat pump systems, refer to the ASHRAE manual, *Geothermal Heating and Cooling: Design of Ground-Source Heat Pump Systems* and the *ASHRAE Journal* article titled "Commercial Open Loop Heat Pump Systems."

A ground-water heat pump (GWHP) system pumps water from a well, and then either returns the used water to the source through a separate re-injection well (Figure 97), or dumps the water into a drain field or sewer system. The water from the well might be circulated directly through each heat pump (an open system), but more commonly an intermediate heat exchanger is used to separate the well water from the water that circulates throughout the building (a closed system).

Figure 97. Ground-water heat pump system



For a building site where an existing (or proposed) well can provide an ample supply of suitable-quality water, a GWHP system might be feasible.

A ground-water system is typically the most efficient of any ground-source system because fluid temperatures in the loop are typically cooler during the cooling season, and warmer during the heating season, than in a ground-coupled system. Also, this type of system typically has the lowest installed cost of any ground-source system because it requires fewer wells and less length of piping.

There are, however, three notable considerations that must be addressed when investigating this use of this type of system.

The first consideration is water quality. If the ground water is allowed to flow through the heat pump refrigerant-to-water heat exchangers, they will be subject to fouling—minerals in the water scaling on the internal surfaces of the heat exchanger tubes—which reduces heat transfer and degrades efficiency. Scaling can be reduced by periodically cleaning the heat exchangers, but the acidic solutions used to de-scale the heat exchanger can also cause corrosion. For this application, the refrigerant-to-water heat exchanger is often made of cupronickel (CuNi), which has a higher tolerance for chlorides (see "Components of the DX refrigeration circuit," p. 14). However, the ASHRAE *Design of Ground-Source Heat Pump Systems* manual cautions that cupronickel does not prevent all water quality problems: "The alloy (CuNi) is ineffective (or only marginally better than copper) at addressing many problems commonly encountered in ground-water applications, such as hydrogen sulfide, low-pH corrosion, or iron, manganese, and carbonate scale or fouling."

To help ensure that the water flowing through the heat pumps is clean, an intermediate plate-and-frame heat exchanger is recommended. This eliminates the problem of scaling in the individual heat pumps, requiring that only the centrally-located, plate-and-frame heat exchanger be cleaned. However, the inefficiency of this additional heat exchanger results in a slightly warmer (during the cooling season) or slightly colder (during the heating season) fluid temperature in the loop, which decreases heat pump efficiency compared to an open, ground-water system. In some applications, two heat exchangers are installed so one can be cleaned while the other is in operation.

An alternate approach is to submerge the intermediate heat exchanger down into the aquifer, isolating the water in the aquifer from the water circulated through the wells. A pump in this heat exchanger circulates aquifer water through the heat exchanger to maximize heat transfer.

The second consideration is the need for an adequate supply of water. The water flow rate through a WSHP is typically between 1.5 and 3 gpm/ton (0.027 and 0.054 L/s/kW). In a commercial or institutional building with many heat pumps, this adds up to a significant quantity of water, often causing a ground-water heat pump system to be subject to local water-resource restrictions.

The final consideration is determining an acceptable way to discharge this large quantity of water after it returns from the heat pumps. The water is typically either re-injected into the ground through a separate well (or separate pipe in the same well) or discharged into a river, lake, or sewer system. The re-injection well must be placed far enough away from the supply well to prevent recirculation. Local codes and regulations may limit some of these practices.

Electrified WSHP System

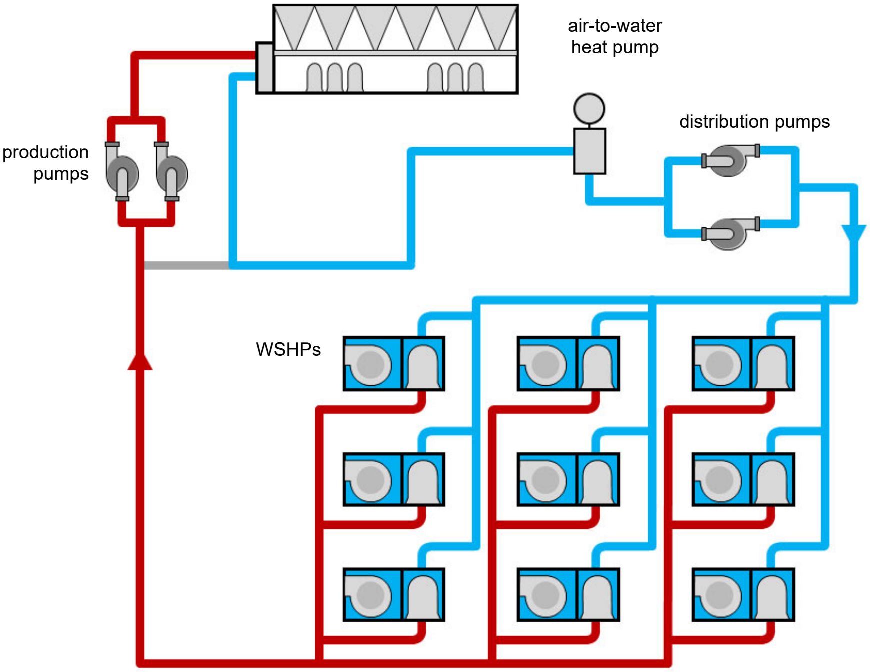
As noted in “Helps achieve decarbonization goals,” p. 10, one strategy commonly used to reduce the carbon dioxide equivalent footprint of a building (often referred to as **decarbonization**) is to reduce the use of fossil fuels by installing electrified HVAC equipment (often referred to as **electrification**).

The water-source heat pumps already use electricity for heating, but the rest of the system can be designed to further reduce on-site use of fossil fuels by either:

- Employing an electric hot-water boiler or electric resistance heaters (see “Electric resistance heat for a “boiler-less” system,” p. 56),
- Adding a ground heat exchanger to eliminate the need for a boiler altogether (see “Ground-Source Heat Pump Systems,” p. 135), or
- Using an air-to-water heat pump in place of both the cooling tower and boiler, which is the focus on this section.

The electrified WSHP system depicted in Figure 98 incorporates an air-to-water heat pump (AWHP) for both heat rejection and heat addition, replacing both the cooling tower and boiler. In addition to reducing on-site CO_2 emissions (by replacing the fossil-fuel boiler), this approach also reduces on-site water use (by replacing the evaporative cooling tower).

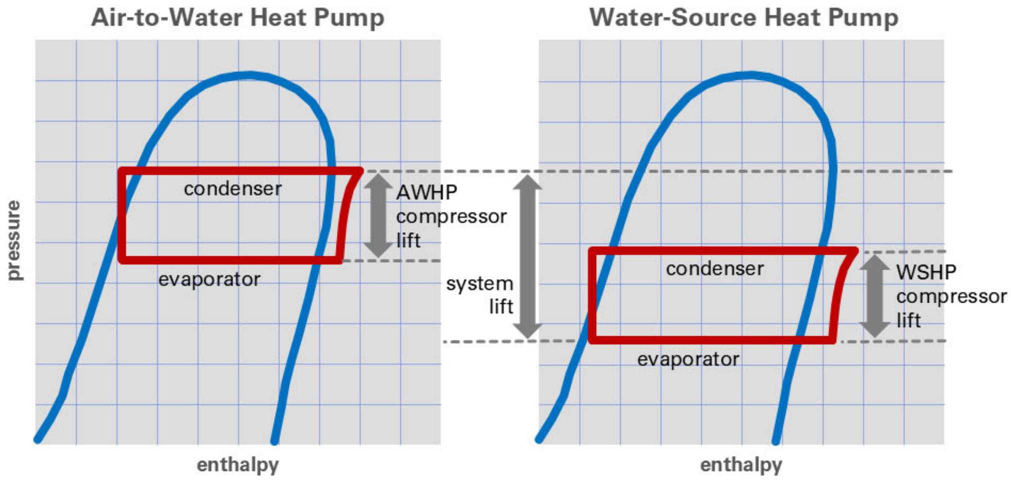
Figure 98. Electrified WSHP system with an air-to-water heat pump



Impact on efficiency and water use

Compressor power draw is proportional to lift, which is the difference between the condenser and evaporator temperatures. System efficiency is related to the total system lift, which in the case of this electrified WSHP system, is split between the AWHP and the WSHP (Figure 99). If the loop temperature is increased, the evaporator temperature in the AWHP warms, which reduces its compressor lift. Meanwhile, the condenser temperature in the WSHP warms, which increases its compressor lift. Total system lift is unchanged. Therefore, changing the water loop temperature does not directly affect system efficiency, it only affects which compressors draw more power.

Figure 99. Compressor lift in an electrified WSHP system

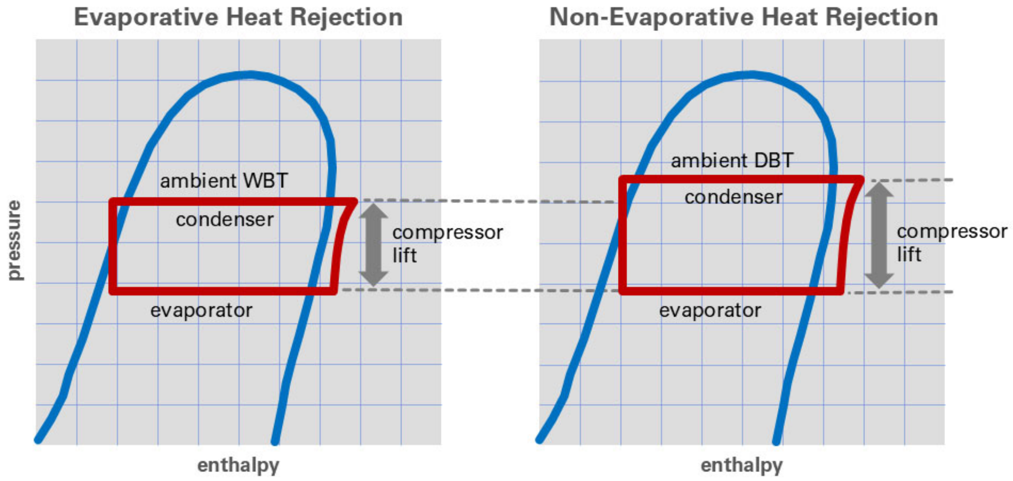


When retrofitting a traditional WSHP system to be an electrified WSHP system, it may be necessary to consider how this will affect system efficiency. In heating mode, this electrified WSHP system should achieve a higher system COP because an AWHP is replacing the boiler.

In cooling mode, however, the system COP may be lower because the AWHP requires more electrical power than a cooling tower. In addition, replacing an evaporative cooling tower with a non-evaporative AWHP will increase system lift. Figure 100 illustrates the difference in system lift between a conventional WSHP system, which rejects heat to the ambient wet-bulb temperature (WBT) using an evaporative cooling tower, versus an electrified WSHP system, which rejects heat to the ambient dry-bulb temperature (DBT) using an air-cooled condenser in the AWHP.

As mentioned previously, system efficiency is typically not the only driver for retrofitting to an electrified WSHP system. This approach also reduces on-site CO_2 emissions (by eliminating the fossil-fuel boiler) and reduces on-site water use (by eliminating the evaporative cooling tower). Cooling towers require a substantial quantity of makeup water due to evaporation loss, as well as “blow-down” operation (draining water from the cooling tower sump to manage water chemistry and levels of impurities).

Figure 100. Lift in a conventional WSHP system versus an electrified WSHP system



Sizing the AWHP

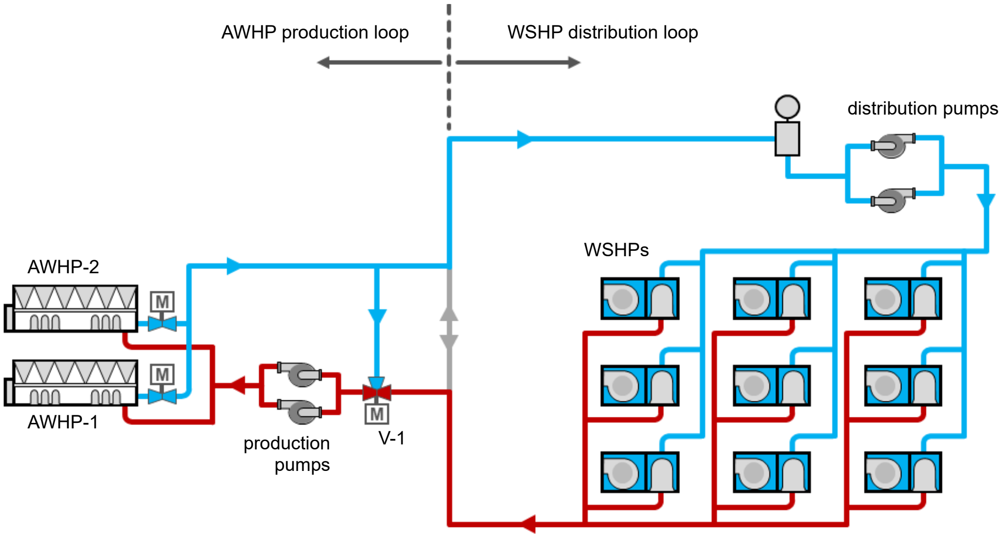
For an electrified WSHP system, the amount of heat that the water-source heat pumps reject to, and extract from, the water loop is the same as in a conventional, boiler/tower WSHP system. Therefore, the sizing criteria described previously can also be used to size the AWHP (see "Sizing the cooling tower," p. 43, and "Sizing the boiler in a system with night setback," p. 50).

Sizing the AWHP should be based on manufacturer's selection software that considers defrost and ambient temperature limitations. If expected ambient temperatures are below the operational limits of the AWHP, consider including an auxiliary boiler (see "Optional system components," p. 159).

Pumping and flow management

This electrified WSHP system includes a production loop (connected to the AWHPs) and a distribution loop (connected to the WSHPs). The loops are hydraulically decoupled with separate distribution and production loop pumps (Figure 101).

Figure 101. Decoupled production and distribution loops



This decoupled arrangement allows the loop flow rates to differ and be optimized to ensure reliable and stable operation of the AWHPs at any load point. This is important for the following reasons:

- The flow rate in the distribution loop is likely to vary widely, from full design flow to a very low flow rate when only a few WSHP compressors are operating.
- The AWHPs likely allow a relatively narrow range of flow rates in the production loop (particularly in heating mode), depending on how many units are operating at a time.
- The AWHP design flow rate for heating mode may differ from its design flow rate for cooling mode.
- The AWHP design flow rate may be near its minimum allowable flow rate, prohibiting or limiting variable flow; this is particularly likely in heating mode.

For more information on pump control for AWHPs, refer to the Trane application guide, *Air-to-Water Heat Pump System with Cascade Option* (SYS-APG003*-EN).

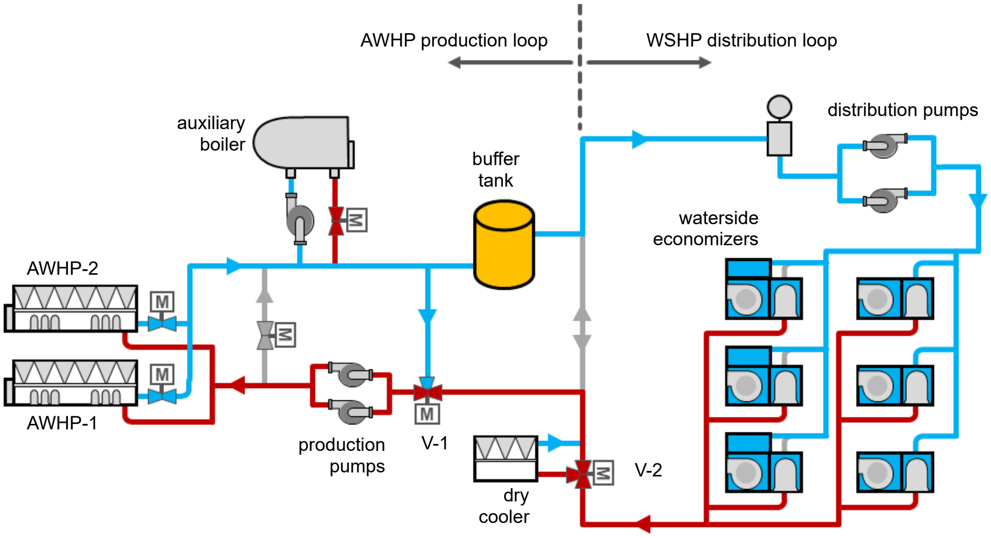
The distribution loop pumps may be controlled for constant or variable flow (see "Constant- versus variable-flow pumping," p. 32), while the production loop pumps are controlled to maintain the design flow rate through the operating AWHPs. As mentioned above, the AWHP design flow rate may differ for heating mode versus for cooling mode. The production pumps may be operated at different speeds, depending on the operating mode of the AWHP, or may be set to operate at the higher of the design flow rates (typically cooling mode). If dual-speed pump control is used, ensure that production pump selections can achieve the full range of flow rates.

A tempering valve (V-1) in the production loop can recirculate fluid leaving the AWHPs to ensure that the temperature of the fluid entering the AWHPs remains within the acceptable range.

Optional system components

Figure 102 depicts several optional components that might be added to the example electrified WSHP system.

Figure 102. Electrified WSHP system with optional components



A **dry cooler** (or cooling tower) could be added to reject heat directly to the atmosphere, when conditions allow, without needing to operate the AWHPs. If the peak cooling load of the building is higher than the peak heating load, this might also reduce the required AWHP capacity. In this layout, a diverting valve (V-2) is used to direct distribution loop flow through the dry cooler (or a dedicated pump could be used instead). Alternatively, a closed-circuit cooling tower could be used to improve annualized system efficiency, but this would increase water usage.

An **auxiliary boiler** could be used to add heat to the water loop, offsetting high peak heating loads without needing to upsize the AWHPs. In many applications, AWHPs sized for half of the peak heating load can offset the vast majority of heating loads throughout the year, maybe as much as 90 percent of the hours. Also, a backup boiler might be needed if the design ambient temperature is below the operational limits of the AWHP.

Some or all of the WSHPs could be equipped with a **waterside economizer**, enabling direct cooling of recirculated air by the water loop while the WSHP compressor can be turned off (see "Economizer control," p. 170).

A **buffer tank** might be added to ensure adequate fluid volume in the loop. The minimum loop volume for an AWHP is based on limiting disturbances to the hot-water supply temperature during defrost mode. Consult the manufacturer for specific recommendations, but a good rule-of-thumb is to ensure that the "loop time" is longer than the defrost cycle duration (5 minutes, for example). This "loop time" is calculated by dividing the total volume of fluid in the loop (in gallons or

Liters) by the design flow rate (in gpm or L/min). If the system includes more than one AWHP, use the design flow rate of only one AWHP (assumes that only one AWHP will be operating in defrost mode at a given time).

Unlike a cooling tower that can be drained in the winter, the AWHP must continue operating during cold winter. Therefore, if outdoor temperatures are expected to drop below freezing, this system will require a minimum of 25 percent **glycol concentration**; a higher concentration may be required depending on expected extreme winter temperatures. This glycol concentration needs to be high enough to provide full "freeze" protection, not only "burst" protection, since the fluid needs to be pumped during winter operation (see Table 7). The volume of glycol required in the system can be reduced by placing a heat exchanger between the indoor distribution loop and the outdoor production loop.

Excerpt from Section 6.5.2.2.3 in ASHRAE Standard 90.1-2022:

6.5.2.2.3 Hydronic (Water Loop) Heat Pump Systems. 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) shall have the following:

a. Controls that are capable of and configured to provide a heat-pump water supply temperature deadband of at least 20°F (11°C) between initiation of heat rejection and heat addition by the central devices (e.g., tower and boiler)...

Exception to 6.5.2.2.3: Where a system loop temperature optimization controller is used to determine the most efficient operating temperature based on real-time conditions of demand and capacity, deadbands of less than 20°F (11°C) shall be allowed.

Loop temperature control

As explained in "Impact on efficiency and water use," p. 156, changing the water loop temperature does not directly affect the efficiency of this electrified WSHP system, it only affects which compressors draw more power. So then how do the loop temperature control requirements in ASHRAE Standard 90.1 affect this type of system?

Section 6.5.2.2.3 of ASHRAE 90.1 includes a requirement for a temperature deadband "between initiation of heat addition and heat rejection by the central devices" (see sidebar). This deadband is intended to ensure that the cooling tower and boiler will not operate simultaneously, and improves heat recovery potential in a WSHP system by allowing the loop temperatures to drift. This deadband requirement was originally developed for conventional boiler/tower WSHP systems, and the ASHRAE committee likely did not anticipate the use of electrified WSHP systems. Nevertheless, these requirements can still serve as a valuable framework for controlling an electrified WSHP system.

The allowable range for the heating and cooling setpoints in an AWHP differ from product to product. Table 29 shows the allowable ranges for three example products. In some cases, the allowable ranges may not allow for 20°F (11°C) of separation between setpoints.

Table 29. Example ranges for allowable setpoints in AWHP products

Example Product	Allowable Range for its Heating Setpoint	Allowable Range for its Cooling Setpoint	Range in Which the Setpoints Can Overlap
Product A	50°F to 140°F (10.0°C to 60.0°C)	20°F to 75°F (-6.7°C to 23.9°C)	50°F to 75°F (10.0°C to 23.9°C)
Product B	55°F to 140°F (12.8°C to 60.0°C)	20°F to 65°F (-6.7°C to 18.3°C)	55°F to 65°F (10.0°C to 18.3°C)
Product C	75°F to 140°F (23.9°C to 60.0°C)	20°F to 65°F (-6.7°C to 18.3°C)	no overlap

However, note that this deadband requirement in ASHRAE 90.1 is based on "initiation of heat rejection and (initiation of) heat addition." That is, there needs to be a 20°F (11°C) difference in loop temperature between when the AWHP starts in heating mode (Initiate Heating setpoint) versus when it starts in cooling mode (Initiate Cooling setpoint). The *Standard 90.1-2019 User's Manual* clarifies: "Note that this section's 20°F (11°C) deadband requirement only establishes the capability of the control system, not the actual setpoints."

The following terminology is used in this discussion of loop temperature control:

- Heating Setpoint: The leaving-water temperature that the AWHP controller attempts to maintain when operating in heating mode.
- Cooling Setpoint: The leaving-water temperature that the AWHP controller attempts to maintain when operating in cooling mode.
- Differential-to-Start: How far the loop temperature is allowed to drop below the Heating Setpoint (or to rise above the Cooling Setpoint) before the AWHP starts.
- Differential-to-Stop: How far the loop temperature is allowed to rise above the Heating Setpoint (or to drop below the Cooling Setpoint) before the AWHP shuts off.
- Initiate Heating: The temperature at which the AWHP turns on in heating mode; this is the Heating Setpoint minus the Differential-to-Start.
- Initiate Cooling: The temperature at which the AWHP turns on in cooling mode; this is the Cooling Setpoint plus the Differential-to-Start.
- Terminate Heating: The temperature at which the AWHP turns off, when operating in heating mode; this is the Heating Setpoint plus the Differential-to-Stop.
- Terminate Cooling: The temperature at which the AWHP turns off, when operating in cooling mode; this is the Cooling Setpoint minus the Differential-to-Stop.

For this electrified WSHP system, loop temperature control must be configured to achieve the following outcomes:

- At least a 20°F (11°C) difference between the Initiate Heating setpoint and the Initiate Cooling setpoint.
- The Terminate Heating setpoint must be lower than the Initiate Cooling setpoint. Table 30 uses a minimum separation of 5°F (2.8°C) between these setpoints.
- The Terminate Cooling setpoint must be higher than the Initiate Heating setpoint. Table 30 uses a minimum separation of 5°F (2.8°C) between these setpoints.
- Heating Setpoint and Cooling Setpoint must be withing the allowable range for the specific AWHP product.

System Design Variations

Table 30 suggests loop temperature control setpoints for the three example products depicted in Table 29.

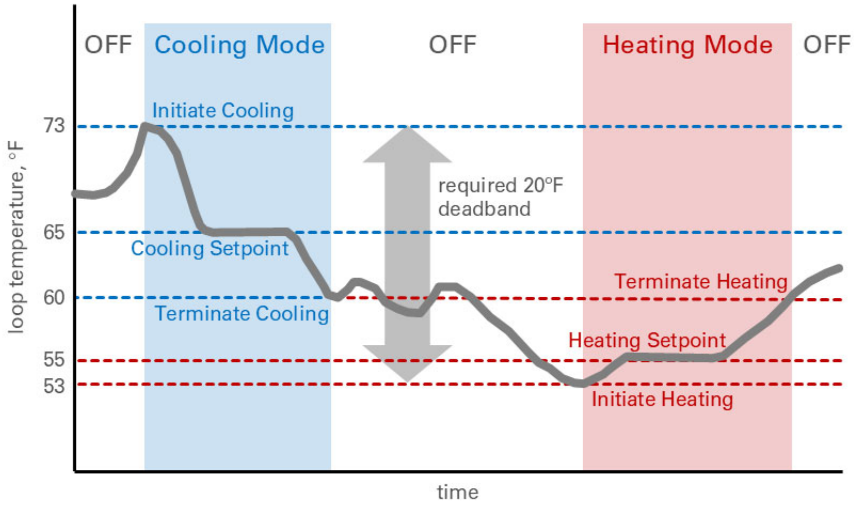
Table 30. Example loop temperature control setpoints for an electrified WSHP system

Example Product	Initiate Heating	Heating Setpoint	Terminate Heating	Initiate Cooling	Cooling Setpoint	Terminate Cooling
Product A	48°F (8.9°C)	50°F (10.0°C)	55°F (12.8°C)	77°F (25.0°C)	75°F (23.9°C)	70°F (21.1°C)
Product B	53°F (11.7°C)	55°F (12.8°C)	60°F (15.6°C)	73°F (22.8°C)	65°F (18.3°C)	60°F (15.6°C)
Product C	58°F (14.4°C)	75°F (23.9°C)	77°F (25.0°C)	82°F (27.8°C)	65°F (18.3°C)	63°F (17.2°C)

For Product A, the allowable setpoint ranges allow for a 20°F (11°C) difference between the Initiate Heating setpoint and the Initiate Cooling setpoint. In this case, the deadband is maximized to allow for the AWHP to operate as efficiently as possible: using the lowest Heating Setpoint allowed and the highest Cooling Setpoint allowed. The Differential-to-Start is 2°F (1.1°C) and Differential-to-Stop is 5°F (2.8°C).

For Product B, the Differential-to-Start for cooling needed to be increased to 8°F (4.4°C) to comply with the required 20°F (11°C) difference between the Initiate Heating and Initiate Cooling setpoints. Figure 103 depicts loop temperature control using the setpoints listed for Product B in Table 30. In the morning, the loop temperature begins to drift upward. When it reaches the Initiate Cooling setpoint of 73°F (22.8°C), the AWHP starts in cooling mode and cools the loop to the Cooling Setpoint of 65°F (18.3°C). Later, when the loop over-cools to the Terminate Cooling setpoint of 60°F (15.6°C), the AWHP turns off and the loop temperature is allowed to drift. When it reaches the Initiate Heating setpoint of 53°F (11.7°C), the AWHP starts in heating mode and warms the loop to the Heating Setpoint of 55°F (12.8°C). Later, when the loop over-heats to the Terminate Heating setpoint of 60°F (15.6°C), the AWHP turns off and the loop temperature is allowed to drift again.

Figure 103. Example loop temperature control for “Product B”



For Product C, the Differential-to-Start (for both cooling and heating) needed to be increased to 17°F (9.4°C) to comply with the required 20°F (11°C) difference between the Initiate Heating and Initiate Cooling setpoints. The Differential-to-Stop, for both cooling and heating, needed to be decreased to 2°F (1.1°C) to keep the

Terminate Heating setpoint at least 5°F (2.8°C) lower than the Initiate Cooling setpoint, and to keep the Terminate Cooling setpoint at least 5°F (2.8°C) higher than the Initiate Heating setpoint.

What about defrost mode for the AWHP? When the ambient temperature drops below 47°F (8°C), frost may build up on the outdoor coil of an AWHP when it operates in heating mode. When defrost mode is initiated, the unit controller temporarily reverses the refrigeration cycle, causing the outdoor coil to warm up and melt this frost. This defrost cycle takes about 5 minutes. During this time window, the hot-water supply temperature may drift colder. Fortunately, WSHPs are designed to handle a wide range of entering fluid temperatures; many can operate in heating mode with an entering fluid temperature of 30°F (-1°C) or colder. Consult the specific manufacturer for operational limitations.

Hybrid WSHP System Configurations

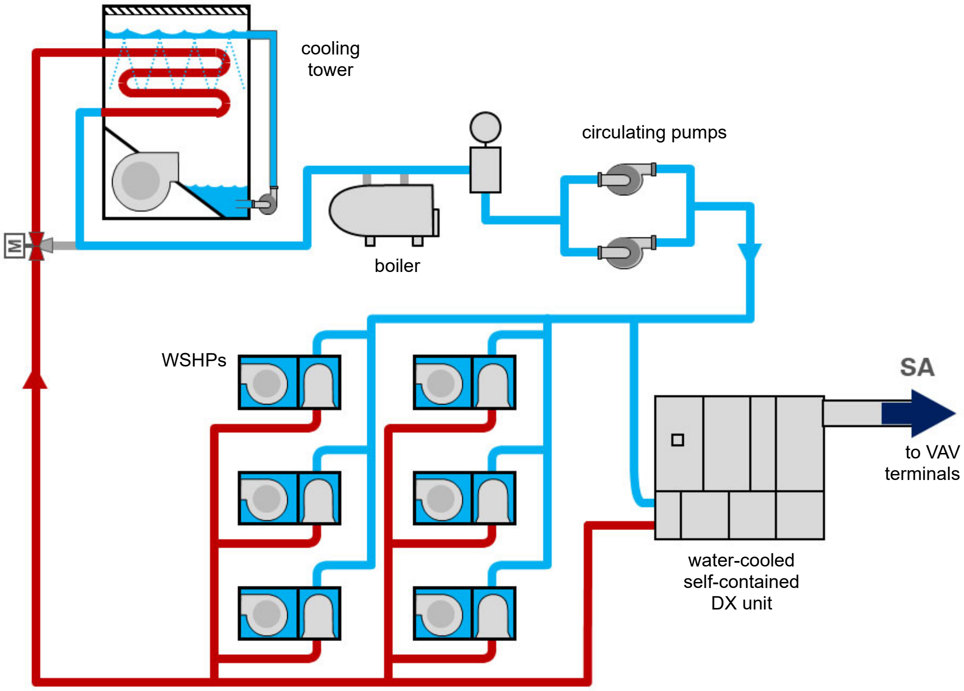
The WSHP system is often viewed solely as an alternative to other types of HVAC systems. “Hybrid” systems comprised of water-source heat pumps and other types of HVAC equipment, however, may be best suited to meet the specific requirements of a given building.

While there are many possible combinations, this section includes two examples.

Water-cooled, self-contained VAV systems used to serve interior zones

The example hybrid WSHP system shown in Figure 104 uses a variable-air-volume (VAV), water-cooled self-contained DX unit to serve the interior zones on each floor of a multi-story building and WSHPs to serve the perimeter zones. These self-contained units include all the components of the refrigeration circuit, including a water-cooled condenser that is connected to the same water distribution loop that serves the heat pumps.

Figure 104. Hybrid WSHP system with self-contained VAV air conditioners



Since the interior zones in this example have variable loads, but will nearly always require cooling, the VAV terminal units vary the airflow supplied to those interior zones. At reduced cooling loads, the VFD on the supply fan results in part-load fan energy savings. Heat is rejected from the self-contained units into the common water loop, where it can be extracted by the heat pumps that are providing heat to the perimeter zones.

Water-to-water heat pump or water-cooled chiller to serve air-handling units (AHUs) or terminal units

Another example of a hybrid WSHP system, shown in Figure 50, p. 77, includes a water-to-water heat pump (which is connected to the water distribution loop) serving an air-handling unit (AHU) that conditions all of the outdoor air required for ventilation (see "Dedicated OA equipment types," p. 73).

As explained previously, this configuration allows the cooling/heating equipment to be distributed throughout the building, often located very close to the dedicated OA unit that it is connected to. This may be advantageous for buildings that are large (in terms of floor area) but are only one or two floors, such as a K-12 school or an extended care facility.

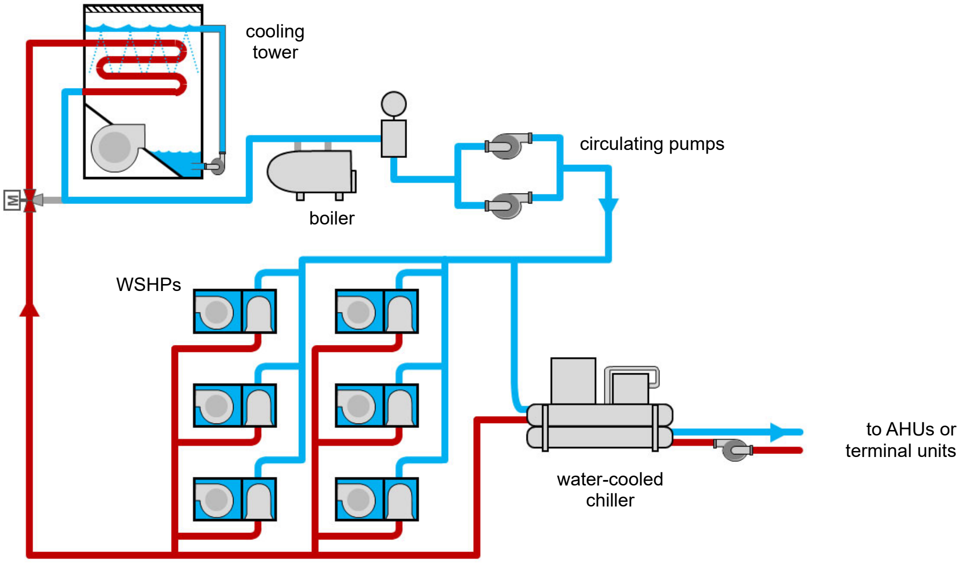
An alternative approach is to use a centralized, water-cooled chiller (which is also connected to the water distribution loop) to serve multiple dedicated OA units. Rather than connecting the water-cooled chiller to a separate cooling tower, the chiller condenser is connected to the same water distribution loop that is used by the heat pumps (Figure 105).

This configuration allows the cooling/heating equipment used to condition the outdoor air to be centralized. This may be advantageous for multi-story buildings that include only a few dedicated OA air-handling units, such as an office building, hotel or apartment building. This chiller could also serve other terminal units (such as fan-coils) that are used in cooling-only or specialty spaces.

For more information on using a centralized chiller/heater in a ground-coupled system, refer to the Trane application manual, *Central Geothermal Systems* (SYS-APM009*-EN).

The chiller may be used for cooling only or it might be piped into the water distribution loop to allow it to be used for either cooling or heating. This concept can be used with a conventional boiler/tower WSHP system or with a ground-coupled system.

Figure 105. Hybrid WSHP system with a water-cooled chiller



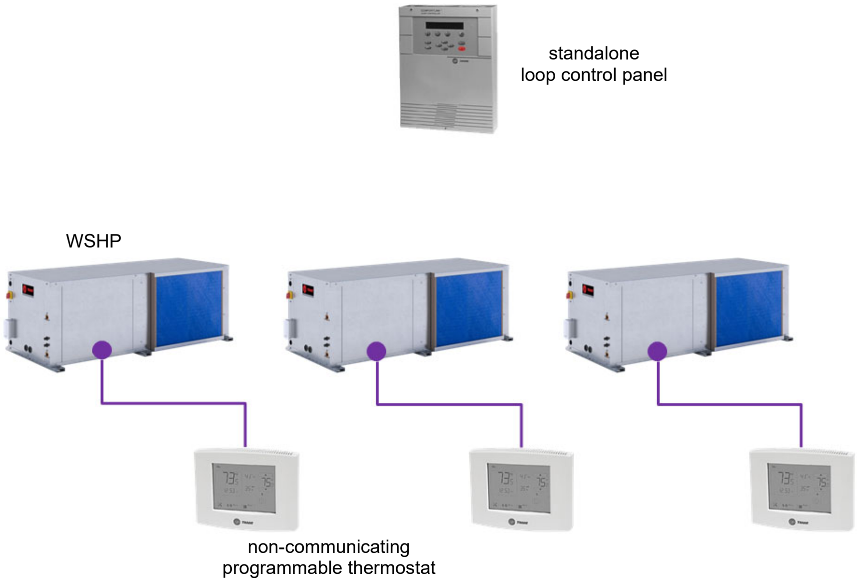
System Controls

This chapter discusses the control of a water-source heat pump system. Unit-level control refers to the functions required to control and protect each individual piece of equipment. System-level control refers to the intelligent coordination of the individual pieces of equipment so they operate together as a reliable, efficient system.

Historically, some WSHP systems have been installed with very simplistic controls (some might argue they were too simplistic). Like other systems, however, WSHP system controls have advanced to make use of communicating, digital controls.

- **Non-communicating thermostat control.** The lowest level of control typically uses a non-communicating, mechanical thermostat for each WSHP and a standalone control panel in the mechanical room (Figure 106). Installed cost is low because no communication wire needs to be pulled to connect the individual heat pumps to the centralized loop control panel. However, functionality is limited.

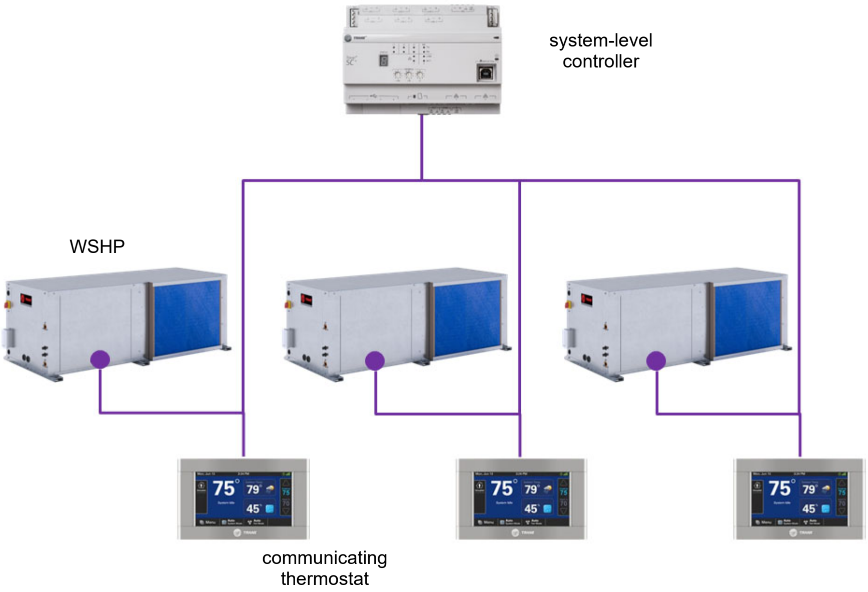
Figure 106. Non-communicating thermostat control



With no communication between the individual heat pumps and loop control panel, the water-circulating pumps are often operated continuously. To avoid this, a daisy-chained wire can be used to connect to a relay in each heat pump, allowing the centralized loop control panel to disable the heat pumps and turn off the water-circulating pumps based on a time-of-day schedule. Unoccupied operation—to maintain setback temperatures—can be enabled using either a single temperature sensor in a representative zone, or a second set of relays can be daisy-chained to allow any zone to request after-hours operation of the system. Of course, when wires need to be pulled to connect the individual heat pumps, it would be more beneficial to use communication wire and take advantage of using networked unit controllers.

- **Communicating thermostat control.** One step up in functionality is to use a communicating thermostat to control each WSHP (Figure 107). Communicating thermostats are more expensive than non-communicating thermostats, but likely less expensive than networked (communicating) unit controllers with zone temperature sensors.

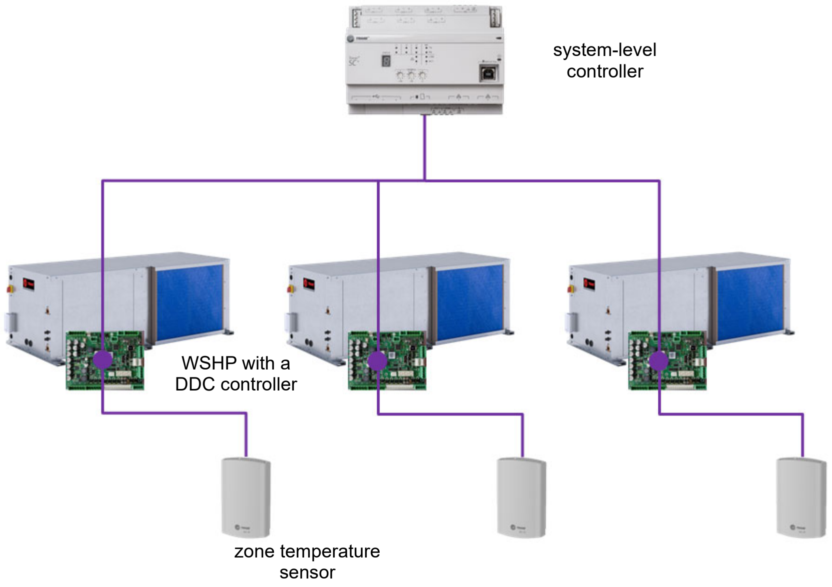
Figure 107. Communicating thermostat control



This configuration allows basic information about the zone to be shared with a centralized system-level controller. For example, any zone can request after-hours operation of the system to maintain setback temperatures, and the system-level controller can be used to centralize system scheduling. However, since the individual unit controllers do not communicate with the system-level controller, this approach does not provide for centralized alarms, troubleshooting, or trend logging of equipment operation.

- **Networked unit controllers with zone sensors.** Achieve more functionality, including the potential for greater energy savings, by using a communicating controller on each WSHP. All the controllers are connected to a network, communicating with a centralized system-level controller (Figure 108). The unit controllers may be more expensive, but the zone temperature sensors will likely be less expensive than either communicating or non-communicating thermostats.

Figure 108. Networked unit controllers with zone sensors



Wireless communications eliminates the wires between the zone temperature sensor and the unit controller on each water-source heat pump, as well as the wires between each unit controller and the centralized, system-level controller (see "Using wireless technology," p. 89). Benefits include faster project completion and easier relocation when space layout or use changes in the future, as well as making it easier to upgrade an older system to reap the benefits of networked unit controls.

This configuration provides an opportunity to optimize system operation with capabilities such as:

1. Centralized system scheduling and shutdown based on occupancy
2. Override to allow the system to operate when a zone is occupied after scheduled hours (indicated through the use of a timed override button on the zone sensor)
3. Enabling morning warm-up and cool-down sequences, including optimal start
4. Centralized alarms to indicate problems, required service, or needed maintenance
5. Trend logging to help anticipate potential system problems
6. Integration with the dedicated outdoor-air unit, or other equipment/systems serving the building, such as lighting, security, and fire safety

Unit-Level Control

Unit-level control for a piece of HVAC equipment typically involves the use of several control loops to employ specific functions, plus various safeties to protect the equipment. In addition, alarms and diagnostic messages assist the building operator or service personnel in responding to, or preventing, problems with the equipment.

While this section identifies many of the unit-level control functions for the primary components of a WSHP system, specific details should be obtained from the equipment manufacturer. Extended discussions in this section are limited to those unit-level control issues that require decisions to be made by the HVAC system designer or system operator.

Water-source heat pump

Typically, each water-source heat pump is equipped with a dedicated, unit-level controller. This controller typically performs the following functions:

Zone temperature control

A sensor in each zone measures the dry-bulb temperature in that zone. The unit-level controller compares this measured temperature to the desired setpoint. If the zone requires cooling, the controller responds by cycling the compressor to match the changing cooling load in the zone. As the cooling load decreases, the compressor operates for a shorter period of time between cycles.

If the zone requires heating, the controller activates the reversing valve to switch operation of the refrigeration circuit to the heating mode, and then cycles the compressor to match the changing heating load in the zone. As the heating load decreases, the compressor operates for a shorter period of time between cycles.

Traditionally, smaller-capacity heat pumps have contained a single, constant-speed compressor that cycles on and off. Larger-capacity heat pumps often have multiple compressors, allowing for multiple stages of capacity control. Currently, some heat pumps are equipped with either a two-stage, variable-capacity, or variable-speed compressor (see “Methods of compressor capacity modulation,” p. 15). Compared to the on/off compressor historically used in this type of equipment, these newer methods of compressor capacity modulation are better able to match cooling or heating capacity with the changing load in the zone, often resulting in more stable temperature control.

Typically, the fan inside the WSHP operates continuously during the occupied mode. However, if a dedicated outdoor-air system delivers conditioned OA directly to each zone, the fan inside the WSHP can be configured to cycle off whenever the compressor cycles off, reducing fan energy use. In addition, if the heat pump is equipped with either a two-stage, variable-capacity, or variable-speed compressor, or if it includes more than one compressor, the fan may be controlled to operate at a reduced speed (and airflow) when the compressor operates at reduced capacity (see “Multiple-speed fan operation,” p. 21).

If an electric resistance heater is installed inside the WSHP or in the downstream ductwork (see “Electric resistance heat for a “boiler-less” system,” p. 56), coordination of compressor and electric heater operation should be handled by the unit-level controller to prevent simultaneous cooling and heating, which wastes energy and is prohibited by many building energy codes. In this configuration, a temperature sensor is installed on the entering-water pipe of the WSHP. The heat pump operates the compressor in normal heating mode until the temperature of the entering water drops below a pre-determined low limit, 55ºF (13ºC) for example. At that time, the compressor is disabled and the electric resistance heater is energized to provide heat to the zone. When the loop temperature rises again—to 60ºF (16ºC) for example—the electric resistance heater is disabled and the heat pump compressor is again allowed to operate in the normal heating mode.

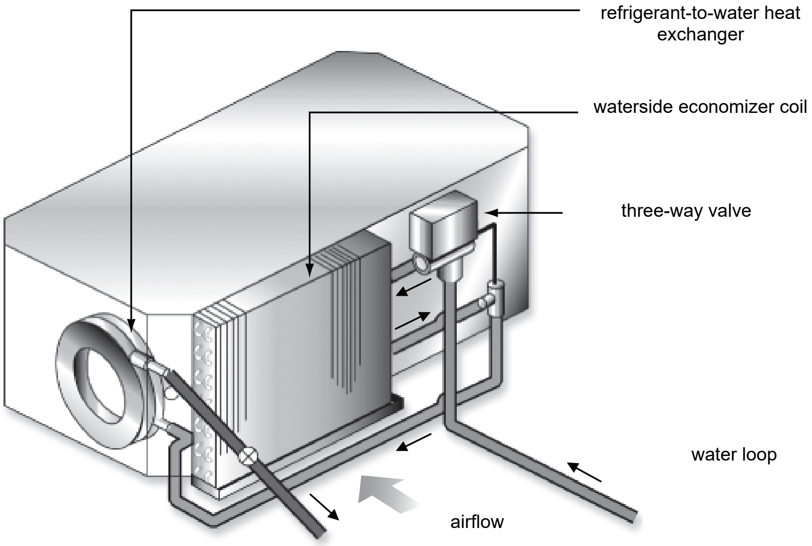
Economizer control

The energy consumed by a WSHP system can often be reduced through the use of a waterside or airside economizer.

A **waterside economizer** can provide an inexpensive means of cooling when used in systems that require perimeter heating and interior cooling. During cold weather, the heat pumps serving interior zones often operate in the cooling mode because of the heat generated by lights, people, and office equipment. Simultaneously, the heat pumps serving perimeter zones may be operating in the heating mode, extracting heat from the loop and lowering the water temperature. This cool loop water can be circulated through a waterside economizer coil mounted in a WSHP that serves an interior zone, providing “free” cooling without the need to operate the heat pump compressor.

In the example shown in Figure 109, a temperature sensor is installed on the pipe entering the WSHP. When the entering-water temperature is below the economizer enable setpoint—50ºF (10ºC) for example—the unit-level controller positions a three-way valve to divert the cool loop water through the waterside economizer coil to cool the entering air. This water then passes through the refrigerant-to-water heat exchanger. This piping configuration allows the economizer coil to be bypassed when not in use, which reduces pump energy use. It also allows the waterside economizer to operate in the “integrated economizer” mode, whereby both the economizer and compressor are used simultaneously to satisfy the cooling load.

Figure 109. Waterside economizer in a horizontal WSHP



If the zone requires cooling and the entering-water temperature is below the economizer enable setpoint, the three-way valve diverts the cool loop water through the waterside economizer coil. If the economizer cannot sufficiently cool the zone by itself, then the unit-level controller will cycle on the compressor to provide more cooling capacity. In this manner, both the waterside economizer and compressor are used simultaneously.

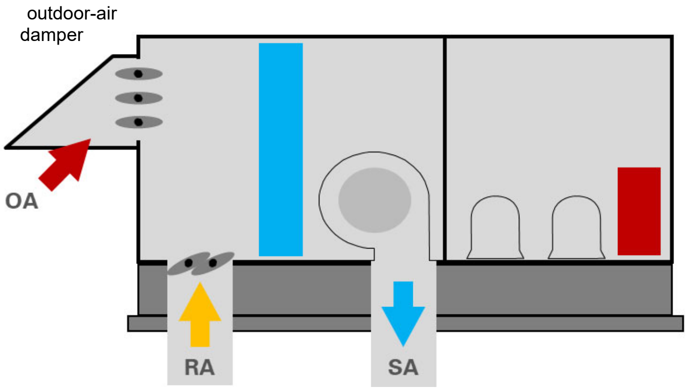
Unlike waterside economizers that are used with other types of HVAC systems, the waterside economizer in a WSHP system typically requires no use of cooling tower energy to create the colder water for free cooling. The cold water is produced by the heat pumps that are serving the perimeter zones, which would be operating in the heating mode regardless.

In a boiler/tower WSHP system, the boiler is controlled to prevent the water loop temperature from dropping below a pre-defined lower setpoint—60ºF (16ºC) for example. However, to allow a waterside economizer to provide a significant benefit, this temperature setpoint may need to be set lower than normal—colder than 60ºF (16ºC) for example—and it must be lower than the economizer enable setpoint. Allowing a colder loop temperature will increase the benefit of waterside economizing in those zones that require cooling, but it will reduce the efficiency of any WSHP compressors that are operating in the heating mode. Therefore, these two setpoints need to be carefully selected to minimize overall system energy use (see "Economizers," p. 116).

Alternatively, the system-level controller could be programmed to automatically change the lower loop temperature setpoint based on current operating conditions. For example, if most of the heat pumps are operating in the cooling mode, the system-level controller could allow the loop temperature to drift colder—to 50°F (10°C), for example—increasing the benefit of the waterside economizers. However, if most of the heat pumps are operating in the heating mode, the controller could maintain a warmer loop temperature—60°F (16°C), for example—to increase the efficiency of the compressors operating in the heating mode.

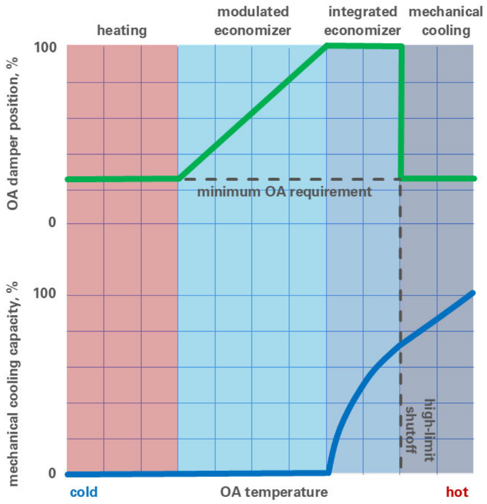
An **airside economizer** is probably the most well-known type of economizer. It uses cool outdoor air as a source of "free" cooling whenever possible. For WSHP systems, its use is typically limited to rooftop-style WSHPs, since they are equipped with an outdoor-air damper (Figure 110).

Figure 110. Airside economizer in a rooftop WSHP



When it is cold outside and the zone requires heating, the outdoor-air damper is closed to a minimum position to bring in the minimum quantity of outdoor air required for ventilation (Figure 111).

Figure 111. Typical airside economizer control for a constant-volume WSHP



As the weather warms, and the zone requires cooling, the linked outdoor- and return-air dampers modulate to bring in more of the cool outdoor air (Figure 111). This is called “modulated economizer” mode. In this mode, the outdoor air is cool enough to provide all the needed cooling capacity to maintain zone temperature at setpoint, without needing to operate the WSHP compressor.

As the cooling load increases, the outdoor-air damper eventually opens to 100 percent and the return-air damper closes completely. To provide the extra cooling capacity needed to maintain the zone temperature at setpoint, the compressor is cycled on. This is called “integrated economizer” mode; 100 percent outdoor airflow provides part of the required cooling capacity and mechanical cooling provides the balance (Figure 111).

At some point, the outdoor air becomes so warm that it provides little or no cooling benefit. When the outdoor-air temperature (or enthalpy) reaches the “high-limit shutoff” setpoint, the unit controller disables airside economizer operation and the outdoor-air damper is closed to a minimum position to bring in only the quantity of outdoor air required for ventilation (Figure 111). At this point, the compressor provides all cooling capacity needed to maintain zone temperature at setpoint.

The three most common high-limit shutoff strategies used to control the airside economizer in a WSHP are:

- **Fixed dry-bulb control** uses a sensor to measure the dry-bulb temperature of the outdoor air. The controller compares this temperature to a predetermined high-limit shutoff setting, and disables the economizer whenever the outdoor dry-bulb temperature is above this limit.

  This method is simple, reliable, and relatively inexpensive. However, in non-arid climates, if the high-limit shutoff setting is too high, this control strategy can bring in cool but humid outdoor air, which may raise indoor humidity levels and increase compressor energy use.
- **Fixed enthalpy control** uses sensors to measure both the dry-bulb temperature and humidity of the outdoor air. The controller then calculates the enthalpy of the outdoor air and compares it to a predetermined high-limit

shutoff setting. The economizer is disabled whenever the outdoor-air enthalpy is above this limit.

This method requires an added outdoor humidity sensor, so it costs more than fixed dry-bulb control. But in most climates, fixed enthalpy control helps prevent bringing in cool, humid outdoor air for economizing, and typically results in more compressor energy savings.

In hot and dry climates, however, there are times when bringing in 100 percent outdoor air can actually increase mechanical cooling, even if the outdoor-air enthalpy is low.

For more information on the various methods of controlling airside economizers, and their impact on system performance, refer to the Trane *Engineers Newsletter* titled “Airside Economizers and ASHRAE Standard 90.1” (ADM-APN054-EN).

- **Differential (or comparative) enthalpy control** uses sensors to measure both the dry-bulb temperature and humidity of both the outdoor air and return air. The controller calculates the enthalpy of both air streams, and uses the lower-enthalpy air to satisfy the cooling load. The economizer is disabled whenever the outdoor-air enthalpy is higher than the return-air enthalpy.

  The installed cost of differential enthalpy control is higher than for the other control methods, because it requires humidity sensing for both outdoor and return air. But it can result in the most compressor energy saved, compared to the other control types.

Climate, building use, and utility costs impact the operating cost differences of these different methods of airside economizer control.

When the outdoor air is cool enough that the airside economizer provides all the needed cooling capacity, the compressor is shut off and the WSHP is not rejecting heat to the water loop. Without heat being rejected from those heat pumps that are operating in cooling mode, the heat pumps that are operating in heating mode may cause the loop water temperature to drop to the point where the boiler must be activated. The energy used by the boiler may exceed the energy saved by turning off the compressors (and using the airside economizer) in the heat pumps operating in cooling mode. Therefore, consider disabling the airside economizer if the temperature of the water loop drops too far—below 65ºF (18ºC), for example.

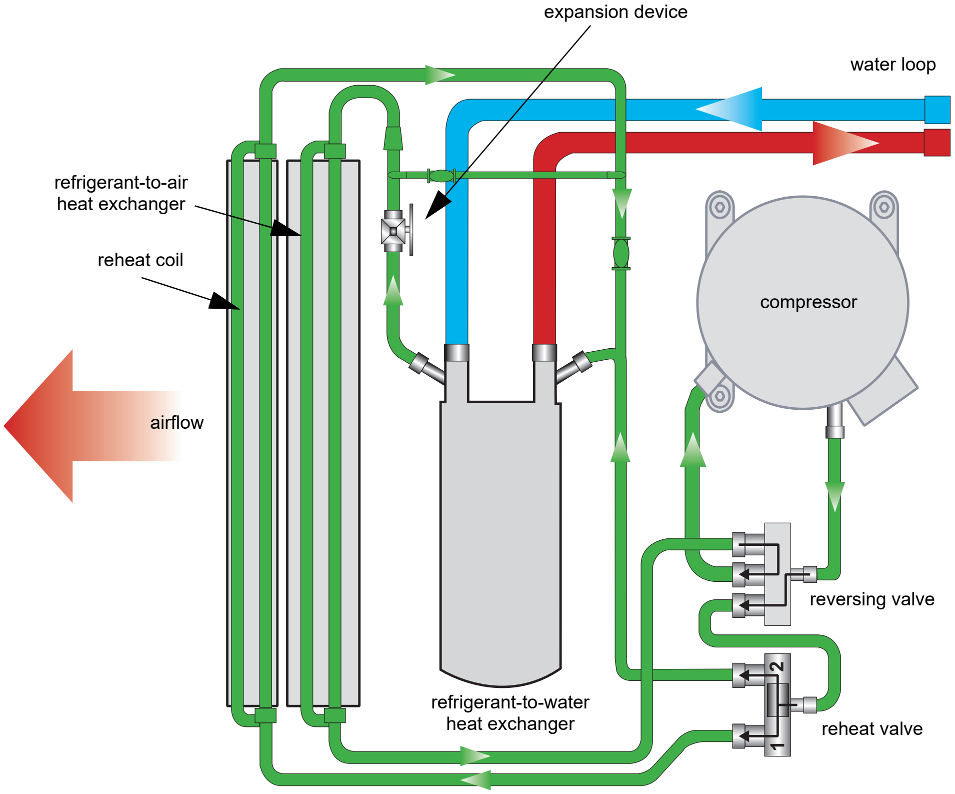
Hot-gas reheat for humidity control

As mentioned in “Methods for improving dehumidification performance,” p. 101, hot-gas reheat can be a cost-effective method of controlling zone humidity.

As long as the zone humidity is less than the desired upper limit—60 percent RH or 60°F (15°C) dew point for example—the WSHP operates in the standard cooling mode and the compressor cycles on and off to maintain zone temperature. When the humidity sensor indicates that zone humidity is too high, the reheat valve diverts hot refrigerant vapor from the compressor through the reheat coil (Figure 112). This allows the compressor to keep operating to dehumidify the air, while warming the supply air to avoid overcooling the zone.

When zone humidity drops back below this upper limit (minus some offset), the reheat valve returns to its original position and the WSHP operates in the standard cooling mode again.

Figure 112. Hot-gas reheat for humidity control



Evaporator freeze protection

When the heat pump is operating in cooling mode, and the air entering the refrigerant-to-air heat exchanger is cooler or drier than normal, the temperature (and pressure) of the refrigerant inside the heat exchanger (operating as the evaporator in the cooling mode) can drop to the point where the coil surface temperature falls below 32°F (0°C). When this occurs, water vapor that condenses out of the air will begin to freeze on the surface of the refrigerant-to-air heat exchanger.

A common approach to prevent this is to attach a temperature sensor to the refrigerant-to-air heat exchanger, and monitor the refrigerant (suction) temperature inside the evaporator. If this temperature drops below a desired limit—30°F (-1°C) for example—a compressor is cycled off, allowing the surface of the heat exchanger to warm back up and avoid condensate from freezing on the surface. When the refrigerant temperature rises back up above the limit (plus a deadband), the compressor is allowed to turn on again.

This condition is most likely to occur in the “integrated economizer” (waterside or airside) mode, if cool outdoor air is brought in directly through the WSHP (unconditioned by a dedicated OA unit), or if the dedicated OA system delivers the conditioned OA at a cold temperature directly to the intake of each WSHP.

Safeties

The unit-level controller for a water-source heat pump typically includes several safeties that protect the equipment from harm. Common examples include:

- Minimum on and off timers to prevent rapid cycling of the compressor(s).
- Cutouts to avoid refrigerant pressures that are too low or too high.
- A condensate overflow float switch that turns off the compressor (and closes the OA damper, if equipped) to prevent the drain pan from overflowing in the event that the condensate drain line is plugged.
- A freeze protection sensor to turn off the compressor if the water leaving the refrigerant-to-water heat exchanger approaches a temperature at which freezing will occur—below 35°F (2°C) for example.

Specific details on safeties should be obtained from the equipment manufacturer.

Water-circulating pump(s)

Before starting the pump, ensure that some isolation valves in the system are open to avoid dead-heading the pump or include a bypass pipe with a pressure-actuated valve in the piping system.

Also, the pump may require a minimum flow rate (or speed) to sufficiently cool the pump motor. A bypass pipe with a pressure-actuated valve can be used to ensure the minimum required flow rate, or the system-level controller can be used to open the valves at several heat pumps when needed to ensure minimum flow.

The simplest approach to controlling the water-circulating pump is to turn on a constant-speed pump whenever the building is expected to be occupied. A simple time clock, or a time-of-day schedule in the building automation system, may be used to turn on the pump at the beginning of the scheduled occupied period and turn it off at the end of the occupied period.

This approach is simple and inexpensive to install because no method of pump capacity control is needed. However, a constant-speed pump consumes a constant amount of pump energy, regardless of building load.

Variable-flow pumping takes advantage of the fact that all the WSHP compressors in the system are not always operating at the same time. For example, when a zone needs neither cooling nor heating the compressor turns off. When the compressor turns off, a two-position isolation valve can be used to shut off water flow to that heat pump, so less total water flow is required in the loop (see "Constant- versus variable-flow pumping," p. 32).

While variable-flow pumping reduces system energy use, it requires some method to modulate pump capacity. The most common method used in a variable-flow system is to vary the speed at which the pump impeller rotates. This is commonly accomplished using a variable-speed drive (or variable-frequency drive, VFD) on the pump motor.

Similar to the supply fan in a variable-air-volume (VAV) system, the pump is often controlled to maintain a set pressure at some location in the piping system. A challenge in a WSHP system is deciding where to locate the differential pressure sensor, since the isolation valves on the individual heat pumps are two-position (open or closed), rather than modulating. Some design engineers locate the differential pressure transducer between the supply and return piping at a location far from the pump (Figure 113). Other design engineers install a small "bleed line" at a location in the piping that is far from the pump, and measure the pressure difference across this bleed line. A controller compares the measured pressure difference to a setpoint, and pump speed is varied to generate enough pressure to maintain the desired pressure difference at the location of the transducer.

In a variable-flow system, consider installing an automatic flow-control device for each heat pump. This device helps ensure proper water flow through the heat pump (when the compressor is operating) as the overall system flow rate and pressure changes (see "Isolation valves and flow-control devices," p. 38).

Figure 113. Pump capacity control in a variable-flow system

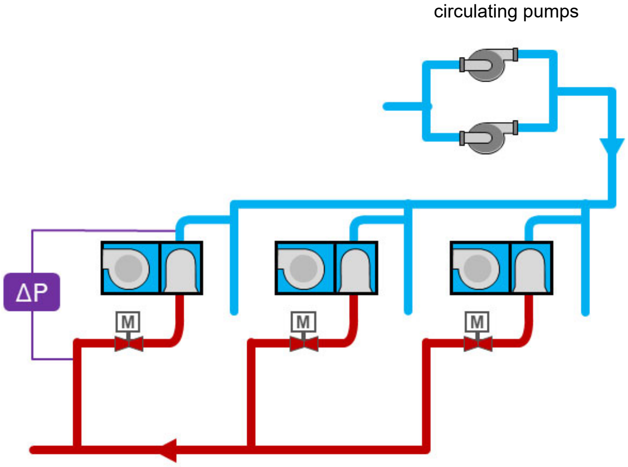
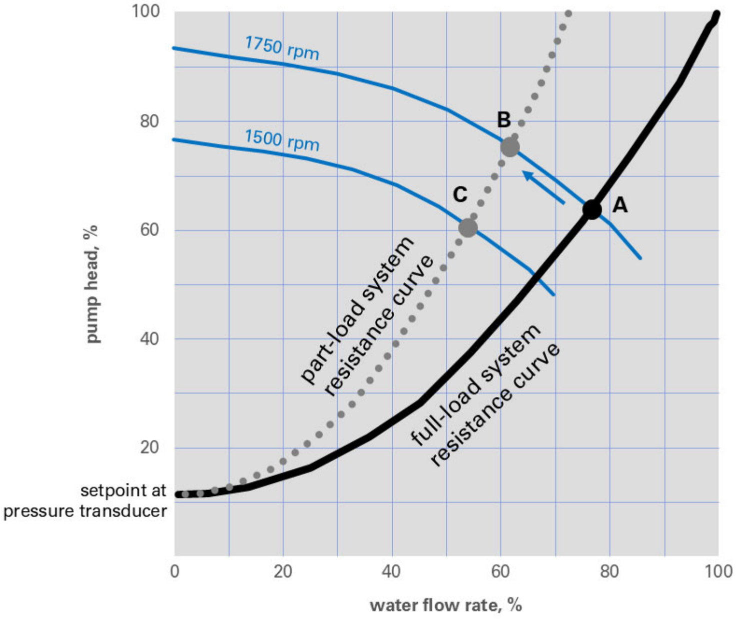


Figure 114 depicts an exaggerated example to illustrate this control loop. When a WSHP compressor turns off, its isolation valve closes to shut off water flow through that heat pump. Closing the valve forces more water to flow through the remaining open valves, increasing the pressure drop (head or resistance) through the system. In response, the pump begins to "ride up" the constant-speed (rpm) performance curve, from the design operating point (A), trying to balance with this new (part-load) system resistance curve (point B).

Figure 114. Variable-flow pump modulation



As a result, the pump delivers less flow at a higher pressure (head). The pressure transducer senses this higher pressure difference, and the controller sends a signal to reduce pump speed. The VFD reduces the speed (rpm) at which the pump impeller rotates, which reduces the system water flow rate until the system balances at a new operating point (C) that brings the pressure difference at the location of the transducer back down to the desired setpoint.

As the VFD reduces the flow rate that the pump produces, it also reduces pump energy use.

A standby pump is often installed to minimize the risk of flow loss in the water loop, in the event that one pump fails. In this case, the controller often rotates operation of the two pumps to equalize runtime and starts. For example, pump 1 would be operated for the week, with pump 2 serving as standby; then the following week, pump 2 would be operated, with pump 1 serving as standby.

Cooling tower

When the loop supply-water temperature reaches the upper setpoint—90ºF (32ºC) for example—the system-level controller activates the cooling tower to reject heat from the water loop.

As mentioned previously, most boiler/tower WSHP systems use either a closed-circuit cooling tower (sometimes called a fluid cooler) or an open cooling tower with an intermediate heat exchanger.

Closed-circuit cooling tower

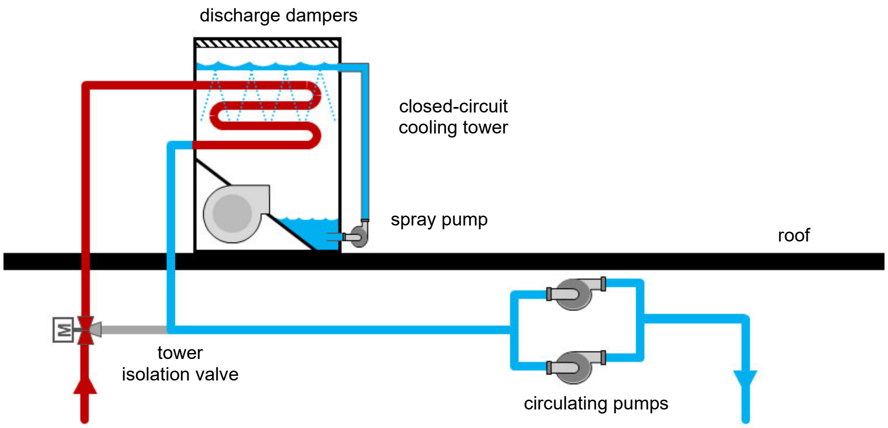
A typical sequence for controlling a closed-circuit cooling tower (Figure 115) is:

Stage 1) Open the discharge dampers and tower isolation valve. If the temperature of the outdoor air is cooler than the water inside the tubes, this allows air to move through the tower via natural convection and a small amount of heat will be rejected from the water loop. No tower fan or circulation pump energy is used, but the amount of heat that can be rejected is limited.

Stage 2) Start the spray pump. This circulates water from the sump to wet the outer surfaces of the tubes. The evaporative cooling effect increases the amount of heat transferred from the water inside the tubes to the air that is flowing across the tubes, still via natural convection. No tower fan energy is used, but the amount of heat than can be rejected is still limited.

Stage 3) Start the cooling tower fan. This forces more air to flow across the tubes, increasing the amount of heat rejected. A two-speed fan, multiple fans, or a fan equipped with a VFD provides additional stages of heat rejection capacity and avoids excessive cycling. Varying airflow allows for closer temperature control, reduces tower fan energy use, and helps prevent freezing during cold weather.

Figure 115. Control of a closed-circuit cooling tower



If a closed-circuit cooling tower is expected to be exposed to ambient temperatures colder than 32°F (0°C), consider operating the tower as a “dry cooler” when the ambient temperature drops below about 45°F (7°C). In this mode, the spray pump is shut off and the fan continues to operate. Heat rejection occurs as the cold outdoor air passes over the dry tubes of the heat exchanger. If a constant-speed fan is used, the reduced airside pressure drop of the dry heat exchanger will result in a higher airflow rate. In this case, contact the tower manufacturer to ensure that the fan motor will not overload due to the reduced pressure drop (and higher airflow).

An alternative approach is to equip the cooling tower with a VFD, and possibly even modulating dampers, to reduce fan airflow. This approach not only provides freeze protection, but results in closer temperature control and less tower fan energy use. In addition, for projects where a small part of the building needs to operate before the remainder of the building is completed, the tower can operate at reduced heat-rejection capacity without excessive cycling of the fan.

Loss-of-flow protection is very important in a system with a closed-circuit cooling tower that is not protected by antifreeze. If the spray pump fails, the water remaining inside the tower sump can freeze rapidly. Therefore, it is essential that the controls respond quickly to protect the tower from freezing in the event of flow loss.

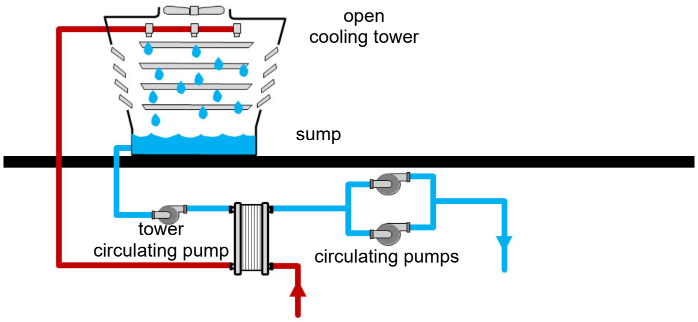
Open cooling tower

Open cooling towers are usually controlled to achieve a desired leaving-water (sump) temperature. A typical sequence for controlling an open cooling tower (Figure 116) is:

Stage 1) Start the tower circulating pump. This circulates water from the sump through the intermediate heat exchanger. If the temperature of the water in the sump is cooler than the water in the closed loop, heat will be rejected from the loop. After passing through the heat exchanger, the sump water is sprayed over the fill inside the tower. Air moves through the tower via natural convection and a small amount of heat will be rejected from the water. No tower fan energy is used, but the amount of heat that can be rejected is limited, typically to about 15 percent of design capacity.

Stage 2) Start the cooling tower fan. This draws more air through the tower fill, increasing the amount of heat rejected. A two-speed fan, multiple fans, or a fan equipped with a VFD provides additional stages of heat rejection capacity and avoids excessive cycling. Varying airflow allows for closer temperature control and reduces tower fan energy use.

Figure 116. Control of an open cooling tower with intermediate heat exchanger



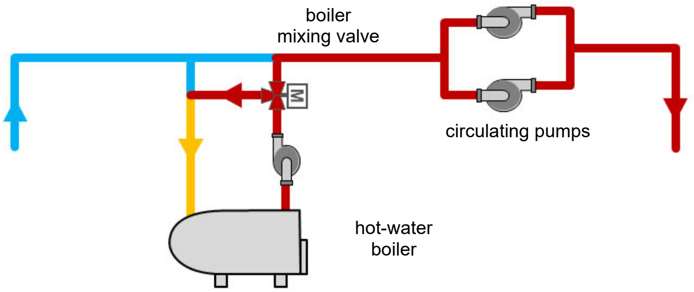
Hot-water boiler

When the loop supply-water temperature reaches the lower setpoint—60ºF (16ºC) for example—the system-level controller activates the boiler to add heat to the water loop.

The hot-water boiler is typically equipped with a dedicated, unit-level controller that varies the heating capacity of the boiler. The controller also monitors boiler operation and protects it from damage by preventing it from operating outside acceptable limits. Specific details about the boiler controller should be obtained from the manufacturer.

In some WSHP systems, the hot-water boiler is decoupled from the main water loop (Figure 117), using a three-way mixing valve to add heat to the water loop by blending in hot water from the boiler. When activated, the boiler controller modulates its capacity to supply hot water at the desired temperature—typically between 140°F (60ºC) and 180°F (82ºC)—and the mixing valve modulates to blend in some of this hot water from the boiler to keep the loop temperature above the lower setpoint.

Figure 117. Boiler control using a three-way mixing valve



This decoupled piping configuration is simple and is suitable for a wide range of packaged boilers. For example, non-condensing boilers require the return-water temperature be no lower than 140ºF (60ºC) to prevent condensing (see “Non-condensing versus condensing boilers,” p. 49). Since the loop water temperature is well below this limit, the decoupled piping configuration diverts hot water leaving the boiler and mixes it with cooler water from the loop, to keep the temperature of the water entering the boiler warm enough to prevent condensing and minimize the risk of “boiler shock.” Sensor location can be critical for proper boiler control. Ensure that the sensor is installed far enough downstream of the boiler so that the hot water leaving the boiler adequately mixes with any bypassed water.

If the system includes a water (thermal) storage tank in the loop, see “Hot-water (thermal) storage,” p. 190, for discussion on how this affects loop temperature control.

In systems that use variable-flow pumping, it may be necessary to ensure constant water flow through the boiler to prevent the leaving-water temperature from getting too hot. This decoupled piping configuration employs a small, constant-speed pump and bypass line to decouple the constant-flow boiler from the variable-flow water distribution loop (Figure 117).

Condensing boilers do not require an elevated return-water temperature (see “Non-condensing versus condensing boilers,” p. 49), so a non-decoupled piping configuration may be possible when they are used.

To avoid large temperature swings as individual steps of boiler capacity energize or de-energize, be sure to provide an adequate number of stages. For an electric boiler, time delays between stages of heating capacity prevent all stages from energizing simultaneously, thereby limiting in-rush current.

Dedicated outdoor-air unit

The most common approach for controlling the dedicated outdoor-air unit is to turn it on when the building is expected to be occupied. The same time-of-day schedule that is used to start and stop the WSHP system is used to start and stop the dedicated OA unit.

For more information on typical operating modes for control for a dedicated OA unit, refer to the Trane application guide, *Dedicated Outdoor Air Systems: Trane DX Outdoor Air Unit* (SYS-APG001*-EN).

The fan in the dedicated OA unit is activated to bring in the required amount of outdoor air for ventilation, and cooling, dehumidification, or heating is modulated to maintain the discharge air at the desired conditions.

The simplest approach operates the supply fan in the dedicated OA unit at a constant speed, to deliver a constant quantity of outdoor air to each zone during the occupied mode, regardless of the level of occupancy. However, if the ventilation ductwork is equipped with VAV terminals to implement occupied standby mode (p. 184) or demand-controlled ventilation (p. 195), the supply fan in the dedicated OA unit is equipped VFD to vary its speed to maintain duct static pressure at setpoint, ensuring each zone receives the necessary airflow (Figure 127).

If the dedicated OA unit is equipped with a relief fan, its capacity might be modulated to maintain a desired static-pressure difference between indoors and outdoors (see “Building pressure control,” p. 192).

System-Level Control

System-level control refers to the intelligent coordination of the individual pieces of equipment so they operate together as a reliable, efficient system. Typically, each water-source heat pump is equipped with a dedicated, unit-level controller that responds to the cooling and heating demands of the zone.

A system-level controller can be used to monitor system operation and to coordinate all these pieces for optimized system control. At a minimum, a system-level controller should be used to operate the water-circulating pumps and dedicated OA unit, coordinate cooling tower and boiler operation to maintain the proper temperature in the water loop, and provide centralized monitoring of system operation.With this configuration, each unit-level controller is capable of performing its functions, even if communication with the system-level controller is lost.

To reduce installation costs, some WSHP systems use simple, residential-style thermostats with no system-level controls. A non-programmable thermostat causes the heat pump to maintain the same temperature, whether the zone is occupied or not.

Use of a programmable thermostat allows a zone to vary the temperature setpoint based on time of day and day of the week. But they also allow occupants to override these setpoints or ignore the schedule altogether (by using the "hold" feature of the thermostat), thus thwarting any potential for energy savings.

A more sustainable approach may be to equip each heat pump with a DDC controller that is connected to a zone temperature sensor, and then use a system-level controller that coordinates the operation of all components of the system. This system-level controller contains a time-of-day schedule that defines when the building is expected to be unoccupied. During these times, the system is shut off and the temperature in each zone is allowed to drift away from the occupied setpoint (often called "night setback").

Allowing the indoor temperature to drift during unoccupied periods saves energy by avoiding the need to operate heating, cooling, and ventilation equipment. Figure 118 shows the potential energy savings of using night setback in an example office building that has a typical boiler/tower WSHP system. Night setback reduced the overall HVAC energy use by 10 to 15 percent for this example building.

Coordination during different operating modes

One of the most important system-level control functions is to coordinate the WSHPs, cooling tower, boiler, and other pieces of equipment during the various modes of operation. The primary system-level control modes in a water-source heat pump system are:

- Occupied mode
- Occupied standby mode
- Unoccupied mode
- Morning warm-up (or cool-down) mode

Typically, a time-of-day schedule in the building automation system is used to define when the system is to operate in these various modes.

Occupied mode

When the building is occupied, the heat pumps maintain the temperature in each occupied zone at the desired setpoint (see Figure 3), and provide the required amount of outdoor air for ventilation. Table 31 describes the typical functions of the different system components during the occupied mode.

Table 31. Coordination of equipment during occupied mode

WSHP	• Activates the fan[1] • Positions the reversing valve and cycles the compressor(s) to maintain zone temperature at the occupied setpoint (cooling or heating)[2]
Dedicated outdoor-air unit	• Activates the fan to bring in the required amount of outdoor air for ventilation • Modulates cooling, dehumidification, or heating to discharge air at the desired conditions • Modulates the central relief fan to maintain indoor-to-outdoor static pressure difference at the desired setpoint
Water distribution loop	• Turns on water-circulating pumps (if a variable-flow system, varies the speed of the pumps to maintain pressure in the piping at the desired setpoint) • Turns on cooling tower if needed to maintain the loop water temperature below the upper setpoint • Turns on boiler if needed to maintain the loop water temperature above the lower setpoint

[1] *Assumes outdoor air is introduced to the occupied zone through the WSHP, and thus requires the fan inside the WSHP to operate to ventilate the zone. If a dedicated OA system delivers outdoor air directly to the zone, the fan inside the WSHP could be configured to cycle off whenever the compressor is off.*

[2] *If the WSHP is equipped for waterside economizing (see "Economizer control," p. 170), the WSHP unit controller may open the waterside economizer valve if the loop water temperature is suitable to provide free cooling. If the WSHP is equipped for airside economizing, the WSHP unit controller may open the outdoor-air damper further if the condition of the outdoor air is suitable to provide free cooling.*

In many buildings, the occupied mode occurs during daytime hours and the unoccupied mode occurs at night. Depending on building usage, however, the occupied mode could extend into the evening.

Occupied standby mode

As mentioned, a time-of-day schedule in the BAS is typically used to define when a zone is to operate in the occupied versus unoccupied mode. In addition, when an occupancy sensor is used in combination with a time-of-day schedule, this sensor can be used to indicate if the zone is actually unoccupied even though the BAS has scheduled it as occupied. This combination can be used to switch the zone to an "occupied standby" mode (see example in Table 32). In this mode, all or some of the lights in that zone can be shut off and the temperature setpoints can be raised or lowered by 1°F to 2°F (0.5°C to 1°C) to reduce energy use.

In addition, if the dedicated outdoor-air system is capable of varying the outdoor airflow delivered to individual zones, the ventilation delivered to that zone can be reduced during this "occupied standby" mode. Section 6.2.6.1.4 of ASHRAE Standard 62.1 allows ventilation to be reduced to zero for some types of spaces during "occupied standby" mode. The example depicted in Table 32 a conference room. For this occupancy category, the standard permits ventilation to be reduced to zero in this mode. For other occupancy categories, ventilation must be no lower than the building-related (or "base") ventilation rate, R_a (see "Minimum ventilation required in breathing zone (Vbz)," p. 90). When the occupancy sensor indicates that the zone is again occupied, the zone is switched back to normal occupied mode.

Table 32. Example of "occupied standby" mode[1]

	"Occupied" mode	**"Occupied standby" mode**
Lights	on	off
Zone cooling setpoint	75°F (24°C)	77°F (25°C)
Outdoor airflow required[2]	310 cfm (153 L/s)	0 cfm (0 L/s)

[1] *Based on a 1000-ft^2 (93-m^2) conference room with a design zone population (P_z) of 50 people.*

[2] *According to Table 6-1 of ASHRAE Standard 62.1-2022, the required outdoor airflow rates for a conference room are: R_p = 5 cfm/p (2.5 L/s/p), R_a = 0.06 cfm/ft^2 (0.3 L/s/m^2). During "occupied" mode: $V_{bz} = R_p \times P_z + R_a \times A_z$ = 5 cfm/p x 50 people + 0.06 cfm/ft^2 x 1000 ft^2 = 310 cfm (2.5 L/s/p x 50 people + 0.3 L/s/m^2 x 93 m^2 = 153 L/s). During "occupied standby" mode, for this occupancy category, V_{bz} is permitted to reduce to zero.*

Unoccupied mode

When the building is unoccupied, the BAS can allow the temperature in the zones to drift away (cooler or warmer) from the occupied setpoints (see Figure 4). But the system must still prevent the zones from getting too cold—perhaps 60°F (16°C)—or too hot—perhaps 90°F (32°C). In addition, when unoccupied, the building does not typically require outdoor air for ventilation or to replace exhaust air, so the dedicated outdoor-air system can be turned off (or for a rooftop-style WSHP, the outdoor-air damper can be closed).

Allowing the indoor temperature to drift during the unoccupied mode, often called "night setback," typically saves energy by avoiding the need to operate heating, cooling, and ventilation equipment (Figure 118).

Figure 118. Energy-saving potential of night setback

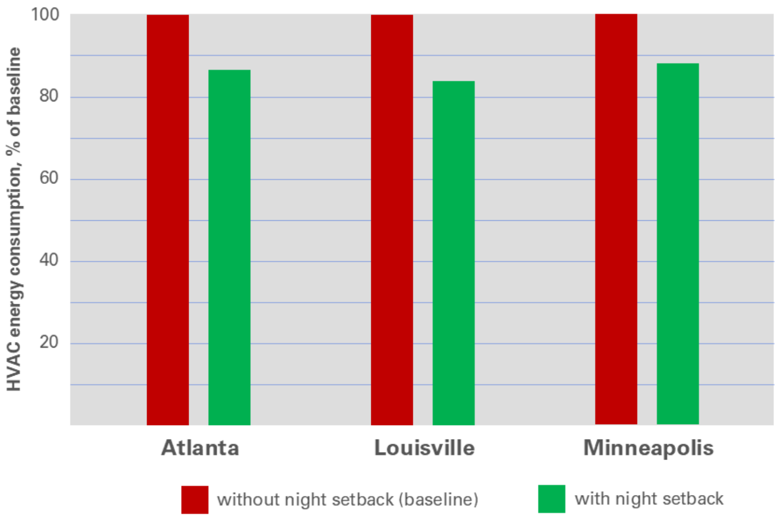


Table 33 describes the typical functions of the different system components during the unoccupied mode.

Table 33. Coordination of equipment during unoccupied mode

WSHP	• Fan is turned off, unless the zone requires cooling or heating • Positions the reversing valve and cycles the compressor(s) to bring the zone temperature to the unoccupied setpoint (cooling or heating)
Dedicated outdoor-air unit	• Fan is turned off • Central relief fan is turned off
Water distribution loop	• Water-circulating pumps are turned off, unless any zone requires cooling or heating (if a variable-flow system, varies the speed of the pumps to maintain pressure in the piping at the desired setpoint) • Turns on cooling tower if needed to maintain the loop water temperature below the upper setpoint • Turns on boiler if needed to maintain the loop water temperature above the lower setpoint

Figure 119. Zone temperature sensor with timed override button

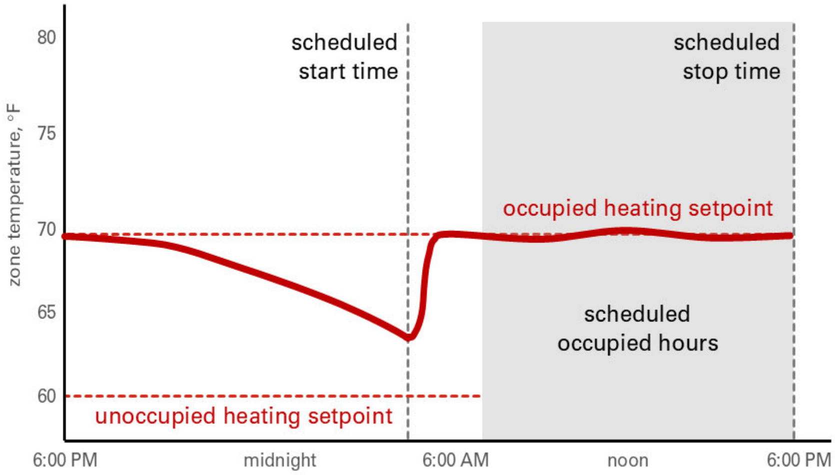
Some systems incorporate a **timed override** feature, which allows the occupant to switch the system into the occupied mode during hours when it is scheduled to be unoccupied. The most common means for enabling this function is a timed override button located on the zone sensor (Figure 119). Typically, pressing this button directs the system to operate in the occupied mode for only a fixed period of time (two hours, for example). After this time period expires, the BAS automatically returns the zone to unoccupied mode.

Finally, in some cases, it may be important to control humidity (in addition to temperature) when the building is unoccupied, to avoid damage to the building structure and furnishings. The BAS can monitor indoor humidity levels and take action if the humidity rises above a maximum limit (see "After-hours dehumidification," p. 108) or drops below a minimum limit.

Morning warm-up (or cool-down) mode

As mentioned previously, the temperature inside a building is typically allowed to drift when unoccupied, usually for the purpose of saving energy. This generally requires the HVAC system to start prior to occupancy, and operate long enough for the temperature inside the building to reach the desired occupied setpoint by the time people are expected to occupy the building (Figure 120). When the building must be heated prior to occupancy, this is called “morning warm-up.” When the building must be cooled, it is called “morning cool-down.”

Figure 120. Morning warm-up



The morning warm-up/cool-down mode typically occurs as a transition from the unoccupied mode to the occupied mode. The system attempts to return the temperature inside the building to the occupied setpoint as rapidly as possible. In this mode, the building does not typically require ventilation because it is not yet occupied. Table 34 describes the typical functions of the different system components during the morning warm-up or morning cool-down modes.

Table 34. Coordination of equipment during morning warm-up or cool-down mode

WSHP	• Activates the fan until zone temperature reaches the occupied heating (for warm-up) or occupied cooling (for cool-down) setpoint, then the fan is turned off • Positions the reversing valve and cycles the compressor(s) to bring the zone temperature to the occupied setpoint (cooling or heating)
Dedicated outdoor-air unit	• Fan is turned off[1] • Central relief fan is turned off[1]
Water distribution loop	• Turns on water-circulating pumps (if a variable-flow system, varies the speed of the pumps to maintain pressure in the piping at the desired setpoint) • Turns on cooling tower if needed to maintain the loop water temperature below the upper setpoint • Turns on boiler if needed to maintain the loop water temperature above the lower setpoint

[1] *In some buildings, outdoor air may be brought into the building during the morning warm-up or cool-down mode to dilute contaminants that have accumulated inside the building during the unoccupied mode. This is often called a* ***preoccupancy purge****. In this case, the fan in the dedicated outdoor-air unit is activated; cooling, dehumidification, or heating is modulated to discharge air at the desired conditions; and the central relief fan should modulate to maintain indoor-to-outdoor pressure difference at the desired setpoint.*

Rather than simultaneously turning on all the heat pumps in the morning to warm up or cool down their respective zones, a system-level controller can be used to stagger the starting of the individual heat pumps to avoid a “spike” in the building electrical demand. Of course, this requires a longer period of time to bring all zones to desired occupied conditions.

Scheduling

Determining the times at which to start and stop the HVAC system is typically based on assumptions regarding building usage. Most building managers or operators want to avoid complaints from the occupants and the time needed to respond to those complaints. For this reason, they usually take a very conservative approach, starting the system very early and stopping it very late. This can be costly from an energy perspective, since the entire building may be operating to maintain occupied temperature setpoints, even though only a few spaces are occupied.

Following are a few ways to minimize comfort complaints and avoid wasting energy:

- *Use “aggressive” scheduling and equip zone temperature sensors with timed override buttons.* If a person wants to use a space during a time when it has been scheduled as unoccupied, they simply press the timed override button (see Figure 119) and the BAS switches that zone into the occupied mode. This returns the temperature to the occupied setpoint and delivers ventilation air to that zone. Typically, the BAS automatically returns this zone to the unoccupied mode after a defined fixed period of time (two hours, for example).

  Using the timed override feature affords the opportunity to be more aggressive with time-of-day operating schedules. This avoids wasting energy by starting and stopping the HVAC system based on typical usage, and allowing the timed override feature to handle the worst-case or once-a-year scenarios. Once occupants are educated about using the timed override feature, energy savings and minimal complaints can coexist.

- *Use separate time-of-day schedules for areas with differing usage patterns.* For simplicity, many building managers or operators define only one time-of-day schedule to operate the entire building. However, if areas of the building have significantly different usage patterns, this approach wastes energy since the entire building may be operating to maintain occupied temperature setpoints, even though only part of the building is in use.

  A more energy-efficient approach is to create separate time-of-day operating schedules for areas of the building with significantly different usage patterns. If the facility already has a building automation system (BAS), it probably includes a time-of-day scheduling function, so the only additional cost is the operator's time to set up the schedules.

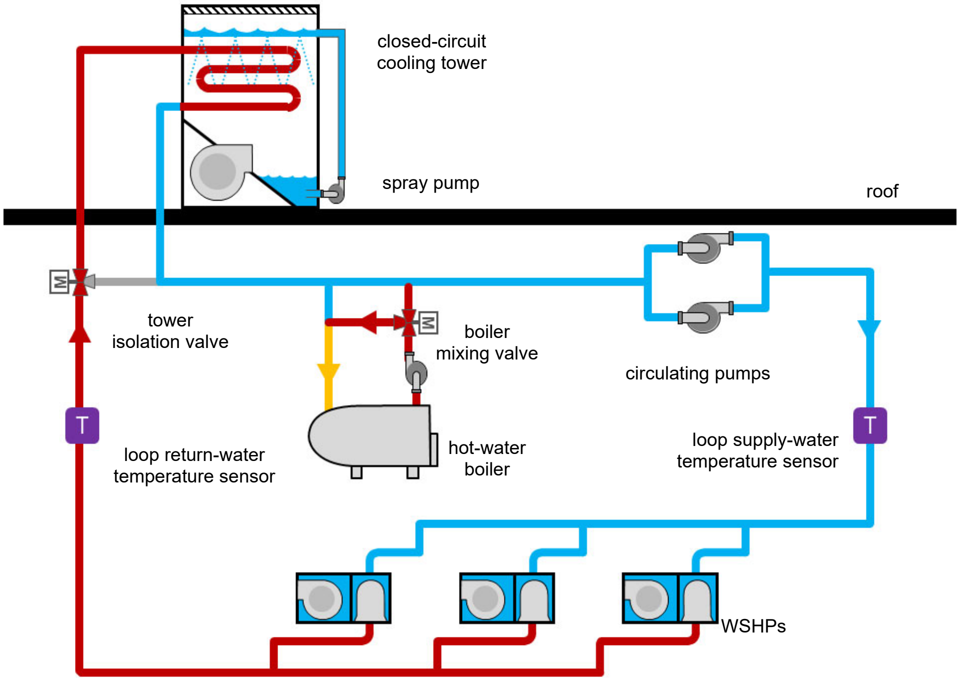
  To reduce the number of schedules that need to be created and maintained, group zones with similar usage patterns together and create one schedule for each group.

Water loop temperature control

Water-source heat pumps can operate in either heating or cooling mode when the water loop temperature is maintained within the recommended range—between about 60ºF (16ºC) and 90ºF (32ºC) for example. Loop temperatures outside the recommended range can severely impact WSHP performance. For this reason, one of the primary functions of the system-level controller is to monitor and maintain an acceptable temperature in the water loop.

The loop supply-water temperature sensor is typically located slightly downstream of the water-circulating pumps (Figure 121). Additionally, a loop return-water temperature sensor is often located downstream of the heat pumps, but upstream of the cooling tower.

Figure 121. Water loop temperature control



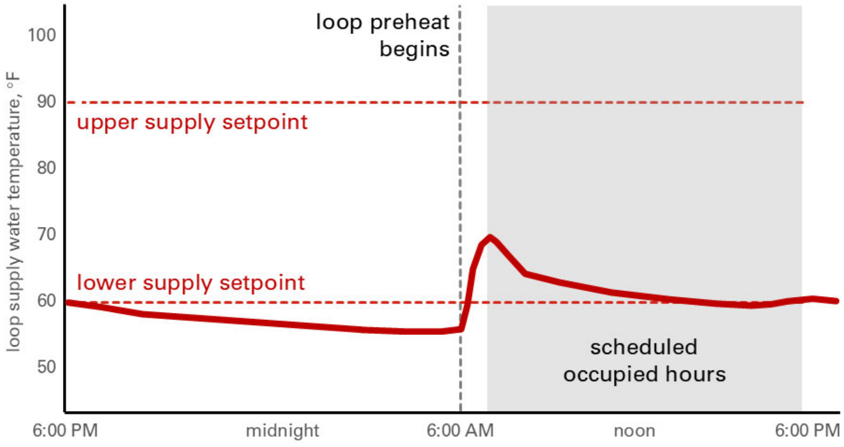
When the loop supply-water temperature reaches the upper setpoint—90ºF (32ºC) for example—the system-level controller activates the cooling tower to reject heat from the water loop (see "Cooling tower," p. 178, for discussion of the various methods of cooling tower control). When the loop supply-water temperature reaches the lower setpoint—60ºF (16ºC) for example—the system-level controller activates the boiler to add heat to the water loop (see "Hot-water boiler," p. 180, for discussion of the various methods of boiler control). Between these two setpoints, both the cooling tower and boiler remain off.

Contact the equipment manufacturer to confirm the acceptable operating limits of the specific heat pump equipment.

Preheat for morning warm-up

On a Monday morning following a cold weekend, the temperature in many zones of the building may have drifted such that a large number of the WSHPs will need to operate in morning warm-up mode prior to occupancy. To avoid overwhelming the boiler, the loop water can be preheated above the lower setpoint before the system enters the occupied mode (Figure 122).

Figure 122. Loop preheat



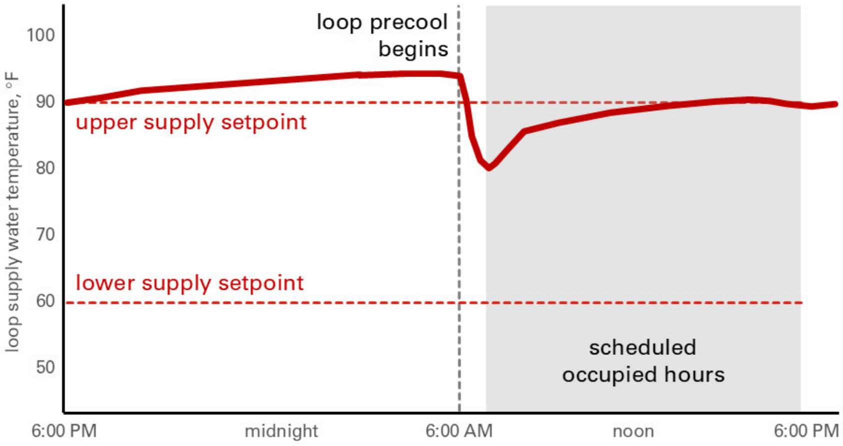
To avoid the unnecessary use of energy during mild weather, the system-level controller can be used to disable this loop preheat sequence if the outdoor temperature is warm—above 40°F (4°C) for example.

Precool loop for morning cool-down

During the hot summer months, outdoor temperatures in the early morning hours may be cool enough that the cooling tower might be activated to precool the loop water below the upper setpoint, before the system enters the occupied mode (Figure 123).

This can reduce building energy use by operating the cooling tower during cooler, drier conditions in the morning, rather than during the hot, humid hours of the day. This can also help avoid overwhelming the cooling tower when a large number of the WSHPs need to operate in morning cool-down mode prior to occupancy.

Figure 123. Loop precool



Hot-water (thermal) storage

As described earlier (see "Hot-water (thermal) storage," p. 53), some systems include a water storage tank in the loop. Control of the blending valve associated with this tank depends on the project-specific goals.

For a **high-temperature storage tank**, the tank's heat source (typically a electric heating element, a gas burner, or solar) is activated to raise the water temperature inside the tank to a desired setpoint—180°F (82°C) for example. During morning warm-up mode, the tank is then used as the first stage of heat addition. The blending valve is modulated to mix hot water from the tank into the loop, maintaining the loop temperature at the lower supply setpoint. When the tank is no longer able to supply hot water, the boiler is activated as the second stage of heat addition.

For systems that use a electric boiler, this same control sequence might also be used during normal occupied mode to avoid using electricity during on-peak utility periods.

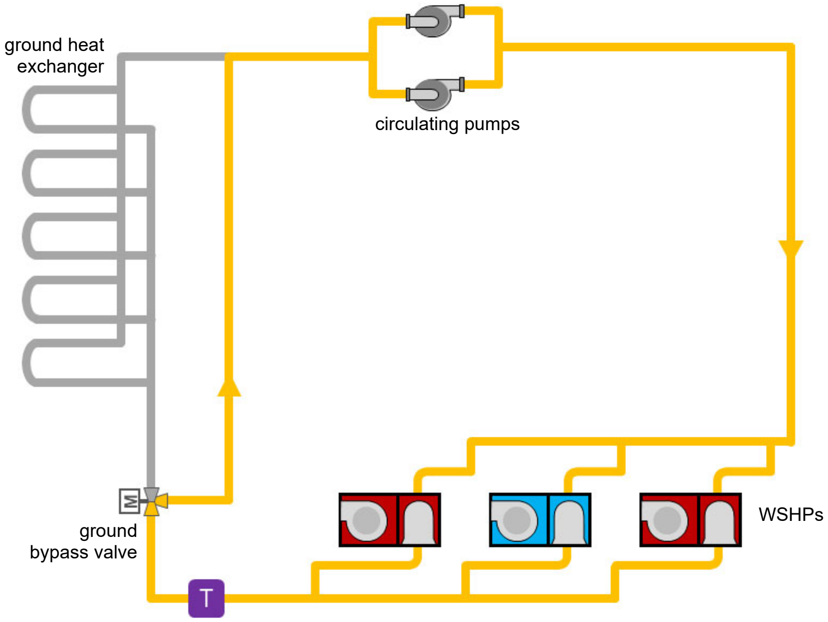
For a **low-temperature storage tank**, the blending valve is normally positioned to flow loop water into the tank. Once the water temperature inside the tank reaches a desired setpoint—100°F (38°C) for example—this valve is positioned such that loop water bypasses the tank. During morning warm-up mode (and maybe even if heat is needed during unoccupied mode), the tank is then used as the first stage of heat addition. The blending valve is modulated to mix warm water from the tank into the loop, maintaining the loop temperature at the lower supply setpoint. When the tank is no longer able to supply hot water, the boiler is activated as the second stage of heat addition.

As described in "Hot-water (thermal) storage," p. 53, this low-temperature storage tank can also be used for other purposes, which would affect the loop controls.

Bypassing the ground heat exchanger

In some ground-source heat pump systems, a bypass valve and pipe are included to avoid pumping water through the ground heat exchanger whenever the temperature of the loop is within the desired range (Figure 124).

Figure 124. Bypass of the ground heat exchanger



During mild weather, heat rejected to the loop by heat pumps serving zones that require cooling may be nearly equal to the heat extracted from the loop by heat pumps serving zones that require heating. In such a case, the loop temperature may remain within a reasonable temperature range, requiring no heat rejection to, or heat extraction from, the ground.

When the fluid temperature upstream of the ground heat exchanger is within a reasonable range, the valve diverts flow to bypass the ground heat exchanger. This lowers the pressure drop that the pump must overcome and reduces pump energy use. It also avoids unwanted heat transfer to (or from) the ground when some zones requires cooling while other zones simultaneously need heating.

While some design engineers include this bypass pipe, others suggest that it provides little benefit. The proponents of no bypass pipe claim the following:

- Such a balanced load condition occurs infrequently, and when it does occur, the system flow rate will likely be relatively low (assuming variable-flow pumping is used), so any pump energy savings is minimal.
- Installing abypass valve adds a risk that it will be controlled improperly in the future.
- Keeping fluid flowing through the ground heat exchanger helps “even out” the ground temperature throughout the borefield, improving performance over the long term.

Safeties

The system-level controller can also include several safeties to protect the equipment from harm. Examples include:

- Sending a signal to disable all WSHP compressors if the water-circulating pump fails, resulting in the loss of water flow. Of course, the controller should also attempt to start the standby (or back-up) pump and then automatically enable all of the WSHP compressors after flow resumes.
- Sending a signal to disable all WSHP compressors if the water loop temperature gets too hot or too cold. This might occur if a cooling tower fan belt were to break, for example. The controller should automatically enable all of the WSHP compressors after loop water temperature returns to normal.
- Install a freeze protection sensor that will turn on the water-circulating pump (or take some other action) if the water in the loop approaches a temperature at which freezing might begin to occur.

Building pressure control

In addition to providing conditioned outdoor air for ventilation, the dedicated outdoor-air system is also used to replace air that is exhausted locally from certain areas of the building (such as restrooms, kitchens, and lab spaces) and control the indoor-to-outdoor pressure difference.

During humid weather, maintaining the pressure inside the building so that it is slightly higher than the pressure outside (“positive” pressure) may improve comfort and helps prevent humid outdoor air from leaking into the building envelope. During cold weather, the pressure inside the building should be equal to (or even slightly less than) the pressure outside. This helps avoid forcing moist indoor air into the building envelope, and helps minimize uncomfortable cold drafts due to infiltration. In either case, excessive building pressure, whether negative or positive, should be avoided.

Because many WSHP systems with a dedicated OA system bring in a constant quantity of outdoor air during occupied periods and do not use an airside economizer cycle, maintaining proper building pressurization is typically thought of as an air balancing issue. Even in a properly balanced system, however, the wind, variable operation of local exhaust fans, and “stack effect” can result in pressure fluctuations.

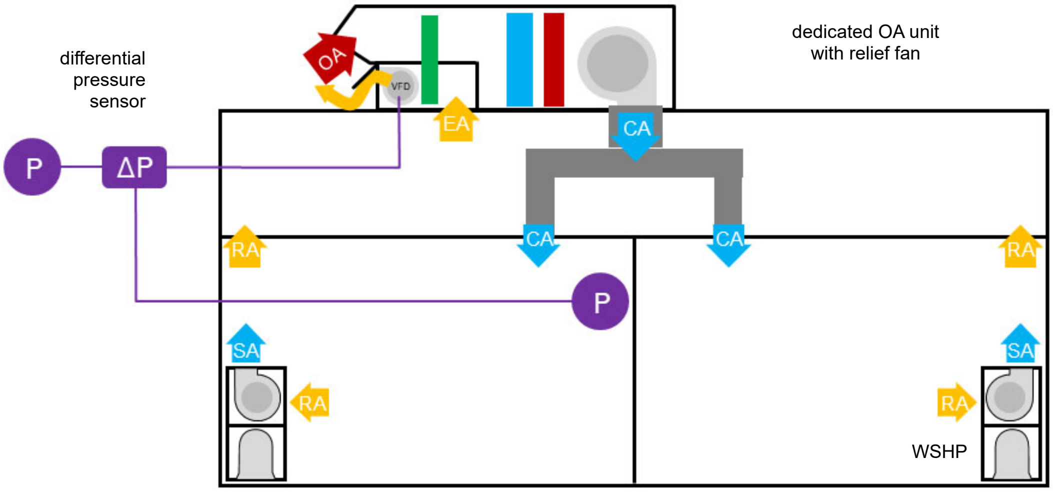
If occupied standby mode, demand-controlled ventilation, or an airside economizer is used, the intake airflow varies during occupied periods. This will also require varying relief (exhaust) airflow to avoid over-pressurization or depressurization.

In most applications, the dedicated OA system is turned off during unoccupied periods. In some cases, however, local exhaust fans are allowed to operate, either by design or as an oversight. Because air is still being exhausted from the building, but no air is being brought in by the ventilation system, a negative pressure is created in the building with respect to the outdoors. One solution may be to use the building automation system to turn off all local exhaust fans whenever the dedicated OA system is not operating. This may require a manual override to allow for after-hours cleaning processes. If some exhaust fans are required to operate at all times, then a solution may be to operate the dedicated OA system at a reduced airflow during unoccupied periods to maintain building pressure 24/7.

This indoor-to-outdoor pressure difference can be controlled by adjusting either the quantity of air brought into, or exhausted from, the building. In most WSHP systems, the quantity of outdoor air brought into the building is based on the minimum ventilation requirements of a local building code, so controlling building pressure typically involves varying the quantity of air exhausted from the building.

When a dedicated OA system is used, this might involve controlling the capacity of a central relief fan to maintain a desired static-pressure difference between indoors and outdoors (Figure 125). A differential pressure sensor monitors the indoor-to-outdoor pressure difference. Its signal is used to adjust relief airflow, directly controlling building pressure. Capacity control can be accomplished by either a) modulating the relief damper and allowing the relief fan to “ride the fan curve” or b) equipping the relief fan with a variable-speed drive.

Figure 125. Direct control of building pressure using a central relief fan



Direct control of building pressure requires a differential pressure sensor to monitor the indoor-to-outdoor pressure difference. A common approach is to use an electronic transducer to convert the pressure difference into an electrical signal, which is then sent to the controller of the central relief fan. Two sensing tubes (one measuring indoor pressure and the other measuring outdoor pressure) are attached to this transducer. Proper location of these pressure sensing tubes is important:

- The **indoor sensor** is typically located on the ground floor, because the effects of over- or under-pressurization are most noticeable at the external doors. Many design engineers locate the sensor in a large open space near the door, while others isolate the indoor sensor from the door (in a central hallway, for example) to dampen the effect of rapid pressure changes caused by door operation.

  In either location, the indoor pressure sensor should include sufficient signal filtering to minimize the effects of high-speed pressure changes. It is also important to avoid perimeter locations that can be influenced by wind-induced pressure fluctuations.

- Many design engineers place the **outdoor sensor** on the roof of the building. Others use multiple sensors—one at each corner of the building, at least 15 ft (4.6 m) above the roof—and average their signals. In any case, select sensors that will minimize wind effect and keep water out of the sensing tube.

System optimization

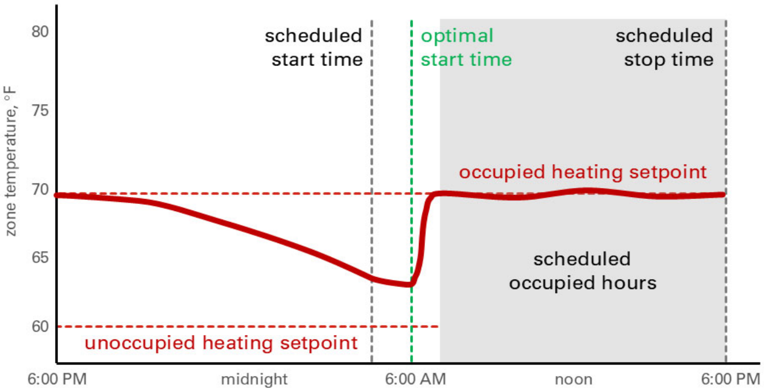
When there is a building automation system (BAS) that provides system-level coordination of the various pieces of equipment, the next logical step is to optimize the control of that system. For this discussion, optimization is defined as minimizing the cost to operate the entire HVAC system, while still maintaining acceptable comfort. In other words, this means maximizing the efficiency of the entire system, not just an individual component.

Optimal start

The morning warm-up (or cool-down) mode was discussed previously in this chapter. In some buildings, a simple time clock or a time-of-day schedule is used to start and stop the HVAC system. In this case, the time at which the morning warm-up (or cool-down) mode begins is typically set to ensure that the indoor temperature reaches the desired occupied setpoint prior to occupancy on the coldest or warmest morning of the year. In other words, the system is programmed to start early enough so the building will warm up or cool down fast enough on the worst-case morning. As a result, for most days, the system starts earlier than it needs to. This increases the number of operating hours and increases energy use.

An alternative approach is a strategy called “optimal start.” The system-level controller is used to determine the length of time required to bring each zone from its current temperature to its occupied setpoint temperature. Then the controller waits as long as possible before starting the system, so that the temperature in each zone reaches its occupied setpoint just in time for scheduled occupancy (Figure 126).

Figure 126. Optimal start



The optimal starting time is determined using the difference between the actual zone temperature and the occupied setpoint temperature (heating or cooling). It compares this difference with the historical performance of how quickly the zone has been able to warm up or cool down. Some systems also compensate for the current outside temperature.

This strategy reduces the number of system operating hours and saves energy by avoiding the need to maintain the indoor temperature at *occupied* setpoint, even though the building is *unoccupied*. This may require many of the heat pumps to operate at full capacity simultaneously, which could impact the size of the cooling tower and/or boiler, and may impact the electrical demand charge from the utility.

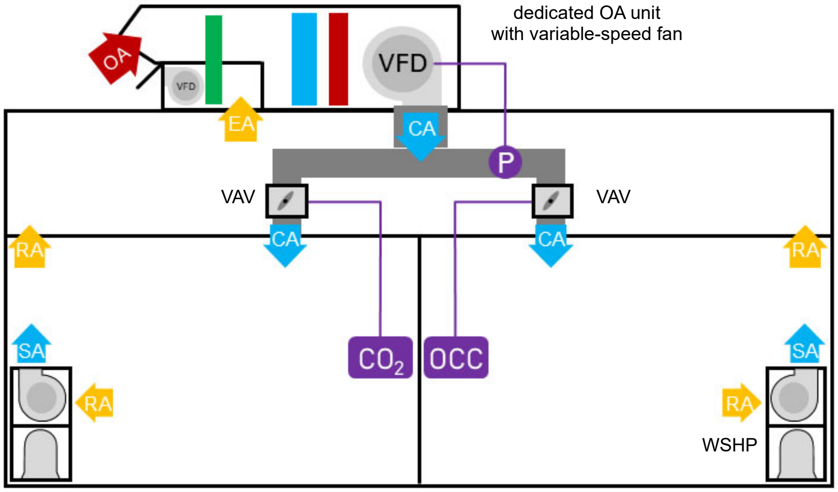
Demand-controlled ventilation

For densely-occupied zones, demand-controlled ventilation may be required by ASHRAE Standard 90.1 (see "Demand-controlled ventilation," p. 119).

As mentioned in "Dynamic reset of intake airflow," p. 96, ASHRAE Standard 62.1 permits dynamic reset of intake (outdoor) airflow as operating conditions change, as long as the system provides at least the required breathing-zone outdoor airflow (V_{bz}) whenever a zone is occupied. As the number of people occupying a zone varies, the quantity of outdoor air required to properly ventilate that zone also varies.

Demand-controlled ventilation (DCV) is a strategy that attempts to dynamically reset the outdoor airflow delivered to a zone based on changing population within that zone. By installing a VAV terminal unit for each zone, and equipping the fan in the dedicated OA unit with a VFD to vary airflow, a dedicated OA system can implement DCV (Figure 127). A carbon dioxide (CO_2) sensor, an occupancy (OCC) sensor, or even a time-of-day schedule, can be used with the pressure-independent VAV terminal to reset the outdoor airflow delivered to a given zone. The VFD in the dedicated OA unit varies airflow to maintain duct static pressure at setpoint, ensuring each zone receives the airflow desired.

Figure 127. Demand-controlled ventilation with a dedicated OA system



- *Time-of-day schedules*

  A time-of-day schedule can be created in the building automation system (BAS) to indicate when each zone is occupied versus unoccupied. For any hour that a zone is scheduled to be unoccupied (even though other zones served by the dedicated OA system are scheduled to be occupied), outdoor airflow for that zone is reduced to zero (or to the building-related ventilation rate, R_a; see "Zone-level ventilation requirements," p. 90). This approach may be well-suited for many classrooms, where occupancy is predictable and the number of occupants does not vary greatly.

Alternatively, a time-of-day schedule can be used to estimate the actual number of people in a zone for any given hour. This variation in population is then communicated to the unit-level controller on a VAV terminal unit and used to reset the outdoor airflow delivered to the zone for that hour. This approach may be well suited for densely-occupied zones that have a predictable occupancy pattern, such as a cafeteria.

- *Occupancy sensors*

  An occupancy sensor, such as a motion detector, can be used to detect the presence of people in a zone, and send a binary signal to the unit-level controller on a VAV terminal unit. When the sensor indicates the zone is occupied, the VAV terminal unit modulates to deliver the design outdoor airflow for that zone. When the sensor indicates the zone is presently not occupied, the damper reduces or shuts off outdoor airflow to that zone.

  If an occupancy sensor is used in combination with a time-of-day schedule, the building may be scheduled as “occupied” while the sensor indicates the zone is unoccupied (Table 35). In this “occupied standby” mode (p. 184), the controller resets the outdoor airflow rate to zero or to a less-than-design (“base”) outdoor airflow rate (typically the building component of the ventilation rate, Ra × Az).

Table 35. Combining occupancy sensors with time-of-day schedule

Time-of-day schedule reads	Occupancy sensor indicates	Operating mode	Ventilation setpoint
occupied	occupied	occupied	design outdoor airflow
occupied	unoccupied	occupied standby	zero or “base” outdoor airflow
unoccupied	n/a	unoccupied	no outdoor airflow

  Occupancy sensors are relatively inexpensive, do not need to be calibrated, and are already used in many zones to control the lights. Zones that are less-densely occupied or have a population that varies only minimally—such as private offices, many open plan office spaces, and many classrooms—are good candidates for occupancy sensing.

- *Carbon dioxide (CO_2) sensors*

  A sensor is used to monitor the concentration of CO_2 in the zone, which is being continuously produced by the occupants. The difference between the CO_2 concentration in the zone and the outdoor CO_2 concentration can be used as an indicator of the per-person ventilation rate (cfm/person [m^3/s/person]).

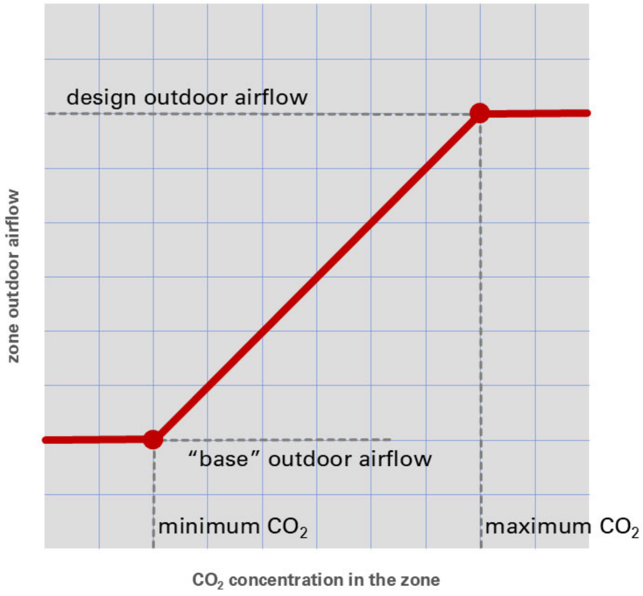
  Note: *In most locations, the concentration of CO_2 outdoors remains relatively constant. Because of this and in lieu of installing an outdoor CO_2 sensor, most designers use either a one-time reading of the outdoor CO_2 concentration at the building site or a conservative value from historical readings. This simplifies control, lowers the installed cost, and often increases accuracy because it avoids the potential inaccuracy of an outdoor sensor.*

  The measured concentration of CO_2 in the zone is then communicated to the unit-level controller on a VAV terminal unit and used to reset the outdoor airflow currently delivered to that zone (Figure 128). If the CO_2 concentration in the zone is less than or equal to the minimum CO_2 limit, the damper modulates to deliver

For more information on CO_2-based demand-controlled ventilation, refer to the Trane *Engineers Newsletter*, titled “CO_2-Based Demand-Controlled Ventilation with ASHRAE Standard 62.1” (ADM-APN017-EN), and the Trane *Engineers Newsletter Live* video, titled “Demand-Controlled Ventilation” (APP-CMC067-EN).

a less-than-design (base) outdoor airflow (typically the building component of the ventilation rate, Ra × Az). On the other hand, if the CO_2 concentration is greater than or equal to the maximum CO_2 limit, the damper modulates to deliver the design outdoor airflow. If the CO_2 concentration is between the minimum and maximum CO_2 limits, the outdoor airflow is adjusted proportionally between the "base" and design airflows.

Figure 128. Varying zone outdoor airflow based on CO_2 concentration



CO_2-based DCV requires a CO_2 sensor in each zone where it is used, which requires periodic calibration and cleaning to ensure proper operation. Zones that are densely occupied and experience widely varying population—such as conference rooms, auditoriums, and gymnasiums— are typically good candidates for CO_2 sensors.

CO_2 sensors increase the installed cost of the system, but they also increase risk. These sensors need to be maintained and calibrated (or periodically replaced) to maintain accuracy. If the sensor goes out of calibration and signals that the CO_2 concentration in the zone is lower than it actually is, the system will reduce ventilation to (under-ventilate) that zone, degrading indoor air quality. On the other hand, if the sensor signals that the CO_2 concentration is higher than it actually is, the system will increase ventilation to (over-ventilate) that zone, wasting energy.

Therefore, CO_2 sensors should not be used indiscriminately. Rather, they should be installed only in those zones where they provide the best return on investment and are worth the risk.

In most cases, the best value is achieved by **combining all three DCV approaches**, using each where it best fits. Those zones that are densely occupied and experience widely varying population—such as conference rooms, auditoriums, and gymnasiums—are typically good candidates for CO_2 sensors. However, zones that are less-densely occupied or have a population that varies only minimally—such as private offices, many open plan office spaces, and many classrooms—are probably better suited for occupancy sensors and/or time-of-day schedules. Zones with predictable occupancy patterns—such as cafeterias—are good candidates for time-of-day schedules.

Combining these various zone-level DCV strategies can ensure that each zone is properly ventilated *without* requiring a CO_2 sensor in every zone. CO_2 sensors are used only in those zones where they will bring the most benefit. This minimizes installed cost and avoids the periodic calibration and cleaning required to ensure proper sensor operation. For the other zones, occupancy sensors and/or time-of-day schedules are used to reduce ventilation.

Reset dedicated OA unit leaving-air temperature

Many dedicated outdoor-air systems are designed to dehumidify the outdoor air to a dew point that is drier than the zone, and then reheat it to dry-bulb temperature that is close to the zone setpoint (neutral). This control approach is simple because it allows the dedicated OA unit to operate independently of the local heat pumps.

However, allowing the dedicated OA unit to deliver the conditioned outdoor air at a cold (rather than neutral) temperature can reduce both the installed cost and energy use of the overall system (see "Neutral- versus cold-air delivery," p. 64). Why reheat the dehumidified outdoor air to a neutral temperature on the hottest day of the summer when all zones need cooling?

However, there are times during the year when it may be more efficient to reheat the dehumidified outdoor air to avoid overcooling the zones. Following are some possible approaches for resetting the dry-bulb temperature of the conditioned air delivered by the dedicated OA unit. (The dew point of the conditioned air is controlled independently to meet the humidity control requirements of the zones.)

- **Activate the reheat coil when it gets cold outside.**

  A very simple control approach could be to activate the reheat coil—reheating the dehumidified outdoor air to a neutral dry-bulb temperature—whenever the outdoor temperature drops to the point where the sensible cooling loads in some of the zones are expected to be low enough that the cold outdoor air may cause overcooling. For example, whenever the outdoor temperature drops below 65°F (18°C)—but this limit could be adjusted after a few months of experience in operating the system.

- **Monitor the zone temperatures and modulate reheat capacity in the dedicated OA unit to avoid overcooling.**

  An even more effective way to implement this strategy is to use a building automation system (BAS) to monitor the current zone temperature for all the heat pumps to determine the zone where the temperature is closest to its heating setpoint. This is the zone that is most at risk of overcooling. Based on a signal from the BAS, the dedicated OA unit could then modulate its reheat capacity, resetting the leaving-air dry-bulb temperature setpoint upward just enough to prevent overcooling any of the zones.

  This strategy delivers the conditioned OA at a temperature that offsets as much of the zone sensible cooling loads as possible, without overcooling any zone, avoiding the need for any heat pumps to operate in the heating mode. Of course, this is only a benefit if the dedicated OA unit uses recovered energy for reheat. If it uses "new" energy for reheat, it would likely be more efficient to allow the heat pumps to operate in the heating mode.

- **Monitor the loop water temperature and modulate reheat capacity to avoid activating the boiler.**

  If the cold, conditioned outdoor air causes only a few heat pumps to operate in the heating mode, they will extract heat from the water loop, reducing the amount of heat that must be rejected by the cooling tower. This likely improves system efficiency, rather than degrading it. In addition, the remaining zones where the heat pumps are operating in the cooling mode continue to benefit from the sensible cooling provided by the cold, conditioned outdoor air.

  However, if enough heat pumps are operating in the heating mode that the temperature of the water loop approaches the lower setpoint—60°F (16°C) for example—and the source of reheat energy in the dedicated OA unit is recovered from another part of the system (hot gas reheat or an air-to-air heat exchanger, for example), it will likely be more efficient to reheat the dehumidified outdoor air to avoid the need to activate the hot-water boiler.

For an application in which very few zones require cooling during the colder months of the year, it may be desirable to heat the outdoor air to a temperature near the desired zone temperature before delivering it directly to the zones.

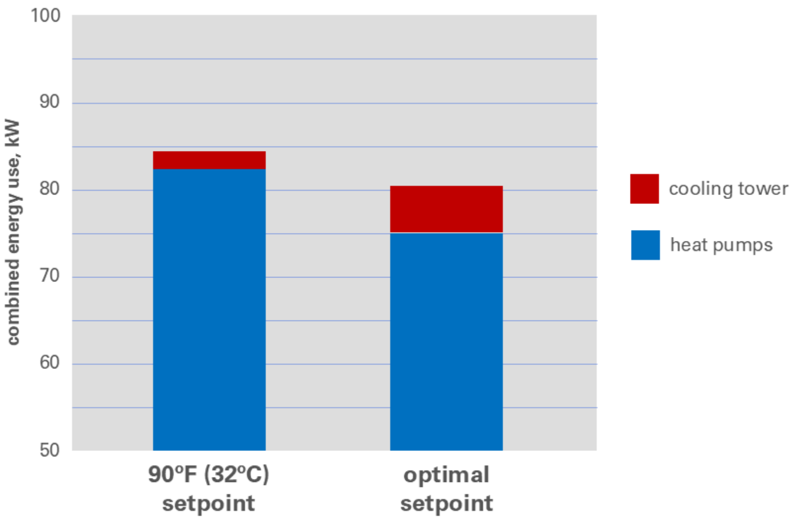
Loop temperature optimization

To maximize the energy-related benefits of a boiler/tower WSHP system, the loop water temperature is typically allowed to float across a wide range—between approximately 60ºF (16ºC) and 90ºF (32ºC) for example ("Water loop temperature control," p. 188).

When communicating, system-level controls are used, there is an opportunity to optimize the loop water temperature in an effort to minimize overall system energy use. In the cooling mode, the compressor in the WSHP uses less energy if the entering water temperature is cooler. However, making cooler water may require the cooling tower fans to use more energy, and the cooler water decreases the efficiency of any compressors operating in the heating mode.

A system-level controller can reset the loop temperature setpoint to minimize the combined energy consumed by the heat pumps and cooling tower under the current operating conditions (Figure 129).

Figure 129. Impact of loop temperature optimization



Coordination with other building systems

System-level control provides the opportunity to coordinate the operation of the HVAC system with other building systems, such as lighting, security, and fire protection. Following are some examples:

- A time-of-day schedule that is used to turn the HVAC system on and off could also be used to turn lights on and off inside or outside the building. In addition, an occupancy sensor could be used to indicate that a zone is actually unoccupied even though the BAS has scheduled it as occupied (see "Occupied standby mode," p. 184), and turn off all or some of the lights and/or plugged-in equipment. When the occupancy sensor indicates that the zone is again occupied, the lights are turned back on.
- An occupancy sensor that is used to turn on and off lights in a private office could also be used to slightly raise or lower the zone temperature setpoints and to reduce the outdoor airflow delivered to that zone when it is unoccupied (see "Occupied standby mode," p. 184).
- A card access security system could be used to turn on lights, start the HVAC system, and increase outdoor airflow delivered to a secure work area when the occupants "card in" for the day.
- A point-of-sale ticket system at a theater could be used to vary the outdoor airflow delivered to an individual theater based on the number of people that purchased tickets for the show.
- Activation of a fire alarm could enlist the help of the fans in the dedicated OA unit to perform a smoke-control function.

Glossary

ACH. Air changes per hour.

ADPI. Air Diffusion Performance Index. A measure of a supply-air diffuser's performance when delivering cool air to the zone.

adsorption. Process by which fluid molecules are concentrated on a surface by chemical or physical forces.

AHRI. Air-Conditioning, Heating, and Refrigeration Institute (www.ahrinet.org).

air-cooled condenser. A type of condenser in which refrigerant flows through the tubes and rejects heat to outdoor air that is drawn across the tubes.

air-source heat pump. A type of heat pump that transfers heat from indoor air to outdoor air and vice versa.

air-to-water heat pump (AWHP). A refrigerant compression device that heats a water loop, using ambient air as a source of heat. An AWHP can also operate in cooling mode with the use of a reversing valve.

air diffusion. Distribution of air within a conditioned space by an outlet discharging supply air in various directions and planes.

air-handling unit (AHU). A piece of equipment used to move, clean, and condition (heat, cool, humidify, dehumidify) air.

air separator. A component of a closed piping system that removes air that is entrained in the water distribution system.

airside economizer. A method of free cooling that involves using cooler outdoor air for cooling instead of recirculating warmer indoor air.

air-to-air energy recovery. The transfer of sensible heat, or sensible plus water vapor (latent heat), between two or more air streams, or between two locations within the same air stream.

ANSI. American National Standards Institute (www.ansi.org)

ARI. Former Air-Conditioning & Refrigeration Institute. See AHRI.

ASHRAE. American Society of Heating, Refrigerating and Air Conditioning Engineers (www.ashrae.org)

aspiration ratio. Total room air circulation divided by the air discharged from the outlet. Also called entrainment ratio.

attenuation. The reduction in the sound level as it travels along the path from a source to the receiver.

block cooling load. Calculated by finding the single instance in time when the sum of the space cooling loads is the highest.

blow-thru. A configuration where the fan is located upstream and blows air through the cooling coil.

boiler. A pressure vessel that typically consists of a water tank (or tubes with water flowing through them), a heat exchanger, fuel burners, exhaust vents, and controls. Its purpose is to transfer the heat generated by burning fuel to either water or steam.

boiler-less system. A WSHP system that does not include a centralized hot-water boiler connected to the water distribution loop. Rather, if the loop temperature gets too cold, the heat pump activates an external heat source, such as an electric heater installed inside the heat pump.

borehole. A narrow shaft bored in the ground, either vertically or horizontally, in which a U-tube is inserted as part of a ground heat exchanger.

brake horsepower (bhp). Actual, rather than theoretical, power required to drive a fan … or that is applied to the shaft of a pump or compressor.

breathing zone. The region within an occupied space between planes 3 in. and 72 in. (75 mm and 1800 mm) above the floor and more than 2 ft. (600 mm) from the walls or fixed air-conditioning equipment.

building automation system (BAS). A centralized control and monitoring system for a building.

building diversity. Calculated by dividing a building's block load by its sum-of-the-peaks load … also referred to as "system diversity."

CA. Conditioned Outdoor Air.

CDQ™. Trane's Cool, Dry, Quiet technology. See series desiccant wheel.

circuit setter. A device installed in the water pipe connected to a heat pump, which is used to measure and adjust the water flow rate.

closed circuit cooling tower. A type of cooling tower that keeps the fluid to be cooled separate from the water used in the evaporation process of the tower

Coefficient of Performance. See COP.

collection efficiency. Describes how well a particulate filter removes particles of various sizes from the air stream.

combustion efficiency. A measure of boiler efficiency that is calculated by dividing the fuel input to the boiler minus stack (flue gas outlet) loss by the fuel input to the boiler. This value generally ranges from 75 to 86 percent for most non-condensing boilers, and from 88 to 95 percent for condensing boilers.

compressor. A mechanical device used in the vapor-compression refrigeration cycle to increase the pressure and temperature of the refrigerant vapor.

condensate trap. Device for collecting liquid formed by the condensation of water vapor on a cooling coil, as it travels out of the drain pan, for the purpose of preventing the passage of air through the drain line.

condenser. The component of the refrigeration system where refrigerant vapor is converted to liquid as it rejects heat to water or air.

condensing boiler. A type of boiler that uses a high-efficiency heat exchanger designed to capture nearly all of the available sensible heat from the fuel, as well as some of the latent heat of vaporization. The result is a significant improvement in boiler efficiency.

condensing pressure. Pressure of the refrigerant vapor when it condenses into a liquid.

connected load. The sum of the capacities of all heat pumps installed in the system

constant-volume system. A type of air-conditioning system that varies the temperature of a constant volume of air supplied to meet the changing load conditions of the zone.

controller. The component of a control loop that compares the measured condition of the controlled variable to the desired condition (setpoint), and transmits a corrective output signal to the controlled device.

cool-down mode. See morning cool-down mode.

cooling dominant. When more WSHP capacity is used for cooling than for heating, meaning that heat must be rejected from the water loop by a cooling tower, the ground, or an AWHP operating in cooling mode.

cooling tower. An enclosed device for evaporatively cooling water by contact with air.

COP. A dimensionless ratio of the rate of heat removal to the rate of energy input (in consistent units) for a complete refrigerating system or some specific portion of that system under designated operating conditions. A higher COP designates a higher efficiency.

CPVC. Chlorinated Polyvinyl Chloride, a plastic piping material.

cycling. The practice of alternating a compressor on and off to match the system load.

damper. A device used to vary the volume of air passing through a confined cross section by varying the cross-sectional area.

deadband. The temperature range between the cooling and heating setpoints.

dedicated outdoor-air system (DOAS). A system that uses a dedicated air-handling unit to cool, heat, dehumidify, or humidify all of the outdoor air brought into the building for ventilation. This system then delivers this conditioned outdoor air directly to the conditioned spaces or to HVAC equipment.

dedicated outdoor-air unit. An air-handling unit used to cool, heat, dehumidify, or humidify all of the outdoor air brought into the building for ventilation. This conditioned outdoor air may be delivered directly to the zone(s) or to other air handlers or terminal equipment. Also called a makeup-air unit or 100 percent outdoor-air unit.

demand-controlled ventilation (DCV). A control strategy that attempts to dynamically reset the system outdoor-air intake based on changing population in the zone.

desiccant. Adsorbent or absorbent (liquid or solid) that removes water or water vapor from an air stream or another material.

dew point temperature (DPT). The temperature at which moisture leaves the air and condenses on surfaces.

diffuser. A device connected to the end of the supply-duct system, used to distribute the supply air into the conditioned space.

direct digital control (DDC). A method of terminal unit control using an electric motor to operate the air-modulation damper actuator. It uses a microprocessor that enables digital communication between the unit controller and a central building automation system.

direct-drive plenum fan. A type of plenum fan in which the motor is mounted directly to the end of the fan wheel shaft, eliminating the need for sheaves or belts.

direct expansion (DX) system. A system that uses the refrigerant directly as the cooling media. The refrigerant inside the finned-tube evaporator absorbs heat directly from the air used for space conditioning.

direct-fired burner. A fuel-burning device in which the heat from combustion and the products of combustion are transferred directly to the air stream being heated.

direct-return piping. A type of piping arrangement that minimizes the amount of piping by returning the water along the same path as it was supplied.

displacement ventilation. See thermal displacement ventilation.

distribution loop. Part of the water loop that connects to the WSHPs.

diversity. See building diversity.

draft. Undesired local cooling of a body caused by low temperature and air movement.

drain pan. A device positioned under a cooling coil to collect condensate and direct it to a drainage system.

draw-thru. A configuration where the fan is located downstream and draws air through the cooling coil.

ECM. Electronically commutated motor. A brushless DC motor that combines a permanent-magnet rotor, wound-field stator, and an electronic commutation assembly to eliminate the brushes.

EER. Energy Efficiency Ratio. The ratio of net cooling capacity (in Btu/hr) to total rate of electric input (in watts) at any given set of rating conditions, in watts per watt.

electrified WSHP system. A WSHP system that does not use a gas-fired boiler.

electronic air cleaner. Particulate filter that uses electrostatic attraction, either passively charged (electret) or actively charged (electrostatic precipitators), to enhance collection efficiency.

Energy Star®. A program, administered by the U.S. Environmental Protection Agency and Department of Energy, that helps reduce energy costs and protect the environment through energy-efficient products and practices (www.energystar.gov).

enthalpy. Describes the total amount of heat energy, both sensible and latent, in one pound of air at a given condition.

enthalpy wheel. See total-energy wheel.

equal friction duct design method. A method of designing an air duct system that results in an equal static pressure drop per foot (meter) of duct. Equal friction duct systems can be easily designed by hand.

evaporative cooling. Sensible cooling obtained by latent heat exchange from water sprays or jets of water.

evaporator. The component of the refrigeration system where cool, liquid refrigerant absorbs heat from air, causing the refrigerant to boil.

exhaust air. Air that is removed from the conditioned space(s) and then discharged to the outdoors.

expansion device. The component of the refrigeration system used to reduce the pressure and temperature of the refrigerant.

expansion tank. A component of a closed piping system that accommodates the expansion and contraction of the water as temperature and, therefore, density changes.

expansion valve. See expansion device.

face velocity. Velocity of the air as it passes through a device (airflow rate divided by the face area of the device).

fan performance curve. A plot of a specific fan's airflow capacity at a given speed (rpm) versus the static pressure it generates.

fan speed control. A method of supply fan modulation that affects a fan's capacity by varying its speed of rotation, commonly accomplished using a variable-speed drive on the fan motor.

flue gases. Exhaust gases from a boiler or gas-fired burner.

fluid cooler. See closed-circuit cooling tower.

four-way valve. See reversing valve.

glycol. A liquid that is mixed with water to lower the freezing point of the solution.

grille. A device used to direct air out of the conditioned space into the ceiling plenum or return duct system.

GSHP. Ground-source heat pump.

ground-coupled system. A type of ground-source heat pump system that uses a closed system of looped, polyethylene pipes that are buried in the ground in a vertical, horizontal, or spiral pattern.

ground heat exchanger. A closed system of looped, polyethylene pipes that are buried in the ground in a vertical, horizontal, or spiral pattern.

ground-source system. A type of water-source heat pump system that takes advantage of the earth’s relatively constant temperature and uses the ground or surface water as the heat rejecter and heat adder.

ground-water system. A type of heat pump system that supplies water from a well directly to each heat pump, and then returns it to the source or a drain field.

heat of compression. Energy, in the form of heat, created from the pressurization of a refrigerant vapor by a compressor.

heat pump. A device that transfers heat from one substance to another substance. It includes the basic refrigeration components of a compressor, condenser, evaporator, and expansion device. The difference is that it can also reverse the refrigeration cycle to perform heating as well as cooling.

heating dominant. When more WSHP capacity is used for heating than for cooling, meaning that heat must be added to the water loop by a boiler, the ground, or an AWHP operating in heating mode.

HEPA. High-efficiency particulate air filter.

hot gas reheat. A process where hot, high-pressure refrigerant vapor is diverted from the compressor through a separate reheat coil located downstream of the cooling coil, to improve part-load dehumidification.

humidity pull-down mode. An operating mode for transition from the unoccupied mode to the occupied mode, in which the HVAC system operates to lower the humidity inside the building to reach the desired occupied humidity setpoint by the time people enter the building.

IEEE. Institute of Electrical and Electronics Engineers (www.ieee.org).

IGSHPA. International Ground-Source Heat Pump Association (www.igshpa.org).

indirect-fired burner. A fuel-burning device in which the products of combustion do not come into contact with the air stream being heated, but are separated from the air stream through the use of a heat exchanger.

infiltration. Leakage of air into a building or space through cracks, crevices, doors, windows or other openings caused by wind pressure or temperature difference.

integrated economizer mode. An operating mode of an airside economizer when the outdoor air is warmer than the current supply-air temperature setpoint. The outdoor-air dampers remain wide open (return-air dampers are closed), but the unit controller activates compressors to provide the balance of the cooling capacity needed to provide supply air at the desired setpoint.

interior zone. A conditioned space that is surrounded by other conditioned spaces, with no perimeter walls/windows. Typically requires some degree of cooling all year long to overcome the heat generated by people, lighting, or equipment.

ISO. International Organization for Standardization (www.iso.org).

latent heat. Heat that causes a change in the moisture content of the air with no change in dry-bulb temperature.

LEED®. Leadership in Energy and Environmental Design. A building rating system created by the U.S. Green Building Council, a building industry coalition (www.usgbc.org).

linear slot diffuser. A type of supply-air diffuser in which jets are formed by slots or rectangular openings with a large aspect ratio. See Coanda effect.

makeup water. Water added to the cooling tower to compensate for the volume of water lost through drift loss, evaporation and blow-down (i.e., water wasted from the system to the sewer to reduce the concentration of solids).

makeup-air unit. See dedicated outdoor-air unit.

MERV. Minimum Efficiency Reporting Value. A rating value, defined by ASHRAE Standard 52.2, that depicts how efficiently a filter removes particles of various sizes.

mixed air. A mixture of outdoor air and recirculated return air.

modulated economizer mode. An operating mode of an airside economizer when the outdoor air is cool enough to handle the entire cooling load, and the compressors are off. The controller modulates the positions of the outdoor-air and return-air dampers so that the mixture of outdoor and return air provides supply air at the desired setpoint.

moisture carryover. Retention and transport of water droplets in an air stream.

morning cool-down mode. A typical operating mode for transition from the unoccupied mode to the occupied mode during the cooling season. It establishes the zone occupied comfort conditions, because they were allowed to drift from the occupied setpoint during the unoccupied mode, usually to save energy.

morning warm-up mode. A typical operating mode for transition from the unoccupied mode to the occupied mode during the heating season. It establishes the zone occupied comfort conditions, because they were allowed to drift from the occupied setpoint during the unoccupied mode, usually to save energy.

night setback. See setback.

Noise Criteria (NC). A single number used to describe sound in a occupied space. It uses a series of curves for plotting sound pressure by octave band and determining the NC value.

non-condensing boiler. A conventional boiler, designed to operate without condensing the flue gases inside the boiler. Only the sensible heat value of the fuel is used to heat the hot water. All of the latent heat value of the fuel is lost up the exhaust stack.

occupied mode. The typical daytime operating mode of a system. The building must be ventilated, and the comfort cooling or heating temperature setpoints must be maintained in all occupied zones.

occupied standby mode. A daytime operating mode of a system, when a zone is expected to be occupied but an occupancy sensor indicates that it is not presently occupied. All or some of the lights can be shut off, the temperature setpoints can be raised or lowered slightly, and the outdoor airflow required can be reduced (typically to the building-related ventilation rate, R_a).

optimal start. An optimized morning warm-up routine that determines the length of time required to bring the zone from its current temperature to the occupied setpoint temperature, and then waits as long as possible before staring the system, so the zone reaches the occupied setpoint just in time for scheduled occupancy.

outdoor air. Air brought into the building from outside, either by a ventilation system or through openings provided for natural ventilation.

perimeter zone. A conditioned space with walls and windows that are exposed to the outdoors. In most climates these spaces would require seasonal cooling and heating.

plenum. The space between the ceiling and the roof or the floor above.

population diversity. The ratio of the actual system population to the sum of the peak zone populations.

pressure-independent. VAV control method that directly controls the actual volume of primary air that flows to the zone. The position of the air-modulation damper is not directly controlled and is simply a by-product of regulating the airflow through the unit. Since the airflow delivered to the zone is directly controlled, it is independent of inlet duct static pressure.

pressure/temperature (P/T) ports. Self-sealing orifices that allow insertion of a probe-type thermometer or pressure gauge directly into the system water.

primary air. Conditioned air delivered by a central supply fan to a terminal unit.

production loop. Part of the water loop that connects to a cooling tower, boiler, or AWHP.

psychrometric chart. A tool used to graphically display the properties of moist air.

pump. Device for transferring a liquid or gas from a source or container through tubes or pipes to another container or receiver.

PVC. Polyvinyl Chloride, a plastic pipe material.

recirculated return air. Air removed from the conditioned space and reused as supply air, usually after passing through an air-cleaning and -conditioning system, for delivery to the conditioned space.

reducer. A transition that reduces the size of the air duct.

refrigerant. A substance used to extract and transport heat for the purpose of cooling.

refrigerant-to-air heat exchanger. A finned-tube coil inside a water-source heat pump. In the cooling mode, it acts like an evaporator and the refrigerant inside tubes extracts heat from the air flowing across the fins and tubes. In the heating mode, it acts like a condenser and the refrigerant rejects heat to the air.

refrigerant-to-water heat exchanger. Typically a coaxial (tube-within-a-tube) heat exchanger inside a water-source heat pump. In the cooling mode, it acts like a condenser. The water flowing through the outer tube extracts heat from the refrigerant flowing through the inner tube. In the heating mode, it acts like an evaporator and the refrigerant extracts heat from the water.

return air. Air that is removed from the conditioned space(s) and either recirculated or exhausted.

return-air grille. See grille.

reverse-return piping. A type of piping arrangement where the water being supplied to each coil travels through essentially the same distance of supply and return pipe, reducing system design and balancing time.

reversing valve. The component of a heat pump that allows it to perform heating as well as cooling. In the heating mode, refrigerant vapor from the compressor is diverted, by the reversing valve, to the refrigerant-to-air heat exchanger.

Room Criteria (RC). A single number used to describe sound in an occupied space. It uses a series of curves and reference lines for plotting sound pressure by octave band and determining the RC value and a descriptor of the sound quality (i.e., hiss, rumble).

sensible-energy recovery. The transfer of sensible heat between two or more air streams or between two locations within the same air stream.

sensible heat. Heat that causes a change in the dry-bulb temperature of the air with no change in moisture content.

sensor. The component of a control loop that measures the condition of the controlled variable and sends an input signal to the controller.

series desiccant wheel. A dehumidification device in which the downstream (process) side of the desiccant wheel is located downstream of the cooling coil and the upstream (regeneration) side of the wheel is located upstream of the cooling coil.

setback. The practice of changing the temperature setpoint of the zone during unoccupied hours in an effort to save energy.

setpoint. The desired condition of the controlled variable in a control loop.

silencer. A device installed in an air distribution system to reduce noise.

SMACNA. Sheet Metal and Air Conditioning Contractors National Association (www.smacna.org).

specific gravity (SG). Weight of a volume of material compared to the weight of the same volume of water.

specific heat. Quantity of heat required to raise the temperature of a definite mass of a material a definite amount compared to that required to raise the temperature of the same mass of water the same amount, expressed in units of Btu/lb•°F (J/kg•°K).

stack effect. When indoor air is warmer than outdoor air, the less dense column of air inside the building results in a negative pressure in the lower floors and a positive pressure in the upper floors. This pressure difference induces outdoor air to enter the lower floors and indoor air to leave the upper floors, while air flows upward within shafts and stairwells.

strainer. A component of the water distribution loop, installed prior to the inlet of each water-circulating pump, to protect the pumps from debris flowing inside the water distribution loop.

supply air. Air that is delivered to the zone by mechanical means for ventilation, heating, cooling, humidification, or dehumidification.

supply-air diffuser. See diffuser.

supply duct system. A system that is typically constructed of ductwork, fittings, and diffusers. This system transports the supply air from the air-conditioning equipment to the conditioned space.

surface-water system. A type of ground-source heat pump system that uses a series of closed loops of piping submerged in a pond or lake.

surge. A condition of unstable fan operation where the air alternately flows backward and forward through the fan wheel, generating noise and vibration.

TAB. Test, adjust, and balance.

thermal conductivity. Time rate of heat flow through a homogeneous material, expressed in units of Btu/hr•ft•°F (W/m•°K). This property characterizes the rate at which heat transfers from the ground heat exchanger to the surrounding soil.

thermal diffusivity. Calculated by dividing the measured thermal conductivity of the ground and by the estimated heat capacity of the ground, often expressed in units of ft^2/day (m^2/day).

thermal displacement ventilation. A method of air distribution in which cool air is supplied at low velocity, directly to the lower part of the occupied space. Heat is carried by convective flows created by heat sources into the upper part of the zone, where is extracted.

thermal expansion valve. A type of expansion device that uses a thermally-actuated valve to sense and control the flow rate of liquid refrigerant to the evaporator.

throw. Horizontal or vertical axial distance an air stream travels after leaving an air outlet before the maximum stream velocity is reduced to a specified terminal velocity, defined by ASHRAE Standard 70.

total-energy recovery. The transfer of sensible and latent (moisture) heat between two or more air streams or between two locations within the same air stream.

total-energy wheel. A rotating, heat-recovery device that recovers sensible (temperature) and latent (humidity) heat from one air stream and releases it to another adjacent air stream. Also known as a rotary heat exchanger, passive desiccant wheel, heat wheel, or enthalpy wheel.

transmission loss. A term used to measure the effect of a barrier on reducing the amount of transmitted sound. It is the ratio of sound power on the receiver side of a barrier to the sound power on the source side.

Traq™ damper. Trane’s flow-measuring outdoor-air damper.

TXV. See thermal expansion valve.

u-tube. The component of a ground heat exchanger that is inserted into the borehole. It is typically constructed of two continuous sections of high-density polyethylene (HDPE) with a factory-attached 180-degree fitting (U-bend) at the bottom.

underfloor air distribution. A method of air distribution in which conditioned air is delivered to the zones under a raised floor and floor grilles.

unoccupied mode. The typical nighttime operating mode of a system. The building does not require ventilation because it is not occupied, and the zone temperatures are allowed to drift to unoccupied setpoints.

variable-frequency drive (VFD). See variable-speed drive.

variable-speed drive (VSD). A device used to vary the capacity of a fan, pump, or compressor by varying the speed of the motor that rotates the drive shaft.

ventilation. The intentional introduction of outdoor air into a zone through the use of the HVAC system in the building.

warm-up mode. See morning warm-up mode.

water chiller. A refrigerating machine used to transfer heat between fluids.

water loop. A hydronic loop that is piped to the individual WSHP units and includes a means for heat addition and heat rejection, depending on operating conditions.

water regulating valve. A type of valve used to vary the flow rate of water through the refrigerant-to-water heat exchanger in a WSHP for the purpose of controlling the refrigerant pressure within a desired range.

waterside economizer. A method of free cooling that diverts cool water from the loop through a separate heat exchanger to precool the entering air before it reaches the refrigerant-to-air heat exchanger.

water-source heat pump. A type of heat pump that transfers heat from air to water and vice versa.

water-to-water heat pump. A type of heat pump that transfers heat from one water loop to another water loop and vice versa.

WRV. See water-regulating valve.

WSHP. Water-source heat pump.

zone. One occupied space or several occupied spaces with similar characteristics (thermal, humidity, occupancy, ventilation, building pressure).

zone air-distribution effectiveness (Ez). A measure of how effectively the air delivered to the zone by the supply-air diffusers reaches the breathing zone.

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