Applications Engineering Manual

Chilled-Water VAV Systems

September 2009

SYS-APM008-EN

Chilled-Water VAV Systems

John Murphy, applications engineer
Beth Bakkum, information designer

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 *chilled-water, variable-air-volume (VAV) 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 chilled-water VAV system, 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.

Contents

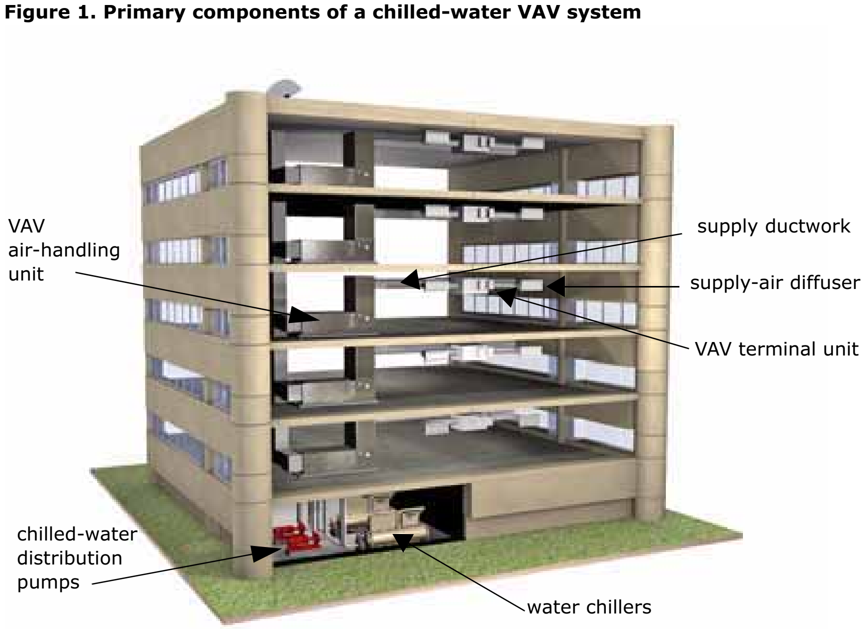
Overview of a Chilled-Water VAV System

A typical chilled-water variable-air-volume (VAV) system consists of a VAV air-handling unit that serves several individually controlled zones. Each zone has a VAV terminal unit that varies the quantity of air delivered to maintain the desired temperature in that zone.

The primary components of a typical chilled-water VAV system (Figure 1) include:

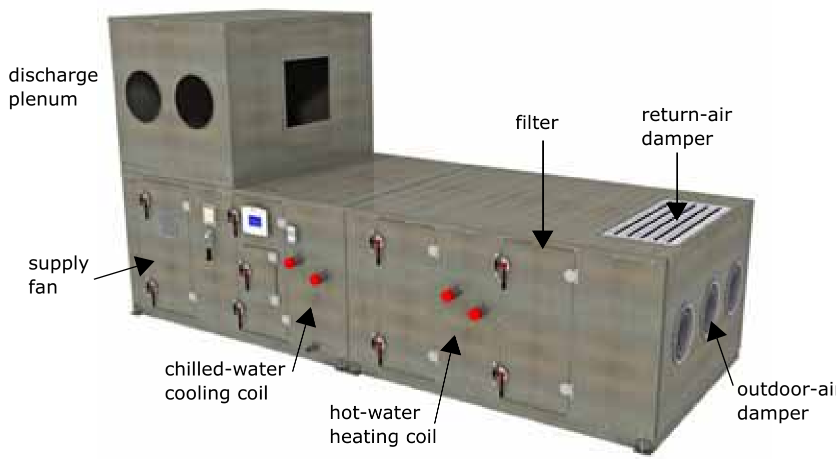
- VAV air-handling unit that contains a mixing box; filters; a chilled-water cooling coil; possibly a heating coil, gas-fired burner, or electric heater; a variable-volume supply fan; possibly a return or relief fan; and controls
- VAV terminal unit with a temperature sensor for each independently controlled zone
- Supply ductwork and supply-air diffusers
- Return-air grilles, ceiling plenum, and return ductwork
- Water chiller(s) with associated water distribution pumps and heat rejection equipment (cooling towers for water-cooled chillers, condenser fans for air-cooled chillers)
- Hot-water boiler(s), with associated water distribution pumps, or electric heat
- System-level controls

Figure 1. Primary components of a chilled-water VAV system



The VAV air-handling unit can be located either outdoors, typically on the roof of the building, or indoors, typically in a penthouse or mechanical equipment room in the basement or on one of the occupied floors of the building. A building may use a single air-handling unit or several units, depending on its size, load characteristics, and function. Return air from inside the building is drawn back to the air-handling unit. Some of this air is exhausted while the rest enters the air-handling unit through a return-air damper to be mixed with outdoor air that enters through a separate damper. This mixed air typically passes through a filter, a heating coil, a chilled-water cooling coil, and a supply fan before it is discharged from the unit (Figure 2).

Figure 2. Typical air-handling unit used in a VAV system



The supply air is distributed through ductwork that is typically located in the ceiling plenum above each floor (Figure 1). The supply ductwork delivers air to each of the VAV terminal units, then this air is introduced into the zones through supply-air diffusers. Each independently controlled zone has a VAV terminal unit that varies the quantity of air delivered to maintain the desired temperature in that zone. Air typically returns from the zones through ceiling-mounted return-air grilles and travels through the open ceiling plenum to a central return duct that directs this return air back to the air-handling unit.

The chilled water for cooling is provided by a chilled-water system, which includes one or more water chillers with associated water distribution pumps and heat rejection equipment (cooling towers for water-cooled chillers, condenser fans for air-cooled chillers).

Heating can be accomplished in several ways. One approach uses a heating coil (hot water, steam, or electric) or gas-fired burner inside the air-handling unit. In this configuration, the air-handling unit can warm the supply air during cold weather or during a morning warm-up period. A second approach uses individual heating coils (hot water or electric) installed in the VAV terminal units. Each coil is controlled to warm the supply air when necessary. A third approach uses perimeter baseboard radiant heat within those zones that require heat. The baseboard heaters can be controlled separately or by the controller on the VAV terminal unit.

When hot water is used for heating, it is provided by a hot-water system, which includes one or more boilers with associated water distribution pumps.

Each VAV terminal unit is equipped with a unit controller that regulates the flow of primary (supply) air to the zone to provide cooling, heating, and ventilation for the zone it serves. The VAV air-handling unit is also equipped with its own controller. A system-level controller ties the individual VAV terminal unit controllers to the controller on the air-handling unit, providing intelligent, coordinated control so that the individual pieces of equipment operate together as a system.

Basic System Operation

Unlike a constant-volume system, which delivers a constant quantity of air at varying temperatures, a VAV system delivers a varying quantity of constant-temperature air. The following section describes, in a very simple manner, how a typical chilled-water VAV system operates. For a more detailed discussion, see “System Controls,” p. 171.

Zone is occupied and requires cooling

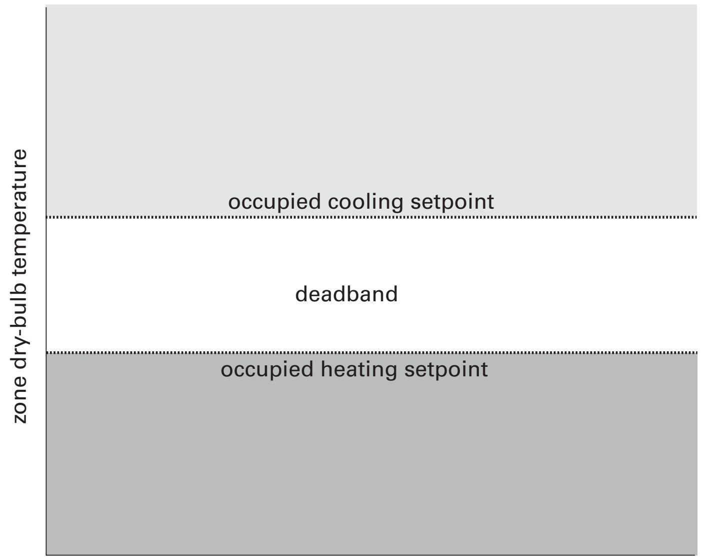
A sensor in each zone compares the dry-bulb temperature in the zone to a setpoint, and the VAV terminal responds by modulating the volume of supply air to match the changing cooling load in the zone. As the cooling load decreases, the VAV terminal responds by reducing the quantity of cold air delivered to the zone.

The VAV air-handling unit is controlled to maintain a constant supply-air temperature. Depending on the condition of the outdoor air, this may involve modulating a control valve on the chilled-water cooling coil, using outdoor air for “free cooling” (airside economizing), or modulating a control valve on the hot-water heating coil or gas-fired burner. The central supply fan modulates to maintain the pressure in the supply ductwork at a setpoint; this pressure ensures that all zones receive their required quantities of cold air. The outdoor air damper allows the required amount of fresh, outdoor air to be brought into the system for ventilation.

Zone is occupied, but requires no cooling or heating

As the cooling load in the zone decreases, the damper in the VAV terminal closes until it reaches the minimum airflow setting. As the load continues to decrease further, the constant quantity of cool air causes the dry-bulb temperature in the zone to drop below the cooling setpoint. If the temperature in the zone falls below the cooling setpoint, but remains above the heating setpoint, the VAV terminal takes no control action, remaining at its minimum airflow setting. The temperature range between the cooling and heating setpoints is called the deadband (Figure 3).

Figure 3. Occupied zone temperature setpoints



The air-handling unit is controlled to maintain a constant supply-air temperature by either modulating the chilled-water control valve, using the airside economizer, or modulating the heating control valve. The supply fan modulates to maintain a constant pressure in the supply ductwork, and the outdoor-air damper brings in at least the minimum required amount of outdoor air for ventilation.

Zone is occupied and requires heating

In many chilled-water VAV systems, zones that require heating include a heating coil (hot water or electric) in the VAV terminal unit. Alternatively, some systems use baseboard radiant heat located along the perimeter walls within the zone.

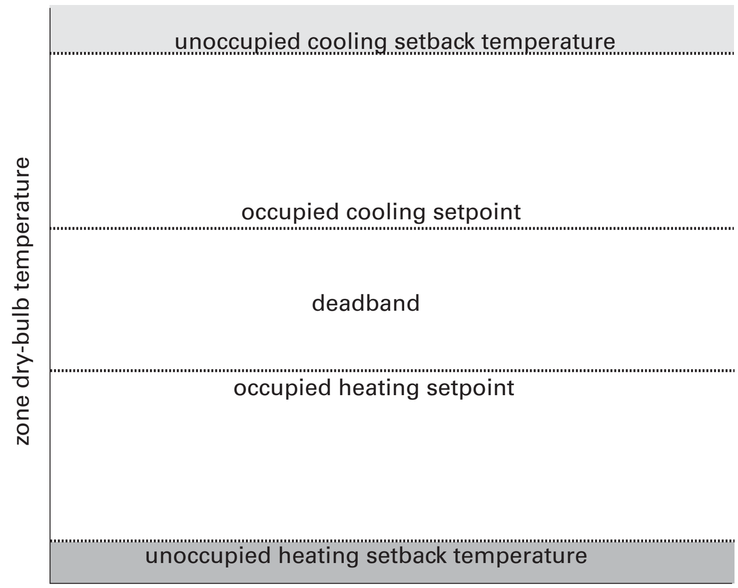
When the temperature in the zone drops below the heating setpoint, the controller on the VAV terminal unit activates the heating coil, warming the air supplied to the zone. If baseboard radiant heat is used instead, it is activated to add heat directly to the zone.

The air-handling unit is controlled to maintain a constant supply-air temperature, by either modulating the chilled-water control valve, using the airside economizer, or modulating the heating control valve. The supply fan modulates to maintain a constant pressure in the supply ductwork, and the outdoor-air damper brings in at least the minimum required amount of outdoor air.

Zone is unoccupied

When the zone is scheduled to be unoccupied, most buildings relax the zone setpoints, allowing the temperature in the zone to either increase or decrease. (In fact, this practice is required in many buildings by local codes or energy standards.) 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 VAV terminal unit closes to prevent any air from being supplied to the zone. Also, any local heat (heating coil in the VAV terminal or baseboard heat within the zone) is off.

If all zones served by the air-handling unit are unoccupied and the zone temperatures are within the deadband, the supply fan typically shuts off. Because the building is unoccupied, no ventilation is required and the outdoor-air damper is closed.

Some systems incorporate a “timed override” button on the zone temperature sensor, which allows the occupant to temporarily switch the system into the occupied mode, even though it is scheduled to be unoccupied. After a fixed period of time (three hours, for example), the system 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 “unoccupied” 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, the ventilation delivered to that zone can be reduced, and the minimum airflow setting of the VAV terminal can be lowered. When the occupancy sensor indicates that the zone is again occupied, the zone is switched back to the normal occupied mode.

Benefits of Chilled-Water VAV Systems

A comprehensive list of system benefits depends on which type of system is the basis of comparison. The following section discusses some of the primary benefits of using a chilled-water VAV system.

Provides multiple zones of comfort control

Chilled-water VAV systems are popular because they are capable of controlling the temperature in many zones with dissimilar cooling and heating requirements, while using a central air-handling unit. This is accomplished by providing a VAV terminal unit and temperature sensor for each independently controlled zone.

When the sun is shining against the west side of the building in the late afternoon, a VAV system can provide an increased amount of cool supply air to keep the perimeter zones along the west exposure comfortable, while throttling back the airflow to the zones along the east exposure so as not to overcool them.

Load diversity results in less supply airflow and a smaller supply fan

For more information on the impact of load diversity in multiple-zone systems, refer to the Trane *Air Conditioning Clinic* titled "Cooling and Heating Load Estimation" (TRG-TRC002-EN).

When an air-handling unit is used to deliver air to multiple zones, the method used to size that supply fan depends on whether the system is designed to deliver a constant or variable quantity of air to each zone. If the system is designed to deliver a constant quantity of air to each zone (a constant-volume system), the supply fan must be sized by summing the peak (design) airflow requirements for each of the zones it serves, regardless of when those peak requirements occur. However, if the system varies the quantity of air delivered to each zone, as is the case in a VAV system, the supply fan can be sized based on the one-time, worst-case overall ("block") airflow requirement of all the zones it serves, since all zones do not require peak (design) airflow simultaneously.

Impact of load diversity on sizing the supply fan

As seen in the example shown in Table 1, the peak space cooling loads do not necessarily occur at the same time for all spaces served by the system. Room 101 has several west-facing windows, so the peak (highest) space sensible cooling load occurs in late afternoon (4:00 p.m.) when the sun is shining directly through the windows. Room 102 has several east-facing windows, so the peak space sensible cooling load occurs in the morning (8:00 a.m.) when the rising sun shines directly through its windows.

If a single, constant-volume system is used to serve these two zones, the system must deliver 3,440 cfm (1.62 m^3/s) to Room 101 and 2,880 cfm (1.36 m^3/s) to Room 102 at all times. So, a constant-volume supply fan needs to be sized to deliver 6,320 cfm (2.98 m^3/s). This is often called the "sum-of-peaks" airflow.

Although Rooms 101 and 102 peak at different times during the day, there will be a single instance in time when the sum of these two space loads is highest. If these two rooms are served by a single VAV system, the supply fan need only be sized for the time when the sum of the space sensible cooling loads is the highest, which occurs at 4:00 p.m. in this example. So, a variable-volume supply fan need only be sized to deliver 5,990 cfm (2.82 m^3/s). This is often called the "block" airflow.

This zone-by-zone load variation throughout the day (called "load diversity") is the reason that VAV systems can deliver less air (18 percent less in this example) and use smaller supply fans, than multiple-zone constant-volume systems.

Similarly, the peak cooling coil loads do not necessarily occur at the same time for all VAV air-handling units served by a central chilled-water plant. This system-by-system load variation ("block load") is the reason that the chilled-water plant serving VAV air-handling units can be sized for less total capacity than if the building is served by multiple rooftop VAV systems.

Table 1. Sum-of-peaks versus block airflow

	8:00 a.m.		4:00 p.m.	
	space sensible cooling load, Btu/hr (W)	**supply airflow, cfm (m^3/s)**	**space sensible cooling load, Btu/hr (W)**	**supply airflow, cfm (m^3/s)**
Room 101 (faces West)	44,300 (13,000)	2,040 (0.96)	74,600 (21,900)	3,440 (1.62)
Room 102 (faces East)	62,400 (18,300)	2,880 (1.36)	55,300 (16,200)	2,550 (1.20)
"Block" airflow		4,920 (2.32)		5,990 (2.82)
"Sum-of-peaks" airflow	3,440 + 2,880 = 6,320 cfm (1.62 + 1.36 = 2.98 m^3/s)			

Opportunity to save energy

The part-load energy savings inherent with a VAV system is twofold. First, reducing the amount of air delivered at part load creates an opportunity to reduce the fan energy required to move this air. The magnitude of this energy savings depends on the method used to modulate the capacity of the fan.

Second, the reduced airflow across the cooling coil allows the chilled-water control valve to reduce water flow in order to deliver a constant supply-air temperature. This results in a reduction in cooling energy compared to a constant-volume reheat system.

In addition, using chilled water as the cooling medium presents further opportunity for energy savings through the use of centralized, higher-efficiency cooling equipment and a water distribution system.

Flexibility of equipment location

Chilled-water VAV systems offer significant flexibility when locating the various components of the system. The design team can maximize the amount of usable floor space in the building by using an air-cooled chiller, outdoor air-handling units, and VAV terminal units installed in the ceiling plenum. Or, equipment can be located indoors (water-cooled chillers and indoor air-handling units) to improve access for maintenance and prolong equipment life.

Centralizing the cooling and heating equipment minimizes disruption of the occupants when maintenance or repair is required. Similarly, the VAV terminals can be installed above corridors to minimize disruption of the occupants.

Typically, the only equipment located within the occupied space is the temperature sensor mounted on the wall. However, as mentioned earlier, in cold climates some VAV systems may use baseboard radiant heat located along the perimeter walls within the occupied space.

This flexibility also makes chilled-water VAV systems a popular choice for taller buildings, which are not well-suited for roof-mounted DX equipment and large areas for vertical air shafts.

Finally, using water chillers for cooling centralizes the refrigerant inside a few pieces of equipment. This minimizes the risks associated with refrigerant leaks compared to having refrigerant-containing equipment spread throughout the facility.

Flexibility of air-handling equipment

In general, air-handling units offer greater flexibility than packaged DX equipment. Air-handling units can typically be applied to a wider range of operating conditions, making them better suited for systems requiring variable airflow, lower cfm/ton (L/s/kW) (such as those with high percentages of outdoor air or colder supply-air temperatures), and tighter space control requirements.

In addition, air-handling units are typically available with a broader range of options, such as energy recovery devices, dehumidification enhancements, fan choices, air cleaning equipment, sound attenuation choices, and casing performance (thermal and leakage) options.

Able to adapt to changes in building use

Most chilled-water VAV systems use an open return-air plenum to allow air from all the zones to return back to the VAV air-handling unit. This, combined with the use of flexible ductwork to connect the VAV terminal units to supply-air diffusers, provides flexibility for the system to adapt to potential changes in building use.

Also, using chilled water as the cooling medium offers more flexibility to adapt to future changes in the building load. It is easier to design a chilled-water air-handling unit with "reserve capacity" than it is for a direct-expansion (DX) refrigeration system. In addition, centralized chilled-water and hot-water plants can also be designed to have "reserve capacity," which can be used by any of the air-handling units they serve.

Drawbacks/Challenges of Chilled-Water VAV Systems

Similar to the discussion of benefits, a list of drawbacks is dependent on which type of system is the basis of comparison. The following section, however, discusses some of the primary challenges related to chilled-water VAV systems, along with some potential ways to address those challenges.

More sophisticated system to design, control, and operate

The flexibility of a chilled-water VAV system, achieved through the use of more sophisticated equipment and controls, also means that it can be more challenging to design, control, and operate properly, than a system that uses more "packaged" components.

Single-zone, packaged DX equipment is simpler to design, install, and operate. But the packaged nature of this equipment limits flexibility, minimizes the potential to reduce energy use, and can increase maintenance since the equipment is distributed throughout the facility.

The challenge for the design team is to take advantage of the benefit of flexibility, while keeping the system easy to operate and maintain. Pre-engineered controls and easy-to-use building automation systems are two technologies that can help achieve this balance.

Ceiling space and vertical shafts are required to deliver conditioned air

Because the conditioned air is delivered by a central supply fan, ceiling space is required to duct the air from the air-handling unit to the VAV terminals, and eventually to the occupied spaces. In addition, for a multi-story building, one or more vertical air shafts may be needed to duct the outdoor air to floor-by-floor air-handling units. These vertical shafts take up some usable floor space in the building.

To minimize impact on the floor plan, these shafts are often located in the core of the building, next to elevator shafts and restrooms.

More sophisticated equipment to maintain and repair

Water chillers and hot-water boilers are more complicated pieces of equipment than packaged DX units. Some facilities prefer to use smaller, simpler equipment that can be maintained by the facility staff.

Many buildings that use chilled-water VAV systems use facility staff to perform regular maintenance (such as changing filters, inspecting and tightening fan belts, and cleaning drain pans) and then hire an outside service provider for repair work and regular maintenance of the larger pieces of equipment.

The right answer for each project depends on the expertise and availability of the facilities maintenance staff.

Common Building Types That Use Chilled-Water VAV Systems

Chilled-water VAV systems are used in almost all building types, but the most common uses include:

- Commercial office buildings
- Schools (both K-12 and higher education)
- Hospitals, clinics, and medical office buildings
- Large hotels and conference centers
- Large retail centers (shopping malls)
- Airports
- Laboratories
- Industrial facilities and manufacturing processes

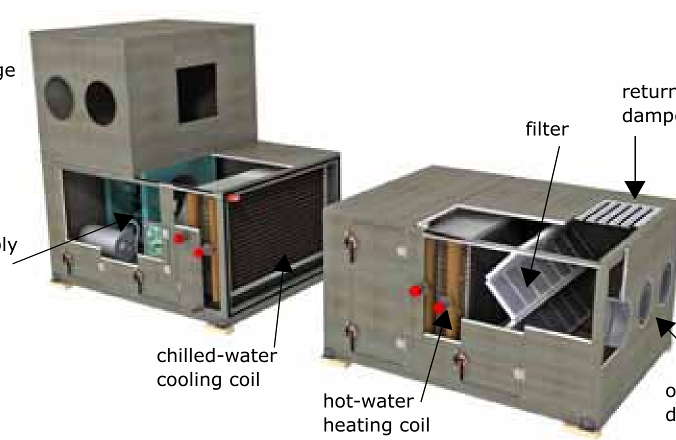
Primary System Components

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

VAV Air-Handling Unit

Return air from inside the building is drawn into the VAV air-handling unit (AHU) (Figure 5) through the return-air damper and is mixed with outdoor air that enters through the outdoor-air damper. This mixed air passes through a filter, a heating coil, a chilled-water cooling coil, the supply fan, and possibly a final filter before it is discharged into the supply ductwork.

Figure 5. Typical air-handling unit used for VAV applications



VAV air-handling units are typically available with a broad range of options, such as energy recovery devices, dehumidification enhancements, fan choices, air cleaning equipment, sound attenuation choices, and casing performance (thermal and leakage) options. Because of this flexibility, there is generally no "standard" configuration for a VAV system.

When multiple units are used, a common approach is to dedicate one air-handling unit (and, therefore, one VAV air distribution system) to serve each floor of the building. An advantage of this approach is that it minimizes the number and/or size of vertical shafts used to route ductwork inside the building. Using fewer and/or smaller vertical air shafts will likely increase the amount of usable floor space in the building. Another advantage is that, if the floors are leased to different organizations, it offers a simple way to bill tenants individually for their HVAC energy use.

An alternative approach is to use one air-handling unit to serve each exposure of the building. For example, all the west-facing zones are served by one unit, all the east-facing zones are served by another unit, and so on. The advantage of this approach is that all the zones served by an air-handling

A few large air-handling units or several smaller units?

A building may use a few large air-handling units or several smaller units, depending on size, load characteristics, and function. A study commissioned by the U.S. General Services Administration (GSA) concluded that using multiple small air-handling units is more desirable than using fewer large air-handling units (Callan, Bolin, and Molinini 2004).

Smaller air-handling units allow more diversity for after-hours operation, provide more flexibility, result in less cross-contamination between zones, allow for greater optimization of setpoints for energy savings, and often avoid the need for return fans. A survey of five newly constructed buildings revealed that using smaller air-handling units (< 50,000 cfm [24 m^3/s]) resulted in 7 to 8 percent energy savings compared to using larger air-handling units.

unit are thermally similar, potentially allowing for reduced energy use. A drawback is that, depending on the shape of the building, it may necessitate the use of multiple, vertical air shafts, which can result in less usable floor space. This approach may be combined with the use of floor-by-floor air-handling units to serve the interior zones.

Indoor versus outdoor air-handling units

The VAV air-handling unit can be located either outdoors or indoors. An outdoor unit is typically installed on the roof of the building. An indoor unit is typically installed in a penthouse, the basement, or a mechanical equipment room located on one of the occupied floors of the building.

Table 2 describes the advantages and drawbacks of each location.

Table 2. Indoor versus outdoor location of air-handling units

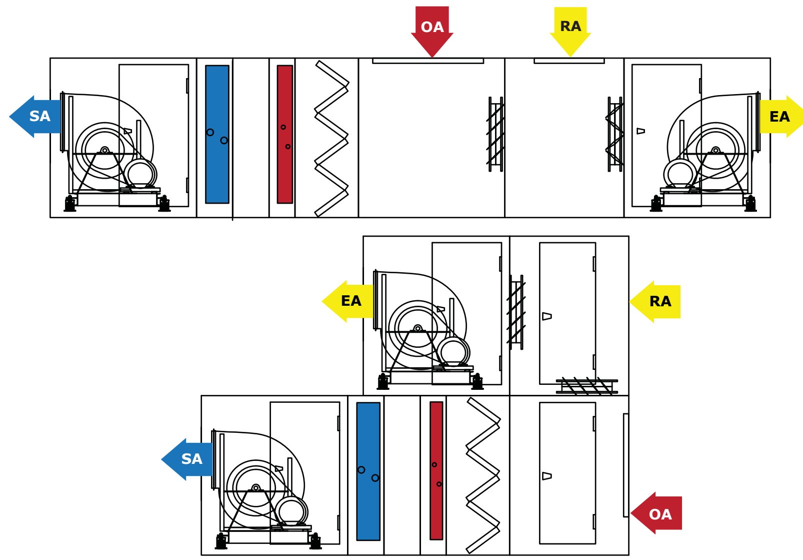
Indoor AHU	
Advantages:	**Disadvantages:**
• Preventive maintenance on the AHU is performed indoors, away from harsh weather. • Thermal performance of the AHU casing is less critical since indoor temperatures are milder, resulting in less heat loss/gain.	• Requires indoor floor space that could have otherwise been used by the building owner or leased to a tenant. • May require the equipment room to be conditioned to prevent condensation (sweating) on the exterior surfaces of the AHU.
Outdoor AHU	
Advantages:	**Disadvantages:**
• Frees indoor floor space that can be used by the building owner or leased to a tenant. • Less concern about condensation (sweating) on the exterior surfaces of the AHU, since any condensation drains onto the roof surface.	• Preventive maintenance on the AHU may need to be performed during harsh weather or require the construction of an outdoor service corridor. • May require more space for vertical air shafts, reducing the amount of usable floor space. • Roof structure may need to be strengthened to support the added weight of the AHUs. • Thermal performance of the AHU casing is more critical since extreme outdoor temperatures result in more heat loss/gain.

Installing the air-handling units indoors requires floor space that could have otherwise been used by the building owner or leased to a tenant. Following are several strategies used to minimize the floor space required by an indoor air-handling unit:

- *Stacked configurations*
  If the frame of the air-handling equipment is sturdy enough, components can be stacked on top of one another to minimize the footprint, or floor space required.

  The top unit in Figure 6 depicts an example VAV air-handling unit, with both a supply fan and relief fan, configured on one level. The overall length is 23.2 ft (7.1 m) and the width is 7.8 ft (2.4 m).

Figure 6. Example of a stacked configuration to reduce unit length



Source: Images from Trane TOPSS program

The bottom unit in Figure 6 depicts the same components configured in a stacked arrangement, where the diverting box and relief fan are stacked on top of the other modules. This reduces the overall length to 13.8 ft (4.2 m). The width of the unit remains the same but, of course, the unit is taller—10.2 ft (3.2 m) compared to 5.1 ft (1.6 m) for the non-stacked unit.

In many buildings with slab-to-slab heights of 12 ft (3.7 m) or more, the added height of a stacked air-handling unit is not an issue. And, by using a stacked configuration, the footprint of this example unit is reduced by 40 percent.

Of course, the larger the airflow capacity, the taller the air-handling unit will be. So, for very large units, stacking may not be possible due to the floor-to-floor height limitation of the building.

For more information on direct-drive fans and using multiple versus single fans, refer to the Trane engineering bulletin titled "Direct-Drive Plenum Fans for Trane Climate Changer™ Air Handlers" (CLCH-PRB021-EN).

- *Multiple fans (dual fans or fan arrays)*
  Using multiple fans, rather than a single supply fan, can also result in a shorter air-handling unit (Figure 27, p. 35). For a given airflow, a unit with multiple fans uses several, smaller-diameter fan wheels, rather than a single, larger-diameter fan wheel. Upstream and downstream spacing (length) requirements are typically a function of the fan wheel diameter—except for very small sizes where access requirements dictate the necessary spacing. Therefore, using multiple, smaller-diameter fan wheels can shorten the upstream and downstream spacing required, and can shorten the length of the overall air-handling unit (see "Fan types," p. 32). However, using multiple fans is typically less efficient and increases the cost of the air-handling unit.
- *Dual-path configuration*
  Another way to reduce AHU footprint is to use a dual-path configuration to separately condition the recirculated return air (RA) and outdoor air (OA). Each air path includes a dedicated cooling coil, but the same fan serves both paths (Figure 7).

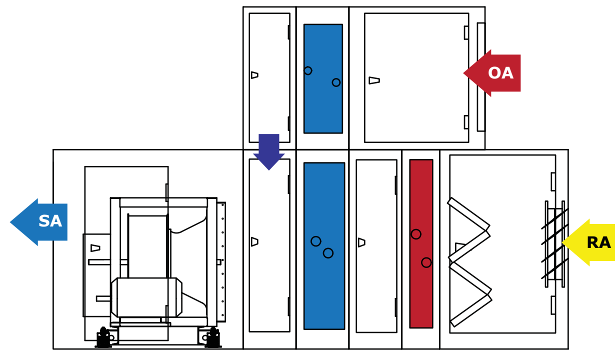
Avoiding fan surge in a dual-path AHU

When a dual-path configuration is used for a VAV air-handling unit, preventing the supply fan from operating in the surge region can be more challenging. As supply airflow is reduced at part load, outdoor airflow through the OA path may remain nearly constant (to ensure proper ventilation) while the recirculated airflow through the RA path is reduced. Therefore, the pressure drop through the OA path remains high while the pressure drop through the RA path decreases. This high-pressure drop through the OA path can cause the supply fan to surge at reduced supply airflow.

To help prevent the fan from operating in the surge region:

- Size the components in the OA path for a low airside pressure drop. This may involve increasing the casing size for the top section (OA path) of the air-handling unit.
- Carefully select the supply fan to reduce the potential for surge. A direct-drive plenum fan (p. 35) often provides the greatest flexibility for selection.
- Implement the fan-pressure optimization control strategy (p. 199), which allows the fan to generate less pressure at part load.
- Implement the ventilation optimization control strategy (p. 204), which reduces outdoor-air intake flow during partial occupancy. With less intake airflow, the pressure drop through the OA path decreases.

Figure 7. Dual-path VAV air-handling unit



Source: Image adapted from Trane TOPSS program

The face area of a cooling coil is dictated by the design airflow through that coil, and the size of the coil typically dictates the footprint of the air-handling unit: the larger the coil, the larger the AHU must be to house it. In a dual-path unit, because the RA cooling coil only conditions the recirculated air, rather than the mixture of outdoor and recirculated air, it can be smaller than it would be for a single-path unit.

Consider an example VAV air-handling unit that is sized for 13,000 cfm (6.1 m^3/s) of supply air, of which 3,500 cfm (1.6 m^3/s) is outdoor air and 9,500 cfm (4.5 m^3/s) is recirculated return air. In a single-path configuration, the single cooling coil must be sized for the total 13,000 cfm (6.1 m^3/s). For this unit, a size 30 AHU casing results in a coil face velocity of 435 fpm (2.2 m/s).

Note: The unit "size" typically represents the nominal face area of the cooling coil, in terms of ft^2. In this example, the face area of the size 30 air-handling unit is 29.90 ft^2 (2.78 m^2).

In a dual-path configuration, the RA (lower) cooling coil need only be sized for the 9,500 cfm (4.5 m^3/s) of recirculated air. For this path, a size 21 AHU casing results in a coil face velocity of 456 fpm (2.3 m/s). The OA (upper) coil, which is sized for the 3,500 cfm (1.6 m^3/s) of outdoor air, requires a size 8 AHU casing, which results in a coil face velocity of 438 fpm (2.2 m/s). The overall footprint of the dual-path unit (size 8 casing stacked on top of a size 21 casing) is smaller than that of a dual-path unit (size 30 casing), although the dual-path unit is taller (Figure 8 and Table 3).

Figure 8. Example footprint reduction from using a dual-path air-handling unit (see Table 3)

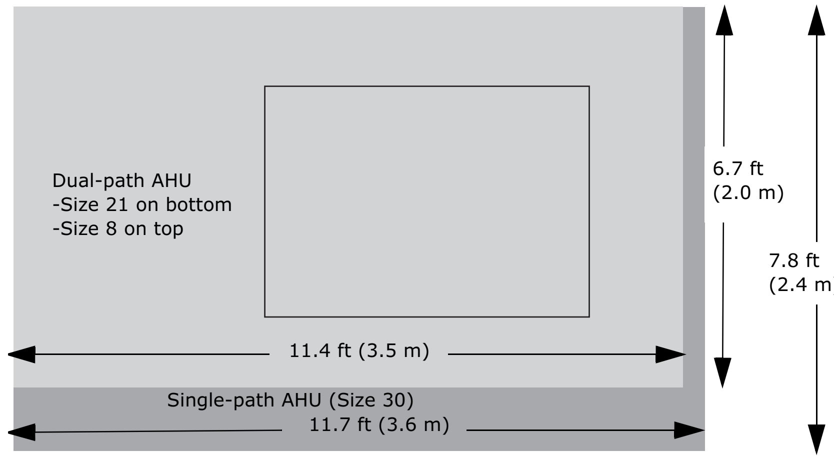


Table 3. Impact of dual-path configuration on AHU footprint and weight

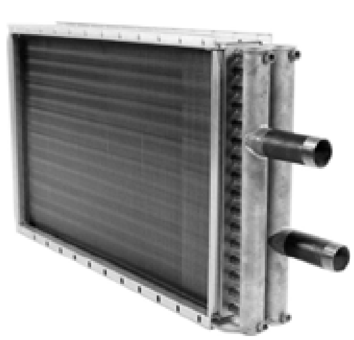
	Single-path AHU Size 30	Dual-path AHU bottom: Size 21 top: Size 8
AHU footprint, ft (m)	11.7 x 7.8 (3.6 x 2.4)	11.4 x 6.7 (3.5 x 2.0)
AHU height, ft (m)	5.1 (1.6)	7.5 (2.3 m)
AHU weight, lbs (kg)	2800 (1270)	2800 (1270)

- *Cold-air distribution*
  By reducing the supply-air temperature, less supply airflow is required to offset the sensible cooling loads in the zones. Reducing supply airflow can allow for the selection of smaller air-handling units, which can increase usable (or rentable) floor space.

  Cold-air VAV systems typically deliver supply air at a temperature of 45°F to 52°F (7°C to 11°C). For more information, see "Cold-Air VAV Systems," p. 147.

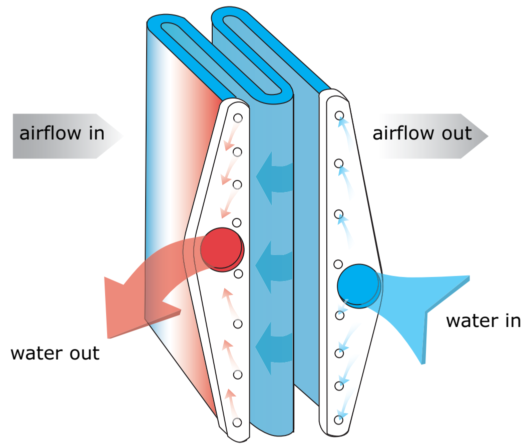
Chilled-water cooling coil

Figure 9. Actual cooling coil



Cooling in a chilled-water VAV system is accomplished using a chilled-water coil in the VAV air-handling unit. Cooling coils are finned-tube heat exchangers consisting of rows of tubes that pass through sheets of formed fins (Figure 9). As air passes through the coil and contacts the cold tube and fin surfaces, heat transfers from the air to the water flowing through the tubes (Figure 10).

Figure 10. Chilled-water cooling coil



In most applications, the cooling coil also dehumidifies as water vapor in the air condenses on the cold fin surfaces of the coil. This water then drains down the coil surfaces, drops into the drain pan located beneath the cooling coil, and is piped away by the condensate drain line.

The coil tubes are usually constructed of copper, and the coil fins of aluminum. For some applications, coils may use copper fins or a manufacturer may cover the coil surfaces with a coating to minimize corrosion.

Selection of the cooling coil impacts the cost of installing, operating, and maintaining both the VAV air-handling unit and the chilled-water system. For example, the amount of material used to construct the coil—overall size, number of tubes, number of fins—determines the initial cost: more material increases the cost. But the size of the cooling coil also dictates the weight and footprint of the air-handling unit: the larger the coil, the larger the AHU must be to house it. A larger AHU may require a larger mechanical room (reducing usable or rentable floor space), limit access for service, impact the amount of structural support needed, or challenge the arrangement of ductwork and piping.

Because the cooling coil is also part of the air distribution system, its geometry—size, number of rows, fin spacing, and fin profile—contributes to the airside pressure drop and affects the energy used and sound generated by the fans. A larger AHU will typically result in a lower airside pressure drop through its components, which can reduce fan energy (see example in Table 4).

Finally, because a chilled-water cooling coil is also part of the chilled-water system, its geometry contributes to the waterside pressure drop and affects the energy used by the pumps. And, the extent to which coils raise the chilled-water temperature dramatically affects both the installed cost of chilled-water piping and pumping energy. Coil performance can even influence the efficiency of the water chiller. For further discussion, see “Chilled-Water System,” p. 79.

Maximum face velocity to prevent moisture carryover

If a cooling coil also dehumidifies, it must be selected to prevent moisture carryover at design air velocities. While a long-time industry rule-of-thumb has been to select cooling coils for a face velocity no greater than 500 fpm (2.5 m/s) at design airflow, many of today's heat exchanger surfaces have been engineered to prevent moisture carryover at much higher velocities.

The footprint of the AHU cabinet is typically dictated by the size of the cooling coil, and the size of the cooling coil is often dictated by the allowable face velocity. An overly restrictive limit on coil face velocity results in the selection of a larger air-handling unit than may be necessary. This increases the cost of the equipment, results in heavier equipment that requires more structural support, and requires more floor space. Table 4 shows an example selection of a 13,000-cfm (6.1-m^3/s) VAV air-handling unit, selected to provide 525 MBh (154 kW) of total cooling capacity. Arbitrarily limiting coil face velocity to 500 fpm (2.5 m/s) results in the need to select a size 30 unit.

Note: The unit "size" typically represents the nominal face area of the cooling coil, in terms of ft^2. In the example depicted in Table 4, the face area of the size 30 air-handling unit is 29.90 ft^2 (2.78 m^2).

A size 25 unit, with its reduced coil face area, would result in a coil face velocity of 521 fpm (2.6 m/s) at design airflow. While this is higher than the industry rule-of-thumb, it is well below the manufacturer's tested limit for preventing moisture carryover.

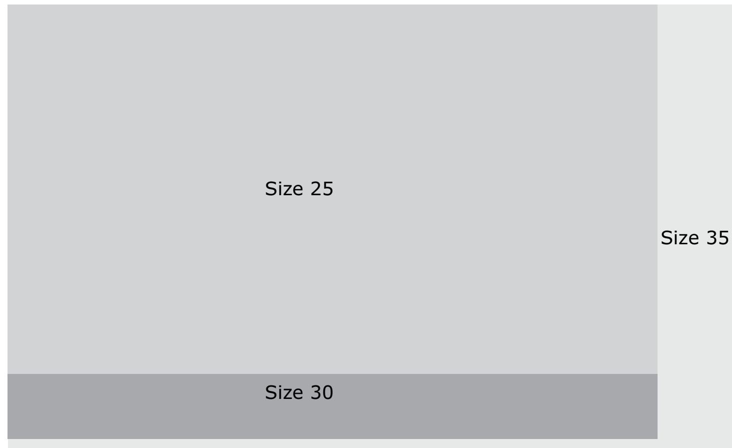
Table 4. Impact of cooling coil face velocity on AHU footprint and weight

	Size 25 AHU	Size 30 AHU	Size 35 AHU
Coil face area, ft^2 (m^2)	24.97 (2.32)	29.90 (2.78)	34.14 (3.17)
Face velocity, fpm (m/s)	521 (2.6)	435 (2.2)	381 (1.90)
Coil rows	6 rows	6 rows	4 rows
Fin spacing, fins/ft (fins/m)	103 (338)	83 (272)	137 (449)
Total cooling capacity, MBh (kW)	525 (154)	525 (154)	525 (154)
Airside pressure drop, in H_2O (Pa)	0.69 (173)	0.50 (125)	0.36 (90)
Fluid pressure drop, ft H_2O (kPa)	12.1 (36.3)	13.6 (40.6)	9.1 (27.1)
AHU footprint[1], ft (m)	12.1 x 6.7 (3.7 x 2.0)	12.1 x 7.8 (3.7 x 2.4)	13.5 x 8.0 (4.1 x 2.4)
AHU height[1], ft (m)	9.1 (2.8)	9.1 (2.8)	9.3 (2.8)
AHU weight[1], lbs (kg)	3350 (1520)	3570 (1620)	4700 (2130)

1 Based on a typical VAV air-handling unit layout consisting of an OA/RA mixing box, high-efficiency filters, hot-water heating coil, chilled-water cooling coil, airfoil centrifugal supply fan, and a top-mounted discharge plenum.

Selecting the smaller (size 25) air-handling unit results in lower cost of the equipment, reduces the weight of the unit by 6 percent, and reduces the footprint (floor space required) by 14 percent (Figure 11).

Figure 11. Impact of coil face velocity on AHU footprint (see Table 4)



Note that in order to deliver equivalent capacity, the cooling coil in the smaller-sized AHU requires more fins than the coil in the larger-sized AHU. This slightly increases the cost of the coil and, along with the smaller coil face area, increases the airside-pressure drop—0.69 in H_2O (173 Pa) for size 25 unit versus 0.50 in H_2O (125 Pa) for the size 30 unit. This does impact fan energy use. However, in a VAV air-handling unit, the airside-pressure drop decreases quickly as supply airflow is reduced at part load, so the actual impact on annual fan energy use is lessened. Finally, the smaller coil reduces the fluid pressure drop—12.1 ft H_2O (36.3 kPa) for size 25 unit versus 13.6 ft H_2O (40.6 kPa) for the size 30 unit—which may decrease pumping energy.

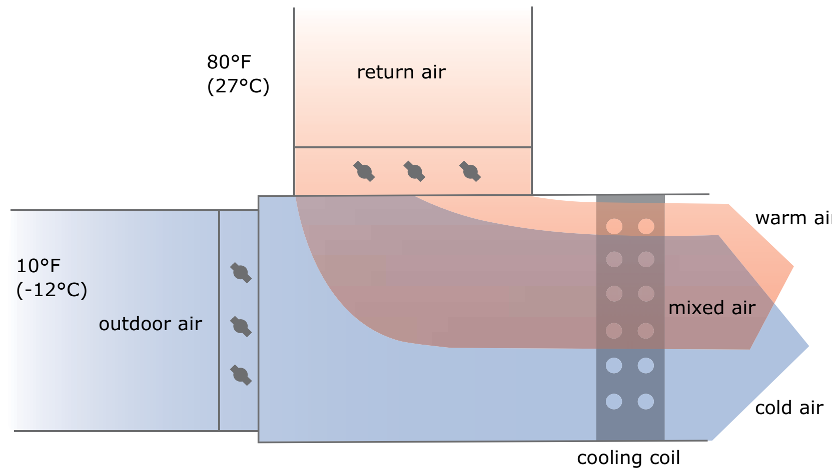
If the goal for a project is to minimize footprint and/or equipment cost, consider specifying that the face velocity at design airflow should not exceed 90 percent (for example) of the manufacturer's tested and published limit to prevent moisture carryover, rather than specifying an arbitrary maximum coil face velocity of 500 fpm (2.5 m/s). For example, if the manufacturer's published limit is 600 fpm (3.0 m/s), specify that the coil face velocity not exceed 540 fpm (2.7 m/s) at design airflow.

However, if the goal for a project is to reduce energy use, consider selecting a slightly larger air-handling unit. The example in Table 4 also shows a selection for a size 35 unit. With more surface area, the cooling coil can deliver equivalent capacity with only four rows of tubes, rather than six. Fewer rows, along with the larger coil face area, decreases the airside-pressure drop to 0.36 in H_2O (90 Pa) and decreases the fluid pressure drop to 9.1 ft H_2O (27.1 kPa). The result is reduced fan and pumping energy. However, this does increase the cost, footprint (Figure 11), and weight of the air-handling unit.

Freeze prevention

As discussed in "Ventilation," p. 101, at part-load conditions a properly controlled VAV air-handling unit typically brings in a high percentage of outdoor air. During cold weather it is difficult to mix the outdoor air and recirculated return air when the two air streams are at widely differing temperatures. Incomplete mixing results in distinct temperature layers (stratification) in the resulting “mixed” air stream (Figure 12). If a layer of sub-freezing air moves through an unprotected chilled-water or hot-water coil, the water can freeze and damage the coil.

Figure 12. Temperature stratification during cold weather



Typically, a low-limit thermostat (or “freezestat”) is installed on the upstream face of the water coil. This sensor measures the lowest temperature in any 12-in. (30-cm) section of the coil face. If the temperature of the air entering any section of the coil approaches 32°F (0°C), the unit controller responds by stopping the supply fan, closing the outdoor-air damper, or both. This adversely affects occupant comfort and indoor air quality.

If a chilled-water coil is likely to be exposed to air that is colder than 32°F (0°C), the system must include some method to protect the coil from freezing. Several common freeze-prevention methods are listed below. Choose the method that best suits the application.

- *Drain the chilled-water cooling coils during cold weather.*
  This requires vent and drain connections on every coil, as well as shutoff valves to isolate the coils from the rest of the chilled-water distribution system. After draining each coil, use compressed air to remove as much water as possible, add a small amount of antifreeze to prevent any remaining water from freezing, and disconnect the freezestat to avoid nuisance trips.

  The advantage of this approach is that it has minimal impact on the cost of the air-handling unit and has no impact on energy use. However, it does increase maintenance cost, especially in locations where the temperature can fluctuate widely during seasonal transitions, requiring the coils to be drained and filled several times each season.

- *Add antifreeze to the chilled-water system.*
  Adding antifreeze (such as glycol) to the chilled-water system lowers the temperature at which the solution will freeze. Given a sufficient concentration of glycol, no damage to the system will occur. For a VAV system, since the cooling coil is typically not used during sub-freezing

weather, a concentration that provides "burst protection" is usually sufficient for the chilled-water system (Table 13, p. 87). A concentration that provides "freeze protection" is only needed in those cases where no ice crystals can be permitted to form (such as a coil loop that operates during very cold weather) or where there is inadequate expansion volume available. Make sure to also use an inhibitor package to help resist corrosion.

The advantage of this approach is that it is predictable and relatively easy to maintain. However, antifreeze degrades the heat-transfer performance of cooling coils and chillers, often increasing the size and cost of these components. In addition, it increases the fluid pressure drop through the coils and chillers, impacting pumping energy use.

- *Preheat the outdoor air before it mixes with the recirculated air.* Using an electric heater, steam coil, or hot-water heating coil to preheat the sub-freezing outdoor air before it enters the mixing box decreases the temperature difference between the two air streams, which improves mixing effectiveness and reduces stratification.

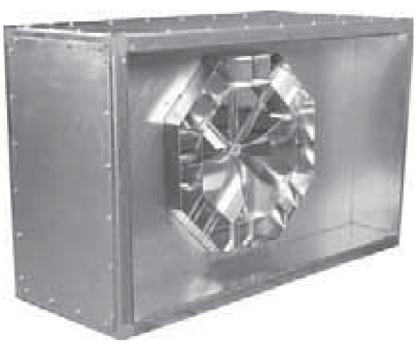
  The advantage of this approach is that it is predictable and effective. In cold climates, a source of heat may already be needed in the centralized VAV air-handling unit. However, because the heat source needs to be located in the outdoor air stream, this approach may limit flexibility or increase the cost of the air-handling unit. And, if a hot-water or steam preheat coil is used, they also require some method of freeze prevention (see "Heating coil," p. 23).

  One common approach is to use a preheat coil with integral face-and-bypass dampers. These dampers modulate to vary the amount of heat transferred to the air, while allowing full water or steam to flow through the coil tubes.

- *Use air-to-air energy recovery to preheat the outdoor air.* As an alternative to the prior method, an air-to-air energy recovery device (such as a coil loop, fixed-plate heat exchanger, heat pipe, or wheel) can be used to preheat the entering outdoor air during cold weather (see "Air-to-Air Energy Recovery," p. 160).

  The advantage of this approach is that it also reduces cooling and heating energy use, and can allow for downsizing of cooling and heating equipment. However, such a device does increase the cost of the air-handling unit and adds a pressure drop to both the outdoor and exhaust air streams, which increases fan energy use.

Figure 13. Air-mixing baffles

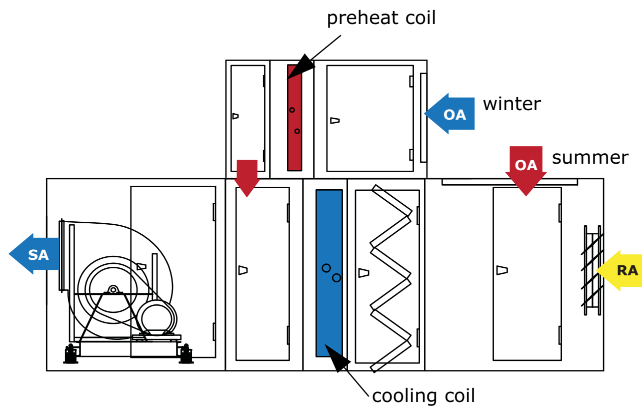


- *Use air-mixing baffles.* This configuration of baffles (Figure 13), located immediately downstream of the mixing box, adds rotational energy and increases the velocity of the air stream, which improves mixing (blending) to prevent or minimize temperature stratification.

  The advantage of this approach is that it works consistently and requires no maintenance. However, it does increase the cost and length of the air-handling unit, since distance is needed downstream for the air to finish mixing and slow down before reaching the filters. Also, the baffles add pressure drop (typically 0.2 in. H_2O [50 Pa]), which increases fan energy use, and mixing effectiveness decreases as airflow is reduced.

- *Introduce cold outdoor air downstream of the cooling coil.*
  This configuration, sometimes called a "winterizer," uses a combination of two different-sized air-handling units (Figure 14) configured to allow the outdoor air to be introduced downstream of the cooling coil whenever the outdoor air is colder than 32°F (0°C). The smaller air-handling unit, sized for the minimum required ventilation airflow, contains filters and possibly a small preheat coil.

Figure 14. "Winterizer" configuration



Source: Image adapted from Trane TOPSS program

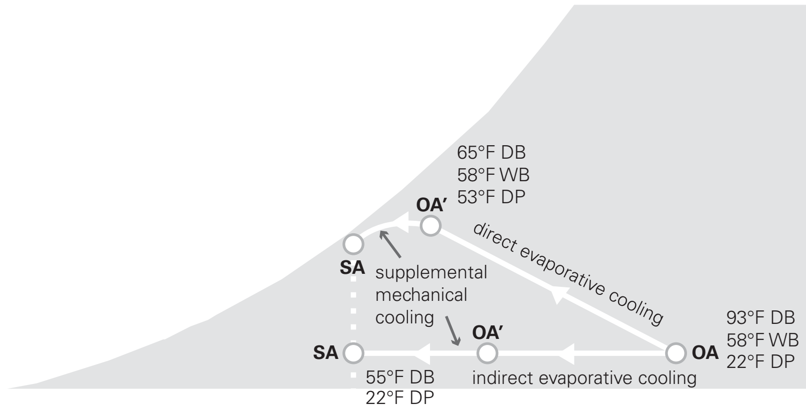
This "winterizer" configuration removes the length and pressure drop associated with a preheat coil from the main air path, resulting in a shorter air-handling unit than if a conventional preheat coil, air-mixing baffles, or an energy-recovery device is used. And, since it adds no static pressure drop to the design of the supply fan, it has less impact on fan energy than these other approaches. However, the cost of the second, smaller air-handling unit is typically higher than an air-mixing baffles, and it requires a second, smaller set of filters that need to be replaced periodically.

Evaporative cooling

Using an evaporative process to cool the air can reduce the energy used by mechanical cooling equipment. However, it 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.

Direct evaporative cooling introduces water directly into the air stream, usually with a spray or wetted media. The water evaporates as it absorbs 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 15).

Figure 15. Direct versus indirect evaporative cooling



For more information on evaporative cooling, refer to Chapter 40, "Evaporative Air-Cooling Equipment," in the 2008 *ASHRAE Handbook - HVAC Systems and Equipment* (www.ashrae.org).

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 DB, 14°C WB), and the direct evaporative process is 80 percent effective, the condition of the leaving air (OA′) will be 65°F DB and 58°F WB (18°C DB, 14°C WB).

$$DBT_{leaving} = DBT_{entering} - \text{effectiveness} \times (DBT_{entering} - WBT_{entering})$$

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

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

In a VAV system that is designed to supply air at 55°F (13°C) dry bulb, a conventional cooling coil is usually required to supplement the evaporative cooling process, and further cool the supply air to the desired setpoint (Figure 15). The system could be designed for warmer supply-air temperature and/or use an aggressive supply-air-temperature reset strategy to minimize the need for supplemental mechanical cooling, but these approaches also increase supply airflow and fan energy use.

Any cooling energy saved is offset somewhat by the increased fan energy use, as the evaporative media increases the airside pressure drop that the supply fan must overcome.

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 15). The evaporation process occurs outside the building in the cooling tower.

In some applications, indirect evaporative cooling is implemented using a stand–alone cooling tower (or similar device) and a separate coil located upstream of the conventional cooling coil. However, in a chilled-water VAV system, because a water distribution system is already part of the system, a more common approach is to add a plate-and-frame heat exchanger to the chilled-water system, allowing cool condenser water (from the cooling tower) to cool the chilled water (see Figure 78, p. 91). This configuration is often called a waterside economizer.

Heat source inside the VAV air-handling unit

Heating in a VAV system can be accomplished in several ways. While many systems include heating coils (hot water or electric) in the VAV terminal units or baseboard radiant heat installed within the zone, some systems also include a heating coil (hot water, steam, or electric) or gas-fired burner inside the air-handling unit. This centralized source of heat is primarily used to: 1) warm up the building in the morning prior to occupancy and 2) maintain the desired supply-air temperature during extremely cold weather, preventing air that is too cold from being delivered to the zones that may still require cooling (such as interior zones).

Heating coil (electric, hot water, or steam)

An electric heater, installed inside the air-handling unit at the factory, simplifies jobsite installation and avoids the need to install a boiler and hot-water (or steam) distribution system in the building or to provide gas service to the unit.

A hot-water or steam heating coil can also be mounted inside the AHU in the factory, but requires a boiler and hot-water (or steam) distribution system to be installed in the building. This approach centralizes the heating equipment and can incorporate various methods of heat recovery to reduce energy use (see "Condenser heat recovery," p. 88).

A hot-water coil that contains pure water (no antifreeze) should not be used if the coil will be exposed to air that is colder than 32°F (0°C). During normal operation, with both the circulation pump and boiler operating, the water inside the coil may not freeze until it is exposed to air that is much colder than 32°F (0°C). But if the pump fails, the stagnant water inside the tubes of the coil will be at risk of freezing, and the consequences of a frozen coil (burst tubes and water leaks) are too severe. If a hot-water coil is likely to be exposed to air that is colder than 32°F (0°C), consider one of the following methods of freeze protection:

- *Add antifreeze to the hot-water system.*
  Adding antifreeze (such as glycol) to the hot-water system lowers the temperature at which the solution will freeze. Given a sufficient concentration of glycol, no damage to the system will occur. For a VAV system, since the hot-water coil operates during sub-freezing weather, a concentration that provides "freeze protection"—to prevent the solution from forming crystals at the coldest expected outdoor temperature—is required (Table 13, p. 87). Make sure to also use an inhibitor package to help resist corrosion. At the warmer fluid temperatures used in the hot-water system, the impact of glycol on pressure drop is much lower than in cooling coils.
- *Use air-to-air energy recovery to preheat the outdoor air.*
  An air-to-air energy recovery device (such as a coil loop, heat pipe, fixed-plate heat exchanger, or wheel) can typically preheat the entering outdoor

air to a temperature warmer than 32°F (0°C), minimizing (or possibly avoiding) the risk of coil freezing (see "Air-to-Air Energy Recovery," p. 160).

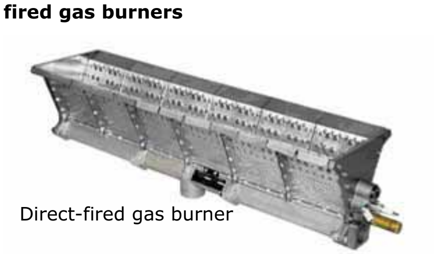
The advantage of this approach is that it also reduces cooling and heating energy use, and can allow for downsizing of cooling and heating equipment. However, such a device does increase the cost of the air-handling unit and adds a pressure drop to both the outdoor and exhaust air streams, which increases fan energy use.

Steam heating coils can also freeze if the condensate is allowed to remain inside the tubes of the coil. If a steam coil is likely to be exposed to air that is colder than 32°F (0°C):

- Use a distributing-type steam coil. This type of coil has steam "distributing" tubes inside the larger "condensing" tubes. Orifices located in the bottom of the distributing tubes are directed toward the condensate return header, improving condensate drainage.
- If possible, pitch the steam coil toward the condensate connection to assist with drainage. Also, make sure the air-handling unit is installed within the manufacturer's tolerance for levelness.
- Properly size, install, and maintain the steam trap.
- Because steam coils are very sensitive to piping practices, it is extremely important to follow the manufacturer's instructions regarding installation of the condensate piping, steam trap, and vacuum breakers.

Gas-fired burner

Figure 16. Direct-fired versus indirect-fired gas burners



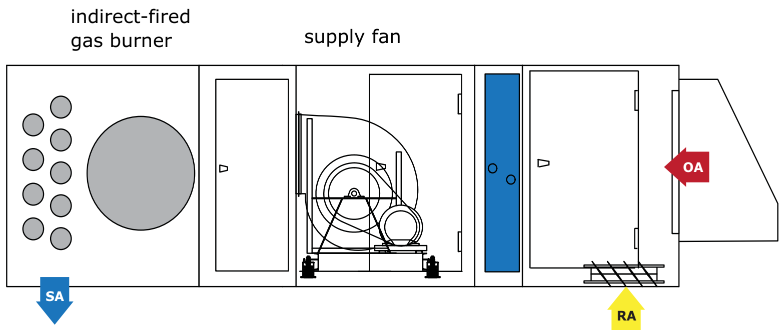
Alternatively, gas-fired burners can be installed inside the air-handling unit at the factory. This simplifies installation at the jobsite and may eliminate the need for a boiler and hot-water (or steam) distribution system in the building. In addition, gas-fired burners do not require freeze protection and may cost less to operate than an electric heater.

Direct-fired burners locate the flame directly in the air stream, while indirect-fired burners separate the combustion process from the air stream through the use of a heat exchanger (Figure 16).

Direct-fired burners are simpler (because no heat exchanger is needed) and more efficient (because there are no heat transfer losses associated with the heat exchanger) than indirect-fired burners. However, direct-fired burners introduce the products of combustion into the air stream. This requires careful control of the combustion process, but is generally considered safe in most commercial and industrial applications. However, many building codes and industry standards prevent the use of direct-fired burners for any parts of a building that contain sleeping quarters.

- *Location within the air-handling unit*
  Indirect-fired burners should be located *downstream* of the supply fan (Figure 17). In this location, the pressure inside the casing of the air-handling unit is greater than the pressure outside. This positive pressure difference reduces the likelihood that combustion gases will be drawn into the supply air stream and, therefore, into the occupied spaces.

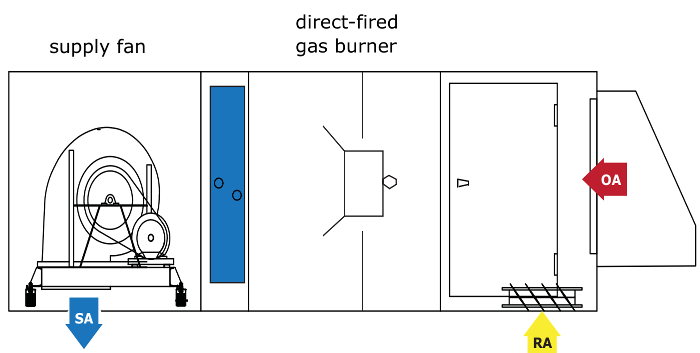
Figure 17. Indirect-fired burner located downstream of supply fan



Source: Image adapted from Trane TOPSS program

Direct-fired burners should be located *upstream* of the supply fan (Figure 18). This helps avoid unstable burner operation and nuisance trips of airflow safety switches, which are often caused by localized high velocities when the burner is located in close proximity to the fan discharge.

Figure 18. Direct-fired burner located upstream of supply fan



Source: Image adapted from Trane TOPSS program

- *Gas supply*
  The supply of natural gas connected to the gas train must be within the range of allowable pressures. A higher inlet gas pressure may require a pressure regulator, while a lower inlet pressure may require an oversized gas train. Consult the manufacturer for specific gas pressure and volume requirements.

- *Combustion gas flue stack (indirect-fired burner only)*
  For air-handling units located outdoors, the manufacturer often provides a combustion gas flue stack to be installed at the jobsite. Because the burner is located outside of the building, concerns about combustion gases are lessened.

  For units located indoors, the engineer must design the combustion gas flue stack based on heat output, horizontal and vertical lengths of the stack, type of material to be used, and all applicable codes. For long or high-pressure-drop stacks, a flue booster fan may be required. And, a barometric damper may be required if a tall chimney produces excessive draft.

- *Controls*
  Because airflow across the gas-fired burner varies, the heating capacity of the burner must be modulated to prevent the temperature rise through the burner from exceeding the maximum allowable limit. For most VAV applications, this likely requires a burner with a 10:1 turndown ratio. This means the burner can operate at a capacity as low as 10 percent of its rated capacity. In addition, air temperature sensors on the entering and leaving sides of the burner are needed to protect it from damage.

  Finally, allow the supply fan to continue to operate for a period of time after the gas-fired burner has been shut off, allowing the heat exchanger to dissipate any residual heat. Consult the manufacturer for the length of this "cool-down" period.

  In VAV applications, the air velocity across a direct-fired burner must remain within a specific range for safe operation. In this case, manufacturers typically provide an adjustable opening that automatically varies the opening size as airflow changes, keeping the air velocity relatively constant.

Recovered heat

In some applications, heat may be recovered from another part of the HVAC system. Common sources of recoverable heat include:

- Warm condenser water leaving a water-cooled chiller or hot refrigerant vapor leaving the compressor in an air-cooled chiller (see "Condenser heat recovery," p. 88)
- Another air stream, or another location in the same air stream, using an air-to-air heat exchanger (see "Air-to-Air Energy Recovery," p. 160)

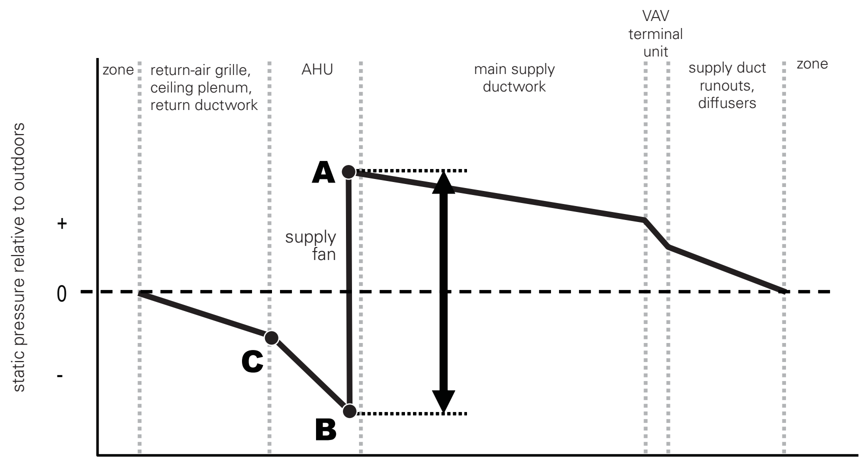
Fans (supply, return, relief)

Fans are used to move air throughout the various components of a VAV system. Depending on the application, the system may include: 1) a supply fan only, 2) a supply fan and a relief (or exhaust) fan, or 3) a supply fan and a return fan.

Supply fan only

In this configuration (Figure 19), the supply fan must create high enough pressure at its outlet (A) to overcome the pressure losses associated with pushing the air through the main supply ductwork, VAV terminal units, supply-duct runouts, and supply-air diffusers. *(Note: If the system uses series fan-powered VAV terminals, the small terminal fan is used to overcome the pressure losses between the terminal unit and the zone. For further discussion, see "Fan-powered VAV terminal units," p. 58.)*

Figure 19. VAV system with supply fan only (100% recirculated air)



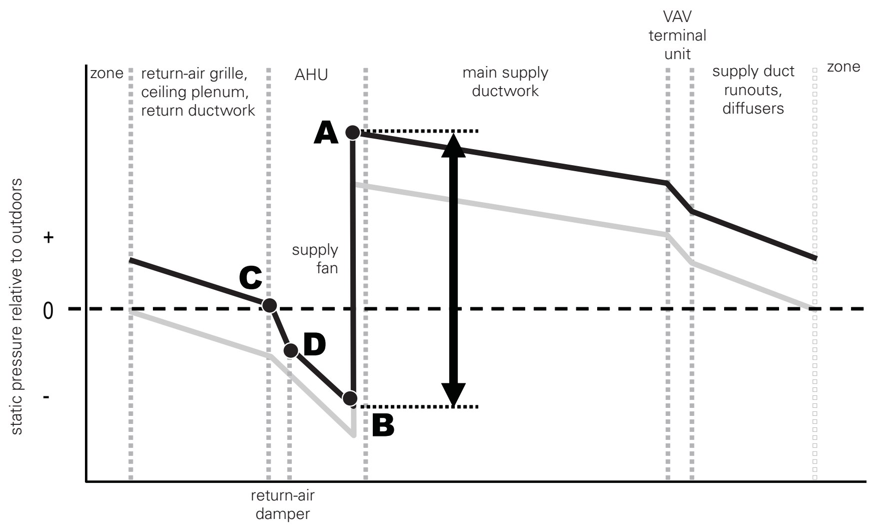
In addition, the supply fan must create low enough pressure at its inlet (B) to overcome the pressure losses associated with drawing the return air out of the zones and through the return-air grilles, through the open ceiling plenum and/or return ductwork, and then through the return-air damper, filter, and coils inside the air-handling unit.

Figure 19 depicts a typical “supply fan only” system operating with 100 percent recirculated air, as it might operate during unoccupied periods or morning warm-up (no outdoor air is being brought into the building). Due to the pressure drop through the return-air path, the pressure at the inlet to the air-handling unit (C) is lower than the ambient pressure. With this negative pressure differential, no air will be forced out of the relief damper.

With this “supply fan only” configuration, the only way for any air to leave the building (which is required if outdoor air is to be brought into the building) is for the pressure in the zone and return-air path to increase.

Figure 20 depicts this same system operating with 25 percent outdoor air. The pressure in the zone and return-air path has increased to the point that the pressure at the inlet to the air-handling unit (C) is now higher than the ambient pressure. This positive pressure differential will force air out through the relief damper.

Figure 20. VAV system with supply fan only (25% outdoor air)



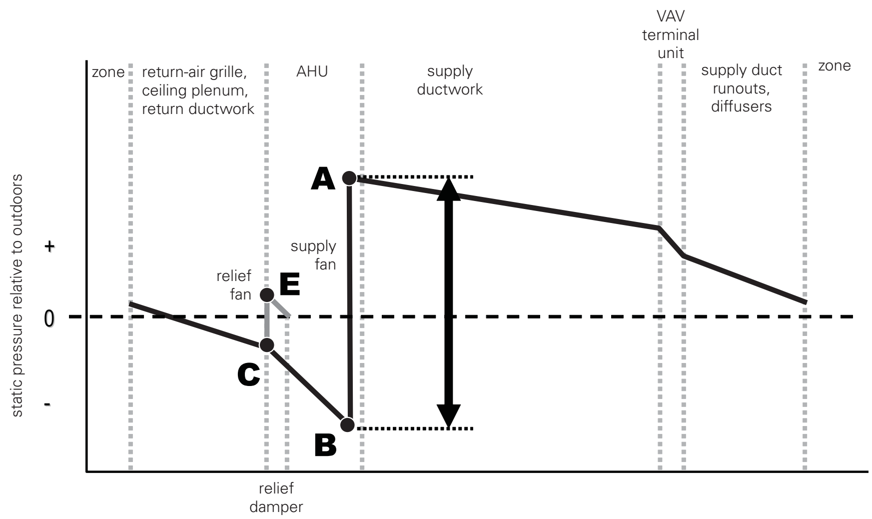
However, the pressure inside the mixing box (D) must still be lower than the ambient pressure in order for air to be drawn in through the outdoor-air damper. So the return-air damper must be closed far enough to create the necessary pressure drop (from C to D). This requires the supply fan to generate a larger pressure differential (inlet to outlet, from B to A) in order to deliver the desired supply airflow and bring in the required amount of outdoor air.

It also results in a higher pressure in the zone, which could cause doors to stand open without latching. For proper control of building pressure, this “supply fan only” configuration should usually be avoided and either a relief fan or return fan should be used.

Supply fan and relief fan

As an alternative, a relief fan can be added to the system. In this configuration (Figure 21), the supply fan must still create a high enough pressure at its outlet (A) to overcome the pressure losses associated with the supply-air path, and create a low enough pressure at its inlet (B) to overcome the pressure losses associated with the return-air path and the components inside the air-handling unit.

Figure 21. VAV system with supply and relief fans (25% outdoor air)



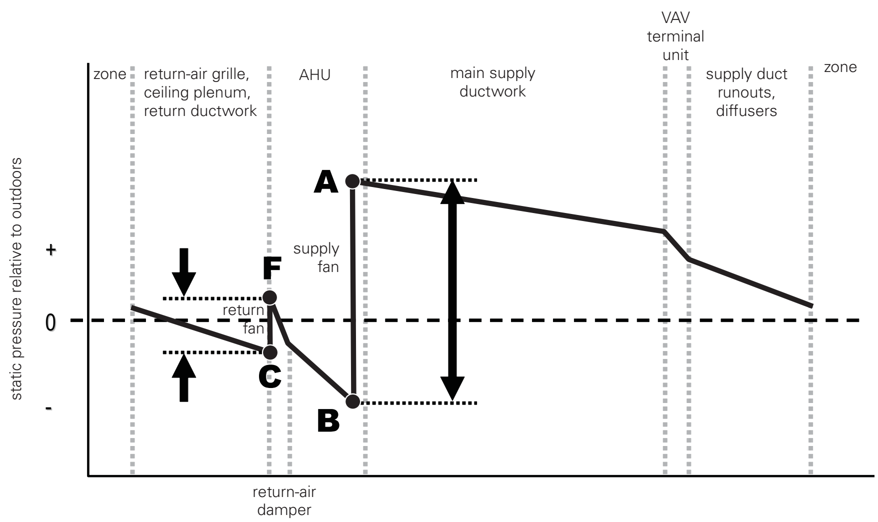
However, the relief fan is used to raise the pressure of the air to be exhausted (from C to E) so that it is high enough to overcome the pressure loss associated with the relief damper, and force the excess air out of the building. Adding the relief fan allows the system to exhaust the air that is to be replaced by fresh, outdoor air, and does so without increasing the pressure in the zone or requiring the supply fan to generate a larger pressure differential.

In smaller VAV systems that do not use an airside economizer cycle, local exhaust fans (serving restrooms or copy centers) and local barometric relief dampers may exhaust enough air without the need for a central relief fan.

Supply fan and return fan

The final configuration uses a fan in the return-air path, rather than in the exhaust air path. When the system includes a return fan, the supply fan must still create the same pressure at its outlet (A) to overcome the pressure losses associated with the supply-air path (Figure 22).

Figure 22. VAV system with supply and return fans (25% outdoor air)



However, the pressure at the inlet of the supply fan (B) only needs to be low enough to overcome the pressure losses associated with drawing the return air through the return-air damper, filter, and coils (or the pressure associated with drawing the outdoor air through the outdoor-air damper, whichever is higher). The return fan is used to overcome the pressure losses associated with drawing the return air out of the zones and through the return-air grilles, open ceiling plenum and/or return ductwork. It must also generate enough pressure (F) to push any relief air out through the relief damper.

Should the system use a relief fan or a return fan?

The "supply fan and relief fan" configuration usually works best in VAV systems that use an open ceiling plenum for part of the return-air path (Table 5). Systems with relief fans are easier to control, typically lower the cost of the air-handling unit, and are often less costly to operate than systems with return fans. (See "Building pressure control," p. 178.)

When the pressure drop through the return-air path is very high (which may be the case in a larger system with a fully ducted return-air path), evaluate both the relief- and the return-fan configurations. If the supply fan is capable of handling the pressure drop of both the supply- and return-air paths, then the relief-fan configuration is preferred for the reasons mentioned above. Use a return fan only if the return-air path adds more pressure drop than the supply fan can handle.

Table 5. Return fan versus relief fan in a VAV system

Return Fan	
Advantages:	**Disadvantages:**
• Lower differential pressure across the supply fan, if the pressure drop of the return air path is greater than the pressure drop of the outdoor air path (which is likely in most VAV systems). • Potentially lower installed cost for a system with a fully ducted, return-air path. A smaller differential pressure across the supply fan can result in a smaller fan motor and variable-speed drive.	• Higher operating costs, especially in applications with extended hours of economizer cooling[1] (the return fan must run whenever the supply fan operates). • Potential for air to leak out through the relief damper, because the return-air plenum inside the air-handling unit operates at a positive pressure (F in Figure 22). *Note: Using low-leak relief dampers can minimize air leakage to the outdoors.* • More complex (expensive) fan-speed control. Controlling the pressure in the return-air plenum inside the air-handling unit requires an additional pressure sensor and modulating device (either a damper actuator or variable-speed drive), and the pressure sensor is difficult to situate because this plenum is usually small and turbulent. (See "Return-fan capacity control," p. 181.) • Requires more fan power at part load. The pressure in the return-air plenum inside the air-handling unit must always be high (positive) enough to force air out through the relief damper. The return-air damper must therefore create a significant pressure drop between the positive return-air plenum and the negative mixed-air plenum. This added pressure drop requires more combined fan power than a system with a relief fan. • Limited layout flexibility. The return fan must be situated between the air-handling unit and the closest leg of the return-air path (usually near the air-handling unit) because it must draw the entire return path negative relative to the occupied spaces. It must also discharge into the return-air plenum during modulated economizer operation.
Relief fan	
Advantages	**Disadvantages**
• Lower operating cost. In some applications, the relief fan can remain off during "non-economizer" hours[1] and operate at low airflow during many "economizer" hours. Also, the return-air damper can be sized for a lower pressure drop. • Simpler control scheme. One less sensor and one less actuator simplifies installation and air balancing. Applications with a relatively low pressure drop through the return-air path (which is common in a system that uses an open ceiling plenum for part of the return-air path) can use lower-cost fans as well as fewer (less costly) controls. • Greater layout flexibility. The relief fan can be positioned anywhere in the return-air path because the supply fan draws the return path negative (relative to the occupied spaces) during modulated economizer operation. An air-handling unit installed in the basement with a central relief fan installed on the roof can take advantage of winter stack effect to lower operating cost.	• Potential for negative building pressure at low loads. This condition can occur when a variable-speed drive controls the relief fan, supply airflow is very low, and required relief airflow is less than is delivered with the relief fan operating at lowest speed. *Note: Using a constant-speed relief fan with a modulating relief damper avoids this problem.* • Potential for air to leak in through the relief damper because the return-air plenum inside the air-handling unit operates at negative pressure whenever the relief fan is turned off (C in Figure 21). *Note: Using low-leak relief dampers can minimize air leakage from outdoors.* • Higher differential pressure across the supply fan than in a system with a return fan. The supply fan must overcome the pressure drop of the return path as well as the supply path. For this reason, an air-handling unit with a relief fan may not be capable of delivering the pressure differential required in a system with a fully ducted return-air path.

1 In many applications, when the system is bringing in minimum ventilation airflow (not in airside economizer mode), local exhaust fans (in restrooms and copy centers, for example) and exfiltration (due to positive building pressurization) are often sufficient to relieve all the air brought into the building for ventilation. In this case, no central relief is needed and the central relief fan can be turned off, only needing to operate during economizer mode.

For more information on fan types, refer to the Trane *Air Conditioning Clinic* titled, "Air Conditioning Fans" (TRG-TRC013-EN).

Fan types

VAV air-handling units are typically available with several choices for fan types and sizes. This affords the opportunity to select a fan that optimizes the balance of energy efficiency, acoustics, and cost.

The most common type of fan used in VAV systems is a centrifugal fan, in which air enters the center of the fan wheel (axially) and follows a radial path through it. A centrifugal fan may be characterized by the shape of the fan blades, whether it contains a fan scroll (housed) or not (plenum), whether it is belt-driven or direct-drive, and whether one or multiple fan wheels are used.

- *Shape of fan blades*
  A forward curved (FC) fan has blades that are curved in the direction of wheel rotation. These fans are operated at relatively low speeds and are used to deliver large volumes of air against relatively low static pressures. Due to the inherently light construction of the fan wheel, FC fans are typically used in smaller systems that require static pressures of 4 in. H_2O (1000 Pa) or less. FC fans are typically less costly, but usually less efficient, than the other types.

  Systems that require greater than 3 in. H_2O (750 Pa) of static pressure are usually best served by the more-efficient backward curved (BC), backward inclined (BI), or airfoil-shaped (AF) fan blades. In larger systems, the higher fan efficiencies can result in significant energy savings.

  A BI fan has flat blades that are slanted away from the direction of wheel rotation, while a BC fan has shaped blades that are curved away from the direction of wheel rotation. Their rugged construction allows them to operate at higher speeds than FC fans, and makes them suitable for moving large volumes of air in higher static-pressure applications. Also, BC and BI fans are typically more efficient than FC fans.

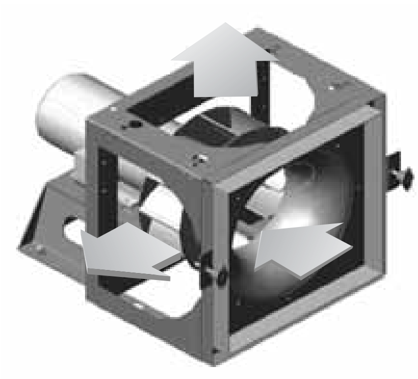
  A refinement of the BI fan changes the shape of the blade from a flat plate to that of an airfoil, similar to an airplane wing. The smooth airflow across the blade surface reduces turbulence and noise within the wheel. The result is that an AF fan typically requires less input power than other fan types (Table 6).

Table 6. Impact of fan type on input power (brake horsepower)[1]

Fan type and size	Input power, bhp (kW)	Rotational speed, rpm
Housed FC, 22.375 in. (568 mm)	16.07 (11.98)	979
Housed AF, 25 in. (635 mm)	14.65 (10.92)	1433
Belt-drive plenum AF, 35.56 in. (903 mm)	16.15 (12.04)	1108
Direct-drive plenum AF, 30 in. (762 mm)	14.85 (11.07)	1388

1 Based on a typical VAV air-handling unit configuration (OA/RA mixing box, high-efficiency filter, hot-water heating coil, chilled-water cooling coil, and draw-thru supply fan with a single discharge opening off the fan section) operating at 13,000 cfm (6.1 m^3/s) and 3 in. H_2O (750 Pa) of external static pressure drop.

Figure 23. Direct-drive plenum fan

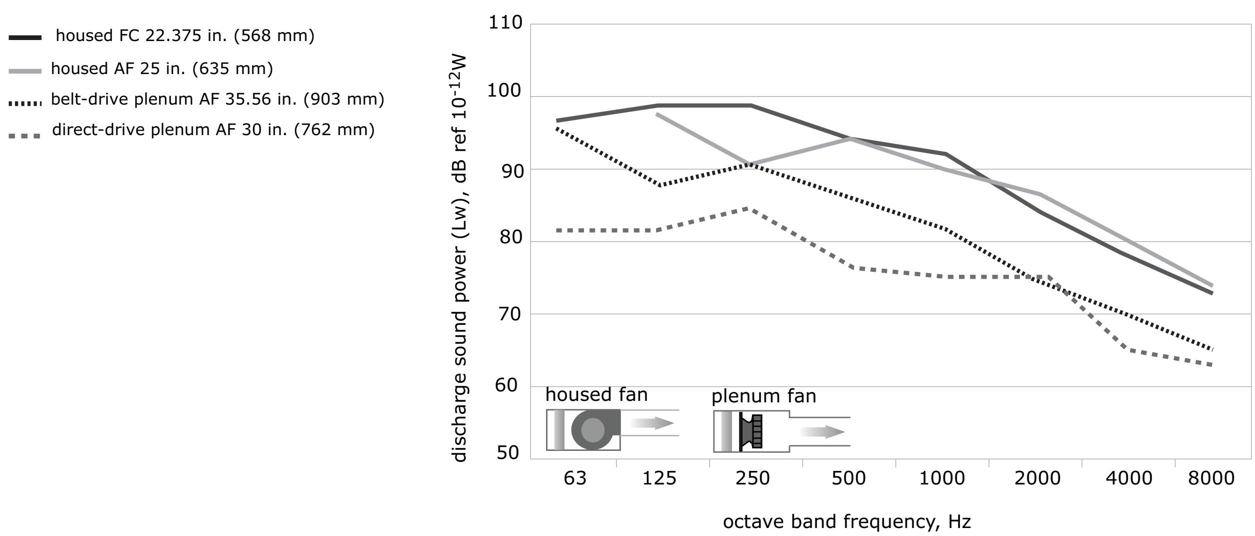


- *Housed versus plenum*
  A variation of the centrifugal fan is a plenum fan. This type of fan consists of an unhoused centrifugal fan wheel with an inlet cone and typically airfoil fan blades (Figure 23). The fan wheel pressurizes the plenum surrounding the fan, allowing the air to discharge in multiple directions.

  Depending on the configuration of the fan inside the air-handling unit, a plenum fan may be more or less efficient than a housed fan. A housed fan is specifically designed to discharge into a straight section of ductwork, which minimizes losses as velocity pressure is converted to static pressure. In the configuration used in this example (a single front discharge opening off the fan section; see diagrams in Figure 24), the input power for the housed AF fan is lower than for either of the plenum fans (Table 6).

  However, a plenum fan typically has lower discharge sound levels than a housed fan (Figure 24). The reduced sound levels occur because air velocity dissipates more quickly as the air pressurizes the plenum surrounding the fan and because the plenum provides an opportunity for some of the sound to be absorbed before the air discharges from the air-handling unit.

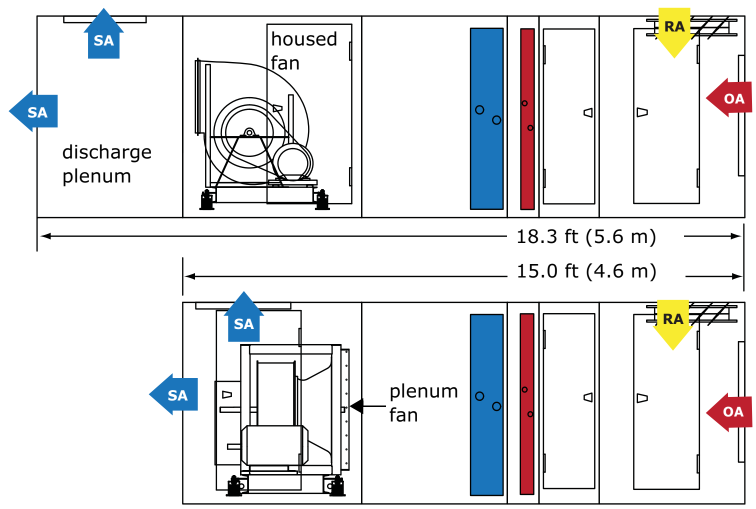
Figure 24. Plenum fan can reduce discharge sound levels[1]



1 Based on a typical VAV air-handling unit configuration with a single (front-top) discharge opening, operating at 13,000 cfm (6.1 m^3/s) and 3 in. H_2O (750 Pa) of external static pressure drop.

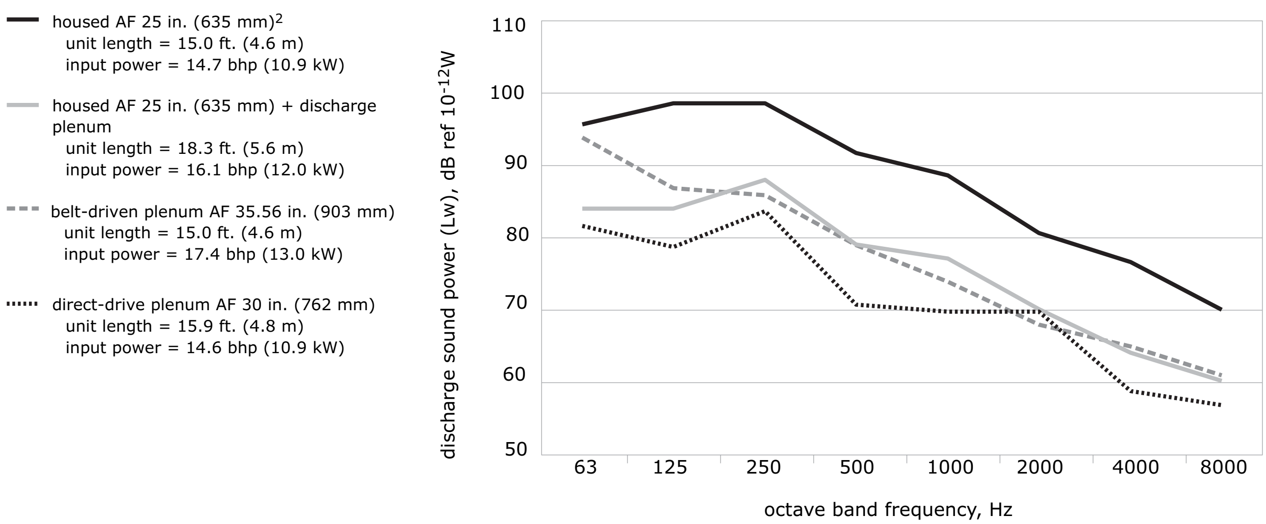
To allow a housed fan to achieve similar discharge sound levels, a discharge plenum can be added to the air-handling unit. While this plenum helps reduce discharge sound levels, it increases fan input power and increases the overall length of the unit (Figure 25 and Figure 26). In this example, when a discharge plenum is added to the housed fan, the input power for the housed AF fan is higher than for the direct-drive plenum fan (Figure 26).

Figure 25. Plenum fan can reduce overall AHU length



Source: Images from Trane TOPSS program

Figure 26. Discharge sound levels with multiple discharge connections[1,2]



1 Based on a typical VAV air-handling unit configuration with multiple discharge openings (Figure 25), operating at 13,000 cfm (6.1 m^3/s) and 3 in. H_2O (750 Pa) of external static pressure drop.

2 Discharge sound power for a housed AF fan with a single (front-top) discharge opening (Figure 24), included here to demonstrate the sound reduction provided by the discharge plenum.

When space is a prime consideration, and multiple supply duct connections are desired, a housed fan requires a discharge plenum to allow for the multiple connections. If a plenum fan is used, however, multiple duct connections can be made to the fan module itself, eliminating the need for a discharge plenum, and resulting in a shorter air-handling unit (Figure 25).

- *Belt-driven versus direct-drive*
  Historically, most large fans used in VAV systems were belt-driven. However, with the increased use of VFDs, direct-drive fans have become popular, primarily with plenum fans. With a direct-drive plenum fan, the fan wheel is mounted directly on the motor shaft, rather than using a belt and sheaves (Figure 23). Because there are no belts or sheaves, and fewer bearings, direct-drive fans are more reliable and require less maintenance. In addition, there are no belt-related drive losses, so direct-drive fans are typically more efficient, quieter, and experience less vibration (Table 6, Figure 24, and Figure 26).

  However, the air-handling unit may be slightly longer since the fan motor is mounted at end of the shaft.

- *Single versus multiple fans*
  Most fans in VAV systems use a single fan wheel. However, using multiple fans can shorten the length of the AHU. This is often referred to as a fan array (Figure 27).

  For a given airflow, a unit with multiple fans uses several, smaller-diameter fan wheels, rather than a single, larger-diameter fan wheel. The distance (length) required both upstream and downstream of the fan is typically a function of the fan wheel diameter. Therefore, using multiple, smaller-diameter fan wheels can shorten the required upstream and downstream spacing required, and can shorten the overall length of the air-handling unit.

  While the overall length can be reduced significantly by changing from one to two fans, the potential length reduction diminishes as the number of fans increases. When more than four to six fan wheels are used, the upstream and downstream spacing requirements begin to be dictated by the need for access rather than by fan wheel diameter, so there is generally little further length reduction benefit.

  Another benefit of using multiple fans is redundancy. If one fan fails, another fan is available to compensate. In most cases, three or four fans are able to provide the required level of redundancy.

  However, using multiple fans is typically less efficient and increases the cost of the air-handling unit.

For more information on direct-drive fans and using multiple versus single fans, refer to the Trane engineering bulletin titled "Direct-Drive Plenum Fans for Trane Climate Changer™ Air Handlers" (CLCH-PRB021-EN).

Figure 27. Fan array with direct-drive plenum fans

upstream (inlet) side

downstream (outlet) side

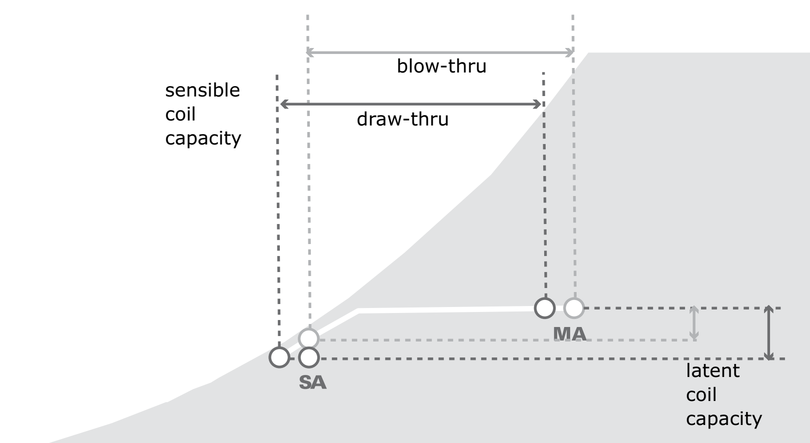
Blow-thru versus draw-thru?

In a blow-thru configuration, the fan blows air through a cooling coil located downstream of the fan (Table 7, p. 37). The heat generated by the fan and motor is added to the air upstream of the cooling coil.

In a draw-thru configuration, the fan draws air through a cooling coil located upstream. The heat generated by the fan and motor is added to the air downstream of the cooling coil.

Assuming equivalent supply airflows, in a draw-thru configuration the air must leave the cooling coil at a colder temperature in order to achieve the same supply-air temperature delivered down the duct. Because the fan heat gain is equal, the sensible cooling capacity required is the same for both configurations (Figure 28). However, because the cooling coil in the draw-thru configuration must make the air colder (since the fan heat is added downstream of the coil), it also makes the air drier (in non-arid climates). This increases the latent cooling capacity. Therefore, achieving the same 55°F (13°C) supply-air temperature with the draw-thru configuration requires slightly more total cooling capacity, but offers the benefit of slightly drier air delivered to the zones.

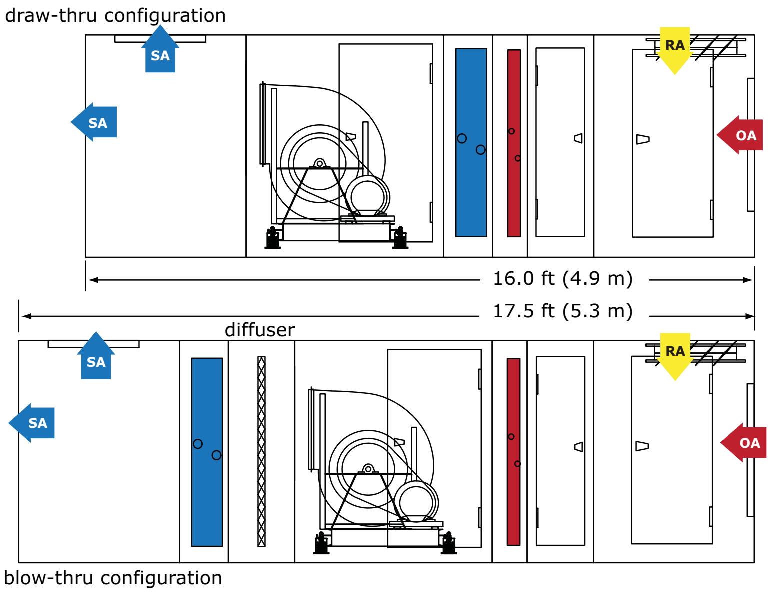
Figure 28. Effect of fan heat gain on cooling capacity (equal supply airflows)



Of course, this colder air leaving the coil in a draw-thru configuration can also be of benefit. At equivalent supply-air temperatures, the draw-thru configuration delivers the supply air at a lower dew point (Figure 28), which improves dehumidification performance.

In a blow-thru configuration, the air-handling unit typically needs to be longer to avoid uneven velocities as the air passes through the cooling coil. Alternatively, a diffuser (set of baffles) can be used to provide even airflow across components downstream of the fan and minimize additional length. In the example shown in Figure 29, the diffuser section increases the overall length of the air-handling unit by 1.5 ft. (0.4 m), or 8 percent.

Figure 29. Effect of draw-thru versus blow-thru (with diffuser) on AHU length



Source: Images from Trane TOPSS program

Note: When using a blow-thru configuration, consider using a plenum fan. The required distance between the fan and cooling coil is much shorter and a diffuser is not needed. In addition, this avoids the negative impact of the abrupt discharge on the performance of a housed fan.

Finally, systems that use the blow-thru configuration have often experienced problems with final filters getting wet. Experience indicates that this problem can often be minimized by using a draw-thru supply fan or by adding a few degrees of heat to the air before it passes through the final filters.

Table 7. Blow-thru versus draw-thru configuration

Blow-thru	
Advantages:	**Disadvantages:**
• Heat generated by the fan and motor is added to the air *upstream* of the cooling coil, allowing for a warmer leaving-coil temperature to achieve a desired supply-air temperature. • Locating the fan upstream of the cooling coil often lowers the discharge sound levels slightly, but also raises inlet sound levels.	• Often results in a longer AHU, or the use of a diffuser section, to develop an acceptable velocity profile for air passing through the cooling coil (Figure 29). • If final filters are used, this configuration often results in problems with final filters getting wet (Figure 38, p. 45). • Greater concern with air leaking out of the AHU since more of the casing is pressurized. (See "Air leakage," p. 51.)
Draw-thru	
Advantages:	**Disadvantages:**
• Typically results in a shorter AHU since less distance is needed between the upstream cooling coil and the fan (Figure 29). • Colder leaving-coil temperature results in increased dehumidification capacity. • Heat added by the supply fan (located between the cooling coil and final filters) typically prevents final filters from getting wet. • Less concern with air leaking out of the AHU since less of the casing is pressurized. (See "Air leakage," p. 51.)	• Heat generated by fan and motor is added to the air *downstream* of the cooling coil, requiring a colder leaving-coil temperature to achieve a desired supply-air temperature. • Proper condensate trapping is critical to avoid wetting the interior of the AHU, because the drain pan is under a negative pressure.

> For more information on fan performance curves, system resistance curves, and VAV fan modulation, refer to the Trane *Air Conditioning Clinic* titled "Air Conditioning Fans" (TRG-TRC013-EN).

Supply fan capacity modulation

To accommodate variable airflow in a VAV system, the supply fan must be selected and controlled so that it is capable of modulating over the required airflow range.

The pressure drop through ducts, fittings, coils, filters, and so forth change as airflow varies. The capacity of the supply fan must be modulated to generate sufficient static pressure to offset these pressure losses, and provide the minimum pressure required for proper operation of the VAV terminal units and supply-air diffusers at all airflows.

To achieve this balance, a simple control loop is used (Figure 30). A pressure sensor measures the static pressure at a particular location in the duct system. A controller compares this static pressure reading to a setpoint, and the supply fan capacity is modulated to generate enough static pressure to maintain the desired pressure setpoint at the location of the sensor.

Figure 30. Supply fan capacity control

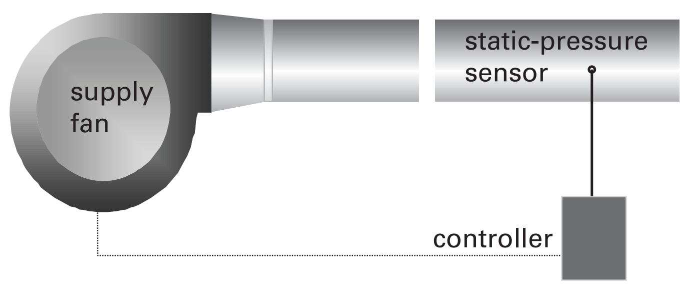
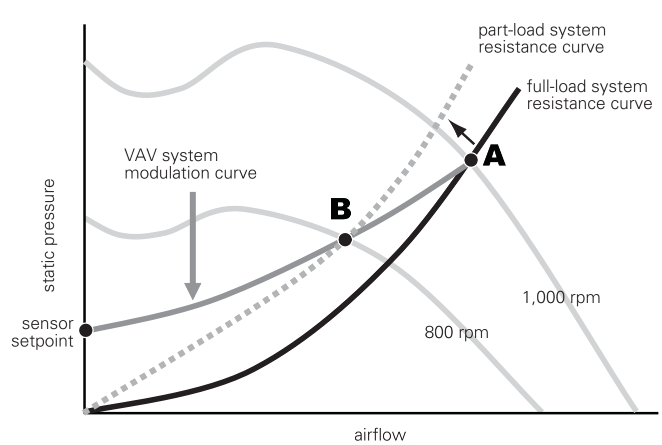


Figure 31 depicts an exaggerated example to illustrate this control loop. As the cooling loads in the zones decrease, the dampers in all or most of the VAV terminal units modulate toward a closed position. This added restriction increases the pressure drop through the system, reducing supply airflow and causing the (part-load) system resistance curve to shift upwards.

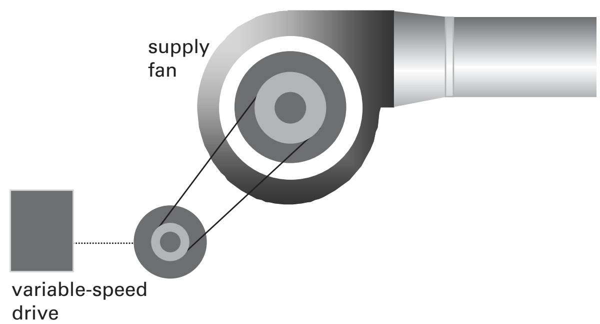
Figure 31. VAV system modulation curve



In response, the fan begins to "ride up" the constant-speed (rpm) performance curve, from the design operating point (A), trying to balance with this new system resistance curve. As a result, the fan delivers less airflow at a higher static pressure. The static pressure controller senses this higher pressure and sends a signal to reduce the capacity of the supply fan. Modulating the fan capacity shifts the performance curve of the fan downward until the system balances at an operating point (B) that brings the system static pressure back down to the setpoint. This response, over the range of system supply airflows, causes the supply fan to modulate along the *VAV system modulation curve*.

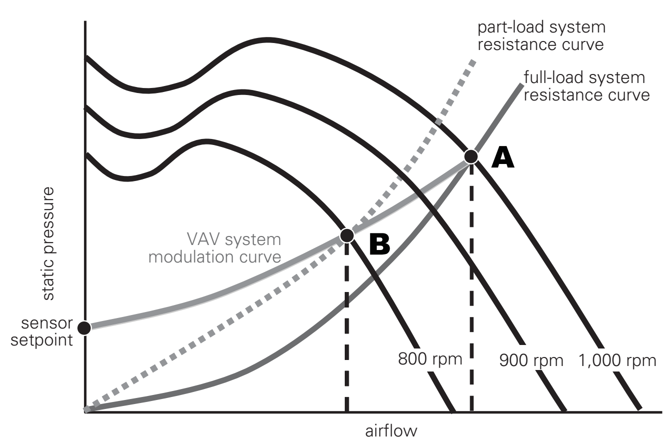
The most common method used to modulate supply fan capacity in a VAV system is to vary the speed at which the fan wheel rotates. This is commonly accomplished using a variable-speed drive (or variable-frequency drive, VFD) on the fan motor (Figure 32).

Figure 32. Fan-speed control



When the system static pressure controller sends a signal to reduce fan capacity, the variable-speed drive reduces the speed at which the fan wheel rotates. Reducing fan speed (rpm) shifts the performance curve of the fan downward, until the system balances at an operating point (B) along the VAV system modulation curve, bringing the system static pressure back down to the setpoint (Figure 33).

Figure 33. Performance of fan-speed control in a VAV system



The modulation range of the supply fan is limited by how far the variable-speed drive can be turned down (typically 30 to 40 percent of design airflow).

Air cleaning

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 coils cleaner, for example) and keeps the air distribution system relatively clean. Some of the contaminants that affect indoor air quality can be classified as particulates, gases, or biologicals. The methods and technologies for effectively controlling these contaminants differ, so it is important to define the contaminants of concern for a given facility.

Particulate filters

For more information on the various types of particulate filters, refer to Chapter 28, "Air Cleaners for Particulate Contaminants," in the 2008 *ASHRAE Handbook—HVAC Systems and Equipment* (www.ashrae.org) or the *NAFA Guide to Air Filtration* (www.nafahq.org).

Particulate matter ("particulates") describes a broad class of airborne chemical and physical contaminants that exist as discrete grains or particles. Common particulates include pollen, tobacco smoke, skin flakes, and fine dust.

Airborne particulates vary in size, ranging from submicron to 100 microns (µm) and larger (Figure 34). Many types of particle filters are available (Figure 35). 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.

Figure 34. Typical particle sizes

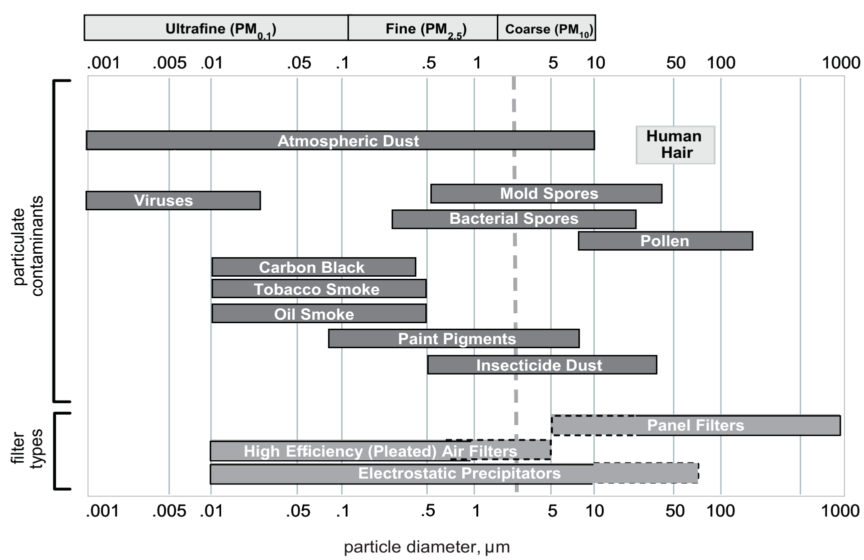


Figure 35. Types of particulate media filters

Images used by permission from CLARCOR Air Filtration Products (www.clcair.com)

Particulate filter efficiency is typically expressed in terms of "dust-spot efficiency" or "minimum efficiency reporting value" (MERV). *Dust-spot efficiency* is defined by ASHRAE Standard 52.1 and relates to the amount of atmospheric dust that a filter captures. ASHRAE 52.1 is now obsolete and has been replaced by ASHRAE Standard 52.2. The *minimum efficiency reporting value (MERV)* is defined by ASHRAE 52.2, and relates to how efficiently a filter removes particles of various sizes, from 0.3 to 1 micron.

Table 8 identifies common types of particulate filters and their typical applications. It also approximates equivalent dust-spot efficiencies for the various MERV levels.

Table 8. Applications guidelines for various filter types

Collection efficiency[1]	Dust-spot efficiency	Typical controlled contaminant	Typical applications and limitations	Typical air filter/cleaner type
IEST Type F (≥ 99.999% on 0.1 to 0.2 µm particles) IEST Type D (≥ 99.999% on 0.3 µm particles) IEST Type C (≥ 99.99% on 0.3 µm particles) IEST Type A (≥ 99.97% on 0.3 µm particles)	n/a n/a n/a n/a	≤ 0.30 µm particles • Virus (unattached) • Carbon dust • Sea salt • All combustion smoke • Radon progeny	• Cleanrooms • Radioactive materials • Pharmaceutical manufacturing • Carcinogenic materials • Orthopedic surgery	**HEPA/ULPA filters**
MERV 16 MERV 15 MERV 14 MERV 13	n/a >95% 90% to 95% 80% to 90%	0.3 to 1.0 µm particles • All bacteria • Most tobacco smoke • Droplet nuclei (sneeze) • Cooking oil • Most smoke • Insecticide dust • Copier toner • Most face powder • Most paint pigments	• Hospital inpatient care • General surgery • Smoking lounges • Superior commercial buildings	**Bag filters** Nonsupported (flexible) microfine fiberglass or synthetic media, 12 to 36 in. deep, 6 to 12 pockets **Box filters** Rigid style cartridge filters, 6 to 12 in. deep, may use lofted (air-laid) or paper (wet-laid) media
MERV 12 MERV 11 MERV 10 MERV 9	70% to 75% 60% to 65% 50% to 55% 40% to 45%	1.0 to 3.0 µm particles • Legionella • Humidifier dust • Lead dust • Milled flour • Coal dust • Auto emissions • Nebulizer drops • Welding fumes	• Superior residential buildings • Better commercial buildings • Hospital laboratories	**Bag filters** Nonsupported (flexible) microfine fiberglass or synthetic media, 12 to 36 in. deep, 6 to 12 pockets **Box filters** Rigid style cartridge filters, 6 to 12 in. deep, may use lofted (air-laid) or paper (wet-laid) media
MERV 8 MERV 7 MERV 6 MERV 5	30% to 35% 25% to 30% <20% <20%	3.0 to 10.0 µm particles • Mold • Spores • Hair spray • Fabric protector • Dusting aids • Cement dust • Pudding mix • Snuff • Powdered milk	• Commercial buildings • Better residential buildings • Industrial workplaces • Paint booth inlet air	**Pleated filters** Disposable, extended surface, 1 to 5 in. thick with cotton/polyester blend media, cardboard frame **Cartridge filters** Graded-density viscous-coated cube or pocket filters, synthetic media **Throwaway** Disposable, synthetic media panel filters
MERV 4 MERV 3 MERV 2 MERV 1	<20% <20% <20% <20%	> 10.0 µm particles • Pollen • Spanish moss • Dust mites • Sanding dust • Spray paint dust • Textile fibers • Carpet fibers	• Minimum filtration • Residential buildings • Window air conditioners	**Throwaway** Disposable, fiberglass or synthetic panel filters **Washable** Aluminum mesh, latex-coated animal hair, or foam rubber panel filters **Passive electrostatic (electret)** Self-charging (passive) woven polycarbonate panel filter

1 Minimum Efficiency Reporting Value (MERV) is defined by ANSI/ASHRAE Standard 52.2-1999, *Method of Testing General Ventilation Air-Cleaning Devices for Removal Efficiency by Particle Size*. HEPA/ULPA classifications are defined by IEST-RP-CC001.4, *HEPA and ULPA Filters*.

Source: 2008 *ASHRAE Handbook—HVAC Systems and Equipment*, Chapter 28, Table 3. © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org.

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 (Figure 34), it is important to define the contaminants of concern for a given facility when selecting the type of filter to be used.
- *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 velocity through a media filter results in a higher static pressure drop.

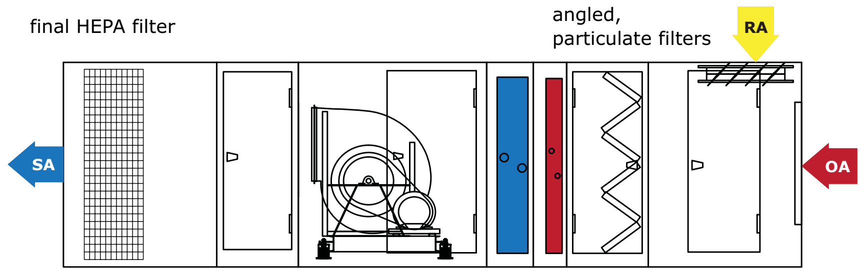
  One common method of reducing airside pressure drop is to angle the filters within the air-handling unit (Figure 36). This decreases air velocity through the media and increases the surface area of the media, which increases its dirt-holding capacity.

Figure 36. Angled filters increase surface area and lower airside 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 higher efficiencies, lower airside pressure drop, and/or greater dirt-holding capacity, require more space than filters that perform more poorly in these categories. This can impact the physical size of the air-handling unit.
- *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.

In a typical VAV system, the mixture of outdoor air and recirculated return air passes through a particulate filter to remove airborne particulate contaminants. Locating these filters upstream of the heating and cooling coils (Figure 37) helps keep the coils cleaner for a longer period of time, and allows the system to operate more efficiently.

Figure 37. Particulate filters in a VAV air-handling unit



Source: Image adapted from Trane TOPSS program

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.

An addendum (c) to ASHRAE Standard 62.1-2007 requires a MERV 11 filter 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 2.5 microns or less (PM2.5).

One of the requirements for earning the "Indoor Chemical and Pollutant Source Control" credit (Indoor Environmental Quality section) of LEED 2009 is to install a MERV 13 (or higher) filter to clean both return and outside air that is delivered as supply air.

ASHRAE Standard 62.1-2007 (Section 5.9) requires that a filter with a MERV rating of at least 6 be installed upstream of all wet surfaces, including cooling coils. *Note: ASHRAE 62.1 (Section 6.2.1) also requires a MERV 6 filter 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).* In general, this requirement can be met with most standard "throwaway" or "pleated" filters (see Table 8). However, when a higher level of filtration is desired, bag or cartridge filters are sometimes used (Figure 35, p. 41). Bag or cartridge filters typically cost more than pleated filters, which may make them a good candidate for using a pre-filter to extend their useful life.

In order to maintain the desired level of cleanliness and minimize system energy use, never operate the air-handling unit without the filters in place, especially during construction of the building. Filters used during construction should be replaced prior to building occupancy.

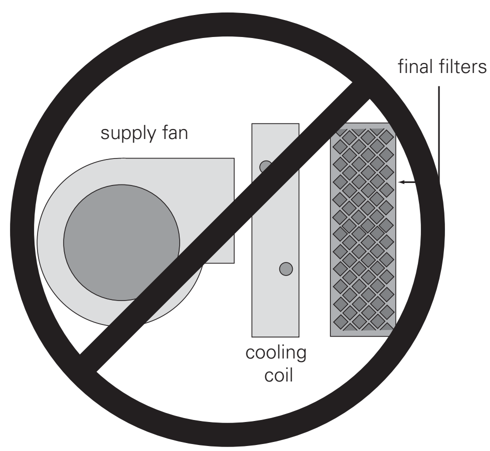
It is important to maintain and replace filters as recommended by the manufacturer. Typically filters are replaced based on the length of time in operation or on the measured air pressure drop across the filter. A differential pressure sensor can be used to indicate when the filters are dirty (the pressure drop exceeds a pre-determined limit) and can send an alarm to the building automation system to indicate that they need to be replaced. Filter manufacturers typically publish a "dirty" (or final) pressure drop. However, this is sometimes overridden by the air-handling equipment manufacturer based on the ability of the fan to overcome this pressure drop. The replacement filters should have similar performance characteristics as the filters originally specified by the design engineer. Critical characteristics include efficiency (MERV rating, for example), 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 tape or gasketing, for example) to minimize airflow through the space between adjacent filters.

Some applications, such as certain areas of hospitals, clinics, laboratories, or manufacturing facilities require **final filters** located downstream of all other components in the air-handling unit (Figure 37).

When HEPA filters are used, locate them in the furthest downstream section of the air-handling unit. If these high-pressure-drop filters are located upstream of the supply fan, the pressure inside the unit casing—between the filters and fan inlet—will likely be at a significantly *negative pressure* with respect to the air surrounding the casing. Any air that leaks into the air-handling unit at this location will introduce particles downstream of the high-efficiency filters, negating some of the benefit of using final filters.

Finally, if possible, avoid combining final filters with a blow-thru supply fan—that is, where the cooling coil is located between the supply fan and final filters (Figure 38). Systems with this configuration often experience problems with final filters getting wet. Experience indicates that this problem can often be minimized by using a draw-thru supply fan—where the supply fan is located between the cooling coil and the final filters—or by adding a few degrees of heat to the air before it passes through the final filters.

Figure 38. Avoid combining a blow-thru supply fan with final filters



Some particulate filters use electrostatic attraction to enhance efficiency. These **electronic air cleaners** are either passively charged (electret) or actively charged (electrostatic precipitators). Passively charged filters are standard panel or pleated media filters that receive an electrostatic charge to the media during the manufacturing process. This charge increases the efficiency rating by one to two MERV levels and typically dissipates over time.

Actively charged air cleaners (electrostatic precipitators) include an active electrical field that charges the media and, in some cases, the particles, as they enter the filter system. This can greatly improve the fine-particle collection efficiency and, in some cases, the dirt-holding capacity of the filter system. Some types of electrostatic precipitators are cleanable, which significantly reduces the cost of system maintenance.

The benefit of this technology is higher collection efficiency and increased dust-holding capacity at a lower airside pressure drop. The primary drawback is an increase in first cost, and it typically increases the cost and size of the air-handling unit. Many actively charged electrostatic precipitators are not MERV-rated, but are rated by collection efficiency at a specific particle size (99.8 percent efficient at capturing 0.5 micron particles, for example).

Gaseous air cleaners

For more information on the various types of gaseous air cleaners, refer to Chapter 29, "Industrial Gas Cleaning and Air Pollution Control," in the 2008 *ASHRAE Handbook—HVAC Systems and Equipment* (www.ashrae.org) or the *NAFA Guide to Air Filtration* (www.nafahq.org).

The presence of undesirable gases and vapors (particularly carbon monoxide, radon, oxidants, and volatile organic compounds, or VOCs, such as formaldehyde) indoors can be detrimental to building occupants, materials, and contents. Controlling VOC concentrations is particularly challenging—hundreds of them are present (often at very low concentrations), few are unique to any one source, and there are many potential sources, some of which emit several VOCs and at various generation rates.

A common way to control the concentration of gaseous contaminants indoors is to dilute them with outdoor air. This approach is appealing because many VOCs defy individual treatment. However, it is only practical if the quality of the outdoor air is suitable, and if the resulting supply airflow is consistent and appropriate and if it mixes effectively with the air in the occupied space.

Removal of gaseous contaminants typically relies on the principles of adsorption, oxidation, or catalysis. Of these, adsorption is the most common.

The media used for adsorption depends on the contaminants to be removed (Table 9). Of these, granular, **activated carbon** is the most widely used. This typically involves installation of a pleated panel or packed-bed media in the air stream (Figure 39).

Figure 39. Activated carbon

pleated panel impregnated with activated carbon

packed-bed media filled with granular activated carbon (shown with prefilters)

Images used by permission from CLARCOR Air Filtration Products (www.clcair.com).

All types of gaseous air cleaners require medium- to high-efficiency particle filtration (typically MERV 13 or higher) upstream to keep the media or catalytic surfaces clean.

Carbon filtration can reliably remove a variety of gases and odors from the air stream. Application considerations include:

- The activated carbon media adds a pressure drop to the airside system, which increases fan energy use.
- Lower air velocity is required through the bed (typically 250 to 300 fpm [1.3 to 1.5 m/s]) for effective operation, often increasing the size of the air-handling unit.
- Filtration efficiency is reduced by high humidity levels.
- Activated carbon does not remove some gases, such as carbon monoxide or carbon dioxide.
- The technology requires frequent maintenance to ensure expected performance, and is messy.

No single media works for all gaseous contaminants (Table 9), so it is important to define the contaminants of concern for a given facility. In addition, the lack of standard performance ratings makes it difficult to effectively evaluate and apply gaseous air cleaners.

Table 9. Recommended removal media for gaseous contaminants

Contaminant	Activated Carbon	Potassium Permanganate Impregnated Media	Caustic Impregnated Carbon	Phosphoric Acid Impregnated Carbon
Acetic acid	•	•	•	
Acetone	•			
Acrolein	•	•		
Amines				•
Ammonia				•
Benzene	•			
Chlorine	•		•	
Ethyl alcohol	•			
Formaldehyde		•		
Gluteraldehyde	•	•		
Hydrogen cyanide			•	
Hydrogen sulfide		•	•	
Methyl alcohol	•	•		
Mercaptans	•	•		
Methylene chloride	•			
Methyl ethyl ketone	•			
Nitric oxide		•		
Nitrogen dioxide		•		
Ozone	•			
Sulfur dioxide		•	•	
Sulfur trioxide		•	•	
Toluene	•			

Source: NAFA Guide to Air Filtration, 4th Edition, Figure 11.5 © National Air Filtration Association, www.nafahq.org

Photocatalytic oxidation (PCO) can also be used to remove VOCs from the air stream. VOCs are adsorbed onto the surface of the catalyst, commonly a bed of titanium dioxide (TiO_2), and ultraviolet light is used to drive a reaction with oxygen and water vapor. Ideally, this process oxidizes all of the organic compounds into carbon dioxide (CO_2) and water (H_2O). (See p. 49 for further discussion of PCO.)

Biologicals

For more information on the various methods for dealing with biological contaminants in the air stream, refer to the *NAFA Guide to Air Filtration* (www.nafahq.org).

Biological, or microbial, contaminants describe a subset of airborne particles that originate from living or once-living organisms (e.g., bacteria, fungi, viruses). Inhaling these agents may cause occupants to experience hypersensitivity and allergic reactions, respiratory disorders, or infectious diseases.

Many of the larger biological particles, such as fungal and bacterial spores, are 3 micron to larger than 10 micron in size, which makes them easily captured with medium-efficiency particle filters (see Figure 34, p. 40). Other biological particles, such as viruses, are sub-micron in size and cannot be practically removed by particle filtration.

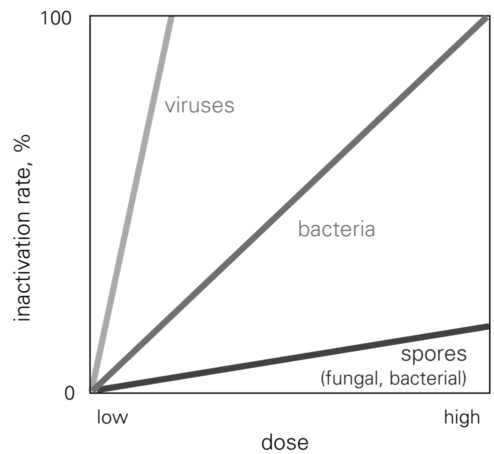
For more information on the benefits and concerns related to using UV-C lights in HVAC systems, including recommendations and precautions for when they are used, refer to the Trane engineering bulletin titled "Using Ultraviolet Light to Control Microbial Growth in Buildings" (CLCH-PRB014-EN) or Chapter 16, "Ultraviolet Lamp Systems," in the 2008 *ASHRAE Handbook—HVAC Systems and Equipment* (www.ashrae.org).

Given sufficient dose (time and intensity), C-band **ultraviolet light (UV-C)** can inactivate (kill) microorganisms. In applications where sufficient residence times exist, such as on stationary surfaces, it is possible and often practical to deliver a "killing" dose of UV energy to the organism. In this application, the UV-C lamps are sized and positioned inside the air-handling unit to irradiate interior surfaces and components (typically the cooling coil and condensate drain pan) to limit fungal and bacterial growth on those stationary surfaces.

In addition to controlling microbial growth on surfaces, UV-C lights are sometimes applied to inactivate microbes in moving air streams (referred to as airborne, or "on-the-fly," kill). The high air velocity through most air-handling units or air distribution systems requires the use of much higher intensity lamps to deliver the required dose of UV energy to the moving microbes (Figure 40). This approach has been effectively applied in some health care applications with specially designed low velocity air systems. However, it is typically impractical for most conventional HVAC systems.

The required dose of UV-C light varies widely depending on the type and form of the microorganism. Viruses require a much lower dose of UV energy than spore forms of fungi (such as mold) and bacteria (Figure 40).

Figure 40. Effectiveness of UV-C light



If using UV-C lamps in air-handling units, be aware of the following safety and material considerations:

- *Include interlock switches and warning signs at each entry location.*
  UV-C energy poses a significant health hazard to unprotected human skin and eyes. To safeguard against inadvertent exposure by operators or maintenance personnel, electrical interlock switches should be provided at all entry locations to the air-handling unit to disconnect power to the lamps when a door is opened. Industry-recognized warning signs should also be placed at all entry locations.
- *Substitute or shield any susceptible materials.*
  UV energy can also prematurely age many polymeric (plastic) materials. Susceptible materials (such as electrical wiring insulation, gaskets, caulk and sealants, and insulation) in close proximity to the lamps should be shielded from direct expose to the UV energy or substituted with other materials. Failure to properly protect these materials can result in serious safety and equipment reliability issues.
- *Clean and replace lamps as specified by the manufacturer.*
  Proper maintenance, including periodic cleaning of the lamp surface (typically every three months), is important to keep the UV lamps operating effectively. The UV intensity degrades over time so lamps need to be replaced at intervals specified by the manufacturer (typically every 9000 operating hours, or annually). Since UV lamps contain mercury, which is a regulated hazardous waste, disposal of lamps is subject to state and federal regulations.

For more information on photocatalytic oxidation (PCO), refer to the Trane engineering bulletin, "Trane Catalytic Air Cleaning System" (CLCH-PRB023-EN).

Photocatalytic oxidation (PCO) uses ultraviolet light shining on the surface of a catalyst, commonly a bed of titanium dioxide (TiO_2), to create highly reactive hydroxyl radicals. The cells of microbiological organisms that contact these hydroxyl radicals are destroyed through a process called lysis (destruction of the cell by rupturing its membrane or cell wall).

Biological organisms also create low levels of VOCs, which typically cause odors. As mentioned earlier (p. 48), PCO technology is able to break down VOCs and remove these biological-related odors.

PCO technology can be applied directly to the air stream since it has a very low airside pressure drop and the reaction takes place at ambient temperatures (Figure 41). The primary drawback is high installed cost and the operating costs associated with cleaning, replacing, and disposing of the UV lamps.

Figure 41. Photocatalytic oxidation (PCO)

Water management

Preventing **blowing snow** from entering through the intake of an outdoor air-handling unit is difficult, so the unit should be designed to minimize the impact of the snow on downstream components (e.g., wetting the filters). Consider including a drain pan in the intake module or mixing box to allow melted snow to drain away.

If a preheat coil is used to help melt snow, configure the unit with the filters located downstream of the preheat coil to keep them from getting wet. Installing a separate throwaway filter upstream of the preheat coil helps keep the coil clean and allows that filter to be removed during winter to avoid it getting wet from snow.

Preventing moisture problems in buildings is a shared responsibility among all parties involved in the design, construction, maintenance, and use of the building. As for preventing water-related problems within the air-handling unit itself, follow these basic practices:

- *Prevent rain from entering through outdoor-air intakes, or from leaking through the equipment casing or roof curb.*
  An outdoor-air intake sized to avoid too high of air velocity and a hood mounted over the intake both help to prevent rain from being drawn into the ductwork or air-handling unit. A mist eliminator just inside the intake opening can be used to remove water droplets that may be carried along with the intake air.

  For an outdoor air-handling unit, the casing should prevent water from leaking in, by including water-tight seams, gasketed doors, and a sloped roof to prevent water from puddling. Puddling (standing) water can lead to rust and eventually water leaking into the unit.

- *Use nonporous, cleanable interior surfaces.*
  Smooth, double-wall interior surfaces allow for easier inspection and cleaning, and also isolate the casing insulation from the air stream. For acoustically sensitive installations, consider a perforated interior surface with matte-faced, coated, or wrapped insulation. The perforated surface is still cleanable, but the casing will absorb some of the generated sound.

  Ensure that the air-handling unit includes easily removable panels and/or access doors to allow for regular inspection and cleaning. Poor location of

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

the air-handling unit 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 air-handling unit within the manufacturer's tolerance for levelness.

- *Properly install condensate traps and periodically clean them out.*
  If the cooling coil and its associated drain pan are located *upstream* of the supply fan (*draw-thru* configuration), the pressure inside the air-handling unit casing at that section is *less than* the pressure outside. Without a drain seal, air can be drawn in through the condensate drain line from outside. Improper condensate trapping results in the wetting of the interior of the unit, and may even allow water to leak into the building.

  If the cooling coil and drain pan are located *downstream* of the supply fan (*blow-thru* configuration), the pressure inside the casing is *greater than* the pressure outside. Without a drain seal, 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 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 cooling coil 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.

Casing performance (leakage and thermal)

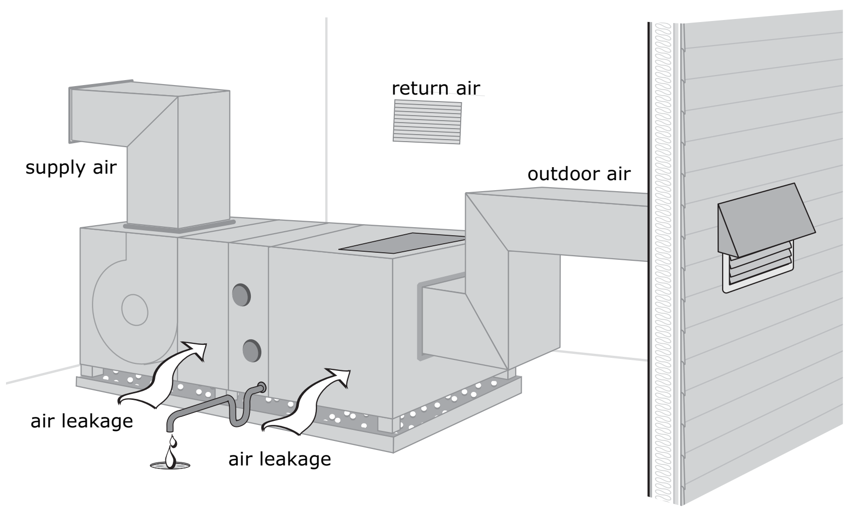
The primary role of the air-handling unit casing is to direct air through the various components (filters, coils, fans, etc.). To accomplish this in an energy-efficient manner, the casing must minimize air leaking into or out of the air-handling unit and minimize heat loss or gain through the casing.

Air leakage

In addition to the casing of the air-handling unit being designed to minimize air leakage, the impact of air leakage on the overall energy use of the HVAC system depends on the configuration of the unit and its location within the system.

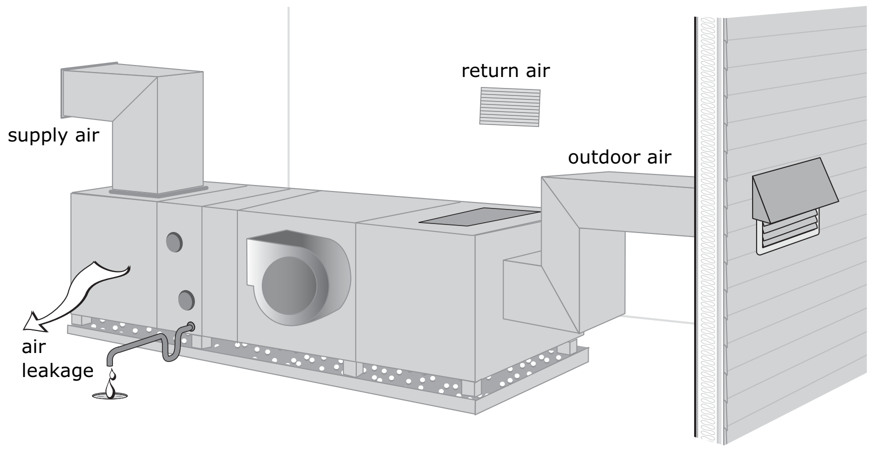
For example, Figure 42 depicts an indoor, *draw-thru* air-handling unit that is installed in a mechanical room, in which the return air passes through the mechanical room before entering the unit. The static pressure inside the unit and upstream of the fan is at a negative pressure relative to the air surrounding the unit. (Only the fan discharge is at a positive pressure.) This will result in air leaking *into* the air-handling unit. Any air that leaks into the unit between the cooling coil and supply fan will raise the temperature of the supply air. Since the VAV air-handling unit is controlled to maintain the supply-air temperature at a setpoint, this leakage results in the cooling coil needing to overcool the portion of air that passes though the coil (in order to offset the mixing of warmer air downstream of the coil). But any air that leaks into the unit upstream of the cooling coil does not impact the supply-air temperature because it will still pass through the cooling coil. This is simply return air that enters the unit via leakage, rather than through the return-air damper. Of course, if the return air is ducted directly to the air-handling unit, the air that leaks in is not conditioned return air and will likely increase cooling energy.

Figure 42. Example impact of air leakage (*draw-thru* indoor AHU)



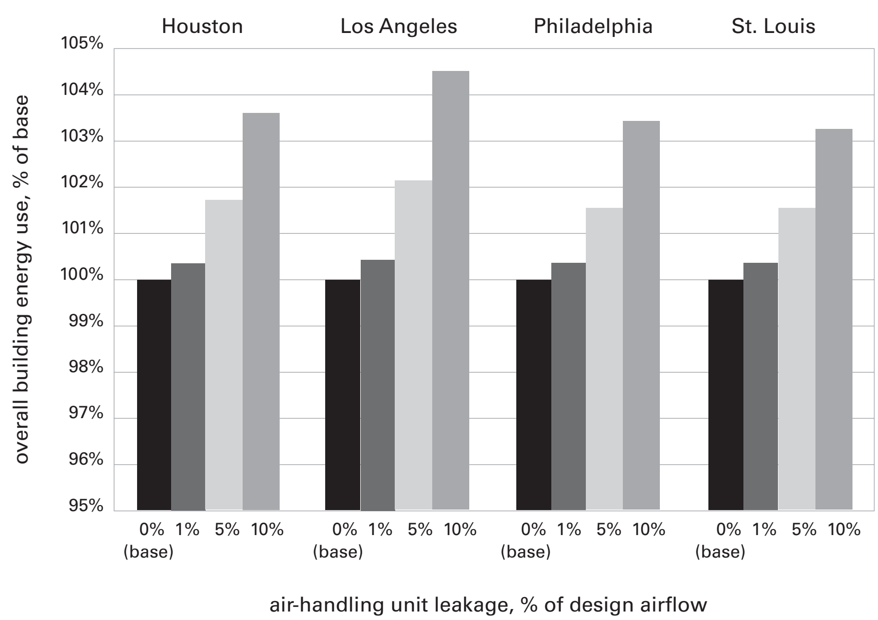
By contrast, Figure 43 depicts the same indoor air-handling unit configured as a *blow-thru* unit. The static pressure inside the unit and downstream of the fan is at a positive pressure relative to the air surrounding the unit. This will result in air leaking *out of* the air-handling unit. This will increase fan airflow and fan energy use, because the fan is moving more air than is being delivered to the spaces. In addition, the air that leaks out of the unit has already passed through the cooling coil, so the unit is using more cooling energy to condition air that is not being sent to the spaces. However, since the return air from the spaces passes through the mechanical room where the air-handling unit is located, any air that leaks out of the unit mixes with (and pre-cools) this return air, so not all of the excess cooling energy is lost. Of course, if the return air is ducted directly to the air-handling unit, the energy used to cool the air that leaks out is lost.

Figure 43. Example impact of air leakage (*blow-thru* indoor AHU)



The actual impact of air leakage on the overall energy use of the HVAC system depends on whether the air-handling unit is located indoors or outdoors, the location of the fan within the overall unit (draw-thru or blow-thru), whether the return air is ducted directly to the air-handling unit or returns through the mechanical room in which the unit is located, and many other factors. As an example, Figure 44 illustrates the impact of air leakage from a blow-thru, indoor air-handling unit on the overall energy use of the building.

Figure 44. Impact of AHU leakage on building energy use (blow-thru indoor AHU with ducted return)



Regardless of unit or system configuration, all penetrations through the casing and base of the air-handling unit should be sealed, including any electrical and piping connections.

Thermal performance

The air-handling unit casing also provides an insulating function to minimize heat loss or gain and to minimize condensation from forming on the interior or exterior surfaces of the unit. If the air surrounding the air-handling unit is unconditioned, or significantly warmer or colder than the conditioned air inside the unit, heat gain/loss and condensation may be a design issue.

For more information on preventing condensation of air-handling units, including design strategies for the mechanical equipment room, refer to the Trane application manual titled *Managing Building Moisture* (SYS-AM-15).

To prevent condensation from forming on the surfaces of the air-handling unit, the casing thermal resistance must maintain the surface temperature of the casing exterior above the dew point of the surrounding air. The following equation defines the cold-spot thermal-resistance ratio (TR) required for a specific application:

$$TR_{min} = (DPT_{out} - DBT_{in}) / (DBT_{out} - DBT_{in})$$

where,

TR_{min} = minimum cold-spot thermal-resistance ratio required
DPT_{out} = dew point of the air outside the casing, °F (°C)
DBT_{in} = dry-bulb temperature of the air inside the casing, °F (°C)
DBT_{out} = dry-bulb temperature of the air outside the casing, °F (°C)

For example, consider an air-handling unit that will be installed in an unconditioned equipment room. The worst-case conditions for the equipment room are 96°F (36°C) dry bulb and 60 percent relative humidity, which equates to a 80°F (27°C) dew point. If the temperature of the air leaving the cooling coil is 55°F (13°C), the thermal-resistance ratio of the air-handling unit casing must be at least 0.61.

$$TR_{min} = (80°F - 55°F) / (96°F - 55°F) = 0.61$$

$$[TR_{min} = (27°C - 13°C) / (36°C - 13°C) = 0.61]$$

To minimize the risk of condensation, either the air-handling unit casing must be above this calculated minimum TR value or the mechanical equipment room can be conditioned to lower the dew point of the air surrounding the unit (see Figure 110, p. 155, and Figure 96, p. 124).

VAV Terminal Units

Pressure-dependent versus pressure-independent control

A pressure-*dependent* VAV controller uses the zone temperature sensor to directly control the position of the modulating damper. The actual airflow delivered to the zone is a by-product of this position and depends on the static pressure at the inlet of the terminal unit. Although the zone temperature sensor will continually correct the position of the damper, the response can be sluggish and may cause unacceptable temperature variations within the zone, particularly if reheat is not used.

In contrast, a pressure-*independent* VAV controller directly controls the actual volume of primary air that flows to the zone. This requires accurate measurement of primary airflow, which is typically accomplished with a multipoint airflow sensor mounted on the inlet. The position of the modulating damper is not directly controlled and is a by-product of regulating the airflow through the unit. Because the airflow delivered to the zone is directly controlled, it is independent of inlet static pressure. Pressure-independent control increases the stability of airflow control, and allows minimum and maximum airflow settings to become actual airflows rather than physical positions of the modulation device.

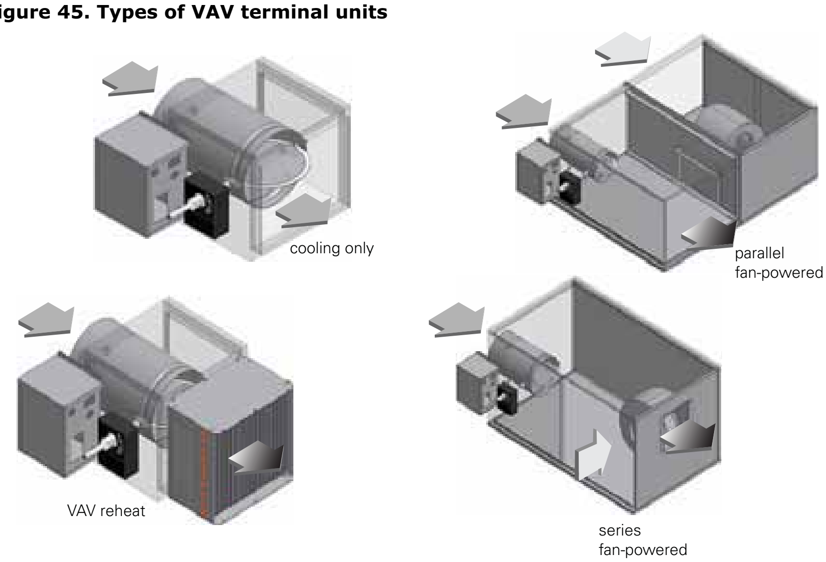
The supply ductwork delivers air to each of the VAV terminal units. Each zone has a VAV terminal unit that varies the quantity of air delivered to maintain the desired temperature in that zone.

A VAV terminal unit is a sheet-metal assembly consisting of an airflow-modulation device, a flow sensor (on units with pressure-independent control), a controller, and possibly a heating coil, small terminal fan, or filter. Modulating airflow to the zone is typically accomplished by using a rotating blade damper that changes airflow resistance by rotating the damper and adjusting the size of the air passage to the zone.

Types of VAV terminal units

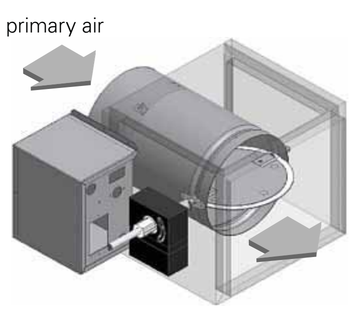
This section describes the various types of VAV terminal units used in VAV systems (Figure 45). See "Typical combinations used in chilled-water VAV systems," p. 66, for further discussion on the typical applications for these different terminal units.

Figure 45. Types of VAV terminal units



Cooling-only VAV terminal units

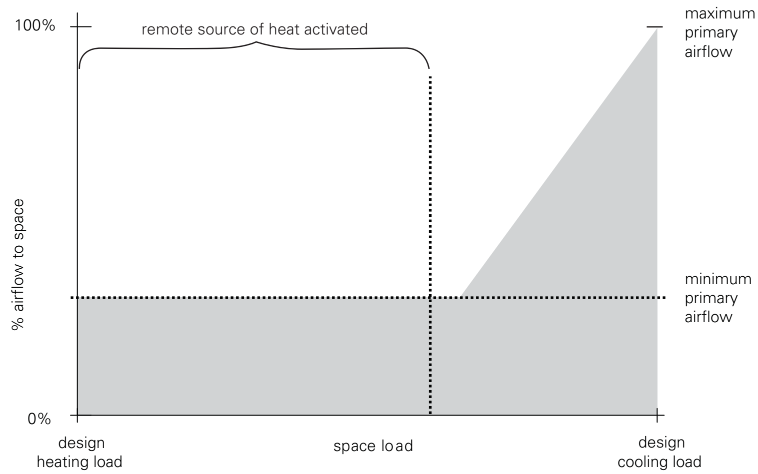
Figure 46. Cooling-only VAV terminal unit



The cooling-only VAV terminal unit (Figure 46) consists of an airflow-modulation device, with a flow sensor and controls packaged inside a sheet metal enclosure. Primary airflow to the zone is reduced as the cooling load decreases. Responding to the zone sensor, primary airflow is modulated between maximum and minimum settings (Figure 47). The maximum setting is determined by the design cooling load of the zone. The minimum setting is normally determined to ensure that the zone is properly ventilated or to meet requirements for proper operation of the terminal unit or supply-air diffusers.

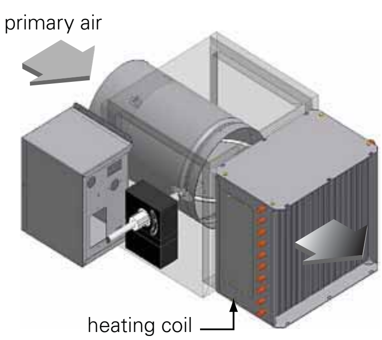
Most cooling-only units are used for zones that require year-round cooling, like the interior zones of a building. When cooling-only units are applied to zones that *do* have a need for heat, the heat is typically provided by a remote source, such as baseboard radiant heat located along the perimeter wall in the zone. When the zone requires heating, primary airflow is at the minimum setting, and the remote heat source is activated (Figure 47). Many VAV unit controllers provide an output signal to control this remote source of heat.

Figure 47. Control of a cooling-only VAV terminal unit



VAV reheat terminal units

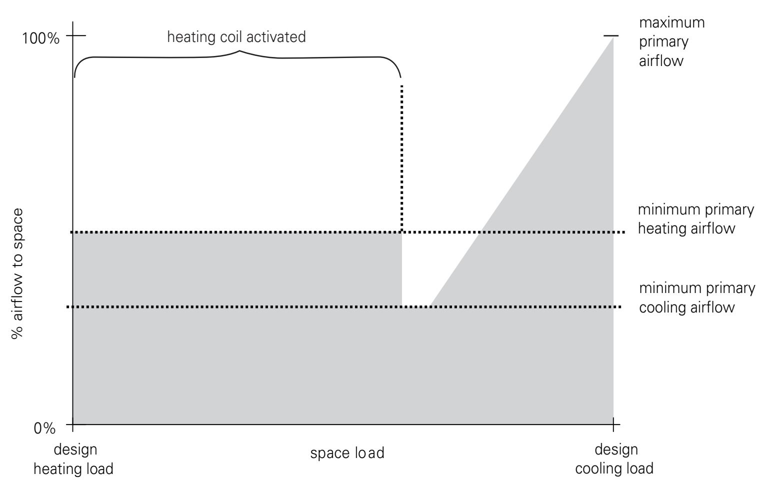
Figure 48. VAV reheat terminal unit



The VAV reheat terminal unit (Figure 48) also contains an airflow-modulation device, flow sensor, and controls, but it also contains an electric or hot-water heating coil. In the cooling mode, the unit is controlled in the same manner as the cooling-only unit. Primary airflow is reduced as the cooling load in the space decreases (Figure 49).

When primary airflow reaches the minimum setting for the unit, and the cooling load continues to decrease, the heating coil warms (tempers) the air to avoid overcooling the zone.

Figure 49. Control of a VAV reheat terminal unit



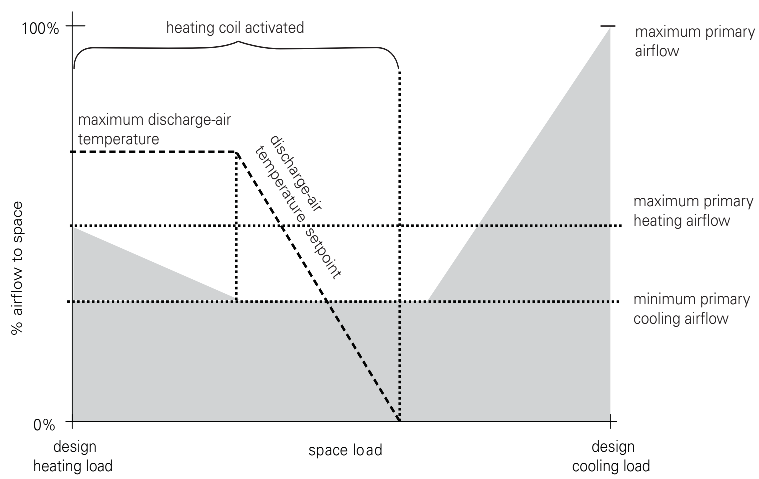
When the zone heating load requires the air to be delivered at a temperature warmer than the zone, the primary airflow may be increased to a higher minimum setting than is used during the cooling mode (Figure 49). This “heating minimum airflow” setting is needed because when warm, buoyant air is supplied from the ceiling, a higher velocity is required to effectively mix the air and avoid temperature stratification in the occupied portion of the zone. Increased airflow may also be needed to meet the minimum requirement for safe and proper operation of an electric heating coil.

Figure 50 shows an alternate method to control a VAV reheat terminal unit. When the zone requires cooling, the control sequence is unchanged; primary airflow is varied between maximum and minimum cooling airflow, as required to maintain the desired temperature in the zone.

For more information on this alternate control strategy (referred to by some as the "dual maximum" strategy), refer to the California Energy Commission's *Advanced Variable Air Volume System Design Guide* (500-03-082-A-11, October 2003).

An addendum (h) to ASHRAE Standard 90.1-2007 allows this alternate control strategy (with a maximum heating primary airflow ≤ 50% of maximum cooling primary airflow) as an exception to comply with the limitation on simultaneous heating and cooling (see "Simultaneous heating and cooling limitation," p. 130).

Figure 50. Alternate control of a VAV reheat terminal unit



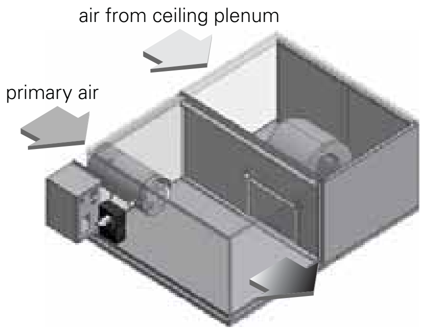
When primary airflow reaches the minimum cooling airflow setting, and the cooling load continues to decrease, the heating coil is activated to warm (temper) the air to avoid overcooling the zone. As more heat is needed, the controller resets the discharge-air temperature setpoint upward to maintain zone temperature at setpoint (dark, dashed line in Figure 50), until it reaches a defined maximum limit. (This control sequence requires a discharge-air temperature sensor installed for each VAV terminal.) The discharge temperature is limited to minimize temperature stratification when delivering warm air through overhead diffusers (see "Best practices for locating supply-air diffusers," p. 75).

When the discharge-air temperature reaches the maximum limit, and the zone requires more heating, primary airflow is increased, while the discharge-air temperature setpoint remains at the maximum limit (Figure 50). The result is that the airflow-modulation damper and hot-water valve will modulate open simultaneously.

By actively controlling the discharge-air temperature, it can be limited so that temperature stratification and short circuiting of supply to return are minimized when the zone requires heating. This improves occupant comfort and results in improved zone air-distribution effectiveness (see "Impact of zone air-distribution effectiveness," p. 102).

Fan-powered VAV terminal units

Figure 51. Parallel fan-powered VAV terminal unit



Fan-powered VAV terminal units include an airflow modulation device, a flow sensor, and a small fan packaged inside an insulated cabinet. The terminal fan mixes warm air from the ceiling plenum with cool primary air from the central air-handling unit to offset heating loads in the zone. A heating coil can be added to the terminal unit and activated when additional heat is required.

A *parallel fan-powered VAV* terminal unit (Figure 51) has a small fan configured inside the terminal unit so that its airflow is in parallel with the primary airflow path. In the cooling mode, the unit is controlled in the same manner as the cooling-only unit. Primary airflow is reduced as the cooling load in the zone decreases.

However, when primary airflow reaches the minimum setting for the unit, and the cooling load continues to decrease, the small fan activates to mix warm plenum air with the cool primary air (Figure 52). This increases the total airflow to the zone, which improves mixing, decreases the risk of temperature stratification, and allows the diffusers to perform better. It also results in a warmer supply-air temperature. As additional heating is required, the terminal fan remains on, and a heating coil is used to further warm the supply air.

Figure 52. Control of a parallel fan-powered VAV terminal unit

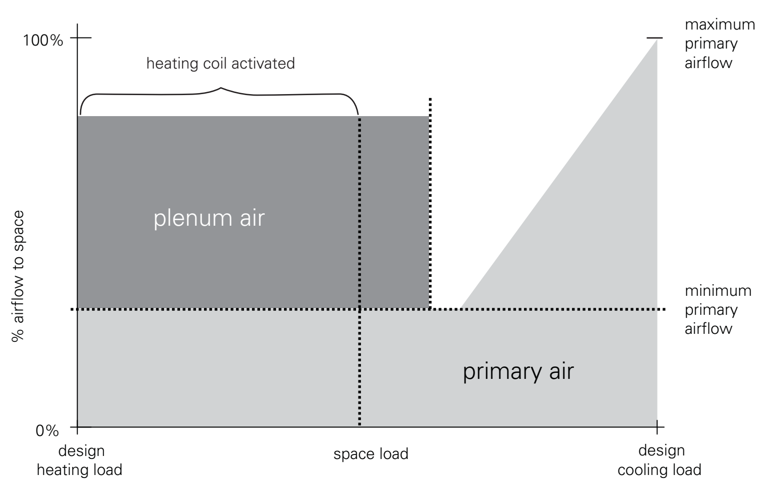
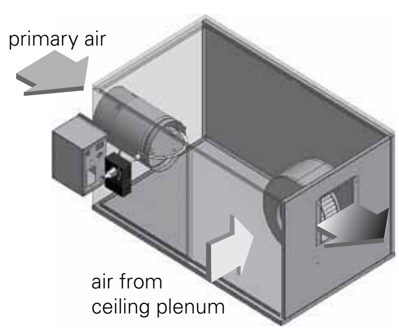


Figure 53. Series fan-powered VAV terminal unit

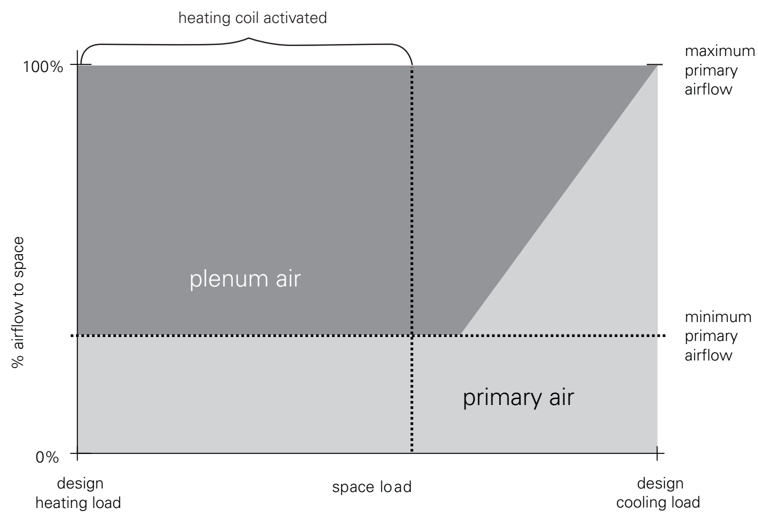


The terminal-unit fan cycles on only when the zone requires heating. Operating the fan is a form of energy recovery. It tempers the supply air with warm return air, which has gained heat from the building and lights. This delays the need to use "new" energy by activating the heating coil.

A *series fan-powered VAV* terminal unit (Figure 53) has a relatively large fan configured inside the terminal unit so its airflow is in series with the primary airflow path. The terminal-unit fan operates continuously whenever the zone is occupied, and draws air from either the primary air stream or the ceiling plenum, based on the cooling or heating requirement of the zone. The result is a constant volume of air delivered to the zone at all times.

In the cooling mode, the primary airflow is reduced as the cooling load in the zone decreases. The total airflow to the zone remains constant, a combination of cool primary air and warm plenum air (Figure 54). When primary airflow reaches the minimum setting for the unit, and the cooling load continues to decrease, a heating coil can be used to warm the air to maintain zone temperature.

Figure 54. Control of a series fan-powered VAV terminal unit



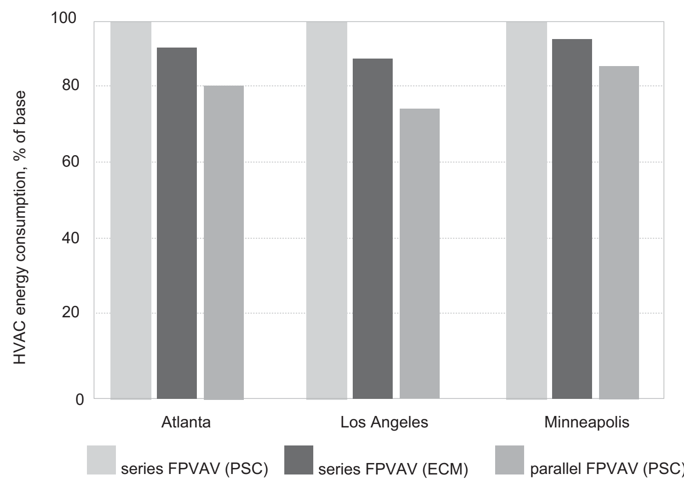
Parallel versus series fan-powered VAV terminals. Series fan-powered units are sometimes selected because the zones receive constant airflow. Constant airflow simplifies selection of the supply-air diffusers and, compared to parallel fan-powered units, increases air motion at part load, which can improve occupant comfort.

However, because the terminal fan runs continuously whenever the zone is occupied, and because it must be sized for the design airflow to the zone, a series fan-powered unit consumes more energy than a parallel fan-powered unit. The use of series fan-powered terminals can decrease the energy used by the central supply fan, because the series fan overcomes the pressure loss between the terminal and zone. However, the actual impact on overall system energy use depends on the efficiency differences between the supply and terminal fans and motors. In general, the larger centralized supply fan has a significant efficiency advantage (both fan and motor efficiency) over the terminal fan, even with the use of ECM technology (see "Electronically commutated motors on fan-powered VAV terminal units," p. 60).

In addition, the heat generated by the terminal fan motor in a series unit is added to the air stream during both cooling and heating operation, while the fan heat in a parallel unit is only added to the air stream when it provides a benefit (that is, only when heating is required).

Figure 55 shows the impact of using parallel versus series fan-powered VAV terminals in an example office building. The "base case" is a chilled-water VAV system that uses series fan-powered units equipped with conventional, permanent split capacitor (PSC) motors. For this example, parallel fan-powered VAV (also with PSC motors) consumed 20 percent less HVAC energy in Atlanta, 28 percent less in Los Angeles, and 12 percent less in Minneapolis. Even when the series fan-powered VAV terminals are equipped with ECMs, they still used more energy than parallel fan-powered VAV.

Figure 55. Energy consumption of parallel versus series fan-powered VAV



Another difference between the two types of fan-powered terminals is sound. Series fan-powered units typically produce higher sound levels in the occupied space, but some people may prefer the constant sound level of a series unit to the on-off sound generated by the cycling fan in a parallel fan-powered unit. Adding an ECM to a parallel fan-powered unit allows the terminal fan to ramp up slowly when activated, which minimizes the distraction of the fan cycling on and off.

Finally, a series fan-powered unit typically costs more than an equivalent parallel fan-powered unit because it requires a larger terminal fan—sized to deliver design airflow to the zone. And, a system that uses series fan-powered VAV terminals typically uses them in *all* zones. This approach costs more than a system that uses parallel fan-powered VAV terminals in the perimeter zones and cooling-only VAV terminals in the interior zones.

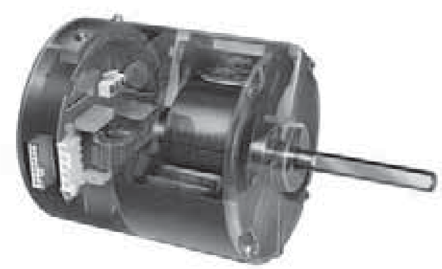
For more information on electronically commutated motors (ECM), refer to the Trane *Engineers Newsletter* titled "Setting a New Standard for Motor Efficiency: Brushless DC Motors" (ADM-APN013-EN).

Electronically commutated motors on fan-powered VAV terminal units. An electronically commutated motor (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 56). They are more efficient than the single-speed, fractional-horsepower motor technologies that have traditionally been used in fan-powered VAV terminals.

To offset some of the increased energy use of series, fan-powered VAV systems, the use of ECMs has become increasingly popular. Some of the benefits include:

- *Energy savings*
  Figure 57 compares the performance of a standard AC motor with that of an ECM, at various airflows, for a series fan-powered VAV terminal unit. Although the efficiency advantage is less at the upper end of the

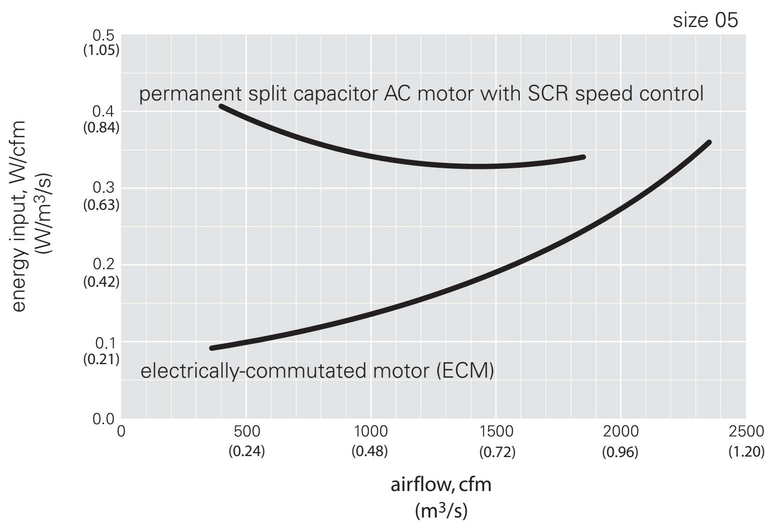
Figure 56. Electronically commutated motor



© GE Industrial Systems

application range for the motor, the efficiency benefit of the ECM is significant at the lower end of the airflow range. This efficiency difference often allows ECMs to offer substantial energy savings compared to conventional motor technologies.

Figure 57. Comparison of motor power in series fan-powered VAV boxes



The added cost of an ECM can be offset more quickly in applications that require a relatively high number of hours of operation. For this reason, they are more commonly used with series fan-powered terminals, in which the terminal fan operates whenever the zone is occupied (see "Parallel versus series fan-powered VAV terminals," p. 59).

However, even if a zone may not be a good candidate for this type of motor based solely on energy savings, the decision to use an ECM may be based on the other benefits listed below.

- *Self-balancing*
  An ECM is capable of maintaining a relatively constant airflow, regardless of future changes to downstream ductwork.
- *Gradually changing sound levels*
  The "soft-start" nature of the ECM allows the fan to ramp up slowly when activated. This minimizes the distraction of the fan cycling on and off in a parallel fan-powered VAV terminal.

Potential drawbacks include:

- *Higher installed cost*
  ECMs require power transistors 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,

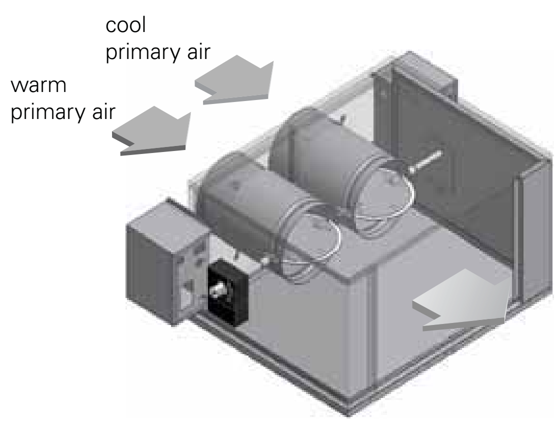
interfere with the operation of sensitive equipment, and in severe cases, burn out transformers and motors.

Determining whether harmonic currents will cause a problem in a particular building requires review of the electrical system *before* it's 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).

Dual-duct VAV terminal units

A dual-duct VAV terminal unit consists of two airflow-modulation devices packaged inside a sheet metal enclosure (Figure 58). One modulation device varies the amount of cold primary air and the other varies the amount of warm primary air. These two air streams mix inside the dual-duct unit before being distributed downstream to the zone.

Figure 58. Dual-duct VAV terminal unit



Dual-duct VAV systems are intended for buildings that require seasonal cooling and heating. The energy use of this system is generally low, and it can provide excellent control of both temperature and humidity. However, this system is relatively uncommon because of the need to install two separate duct systems.

For more discussion, see “Dual-Duct VAV Systems,” p. 165.

Minimum primary airflow settings

In most applications, each VAV terminal unit has a minimum primary airflow setting to ensure proper ventilation or to ensure proper operation of the terminal unit or supply-air diffusers. Providing less than this required minimum airflow may:

- Underventilate the zone and degrade indoor air quality
- Result in poor distribution (mixing) of supply air into the conditioned space (some types of supply-air diffusers require a minimum discharge velocity)

- Cause erroneous airflow readings that interfere with proper control of the VAV terminal unit (the accuracy of the flow sensor in the VAV terminal is based upon a specific airflow range)

For multiple-zone VAV systems, this minimum primary airflow setting should be greater than the zone ventilation requirement (V_{oz}). This avoids excessively high zone primary outdoor-air fractions (Z_p), which could significantly increase the required system intake airflow (see “Ventilation,” p. 101). However, ASHRAE Standard 90.1 places limits on using “new” energy to reheat air that has been previously cooled. See “Simultaneous heating and cooling limitation,” p. 130, for more information on how to select minimum primary airflow settings that comply with both ASHRAE Standards 62.1 and 90.1.

Finally, for certain types of spaces, building codes or process requirements dictate a certain minimum air change rate (air changes per hour, or ACH) for the zone. This is most common in certain health care facilities, laboratories, and manufacturing plants. In this case, the minimum primary airflow setting for the VAV terminal unit is set equal to this minimum air change rate:

minimum airflow setting, cfm = minimum required air change rate x volume of zone, ft^3 / 60 min/hr

(minimum airflow setting, m^3/s = minimum required air change rate x volume of zone, m^3 / 3600 s/hr)

Perimeter versus interior zones

For simplicity, a typical building can be described as having two types of thermal zones, perimeter and interior. As mentioned earlier, each zone is typically served by an individual VAV terminal unit, allowing independent control of cooling and heating.

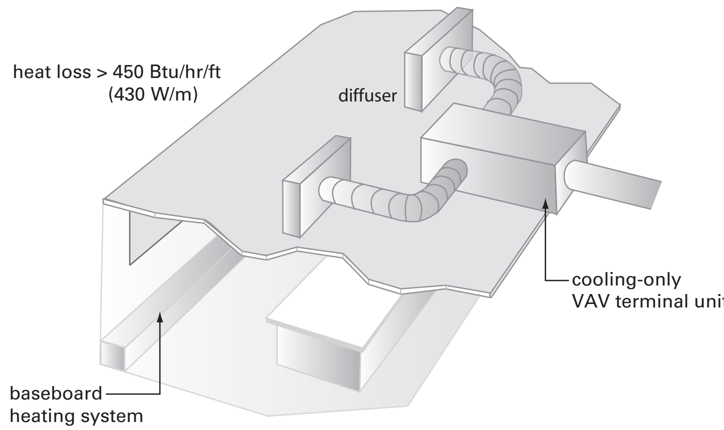
Perimeter zones

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: 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.

Before a VAV terminal unit can be selected to serve a perimeter zone, the design engineer must determine the heating load for that zone. This determines whether the heating load can be satisfied by supplying warm air through overhead diffusers or if the load must be offset by a separate perimeter heating system (e.g., baseboard radiant heat). The following guidelines for heating a perimeter zone are based on the heat loss per unit length of perimeter wall.

If the design heat loss in a perimeter zone exceeds 450 Btu/hr per linear foot (430 W/m) of perimeter wall, an under-the-window (or baseboard) heating system is typically used (Figure 59). With this much heat loss, supplying a high quantity of warm air from overhead diffusers can cause downdrafts, leading to occupant discomfort.

Figure 59. Perimeter heating in a zone with a *high* amount of heat loss

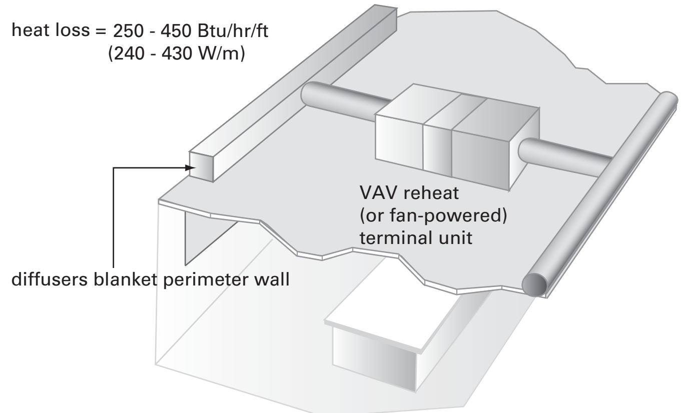


The cooling requirements of these perimeter zones are satisfied by a cooling-only VAV terminal unit. The supply-air diffusers are located in the center of the room to distribute cool air throughout the zone. During the heating mode, the terminal unit provides a minimum airflow to the zone to meet ventilation requirements. The baseboard heating system is separate, but can be controlled by the same VAV unit controller. Having only one controller for the zone ensures proper sequencing of the cooling and heating systems.

If the design heat loss in a perimeter zone is less than 450 Btu/hr per linear foot (430 W/m) of perimeter wall, downdrafts are typically less problematic and heated air can be supplied through ceiling-mounted diffusers. When overhead heating is acceptable, VAV reheat or fan-powered VAV terminal units can be used to provide both cooling and heating for perimeter zones. The rate of heat loss is then used to determine proper diffuser location.

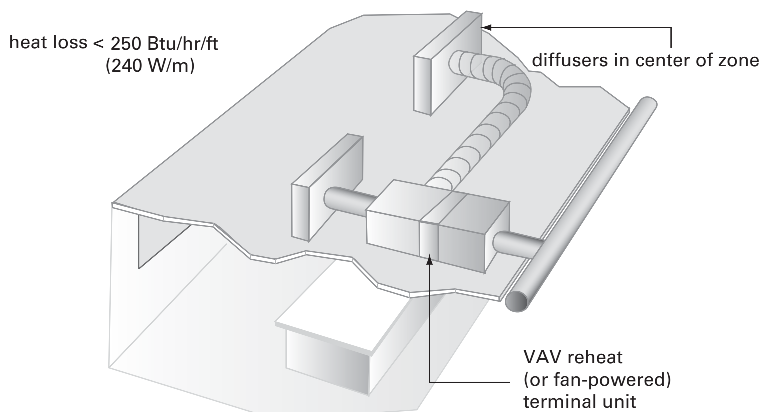
If the design heat loss is between 250 and 450 Btu/hr per linear foot (240 to 430 W/m) of perimeter wall, the diffuser airflow pattern should blanket the perimeter walls with warm air (Figure 60).

Figure 60. Perimeter heating in a zone with a *moderate* amount of heat loss



If the design heat loss is less than 250 Btu/hr per linear foot (240 W/m) of perimeter wall, diffusers can typically be located in the center of the room and still provide adequate blanketing to handle the perimeter heat loss (Figure 61).

Figure 61. Perimeter heating in a zone with a *minimal* amount of heat loss



Interior zones

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 losses 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 heat loss through the roof.

Most interior zones are served by cooling-only VAV terminal units that modulate supply airflow in response to the changing cooling load. Some interior zones, such as conference rooms, may require some amount of tempering (reheat) to avoid overcooling the zone at partial occupancy.

Typically, either VAV reheat or fan-powered VAV terminal units are used to provide the tempering needed at lower cooling loads. They also have a minimum airflow setting to serve ventilation requirements.

Typical combinations used in chilled-water VAV systems

Most buildings use a combination of VAV terminal unit types. Following is a discussion of the most common combinations:

Interior zones: Cooling-only VAV
Perimeter zones: Cooling-only VAV with baseboard radiant heat

All zones, interior and perimeter, are served by cooling-only VAV terminal units. However, the perimeter zones also include baseboard radiant heat along the perimeter walls.

Benefits

- Simple, dependable heating system
- Radiant heat below the windows helps minimize downdrafts
- Simple control of morning warm-up operation
- No hot water pipes (to serve VAV terminals) need to be installed in the ceiling plenum
- No VAV terminal fan energy consumption

Drawbacks/Challenges

- Requires coordinated control to prevent the baseboard heating system from “fighting” with the cooling-only VAV system
- Hot water baseboard radiant heat normally has a relatively high installed cost
- Baseboard heating systems require floor space within the perimeter zones
- Hot water or steam radiant heat requires additional space in the building for a boiler and water or steam distribution system

Interior zones: Cooling-only VAV
Perimeter zones: VAV reheat

Most of the interior zones are served by cooling-only VAV terminal units. The perimeter zones, and certain densely occupied interior zones that experience widely varying occupancy (such as conference rooms), are served by VAV reheat terminal units.

VAV reheat typically has the lowest installed cost among VAV systems, particularly when electric heat is used in the VAV terminals. But this combination is typically less efficient than systems that use parallel, fan-powered VAV terminals (fan-powered VAV terminals recover heat from the ceiling plenum). See “Fan-powered VAV terminal units,” p. 58.

The magnitude of this trade-off varies with building usage and climate. For example, consider a well-constructed building in a mild climate. During the day, the heat generated by people and lights may be more than enough to overcome the heat loss through the building envelope. This can result in a net cooling load on most days, even in perimeter zones. Therefore, during occupied hours, the system may not need to activate the heating coils in the VAV terminal units.

During the night, on the other hand, when people are generally gone and most of the lights are turned off, the heat loss through the building envelope causes the temperature in the zone to drop. In the morning, prior to occupancy, the supply fan is turned on, the outdoor-air damper is closed, and the heating coils in the VAV reheat terminal units are used to warm the zone.

In this example, the VAV reheat terminals either operate in the cooling mode or in the morning warm-up mode, and the heating coils are never used to reheat previously cooled air. The result is that, for a well-constructed building in a mild climate, overall energy use is likely to be very close to that of a fan-powered VAV system, and the installed cost is typically lower. Of course, this changes with climate and building type.

Benefits

- Normally has the lowest installed cost among VAV systems, particularly when electric heat is used in the VAV terminal units
- Often preferred in retrofit projects because cooling and heating are both provided by a single VAV terminal unit, and VAV reheat terminals are smaller than fan-powered VAV terminals
- Economical method of providing heat to perimeter zones in a building with relatively small heating loads
- No floor space is required in the zone for the heating system
- Does not require a boiler and water distribution system, if electric heating coils are used in the VAV terminal units

Drawbacks/Challenges

- Uses more terminal heating energy than fan-powered VAV units
- VAV terminals with hot water coils require piping and valves for each unit to be installed in the ceiling plenum
- Units with electric heat typically require larger power wiring to each unit
- Potential for downdraft problems in perimeter zones with very high heat loss (Figure 59, p. 64)
- High minimum airflow settings may require boiler operation during the cooling season
- High minimum heating airflow settings may be required to minimize drafts and provide adequate mixing in the zone

Interior zones: Cooling-only VAV
Perimeter zones: Parallel fan-powered VAV

Most of the interior zones are served by cooling-only VAV terminal units. The perimeter zones, and certain densely occupied interior zones that experience widely varying occupancy (such as conference rooms), are served by parallel, fan-powered VAV terminal units.

Benefits

- Minimizes terminal heating energy by using warm air from the ceiling plenum, which has been warmed from heat generated by lights
- Often suitable for retrofit projects because both cooling and heating can be provided by a single VAV terminal
- No floor space is required in the zone for the heating system
- Does not require a boiler and water distribution system, if electric heating coils are used in the VAV terminal units
- Fan-powered terminals equipped with a heating coil can provide unoccupied heating to the zone without the need to turn on the central supply fan

Drawbacks/Challenges

- Fan-powered VAV terminal units typically cost more than VAV reheat terminal units
- Terminals with hot water heating coils require piping and valves for each unit to be installed in the ceiling plenum
- Typically require larger power wiring to each unit because of the terminal fan
- Potential for downdraft problems in perimeter zones with very high heat loss (Figure 59, p. 64)
- Additional source of noise (terminal fan) is located above, or very near, the occupied space
- Operation of the terminal fan consumes energy

Fan-powered terminal units *without a heating coil* are typically used for those zones that require year-round cooling and have relatively high minimum airflow settings, such as interior conference rooms. Fan-powered terminal units *with a heating coil* are typically used for zones that require seasonal cooling and heating, such as the perimeter zones of a building.

Interior zones: Series fan-powered VAV
Perimeter zones: Series fan-powered VAV

All of the zones, both interior and perimeter, are served by series fan-powered VAV terminal units.

Benefits

- Reduces terminal heating energy by using warm air from the ceiling plenum, which has been warmed from heat generated by lights
- Series fan-powered terminals deliver constant airflow to the zone at all times
- Series fan-powered terminals can allow for a smaller, central supply fan because the terminal fans overcome the pressure loss between the terminal and the zone
- Often suitable for retrofit projects because both cooling and heating can be provided by a single VAV terminal
- No floor space is required in the zone for the heating system
- Does not require a boiler and water distribution system, if electric heating coils are used in the VAV terminal units
- Fan-powered terminals equipped with a heating coil can provide unoccupied heating to the zone without the need to turn on the central supply fan

Drawbacks/Challenges

- Series fan-powered VAV terminal units typically cost more than either VAV reheat or parallel fan-powered VAV terminal units
- A system that uses series fan-powered VAV terminals in all zones costs more than a system that uses parallel fan-powered VAV terminals in perimeter zones and cooling-only VAV terminals in the interior zones
- Terminals with hot-water heating coils require piping and valves for each unit to be installed in the ceiling plenum
- Typically require larger power wiring to each unit because the terminal fan in a series fan-powered unit is larger than the fan in a parallel fan-powered unit
- Potential for downdraft problems in perimeter zones with very high heat loss (Figure 59, p. 64)
- Additional source of noise (terminal fan) is located above, or very near, the occupied space
- Continuous operation of the terminal fan (operates whenever the zone is occupied) consumes more energy than a parallel fan-powered unit (where the terminal fan only operates when heating is required)

Interior zones: Cooling-only VAV
Perimeter zones: Dual-duct VAV

Most of the interior zones are served by cooling-only VAV terminal units. The perimeter zones, and certain densely occupied interior zones that experience widely varying occupancy (such as conference rooms), are served by dual-duct VAV terminal units.

Benefits

- Eliminates terminal heating energy by using warm, centrally recirculated air
- No heating coils in the dual-duct VAV terminals, and no associated water distribution system
- No fans located in the dual-duct VAV terminals
- No floor space is required in the zone for the heating system

Drawbacks/Challenges

- Two duct systems are required
- Dual-duct VAV terminal units typically cost more than VAV reheat terminal units
- Potential for downdraft problems in perimeter zones with very high heat loss (Figure 59, p. 64)

Air Distribution

For an indoor installation, a VAV air-handling unit can be configured to discharge the supply air horizontally or vertically, depending on the floor space available and the layout of the mechanical room with respect to the area of the building to be conditioned. Additionally, the return air can enter the air-handling unit from one of many directions.

For an outdoor installation, both the supply-air and return-air paths are typically routed downward, through the roof curb and building roof, into the building. However, architectural or acoustic limitations may require the supply air to be discharged horizontally from the air-handling unit. The return air may also need to be connected horizontally, rather than from below.

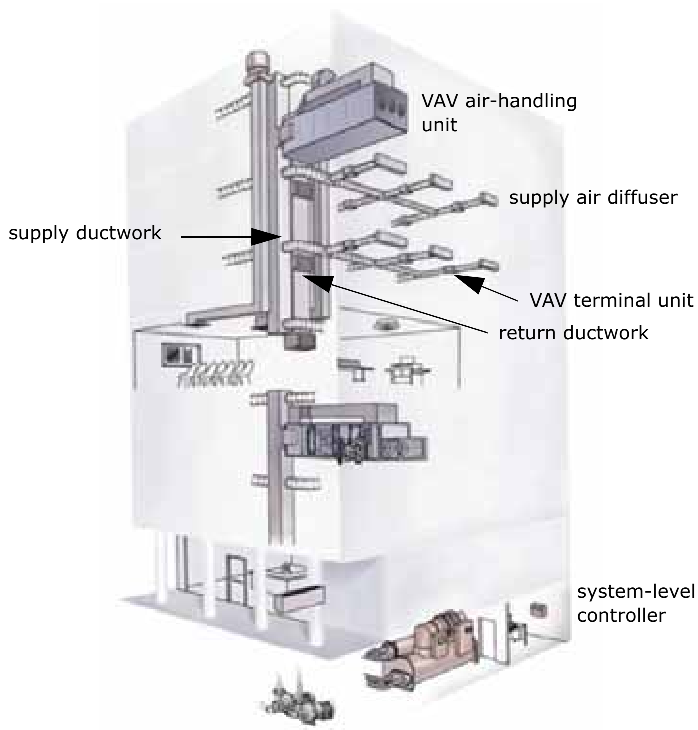
Contact the equipment manufacturer for specific information on available configurations.

Supply duct system

The supply duct system transports air from the air-handling unit to each of the VAV terminal units, and then on to the supply-air diffusers (Figure 62). From its connection to the air-handling unit, the supply duct may be routed through a central, vertical shaft and connected to ductwork that is located in the ceiling plenum above each floor of the building.

Between the air-handling unit and the VAV terminal units, most of the supply ductwork is constructed of sheet metal. However, a short section of flexible duct is commonly used to connect the sheet metal duct to the inlet of each VAV terminal unit. This allows for some flexibility when the sheet metal ductwork and VAV terminals are not installed at the same time.

Figure 62. Air distribution components of a chilled-water VAV system



The supply ductwork between the VAV terminal units and the supply-air diffusers is constructed of either sheet metal or flexible duct. It is best, however, to limit the use of flexible duct to no longer than 6-ft (2-m) sections to reduce the turbulence and high pressure drop associated with flexible duct. If the overall distance between the VAV terminal and diffuser is greater than 6 ft (2 m), 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.

A successful design of the supply duct system achieves the following:

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

To achieve all of the above, system designers commonly use one of two methods to design the supply duct system: the equal friction method or the static regain method.

Equal friction method

Duct systems designed using the equal friction method are sized to have a nearly equal static-pressure drop per foot (meter) of duct length. The result is that the static pressure is very high near the discharge of the fan, and steadily decreases until it is very low at the inlet to the farthest VAV terminal unit.

Figure 63. Trane Ductulator®

The equal friction method is often used because the calculations are simple, and are easily performed using a hand-held duct sizing calculator like the Trane Ductulator® (Figure 63). Drawbacks include the possibility of a higher total pressure drop and higher operating costs.

Static pressures throughout the duct system are balanced at design airflow through the use of balancing dampers, but will not remain balanced at part-load airflows. For this reason, the equal friction method is better suited for constant-volume systems than for VAV systems.

If the equal friction method is used for the main supply ductwork in a VAV system, the VAV terminal units should have pressure-independent control capability and possibly some balancing dampers to avoid excessive flow rates when upstream duct pressures are high.

Static regain method

For more information on the static regain method of duct design, refer to the Trane application manual titled *Variable-Air-Volume Duct Design* (AM-SYS-6). For more information on the VariTrane™ Duct Designer software, which can use either the equal friction or static regain method; visit www.trane.com.

Duct systems that are designed using the static regain method strive to maintain a fairly constant static pressure in each section of the entire duct system. With this method, static pressure is "regained" as the duct size decreases by converting velocity pressure to static pressure.

Advantages of using the static regain method include the possibility of reduced overall static pressure drop and lower fan operating costs, and more equally balanced pressures throughout the system. The drawback of this method is the time-consuming, iterative calculation procedure. For large systems, this often requires the use of a computer program.

The static regain method is recommended for sizing the main supply ducts upstream of the VAV terminal units. The higher air velocity increases the benefit of "regaining" static pressure. Because of the lower pressures associated with the ductwork downstream of the VAV terminal units, these benefits are not as significant. So, the equal friction method is often used for this portion of the system.

Best practices for the design and layout of the 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 titled *HVAC Systems Duct Design*.

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 downstream of the discharge from the air-handling unit, before the first elbow, junction, etc.*
  Satisfactory fan performance and distribution of air throughout the system requires unrestricted and relatively uniform airflow from the discharge of the air-handling unit. This first section of supply ductwork should be straight for at least three duct diameters to allow turbulence to decrease and for a uniform velocity profile to develop before the air encounters an elbow or junction.

  However, when jobsite conditions dictate that an elbow or junction be installed near the air-handling unit, consider using a plenum fan or adding a discharge plenum downstream of a housed fan. In either case, the plenum can often eliminate the need for the elbow or junction by allowing for straight duct takeoffs in multiple directions (Figure 64). This reduces pressure losses and the associated impact on fan energy and sound.

Figure 64. Discharge plenum with multiple, straight connections

- *Place main duct runs and, when possible, branch runs and VAV terminal units, above hallways and other "unoccupied" areas.*
  This typically eases installation and maintenance and helps minimize sound radiated to the occupied spaces.
- *Limit the use of flexible ductwork upstream of VAV terminal units.*
  While flexible ductwork has many benefits, improper use can cause numerous problems in a VAV system. Flexible ductwork causes turbulent airflow and relatively large static pressure loss. In addition, using flexible ductwork at the inlet of a VAV terminal unit may impact the accuracy and consistency of the flow sensor, due to the turbulence it causes. Therefore, the use of flexible ductwork upstream of VAV terminal units should be kept to an absolute minimum.

  Ideally, flexible ductwork should only be used downstream of the VAV terminal units. All runs of flexible ductwork should be kept as short as possible, even though the ease of installation is an enticement to push the limits of acceptable practice.

- *If needed, reducers should be located several duct diameters upstream of VAV terminal units.*
  A reducer (a transition that reduces the size of the duct) causes a significant drop in static pressure (velocity pressure increases as air velocity increases, then static pressure decreases as velocity pressure increases) and it creates turbulence. The turbulence is greatest just downstream of the reducer. If installed too close to the inlet of the VAV terminal unit, this turbulence can impact the accuracy and consistency of the flow sensor, possibly increasing fan energy use.

  After the VAV terminals have been selected and approved for purchase, the design engineer should coordinate with the installing contractor. Installing an upstream duct that is the same diameter as the inlet to the VAV terminal will eliminate the need for a reducer.

  If a reducer is needed, locate it at least three duct diameters upstream of the inlet to the VAV terminal unit. This allows most of the turbulence to dissipate before air enters the VAV terminal unit, improving airflow measurement accuracy and control stability. Be sure to consider the pressure loss of reducers in the sizing of the supply ductwork and selection of the supply fan.

- *Add balancing dampers to the runout ducts downstream of the VAV terminal units.*
  This allows adjustment to deliver the desired airflow to each diffuser served by the VAV terminal. Many types of diffusers are available with an integral balancing damper to simplify installation.

  Balancing dampers may also be necessary upstream of some VAV terminals, especially those located closest to the supply fan. Pressure-independent control and using the static regain method may reduce the need for upstream balancing dampers.

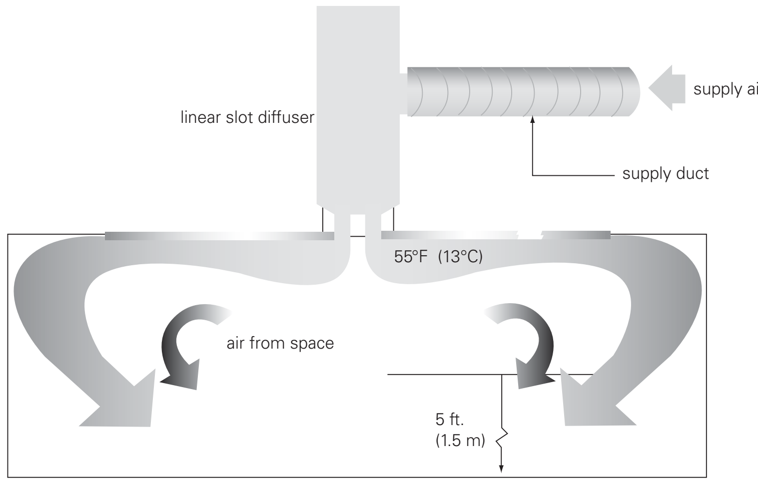
Supply-air diffusers

Each VAV terminal unit is connected, via sheet metal or flexible ductwork, to remotely located diffusers. Supply-air diffusers are used to introduce supply air into the conditioned space. Proper air diffusion is an important comfort consideration, especially in VAV systems. Because most VAV systems require the diffuser to deliver air into the space over a wide range of airflows, diffusers that are specifically intended for use in VAV applications should be used to increase circulation and prevent cold air “dumping” at lower airflow rates.

Figure 65. Linear slot diffuser

Linear slot diffusers (Figure 65) are designed to effectively distribute air over a wide range of airflows, making them the preferred diffuser for VAV systems. They use a principle known as the Coanda effect to distribute air into the conditioned space (Figure 66). When air is discharged at a relatively high velocity along the surface of the ceiling, it creates an area of low pressure that causes the supply air to hug the ceiling and air from the space to be drawn into, and mixed with, the supply air stream. This increases air circulation and allows the air to reach an average temperature before it settles to the occupied levels of the space.

Figure 66. Coanda effect of linear slot diffusers



Best practices for locating supply-air diffusers

For more information on space air diffusion, refer to:

- 2005 *ASHRAE Handbook—Fundamentals*, Chapter 33 (www.ashrae.org)
- ASHRAE's "Designer's Guide to Ceiling-Based Air Diffusion" (www.ashrae.org)

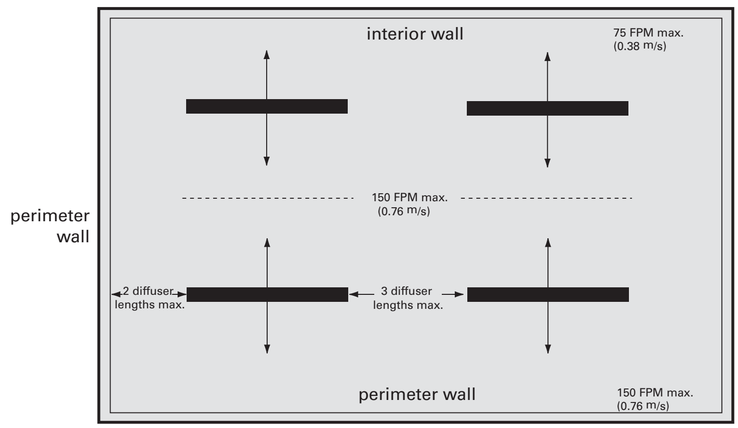
Proper selection and placement of supply-air diffusers generates air movement throughout the conditioned space, eliminating areas of stagnant and stratified air. Other publications contain more complete details related to sizing and locating supply-air diffusers, but the following are a few general recommendations:

- *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 the diffuser's performance when delivering cool air to the zone.
- *Keep air throw as long as possible to maximize the effectiveness of air diffusion.*
  For a VAV system, some amount of "overthrow" at design airflow is often acceptable in order to improve performance at reduced airflows.
- *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 15°F (8.3°C).*
  Limiting the supply-air temperature during heating avoids excessive temperature stratification when supplying warm air from overhead. This may also increase the zone air-distribution effectiveness used to determine required ventilation airflow (see "Impact of zone air-distribution effectiveness," p. 102).
- *In perimeter zones with high heat loss through the building envelope, discharge diffusers to "blanket" the perimeter wall or window area (Figure 60, p. 65).*
  This helps prevent downdraft problems that can occur when large volumes of heated air are distributed through ceiling-mounted diffusers.

In addition, when using linear slot diffusers:

- *Maintain recommended separation between diffusers (Figure 67).*
  The maximum recommended end-to-end distance between linear slot diffusers is three diffuser lengths. The maximum recommended distance between the end of a linear slot diffuser and a perimeter wall (when airflow is parallel to the wall) is two diffuser lengths.

Figure 67. Maximum separation distances and collision velocities

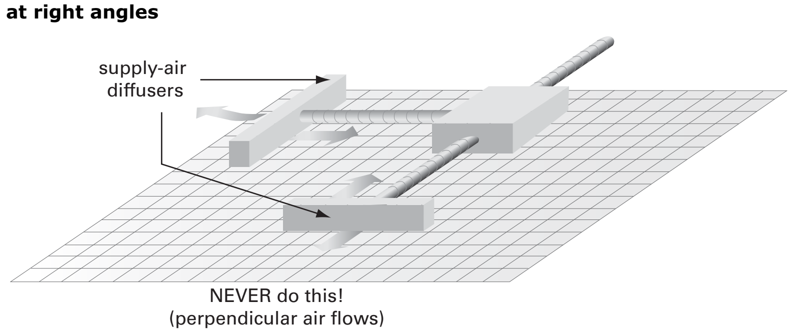


- *Avoid high collision velocities (Figure 67).*
  The collision velocity is the speed at which moving air meets a wall or another air stream. Excessively high collision velocities often result in uncomfortable drafts for the occupants.

  For perimeter walls, the collision velocity should not exceed 150 fpm (0.76 m/s). For interior walls, the collision velocity should not exceed 75 fpm (0.38 m/s). When two air streams collide, the collision velocity (determined by adding the velocities of both air streams at the point of collision) should not exceed 150 fpm (0.76 m/s).

- *Locate linear slot diffusers to maintain parallel flows, and avoid air streams colliding at a right angle (Figure 68).*
  This improves air circulation and avoids high collision velocities.

Figure 68. Avoid locating linear slot diffusers that cause air streams to collide at right angles



Return-air path

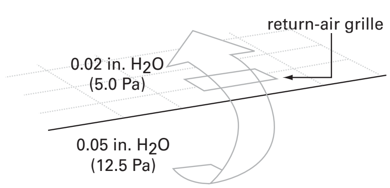
Because a VAV system uses a central supply fan to serve many zones, a path is needed to allow air to return from the zones back to the air-handling unit. Air typically leaves the zones through ceiling-mounted return-air grilles and travels through the open ceiling plenum to either the mechanical room (when floor-by-floor indoor air-handling units are used) or a central, vertical air shaft (when outdoor air-handling units are used). A return-air duct may be used to direct the air returning from the open ceiling plenum into the mechanical room or up the vertical shaft (Figure 62, p. 71).

Open ceiling plenum versus fully ducted return

Most VAV systems use the open plenum space above the ceiling to return air from the zones. This minimizes installed cost and lowers the pressure loss through the return-air path.

Alternatively, some applications use sheet metal ductwork from the return-air grilles all the way back to the return-air opening of the air-handling unit. This increases installed cost, adds more pressure loss that the fans need to overcome (typically requiring a return fan, and possibly VAV terminals in the return duct system), and makes the system more difficult to balance and control. 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.

Figure 69. Static pressure in plenum versus occupied space



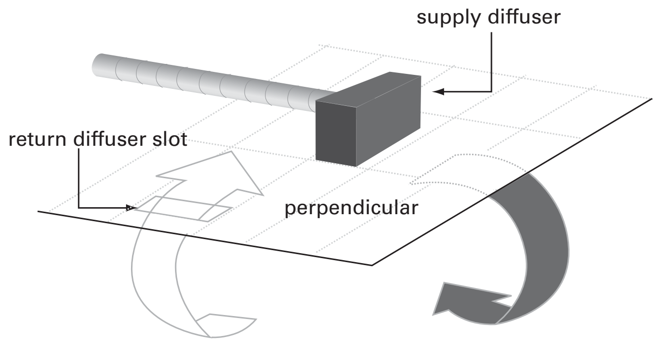
Some in the industry recommend using a ducted return to prevent moisture-related problems in the ceiling plenum. They claim that using an open plenum for the return-air path causes the air pressure inside the plenum to be lower than outdoors, resulting in humid air being drawn in from outdoors. However, this problem can be minimized or avoided by controlling building pressure and properly sizing return-air grilles. When properly sized, the pressure drop through a return-air grille is 0.02 to 0.03 in. H_2O (5.0 to 7.5 Pa). This means that the pressure in the ceiling plenum will be 0.02 to 0.03 in. H_2O (5.0 to 7.5 Pa) less than the pressure in the occupied space. Consider this example (Figure 69): If building pressure is controlled to 0.05 in. H_2O (12.5 Pa) above the outdoor pressure, then the pressure in the ceiling plenum will be 0.02 in. H_2O (5.0 Pa) *higher* than outdoors.

Best practices for the return-air path

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

- *When using return-air linear slots (to match the appearance of linear slot supply-air diffusers), place them perpendicular to the supply-air slot diffusers (Figure 70).*
  This helps avoid supply air from bypassing the occupied portion of the space and allows for proper air circulation.

Figure 70. Place return-air slots perpendicular to supply-air slot diffusers



- *Avoid undersizing return-air grilles.*
  If the return-air openings are too small, they create too much 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 typically causes 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.*
  When 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 (Table 10).*
  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, or when space-to-space pressure differentials must be controlled.

Table 10. Open-plenum versus fully ducted return-air path

Advantages	Disadvantages
Open plenum return	
• Lower installed cost (little or no return ductwork, smaller fan motor due to lower pressure drop through the return-air path) • Often results in lower supply airflow because some of the heat generated by recessed lights and some of the heat conducted through perimeter walls or the roof is picked up by the return air stream, rather than being transferred through the ceiling into the zone • Less fan energy use due to lower pressure drop through the return-air path (and possibly less supply airflow) • Allows use of barometric relief dampers or a central relief fan rather than a return fan • Often allows the use of a shorter ceiling plenum because space is not required for ducting	• Potential exists to lower the pressure in the ceiling plenum below outdoor pressure, if return-air grilles are not sized properly and building pressure is not properly controlled • Local code may require surfaces and materials used in the plenum to meet a certain fire rating • Space-to-space pressure control is not possible • More difficult to clean than a fully ducted return-air path
Fully ducted return	
Advantages	**Disadvantages**
• May avoid the need for surfaces and materials used in the plenum to meet a certain fire rating required by local code • While still difficult to clean due to limited access, a fully ducted return-air path is easier to clean than an open ceiling return • Allows the system to be designed for space-to-space pressure control, which may be required in certain areas of healthcare facilities and in some laboratories	• Higher installed cost due to more return ductwork, a larger fan motor to offset the increased pressure drop through the return-air path, and possibly VAV terminals in the return duct system • More fan energy use due to higher pressure drop through the return-air path • Often dictates the need to use a return fan, rather than barometric relief dampers or a central relief fan • Often requires a taller ceiling plenum to allow space for ducting

Chilled-Water System

For more information on chilled-water systems, refer to the Trane application manuals titled *Chiller System Design and Control* (SYS-APM001-EN) and *Absorption Chiller System Design* (SYS-AM-13).

The chilled-water cooling coil, located inside each VAV air-handling unit, is connected to one or more water chillers by the chilled-water distribution system.

Types of water chillers

Water chillers are used to cool water that is subsequently transported to the chilled-water cooling coils by pumps and pipes.

The refrigeration cycle is a key differentiating characteristic between chiller types. Vapor-compression water chillers use a compressor (reciprocating, scroll, helical-rotary or screw, centrifugal) to move refrigerant around the chiller. The most common energy source to drive the compressor is an electric motor. Absorption water chillers do not have a mechanical compressor, but instead use heat to drive the refrigeration cycle. Steam, hot water, or the burning of oil or natural gas are the most common energy sources for absorption chillers.

Air-cooled versus water-cooled

Besides the refrigeration cycle and type of compressor, the most distinctive difference is the type of condensing used—air-cooled versus water-cooled (Figure 71).

Figure 71. Air-cooled versus water-cooled chiller

A packaged air-cooled chiller contains all the refrigeration components (compressor, air-cooled condenser, expansion device, evaporator, and interconnecting refrigerant piping), wiring, and controls. The components are usually assembled and tested in the factory, which can result in faster installation and improved system reliability. Air-cooled chillers are typically available in packaged chillers ranging from 7.5 to 500 tons (25 to 1,760 kW).

A packaged water-cooled chiller also contains all the refrigeration components, assembled and tested in the factory. But a water-cooled system requires the design and installation of a condenser-water system (piping, pumps, cooling tower, and associated controls). Packaged water-cooled chillers are typically available from 10 to 3,800 tons (35 to 13,000 kW).

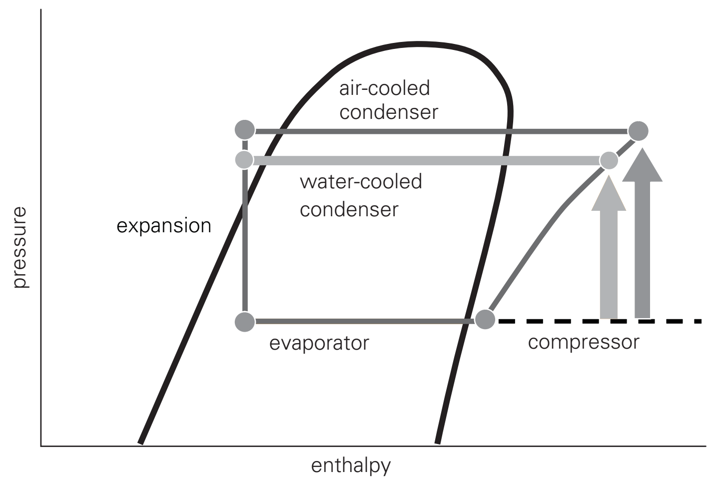
Air-cooled chillers are popular because of their simplicity and convenience, due to the elimination of the cooling tower and condenser-water distribution system. This eliminates the maintenance, chiller condenser-tube cleaning, freeze protection, and water treatment associated with a cooling tower. In addition, it eliminates the concern for the availability and quality of makeup water. This often favors air-cooled chillers in areas that have an inadequate or costly water supply, or where the use of water for the purpose of air conditioning is restricted.

On the other hand, water-cooled chillers typically last longer than air-cooled chillers, since an air-cooled chiller is typically installed outdoors, whereas a water-cooled chiller is installed indoors. In addition, water-cooled chillers are typically more energy efficient.

In an air-cooled chiller, the condensing temperature of the refrigerant is dependent on the *dry-bulb* temperature of the ambient air. For example, if the ambient dry-bulb temperature is 95°F (35°C), the refrigerant condensing temperature might be 125°F (52°C).

In a water-cooled chiller, however, the refrigerant condensing temperature is dependent on the condenser-water temperature, which is dependent on the *wet-bulb* temperature of the ambient air. For the same 95°F (35°C) ambient dry-bulb temperature, if the wet-bulb temperature is 78°F (26°C), the cooling tower likely delivers 85°F (29°C) water to the water-cooled condenser, and the refrigerant condensing temperature might be 105°F (40°C). This lower condensing temperature (pressure) reduces the amount of work required by the compressors (Figure 72), which reduces compressor energy use. This efficiency advantage, however, may lessen at part-load conditions because the dry-bulb temperature tends to drop off faster than the wet-bulb temperature.

Figure 72. Comparison of condensing pressures



Finally, when comparing the efficiency of a water-cooled versus air-cooled chilled-water system, be sure to consider the impact of cooling tower fan and condenser-water pump energy in the water-cooled system, and condenser fan energy in the air-cooled system. Performing a comprehensive energy analysis is the best method of estimating the operating-cost difference.

Chilled-water and condenser-water distribution

The chilled-water distribution system consists of piping, pumps, valves, an air separator and expansion tank, and other accessories. When a water-cooled chiller is used, the condenser-water distribution system consists of piping, pumps, valves, and other accessories.

Design temperatures and flow rates

For more information on "low-flow" chilled-water systems and condenser-water systems, refer to the Trane application manual titled *Chiller System Design and Control* (SYS-APM001-EN).

It is important to remember that water temperatures and flow rates are variables. They should be selected to design an efficient and flexible water distribution system.

Table 11 shows an example of selecting a chilled-water cooling coil in a 13,000-cfm (6.1-m^3/s) VAV air-handling unit. The left-hand column shows the performance of this coil when it is selected with a 44°F (6.7°C) entering fluid temperature and a 10°F (5.5°C) fluid temperature rise (ΔT). To provide the required 525 MBh (154 kW) of cooling capacity, the coil requires 105 gpm (6.6 L/s) of water.

The right-hand column shows the performance of same coil, but in this case it is selected with 40°F (4.4°C) entering fluid and a 15.5°F (8.7°C) ΔT. To provide the equivalent capacity, the coil requires only 67.5 gpm (4.3 L/s) of water.

Table 11. Impact of entering fluid temperature and flow rate on cooling coil

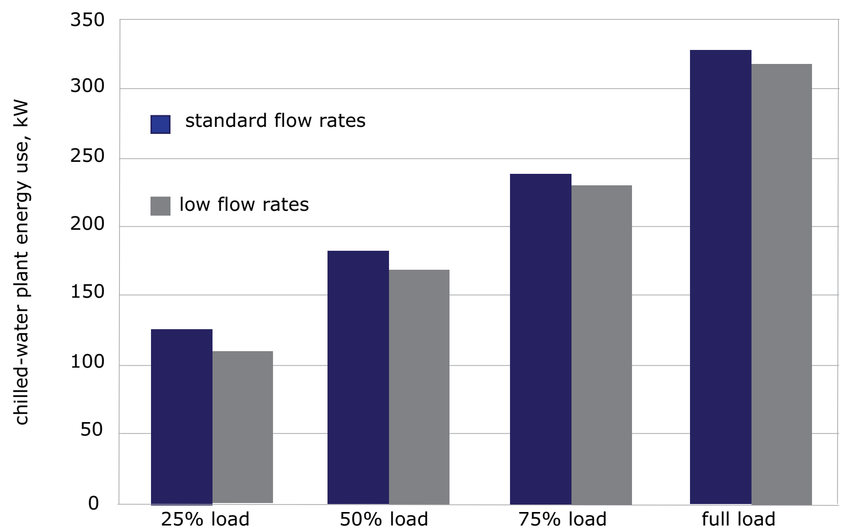
	"Conventional" system design	"Low-flow" system design
Coil face area, ft^2 (m^2)	29.90 (2.78)	29.90 (2.78)
Face velocity, fpm (m/s)	435 (2.2)	435 (2.2)
Coil rows	6 rows	6 rows
Fin spacing, fins/ft (fins/m)	83 (272)	83 (272)
Total cooling capacity, MBh (kW)	525 (154)	525 (154)
Entering fluid temperature, °F (°C)	44 (6.7)	40 (4.4)
Leaving fluid temperature, °F (°C)	54 (12.2)	55.5 (13.1)
Fluid ΔT, °F (°C)	10 (5.5)	15.5 (8.7)
Fluid flow rate, gpm (L/s)	105 (6.6)	67.5 (4.3)
Fluid pressure drop, ft H_2O (kPa)	13.6 (40.6)	6.2 (18.4)

By lowering the entering fluid temperature, this coil can deliver the same cooling capacity with 36 percent less flow, at less than half of the fluid pressure drop, with no impact on the airside system.

For water-cooled systems, this "low-flow" strategy can also be used for the condenser-water distribution system.

The benefits of applying this coil in a "low-flow" chilled-water system include reduced installed cost (lower flow rates can allow the pipes, fittings, pumps, valves, and cooling tower to be smaller) and/or reduced pumping energy (since the system has to pump less water). And, although the chiller will consume more energy to make the colder water, in most applications, the total *system* (chillers, pumps, and cooling tower for water-cooled systems) energy consumption will be reduced (Figure 73).

Figure 73. Chilled-water system energy use: standard flow versus low flow



Standard flow conditions are 2.4 gpm/ton (0.043 L/s/kW) chilled water and 3.0 gpm/ton (0.054 L/s/kW) condenser water. Low flow conditions are 1.5 gpm/ton (0.027 L/s/kW) chilled water and 2.0 gpm/ton (0.036 L/s/kW) condenser water.

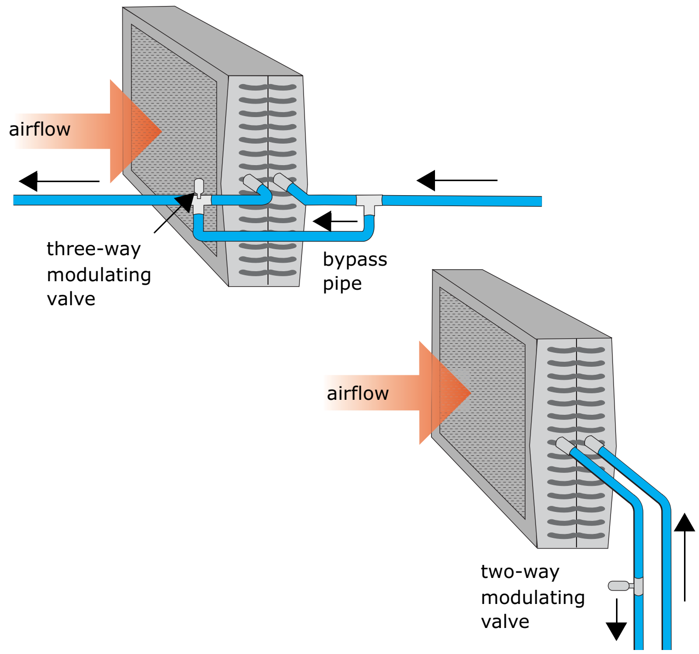
If further energy savings is desired, consider keeping the same-sized pipes (not downsizing them for installed cost savings). This also improves the ability of the chilled-water system to respond to possible future increases in load, since the pipes will be capable of handling an increased flow rate if needed.

Control valve selection

The purpose of the control valve on a chilled-water coil in a VAV air-handling unit is to vary (modulate) the flow of chilled water through the coil to maintain the desired discharge-air temperature.

Either a three-way or two-way control valve can be used (Figure 74). As a three-way control valve modulates to allow less water to flow through the coil, thus decreasing its capacity, the excess water bypasses the coil and mixes downstream with the water that flows through the coil. A two-way modulating valve does not bypass any unused water. It simply throttles the amount of water passing through the coil. The coil experiences no difference in the cooling effect of using a three-way versus a two-way valve. The chilled-water distribution system, however, sees a great difference (Table 12, p. 84).

Figure 74. Three-way versus two-way control valves



With a three-way valve, the "terminal" water flow rate (water flowing through the coil plus the water bypassing the coil) is relatively constant at all loads. Therefore, pumping energy will remain relatively constant regardless of the cooling load. In addition, because cold water is bypassing the coil the temperature of the water returning to the chiller decreases as the zone cooling load decreases.

With a two-way valve, however, the "terminal" water flow varies proportionately with the load. This provides the opportunity to significantly reduce pumping energy at part load. Because there is no mixing of coil and bypassed water, the temperature of the water leaving the terminal remains relatively constant at all load conditions.

Table 12. Three-way versus two-way control valves

Characteristics of system that uses three-way, modulating control valves	Characteristics of system that uses two-way, modulating control valves
• Water flow rate through each load terminal (water flowing through the coil plus water bypassing the coil) remains relatively constant at all load conditions, which results in relatively constant pumping energy • Temperature of the water returning to the chiller decreases as the cooling load decreases • Water-flow balance is critical to ensure proper operation because flow is constant	• Water flow rate through each load terminal varies proportionately to the load, providing the opportunity to significantly reduce pumping energy at part load • Temperature of the water returning to the chiller remains relatively constant as the cooling load decreases • A variable-flow system is less sensitive to water balance than most constant-flow systems

The most common approach to select the control valve is the valve flow coefficient (Cv or Kv):

$$Cv = \frac{Q}{\sqrt{\Delta P / SG}} \qquad \left(Kv = \frac{36 \times Q}{\sqrt{\Delta P / \rho}} \right)$$

where,

Cv = valve flow coefficient (Kv)
Q = fluid flow rate, gpm (L/s)
ΔP = pressure drop across the valve, psi (kPa)
SG = specific gravity of the fluid (1.0 for water)
ρ = density of fluid, kg/m^3 (1,000 kg/m^3 for water)

In general, the pressure drop (ΔP) across the valve should be equal to, or slightly greater than, the pressure drop through the chilled-water coil (obtained from the manufacturer).

For example, consider a chilled-water coil that has a design flow rate of 68 gpm (4.3 L/s) with a pressure drop of 2.5 psi (17.2 kPa). Ideally the pressure drop across the valve should be equal to the pressure drop through the coil. Therefore, if the fluid is pure water, the desired valve flow coefficient is 43.0 (1180).

$$Cv = \frac{68 gpm}{\sqrt{2.5 psi / 1.0}} = 43.0 \qquad \left(Kv = \frac{36 \times 4.3 L/s}{\sqrt{17.2 kPa / 1000 kg/m^3}} = 1180 \right)$$

If a valve with a smaller Cv (Kv) is selected, the pressure drop through that valve will be larger than the pressure drop through the coil, at design flow. This is generally considered good for valve controllability, but make sure that the pump can handle this increase in pressure drop. If this valve is located near the pump, the added pressure drop will probably not impact the size of the pump. On the other hand, if this valve is part of the “critical” path (highest pressure drop path), the added pressure drop through the valve may necessitate the selection of a larger pump.

If a valve with a larger Cv (Kv) is selected, the pressure drop through the valve will be less than the pressure drop through the coil. If this difference in pressure drops is too large, it could result in poor controllability (low valve authority).

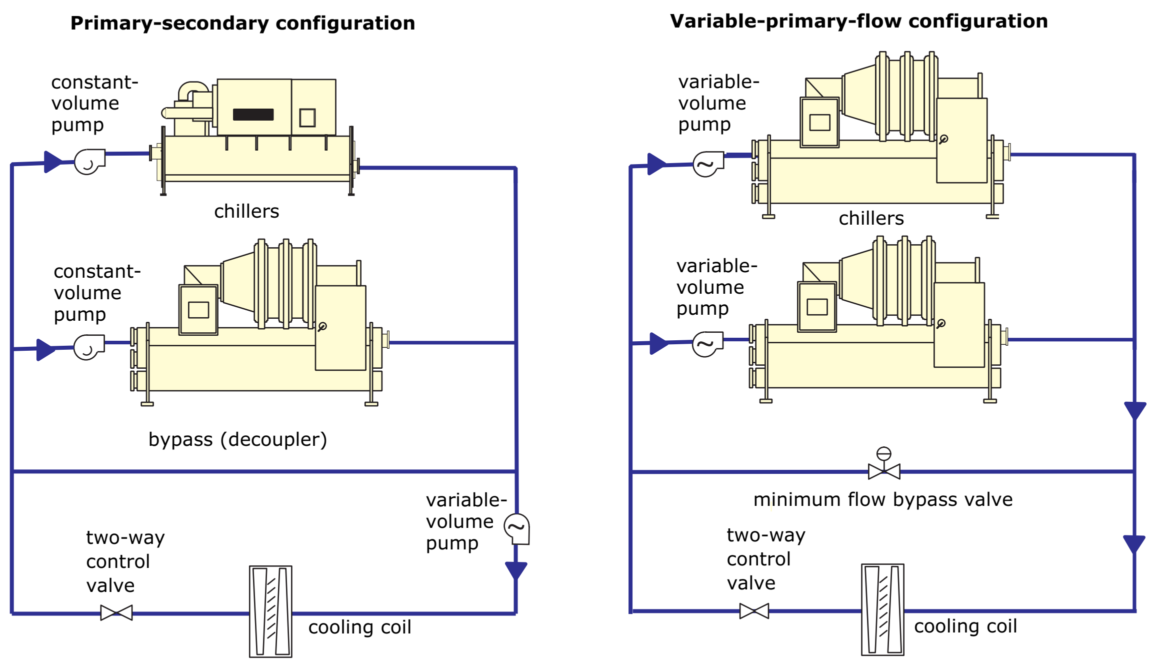
For more information on variable-flow chilled-water or condenser-water systems, refer to the Trane application manual titled *Chiller System Design and Control* (SYS-APM001-EN).

Variable- versus constant-flow pumping

As previously mentioned, using two-way control valves results in variable water flow through the chilled-water system, which provides the opportunity to significantly reduce pumping energy at part load.

A variable-flow chilled-water distribution system (Figure 75), however, requires the chillers be equipped to handle variable water flow (in a variable-primary-flow configuration) or the system be designed to provide constant water flow through the chillers (such as a primary-secondary, or decoupled, configuration).

Figure 75. Variable-flow chilled-water systems



While much less common, the condenser-water distribution system can also be designed for variable flow. Reducing the condenser-water flow rate does reduce pumping energy, but it increases chiller energy use and can sometimes increase cooling tower energy use. As the flow rate through the chiller condenser decreases, the temperature of the water leaving the condenser (and, therefore, the refrigerant condensing pressure) increases. This increases compressor lift and energy use. In addition, the condenser-water flow rate must be maintained above the minimum required for proper operation of the chiller, cooling tower, and pumps (motor cooling). For these reasons, controlling a variable-flow condenser-water system to minimize overall energy use is complicated. Instead, many projects choose to design a "low-flow" condenser-water system to minimize pump energy and simplify system-level control (see "Design temperatures and flow rates," p. 81).

Freeze prevention for the chilled-water distribution system

Adding antifreeze (such as glycol) to the chilled-water system lowers the temperature at which the solution will freeze. Given a sufficient concentration of glycol, no damage to the system will occur.

As the temperature drops below the glycol solution's 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. "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 13).

Table 13. Concentration required for freeze protection vs. burst protection

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

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

For a chilled-water VAV system, since the cooling coil is typically shut off during sub-freezing weather, burst protection is usually sufficient. Freeze protection is mandatory in those cases where no ice crystals can be permitted to form (such as a coil loop that operates during very cold weather) or where there is inadequate expansion volume available.

When an air-cooled chiller is used, an alternative approach is to use a packaged condensing unit (condenser and compressor) located outdoors, with a remote evaporator barrel located in an indoor equipment room. The two components are connected with field-installed refrigerant piping. This configuration locates the part of the system that is susceptible to freezing (evaporator) indoors and still uses an outdoor air-cooled condenser.

Freeze prevention for the condenser-water distribution system

In a water-cooled system, the cooling tower and sections of the condenser-water piping located outdoors are often drained during cold weather to avoid freezing. Alternatively, some systems use special control sequences, sump basin heaters, heat tape on outdoor piping, or even locate the sump indoors to prevent freezing.

In another configuration, the cooling tower can be located indoors, with outdoor air ducted to and from the tower. This typically requires the use of a centrifugal fan to overcome the static-pressure losses due to the ductwork, which increases tower fan energy are compared to a conventional outdoor cooling tower with propeller fans.

For systems with year-round cooling requirements that cannot be met with an airside economizer, air-cooled chillers are often used. Air-cooled condensers have the ability to operate in below-freezing weather, and can do so without the problems associated with operating the cooling tower in these conditions. Alternatively, a dry cooler can use the cold ambient air to cool the water directly, without needing to operate a chiller or cooling tower.

Condenser heat recovery

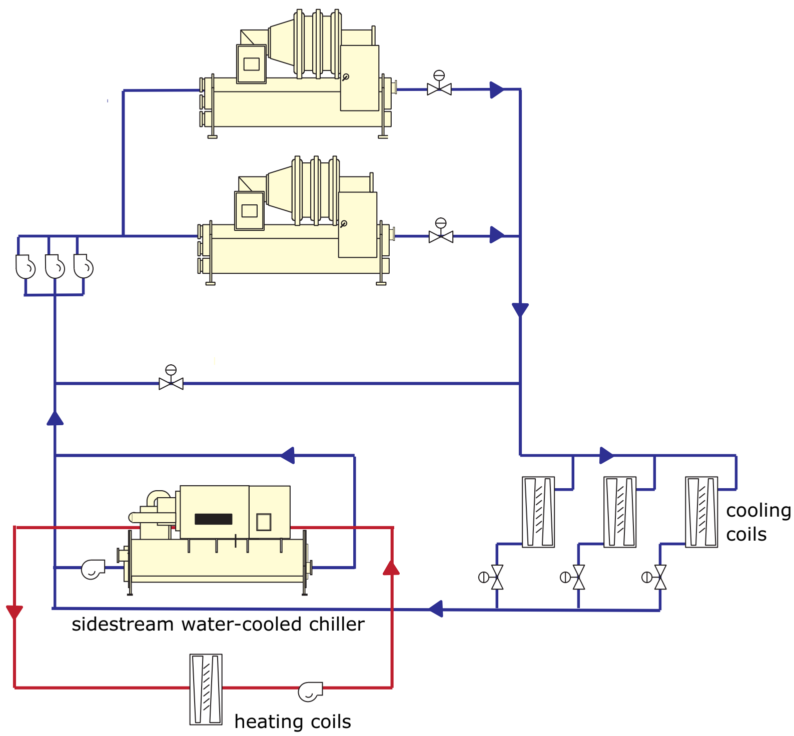
For more information on employing heat recovery in a water-cooled chilled-water system, refer to the Trane application manual titled *Waterside Heat Recovery in HVAC Systems* (SYS-APM005-EN).

A water-cooled chiller rejects heat to the condenser water, then the cooling tower rejects this heat to the outdoors. An air-cooled chiller rejects heat from the hot refrigerant vapor leaving the compressor to the outside air flowing across the air-cooled condenser coils. In some systems, all or part of this heat can be recovered and used within the facility, rather than rejecting it to the outdoors.

Locating a water-cooled chiller in the "sidestream" configuration (Figure 76) allows heat rejected from the condenser to be used within the facility, while chiller capacity is controlled to provide only the amount of condenser heat required at a given time. Alternatively, some water-cooled chillers can be equipped with a separate heat-recovery condenser bundle to allow for simultaneous heat recovery and heat rejection.

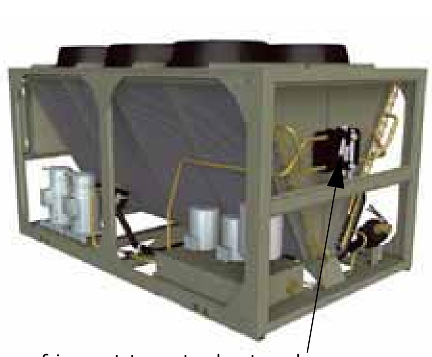
Some air-cooled chillers are available with an integral refrigerant-to-water heat exchanger (Figure 77), which recovers heat from the hot refrigerant vapor for use within the facility.

Figure 76. Water-cooled chiller in sidestream configuration for heat recovery



In a chilled-water VAV system, this recovered heat is typically used for preheating domestic (or service) hot water or for reheat (or tempering) at VAV terminals. Many times, reheat loads can be met with water temperatures of 105°F (40.6°C) or lower, because the heat required for tempering air is minimal. This is an ideal application for heat recovery from any chiller.

Figure 77. Air-cooled chiller with heat-recovery heat exchanger



The example in Table 14 depicts a VAV terminal selected to deliver 2000 cfm (0.94 m^3/s) at cooling design conditions, with a minimum airflow setting of 600 cfm (0.28 m^3/s).

During reheat (tempering) mode, the heating coil needs to reheat the 600 cfm (0.28 m^3/s) of cool primary air just enough to avoid overcooling the space. When the space cooling load is zero (when the "reheat" load is highest), it requires heating the air from 55°F (12.8°C) to 75°F (23.9°C) in this example.

In order to use 105°F (40.6°C) water for reheat, the VAV terminal unit must be equipped with a two-row heating coil. Table 14 compares the performance of a two-row coil, operating in the reheat mode, to a one-row coil operating at the same condition using 150°F (65.6°C) water. While the two-row coil is able to provide the required reheat capacity with the lower water temperature, both the airside pressure drop and fluid flow rate are higher, which increase fan and pumping energy.

Table 14 also compares the performance of the same coils operating at design heating conditions—delivering 90°F (32.2°C) air to the space. The two-row coil is able to provide the required heating capacity using a lower hot-water temperature—150°F (65.6°C) compared to 180°F (82.2°C) for the one-row coil. When combined with the use of a condensing boiler, this lower water temperature can significantly increase the efficiency of the heating system (see "Non-condensing versus condensing boilers," p. 92). In addition, the fluid pressure drop due to the coil is larger for the single-row coil, increasing the size and energy use of the pumps.

Table 14. Using condenser-water heat recovery and 105ºF (40.6ºC) water for reheat (tempering)

	2-row coil	1-row coil
Airside pressure drop at design cooling airflow, in H_2O (Pa)	0.79 (196)	0.45 (112)
Operating in reheat (tempering) mode		
	2-row coil	**1-row coil**
Entering air temperature, ºF (ºC)	55 (12.8)	55 (12.8)
Leaving air temperature, ºF (ºC)	75 (23.9)	75 (23.9)
Heating capacity, MBh (kW)	13.0 (3.8)	13.0 (3.8)
Entering fluid temperature, ºF (ºC)	105 (40.6)	150 (65.6)[2]
Returning fluid temperature, ºF (ºC)	91 (32.8)	107 (41.9)
Coil flow rate, gpm (L/s)	1.88 (0.12)	0.61 (0.04)
Fluid pressure drop, ft H_2O (kPa)	0.23 (0.68)	0.29 (0.85)
Airside pressure drop at reheat airflow[1], in H_2O (Pa)	0.07 (17.6)	0.04 (10.1)
Operating at design heating conditions		
	2-row coil	**1-row coil**
Entering air temperature, ºF (ºC)	55 (12.8)	55 (12.8)
Leaving air temperature, ºF (ºC)	90 (32.2)	90 (32.2)
Heating capacity, MBh (kW)	22.8 (6.7)	22.8 (6.7)
Entering fluid temperature, ºF (ºC)	150 (65.6)[3]	180 (82.2)[2]
Returning fluid temperature, ºF (ºC)	116 (46.7)	146 (63.3)
Coil flow rate, gpm (L/s)	1.34 (0.085)	1.33 (0.084)
Fluid pressure drop, ft H_2O (kPa)	0.12 (0.35)	1.10 (3.3)
Airside pressure drop at heating airflow[1], in H_2O (Pa)	0.07 (17.6)	0.04 (10.1)

1 Assumes airside pressure drop changes with the square of the airflow reduction: design cooling airflow = 2000 cfm (0.94 m^3/s), reheat/heating airflow = 600 cfm (0.28 m^3/s).

2 Assumes the hot-water supply temperature has been reset from 180ºF (82.2ºC) to 150ºF (65.6ºC) during the months when reheat (rather than heating) is needed (see "Hot-water temperature reset," p. 209).

3 The two-row coil is able to provide the required heating capacity using a lower hot-water temperature, which presents the opportunity to use a condensing boiler (see "Non-condensing versus condensing boilers," p. 92).

Some engineers express concern about the increase in air-pressure drop, and this certainly should be considered—it requires higher static pressure and fan horsepower at design conditions. However, because the coil air-pressure difference drops quickly as airflow is reduced, the actual impact on annual fan energy use is small.

In many cases, the energy-related benefits of using waterside heat recovery and condensing boilers will outweigh the increase in annual fan energy. Pumping energy may increase during reheat mode, but will likely decrease during heating mode. The overall impact on annual pumping energy use depends on climate and building operating characteristics.

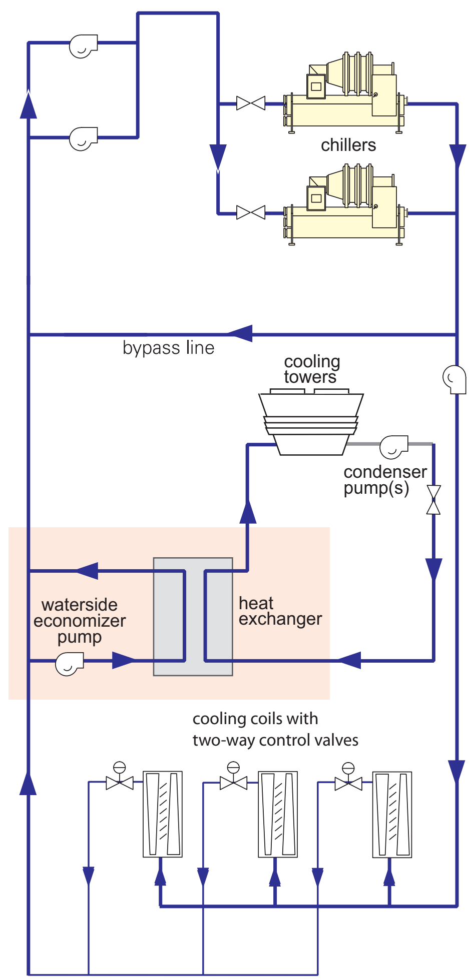
Waterside economizer

For more information on incorporating a waterside economizer into a chilled-water system, refer to the Trane application manual titled *Chiller System Design and Control* (SYS-APM001-EN).

In a chilled-water system, there are several ways to accomplish "free" cooling through the use of a waterside economizer. The three most common approaches are to 1) use a plate-and-frame heat exchanger in the condenser-water system, 2) allow for refrigerant migration inside a centrifugal chiller, or 3) use cool water from a well, river, or lake.

In the case of the plate-and-frame heat exchanger, it is used to keep the chilled-water distribution system separate from the condenser-water system. When the outdoor wet-bulb temperature is low enough, the cooling tower cools the condenser water to a temperature that is colder than the water returning to the chiller(s). In the "sidestream" configuration shown in Figure 78, this plate-and-frame heat exchanger is able to pre-cool the warm water returning from the cooling coils, reducing chiller energy use.

Figure 78. Example waterside economizer using a plate-and-frame heat exchanger



Thermal storage

For more information on adding thermal storage to a chilled-water system, refer to the Trane *Air Conditioning Clinic* titled "Ice Storage Systems" (TRG-TRC019-EN) and the Trane application manuals titled *Ice Storage Systems* (SYS-AM-10) and *Control of Ice Storage Systems* (ICS-AM-4).

Adding thermal storage to the chilled-water system can reduce utility costs by shifting the operation of the chiller from periods when the cost of electricity is high (e.g., daytime) to periods when the cost of electricity is lower (e.g., nighttime). This reduces the electricity required to operate the chiller during the periods of high-cost electricity, and shifts operation of the chiller to the period of low-cost electricity.

The chiller is used during the period of low-cost electricity to cool or freeze water inside storage tanks. During the nighttime hours, the outdoor dry-bulb and wet-bulb temperatures are typically lower than during the day. This allows the chiller to operate at a lower condensing pressure and regain some of the capacity and efficiency lost by producing the colder fluid temperatures needed to "recharge" the storage tanks.

Another potential benefit of thermal storage is to reduce the required capacity of the chiller. When thermal storage is used to satisfy all or part of the cooling load, the chiller may be able to be downsized as long as there is still enough time to recharge the storage tanks.

Hot-Water System

As mentioned earlier, heating in a VAV system can be accomplished using either:

- Baseboard radiant heat installed in the zone
- Heating coils (hot water or electric) in the individual VAV terminal units
- A heating coil (hot water, steam, or electric) or a gas-fired burner that is located inside the VAV air-handling unit

If any of these options uses hot water, a boiler and hot-water distribution system is needed.

Types of hot-water boilers

For more information on the various types of boilers, refer to Chapter 31 (Boilers) of the 2008 *ASHRAE Handbook—HVAC Systems and Equipment* (www.ashrae.org), *The Boiler Book* from Cleaver-Brooks (www.boilerspec.com), or the *Gas Boilers* design guide from the New Buildings Institute (www.newbuildings.org).

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 VAV systems are low-pressure (<160 psig [1100 kPa] and <250°F [120°C]), hot-water boilers.

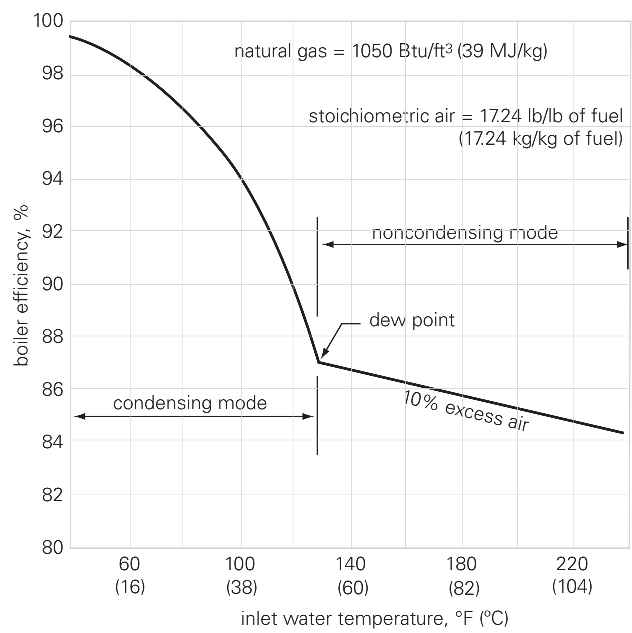
Non-condensing versus condensing boilers

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 140°F (60°C) 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 to over 95 percent, while non-condensing boilers have combustion efficiencies that range from 80 to 86 percent.

Condensing of the flue gases also allows for a lower return-water temperature, much lower than the 140°F (60°C) limit that is common with non-condensing boilers. In fact, the efficiency of a condensing boiler *increases* as the return-water temperature decreases (Figure 79). Therefore, to maximize the efficiency of a condensing boiler, it is important that the rest of the heating system be designed to operate at these lower return-water temperatures.

Figure 79. Impact of return-water temperature on boiler efficiency



Source: 2008 *ASHRAE Handbook—HVAC Systems and Equipment*, Chapter 31, 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 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.

Hot-water distribution

For more information on the design of hot-water distribution systems, refer to Chapters 12 (Hydronic Heating and Cooling System Design) and 46 (Valves) of the 2008 *ASHRAE Handbook—HVAC Systems and Equipment* (www.ashrae.org) or *The Boiler Book* by Cleaver-Brooks (www.boilerspec.com).

The hot-water distribution system consists of piping, pumps, valves, an air separator and expansion tank, and other accessories.

Design temperatures and flow rates

It is important to remember that hot-water temperatures and flow rates are variables. They should be selected to design an efficient and flexible hot-water distribution system.

Many hot-water heating systems are currently designed for 180°F (82.2°C) fluid temperature. Designing the system for a lower temperature may require more coil surface area (more rows, more fins) to deliver the same heating capacity.

Table 15 shows an example of the same VAV terminal unit with heating coils selected to deliver 22.8 MBh (6.7 kW) of capacity using either 180°F (82.2°C) or 150°F (65.6°C) hot water. Note that in order to deliver equivalent capacity, the coil selected for 150°F (65.6°C) hot water requires two rows of tubes rather than one row. This increases the cost of the VAV terminal unit.

Table 15. Comparison of 180ºF (82.2ºC) and 150ºF (65.6ºC) supply-water temperature

	180ºF (82.2ºC) hot water	150ºF (65.6ºC) hot water
Coil rows	1 row	2 rows
Heating capacity, MBh (kW)	22.8 (6.7)	22.8 (6.7)
Coil flow rate, gpm (L/s)	1.33 (0.084)	1.34 (0.085)
Fluid ΔT, °F (°C)	34 (18.9)	34 (18.9)
Returning fluid temperature, ºF (ºC)	146 (63.3)	116 (46.7)
Fluid pressure drop, ft H_2O (kPa)	1.10 (3.3)	0.12 (0.35)
Airside pressure drop at design cooling airflow, in H_2O (Pa)	0.45 (112)	0.79 (196)
Airside pressure drop at heating airflow[1], in H_2O (Pa)	0.04 (10.1)	0.07 (17.6)

1 Assumes airside pressure drop changes with the square of the airflow reduction: design cooling airflow = 2000 cfm (0.94 m^3/s), reheat/heating airflow = 600 cfm (0.28 m^3/s).

However, using a lower entering-water temperature and/or a larger fluid ΔT, results in a lower return-water temperature. In this example, with 150°F (65.6°C) supply-water temperature, the temperature of the water returning to the boiler decreases to 116°F (46.7°C). While this may be too low for a conventional, non-condensing boiler, it can significantly increase the efficiency of a condensing boiler (Figure 79).

In addition, the larger, two-row coil reduces the fluid pressure drop—1.10 ft H_2O (3.3 kPa) for the one-row coil versus 0.12 ft H_2O (0.35 kPa) for the two-row coil—which may decrease pumping power.

Some engineers raise concerns about the increase in airside-pressure drop—0.45 in H_2O (112 Pa) for the one-row coil versus 0.79 in H_2O (196 Pa) for the two-row coil—and its impact on increasing fan power. However, because the airside pressure drop decreases quickly as airflow is reduced at part load, the actual impact on annual fan energy use is typically small.

In addition to the supply-water temperature, the design temperature difference (ΔT) also impacts the cost and energy consumption of a hot-water system. Many hot-water heating systems are currently designed for either a 20°F (11°C) or 30°F (17°C) fluid temperature drop (ΔT). Designing the system for a larger ΔT allows a lower fluid flow rate to deliver the same heating capacity. Lower flow rates can allow the pipes, pumps, and valves to be smaller, reducing system installed cost. In addition, lower flow rates can reduce pumping energy, at both full load and part load. If further energy savings is desired, consider keeping the same-sized pipes (not downsizing them for installed cost savings). This also improves the ability of the hot-water system to respond to possible future increases in load, since the pipes will be capable of handling an increased flow rate if needed.

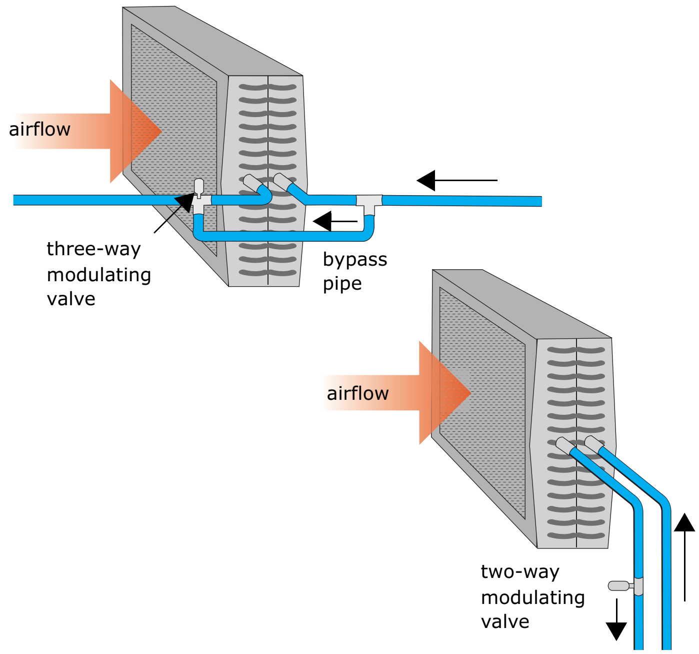
Finally, recognize that hot-water heating coils in VAV terminal units may be used for two purposes: 1) to provide *heated* supply air to offset heating loads in the zone, and 2) to *reheat* the cool supply air to avoid overcooling the zone at low cooling loads. When the zone requires heating, the hot-water coil heats the supply air to a temperature that is significantly warmer than the zone temperature. When the air must be reheated to avoid overcooling, however, the coil warms the supply air to a temperature that is no higher than the zone temperature. The water temperature required for *reheating* can often be much lower than the temperature required for *heating*. Therefore, consider using heat recovery (see “Condenser heat recovery,” p. 88) or temperature-reset control strategies (see “Hot-water temperature reset,” p. 209) to reduce boiler energy use.

Control valve selection

The purpose of the control valve is to vary the flow of water through the coil to maintain space comfort. Hot water coils in VAV terminal units are typically either staged (on/off) or modulated. With staged control, the control valve is either fully opened or fully closed. With modulated control, the position of the control valve is varied to maintain zone temperature.

Either a three-way or two-way control valve can be used (Figure 80). As a three-way control valve modulates to allow less water to flow through the coil, thus decreasing its capacity, the excess water bypasses the coil and mixes downstream with the water that flows through the coil. A two-way modulating valve does not bypass any unused water. It simply throttles the amount of water passing through the coil. The coil experiences no difference in the heating effect of using a three-way versus a two-way valve. The hot-water distribution system, however, sees a great difference (Table 16).

Figure 80. Three-way versus two-way control valves



With a three-way valve, the "terminal" water flow rate (water flowing through the coil plus the water bypassing the coil) is relatively constant at all loads. Therefore, pumping energy will remain relatively constant regardless of the heating load. In addition, because hot water is bypassing the coil the temperature of the water returning to the boiler increases as the zone heating load decreases.

With a two-way valve, however, the "terminal" water flow varies proportionately with the load. This provides the opportunity to significantly reduce pumping energy at part load. Because there is no mixing of coil and bypassed water, the temperature of the water leaving the terminal remains relatively constant at all load conditions.

Table 16. Three-way versus two-way control valves

Characteristics of system that uses three-way, modulating control valves	Characteristics of system that uses two-way, modulating control valves
• Water flow rate through each load terminal (water flowing through the coil plus water bypassing the coil) remains relatively constant at all load conditions, which results in relatively constant pumping energy • Temperature of the water returning to the boiler increases as the heating load decreases • Water-flow balance is critical to ensure proper operation because flow is constant	• Water flow rate through each load terminal varies proportionately to the load, providing the opportunity to significantly reduce pumping energy at part load • Temperature of the water returning to the boiler remains relatively constant as the heating load decreases • A variable-flow system is less sensitive to water balance than most constant-flow systems

The most common approach to select the control valve is the valve flow coefficient (Cv or Kv):

$$Cv = \frac{Q}{\sqrt{\Delta P / SG}} \qquad \left(Kv = \frac{36 \times Q}{\sqrt{\Delta P / \rho}} \right)$$

where,

Cv = valve flow coefficient (Kv)
Q = fluid flow rate, gpm (L/s)
ΔP = pressure drop across the valve, psi (kPa)
SG = specific gravity of the fluid (1.0 for water)
ρ = density of fluid, kg/m^3 (1,000 kg/m^3 for water)

In general, the pressure drop (ΔP) across the valve should be equal to, or slightly greater than, the pressure drop through the hot water coil. (The coil pressure drop can be obtained from the manufacturer of the VAV terminal unit.)

For example, consider a hot water coil that has a design flow rate of 1.5 gpm (0.095 L/s) with a pressure drop of 0.61 psi (4.2 kPa). Ideally the pressure drop across the valve should be equal to the pressure drop through the coil. Therefore, if the fluid is pure water, the desired valve flow coefficient is 1.92 (52.8).

$$Cv = \frac{1.5gpm}{\sqrt{0.61psi / 1.0}} = 1.92 \qquad \left(Kv = \frac{36 \times 0.095 L/s}{\sqrt{4.2kPa / 1000 kg/m^3}} = 52.8 \right)$$

If a valve with a smaller Cv (Kv) is selected, the pressure drop through that valve will be larger than the pressure drop through the coil, at design flow. This is generally considered good for valve controllability, but make sure that the pump can handle this increase in pressure drop. If this valve is located near the pump, the added pressure drop will probably not impact the size of the pump. On the other hand, if this valve is part of the "critical" path (highest pressure drop path), the added pressure drop through the valve may necessitate the selection of a larger pump.

If a valve with a larger Cv (Kv) is selected, the pressure drop through the valve will be less than the pressure drop through the coil. If this difference in pressure drops is too large, it could result in poor controllability (low valve authority).

Variable- versus constant-flow pumping

As previously mentioned, using two-way control valves results in variable water flow through the system, which provides the opportunity to significantly reduce pumping energy at part load.

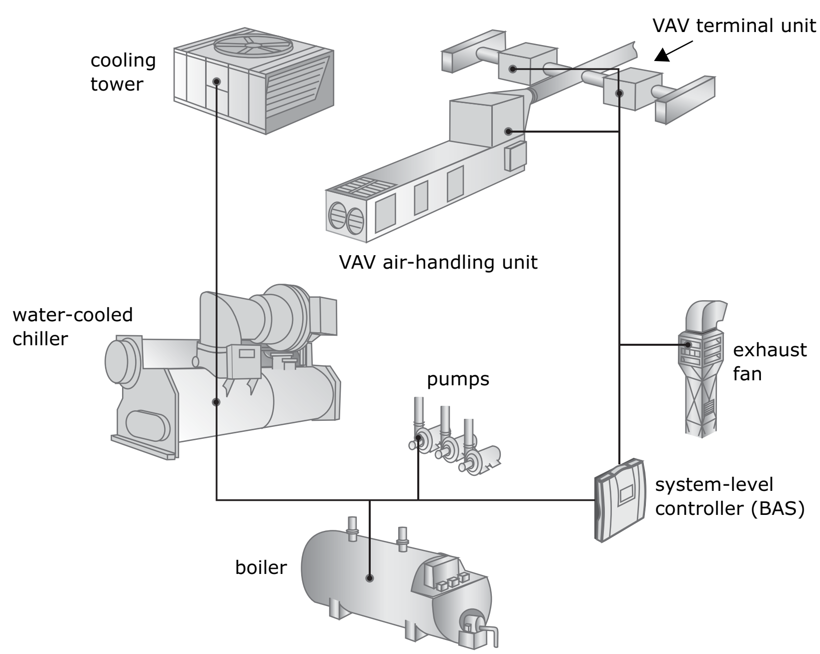
A variable-flow hot-water distribution system, however, may require some means to provide constant water flow through the boilers, or the boilers must be equipped to handle variable water flow. Common approaches include a pressure-actuated bypass valve, primary-secondary pumping system, or a temperature-actuated valve (internal to the boiler) to divert hot water from the supply and mix it with cooler return water.

Controls

The control of a chilled-water VAV 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.

In a typical direct digital control (DDC) system, the VAV air-handling unit, VAV terminal units, water chiller, and boiler (if included) are each equipped with a separate unit-level controller. These unit-level controllers are connected to a centralized, system-level controller (Figure 81). With this configuration, each unit-level controller is capable of performing its functions, even if communication with the system-level controller is lost. A common analogy is to view the individual unit-level controllers as members of an orchestra, and the system-level controller as the conductor.

Figure 81. Control of a chilled-water VAV system



Note: Common unit-level and system-level control functions for a chilled-water VAV system are discussed in detail in the chapter, "System Controls," p. 171. Specific details should be obtained from the equipment or controls manufacturer.

System Design Issues and Challenges

This chapter proposes solutions to several common challenges of designing a chilled-water VAV system. This is not an exhaustive list of all challenges or all solutions, but is meant to cover the most common. The chapter, "System Design Variations," p. 147, addresses several variations on the typical chilled-water VAV system.

Thermal Zoning

Single or multiple air-handling units?

A building may use a few large air-handling units or several smaller units, depending on size, load characteristics, and function. See "VAV Air-Handling Unit," p. 11.

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

- *A single room separated by physical boundaries (walls, windows, doors, floor, and ceiling)*
  The individual offices in an office building or individual classrooms in a school could each be a separate zone. In this case, each office or classroom would be served by a dedicated VAV terminal unit and zone 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 VAV terminal unit would be used to serve the entire group of rooms, and a zone 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 in order 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. Whenever possible, a zone should have definite, physical boundaries (walls, windows, doors, floor, and ceiling). Loss of temperature and humidity control can result if air can be supplied to the zone by a VAV terminal unit other than the one connected to the zone sensor.

Optimizing the number of zones

Definition of a zone

A space or group of spaces within a building with heating and cooling requirements that are sufficiently similar so that desired conditions (e.g., temperature) can be maintained throughout using a single sensor (e.g., temperature sensor).

If a VAV system is designed with too few thermal zones, it may result in undesirable temperature variations for many occupants within the zone. 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 best balances occupant comfort requirements with the budgetary limits of the project. The first step is to determine the maximum number of potential thermal zones, ignoring cost. Each room separated by physical boundaries should be a separate zone. Larger open areas should be divided up 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.

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 sensor where it will be affected by air discharged from a supply-air diffuser.
- Make certain that only the VAV terminal unit that is connected to the zone sensor can influence the temperature being sensed by that sensor. This also means that a zone should only be served by diffusers connected to the VAV terminal that is controlled by the temperature sensor located within that zone. Do not serve a zone with diffusers connected to two separately controlled VAV terminals.
- 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).

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.

Figure 82. Wireless zone sensor

By eliminating the wires between a zone temperature sensor and a VAV terminal, the sensor (Figure 82) 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.

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 (at least five years) and a visible "low battery" indicator to minimize ongoing maintenance.

Ventilation

For this manual, ventilation refers to the introduction of outdoor air to dilute contaminants that are generated inside the building (by people, equipment, processes, or furnishings). Of course, the introduction of outdoor air requires the removal of an equal quantity of air from the building.

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 needed at 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 2007 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 6-2 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 rate required in breathing zone (V_{bz})

Caution: 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 peak zone population (P_z) used for designing the ventilation system and for calculating cooling loads. Using occupant load, rather than zone population, to calculate ventilation requirements will often result in oversized HVAC equipment and excessive energy use.

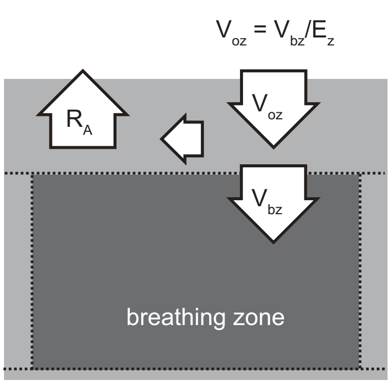
Table 6-1 of ASHRAE 62.1 prescribes two ventilation rates for each occupancy category: one for people-related sources of contaminants and another for building-related sources.

For step 1, determine the occupancy category for the zone and look up the corresponding minimum outdoor air rates in Table 6-1. 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 peak number of people expected to occupy the zone during typical usage (P_z) and occupiable floor area (A_z). Finally, solve the following equation to find the minimum outdoor airflow required for the breathing zone (V_{bz}).

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

Impact of zone air-distribution effectiveness

Figure 83. Impact of zone air-distribution effectiveness (E_z)



In addition to defining the breathing-zone outdoor airflow (V_{bz}), ASHRAE 62.1 also prescribes a zone air-distribution effectiveness (E_z) that accounts for how well the ventilation air, which is delivered to the zone by supply-air diffusers, actually gets into the breathing zone (Figure 83). 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}).

Table 17 is an excerpt from ASHRAE 62.1, and provides default values for E_z for air distribution configurations commonly used in VAV systems. It is based on the placement of supply-air diffusers and return-air grilles, and the temperature of the air being supplied.

Location of supply-air diffusers	Location of return-air grilles	Supply-air temperature (T_{SA})	E_z
ceiling	ceiling	cooler than zone	1.0
ceiling	floor	cooler than zone	1.0
ceiling	ceiling	warmer than zone $\geq T_{zone}$ + 15°F (8°C)	0.8
ceiling	ceiling	warmer than zone $< T_{zone}$ + 15°F (8°C)	1.0
ceiling	floor	warmer than zone	1.0
floor (thermal displacement ventilation)	ceiling	cooler than zone	1.2
floor (underfloor air distribution)	ceiling	cooler than zone	1.0
floor	ceiling	warmer than zone	0.7

Table 17. Zone air-distribution effectiveness (E_z)*

*Excerpt from Table 6-2 of ANSI/ASHRAE Standard 62.1-2007

In most VAV systems, the supply-air diffusers are located in or near the ceiling. When cool air ($T_{SA} < T_{zone}$) 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 ($T_{SA} > T_{zone}$ + 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 will tend 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 $T_{SA} > T_{zone}$ + 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.
- Locate the return-air grilles in the floor or at the base of a side wall (E_z = 1.0 during both cooling and heating modes).
- Design the system so that the supply-air temperature (T_{SA}) during the heating mode is less than 15°F (8°C) above the zone temperature (T_{zone}), and select the supply-air diffusers to achieve a velocity of 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. (See “VAV reheat terminal units,” p. 56.)
- Use baseboard radiant heat as the source of heat within the zone. Since the supply air is not used for heating, the supply-air temperature (T_{SA})

does not need to be warmer than the zone (T_{zone}) during the heating mode, and a zone air-distribution effectiveness (E_z) of 1.0 can be achieved.

Note: The only configuration that has a zone air-distribution effectiveness greater than 1 is when cool air is delivered to the zone using low-velocity, thermal displacement ventilation (TDV). Typical underfloor air distribution (UFAD) systems, which result in partially mixed zones, do not use thermal displacement ventilation and E_z = 1.0 during cooling. However, if either of these two systems is used to deliver warm air through their floor-mounted diffusers, and return air through ceiling-mounted grilles, zone air-distribution effectiveness (heating) is only 0.7.

System-level ventilation requirement

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

For a single-zone VAV system (see p. 157), in which one air-handling unit delivers a mixture of outdoor air and recirculated air to only one zone, ASHRAE 62.1 prescribes that the system-level intake (V_{ot}) needs to equal the calculated zone outdoor airflow (V_{oz}):

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

There is no need to account for system ventilation efficiency (E_v) when designing a single-zone system.

*Note: For more information on calculating system-level intake (V_{ot}) for a **dedicated OA system**, see p. 115.*

For more information on ASHRAE Standard 62.1, and its procedures for calculating zone-level and system-level outdoor airflow requirements in single-zone or 100 percent outdoor-air systems, refer to the Trane *Engineers Newsletter Live* broadcast DVD titled "ASHRAE Standard 62.1: Ventilation Requirements" (APP-CMC023-EN) or to the October 2004 *ASHRAE Journal* article, titled "Addendum 62n: Single-zone & Dedicated-OA Systems" (available at www.ashrae.org or www.trane.com).

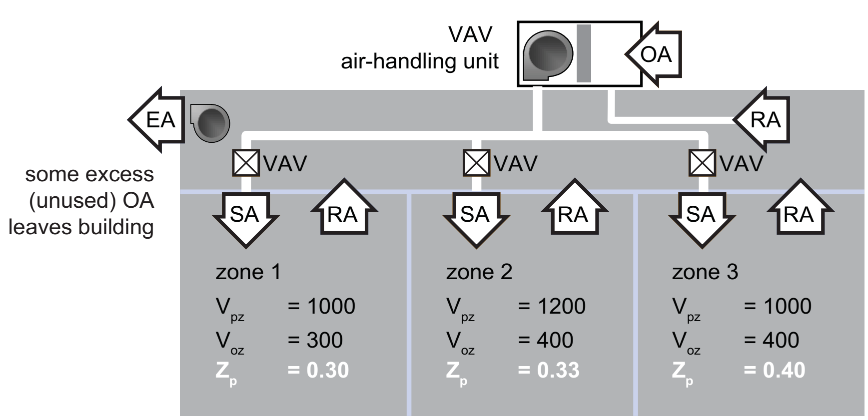
As mentioned earlier, ASHRAE Standard 62.1 also prescribes a process to calculate the outdoor airflow needed at the system-level intake (Vot) in order to make sure that the required quantity of outdoor air is delivered to each zone (V_{oz}).

Challenge of ventilating a multiple-zone VAV system

In a multiple-zone recirculating system, one supply fan delivers a mixture of outdoor air and recirculated return air to more than one zone. This supply air contains a certain fraction (percentage) of outdoor air, and all zones receive the same fraction. That is, if the supply air leaving the air-handling unit contains 40 percent outdoor air (some "first pass" outdoor air that enters through the intake and some "unused" outdoor air that returns from the zones), the air delivered to each zone served by that supply fan will contain 40 percent outdoor air, regardless of the overall airflow actually being delivered to the zone.

In the simple example shown in Figure 84, the outdoor airflow required in Zone 1 is 300 cfm (0.14 m³/s), while the total supply airflow is 1000 cfm (0.47 m³/s). So 30 percent of the air being supplied to Zone 1 must be outdoor air. The "critical" zone (that is, the zone that requires the highest fraction of outdoor air, Z_p) is Zone 3. It requires 40 percent of the air being supplied to be outdoor air.

Figure 84. Challenge of ventilating a multiple-zone VAV system



As a result, in order to properly ventilate the "critical" zone (Zone 3, in this example), the system must over-ventilate all other zones. Therefore, the air that returns from the "non-critical" zones (Zones 1 and 2) contains some "unused" or excess outdoor air. Most of this unused outdoor air recirculates back through the system, which allows the air-handling unit to bring in less than 40 percent outdoor air through the intake. But some of the unused outdoor air leaves the building in the exhaust air stream. In other words, some outdoor air enters the system, passes through non-critical zones, and then leaves the building without being fully used to dilute indoor contaminants.

For this reason, multiple-zone recirculating systems have a system ventilation efficiency (E_v) that is less than 100 percent. Some of the outdoor air brought into the system is "wasted" because of over-ventilation in non-critical zones. However, as demonstrated below, multiple-zone recirculating systems can account for population diversity, which helps offset some of this inherent inefficiency.

Calculating system intake airflow (V_{ot})

ASHRAE Standard 62.1 defines a procedure to account for system ventilation efficiency, and to determine the system-level intake airflow (V_{ot})—the quantity brought through the outdoor-air damper in the air-handling unit—required to deliver the proper quantity of outdoor air to each of the individual zones.

For more information on ASHRAE Standard 62.1 and its procedures for calculating zone-level and system-level outdoor airflow requirements in a VAV system, refer to the Trane *Engineers Newsletter Live* broadcast DVD titled "ASHRAE Standard 62.1: Ventilation Requirements" (APP-CMC023-EN) or the following *ASHRAE Journal* articles:

1) Stanke, D. "Addendum 62n: Single-Path Multiple-Zone System Design." *ASHRAE Journal* 47 (January 2005): 28–35. Available at www.ashrae.org or www.trane.com.

2) Stanke, D. "Standard 62.1-2004: Designing Dual-Path, Multiple-Zone Systems." *ASHRAE Journal* 47 (May 2005): 20–30. Available at www.ashrae.org or www.trane.com.

The standard provides two methods for determining system ventilation efficiency, E_v:

- Table 6-3, "default E_v" method
- Appendix A, "calculated E_v" method

The default approach is simpler and takes less time than the calculated approach. However, the calculated approach is more accurate and usually results in higher efficiency and lower intake flow, and may be worth the added effort.

Note: The examples below are not intended to reflect a real-life system, but rather to demonstrate the calculation methods, so the numbers used are simple rather than realistic. For more realistic examples, refer to the ASHRAE Journal *articles listed in the sidebar.*

Table 6-3, "default E_v" method. Beginning with the default method, which uses Table 6-3 in the standard, determining system ventilation efficiency and the required outdoor-air intake flow involves several steps:

1. Calculate breathing-zone outdoor airflow (V_{bz})
2. Determine the zone air-distribution effectiveness (E_z)
3. Calculate zone outdoor airflow (V_{oz})
4. Calculate the zone primary outdoor-air fraction (Z_p)
5. Determine the "uncorrected" outdoor-air intake (V_{ou})

6 Determine system ventilation efficiency (E_v)

7 Calculate the system outdoor-air intake (V_{ot})

Figure 85 shows an example, eight-zone VAV system serving an office building. Supply air is delivered to the zones through ceiling-mounted diffusers and return air leaves the zones through ceiling-mounted return-air grilles. All zones have VAV reheat terminals. The first three steps (zone-level calculations) have already been completed (Table 18); see "Zone-level ventilation requirements," p. 101.

Figure 85. Example office building with a VAV system

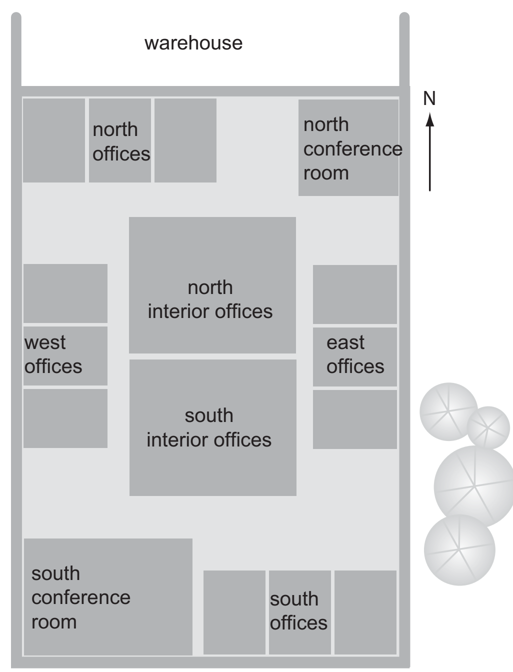


Table 18. Zone-level ventilation calculations for example office building (cooling design)

	R_p (cfm/p) x	P_z (qty) =	$V_{bz,p}$ (cfm)	R_a (cfm/ft^2) x	A_z (ft^2) =	$V_{bz,a}$ (cfm)	V_{bz} (cfm)/	E_z =	V_{oz} (cfm)
South offices	5	18	90	0.06	2000	120	210	1.0	210
West offices	5	20	100	0.06	2000	120	220	1.0	220
South conf room	5	30	150	0.06	3000	180	330	1.0	330
East offices	5	20	100	0.06	2000	120	220	1.0	220
South interior offices	5	50	250	0.06	10,000	600	850	1.0	850
North interior offices	5	50	250	0.06	10,000	600	850	1.0	850
North offices	5	16	80	0.06	2000	120	200	1.0	200
North conf room	5	20	100	0.06	2000	120	220	1.0	220
System totals		224 ΣP_z	1120 $\Sigma(R_p \times P_z)$			1980 $\Sigma(R_a \times A_z)$			

The **fourth step** is to calculate the fraction of outdoor air that is required in the primary air for each zone. The primary outdoor-air fraction, Z_p, is the ratio of zone outdoor airflow to the primary airflow being delivered to the zone. The larger the ratio, the richer the concentration of outdoor air required to meet the zone ventilation demand.

$$Z_p = V_{oz}/V_{pz}$$

where,

V_{oz} = zone outdoor airflow, cfm (m^3/s)

V_{pz} = primary airflow, cfm (m^3/s)

In a VAV system, the primary airflow changes with the cooling load, so Z_p increases when the zone primary airflow decreases. For this reason, for design purposes ASHRAE 62.1 requires using the "minimum expected primary airflow" ($V_{pz} = V_{pzm}$) for VAV systems. Determining this "minimum expected" value requires judgment by the design engineer. It may be the minimum airflow setting for the VAV terminal, or it might be a higher value (see sidebar).

Note: Do not set the minimum primary airflow (V_{pzm}) equal to the zone outdoor airflow (V_{oz}). This approach results in an intake airflow equal to the system primary airflow (that is, 100 percent outdoor air).

Using the minimum primary airflow (V_{pzm}), the outdoor-air fractions (Z_p) vary from 0.40 to 0.50 for this example system (Table 19). So 0.50 represents the critical (highest) demand of the system.

Table 19. System-level ventilation calculations based on "default E_v" method

	design V_{pz} (cfm)	V_{pzm} (cfm)	V_{oz} (cfm)	Z_p (V_{oz}/V_{pzm})
South offices	1900	475	210	0.44
West offices	2000	500	220	0.44
South conf room	3300	825	330	0.40
East offices	2000	500	220	0.44
South interior offices	7000	1750	850	0.49
North interior offices	7000	1750	850	0.49
North offices	1600	400	200	0.50
North conf room	1800	450	220	0.49

The **fifth step** is to determine the "uncorrected" outdoor-air intake, V_{ou}. This is the intake airflow that would be required if the system was 100 percent efficient at delivering the outdoor air to the breathing zone.

$$V_{ou} = D \times \Sigma(R_p \times P_z) + \Sigma(R_a \times A_z)$$

Most of these values were determined for the zone-level ventilation calculations (see Table 18). As part of this step, the designer can take credit for population diversity within the system (D), since buildings are seldom occupied with peak population in all zones simultaneously. For example, if all

What is the "minimum expected primary airflow"?

Conservative designers use the minimum airflow setting for the VAV terminal (see "Minimum primary airflow settings," p. 62) as the "minimum expected primary airflow" in this calculation. While straightforward, this approach results in a lower system ventilation efficiency (E_v) and a higher outdoor-air intake flow (V_{ot}) at design.

Less conservative designers might use a value for "minimum expected primary airflow" that exceeds the minimum airflow setting for the VAV terminal. For most VAV systems, the highest outdoor-air intake flow (V_{ot}) occurs when the total system primary airflow is at design (block) airflow. When system airflow is at design, it is unlikely that any VAV terminal will be closed down to its minimum primary airflow setting. Using a higher, more realistic value for "minimum expected primary airflow" will result in a higher system ventilation efficiency (E_v) and a lower outdoor-air intake flow (V_{ot}) at design.

One of the keys at this step is to be realistic. ASHRAE 62.1 does not require the system to be designed to handle ANY combination of conditions that might conceivably occur, just those that can reasonably be expected to occur under normal operation. The 62MZCALC spreadsheet, provided with the *Standard 62.1 User's Manual*, is a useful tool that allows design engineers to analyze the impact of various operating scenarios.

For more information on selecting minimum airflow settings that comply with both ASHRAE Standard 62.1 and 90.1, refer to the Trane *Engineers Newsletter* titled "Potential ASHRAE Standard Conflicts: Indoor Air Quality and Energy Standards" (ADM-APN030-EN) and to the Trane *Engineers Newsletter Live* broadcast DVD titled "ASHRAE Standards 62.1 and 90.1 and VAV Systems" (APP-CMC034-EN).

of the work spaces in an office building are fully occupied, the conference rooms will be sparsely occupied.

Mathematically, occupant diversity (D) is the ratio of the expected peak system population (P_s) to the sum of the peak populations in the individual zones (ΣP_z).

$$D = P_s / \Sigma P_z$$

In this same example, the sum of the peak zone populations (ΣP_z) is 224 (Table 18). Assuming the maximum expected system population (P_s) is only 164 people, the occupant diversity factor (D) is 0.73 (164 / 224). The architect or owner usually prepares a program indicating the building's expected occupancy and use. This program may be a good source for information related to expected system population.

Note that this occupant diversity value is multiplied by only the people-related component of the ventilation requirement ($R_p \times P_z$), not the building-related component ($R_a \times A_z$). For this example, the uncorrected outdoor-air intake is 2800 cfm (1.3 m^3/s).

$$V_{ou} = 0.73 \times 1120 \text{ cfm} + 1980 \text{ cfm} = 2800 \text{ cfm}$$

The **sixth step** is to determine system ventilation efficiency, E_v. Table 20 is an excerpt from ASHRAE 62.1. It lists the default value for system ventilation efficiency based on the largest zone outdoor-air fraction (Z_p). Interpolating in this table is allowed.

Table 20. E_v defaults from Table 6-3 of ASHRAE 62.1-2007

Maximum Z_p	E_v
≤ 0.25	0.90
≤ 0.35	0.80
≤ 0.45	0.70
≤ 0.55	0.60
> 0.55	Use Appendix A

For the zones served by the system in this example, the largest Z_p is 0.50 (Table 19). Interpolating in Table 20, the system ventilation efficiency (E_v) is determined to be 0.65.

Notice that if the largest Z_p is greater than 0.55, the standard requires the use of the "calculated E_v" method that is outlined in Appendix A of the standard.

The **final step** is to calculate the required outdoor-air intake for the system, V_{ot}. This is determined by dividing the uncorrected outdoor-air intake (V_{ou}) by system ventilation efficiency (E_v):

$$V_{ot} = V_{ou} / E_v$$

For this example, the uncorrected outdoor-air intake is 2800 cfm (1.3 m^3/s) with a system ventilation efficiency of 0.65, resulting in a required outdoor-air intake (V_{ot}) flow of 4310 cfm (2.0 m^3/s).

Appendix A, "calculated E_v" method

As mentioned earlier, ASHRAE 62.1 provides a second method for determining system ventilation efficiency. This "calculated E_v" method may be used for any system, but it *must* be used if the maximum primary outdoor-air fraction (Z_p) is greater than 0.55. Calculating E_v, rather than using the default value from Table 6-3 of the standard, usually results in a lower outdoor-air intake requirement (V_{ot}).

The steps involved in this method are very similar to those in the default method:

1. Calculate breathing-zone outdoor airflow (V_{bz})
2. Determine the zone air-distribution effectiveness (E_z)
3. Calculate zone outdoor airflow (V_{oz})
4. Calculate the zone discharge outdoor-air fraction (Z_d)
5. Determine the "uncorrected" outdoor-air intake (V_{ou})
6. Determine system ventilation efficiency (E_v)
7. Calculate the system outdoor-air intake (V_{ot})

Using the same, eight-zone VAV system example, the first three steps (zone-level calculations) have already been completed (Table 18).

The **fourth step** is to calculate the fraction of outdoor air that is required in the discharge air for each zone. The discharge outdoor-air fraction, Z_d, is the ratio of zone outdoor airflow to the discharge airflow being delivered to the zone.

$$Z_d = V_{oz}/V_{dz}$$

where,
V_{oz} = zone outdoor airflow, cfm (m^3/s)
V_{dz} = discharge airflow, cfm (m^3/s)

Note that Appendix A uses the term *discharge* outdoor-air fraction, Z_d, rather than *primary* outdoor-air fraction, Z_p. In this way, the equations can be applied to systems with multiple recirculation paths, such as fan-powered VAV or dual-duct systems. However, in a single-path system (VAV reheat) like the one used in this example, outdoor air only enters the zone from the central air-handling unit. Therefore, discharge airflow and primary airflow are the same, and Z_d and Z_p are also the same.

In a VAV system, the discharge airflow may change with the space load, so Z_d increases when the zone discharge airflow decreases. For this reason, for design purposes ASHRAE 62.1 recommends using the "minimum expected discharge airflow" (V_{dzm}) for VAV systems. Determining this "minimum expected" value requires judgment by the design engineer. It may be the minimum airflow setting for the VAV terminal, or it might be a higher value (see the sidebar, "What is the minimum expected primary airflow?" p. 107).

For this example, the discharge outdoor-air fractions are the same as the primary outdoor-air fractions calculated in the previous section, ranging from 0.40 to 0.50 (Table 19).

The **fifth step** is to determine the "uncorrected" outdoor-air intake, V_{ou}. Again this step is identical to the "default E_v" method discussed earlier (p. 107). Taking credit for occupant diversity, the people-related component is reduced, and the uncorrected outdoor-air intake for this example is 2800 cfm (1.3 m^3/s).

The **sixth step** is to calculate system ventilation efficiency, E_v. The equations for this step are included in Appendix A of ASHRAE 62.1. For systems with a single recirculation path (cooling-only VAV terminals or VAV reheat terminals), like the system used in this example, this calculation is a fairly simple process. For systems with multiple recirculation paths (fan-powered VAV terminals or dual-duct VAV systems), the process is more complex. (See "Systems with multiple recirculation paths," p. 113.)

Calculating system ventilation efficiency for a system with a single recirculation path requires three steps. **Step 6a** is to calculate the average fraction of outdoor air in the system, X_s:

$$X_s = V_{ou} / V_{ps}$$

where,

V_{ou} = uncorrected outdoor-air intake, cfm (m^3/s)
V_{ps} = system primary airflow, cfm (m^3/s)

Appendix A of the standard defines system primary airflow (V_{ps}) as the sum of the zone primary airflows. In a VAV system, for cooling design calculations, the maximum airflow delivered by the supply fan (or "block" airflow) is less than the sum of the zone peak airflows because of load diversity (Table 21). This block airflow can be obtained from a load calculation program or by summing the peak airflows for all the zones, and multiplying this sum by an assumed load diversity factor.

For this example system, the system block airflow (system primary airflow at cooling design conditions) is 18,600 cfm (8.8 m^3/s). So, the average outdoor-air fraction for the system, X_s, is 0.15 (2800/18,600 [1.3/8.8]). This means that if the system was 100 percent efficient at delivering outdoor air from the intake to the breathing zones, then 15 percent of the system primary airflow would need to be outdoor air.

Table 21. System-level ventilation calculations based on "calculated E_v" method (cooling design)

	design V_{pz} (cfm)	V_{pzm} (cfm)	V_{oz} (cfm)	Z_d (V_{oz}/V_{pzm})	E_{vz} ($1 + X_s - Z_d$)
South offices	1900	475	210	0.44	0.71
West offices	2000	500	220	0.44	0.71
South conf room	3300	825	330	0.40	0.75
East offices	2000	500	220	0.44	0.71
South interior offices	7000	1750	850	0.49	0.66
North interior offices	7000	1750	850	0.49	0.66
North offices	1600	400	200	0.50	0.65
North conf room	1800	450	220	0.49	0.66
System totals	ΣV_{pz} = 26,600 V_{ps} = 18,600 ("block" airflow)				

Step 6b is to determine the ventilation efficiency for each zone (E_{vz}). This describes how efficiently the system delivers outdoor air from the system-level intake to the individual breathing zone. For single-path systems, Appendix A provides the following equation for zone ventilation efficiency:

$$E_{vz} = 1 + X_s - Z_d$$

X_s was calculated in step 6a, and Z_d for each zone was calculated in step 4. For this example, zone ventilation efficiencies vary from 0.65 to 0.75 (Table 21).

For **Step 6c**, system ventilation efficiency (E_v) is equal to the worst-case, or lowest, zone ventilation efficiency (E_{vz}). That is 0.65 for the "north offices" in this example.

The **final step** is to calculate the required outdoor-air intake at the air-handling unit, V_{ot}. This is determined by dividing the uncorrected outdoor-air intake (V_{ou}) by system ventilation efficiency (E_v):

$$V_{ot} = V_{ou} / E_v$$

For this example, the uncorrected outdoor-air intake is 2800 cfm (1.3 m^3/s) with a system ventilation efficiency of 0.65, resulting in a required outdoor-air intake flow of 4310 cfm (2.0 m^3/s).

In this example, using Appendix A resulted in an intake airflow that is the same as if determined using Table 6-3. The default values in that table are based on an assumed average outdoor air fraction, X_s, of 0.15. For systems with a higher average fraction, Appendix A results in a higher system ventilation efficiency, and a lower intake airflow.

Heating versus cooling design

As mentioned earlier, supplying warm air to the zone may result in a zone air-distribution effectiveness (E_z) that is less than one. How does this impact system intake airflow? Table 22 and Table 23 include the calculations for this same example system at the heating design condition, with several zones having a zone air-distribution effectiveness of 0.8.

Table 22. Zone-level ventilation calculations for example office building (heating design)

	R_p (cfm/p) x	P_z (qty) =	$V_{bz,p}$ (cfm)	R_a (cfm/ft^2) x	A_z (ft^2) =	$V_{bz,a}$ (cfm)	V_{bz} (cfm)/	E_z =	V_{oz} (cfm)
South offices	5	18	90	0.06	2000	120	210	0.8	263
West offices	5	20	100	0.06	2000	120	220	0.8	275
South conf room	5	30	150	0.06	3000	180	330	0.8	413
East offices	5	20	100	0.06	2000	120	220	0.8	275
South interior offices	5	50	250	0.06	10,000	600	850	1.0	850
North interior offices	5	50	250	0.06	10,000	600	850	1.0	850
North offices	5	16	80	0.06	2000	120	200	1.0	200
North conf room	5	20	100	0.06	2000	120	220	1.0	220
System totals		224 (ΣP_z)	1120 Σ ($R_p P_z$)			1980 Σ ($R_a \times A_z$)			

Table 23. System-level ventilation calculations based on "calculated E_v" method (heating design)

	design V_{pz} (cfm)	V_{pzm} (cfm)	V_{oz} (cfm)	Z_d (V_{oz}/V_{pzm})	E_{vz} (1 + X_s - Z_d)
South offices	1900	475	263	0.55	0.87
West offices	2000	500	275	0.55	0.87
South conf room	3300	825	413	0.50	0.92
East offices	2000	500	275	0.55	0.87
South interior offices	7000	1750	850	0.49	0.94
North interior offices	7000	1750	850	0.49	0.94
North offices	1600	400	200	0.50	0.92
North conf room	1800	450	220	0.49	0.93
System totals		V_{ps} = 6650 (ΣV_{pzm})			

Because zone primary airflows are lower at the heating design condition, the overall system primary airflow (V_{ps}) is also lower (Table 23). This results in a higher average outdoor-air fraction, X_s.

$$X_s = V_{ou} / V_{ps} = 2800 \text{ cfm} / 6650 \text{ cfm} = 0.42$$

For this example, higher required zone outdoor airflows (V_{oz}) increase the outdoor-air fractions (Z_d), but the average outdoor-air fraction (X_s) also increases. The overall impact is a higher system ventilation efficiency (E_v = 0.87) and a lower intake airflow.

$$V_{ot} = V_{ou} / E_v = 2800 \text{ cfm} / 0.87 = 3220 \text{ cfm}$$

Using the "calculated E_v" method from Appendix A of the standard, the system intake airflow is calculated to be 3220 cfm (1.5 m^3/s) at heating design, compared to 4310 cfm (2.0 m^3/s) at cooling design. Note that if the "default E_v" method had been used for heating design calculations in this example, the default value for E_v would be 0.60 (maximum Z_p = 0.55) and the required system intake would have been much higher, 4670 cfm (2.2 m^3/s).

Note: Unlike multiple-zone recirculating systems, single-zone and most 100 percent outdoor-air systems (for which V_{ot} depends strictly on V_{oz}), usually require more intake airflow during heating operation than during cooling operation.

Systems with multiple recirculation paths (fan-powered VAV and dual-duct)

For a detailed discussion of calculations for systems with multiple recirculation paths, refer to the May 2005 *ASHRAE Journal* article, titled "Standard 62.1: Designing Dual-Path, Multiple-Zone Systems."

Fan-powered VAV systems (Figure 51 and Figure 53, p. 58) have two paths for outdoor air to get into the zone. One path is the primary air stream from the central air-handling unit, and the other path is the recirculation of "unused" outdoor air that the fan-powered VAV terminal draws in from the ceiling plenum.

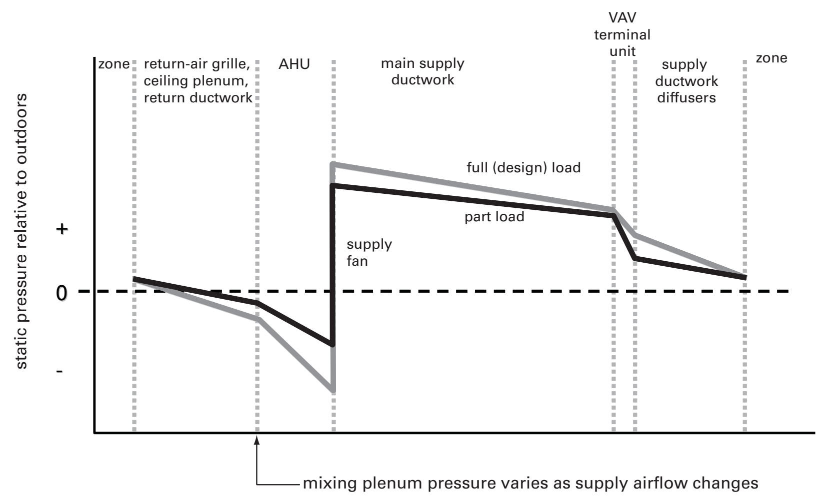
Dual-fan, dual-duct VAV systems (Figure 116, p. 166) also have two ventilation paths. One path is the primary air stream from the central cooling air-handling unit, which brings in outdoor air and mixes it with recirculated return air. The other path is from the central heating air-handling unit that recirculates return air from all zones.

Fixed outdoor-air damper position

One common practice (which rarely works as intended) for ventilating a VAV system is to set a fixed, minimum position for the outdoor-air damper. With the system delivering design supply airflow, this position is set by the test, adjust, and balance (TAB) contractor when the system is first started, and the damper remains at this same position throughout the full range of system airflows. The outdoor-air damper is allowed to open further for economizer cooling. (See "Airside economizer control," p. 174).

As the system supply airflow drops during part-load conditions, the pressure losses due to return-air grilles, ceiling plenum, and return ductwork are lessened (Figure 86). This causes the pressure inside the mixing plenum (where the outdoor air mixes with the recirculated return air) to increase. That is, it is not as negative (with respect to the pressure outside) as it was at design supply airflow. With the outdoor-air damper set at a fixed position, the quantity of outdoor air entering through the OA damper will decrease. That is, outdoor airflow will not be held constant, but instead will decrease, as supply airflow decreases.

Figure 86. Impact of variable airflow on mixing plenum pressure



ASHRAE Standard 62.1-2007 addresses this practice in Section 5.4:

> Mechanical ventilation systems ... shall be designed to maintain the minimum outdoor airflow as required by Section 6 under any load condition. **Note:** VAV systems with fixed outdoor air damper positions must comply with this requirement at minimum supply airflow.

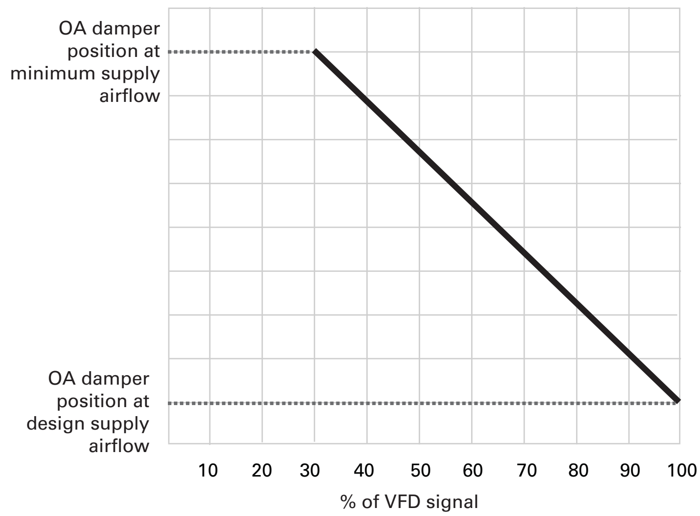
To comply with ASHRAE 62.1 using a fixed OA damper position, the OA damper must be set at a fairly wide open position to bring in the required outdoor airflow when supply airflow is at minimum. This increases the energy required to heat, cool, and dehumidify the excess OA that is brought into the system whenever supply airflow is higher than the minimum.

Proportional outdoor-air damper control

A relatively inexpensive enhancement to using a fixed OA damper position is to vary the position of the OA damper in proportion to the change in supply airflow.

First, with the system operating at design (full load) supply airflow, the OA damper position is set to bring in the minimum required quantity of outdoor airflow (V_{ot}). Then, with the system operating at the minimum expected system airflow, the OA damper is adjusted to a more open position so it brings in the same quantity (cfm [m^3/s]) of outdoor air as it did at design airflow. During system operation, the position of the OA damper is varied in proportion to the change in supply airflow, which is determined by the signal being sent to the variable-speed drive on the supply fan (Figure 87).

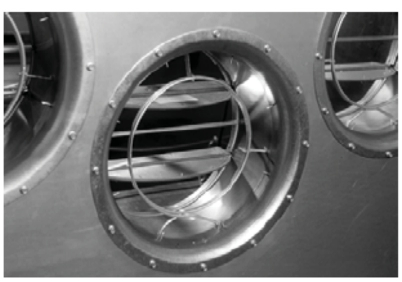
Figure 87. Proportional control of the outdoor-air damper



This method is somewhat inaccurate over the entire range of supply airflows (resulting in some over-ventilation) and is unable to respond to pressure fluctuations caused by wind or stack effect. But it results in significantly less over-ventilation than a fixed OA damper position, thus saving energy, and it is relatively inexpensive.

Direct measurement and control of outdoor airflow

Figure 88. Flow-measuring outdoor-air damper (Trane Traq™ damper)



One of the best methods for controlling system-level intake airflow (V_{ot}) in a multiple-zone VAV system is to actually measure the outdoor airflow and control it directly. This is typically accomplished by using a flow-measuring device in the outdoor air stream, such as a flow-measuring OA damper (Figure 88) or a field-installed airflow measurement station.

This method is accurate over a wide range of airflows, and can respond to pressure fluctuations caused by wind or stack effect. It reduces the energy use associated with over-ventilation and has the added benefit of providing a means to document the outdoor airflow brought into the system over time. However, outdoor airflow measurement does increase the cost of the system.

Dedicated outdoor-air systems

As described earlier, in a conventional VAV system, the "Ventilation Rate Procedure" in ASHRAE 62.1 often requires the air-handling unit to bring in more outdoor air than simply the sum of zone ventilation requirements. However, some VAV system designs use a dedicated outdoor-air unit to condition all the outdoor air for the system. This conditioned outdoor air (CA) is then either:

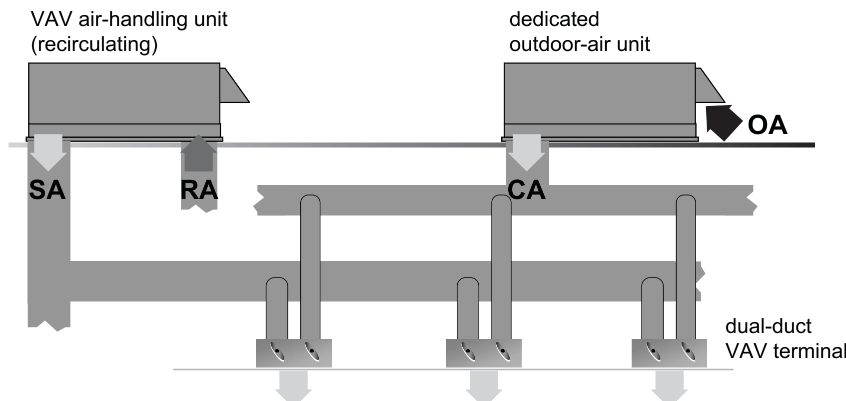
1 Ducted directly to each zone

2 Ducted directly to individual, dual-duct VAV terminals that serve each zone (Figure 89: the "ventilation" damper in the dual-duct VAV terminal maintains the required quantity of outdoor air from the dedicated

outdoor-air unit, while the "primary-air" damper regulates recirculated supply air from the air-handling unit to control zone temperature)

3 Ducted directly to the outdoor-air intake of one or more VAV air-handling units (Figure 96, p. 124)

Figure 89. Dedicated outdoor-air unit delivering OA to dual-duct VAV terminals



In either of the first two configurations, since the outdoor air is delivered directly to the zones or to zone-level VAV terminals, this is not considered a multiple-zone *recirculating* ventilation system. The system-level intake airflow (V_{ot}) delivered by the dedicated OA unit must be the sum of the calculated zone outdoor airflows (V_{oz}):

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

This may be less than what ASHRAE 62.1 would require of a conventional VAV system (multiple-zone recirculating). Using the same example (see Table 18, p. 106), if the dedicated OA system delivers 100 percent outdoor air (no mixing with recirculated air) to the zones, the required outdoor-air intake flow ($V_{ot} = \Sigma V_{oz}$) at cooling design conditions would be 3100 cfm (1.5 m^3/s), compared to 4310 cfm (2.0 m^3/s) for the multiple-zone recirculating VAV system. However, since multiple-zone recirculating systems are allowed to account for population diversity (see p. 107), system-level intake airflow for a multiple-zone VAV system can be close to (or even less than) either of these two dedicated OA system configurations.

At heating design conditions (Table 23, p. 112), the required intake airflow for the dedicated OA system would be 3350 cfm (1.6 m^3/s), compared to 3220 cfm (1.5 m^3/s) for the multiple-zone VAV system. This difference occurs because a multiple-zone recirculating system recirculates most of the "unused" outdoor air that bypasses the breathing zone (when $E_z < 1.0$).

In the third configuration, the dedicated OA unit delivers outdoor air to one or more multiple-zone, recirculating systems. In this case, intake at the dedicated OA unit is the sum of the system-level intake airflows (V_{ot}) required for each VAV air-handling unit, so it does not result in an overall reduction in outdoor airflow.

For more discussion of using a dedicated OA system with VAV, see “Methods for improving dehumidification performance,” p. 121.

Dynamic reset of intake airflow

Section 6.2.7 of ASHRAE 62.1 explicitly permits dynamic reset of intake (outdoor) airflow (V_{ot}) 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. The standard specifically mentions the following reset control strategies:

- *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,” attempts to dynamically reset the system outdoor-air intake (V_{ot}) based on changing population in the zone. Some of the commonly used methods of assessing zone population include:

  1. Time-of-day occupancy schedules in the building automation system (BAS) that are used to either indicate when a zone is normally occupied versus unoccupied or to vary the zone ventilation requirement based on anticipated population.
  2. Occupancy sensors, such as motion detectors, that indicate when a zone is occupied or unoccupied. When unoccupied, the zone ventilation requirement is reduced.
  3. Carbon dioxide (CO_2) sensors that monitor the concentration of CO_2 in the zone as an indicator of the per-person ventilation rate (cfm/person [m^3/s/person]) actually being delivered to the zone.

- *Resetting intake airflow in response to variations in ventilation efficiency.* The ventilation efficiency (E_v) of a multiple-zone VAV system changes as zone-level and system-level primary airflows change with variations in building load. This strategy, which some have referred to as “ventilation reset,” dynamically resets the system outdoor-air intake (V_{ot}) based on this changing efficiency.

- *Resetting VAV minimums in response to variations in intake airflow.* In most VAV applications, each terminal unit has a minimum airflow setting. When the airside economizer is activated, the air-handling unit brings in more than minimum outdoor air for the purposes of free cooling. The supply air, therefore, contains a higher percentage of outdoor air than is required for ventilation. This may allow the minimum airflow settings on some VAV terminal units to be lowered, possibly saving reheat energy or supply fan energy if any of the VAV terminals are closed to their minimum primary airflow settings.

For more information on these ventilation control strategies, see “Ventilation control,” p. 184, and “Ventilation optimization,” p. 205.

Humidity Control

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

Dehumidification

Section 5.10 of ASHRAE 62.1-2007 requires that systems be designed to limit the relative humidity in occupied spaces to 65 percent or less at the peak outdoor dew-point design condition and at the peak indoor design latent load.

A VAV system typically dehumidifies effectively over a wide range of indoor loads because it supplies air at a relatively constant, low dew-point temperature at all load conditions. As long as any zone needs cooling, the VAV air-handling unit provides supply air at a dew point that is usually low enough (sufficiently dry) to offset the latent loads in the zones.

Full-load versus part-load dehumidification performance

For more information on the dehumidification performance of a VAV system, including the impact of minimum airflow settings and SAT reset, refer to the Trane application manual titled *Dehumidification in HVAC Systems* (SYS-APM004-EN).

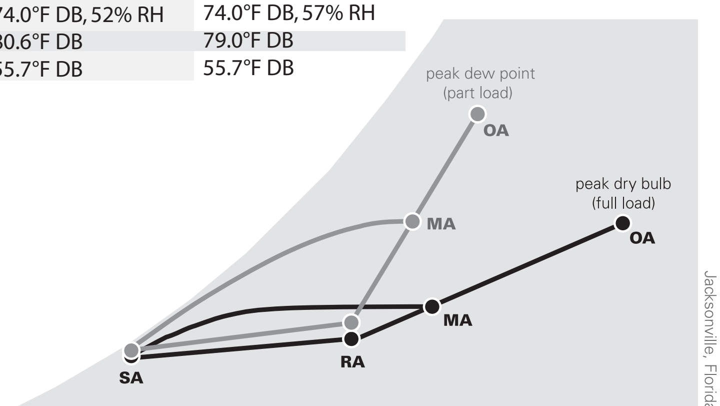
To demonstrate, consider a 10,000 ft^3 (283 m^3) classroom in Jacksonville, Fla., that accommodates 30 people. During cooling mode, the zone sensor setpoint is 74°F (23.3°C) dry bulb.

At the traditional design condition (peak outdoor dry-bulb temperature), the system delivers 1500 cfm (0.7 m/s) of supply air—of which 450 cfm (0.2 m^3/s) is outdoor air required for ventilation—at 55.7°F (13.1°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 relative humidity in the zone is 52 percent at this design condition (Figure 90).

At part-load conditions, the VAV system responds to the lower sensible cooling load in the zone by reducing the quantity of air supplied to the zone, while maintaining a relatively constant supply-air temperature (55.7°F [13.1°C] in this example). At this example part-load (peak outdoor dew point) condition, the supply airflow is reduced to 900 cfm (0.42 m^3/s) to avoid overcooling the zone. Because the supply air is still cool and dry (low dew point), however, the relative humidity in the classroom only rises to 57 percent (Figure 90).

Figure 90. Full- versus part-load dehumidification performance of a VAV system

Design condition	Full Load Peak dry bulb	Part Load Peak dew point
OA	96.0°F DB, 76.0° WB	76.0°F DP, 84.0°F DB
RA	74.0°F DB, 52% RH	74.0°F DB, 57% RH
MA	80.6°F DB	79.0°F DB
SA	55.7°F DB	55.7°F DB



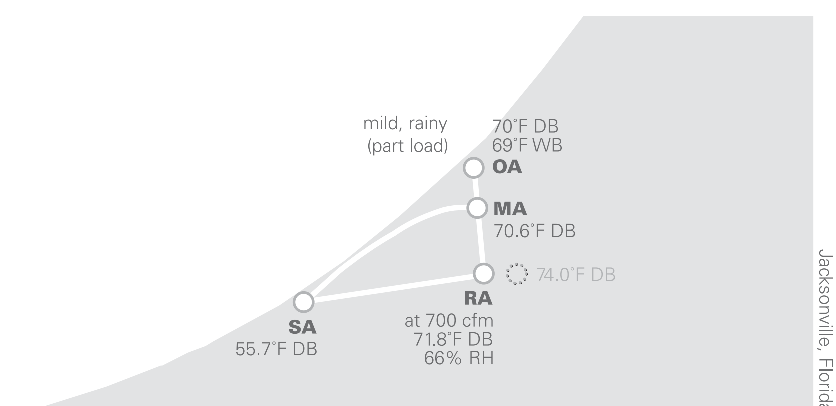
By continuing to deliver cool, dry air at part load, VAV systems typically provide effective dehumidification over a wide range of load conditions.

Impact of minimum airflow settings for VAV terminal units

Eventually, the sensible cooling load in the zone becomes small enough that the required supply airflow is less than the minimum airflow setting of the VAV terminal unit (see “Minimum primary airflow settings,” p. 62). For this same example classroom, assume that the minimum airflow setting is 700 cfm (0.33 m^3/s).

Now, consider a cool, rainy day—70°F dry bulb, 69°F wet bulb (21.1°C DB, 20.6°C WB). At this condition, if 700 cfm (0.33 m^3/s) is supplied at 55.7°F (13.1°C), the space will be overcooled, to 71.8°F (22.1°C). As the dry-bulb temperature in the space decreases, the relative humidity increases—to 66 percent in this example—and the classroom will feel cool and damp (Figure 91).

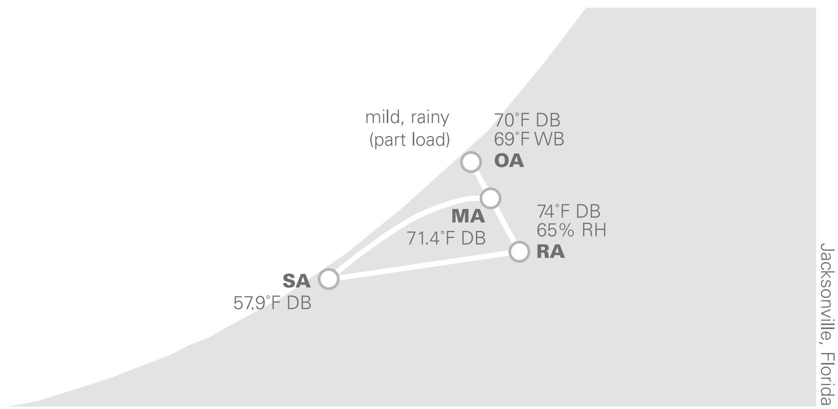
Figure 91. Potential for overcooling due to minimum airflow settings on VAV terminal units



Resetting supply-air temperature

One way to prevent this type of overcooling is to reset the supply-air temperature upward at low-sensible-load conditions. If the minimum airflow setting is 700 cfm (0.33 m³/s), raising the supply-air temperature to 57.9°F (14.4°C), for example, avoids overcooling the classroom on this example mild, rainy day … but the cooling coil also removes less moisture from the supply air. As a result, the relative humidity in the space remains high at 65 percent (Figure 92).

Figure 92. Impact of supply-air-temperature reset



It may be tempting to raise the supply-air temperature (SAT) at part-load conditions to save cooling or reheat energy. But, in non-arid climates, warmer supply air means less dehumidification at the coil and higher humidity levels in the zones. And, because the supply air is warmer, those zones that require cooling will need *more* air to offset the cooling load, which increases supply fan energy consumption. (See "Supply-air-temperature reset," p. 202.)

If dehumidification is a concern, avoid using SAT reset when it is humid outside, unless a system analysis indicates that humidity in the zones will not rise to unacceptable levels. For applications that include SAT reset, consider either 1) providing an outdoor dew point sensor to disable reset when it is humid outside, or 2) providing one or more zone humidity sensors to override the SAT reset function if humidity in the zone (or return air) exceeds a maximum limit.

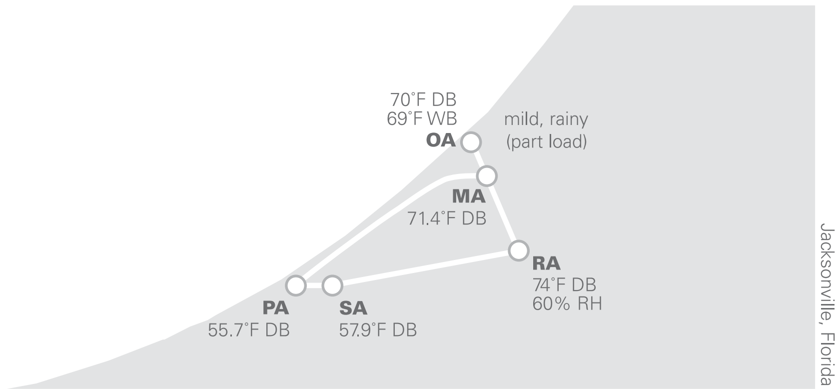
Doesn't ASHRAE Standard 90.1 prohibit the use of new-energy "reheat" in VAV terminals?

Not necessarily. Section 6.3.2.3 (of the 2007 edition) defines several exceptions for which new-energy reheat is permitted. Exception A in the standard permits the use of new energy for reheat after the supply airflow is reduced to a defined limit. The minimum airflow setting for most zones in a VAV system is less than the limits defined by this section of Standard 90.1. For further discussion, see "Simultaneous heating and cooling limitation," p. 130.

Reheating (tempering) supply air at the VAV terminal units

Both overcooling and increased humidity levels can be avoided by reheating (tempering) the supply air after it reaches the minimum airflow setting for the VAV terminal unit. Figure 93 illustrates the effect of adding a heating coil to the VAV terminal unit that serves this same example classroom. When the primary airflow is reduced to the minimum airflow setting of 700 cfm (0.33 m³/s), the heating coil in the VAV terminal warms the 55.7°F (13.1°C) primary air (PA) to 57.9°F (14.4°C) before delivering it to the zone. This avoids overcooling the classroom and, on this example mild and rainy day, results in a relative humidity of 60 percent.

Figure 93. Reheat (tempering) at the VAV terminals



Options include heating coils mounted on the VAV terminals, fan-powered VAV terminals, dual-duct air distribution, and even radiant heat located within the zone.

Considerations for reheating (tempering) in a VAV system:

- Certain zones in a VAV system (such as perimeter zones and interior conference rooms) may require reheat, even when high sensible-cooling loads exist elsewhere. To curb operating costs, consider using parallel fan-powered VAV terminals in these zones.
- When using electric heating coils, comply with the manufacturer's guidelines for minimum airflow limits across the heating elements to ensure safe operation.
- If using a boiler as the heat source for reheat, make sure that it is available to operate during the cooling season when reheat (tempering) may be needed.
- Consider the use of on-site recovered heat to reduce operating costs. For example, the heat rejected by the condenser of a water-cooled chiller can be distributed to VAV terminals throughout the building. Condenser heat recovery is particularly well-suited for supply-air tempering applications because it provides the relatively small amount of heat needed for tempering and allows the primary heating equipment (boilers, for example) to be turned off during the summer. Any water-cooled chiller can be used to provide sensible heat for supply-air tempering (see "Condenser heat recovery," p. 88).

Methods for improving dehumidification performance

As mentioned previously, VAV systems can provide effective dehumidification over a wide range of load conditions for most applications. Avoiding supply-air-temperature reset during humid weather, and providing heat for tempering at low load conditions, are generally sufficient for most comfort-cooling applications.

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

If necessary, the basic design of the VAV system can be altered to enhance dehumidification performance:

- *Investigate the practicality of delivering colder supply air.*
  Lowering the dry-bulb temperature of the air leaving the cooling coil causes more moisture to condense out of the supply air. In a VAV system, this colder, drier supply air results in a drier zone (lower humidity) at all load conditions.

  Another benefit of a cold-air VAV system is reduced supply airflow, which can allow for the use of smaller VAV terminals, air-handling units, and ductwork, and can save supply-fan energy. Increased reheat energy and fewer hours of airside economizer operation partially offset the supply-fan energy savings. Therefore, intelligent system control is crucial to fully realize the potential savings. (See "Cold-Air VAV Systems," p. 147.)

- *Add a desiccant dehumidification wheel in series with the cooling coil.*
  Figure 94 depicts a VAV air-handling unit with a desiccant dehumidification wheel configured in series with the cooling coil. The desiccant 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 94. VAV air-handling unit with a series desiccant wheel

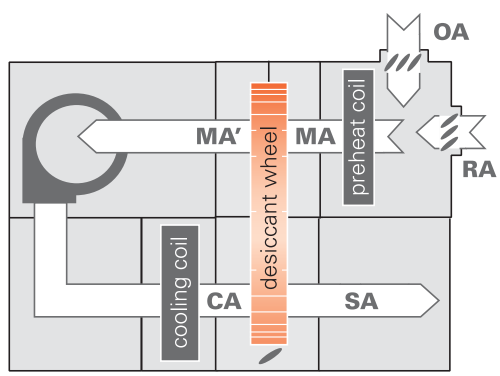
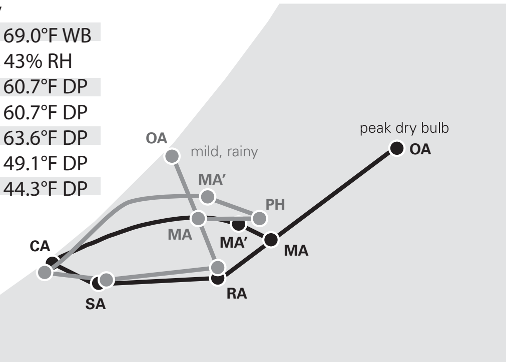


Figure 95 shows the performance of this VAV system with a series desiccant wheel. Air leaves the cooling coil (CA) at a dry-bulb temperature of 51.3°F (10.7°C) and a dew point of 50°F (10°C). The series desiccant wheel adsorbs water vapor, drying the supply air (SA) to a dew point of 42.9°F (6.1°C). Sensible heat added by the adsorption process raises the dry-bulb temperature of the supply air to 55.7°F (13.1°C). The wheel rotates into the mixed air stream (MA), where water vapor released from the wheel passes into the mixed air (MA') and then condenses on the cold coil surface. Using the same example classroom from the previous section, delivering the supply air at a dew point of 42.9°F (6.1°C) results in 40 percent relative humidity in the zone.

Figure 95. Example performance of a series desiccant wheel in a VAV system

Design condition	Peak dry bulb	Mild, rainy
OA	96.0°F DB, 76.0°F WB	70.0°F DB, 69.0°F WB
RA	74.0°F DB, 40% RH	74.0°F DB, 43% RH
MA	80.6°F DB, 54.9°F DP	72.0°F DB, 60.7°F DP
PH		78.7°F DB, 60.7°F DP
MA’	76.2°F DB, 59.8°F DP	70.9°F DB, 63.6°F DP
CA	51.3°F DB, 50.0°F DP	50.1°F DB, 49.1°F DP
SA	55.7°F DB, 42.9°F DP	57.9°F DB, 44.3°F DP



Basically, adding the series desiccant wheel changes the dehumidification performance of the traditional cooling coil, trading sensible capacity for more latent capacity. This system is able to deliver drier air without requiring a significantly colder leaving-coil temperature—without the desiccant wheel, the cooling coil would need to cool the air to about 43°F (6.1°C) dry bulb in order to achieve a supply-air dew point of 42.9°F (6.1°C).

When the relative humidity of the air entering the upstream side of the wheel is high (on a mild rainy day, for example), it may be necessary to preheat this air to lower its relative humidity (RH) and allow the desiccant to regenerate. Figure 95 also shows the performance of the series desiccant wheel on a mild, rainy day. In this example, the mixed air (MA) is preheated from 72.0°F (22.2°C) to 78.7°F (25.9°C) in order to lower the RH of the air entering the upstream side of the wheel (PH). This allows the wheel to supply air at the desired 57.9°F (14.4°C) dry bulb, but at a dew point of 44.3°F (6.8°C). The resulting relative humidity in the zone is 43 percent.

For more information on using a dedicated outdoor-air system to improve dehumidification performance, refer to the Trane application manual titled *Dehumidification in HVAC Systems* (SYS-APM004-EN).

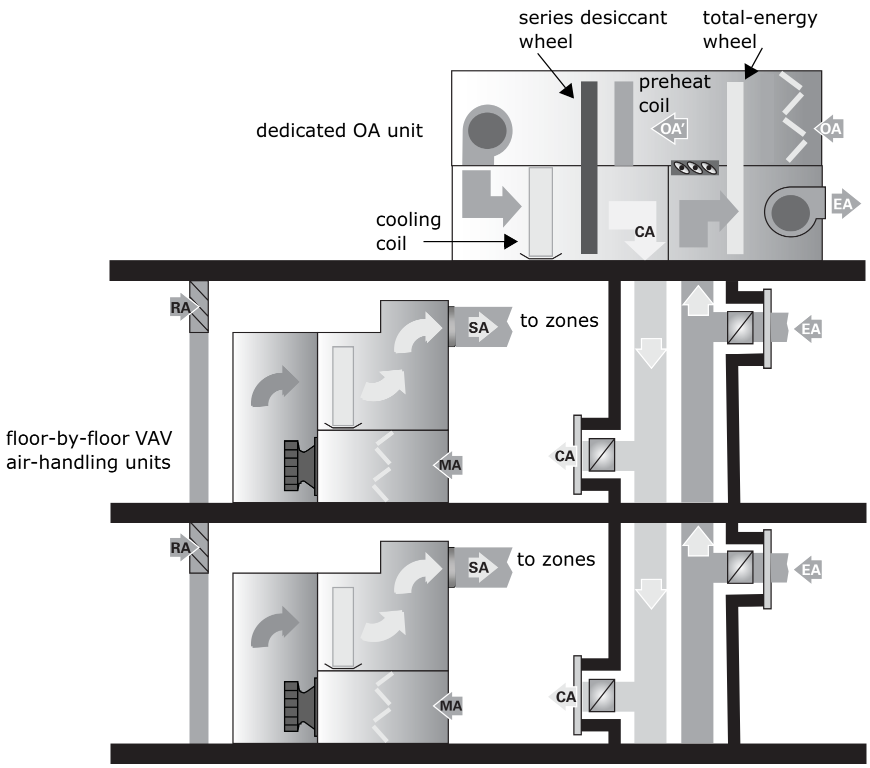
While this manual is focused on chilled-water VAV systems, dedicated outdoor-air systems (such as the example shown in Figure 96) are also commonly used with fan-coils, water-source heat pumps, PTACs, small packaged rooftop units, DX split systems, chilled beams or chilled ceilings, and variable-refrigerant-flow (VRF) systems.

- *Condition the outdoor air with a separate, dedicated unit.* Another way to improve dehumidification performance is to use a dual-path air-handling unit (see "Dual-path configuration," p. 13) or a dedicated outdoor-air unit to separately dehumidify all of the outdoor air to a dew point that is drier than the zones. This conditioned outdoor air is then either:

  1. Ducted directly to each zone
  2. Ducted directly to individual, dual-duct VAV terminals that serve each zone (see Figure 89, p. 116)
  3. Ducted to the outdoor-air intake of one or more VAV air-handling units (Figure 96)

  The example dedicated OA unit depicted in Figure 96 includes a total-energy wheel to precondition the entering OA (see “Air-to-Air Energy Recovery,” p. 160) and a series desiccant wheel to enable the unit to deliver drier air without requiring a significantly colder leaving-coil temperature. In this configuration, the dedicated OA unit delivers the conditioned outdoor air (CA) to floor-by-floor VAV air-handling units.

Figure 96. Example dedicated OA system delivering conditioned OA to floor-by-floor VAV air-handling units



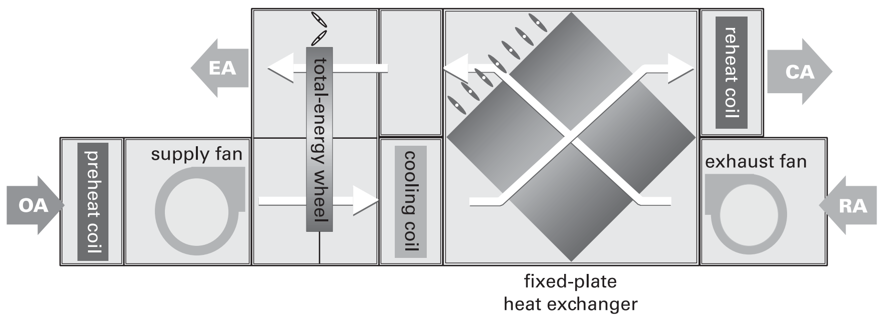
When a series desiccant wheel is used in a dedicated OA unit, it may be necessary to preheat the entering outdoor air (OA) when the relative humidity is high (on a mild rainy day, for example). Using the preheat coil to raise the dry-bulb temperature slightly (5°F to 20°F [3°C to 11°C]) lowers the relative humidity. Lowering the relative humidity of the air entering the regeneration side of the wheel allows the desiccant to reject water vapor to the air, thus enabling it to adsorb water vapor from the 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 (see “Condenser heat recovery,” p. 88). A small, inexpensive electric heater is another option.

Alternatively, a total-energy wheel can be added to the system (Figure 96). When high RH conditions occur, the total-energy wheel transfers moisture 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.

Another example dedicated OA unit configuration includes a total-energy wheel to precondition the entering OA and a fixed-plate heat exchanger to reheat the dehumidified OA when desired (Figure 97).

Figure 97. Example dedicated OA unit with total-energy recovery and a fixed-plate heat exchanger



The fixed-plate heat exchanger is equipped with integral face-and-bypass dampers (an empty core through the middle of the heat exchanger allows air to bypass) to modulate the amount of heat recovered and avoid overheating the conditioned air (CA). This allows the dedicated OA unit to deliver the dehumidified air cold (rather than reheated to “neutral”) when it is hot outside, but then to reheat the dehumidified air to avoid overcooling spaces during mild weather.

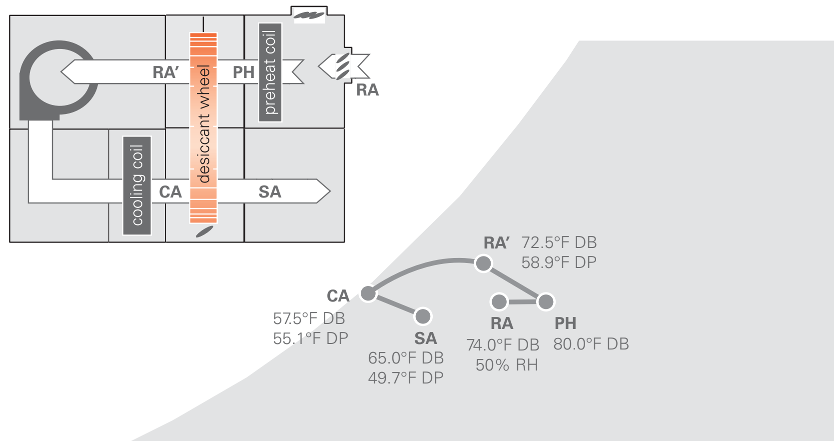
This configuration may be more typical for applications that do not require dehumidification to as low a dew point as can be achieved with the series desiccant wheel (Figure 96).

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 may be to turn on the central air-handling unit and dehumidify recirculated air (OA damper is closed). However, there is typically very little sensible load in the spaces at this time, so the reheat coils in the VAV terminals may need to be activated to prevent overcooling. If a hot-water heating system is used, this also requires turning on the boiler and circulation pumps.

An energy-saving alternative is to equip the VAV air-handling unit with a series desiccant wheel. When after-hours dehumidification is needed, the AHU conditions only recirculated air. The series desiccant wheel allows the air-handling unit to dehumidify the air to a lower dew point, but deliver it at a warmer dry-bulb temperature than if using a cooling coil alone (Figure 98). This saves energy by minimizing the need for using “new” energy to reheat remotely at the VAV terminals, maybe even allowing the hot-water boiler and pumps to remain off.

Figure 98. Example performance of a series desiccant wheel used for after-hours dehumidification



Humidification

In some buildings, or specific areas within a building, *minimum* humidity levels must be maintained for comfort or process requirements.

Types of humidifiers

For buildings that commonly use VAV systems, steam humidifiers are the most common type of humidifier used.

Application considerations

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 21, "Humidifiers," in the 2008 *ASHRAE Handbook—HVAC Systems and Equipment* (www.ashrae.org).

When including humidification equipment in a VAV system, consider the following:

- *Add moisture to the supply air stream, not the outdoor air stream.* When it is cold outside, the outdoor air being brought in for ventilation does not have much capacity to hold additional moisture. After the outdoor air has mixed with the warmer return air and/or has been warmed by a heating coil, it has a much greater capacity to absorb moisture.
- *Avoid oversizing the humidification equipment.* An oversized humidifier typically results in unstable control, with large swings in humidity levels. In applications 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 surface temperatures below the indoor dew point are likely.

  Sizing the humidifier can be particularly challenging if the VAV system includes an airside economizer. 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 economizer may open the OA damper when it is relatively 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 steam 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.

Energy Efficiency

Decisions based solely, or primarily, 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, chilled-water VAV systems are, in many ways, inherently energy efficient. Reducing the quantity of air delivered at part load allows this system to reduce the fan energy required to move this air. And, the reduced airflow across the cooling coil allows the chilled-water plant to unload, reducing cooling energy at part load. In addition, various control strategies and design options provide the opportunity to further reduce the energy use of this type of system.

Minimum efficiency requirements

For more information, refer to ANSI/ASHRAE/IESNE 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 chillers or boilers, 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 codes. Its purpose is "to provide minimum requirements for the energy-efficient design of 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 focuses on only a few of the HVAC-related requirements that are of specific interest to designers of typical chilled-water VAV systems.

Note: Because ASHRAE 90.1 is under continuous maintenance, it can change frequently. This manual is based on the 2007 published version of the standard. Refer to the most current version for specific requirements.

Minimum equipment efficiencies

ASHRAE 90.1 contains minimum full- and part-load efficiency requirements for various types of HVAC equipment (water chillers, packaged rooftop units, boilers, gas-fired burners, cooling towers, and so on). Meeting both the full- and part-load efficiencies is mandatory, whether the prescriptive or Energy Cost Budget (ECB) method of compliance is used.

Maximum allowable fan system power

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. Therefore, ASHRAE 90.1 prescribes a limit to the allowable fan system power.

This limit applies to all fans that operate at peak design (cooling) conditions. For VAV systems, this typically includes supply, return, relief, and exhaust fans, as well as series fan-powered VAV terminals.

- *Parallel fan-powered VAV terminals*
  Since the fans inside parallel fan-powered VAV terminals do not operate in the cooling mode, when the supply fan is operating at peak design conditions, they do not need to be included.
- *Series fan-powered VAV terminals*
  Since the fans inside series fan-powered VAV terminals operate continuously during the occupied mode, they are operating when the supply fan is operating at peak design conditions, so they must be included.
- *Central relief fan*
  In some systems, the central relief fan operates only during the economizer mode, but *not* at the peak design (cooling) condition. In this case, it does not need to be included. On the other hand, if the central relief fan *does* operate at the peak design (cooling) condition, it must be included.
- *Central return fan*
  A central return fan operates at the peak design (cooling) condition, so it must be included.
- *Small, local 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.

The 2007 version of ASHRAE 90.1 includes two options for compliance (Table 24).

Table 24. Fan power limitation for VAV systems*	
Option 1: Allowable nameplate motor power	hp <= $CFM_{supply} \times 0.0015$ (kW <= $L/s_{supply} \times 0.0024$)
Option 2: Allowable fan input (brake) power	bhp <= $CFM_{supply} \times 0.0013 + A$ (kW <= $L/s_{supply} \times 0.0021 + A$)

* Excerpt from Table 6.5.3.1.1A of ASHRAE Standard 90.1-2007. © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org

Option 1 is based on nameplate motor power for the fan. It is easier to apply than Option 2, but not as flexible. To comply using Option 1, the sum of the nameplate motor powers for all fans that operate at peak design conditions must be no greater than the value listed in Table 24. For example, if the design supply airflow is 30,000 cfm (14.2 m^3/s or 14,200 L/s), the total allowable nameplate motor power for the VAV system is 45 hp (34 kW).

Allowable Nameplate Motor Power = 30,000 cfm x 0.0015 = 45 hp

(Allowable Nameplate Motor Power = 14,200 L/s x 0.0024 = 34 kW)

As mentioned previously, this limit applies to the sum of all fans that operate at peak design (cooling) conditions.

Option 2 is based on input power to the fan shaft (brake horsepower). To comply using Option 2, the sum of the fan input (brake) powers for all fans that operate at peak design conditions must be no greater than the value listed in Table 24. This fan power limitation includes the following adjustment factor to account for special filters and other devices:

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

$$[A = \Sigma\,(PD \times L/s_{device} / 650{,}100)]$$

where,

PD = pressure drop adjustment for each applicable device (Table 25), in. H_2O (Pa)

CFM_{device} (L/s_{device}) = design airflow through each applicable device (Table 25), cfm (L/s)

Table 25. Fan power limitation pressure drop adjustments*	
Pressure drop credit	**Adjustment (PD)**
Fully ducted return and/or exhaust air systems	0.5 in. H_2O (125 Pa)
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	2x 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
Heat recovery device (e.g., wheel, coil loop, heat pipe, fixed-plate heat exchanger)	Pressure drop through device at fan system design condition
Evaporative humidifier/cooler in series with another cooling coil	Pressure drop through device at fan system design condition
Sound attenuation section	0.15 in. H_2O (38 Pa)

* Excerpt from Table 6.5.3.1.1B of ASHRAE Standard 90.1-2007. © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org

For example, consider a 30,000-cfm (14,200-L/s) VAV system that includes a MERV 13 filter and a total-energy wheel. The filter is installed in the mixed air stream (outdoor and recirculated air), so the design airflow through it is 30,000 cfm (14,200 L/s). At design cooling conditions, the outdoor airflow through the supply-side of the wheel is 10,000 cfm (4,720 L/s) with a pressure drop of 0.8 in. H_2O (200 Pa). The airflow through the exhaust-side of the wheel is 8,000 cfm (3,780 L/s) with a pressure drop of 0.7 in. H_2O (175 Pa).

For this example, the pressure drop adjustment (A) is 9.8 bhp (7.3 kW)...

A = (0.9 in. H_2O × 30,000 cfm / 4131) + (0.8 in. H_2O × 10,000 cfm / 4131) + (0.7 in. H_2O × 8,000 cfm / 4131) = 9.8 bhp

[A = (225 Pa × 14,200 L/s / 650,100) + (200 Pa × 4,720 L/s / 650,100) + (175 Pa × 3,780 L/s / 650,100) = 7.3 kW]

...and the total allowable fan input (brake) power for the VAV system is adjusted to 48.8 bhp (37.1 kW).

Allowable Fan Input Power = 30,000 cfm x 0.0013 + 9.8 = 48.8 bhp

(Allowable Fan Input Power = 14,200 L/s x 0.0021 + 7.3 = 37.1 kW)

For more information on selecting minimum airflow settings that comply with both ASHRAE Standard 62.1 and 90.1, refer to the Trane *Engineers Newsletter* titled "Potential ASHRAE Standard Conflicts: Indoor Air Quality and Energy Standards" (ADM-APN030-EN) and to the Trane *Engineers Newsletter Live* broadcast DVD titled "ASHRAE Standards 62.1 and 90.1 and VAV Systems" (APP-CMC034-EN).

Simultaneous heating and cooling limitation

For a comfort-cooling application, Section 6.5.2.1 of ASHRAE 90.1-2007 places limits on the energy used to reheat air that has been previously cooled (either by refrigeration equipment or an airside economizer cycle). This impacts VAV systems that use heating coils in the VAV terminal units to prevent overcooling zones under low cooling loads. Before this heating coil can be activated, the primary airflow must first be reduced to the largest of the following:

a1) The volume of outdoor air required to meet the ventilation requirements of Section 6.2 of ASHRAE Standard 62.1 for the zone (V_{oz})

a2) 0.4 cfm/ft² (2 L/s/m²) of the zone conditioned floor area

a3) 30 percent of zone design supply airflow

a4) 300 cfm (140 L/s) … this exception is for zones whose design supply airflows total no more than 10 percent of the total system airflow

a5) Any higher rate that can be demonstrated, to the satisfaction of the authority having jurisdiction, to reduce overall system annual energy usage by offsetting reheat/recool energy losses through a reduction in outdoor air intake for the system

Exception a1. For multiple-zone VAV systems, the minimum primary airflow setting typically must be greater than the zone ventilation requirement (V_{oz}) to avoid excessively high zone outdoor-air fractions (Z_d), which could significantly increase the required system intake airflow (see "Ventilation," p. 101). Therefore, exception (a1) does not usually apply to multiple-zone VAV systems, but it does apply to single-zone VAV systems (see "Single-Zone VAV," p. 157) and dedicated outdoor-air systems (see "Dedicated outdoor-air systems," p. 115).

An addendum (h) to ASHRAE Standard 90.1-2007 removes exceptions a2 (0.4 cfm/ft² [2 L/s/m²]) and a4 (300 cfm [140 L/s]) from Section 6.5.2.1, and adds a new exception that allows the "dual-maximum" control strategy as long as the maximum heating primary airflow ≤ 50% of maximum cooling primary airflow (see Figure 50, p. 57).

Exceptions a2 and a3. In many VAV systems, the minimum airflow setting for each VAV terminal can be set equal to either 0.4 cfm/ft² (2 L/s/m²) or 30 percent of the design supply airflow (whichever is the largest). In this case, ASHRAE 90.1 permits the use of "new" energy for reheat.

Exception a4. For some smaller zones, the 300 cfm (140 L/s) limit may allow for a minimum airflow setting that is higher than 0.4 cfm/ft² (2 L/s/m²) or 30 percent of the design supply airflow. However, this exception can only be used for a limited number of zones since the sum of the design airflows for the zones using this exception can represent no more than 10 percent of design system (fan) airflow.

Exception a5. Alternatively, ASHRAE 90.1 allows the use of reheat at higher minimum airflow settings if it results in a reduction of overall system energy use. This exception recognizes that increasing the minimum airflow setting in the "critical" zone increases system ventilation efficiency and results in lower system intake airflow. However, this has to be demonstrated to and approved by the authority having jurisdiction (AHJ), so it may not be possible in all cases. Some engineers use a spreadsheet analysis to demonstrate this, while others use whole-building energy simulation software. One approach is to use a control strategy, such as ventilation reset (see "Ventilation optimization," p. 205) to minimize overall system energy use.

In addition, exception c under Section 6.5.2.1 allows any amount of air to be reheated if at least 75 percent of the energy required for reheat is recovered on-site. In chilled-water VAV systems that use hot-water coils in the VAV terminals, heat recovery from the chilled-water system may provide enough of the reheat energy to meet this exception (see "Condenser heat recovery,"

p. 88). In this case, the minimum primary airflow setting can be higher than the limits listed in exception a.

VAV fan control

If a variable-volume supply fan motor is larger than 10 hp (7.3 kW), ASHRAE 90.1 requires the motor to be equipped with a variable-speed drive (or some other device that provides a comparable reduction in fan energy at part load).

In addition, the standard requires that:

> Static pressure sensors used to control variable air volume fans shall be placed in a position such that the controller setpoint is no greater than one-third the total design fan static pressure, except for systems with zone reset control [fan-pressure optimization] ... If this results in the sensor being located downstream of major duct splits, multiple sensors shall be installed in each major branch to ensure that static pressure can be maintained in each.

This prohibits locating the duct static pressure sensor at the discharge of the supply fan, unless the fan-pressure optimization control strategy is used. Finally, for VAV systems that use direct digital controls (DDC) on the VAV terminal units, fan-pressure optimization is required. (See "Supply-fan capacity control," p. 172, and "Fan-pressure optimization," p. 200.)

Demand-controlled ventilation

For more information on demand-controlled ventilation, see "Ventilation control," p. 184, and "Ventilation optimization," p. 205.

For VAV systems that use direct digital controls (DDC), if the design system-level outdoor airflow is greater than 3000 cfm (1.4 m^3/s) or if the air-handling unit is equipped with a modulating outdoor-air damper (or airside economizer), ASHRAE 90.1 requires some method of demand-controlled ventilation for any zone larger than 500 ft^2 (50 m^2) that has a design occupancy of more than 40 people/1000 ft^2 of floor area (40 people/100 m^2).

Small systems, in which the design system-level outdoor airflow is less than 1200 cfm (0.6 m^3/s), or systems for which ASHRAE 90.1 requires exhaust-air energy recovery, are exempt from this requirement. In addition, any zone in which the design supply airflow is less than 1200 cfm (0.6 m^3/s) is exempt.

Opportunities to further reduce system energy use

For more information on energy and high-performance-building initiatives, refer to the Trane *Engineers Newsletter* titled "Energy and Environmental Initiatives" (ADM-APN008-EN).

While local building codes might include requirements for *minimum* levels of energy efficiency, many building owners desire even higher efficiency levels for their buildings. In addition, programs like ENERGY STAR® (administered by the U.S. Environmental Protection Agency and Department of Energy [DOE]), Rebuild America (administered by the 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 chilled-water VAV 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 chilled-water VAV systems.

Table 26. Potential energy-savings strategies for chilled-water VAV systems

VAV air-handling unit	
Upsize AHU casing	p. 18
High-efficiency fans	p. 32
Lower pressure drop filters	p. 43
Cold-air distribution	p. 147
Precondition outdoor air with air-to-air energy recovery	p. 160
Series desiccant wheel for low dew point applications	p. 122
Dedicated OA system	p. 123
Dual-fan, dual-duct VAV system	p. 165
Fixed- or differential-enthalpy control of the airside economizer	p. 174
Evaporative cooling	p. 21
Chilled-water and hot-water systems	
High-efficiency water chillers	p. 79
Water-cooled chiller plant	p. 79
Lower flow rates on the chilled-water and condenser-water distribution systems	p. 81
Variable-flow pumping (including variable-primary-flow systems)	p. 86
Waterside economizer	p. 91
Chillers in series	SYS-APM001-EN
Condenser heat recovery for reheat	p. 88
Thermal storage	p. 92
Condensing boilers and lower water temperatures	p. 92
VAV terminal units	
Parallel, fan-powered VAV terminals for those zones that require heat	p. 59
Electronically commutated motors (ECM) on fan-powered VAV terminals	p. 60
System-level controls	
Optimal start/stop	p. 198
Unoccupied (nighttime) economizing	p. 199
Fan-pressure optimization	p. 200
Supply-air-temperature reset	p. 202
Ventilation optimization (demand-controlled ventilation + ventilation reset)	p. 205
Pump-pressure optimization	p. 208
Chilled-water-temperature reset	p. 208
Chiller-tower optimization	p. 210

For more information on the Trane's TRACE™ 700 or System Analyzer™ building analysis software programs, visit www.trane.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 99 shows the potential energy savings of using various HVAC strategies in an example office building that has a water-cooled, chilled-water

VAV system. The system in the "proposed building" incorporates the following energy-saving strategies:

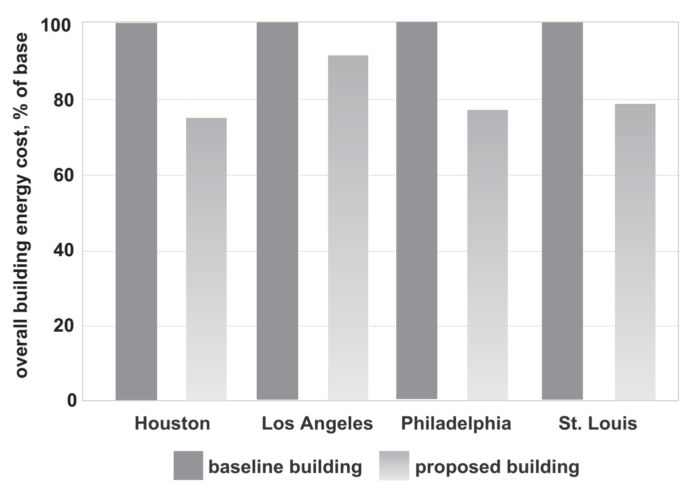
- Low-flow condenser-water distribution system (15°F [8.3°C] ∆T)
- Low-flow chilled-water distribution system (14°F [7.8°C] ∆T)
- High-efficiency, water-cooled centrifugal chillers (5.96 COP, or 7.33 COP for Los Angeles)*
- Chiller-tower optimization control strategy
- High-efficiency, direct-drive plenum fans in the VAV air-handling units
- Total-energy wheel to precondition the entering outdoor air (except for Los Angeles)
- Cold-air distribution (48°F [9°C] supply-air temperature) with same size ductwork
- Improved supply-air-temperature reset control strategy
- Ventilation optimization control strategy
- Parallel fan-powered VAV terminal units serving the perimeter zones
- Airside economizer with fixed enthalpy control (fixed dry-bulb control in Los Angeles)

* For a centrifugal chiller operating at these water temperatures and flow rates, which differ from the ARI standard rating conditions, the minimum efficiency required by ASHRAE 90.1-2007 is 5.11 COP (5.90 COP for Los Angeles).

Note that the energy savings in this example is due only to changes in the design and control of the HVAC system. There were no changes made to the building envelope, no lighting or plug load reductions, and no cross-cutting strategies like shading or daylighting. For any actual project, it is strongly recommended that these other types of energy-saving strategies also be considered.

For this example, even with no improvements to the building envelope, lighting, or other systems, the "proposed" chilled-water VAV system reduced the overall building energy cost by 26 percent for the building in Houston, by 10 percent in Los Angeles, by 24 percent in Philadelphia, and by 21 percent in St. Louis.

Figure 99. Example energy savings versus a LEED 2009 baseline building



There is a real potential to save energy in chilled-water VAV systems through optimized system design and control strategies. This energy 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 titled *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 building.

Sound from HVAC equipment impacts the sound levels in the building, 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 job. With sufficient experience, this may be all a design engineer needs to create an installation that is free of sound 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. Chilled-water VAV systems contain several sound sources and each should be reviewed separately. Each source has a unique sound quality and level, but 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 ARI Standard 260, *Sound Rating of Ducted Air Moving and Conditioning Equipment,* and ARI Standard 880, *Performance Rating of Air Terminals*. This ensures that the sound data accurately reflects the contributions of all the sound sources, and accounts for the effects of the cabinetry. Outdoor sound data should be measured in accordance with ARI Standard 370, *Sound Rating for Large Outdoor Refrigerating and Air-Conditioning Equipment*. Indoor water chillers should be rated in accordance with ARI Standard 575, *Method of Measuring Machinery Sound within an Equipment Space*.

The indoor sound data is used for the supply and return sound paths, while outdoor sound data is used for the wall/roof transmission sound path.

Outdoor sound data is also used to calculate the sound level for outdoor receiver locations, such as at the lot line.

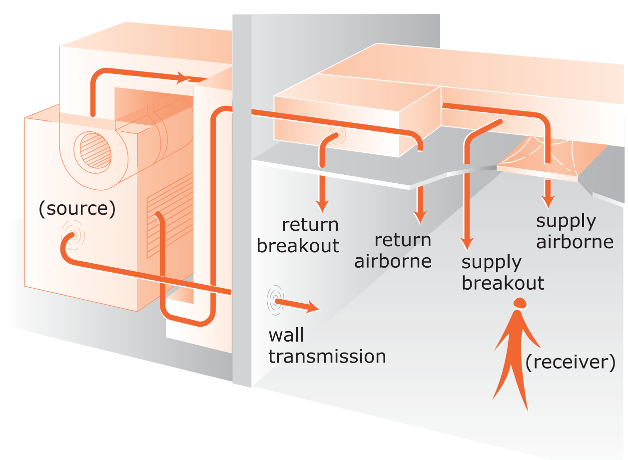
Receiver

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 theater, the neighboring lot line, and so on. Typically, a building has several indoor receiver locations, and possibly even a few outdoor locations if there is a concern about sound at neighboring properties.

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 100). For example, sound from the supply fan follows the supply ductwork and enters the occupied space through the supply-air diffuser. Supply fan sound also travels through the walls of the supply duct (breakout), and then through the ceiling into the space. Sound from the intake of the air-handling unit makes its way to the space through the return ductwork and grilles, or through the wall that separates the mechanical room from the occupied space.

Figure 100. Sound paths in a VAV air-handling unit



There are several different types of equipment that make up chilled-water VAV systems and each requires acoustical analysis. For each sound source it is necessary to determine the path 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. In VAV systems, the supply airborne path is also influenced by the discharge sound from the VAV terminal unit. Return airborne sound travels against the direction of airflow back through the return air path.

- *Duct breakout*
  Sound passes through walls of the ductwork (supply or return), into the ceiling plenum, and then through the ceiling into the occupied space.
- *Radiated*
  Sound radiated from the casing of the equipment (VAV terminal, air-handling unit, water chiller, etc.) and travels through whatever is between the equipment and the receiver location. For a piece of equipment that is located outdoors, this radiated sound travels through a wall or roof. For equipment located indoors, the radiated sound travels either through walls or, in the case of a VAV terminal unit, through the ceiling and into the occupied space.
- *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, 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 HVAC project. There are always implicit, often subjective, expectations, and it is much easier if you understand these expectations before designing the system.

Acoustical goals vary depending on how the space is used. Once the 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.
- The entire building does not have the same acoustical requirements. Bathrooms and hallways do not need to be as quiet as executive offices and conference rooms. A low-cost, quiet installation takes advantage of this point.
- Successful acoustics requires a team effort, including the owner, 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 spaces closest to the sound sources are typically the loudest. If spaces distant from the unit have sound targets that are considerably below the level required in the space closest to the source, these spaces should also be analyzed. Common examples include conference rooms, executive offices, 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.trane.com.

Once each path has been identified, individual elements are 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 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 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 and 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 receiver location, the design does not need to be changed, although it may be reviewed for potential cost reductions.

If the estimate from the analysis 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 costs 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

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 may be used to identify potential problems for each component of the chilled-water VAV system. Consider both source attenuation and path attenuation when determining the most cost effective way to achieve the acoustical goals.

Air-cooled chillers

The acoustical advantage of an air-cooled chiller is that the compressor sound is generated outside of the building, allowing this sound source to be placed away from the building. If the air-cooled chiller is located near the building, it must be far away from operable windows, doors, and sound-sensitive areas. When possible, locate the chiller near a space with a high tolerance for sound, such as a storage or equipment room. Walls near the chiller must have sufficient sound transmission loss to meet the acoustical goals of the adjacent space. Special consideration is required for air-cooled chillers that are located on the roof. Typical built-up roofs have low transmission loss, which makes unit placement critical. Vibration isolation must also be tailored to the specific roof structure.

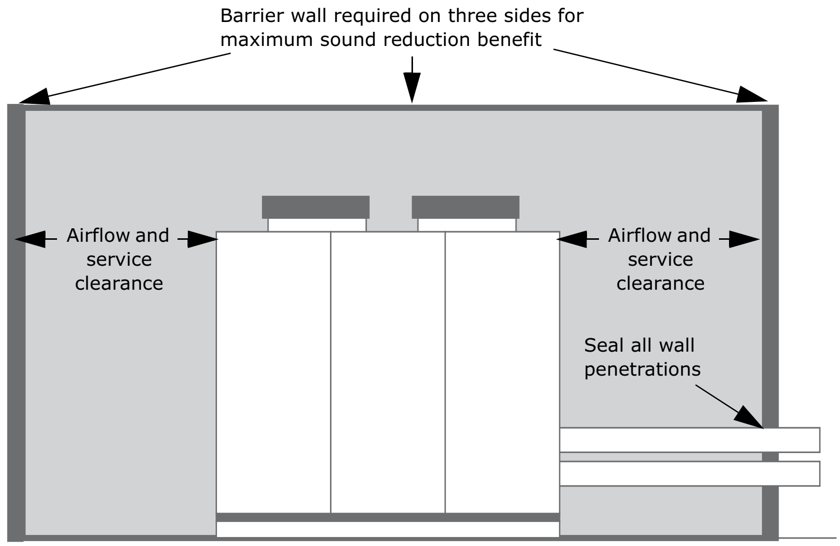
The majority of noise complaints from air-cooled chillers come from neighboring properties or buildings. Most areas of the country have sound ordinances that define the maximum amount, and sometimes quality, of sound permitted at the property boundary (or lot line). In most cases, the lot line maximums are given in terms of dBA, with different maximum levels for daytime and nighttime periods. However, sometimes an ordinance includes limits by octave band and limits for special types of sound, such as tones or impulse sound. In any case, the ordinance defines the acoustical target for the project.

If the acoustical analysis indicates that the air-cooled chiller will exceed the ordinance, consider moving the chiller, adding sound attenuation options to the chiller, or building a barrier wall.

- *Move the chiller.*
  Sound diminishes with distance, so consider moving the chiller away from the sound-sensitive areas and using existing barriers (such as hills or buildings) to block the sound.
- *Investigate sound attenuation options available from the chiller manufacturer.*
  Air-cooled chillers are considered to contain three sound sources: compressors, condenser fans, and structure. Compressors and portions of the structure can be attenuated using sound wraps. Attenuating the fans is more difficult because airflow through the condenser coils and fans cannot be blocked by any type of enclosure. Purchasing the chiller with special low-noise fans is the simplest way to attenuate fan sound.
- *Build a barrier wall.*
  Sound from the entire chiller can be directed away from the lot line, or other sensitive areas, by using a barrier wall. Barrier walls generally are not designed to absorb sound (although sound absorptive materials are sometimes added to the walls). Rather, barriers are most effective when used to redirect the sound toward a less sound-sensitive area. For this reason, barriers should be constructed around three sides of the chiller

with the open side facing away from the sound-sensitive area (Figure 101).

Figure 101. Barrier wall for an air-cooled chiller (elevation view)



Barrier walls should be constructed with sufficient mass to lower transmitted sound by 10 dB in all octave bands and must be free of any openings. They should be at least 2 ft (0.6 m) taller than the chiller, but may need to be even taller to achieve the desired sound reduction. Follow the manufacturer's recommendations for airflow clearance between the barrier and the chiller. In cases where an extreme sound reduction is required, it is possible to build a barrier with built-in sound reduction baffles.

Water-cooled chillers

Water-cooled chillers are typically installed inside the building in a mechanical equipment room. The critical sound path for this equipment is radiated sound from the chiller that passes through the walls or other parts of the structure and into the occupied spaces. Vibration isolation should be considered for the chiller, as well as any connections to the chiller (such as piping and electrical conduit).

Attempting to attenuate the source is not a common approach for water-cooled chillers, but occasionally portions of the chiller may be encased in a sound blanket. Beside selecting equipment with lower sound levels, the most common approach to avoid sound problems with water-cooled chillers is to locate the equipment room away from any sound-sensitive areas and/or construct the surrounding walls with sound transmission loss sufficient to achieve the acoustical goals in the adjacent spaces. Pay particular attention to doorways and any penetrations through the walls. Keep gaps to a minimum and seal them with an acoustical mastic compound.

The most common sound problems with chillers installed indoors are due to replacing an older chiller with a new chiller that has higher sound levels or a different sound spectrum. In this case, it is generally not possible to relocate the chiller, and it can be very difficult to increase the sound transmission loss of the structure. The best approach may be to build an enclosure around the chiller. In general, these enclosures must be large enough to allow an operator to enter and log the performance of the chiller. In addition, they should be designed to be disassembled and moved out of the way when the chiller requires major service.

VAV air-handling units

For more information, refer to the Trane *Engineers Newsletter* titled "Sound Ratings and ARI Standard 260" (ENEWS-29/1).

Air-handling units have the greatest diversity in sound paths of all the equipment in a chilled-water VAV system. Sound from these units enters the occupied space via all of the sound paths described earlier in this section: sound follows the supply and return airflow paths and is radiated from the casing and from the inlet and outlet openings.

However, air-handling units also offer the greatest flexibility of attenuation options. It may be possible to configure the unit in a blow-thru or draw-thru arrangement, individual modules can be stacked on top of each other, they offer a variety of fan types and locations, and they can be equipped with acoustical lining, silencers, or discharge plenums. All of these options impact the sound from the unit. In addition, the air-handling unit can be located inside or outside of the building.

Consult with the equipment manufacturer to ascertain how the sound data is determined. Indoor sound data for air-handling units should be measured in accordance with ARI Standard 260, *Sound Rating of Ducted Air Moving and Conditioning Equipment*, to properly account for the acoustical effect of every unit configuration. In addition, work with the manufacturer to determine which configurations result in the desired sound level for the application.

An acoustical analysis helps determine which of the sound paths are critical, and which source of unit sound is most important to attenuate.

- *Casing radiated sound*
  Casing radiated sound from *outdoor* air-handling units is rarely a problem, unless the unit includes an exhaust fan and is close to a sound-sensitive area. If the unit cannot be selected with sufficiently low casing radiated sound to meet the outdoor acoustical goals, the solutions are similar to those used with air-cooled chillers. Attempt to locate the air-handling unit away from the sound-sensitive area or increase the transmission loss of the adjacent roof or walls. Barrier walls can be used to redirect sound away from sensitive areas, but the barrier must allow outdoor air to reach the air-handling unit and avoid recirculating exhaust air back through the intake.

  Note: If the unit is relocated, and the supply and return ducts are routed above the roof, some of the airborne sound will break out to the outdoors, thereby reducing the sound level in the supply and return airborne paths.

  Casing radiated sound from *indoor* air-handling units must not be overlooked, especially when the return air is not ducted directly to the unit. Sound radiated from the casing can be transmitted through the walls of the equipment room into adjacent spaces. When the return air is not ducted directly to the unit, sound radiates into the equipment room from both the casing and the unit inlet. This increases the equipment room sound level, so more sound is available to transfer through the equipment room walls and/or follow the return air path (opposite the direction of airflow) to the occupied space.

Finally, use sufficient vibration isolation to prevent structure-borne sound. Fan housings should have flexible materials where they connect to the air-handling unit, and the base structure should be isolated.

- *Return airborne sound*
  If the return air is not ducted directly to the air-handling unit, the acoustical analysis for this sound path should begin with the "inlet plus casing" sound data for the air-handling unit. In this configuration, the return air typically enters the equipment room from the ceiling plenum through one or more holes in the equipment room wall. A portion of the sound inside the equipment room is transmitted through these holes into the ceiling plenum, and is then transmitted through the acoustical ceiling tile into the occupied space. As such, this path has very little attenuation.

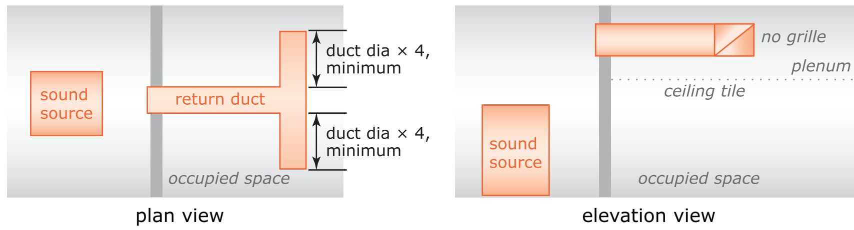
  If necessary, an unducted return path can be attenuated by adding one of the following elements:

  1. A silencer can be installed in the return-air opening(s) to the equipment room. Silencers can be effective, but they add static pressure loss, which makes the fan work harder and generate more sound. Also, their performance (both pressure drop and attenuation) is affected by inlet and outlet conditions.

  2. A section of return ductwork with 2-in. (5-cm) thick, exposed duct lining can provide effective attenuation. The length of lined duct depends on how much attenuation is needed.

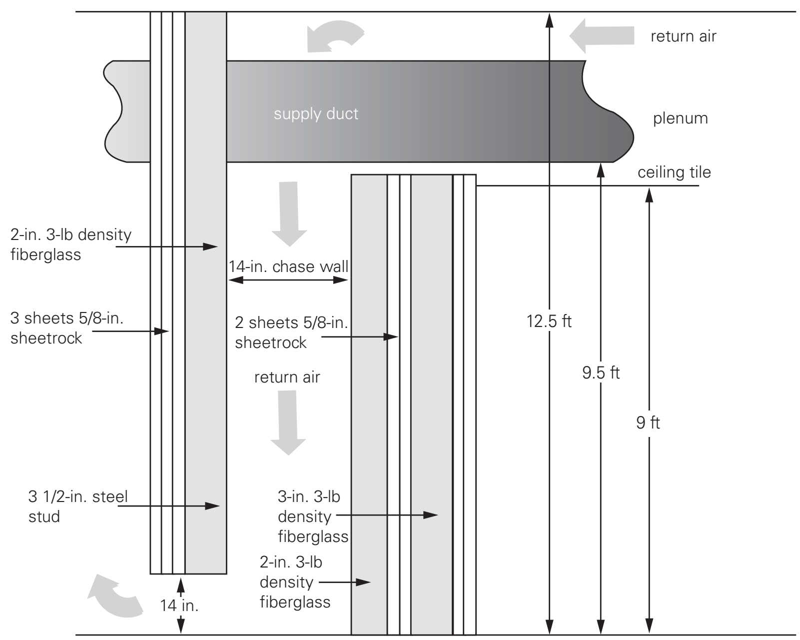
     When possible, use either a T- or H-shaped section of duct to take advantage of duct end reflection loss (Figure 102). The return duct should terminate in the ceiling plenum without an obstruction (diffuser, elbow, or grille) on the end, and should include at least four equivalent duct diameters of unobstructed straight duct prior to the opening. Duct end reflection loss also increases as the duct size decreases, so splitting the return duct into multiple, smaller ducts increases this attenuation. Branching into smaller ducts also provides attenuation at the T-junction and splits the one large sound path into several smaller paths that can be moved far apart from each other before the return ducts terminate above the ceiling. When sizing the return ducts, remember that elbows and junctions generate sound if air velocity is too high or if the inside corners are not sufficiently rounded.

Figure 102. T-shaped section to attenuate return airborne sound



3 An “air chase wall” routes the return air from the ceiling plenum down a chase that is constructed along the wall of the mechanical equipment room (Figure 103). The chase should be large enough so that the velocity of the return air does not exceed 1000 fpm (5.1 m/s). The extra turns and absorptive material inside the chase wall help to attenuate the sound that travels along the return-air path.

Figure 103. Example air chase wall (section view)



Systems with a fully ducted return-air path have several acoustical advantages (see “Open ceiling plenum versus fully ducted return,” p. 77). First, the casing radiated sound does not enter the return airborne path, so less sound is transmitted with the return air. Second, the additional ductwork provides attenuation, especially if duct liner is used. Sound can also break out though the walls of the return ductwork, so be sure to check the impact of the return duct breakout path for spaces nearest the equipment room. If necessary, the return duct breakout path can be attenuated by routing the duct over a less sound-sensitive area, using multiple smaller return ducts that enter on different sides of the equipment room, or switching to round ductwork (which has a much higher transmission loss than rectangular ductwork).

- *Supply airborne sound*
  The acoustical analysis for the supply airborne sound path begins with the "discharge" sound data for the air-handling unit. Since the supply airflow is typically ducted, the sound paths are duct breakout for spaces nearest the equipment room and airborne sound that follows the airflow path to the nearby spaces.

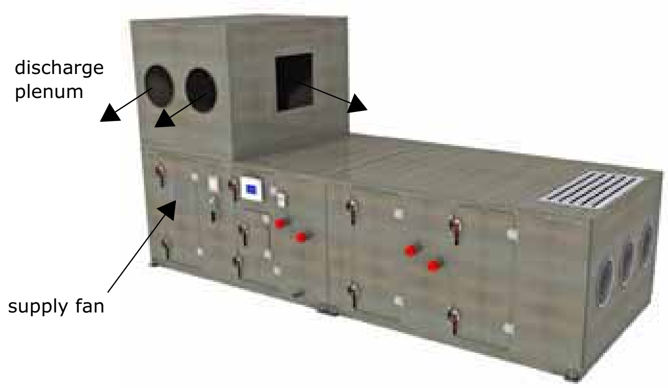
  If necessary, the supply airborne path can be attenuated by changing the configuration of the air-handling unit to lower the discharge sound levels. Potential strategies include:

  1. Selecting a fan with a different blade shape (such as airfoil, or AF), or a fan wheel with a larger diameter, may reduce discharge sound levels. See "Fan types," p. 32.
  2. Using a plenum fan can result in lower discharge sound levels than a housed fan because air velocity dissipates more quickly as the air pressurizes the plenum surrounding the fan and because the plenum provides an opportunity for some of the sound to be absorbed before the air discharges from the air-handling unit. Figure 24 (p. 33) includes an example comparison of discharge sound power for plenum fans versus housed fans.
  3. Adding a discharge plenum to a housed fan allows some of the sound to be absorbed before the air discharges from the air-handling unit and it allows for the multiple duct connections, which splits the sound before it leaves the unit (Figure 104). If a plenum fan is used, however, multiple duct connections can be made to the fan module itself, eliminating the need for a discharge plenum. Figure 26 (p. 34) includes an example of the impact of multiple plenum connections on discharge sound power.
  4. Using a perforated interior casing surface in the discharge plenum (or in the fan section for a plenum fan) allows some of the sound to be absorbed before the air discharges from the air-handling unit.
  5. Locating the supply fan upstream of the cooling coil (blow-thru configuration) often lowers the discharge sound levels slightly, although it may increase inlet sound levels (see Table 7, p. 37).

  If further attenuation is necessary, consider the following changes to the supply airborne path:

  1. Locate main ducts (and, when possible, branch ducts and VAV terminal units) above corridors and other less sound-sensitive areas (such as restrooms or copy rooms).
  2. Add 2-in. (5-cm) thick acoustical lining to the duct. The length of lined duct depends on how much attenuation is needed.
  3. Split the supply duct into multiple, smaller ducts that exit from different sides of the equipment room. This is most easily accommodated by adding a discharge plenum to the air-handling unit that extends to the height of the ceiling plenum (Figure 104). This allows for straight duct takeoffs in multiple directions, minimizing pressure losses and the associated impact on fan energy and noise.

Figure 104. Discharge plenum with multiple straight connections



4 Switch to round, instead of rectangular, ductwork. Round duct has significantly higher transmission loss than rectangular duct, so it will reduce the sound that breaks out of the ductwork. However, with less breakout, this means that the sound continues to travel down the duct (airborne).

5 Add a silencer to the supply airborne path to absorb sound energy. Realize, however, that the added airside pressure drop of the silencer makes the fan work harder, and generate more noise. Therefore, adding a silencer should be considered only after other air-handling unit and ductwork configurations have been explored.

VAV terminal units

VAV terminal units can impact the acoustics of the occupied space in two ways: 1) by adding sound to the supply airborne path, and 2) by radiating sound from the terminal unit itself. Sound data for VAV terminal units should be measured in accordance with ARI Standard 880, *Air Terminals*. Using terminal units with certified sound data guarantees that they will perform as described in the manufacturer's literature.

- *Install VAV terminal units over less sound-sensitive areas.*
  Good engineering practice is to install VAV terminals above corridors and other less sound-sensitive areas (such as restrooms or copy rooms). This helps reduce sound levels in the occupied space and minimizes disruptions during filter changes or periodic maintenance.

  Installing VAV terminal units directly over sound-sensitive spaces can be problematic. The ceiling tile is the only barrier between the sound source and the receiver. Sufficiently increasing the mass of the ceiling to attenuate the sound can be expensive and may reduce service access to the equipment.

  When a VAV terminal is located directly over an occupied space, the length of supply duct downstream of the terminal unit is also typically very short. This short section of duct provides minimal opportunity to attenuate the supply airborne sound before it passes through the diffuser (which also adds sound) and enters the space.

- *Use lined flexible ductwork to connect VAV terminal units to supply-air diffusers.*
  Lined flexible ductwork is very effective at attenuating high-frequency noise. However, it also causes turbulent airflow and relatively large static pressure drops. It is best to limit the use of flexible duct to no longer than 6 ft (2 m). If the overall length of duct between the VAV terminal unit and diffuser is greater than this, sheet metal should be used for the initial section 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.

System Design Variations

This chapter explores several variations to the typical chilled-water VAV system design.

Cold-Air VAV Systems

For more information on the benefits, challenges, and proper application of cold-air VAV systems, refer to the Trane *Engineers Newsletter* titled "Cold Air Makes Good $ense" (ENEWS-29/2) and the *Cold Air Distribution System Design Guide* published by ASHRAE (www.ashrae.org).

Many choices in the design of an HVAC system are "predetermined" by experience. System design engineers repeatedly choose to supply 55°F (13°C) supply air because they know it has worked in the past. The supply airflows that result from this choice directly impact the size (and cost) of fans, air-handling units, VAV terminal units, diffusers and ductwork. The size of fan motors is also affected, which extends the cost impact to the electrical distribution system.

Cold-air VAV systems typically deliver supply air at a temperature of 45°F to 52°F (7°C to 11°C). The appeal of cold-air distribution lies in the reduction in the airflow required to offset the sensible cooling loads in the zones. As the example in Table 27 suggests, lowering the supply-air temperature from 55°F (13°C) to 48°F (9°C) can reduce the supply-air volume by 29 percent.

Table 27. Conventional versus cold-air VAV systems

	conventional VAV system	cold-air VAV system
Supply-air temperature	55°F (13°C)	48°F (9°C)
Zone setpoint	75°F (24°C)	76°F (24.5°C)
ΔT ($T_{zone} - T_{SA}$)	20°F (11°C)	28°F (15.5°C)
Supply airflow per ton (kW) of zone sensible cooling load	553 cfm/ton (0.074 $m^3/s/kW$)	395 cfm/ton (0.053 $m^3/s/kW$)

Benefits of cold-air distribution

For further energy savings, consider keeping the same size ductwork and air-handling units (not downsizing for installed cost savings). This also improves the ability of the system to respond to possible future increases in load, since the system will be capable of handling an increased airflow rate if needed.

Reducing supply airflow can trigger a series of related benefits:

- Smaller supply fan (and return or relief fan, if equipped)
- Smaller air-handling units, which can increase usable (or rentable) floor space (see example below)
- Smaller vertical air shafts, which can increase usable (or rentable) floor space
- Smaller VAV terminals, which ease tight installations, are less expensive, and may be quieter
- Smaller ductwork, which requires less sheet metal, simplifies installation, and leaves more space above the ceiling for other services
- Shorter floor-to-floor height (attributable to smaller ductwork) may reduce the cost of glass and steel in a multi-story building

- Smaller supply (and return or relief, if equipped) fans reduce the cost of the electrical distribution system and lowers operating costs (it may also reduce fan-generated noise)
- Potential for lower space humidity levels due to the delivery of colder (and, therefore, drier) air

The lower relative humidity in a cold-air system often allows the zone dry-bulb temperature to be slightly warmer than in a conventional system, while still achieving an equivalent sensation of comfort.

Table 28 demonstrates the benefit of smaller air-handling units. The left-hand column in the table depicts a conventional, 55°F (13°C) supply-air system, which has a design airflow of 13,000 cfm (6.1 m^3/s). The VAV air-handling unit is a size 30, resulting in a coil face velocity of 435 fpm (2.2 m/s).

Note: The unit "size" typically represents the nominal face area of the cooling coil, in terms of ft^2. In this example, the face area of the size 30 air-handling unit is 29.90 ft^2 (2.78 m^2).

The middle column in the table depicts the same system, but designed for a 48°F (9°C) supply-air temperature and a 1°F (0.5°C) warmer zone temperature setpoint. This reduces the design airflow to 9300 cfm (4.4 m^3/s). The air-handling unit can be downsized two sizes to a size 21 and still maintain a similar coil face velocity. This reduces the footprint of the unit by 25 percent (Figure 105) and reduces the installed weight by 22 percent. This decreases the cost of the equipment, and requires less floor space and less structural support.

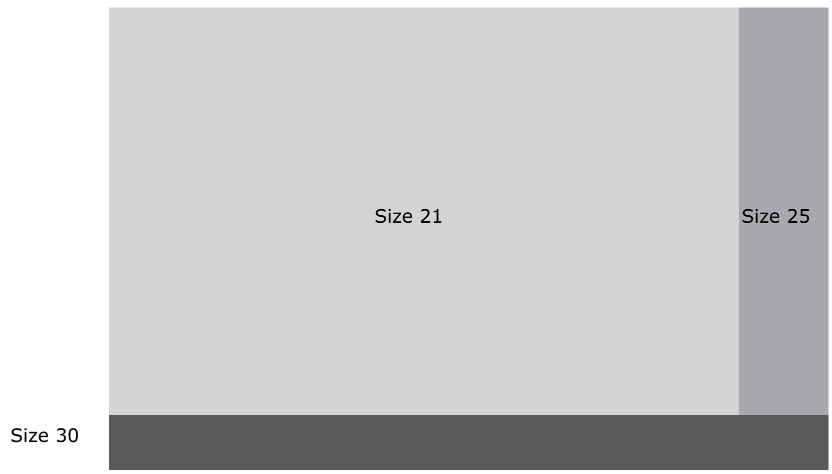
Table 28. Impact of cold-air distribution on AHU footprint and weight

	Size 30 AHU	Size 21 AHU	Size 25 AHU
Supply-air dry-bulb temperature, °F (°C)	55 (13)	48 (9)	48 (9)
Zone temperature setpoint, °F (°C)	75 (24)	76 (24.5)	76 (24.5)
Design airflow, cfm (m^3/s)	13000 (6.1)	9300 (4.4)	9300 (4.4)
Coil face area, ft^2 (m^2)	29.90 (2.78)	20.81 (1.93)	24.97 (2.32)
Face velocity, fpm (m/s)	435 (2.2)	447 (2.3)	372 (1.9)
Fan input power, bhp (kW)	12.8 (9.5)	12.2 (9.1)	8.7 (6.5)
AHU footprint[1], ft (m)	12.1 x 7.8 (3.7 x 2.4)	10.6 x 6.7 (3.2 x 2.0)	12.1 x 6.7 (3.7 x 2.0)
AHU height[1], ft (m)	9.1 (2.8)	8.4 (2.5)	9.1 (2.8)
AHU weight (installed), lbs (kg)[1]	3570 (1620)	2770 (1260)	3110 (1410)

1 Based on a typical VAV air-handling unit layout consisting of an OA/RA mixing box, high-efficiency filters, hot-water heating coil, chilled-water cooling coil, airfoil centrifugal supply fan, and a top-mounted discharge plenum

However, consider the impact of only reducing the air-handling unit by one size. The right-hand column in Table 28 depicts the same cold-air system design, but the air-handling unit is only downsized one size to a size 25. This still results in a significant reduction in footprint (14 percent, see Figure 105) and weight (13 percent), but the larger coil face area results in a lower airside pressure drop and even greater fan energy savings—a 32 percent reduction in fan input power compared to the conventional, 55°F (13°C) supply-air system. This decision often results in the best balance of installed cost savings and energy savings.

Figure 105. Impact of cold-air distribution on AHU footprint (plan view)



Challenges of cold-air distribution

Concerns that design engineers have about cold-air distribution typically focus on the following three issues:

- Effects of delivering cold air into the zone on occupant comfort
- Impact on overall system energy consumption
- Avoiding condensation on components of the air distribution system

Effects of delivering cold air into the zone on occupant comfort

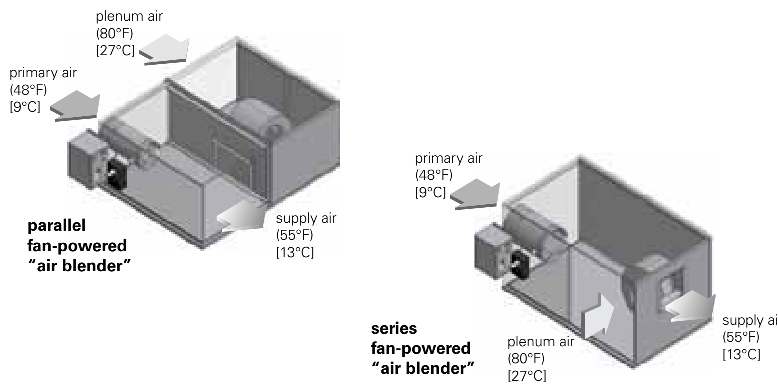
Design engineers typically use one of two approaches to avoid "dumping" cold air into the occupied space: 1) supply-air diffusers with a high aspiration ratio, or 2) fan-powered VAV terminal units used as "air blenders."

- *High-aspiration diffusers*
  A diffuser with a high aspiration ratio induces air from within the occupied space toward the supply-air diffuser. It enhances comfort in cold-air VAV systems by increasing both air motion and diffuser throw. A common example of a diffuser with a high aspiration ratio is a linear slot diffuser (see "Supply-air diffusers," p. 74). It recirculates approximately 1 cfm (m^3/s) of air from within the zone for each 1 cfm (m^3/s) of supply air that it delivers through the diffuser.

  Note: Non-aspirating diffusers, such as perforated plates or concentric grilles, may not perform as well in cold-air applications because cold, dense supply air tends to drop quickly to the floor ("dumping" on the occupants). If this type of diffuser is used, consider using either series or *parallel fan-powered VAV terminals to blend locally recirculated air with the cold primary air, before delivering it to the zone.*

- *Fan-powered VAV terminals as "air blenders"*
  Operating the terminal fan continuously during occupied hours blends warm air from the ceiling plenum with the cold primary air before it is delivered to the zone. Either a series or a parallel fan-powered VAV terminal can be used as an "air blender" in this manner (Figure 106).

Figure 106. Fan-powered VAV terminals used as "air blenders" in cold-air VAV systems



Typically, the fan in a series fan-powered VAV terminal operates continuously during occupied hours. When used as an "air blender," the terminal fan is sized to mix cold primary air—48°F (9°C) in this example—with warm air from the ceiling plenum—80°F (27°C) in this example—to deliver 55°F (13°C) air to the zone at design cooling load (when the primary air damper is wide open). As the cooling load in the zone decreases, and the primary air damper begins to close, the series terminal fan continues to deliver a constant quantity of air to the zone. However, the resulting temperature of the supply air increases at part load.

Typically, the fan in a parallel fan-powered VAV terminal operates only when heat is needed. When used as an "air blender," however, the terminal fan is controlled to operate continuously during occupied hours. It is sized to mix cold primary air with warm plenum air and, in this example, deliver 55°F (13°C) air to the zone at design cooling load (when the primary air damper is wide open). As the cooling load in the zone decreases, and the primary air damper begins to close, the parallel terminal fan continues to operate, but the quantity of air delivered to the zone decreases. While the supply airflow decreases at part load, the supply-air temperature remains the same or increases slightly.

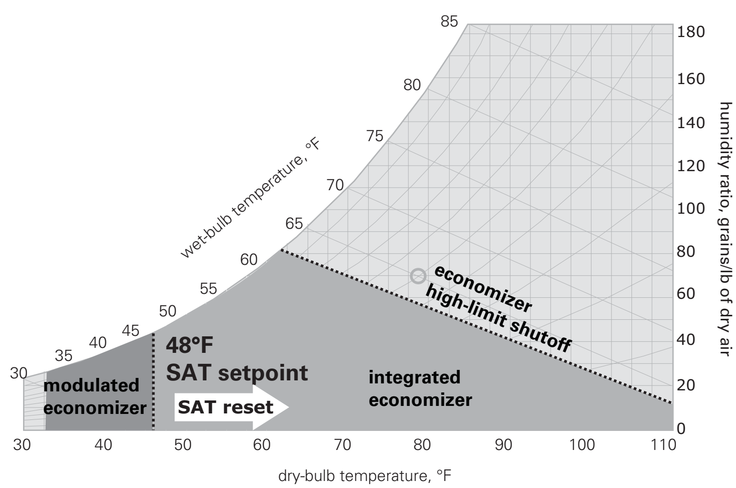
For "air blender" applications, series fan-powered VAV terminals are often preferred for large conference rooms or other zones where constant airflow is desirable. Parallel fan-powered VAV terminals (with constant fan operation during occupied periods) are well suited for zones where less air motion during off-peak conditions is acceptable. The advantage of using a parallel fan-powered terminal as an air blender is that the terminal fan is smaller than in a series fan-powered terminal. Therefore, the VAV terminals are less expensive to install and operate.

Impact on overall system energy consumption

Cold-air VAV systems require less "transport" energy (energy used to move the air) than conventional designs since primary airflow is reduced. But not every component of the system contributes to the energy savings:

- The colder temperature leaving the cooling coil may require the water chiller to operate at a colder refrigerant suction temperature to produce a colder chilled-water supply temperature. This increases the energy required to operate the chiller.
- In non-arid climates, delivering the air colder also results in delivering the air drier (at a lower dew point), which reduces relative humidity in the building and increases the amount of latent cooling by the coil. Offsetting this increase in latent coil load, however, is the substantial and continuous reduction of sensible heat generated by the supply and return fans.
- The economizer in a cold-air VAV system is not able to shut off the water chiller as soon as it can in a conventional VAV system. For the example in Figure 107, when the outdoor temperature is between 48°F and 55°F (9°C and 13°C), the conventional system will be in the "modulated" economizer mode with the chiller off (see "Airside economizer control," p. 174). The cold-air system, however, will be in the "integrated" economizer mode: outdoor-air damper is wide open, but the chiller is still required to operate to deliver the air at 48°F (9°C).

Figure 107. Impact of supply-air temperature on airside economizer operation



To minimize this loss of modulated economizer hours in a cold-air VAV system, consider using supply-air-temperature (SAT) reset (see "Supply-air-temperature reset," p. 202). At part load, when the supply-air temperature is reset upwards, the line separating these two modes will slide to the right on the psychrometric chart (Figure 107). This strategy can help recover some of the lost modulated economizer hours, but likely increases fan energy.

- While a colder supply-air temperature reduces the quantity of primary air sent to each zone, the outdoor-air requirement for the zone remains unchanged. The result is that the minimum airflow settings at the VAV terminal units may need to be a higher percentage of design airflow in order to ensure each zone delivers the required outdoor airflow under low-load conditions. This may cause the reheat coils in the VAV terminal units to activate sooner in a cold-air system than in a traditional VAV system.

  Using supply-air-temperature reset at part load, and optimizing the control of ventilation (see "Ventilation optimization," p. 205), can help reduce the operating-cost impact of reheat.

- When fan-powered VAV terminals are used as "air blenders," the power needed for continuous operation of the terminal fans during occupied hours adds up. Consider using ECMs on terminal fans to minimize energy use (see "Electronically commutated motors on fan-powered VAV terminal units," p. 60). Also, parallel terminals with constant fan operation during occupied hours consume less energy than series terminals.

The varying impact on overall system energy use reflects the complex relationship between building usage, climate, and the design and control of the HVAC system. Increased reheat energy and fewer hours of airside economizer operation partially offset the fan energy savings. Intelligent system control is crucial to fully realize the potential energy savings of cold-air VAV systems.

In addition, it can be a challenge to achieve both energy savings and first cost savings with a cold-air VAV system. Downsizing the ductwork saves first cost, but results in less fan energy savings than keeping the same-sized ductwork. Building analysis software, like Trane's TRACE™ 700, can be used to determine the most desirable balance between energy savings and first cost for any particular project.

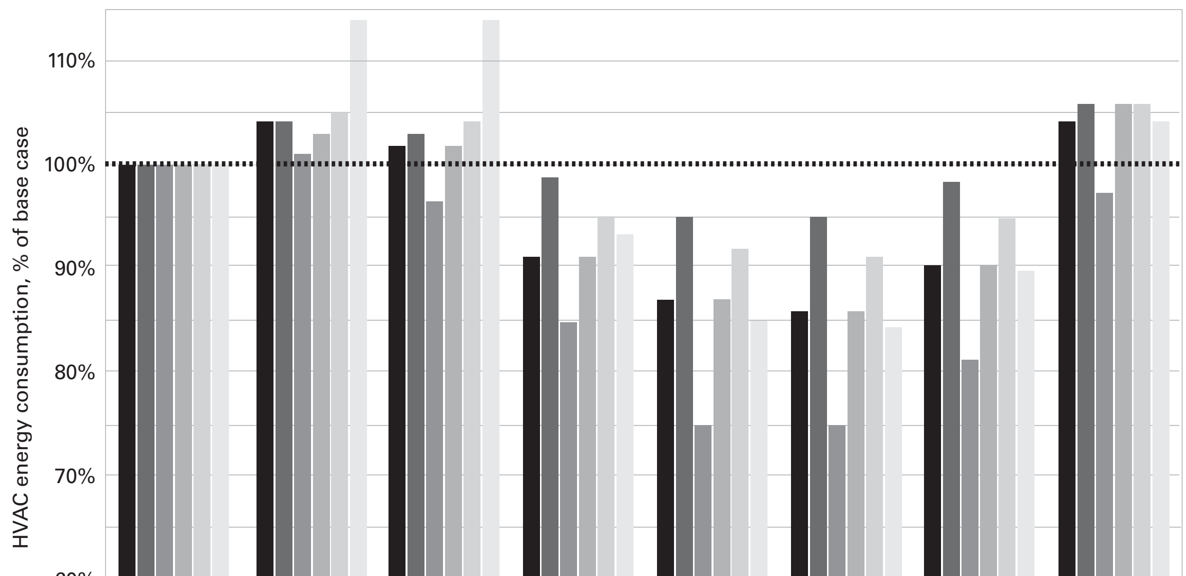
For further energy savings, consider keeping the same size ductwork and air-handling units (not downsizing for installed cost savings). This also improves the ability of the system to respond to possible future increases in load, since the system will be capable of handling an increased airflow rate if needed.

Figure 108 includes the results of a TRACE analysis for an office building in six different climates. The building uses a chilled-water VAV system.

- Run 2 versus Run 1: Simply lowering the supply-air temperature to 48°F (9°C), without changing zone setpoints or using supply-air temperature reset, may use more energy than the conventional 55°F (13°C) design.
- Runs 3 and 4 versus Run 1: Raising the zone cooling setpoint by 1°F (0.5°C) and resetting the supply-air temperature up to 55°F (13°C) at part load resulted in lower overall HVAC energy use.
- Run 5 versus Run 4: Resetting the supply-air temperature even further at part load (up to 60°F [15.6°C], in this example) was beneficial in all climates, but it had the greatest impact in the economizing-dominated climates (Los Angeles and Portland, Ore.).

- Run 7 versus Run 8: When fan-powered VAV terminals were used as "air blenders," parallel fan-powered terminals (with the terminal fan operating continuously during occupied hours) used less energy than series fan-powered terminals. In this example, both the parallel and series fan-powered terminals were equipped with ECMs.

Figure 108. TRACE analysis of a cold-air, chilled-water VAV system



	Run 1 base case (55°F SA)	Run 2 cold air (48°F SA)	Run 3 raise zone setpoint 1°F	Run 4 SAT reset to 55°F	Run 5 SAT reset to 60°F	Run 6 parallel FPVAV (cycled)	Run 7 parallel FPVAV w/ECM (constant)	Run 8 series FPVAV w/ECM
Atlanta	100%	104%	102%	91%	87%	86%	90%	104%
Minneapolis	100%	104%	103%	97%	95%	95%	98%	106%
Los Angeles	100%	101%	96%	85%	75%	75%	81%	97%
Denver	100%	103%	102%	91%	87%	86%	90%	106%
Philadelphia	100%	105%	104%	95%	92%	91%	95%	106%
Portland, Ore.	100%	114%	114%	93%	85%	84%	89%	104%

Avoiding condensation on components of the air-distribution system

For more information on avoiding condensation, including various design strategies for indoor equipment rooms, refer to the Trane application manual titled *Managing Building Moisture* (SYS-AM-15).

Condensation can occur on the outside surface of air-handling units, ductwork, or supply-air diffusers if the surface temperature is at or below the dew point temperature of the air that comes in contact with it. Two diffuser surfaces may be prone to condensation: the surface exposed to the occupied zone and the surface exposed to the ceiling plenum.

The following practices help minimize the risk of condensation on the components of the air-distribution system:

- *Slowly ramp down supply-air temperature during morning cool-down.* In humid climates, indoor humidity levels may increase if the system is shut off for a significant period of time (overnight or over the weekend). When the system starts up again, the elevated dew point temperature in the zone may result in moisture condensing on the cold, zone-side surface of the supply-air diffusers.

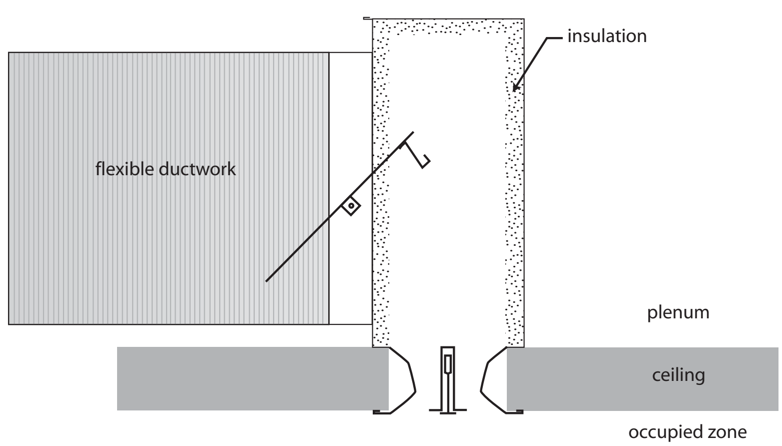
To avoid condensation problems during start-up, the supply-air dry-bulb temperature should be ramped down slowly to lower surface temperatures and to dehumidify the building and lower the indoor dew point temperature [see “Morning warm-up (or cool-down) mode,” p. 194]. The temperature of the zone-side surface will be somewhere between the supply-air dry-bulb temperature and the zone dry-bulb temperature. The ASHRAE *Cold Air Distribution System Design Guide* suggests that, for metal diffusers, the zone-side surface temperature is generally 3°F (1.7°C) warmer than the supply-air dry-bulb temperature.

During morning cool-down, if the supply-air temperature is controlled so that it is no more than 3°F (1.7°C) below the dew point of the zone, the risk of condensation on the zone-side surface of the diffuser should be minimal. Use of a draw-thru supply fan (which raises the dry-bulb temperature of the supply air a few degrees above the dew point of the supply air) and heat gain through the supply ductwork allow for a wider margin and quicker pull-down of indoor humidity, and provide a greater safety factor to avoid condensation.

In addition, monitoring the indoor dew point during unoccupied periods, and turning on the system as necessary to prevent indoor humidity from rising too high, also helps minimize the risk of condensation problems during morning cool-down. (See “After-hours dehumidification,” p. 125.)

- *Properly insulate supply ductwork and supply-air diffusers.*
  Insulate the supply ductwork and supply-air diffusers to prevent condensation on the surfaces exposed to the ceiling plenum (Figure 109). However, insulation alone cannot prevent condensation. A properly sealed vapor retarder must be included on the warm side of the insulation to prevent condensation within the insulation itself. If the ductwork or diffusers use internal insulation, the sheet metal typically acts as the vapor retarder. If external insulation is used, it must be covered with a well-sealed vapor retarder.

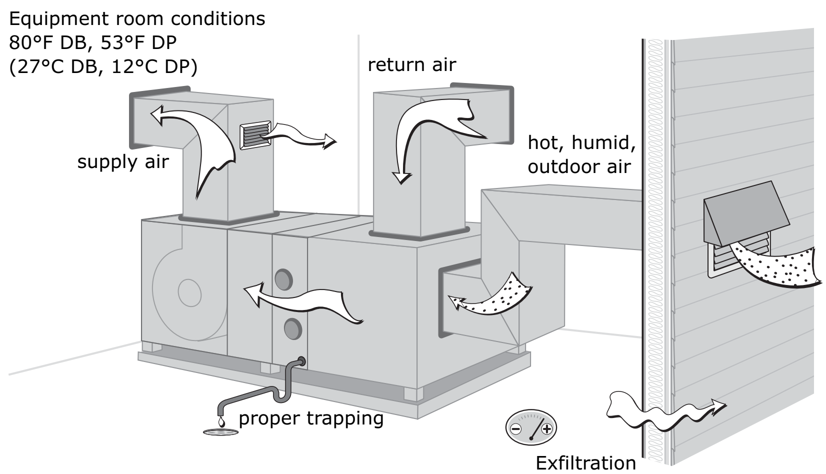
Figure 109. Insulation on supply-air diffusers



- *Avoid high dew point temperature in the mechanical room.*
  Condensation can occur on the outer surface of air-handling units if the surface temperature is below the dew point temperature of the air in the

mechanical room. The dew point temperature can be reduced by ducting the outdoor air directly to the intake of the air-handling unit and then pressurizing the mechanical room with a small amount of dehumidified supply air (Figure 110). The dry supply air limits infiltration into the equipment room and maintains a low dew point by filling the room with low-dew-point air.

Figure 110. Pressurize the mechanical room with dry supply air



Alternatively, a dedicated outdoor-air unit can be used to separately dehumidify all of the outdoor air to a low dew point, and then duct the conditioned outdoor air (CA) directly to floor-by-floor mechanical rooms (Figure 96, p. 124). The dry outdoor air maintains a low dew point in the mechanical room.

Best practices when using cold-air distribution

Proper engineering and construction practices are critical ingredients in any successful HVAC system. Consider the following when designing a cold-air VAV system:

Building construction

- Use vapor retarder on the *warm* side of the insulation in perimeter walls to minimize vapor-pressure diffusion.
- Seal all perimeter wall penetrations (e.g., electrical and plumbing services) to minimize infiltration.

VAV air-handling unit

- Seal all penetrations, including connections for coil piping, electrical service, and controls.
- Gasket access panels, door openings, and inspection windows, paying special attention to positive-pressure sections.
- Insulate and vapor-seal condensate drain pipes if they run though the building.

- Avoid high dew point temperatures in the mechanical room by pressurizing the mechanical room with dehumidified supply air (Figure 110) or using a dedicated outdoor-air unit to supply dehumidified outdoor air to floor-by-floor mechanical rooms (Figure 96, p. 124). Consult the manufacturer to determine the allowable dew point in the mechanical room, based on the thermal performance of the air-handling unit casing. [See "Casing performance (leakage and thermal)," p. 51.]
- Because intake airflow is a larger percentage of supply airflow, freeze protection for the coils is more important (see "Freeze prevention," p. 18). SAT reset can also help reduce the risk of freezing coils because it increases supply airflow, which lowers the percentage of outdoor air.

Supply ductwork, VAV terminal units, and supply-air diffusers

- Insulate and vapor-seal all supply ductwork and plenum-side surfaces of supply-air diffusers, and follow the manufacturer's recommendations for insulating and sealing VAV terminal units.
- Select linear slot diffusers with a high aspiration ratio to provide sufficient air movement.
- Use parallel fan-powered VAV terminals (with the terminal fan cycling on only as the first stage of heat) in perimeter zones and in interior zones that experience wide variations in cooling load to recover heat from the ceiling plenum and minimize reheat energy.
- If using conventional diffusers, use a fan-powered VAV terminal as an "air blender" to blend warm air from the ceiling plenum with the cold primary air before delivering it to the zone (Figure 106). Use parallel, fan-powered terminals (with the terminal fan operating continuously during occupied periods) whenever possible, to minimize the installed cost and energy consumption of these terminal fans.
- Use an open ceiling plenum return (rather than a fully ducted return), if possible. This results in a "conditioned" ceiling plenum, which can reduce the risk of condensation on components of the air distribution system.
- When specifying VAV terminal units, request the manufacturer provide actual thermal performance of the cabinet to determine the risk of condensation.

HVAC system controls

- Consider raising the zone cooling setpoint 1°F or 2°F (0.5°C to 1.1°C) to further reduce the supply airflow in cold-air systems. The lower relative humidity in a cold-air system often allows the zone dry-bulb temperature to be slightly warmer than in a conventional system, while still achieving an equivalent sensation of comfort.
- Maintain positive building pressure during the cooling season to minimize infiltration of humid outdoor air.
- Use supply-air-temperature reset (see "Supply-air-temperature reset," p. 202) to minimize the use of reheat and maximize the benefit of the airside economizer.

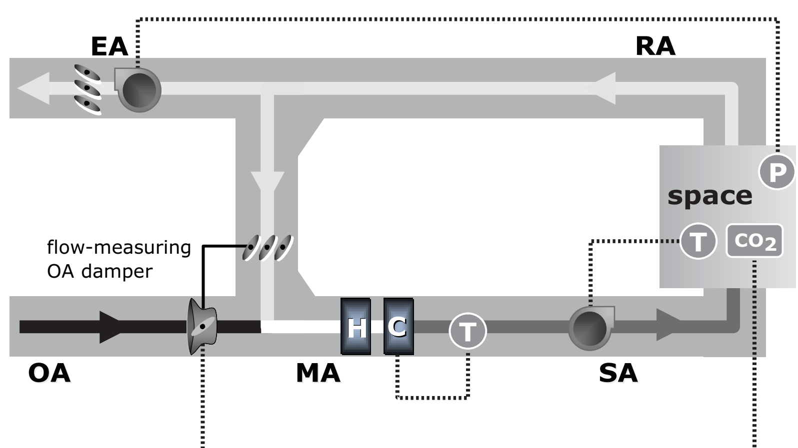
- Implement fan-pressure optimization (see "Fan-pressure optimization," p. 200) to minimize supply fan energy and improve zone control.
- Consider using a differential enthalpy economizer (see "Airside economizer control," p. 174). In a cold-air VAV zone, the relative humidity (and, therefore, the enthalpy) of the return air is typically lower than a conventional system. A differential enthalpy economizer will maximize economizer savings, and avoid bringing in outdoor air that has higher enthalpy than the return air.
- During startup (after the system has been shut off), slowly ramp down the supply-air temperature to pull down the humidity in the building and avoid condensation problems. [See "Morning warm-up (or cool-down) mode," p. 194.]
- Monitor indoor humidity during unoccupied periods, and turn on the system as necessary to prevent indoor humidity from rising too high and causing condensation problems during morning cool-down.

Single-Zone VAV

Recall that in a conventional, *multiple-zone* VAV system, a damper in the VAV terminal unit modulates to maintain the zone temperature by varying the volume of supply air delivered to that zone. The central supply fan modulates to maintain the static pressure setpoint in the supply ductwork.

A *single-zone* VAV system, on the other hand, serves only one zone, so no VAV terminal units are required. Instead, a temperature sensor in the zone is used to directly vary the volume of supply air delivered by the fan (Figure 111). Just like in a conventional VAV system, cooling capacity is modulated to maintain the discharge-air temperature at a setpoint.

Figure 111. Single-zone VAV system



Single-zone VAV is an excellent system to use in large, densely occupied zones that have variable cooling loads. Common examples include gymnasiums, lecture halls, arenas, auditoriums, and places of worship.

Note: Effective January 1, 2010, ASHRAE Standard 90.1 (addendum n to ASHRAE 90.1-2007) will require single-zone VAV control for any air-handling unit that contains a chilled-water cooling coil and a supply fan that has a motor greater than or equal to 5 hp (4 kW).

Compared to a constant-volume system that is often used for large zones such as these, a single-zone VAV system results in fan energy savings, and less fan-generated noise, at part-load conditions. It also significantly improves dehumidification, because unlike a constant-volume system, a VAV system continues to deliver cool, dry air at part-load conditions (see "Full-load versus part-load dehumidification performance," p. 118). However, because supply airflow varies, the air distribution system must be designed carefully to prevent "dumping" under low-load conditions.

Best practices in a single-zone VAV application

Consider the following when designing a single-zone VAV system:

Uniformity of loads throughout the zone

Like any other single-zone system, a single-zone VAV system does not have the capability to satisfy simultaneous heating and cooling requirements. When used to provide comfort for a large zone, it may result in undesirable temperature variations in the areas of the zone that are further away from the zone sensor. Therefore, the loads should be fairly uniform throughout all areas of the zone.

Single-zone VAV systems should be used to condition relatively large, open areas, rather than numerous, small zones, so that any changes in air distribution that occur as the supply fan modulates will have minimal impact on occupant comfort.

As a general rule, a single-zone VAV system can be successfully applied in any zone where a single-zone, constant-volume system may traditionally have been applied.

Design the air distribution system for variable supply airflow

In a single-zone VAV system, it is important to ensure proper air distribution throughout the zone as supply airflow is modulated. In general, follow these guidelines:

- *Keep the supply duct system as short and as symmetrical as possible.*
  As the supply fan modulates, systems with long or asymmetric duct runs are more susceptible to unequal air distribution between diffusers that are located at various distances from the fan.
- *Size the ducts for low-to-medium air velocities.*
  Because this system serves a single zone, the supply fan is typically located near the zone. Using lower air velocities helps to minimize noise generated in the ductwork.
- *Use diffusers that will provide proper air distribution at low airflows.*
  For some applications, supply-air diffusers that are appropriate for

conventional VAV systems (such as linear slot diffusers, Figure 65, p. 74) are used. For spaces with tall ceilings, it may be desirable to distribute the air from the side walls and allow for temperature stratification within the zone.

Consider the need for heating

A single-zone VAV system has no VAV terminal units to be equipped with heating coils. Therefore, in most cases, the air-handling unit must be capable of providing heat because the zones that these systems typically serve (gymnasiums, arenas, auditoriums, etc.) often do not include baseboard radiant heat.

The modulation range of the supply fan is limited by how far the variable-speed drive can be turned down (typically 30 to 40 percent of design airflow). In some applications, the sensible cooling load in the zone may decrease to the point where it is less than the cooling capacity of the unit at the minimum airflow of the supply fan. Under this condition, the zone will be at risk of being overcooled.

To prevent overcooling at low cooling loads, consider one of the following strategies:

- *Reset the supply-air temperature.*
  Reset the supply-air temperature upward at low-load conditions, after the supply fan reaches its minimum airflow. This avoids overcooling the zone, but the cooling coil also removes less moisture, so the humidity level in the zone increases. (See "Supply-air-temperature reset," p. 202.)
- *Activate the source of heat in the air-handling unit.*
  In some applications, the supply fan is ramped up to 100 percent airflow when the air-handling unit operates in the heating mode. This prevents the unit from tripping due to the high temperature rise across electric heaters or conventional, non-modulating gas-fired burners. In many climates and for many space types that use single-zone VAV systems, the air-handling unit may only need to provide heat for morning warm–up and for a relatively small number of hours during the year.

  When a modulating gas-fired burner is used, it may allow for varying airflow during the heating mode. In this case, the supply fan can be modulated based on zone temperature and heating capacity can be modulated to maintain the discharge-air temperature at a setpoint.

Employ demand-controlled ventilation

For information on calculating the required ventilation airflow for a single-zone VAV system, see the sidebar on p. 104.

Because a single-zone VAV system is often used for densely occupied zones with a highly variable population, it is typically a good application for CO_2-based demand-controlled ventilation (DCV). (See "Ventilation control," p. 184.)

Carbon dioxide (CO_2) is produced continuously by the occupants and diluted by the outdoor air, so the difference between indoor and outdoor CO_2 levels can be used as an indicator of the per-person outdoor airflow rate (cfm/person [m^3/s/person]) being delivered to the zone. A sensor is used to monitor the concentration of CO_2 in the zone, and this concentration is then communicated to the air-handling unit controller and used to reset the intake airflow required for the zone (see Figure 111, p. 157).

For more information on implementing CO_2-based DCV in a single-zone system, refer to "Standard 62.1-2004 System Operation: Dynamic Reset Options," published in the December 2006 issue of the *ASHRAE Journal* (www.ashrae.org).

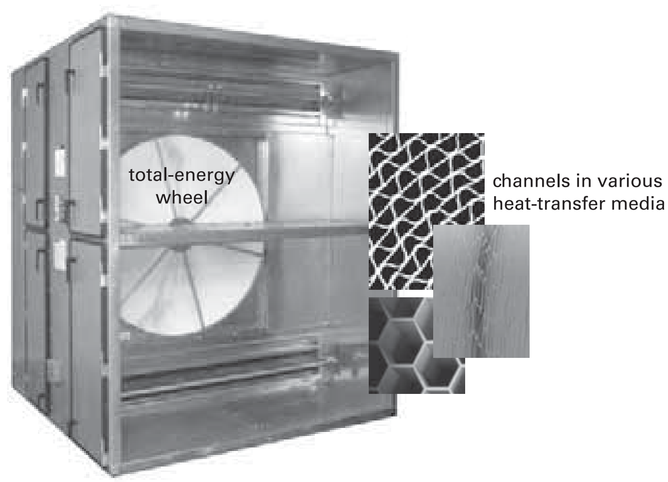
Because mixing box pressure varies as supply fan airflow changes (see Figure 86, p. 114), it is important to make some provision for maintaining the minimum required intake airflow. In a VAV air-handling unit, this typically involves the use of an airflow-measuring station or flow-measuring OA damper to directly measure intake airflow, and prevent it from dropping below the minimum ventilation requirement. And, because DCV reduces OA intake flow during periods of partial occupancy, some method of directly controlling building pressure should be included (see Figure 111). (See "Building pressure control," p. 178.)

Air-to-Air Energy Recovery

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

Air-to-air energy recovery refers to the transfer of sensible heat, or sensible heat and moisture (latent heat), between air streams.

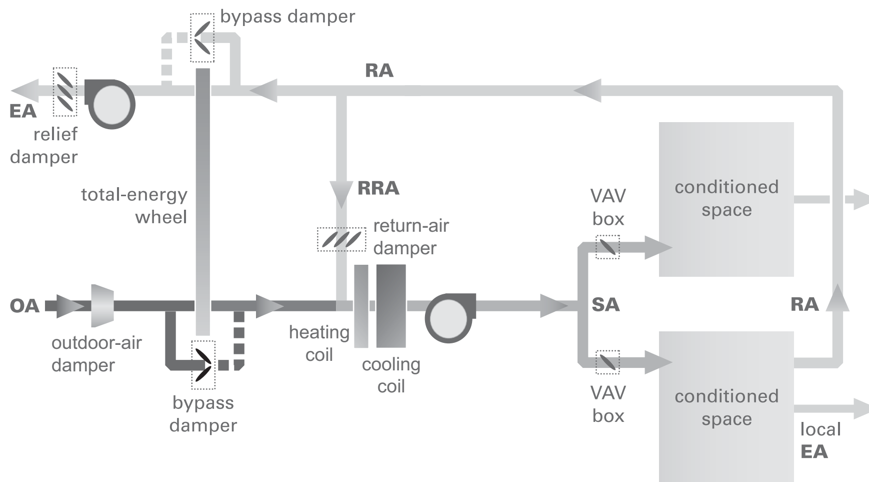
Figure 112. Total-energy wheel in a VAV air-handling unit



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 (sensible-energy 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, Figure 112) and fixed-membrane heat exchangers.

In a chilled-water VAV system, the most common application of an air-to-air energy recovery device is to precondition outdoor air as it enters the building for ventilation (Figure 113). An air-to-air energy-recovery device is arranged to precondition the outdoor air (OA) by exchanging energy with the exhaust air (EA) stream. While any sensible- or total-energy recovery technology can be used for this purpose, total-energy wheels are the most common.

Figure 113. Total-energy wheel used to precondition outdoor air



Benefits of outdoor-air preconditioning

Preconditioning the outdoor air with air-to-air energy recovery offers the following benefits:

- *Reduces cooling, dehumidification, heating, and humidification energy*
  During the cooling season, in a climate where it is hot and humid outside, a *total*-energy recovery device precools and “pre-dries” (dehumidifies) the outdoor air by transferring both sensible heat and water vapor to the exhaust air stream. During the heating season, when outdoor conditions are cold and dry, the same total-energy recovery device preheats and prehumidifies the outdoor air by removing both sensible heat and water vapor from the exhaust air stream and releasing it into the supply air stream.

  A *sensible*-energy recovery device does not transfer water vapor, so it precools the entering outdoor air during warm weather and preheats it during cold weather.

- *Allows downsizing of cooling, dehumidification, heating, and humidification equipment*
  At the design cooling condition, air-to-air energy recovery reduces the cooling load due to the outdoor air, allowing the cooling equipment to be downsized. At the design heating condition, it reduces the heating load due to the outdoor air, allowing the heating equipment to be downsized. Sometimes a source of heat in the VAV air-handling unit can be eliminated if morning warm–up can be accomplished at the VAV terminals.

  Also, if the system includes mechanical humidification, the ability of a total-energy recovery device to “pre-humidify” the entering outdoor air reduces the required capacity of the humidification equipment.

Drawbacks of outdoor-air preconditioning

However, there are some drawbacks that need to be justified:

- *Increases airside pressure drop, which increases fan energy and may require larger fan motors*
  Adding an air-to-air energy-recovery device increases the static pressure drop in both the outdoor- and exhaust-air paths. The magnitude of this pressure drop and the configuration of the fans determine how much additional energy is consumed. The operating cost savings provided by recovered energy must exceed the increased cost of operating the fans in order to justify the cost of the energy-recovery device.
- *May require additional exhaust ductwork*
  Routing most of the exhaust air back to the energy-recovery device, so that the path is adjacent to that of the entering outdoor air, may require more ductwork than a system without energy recovery.

  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 them well suited for retrofit situations. Also, a coil loop can be used to recover heat from multiple, separate exhaust air streams (using multiple exhaust-side coils).

Best practices for preconditioning outdoor air using air-to-air energy recovery

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

- *Properly size the energy-recovery device.*
  In a mixed-air VAV system, size the air-to-air energy-recovery device to precondition only the minimum outdoor airflow required for ventilation, not the maximum airflow expected during economizer operation. This minimizes the first cost of the device.

  Bypass dampers, or a separate OA path, should be used to allow for the increased outdoor airflow needed during airside economizing (see Figure 114).
- *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.

  In applications with a moderate amount of outdoor air for ventilation (20 to 30 percent, for example), when the system is bringing in minimum ventilation airflow (not in airside economizer mode), local exhaust fans (in restrooms and copy centers, for example) and exfiltration (due to positive building pressurization) may be sufficient to relieve all the air brought into the building for ventilation, so the central relief (exhaust) fan is turned off. In this case, there is no air passing through the exhaust-side of the energy-recovery device and, therefore, no energy is saved.

  Therefore, air-to-air energy recovery may only make economic sense for systems with higher amounts of outdoor air or for tightly constructed buildings (little exfiltration due to positive building pressurization) with a minimal amount of local exhaust.

In addition, 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 must be exhausted. Air exhausted by local exhaust fans and exfiltration due to building pressurization are relatively constant, so when DCV reduces intake airflow, less central exhaust air is available for energy recovery.

- *Sensible- or total-energy recovery?*
  In most climates, a total-energy recovery device improves the opportunity for downsizing 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 exceptions are in dry climates where it is often unnecessary to mechanically dehumidify the outdoor air. In this case, coil loops and fixed-plate heat exchangers typically provide 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 many sensible-energy recovery devices, so they save more heating energy and may permit further downsizing of the heating equipment.

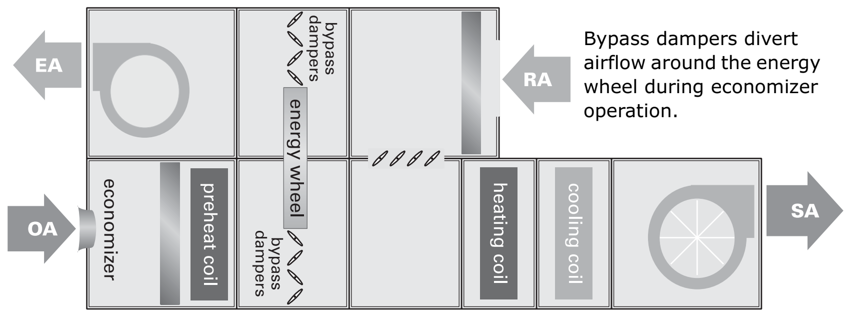
  – Water vapor transferred by a total-energy-recovery device humidifies the entering outdoor air during the heating season, which helps keep the space from becoming too dry. "Free" humidification also reduces the energy used by the mechanical humidification system (if installed) and permits this equipment to be downsized.

  – Most heating climates also include a cooling season. Applying a total-energy-recovery device enables a larger reduction in cooling capacity, which can reduce the first-cost premium for energy recovery.

- *Integrate control with airside economizer operation.*
  In many climates, an airside economizer can provide the benefits of "free" cooling for much of the year. While the economizer operates, air-to-air energy recovery offers no additional benefit. In fact, unless it is turned off, the energy-recovery device actually increases the cooling load by transferring heat to the outdoor air stream. Achieving maximum energy savings depends on proper economizer operation as well as on proper integration of the energy-recovery device into the control of the system.

To accommodate economizer operation when the energy-recovery device is idle, add bypass dampers (Figure 114) to allow full economizer airflow without significantly increasing the airside pressure drop and corresponding fan energy consumption. Alternatively, provide two separate paths for outdoor air: one for "minimum ventilation" airflow and the other for "economizer" airflow.

Figure 114. Bypass dampers allow for airside economizing and capacity control



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 titled *Air-to-Air Energy Recovery in HVAC Systems* (SYS-APM003-EN).

- *Provide a means to control the capacity of the device at part load.* During cool weather, most VAV systems will require a means to modulate the capacity of the energy-recovery device 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 heat exchangers often use a modulating damper to bypass some of the exhaust air. Heat pipes may use a tilt controller, 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 (Figure 114) or vary the rotational speed of the wheel. For wheels, exhaust-air bypass is recommended because it provides more linear control and a wider range of capacity control than a VFD, and in a VAV air-handling unit, the bypass dampers are likely already provided for economizer operation or to bypass the wheel when it shuts off.

- *Provide a method for frost prevention in cold climates.* Any air-to-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.

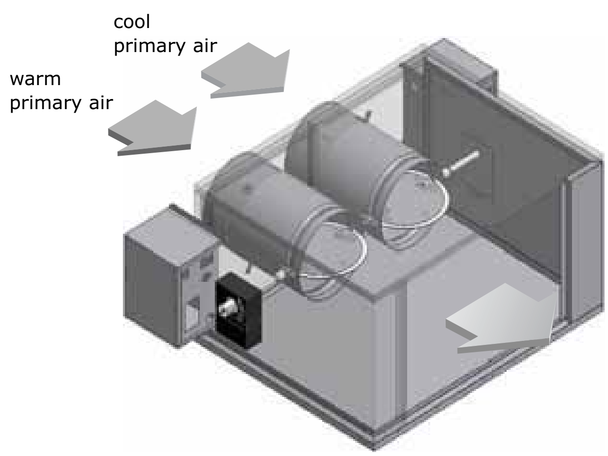
For most applications and climates, reducing capacity (by modulating an OA bypass damper, for example) is sufficient for frost prevention. And, if the VAV air-handling unit is equipped with an airside economizer, this bypass damper is probably already included (see Figure 114). However, for applications with extremely cold outdoor air and higher indoor humidity levels during cold weather, preheat 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. Cross-leakage between the exhaust- and supply-air streams is seldom problematic in VAV systems since most of the air that returns (RA) from the spaces is recirculated (RRA) to the spaces as supply air anyway (Figure 113, p. 161).

Dual-Duct VAV Systems

Dual-duct VAV systems are intended for buildings in which some zones require cooling at the same time that other zones require heating. They are characterized by two separate duct systems: a cold duct and a hot duct. Each zone is served by a dual-duct VAV terminal unit, which consists of two airflow-modulation devices (Figure 115). One modulation device varies the amount of cool primary air and the other varies the amount of warm primary air. These two air streams mix inside the dual-duct unit before being distributed downstream to the zone.

Figure 115. Dual-duct VAV terminal unit



This system can provide excellent control of both temperature and humidity, and it can be very energy efficient when the mixing of cooled and heated air is avoided and two supply fans are used—one for heating, the other for cooling (Figure 116). In addition, all heating is done in the central air-handling unit. There are no heating coils or fans in the dual-duct VAV terminals and no associated water distribution system.

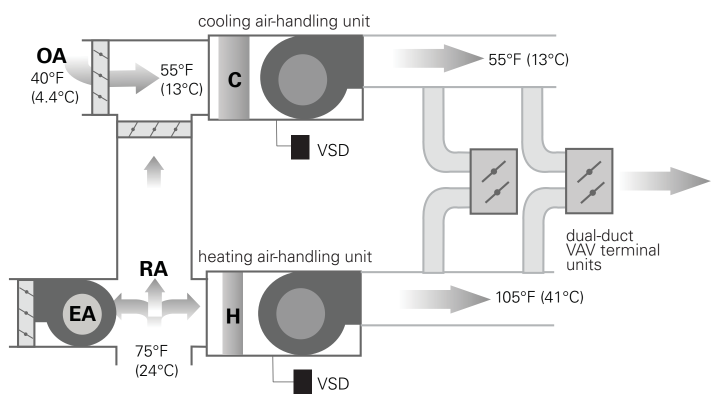
However, this system is relatively uncommon because of the need to install two separate duct systems. In addition, dual-duct VAV terminal units typically cost more than VAV reheat terminal units.

Dual- versus single-fan system

Dual-duct VAV systems may use either one or two central air-handling units.

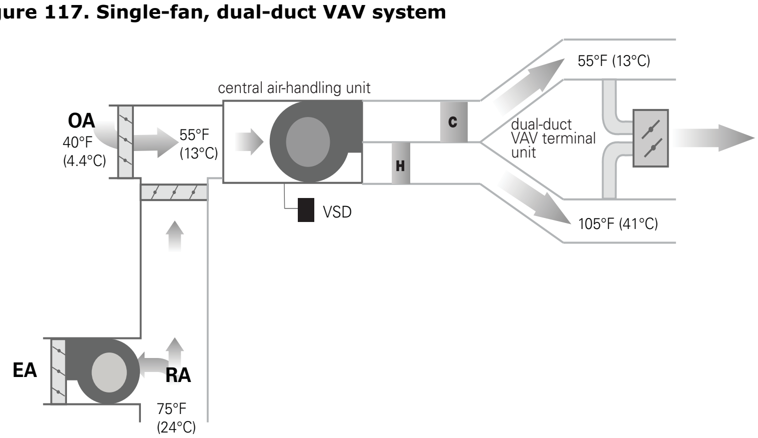
In the *dual-fan* configuration (Figure 116), the cooling air-handling unit mixes all of the outdoor air with a portion of the recirculated return air. This mixture is then cooled and delivered through the cold duct system to the cooling airflow-modulation device in each dual-duct VAV terminal unit. The heating air-handling unit typically conditions only recirculated return air, no outdoor air. This air is heated and delivered through the hot duct system to the heating airflow-modulation device (usually a damper) in each terminal unit.

Figure 116. Dual-fan, dual-duct VAV system



In the *single-fan* configuration (Figure 117), all the recirculated return air is mixed with the outdoor air inside a single air-handling unit. This mixture is then diverted through either the cooling coil or the heating coil and delivered down the respective duct system to the individual dual-duct VAV terminal units.

Figure 117. Single-fan, dual-duct VAV system



While the single-fan, dual-duct configuration requires only one air-handling unit, it is complicated to control efficiently. At the example operating conditions depicted in Figure 117, the airside economizer is modulating the outdoor- and return-air dampers to deliver the supply air at the desired temperature of 55°F (13°C). This economizer strategy saves cooling energy by avoiding the need to operate the cooling equipment (see "Airside economizer control," p. 174). However, the heating coil must use more energy to warm the air from 55°F (13°C) to the 105°F (41°C) primary air temperature. If the economizer was not activated, the mixed air temperature would be about 65°F (18°C), requiring less heating energy, but increasing cooling energy. For this reason, a single-fan, dual-duct system is not as energy efficient as a two-fan, dual-duct system.

In contrast, the dual-fan, dual-duct arrangement avoids this energy penalty. The cooling air-handling unit uses the airside economizer to save cooling energy and to use the cool outdoor air for "free cooling." Because the heating air-handling unit only conditions recirculated air, it is not penalized by the airside economizer, and need only warm the air from 75°F (24°C) to the 105°F (41°C) primary air temperature (Figure 116). This eliminates the wasteful reheating of mixed air and allows an airside economizer to provide free cooling whenever possible.

Variable- versus constant-volume to the zone

A dual-duct VAV terminal unit can be controlled to provide either a variable volume or a constant volume of supply air to the zone.

When designed to deliver a *variable* volume of supply air to the zone (Figure 118 and Figure 119), the cool primary airflow is reduced as the cooling load in the zone decreases. When the cool primary airflow reaches the minimum primary airflow setting for the unit and the zone cooling load continues to decrease, the heating damper begins to open. This allows the warm primary air to mix with the cool primary air and provide warmer supply air to the zone.

In a dual-fan, dual-duct VAV system (Figure 116), all of the outdoor air enters through the cooling air-handling unit. To ensure proper ventilation of the zone, therefore, the VAV terminal needs to maintain a minimum cool primary airflow through the cooling damper (Figure 118). In a single-fan, dual-duct VAV system (Figure 117), outdoor air enters through the common air-handling unit, so it can be delivered by either the cold or hot duct system. To ensure proper ventilation, therefore, the VAV terminal needs to maintain a minimum combined primary airflow, but the cooling damper can close (Figure 119).

Figure 118. Control of a dual-duct VAV terminal: Variable-volume to the zone (dual-fan system)

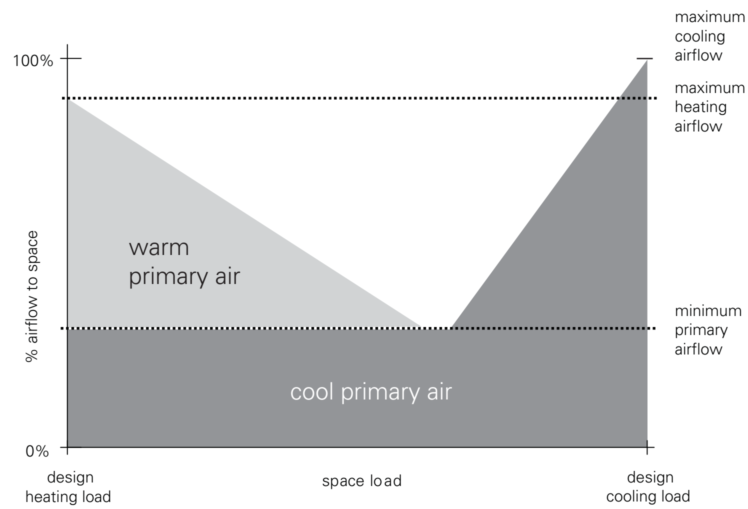
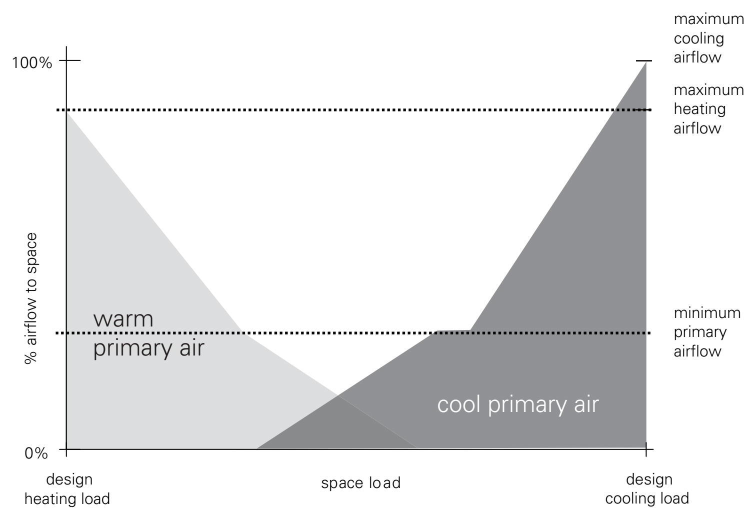


Figure 119. Control of a dual-duct VAV terminal: Variable-volume to the zone (single-fan system)



When designed to deliver a *constant* volume of supply air to the zone (Figure 120 and Figure 121), as the cooling load in the space decreases, the amount of cool primary air is reduced, and the amount of warm primary air is increased. The zone receives a constant total supply airflow (at a variable temperature). Because of this constant airflow to the zone, little or no fan energy savings is realized at part-load conditions.

For the same reasons described previously, in a dual-fan system (Figure 116) the VAV terminal needs to maintain a minimum cool primary airflow through the cooling damper to maintain proper ventilation of the zone (Figure 120). In a single-fan system (Figure 117), however, the cooling damper can close completely, as long as the VAV terminal maintains a constant airflow to the space (Figure 121).

Figure 120. Control of a dual-duct VAV terminal: Constant-volume to the zone (dual-fan system)

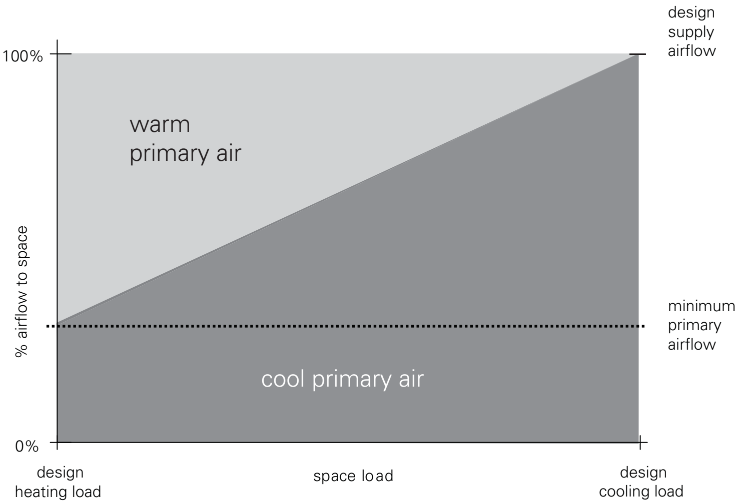
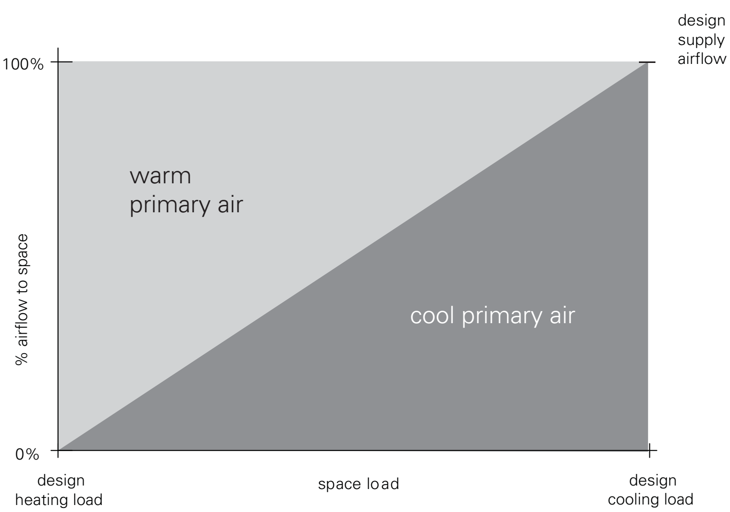


Figure 121. Control of a dual-duct VAV terminal: Constant-volume to the zone (single-fan system)



Best practices for dual-duct VAV systems

When using a dual-duct VAV system, consider the following general recommendations:

- *Use a dual-fan, duct-duct configuration to maximize energy efficiency.* As described earlier, the dual-fan arrangement (Figure 116) results in less overall energy use, because it avoids wasteful reheat of the mixed air when the airside economizer is activated to save cooling energy.

  In addition, the dual-fan arrangement allows the heating air-handling unit to deliver "neutral" air (unheated, recirculated air) down the hot duct during the cooling season. During very low cooling loads, the dual-duct VAV terminal mixes this recirculated air with cooling primary air, "tempering" the supply air to avoid overcooling the zone. Similar to a

For more information on dual-duct VAV systems, refer to the following *ASHRAE Journal* articles:

1) Warden, D. "Dual Fan, Dual Duct Systems: Better Performance at a Lower Cost," *ASHRAE Journal* (January 1996): pp. 36-41. Available at www.ashrae.org.

2) Warden, D. "Dual Fan, Dual Duct Goes to School," *ASHRAE Journal* (May 2004): pp. 18-25. Available at www.ashrae.org.

For a detailed discussion of the ventilation calculations for a dual-fan, dual-duct VAV system, refer to the May 2005 *ASHRAE Journal* article, titled "Standard 62.1: Designing Dual-Path, Multiple-Zone Systems" (available at www.ashrae.org).

fan-powered VAV terminal, mixing this recirculated air with cool primary air avoids the need to use "new" energy for heating. Elimination of reheat is the greatest energy benefit of the dual-fan, dual-duct VAV system.

- *Use Appendix A ("calculated E_v" method) from ASHRAE Standard 62.1 to calculate the system-level ventilation requirement.*
  Dual-fan, dual-duct VAV systems have two paths for delivery of outdoor air to the zone: one path is the primary air stream from the cooling air-handling unit, which brings in outdoor air and mixes it with recirculated return air, and the other path is the heating air-handling unit, which recirculates "unused" ventilation air with return air from all zones.

  Appendix A in ASHRAE 62.1 provides a calculation method for determining system ventilation efficiency ("calculated E_v" method), which gives credit to systems with multiple recirculation paths and typically results in a lower outdoor-air intake requirement (V_{ot}).

- *Implement variable (rather than constant) airflow to the zone wherever possible.*
  While some zones benefit from constant airflow, many zones can accommodate variable airflow. Controlling the dual-duct VAV terminal to deliver variable airflow to the zone (Figure 118 and Figure 119) results in less fan energy used.

  However, as discharge airflow (V_{dz}) to the zone decreases, the zone outdoor-air fraction (Z_d) increases, which can result in a high system intake flow (V_{ot}). In some cases, this benefit of additional fan energy savings may be accompanied by an increase in the energy required to condition this additional outdoor air brought in at the air-handling unit.

System Controls

This chapter discusses the control of a chilled-water VAV 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.

Unit-Level Controls

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 chilled-water VAV system, specific details should be obtained from the manufacturer of the equipment. Extended discussions in this section will be limited to those unit-level control issues that require decisions to be made by the HVAC system designer or system operator.

VAV air-handling unit

Typically, the central air-handling unit is equipped with a dedicated, unit-level controller that communicates with the building automation system (BAS). In a VAV application, this controller typically performs the following functions:

Discharge-air temperature control

A sensor measures the temperature of the air leaving the air-handling unit. The controller compares this measured temperature to the desired setpoint, and modulates a control valve on the chilled-water cooling coil, varies the position of the outdoor-air damper (for economizer operation), or modulates a control valve on the hot-water heating coil or gas-fired burner.

When on-off (or step) control of cooling or heating capacity air is used, such as electric heaters or two-position valves, the actual temperature of the discharge air may swing above and below the desired setpoint. However, this typically has little impact on comfort within the zone.

Ventilation control

In a VAV application, as supply airflow drops at part-load conditions, so does return airflow. With less return airflow, the pressure drop from the zone to the air-handling unit is reduced. If the outdoor-air damper remains at a fixed position, the pressure inside the mixing box (where outdoor air mixes with recirculated return air) becomes less negative, and outdoor airflow will decrease (Figure 86, p. 114). The ventilation control loop ensures that the required quantity of outdoor air is brought into the system at all operating conditions.

Two common methods of controlling the outdoor-air (OA) damper are:

1 Varying the position of the OA damper in proportion to the change in supply airflow, which is determined by the signal being sent to the variable-speed drive on the supply fan.

2 Using an airflow-measuring device to directly measure and control the outdoor airflow.

Both of these methods were discussed in "Ventilation," p. 101. In addition, system-level ventilation optimization is discussed in "Ventilation optimization," p. 205.

Supply-fan capacity control

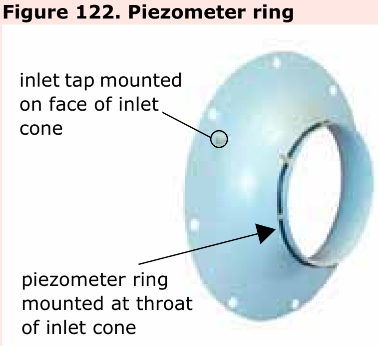
The modulation range of the supply fan is limited by how far the variable-speed drive can be turned down (typically 30 to 40 percent of design airflow).

As cooling load changes, the VAV terminal units modulate to vary the airflow supplied to the zones. This causes the pressure inside the supply ductwork to change. A sensor measures the static pressure at a particular location in the supply duct. The controller compares this measured pressure to the desired setpoint, and adjusts the capacity of the supply fan to deliver enough air to maintain the desired static pressure in the supply duct.

Measuring fan airflow

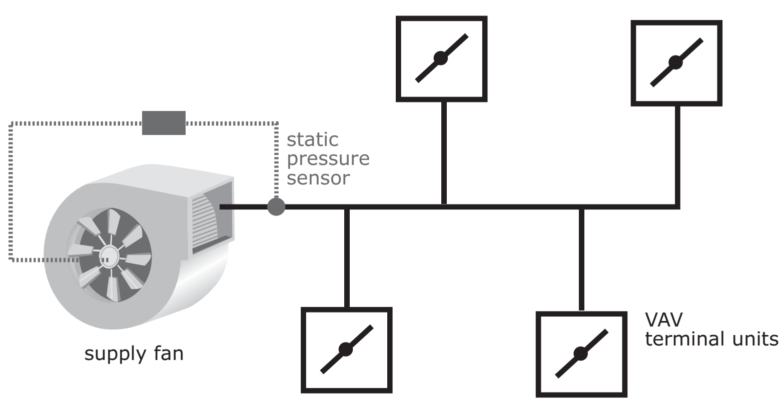
In a VAV system, measurement of actual fan airflow can be very useful for troubleshooting and ensuring proper system operation. A piezometer ring (Figure 122) is a device that can be mounted within the inlet cone for many types of fans. It measures the pressure drop from the inlet of the cone to the throat, which is used to estimate airflow, without obstructing airflow through the inlet of the fan.

Figure 122. Piezometer ring



The location of this duct static pressure sensor impacts the energy use of the supply fan, as well as the ability of the system to provide acceptable comfort to the building occupants. In some VAV systems, this static pressure sensor is mounted near the outlet of the supply fan (Figure 123). The appeal of this approach is that the sensor can be factory-installed and -tested, resulting in greater reliability and no field installation cost. If fire dampers are included in the supply duct system, the sensor will be on the fan side of the fire dampers. In this location, this sensor can also be used as the high-pressure cutout to shut off the fan and protect the ductwork from damage in the event that the fire dampers close. Also, depending on the layout of the duct system, this method may eliminate the need for multiple duct-mounted sensors.

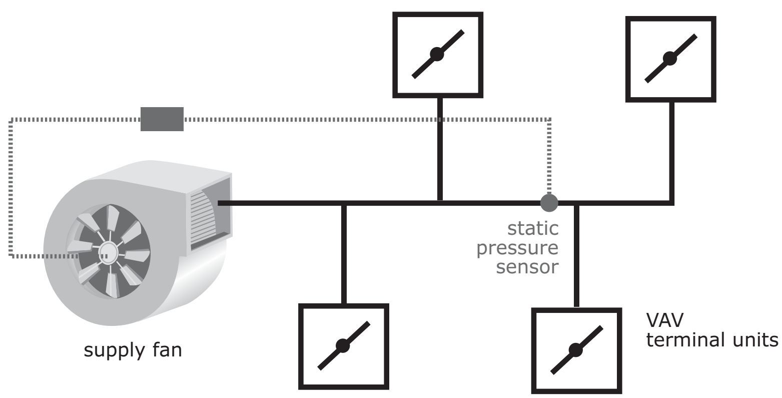
Figure 123. Static pressure sensor located at outlet of supply fan



It is not as energy efficient as other methods, however, because the setpoint must be equal to the pressure at the fan discharge when the supply fan is delivering full (design) airflow. For this reason, ASHRAE 90.1 prohibits the sensor from being installed in this location, unless the fan-pressure optimization control strategy is used (see “VAV fan control,” p. 132).

A more common practice is to locate the static pressure sensor approximately two-thirds of the distance between the supply fan outlet and the inlet to the “critical” VAV terminal unit (Figure 124). The “critical” VAV terminal unit is at the end of the supply duct path that represents the largest overall static pressure drop.

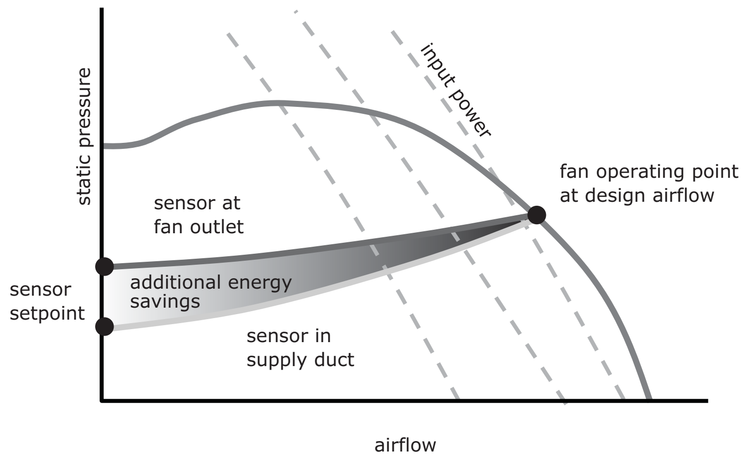
Figure 124. Static pressure sensor located 2/3 of distance down supply duct



For more information on the impact of the static pressure sensor location on supply fan energy use in VAV systems, refer to the Trane *Engineers Newsletter* titled “VAV System Optimization: Critical Zone Reset” (ENEWS-20/2).

In this configuration, the pressure sensor(s) must be field-installed. In larger systems with many VAV terminals, determining the best location for this sensor at all load conditions can be difficult—often determined by trial and error or requiring the installation of multiple sensors. Field installation and adjustment of one, or possibly several, duct pressure sensors increases installation cost. However, using this method typically allows for more fan energy savings than compared to the fan outlet method, because the setpoint is just high enough to maintain the pressure corresponding to that location in the duct system at design supply airflow (Figure 125). In addition, the sensor in the supply duct allows fan airflow to be reduced before the fan enters the surge region.

Figure 125. Fan energy saved using static pressure sensor in supply duct rather than at fan outlet



ASHRAE 90.1 requires the static pressure sensor be located "such that the controller setpoint is no greater than one-third the total design fan static pressure," unless the fan-pressure optimization control strategy is used (see "VAV fan control," p. 132).

A third approach uses the communicating controllers mounted on the VAV terminal units to optimize the static pressure setpoint based on the position of the damper in the "critical" terminal unit, that is, the VAV terminal that requires the highest pressure at the inlet. This strategy, called "fan-pressure optimization," is discussed further on p. 200.

Fan-pressure optimization results in the greatest amount of fan energy savings at part load, and allows the static-pressure sensor to be located anywhere in the supply duct system. This presents the opportunity to have it factory-installed and -tested at the fan outlet inside the air-handling unit.

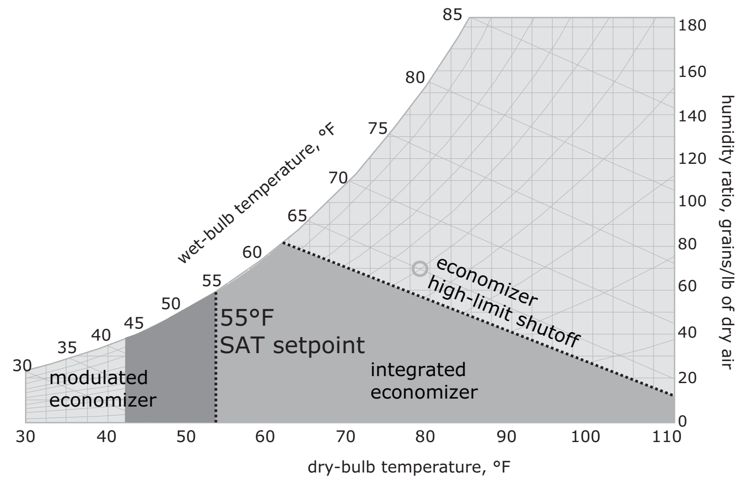
For a chilled-water VAV system in which providing an airside economizer is difficult or costly (because the outdoor-air ductwork must be sized for full economizer airflow), or where a significant amount of humidification is needed, the system can be designed with a **waterside economizer** (p. 91).

Airside economizer control

An airside economizer is a common energy-saving control strategy that uses outdoor air as a source of "free" cooling whenever possible. When the outdoor air is cool enough, the air-handling unit uses it to offset as much of the cooling load as possible.

When the dry-bulb temperature of the outdoor air is colder than the current supply-air-temperature (SAT) setpoint (55°F [13°C] for the example shown in Figure 126), 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. In this "modulated economizer" mode, the outdoor air is cool enough to handle the entire load, and the control valve on the chilled-water cooling coil is closed. The outdoor-air damper is allowed to modulate between the minimum position required for proper ventilation and wide open.

Figure 126. Modulated versus integrated economizer modes



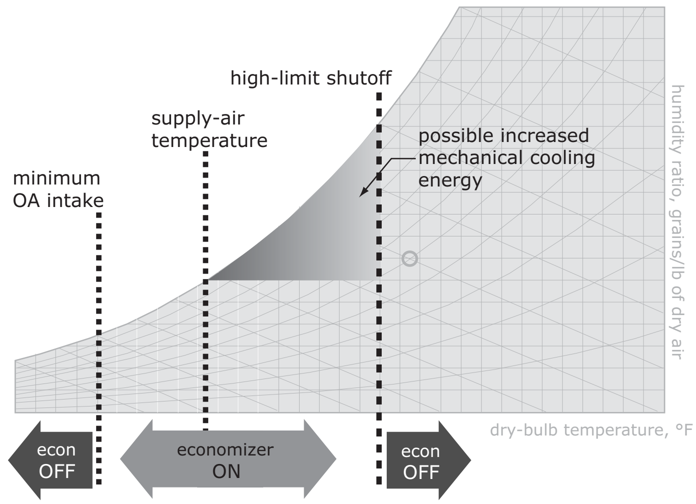
When the dry-bulb temperature of the outdoor air rises above the current supply-air temperature setpoint, the system needs more cooling capacity than the outdoor air alone can provide. The outdoor-air dampers remain wide open (return-air dampers are closed), but the unit controller modulates the valve on the cooling coil to provide the balance of the cooling capacity needed to achieve the desired supply-air temperature. This is called "integrated economizer" mode (Figure 126) because both the economizer and cooling coil are used to satisfy the cooling load.

As outdoor air temperature continues to rise, it eventually takes more mechanical energy to cool all outdoor air than it would take to cool a mixture of outdoor air and recirculated return air. At this point, the outdoor-air damper closes to the minimum position required for proper ventilation, and the valve on the cooling coil modulates as necessary to achieve the desired supply-air temperature. This decision to disable economizer operation is typically made automatically, by comparing the condition of the outdoor air to a setpoint called the "high-limit shutoff setting" (Figure 126).

While there are numerous high-limit shutoff strategies, the most common strategies used to control the economizer in a VAV system are: fixed dry-bulb control, fixed enthalpy control, and differential (or comparative) enthalpy control.

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 (Figure 127).

Figure 127. Fixed dry-bulb control of the airside economizer

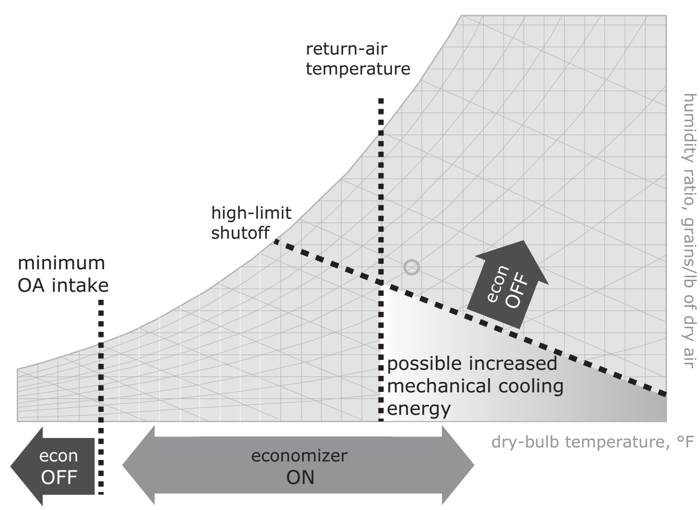


For more information on the various methods of controlling airside economizers, and their impact in VAV systems, refer to the Trane *Engineers Newsletter Live* broadcast DVD titled "HVAC Systems and Airside Economizers" (APP-CMC026-EN) and the Trane *Engineers Newsletter* titled "Airside Economizers" (ADM-APN020-EN).

While this method is simple and relatively inexpensive, fixed dry-bulb control can result in suboptimal performance. For instance, in many 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 actually increase mechanical cooling energy (Figure 127).

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 (Figure 128).

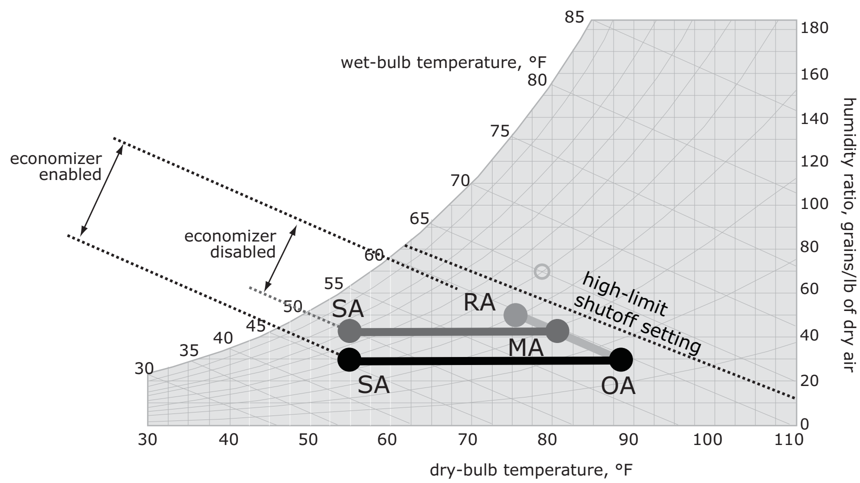
Figure 128. Fixed enthalpy control of the airside economizer



Fixed enthalpy control requires an added outdoor humidity sensor, so it costs more than fixed dry-bulb control. But for a VAV system in most climates, fixed enthalpy control saves more cooling energy, so it might be worth the added first cost. However, in hot and dry climates, bringing in 100 percent outdoor air can actually increase mechanical cooling energy, even if the outdoor-air enthalpy is low (Figure 128). *Note: Because of this, Table 6.5.1.1.3A of ASHRAE Standard 90.1-2007 prohibits the use of a fixed enthalpy high-limit shutoff strategy in many of the dry and marine climate zones.*

Figure 129 shows one of these hot, dry days. The enthalpy of the outdoor air (OA) is below the high-limit shutoff setting, so the fixed enthalpy economizer opens the outdoor-air damper to 100 percent. The controller then modulates the valve on the cooling coil to cool this air to the desired supply-air (SA) temperature setpoint. However, if the economizer was disabled, recirculated return air (RA) would mix with the minimum required outdoor air (OA), and then this mixed air (MA) would pass through the cooling coil.

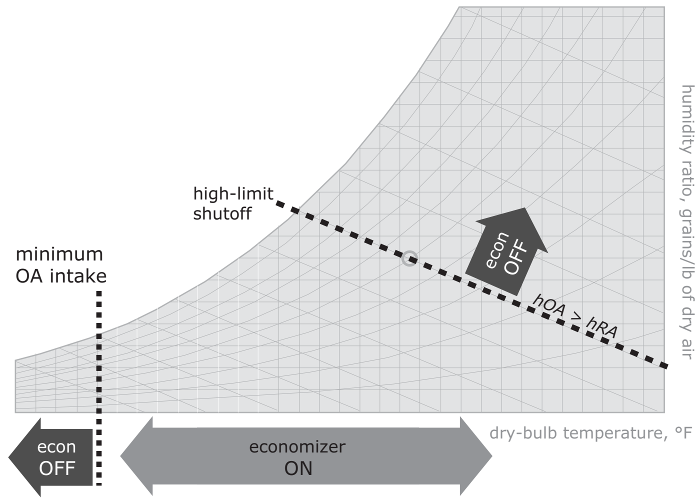
Figure 129. Fixed enthalpy control in hot, dry climates



If the economizer is enabled, the enthalpy difference across the cooling coil is greater than if the economizer is disabled. Because of the dry outdoor conditions, the cooling coil is not dehumidifying; it is only removing sensible heat. So, if the outdoor air (OA) is warmer than the return air (RA), it might take more cooling energy to achieve the same supply-air temperature.

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 (Figure 130).

Figure 130. Differential enthalpy control of the airside economizer



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 typically results in the most cooling energy saved, compared to the other control types, which may make up for the first-cost increase.

Climate, building use, and utility costs impact the operating cost differences of these different methods of economizer control.

Building pressure control

For more information on the issues related to improper building pressure control, and the various methods of controlling building pressure in VAV systems, refer to the Trane *Engineers Newsletter* titled "Commercial Building Pressurization" (ADM-APN003-EN) and the Trane *Engineers Newsletter Live* broadcast DVD titled "Commercial Building Pressurization" (APP-APV013-EN).

The use of an airside economizer, dynamic reset of ventilation air (e.g. demand-controlled ventilation), the presence of wind, stack effect, and intermittent operation of local exhaust fans can all cause undesirable changes in building pressure. During humid weather, maintaining the pressure inside the building so that it is slightly higher than the pressure outside ("positive" pressure) increases comfort and helps reduce the leakage of humid outdoor air 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 reduces the likelihood of 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.

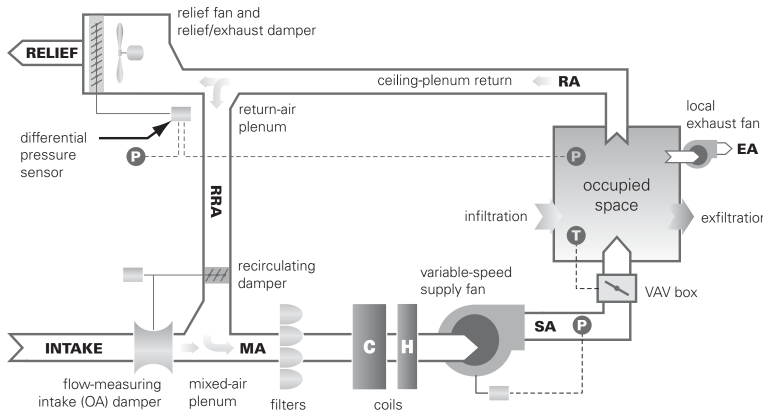
This indoor-to-outdoor pressure difference can be controlled by adjusting either the quantity of air brought into, or exhausted from, the building. In most VAV systems, the minimum quantity of outdoor air brought into the building is based on the ventilation requirements of a local building code, so controlling building pressure typically involves varying the quantity of air exhausted from the building.

The method used to control building pressure depends on the configuration of the fans in the system. For a discussion of the different fan configurations, including the benefits and drawbacks of each, see "Fans," p. 26.

If the air-handling unit includes **a supply fan and a relief fan**, building pressure is directly controlled by varying the capacity of the central relief fan (Figure 131):

- A differential pressure sensor monitors the indoor-to-outdoor pressure difference. Its signal is used to adjust relief airflow, directly limiting building pressure. Relief 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.
- The relief fan only operates when necessary to relieve excess building pressure. When intake airflow is at the minimum required for ventilation (as is often the case when not economizing), operation of the central relief fan may be unnecessary. In some buildings, local exhaust fans and exfiltration due to the pressurized building are sufficient to relieve the intake airflow.

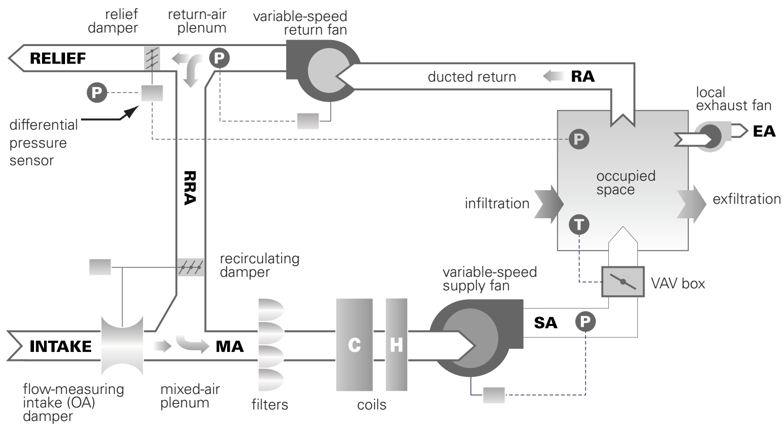
Figure 131. VAV system with central relief fan



If the air-handling unit includes **a supply fan and a return fan**, building pressure is directly controlled by varying the position of the relief damper (Figure 132):

- A differential pressure sensor monitors the indoor-to-outdoor pressure difference. Its signal modulates the position of the relief damper, directly limiting building pressure. The outdoor- and recirculating-air dampers may share the same actuator, but the relief damper must be controlled separately to accommodate varying building-pressure setpoints.
- The return fan, which operates whenever the supply fan does, pulls return air from the occupied zones and pressurizes the return-air plenum in the air-handling unit. Air from this plenum either passes through the recirculating damper into the mixed-air plenum or exits the building through the relief damper. Various methods for controlling the capacity of the return fan are discussed in "Return-fan capacity control," p. 181.

Figure 132. VAV system with central return fan



Regardless of fan configuration, 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 that is sent to the controller on the air-handling unit. Two sensing tubes (one measuring indoor pressure and the other measuring outdoor pressure) are attached to the transducer. Proper location of these pressure sensing tubes is important:

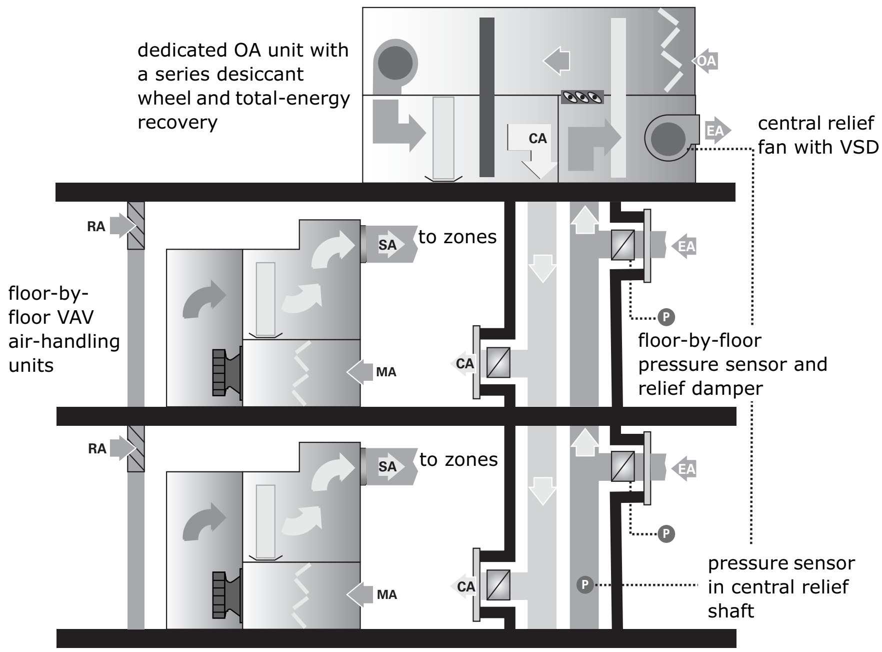
- The *indoor sensor* is typically located on the ground floor, because the effects of over- or under-pressurization are most noticeable at the external doors. Many design engineers locate the sensor in a large open space near the door, while others isolate the indoor sensor from the door (in a central 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, and locate them to minimize the effects of wind.

Finally, due to wind, stack effect, and different operating patterns of floor-by-floor air-handling units, building pressure control can be particularly difficult in taller buildings. Figure 133 demonstrates one approach to control building pressure at each floor.

Figure 133. Floor-by-floor building pressure control



A differential pressure sensor monitors the indoor-to-outdoor pressure difference on each floor. Its signal modulates the position of the relief damper for that floor, directly controlling building pressure. The floor-by-floor relief dampers are all connected to a relief (exhaust) shaft with a central relief fan. The signal from a pressure sensor in this relief shaft is used to modulate the capacity of the relief fan through the use of a variable-speed drive (VSD).

The "fan-pressure optimization" (see p. 200) can also be used to control the central relief fan. In this case, the BAS continually polls the individual relief dampers, looking for the one with the most-open damper. The static-pressure setpoint is then reset so that at least one of the relief dampers is nearly wide open. The result is that the relief fan generates only enough negative pressure in the relief shaft to maintain building pressure on the "critical" floor (furthest open relief damper).

Return-fan capacity control

In a system that includes a return fan, this fan operates whenever the supply fan does, and pulls return air from the occupied zones back to the return-air plenum inside the air-handling unit (Figure 132). The air in this plenum should be at a positive pressure, relative to outdoors, so that it either passes through the recirculating damper into the mixed-air plenum or exits the building through the relief damper.

Historically, system design engineers have used several approaches for controlling the capacity of a return fan in a VAV system, including:

- **Return-air plenum pressure control** uses a pressure sensor to measure the static pressure inside the return-air "plenum" in the air-handling unit and modulates the capacity of the return fan to maintain this pressure at a desired setpoint (Figure 132). This approach is recommended by ASHRAE Guideline 16, *Selecting Outdoor, Return, and Relief Dampers for Air-Side Economizer Systems*, for VAV systems. With this approach, the relief damper is modulated to directly control building pressure at the desired setpoint, and the OA and RA dampers are controlled to ensure that the proper amount of outdoor air is brought in for ventilation.

- **Flow tracking** uses airflow measurement devices to measure both supply and return airflows. The capacity of the return fan "tracks" supply airflow to maintain a fixed airflow differential between the two (accounting for local exhaust fans and building pressurization). Flow tracking works best in applications that have a constant amount of local exhaust and are only minimally affected by wind and stack effect. Successful implementation requires well-calibrated flow sensors because the difference between supply and return airflows can be a very small fraction of the overall sensed airflow.
- **Signal tracking** monitors the pressure in the supply duct and uses that signal to modulate the speeds of both the supply *and* return fans. Building pressure is controlled indirectly (and ineffectively) because of the disparity between the performance curves of the two fans. Therefore, this approach is not recommended.

Safeties

The unit-level controller for the VAV air-handling unit typically includes several safeties that protect the equipment from harm. Common examples include:

- A freeze protection sensor that turns off the fan and/or closes the OA damper when a coil containing water is exposed to air that is cold enough to freeze (see "Freeze prevention," p. 18).
- A pressure switch that turns off the fan(s) to prevent damaging the ductwork when it measures an excessively high static pressure in the duct.
- One or more temperature sensors that avoid operating a gas-fired burner or electric heater at temperatures above the manufacturer's recommendation for safe operation.
- A condensate overflow float switch that turns off the fan, closes the chilled-water control valve, and closes the OA damper to prevent overflowing the drain pan in the event that the condensate drain line is plugged.
- Electrical interlocks that turn off ultraviolet lights (if installed) when a door is opened to avoid inadvertent exposure of service personnel to UV-C light.

These are just examples. Specific details on safeties should be obtained from the equipment manufacturer.

VAV terminal units

Similar to the air-handling unit, each VAV terminal unit is typically equipped with a dedicated, unit-level controller that communicates with the building automation system. This controller typically performs the following functions:

Zone temperature control

A zone sensor measures the temperature in the zone. The unit-level controller compares this measured temperature to the desired setpoint and, depending on the type of VAV terminal unit, varies the quantity of air delivered to the zone, cycles the terminal fan on and off, and/or modulates or stages a heating coil (see "Types of VAV terminal units," p. 55).

When a heating coil is part of the VAV terminal unit, the coordination of cooling and heating is done by the unit-level controller. However, when a remote heat source is used, such as baseboard radiant heat installed under windows, careful coordination is required to prevent the two systems from "fighting each other" (providing cooling and heating simultaneously) and wasting energy.

The most straightforward approach to prevent simultaneous cooling and heating is to allow the unit-level controller on the VAV terminal unit to also control the remote heat source. When the zone requires heating, the unit-level controller responds just as if it was controlling a heating coil inside the VAV terminal unit:

- For a cooling-only VAV terminal or series fan-powered VAV terminal with no heating coil, primary airflow is at the minimum setting and the unit-level controller sends signals to modulate (or stage) the remote heat source.
- For a parallel fan-powered VAV terminal with no heating coil, primary airflow is at the minimum setting. The first stage of heating is provided by activating the small terminal fan to mix warm plenum air with the cool primary air. If further heating is required, the unit-level controller sends signals to modulate (or stage) the remote heat source.

Some systems use a separate, outdoor-air temperature sensor to control the baseboard heating system. As the temperature outside gets colder, the capacity of the baseboard heat is increased to provide more heat. This approach is not recommended in VAV systems, because it is difficult to coordinate and often results in simultaneous heating (baseboard) and cooling (VAV terminal).

Another approach uses two separate zone temperature sensors: one to control the cooling provided by the VAV terminal unit and the other to control the baseboard heating system. As long as there is a suitable deadband, and the two temperature sensors are well calibrated and located next to each other, this approach can be effective at preventing the cooling and heating systems from fighting each other and wasting energy.

Oftentimes, in an attempt to reduce installation costs, the heating control zones are larger than the cooling control zones. In this case, a single temperature sensor might be used to control the baseboard heat for an area of the building that contains several temperature sensors to control the VAV terminal units. In this case, it is possible for the areas within the heating zone that do not contain the heating sensor to fight with the cooling zones due to variations in loads within the larger heating zone. To avoid this problem, increase the number of heating control zones or increase the deadband between the heating and cooling setpoints. While increasing the deadband may limit the energy wasted when the systems fight each other, it may also result in more occupant complaints regarding temperature control.

Again, the best control approach is to allow the unit-level controller on the VAV terminal unit to also control the remote heat source.

Ventilation control

For VAV systems that use direct digital controls (DDC), demand-controlled ventilation may be required by ASHRAE Standard 90.1 for densely occupied zones (see "Demand-controlled ventilation," p. 132).

An addendum (j) to ASHRAE Standard 62.1-2007 helps clarify the intent of the standard by adding the following to Section 8.3: "Systems shall be operated such that spaces are ventilated when they are expected to be occupied."

While the outdoor airflow delivered to the breathing zone (V_{bz}) can be reset as zone population (P_z) varies (per Section 6.2.7), the system must deliver at least the building-related (or "base") ventilation rate, R_a, whenever the zone is expected to be occupied [see "Minimum ventilation rate required in breathing zone (V_{bz})," p. 102].

In a VAV system that serves multiple zones, zone-level DCV must be coordinated to determine how to control the system-level outdoor-air intake (see "Ventilation optimization," p. 205).

In addition to controlling the temperature in the zone, the unit-level controller on a VAV terminal unit can also help ensure that the zone receives the required amount of outdoor air for ventilation.

As mentioned in "Dynamic reset of intake airflow," p. 117, ASHRAE Standard 62.1 permits dynamic reset of intake (outdoor) airflow (V_{ot}) 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. Commonly used methods for assessing zone population include:

- *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 system are scheduled to be occupied), minimum outdoor airflow for that zone is reset to zero (or to the building-related ventilation rate, R_a; see "Zone-level ventilation requirements," p. 101).

  Alternatively, a time-of-day schedule can be used to predict the actual number of people in a zone for any given hour. This variation in population is then communicated to the unit-level controller for the VAV terminal unit and used to reset the minimum outdoor airflow currently required for the zone for that hour.

  If the VAV system includes a BAS, it probably includes a time-of-day scheduling function, so the only additional cost is time to set up the schedules. This approach may be well-suited for a classroom, where occupancy is predictable and the number of occupants does not vary greatly.

- *Occupancy sensors*
  A 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 for the VAV terminal unit, indicating whether that zone should be considered "occupied" or "unoccupied." When the sensor indicates the zone is occupied, it requires the design outdoor airflow. When the sensor indicates the zone is unoccupied, the required outdoor airflow rate is reset to zero.

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 29). In this mode, called "occupied standby," the zone requires a less-than-design (base) outdoor airflow (typically the building-related ventilation rate, R_a).

Table 29. Combining an occupancy sensor with a time-of-day schedule

Time-of-day schedule reads	Occupancy sensor indicates	Operating mode of VAV terminal unit	Outdoor airflow setpoint
occupied	occupied	occupied	design outdoor airflow
occupied	unoccupied	occupied standby	"base" outdoor airflow (less than design)
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.

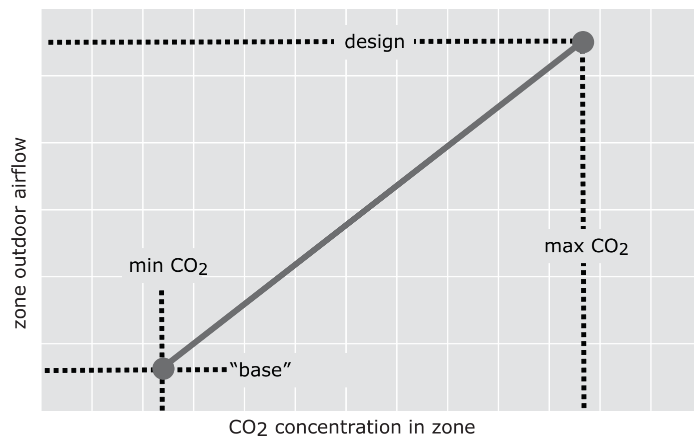
For more information on CO_2-based demand-controlled ventilation, refer to the Trane *Engineers Newsletter* titled "CO_2-Based Demand-Controlled Ventilation with ASHRAE Standard 62.1-2004" (ADM-APN017-EN), and to the Trane *Engineers Newsletter Live* broadcast DVD titled "CO_2-Based Demand-Controlled Ventilation" (APP-CMC024-EN).

- *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 may actually increase 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 for the VAV terminal unit and used to reset the outdoor airflow currently required for the zone (Figure 134). If the CO_2 concentration in the zone is less than or equal to the minimum CO_2 limit, the zone ventilation setpoint is at a less-than-design (base) outdoor airflow. On the other hand, if the CO_2 concentration is greater than or equal to the maximum CO_2 limit, the zone ventilation setpoint is equal to the design outdoor airflow requirement. If the CO_2 concentration is between the minimum and maximum CO_2 limits, the ventilation setpoint is adjusted proportionally between the "base" and design outdoor airflows.

Figure 134. Varying zone-level outdoor airflow requirement based on measured CO_2 concentration



Note: The simplest approach is to use the assumed outdoor CO_2 concentration as the minimum CO_2 limit. In a multiple-zone recirculating system, however, if a zone is unoccupied while other zones are occupied, the concentration of CO_2 in the unoccupied zone cannot drop down to the outdoor concentration. (Recirculated air from the occupied zones causes the CO_2 concentration of the supply air to be higher than the outdoor CO_2 concentration.) The result will be over-ventilation at very low occupancy levels. A more aggressive approach would be to set this minimum CO_2 limit higher than the outdoor CO_2 concentration, but determining this value is difficult and requires making many assumptions.

CO_2-based DCV requires a CO_2 sensor in each zone where it is used, and 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 good candidates for CO_2 sensors. Other "non-critical zones," which are likely to be over-ventilated at all times, never approach their maximum CO_2 levels, so CO_2 sensors in these zones are unnecessary.

It takes some time for the indoor concentration of CO_2 to decrease when people leave a room. An occupancy sensor can be used in combination with a CO_2 sensor to reduce zone ventilation more quickly, thus saving energy. When all the people have left the room, the occupancy sensor will indicate that the zone is unoccupied and this signal can be used to reduce the zone ventilation setpoint to the "base" outdoor airflow, even though the measured CO_2 concentration is still decreasing and has not yet reached the minimum CO_2 limit. In addition, when the minimum CO_2 limit is set equal to the assumed outdoor CO_2 concentration (see note above), the occupancy sensor can help prevent over-ventilation (and wasted energy) when the zone is unoccupied while other zones served by the same system are still occupied.

Alternatively, if the occupancy pattern of a zone is predictable, a time-of-day schedule in the BAS can be used in combination with a CO_2 sensor to reduce over-ventilation (and wasted energy) when the zone is unoccupied while other zones served by the same system are still occupied.

In most cases, the best value for a VAV system is to combine 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 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 and some classrooms—are good candidates for time-of-day schedules.

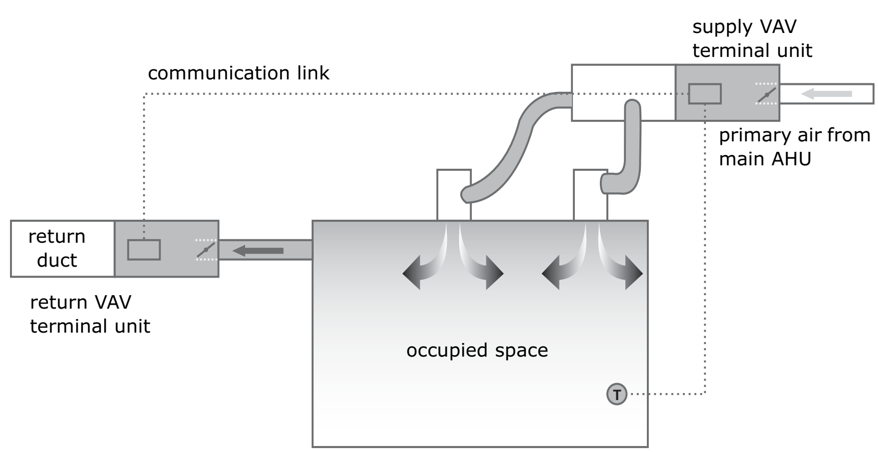
In a VAV system that serves multiple zones, zone-level ventilation control must be coordinated to determine how to control the system-level outdoor-air intake (see “Ventilation optimization,” p. 205).

Space pressure control

For certain types of spaces, the air pressure within the space is required to be controlled to be either higher (positive pressure) or lower (negative pressure) than the adjacent spaces. This is most common in certain health care facilities and laboratories.

One common approach for controlling space pressure is to install a second VAV terminal unit in the return duct leaving the space (Figure 135). The VAV terminal unit in the supply duct modulates to maintain space temperature, and to ensure the minimum air change rate required. The VAV terminal unit in the return duct modulates to maintain either a positive or negative pressure in the space. This can be done either through direct pressure measurement or by controlling the return airflow to a pre-defined offset from the current, measured supply airflow (airflow tracking).

Figure 135. Space pressure control via airflow tracking



When the airflow tracking method is used, this offset (which is determined by the air balancing contractor when the system is commissioned) is the amount of excess (or deficit) return air needed to maintain the space at the desired negative (or positive) pressure with respect to the adjacent space(s). For example, if the design supply airflow for the space is 1000 cfm (0.47 m^3/s) and the air balancing contractor determines that the return airflow must be 800 cfm (0.38 m^3/s) in order for the space to be at the desired positive pressure relative to the adjacent spaces, this offset is a negative 200 cfm (0.09 m^3/s). During operation, the VAV terminal unit in the return duct is controlled to an airflow setpoint that is 200 cfm (0.09 m^3/s) lower than the measured airflow currently passing through the VAV terminal unit in the supply duct.

Safeties

The unit-level controller for a VAV terminal unit typically includes several safeties that protect the equipment from harm. A common example is a low airflow limit to avoid operating electric heating coils at airflows below the manufacturer's recommendation for safe operation. Specific details on safeties should be obtained from the equipment manufacturer.

Water chiller

Each water chiller is equipped with a dedicated, unit-level controller that varies the cooling capacity of the chiller to supply chilled water at the desired temperature. For a chiller using the vapor-compression refrigeration cycle, this is accomplished by varying the capacity of the compressor. Scroll compressors generally cycle on and off. Reciprocating compressors either cycle on and off or use cylinder unloaders. Helical-rotary (or screw) compressors typically use a slide valve or a similar unloading device. Centrifugal compressors typically use inlet vanes or a variable-speed drive in combination with inlet vanes.

The controller also monitors chiller operation and protects it from damage by preventing it from operating outside acceptable limits. Some water chiller controllers are capable of adapting to unusual operating conditions, keeping the chiller operating by modulating its components and sending a warning message, rather than doing nothing more than shutting it down when a safety setting is violated.

Specific details about the water chiller controller should be obtained from the manufacturer.

Condensing pressure control

Every vapor-compression chiller requires a minimum refrigerant pressure difference between the evaporator and the condenser, in order to ensure that refrigerant and oil circulate properly inside the chiller. This pressure difference varies based on the chiller design and operating conditions.

For a packaged *air-cooled chiller*, this function is performed by the unit-level controller. However, if the chiller is expected to operate when it is very cold outside, it may need to be equipped with either a discharge damper or variable-speed drive on one or more of the condenser fans. This is typically an option available from the chiller manufacturer.

For a *water-cooled chiller*, the condenser-water system may need to be designed to vary either the temperature or flow rate of the water through the condenser. For example, consider an office building that has been unoccupied during a cool autumn weekend, during which time the temperature of the water in the cooling tower sump has dropped to 40°F (4°C). Monday is sunny and warm, and the building cooling load requires a chiller to be started. Pumping this cold water through the chiller condenser results in a very low refrigerant pressure inside the condenser. Because the chiller is operating at part load and the tower sump is relatively large, the minimum pressure difference may not be reached before the chiller is turned off on a safety.

The solution is to either increase the temperature of the water entering the condenser or decrease the flow rate of water through the condenser. Both of these strategies will increase the temperature and pressure of the refrigerant inside the condenser. After the minimum pressure difference is reached, system operation may return to normal.

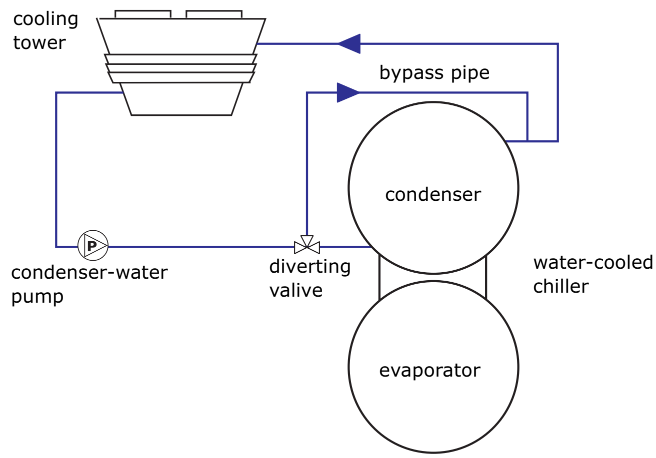
Common approaches for increasing the evaporator-to-condenser pressure difference include:

For more information on condensing pressure control for water-cooled chillers, refer to the Trane engineering bulletins titled "Condenser Water Temperature Control For CenTraVac® Centrifugal Chiller Systems" (CTV-PRB006-EN) and "Water-Cooled Series R® Chiller - Models RTHB & RTHD: Condenser Water Control" (RLC-PRB017-EN).

- Cycling or varying the speed of cooling tower fans to increase the temperature of water leaving the cooling tower
- Modulating a throttling valve to reduce the flow rate of water through the condenser
- Using a chiller condenser bypass pipe to reduce the flow rate of water through the condenser (Figure 136)
- Using a cooling tower bypass pipe to mix warm water leaving the condenser with the cold tower water, thus increasing the temperature of water entering the condenser
- Using a variable-speed drive on the condenser water pump to reduce the water flow rate through the condenser

Each of these strategies has its advantages and disadvantages. Selecting the appropriate condensing pressure control scheme will depend on the specific requirements of the application.

Figure 136. Chiller bypass for condensing pressure control



Hot-water boiler

For more information on the design of hot-water distribution systems, refer to Chapters 12 (Hydronic Heating and Cooling System Design) and 46 (Valves) of the 2008 *ASHRAE Handbook—HVAC Systems and Equipment* (www.ashrae.org) or *The Boiler Book* by Cleaver-Brooks (www.boilerspec.com).

Each hot-water boiler is equipped with a dedicated, unit-level controller that varies the heating capacity of the boiler to supply hot water at the desired temperature. 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.

Return water temperature control

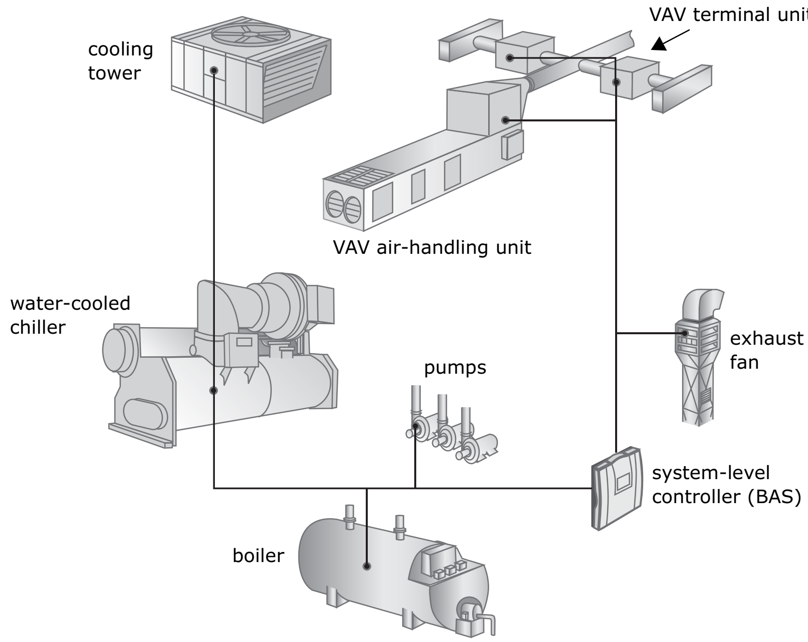
Non-condensing boilers require that the return-water temperature be no lower than 140°F (60°C) to prevent condensing. In some boilers, this function is performed using a temperature-actuated valve (internal to the boiler) to divert hot water from the supply and mix it with cooler return water.

Alternatively, the hot-water system may need to be designed to ensure a suitably warm return-water temperature. Common approaches include a pressure-actuated bypass valve or a primary-secondary pumping configuration.

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, the central air-handling unit, VAV terminal units, water chillers, and boilers are each equipped with separate unit-level controllers, and these unit-level controllers are connected to a centralized, system-level controller (Figure 137). With this configuration, each unit-level controller is capable of performing its functions, even if communication with the system-level controller is lost.

Figure 137. System-level control of a chilled-water VAV system



Coordination during different operating modes

One of the primary, system-level control functions is to coordinate the central air-handling unit, VAV terminal units, and other pieces of equipment during the various modes of operation. The primary system-level operating modes in a VAV system are:

- Occupied 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 VAV system must maintain the temperature in each occupied zone at the desired setpoint (cooling or heating), and provide the required minimum amount of outdoor air for ventilation. Table 30 describes the typical functions of the different system components during the occupied mode.

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.

Table 30. Coordination of equipment during occupied mode

Central air-handling unit	• Modulates supply fan to maintain static pressure in the supply duct at the desired setpoint • Modulates the cooling valve or heating valve to discharge air at the desired setpoint • Modulates the position of the outdoor-air damper to bring in at least the minimum required amount of outdoor air for ventilation • May open the outdoor-air damper further if the condition of the outdoor air is suitable to provide free cooling (economizer) • Modulates the central relief damper or relief fan to maintain indoor-to-outdoor static pressure difference at the desired setpoint • Modulates the central return fan (if equipped) to maintain static pressure in the return-air plenum of the air-handling unit at the desired setpoint[1]
VAV terminal	• Varies primary airflow, cycles terminal fan (if included), and/or modulates (or stages) a local or remote heat source to maintain zone temperature at the occupied setpoint (cooling or heating)
Chilled-water plant	• Turns on chilled-water pumps when cold water is needed (if a variable-flow system, varies the speed of the pumps to maintain pressure in the chilled-water piping at the desired setpoint) • Turns on chillers and varies chiller capacity to cool water to the desired setpoint • If water-cooled, turns on condenser-water pumps when chillers need to operate and modulates cooling tower fans to cool condenser water to the desired setpoint
Hot-water plant (if included)	• Turns on hot-water pumps when hot water is needed (if a variable-flow system, varies the speed of the pumps to maintain pressure in the hot-water piping at the desired setpoint) • Turns on boilers and varies boiler capacity to heat water to the desired setpoint

1 If flow tracking is used, the central return fan modulates to maintain a fixed airflow differential from the supply fan airflow.

Occupied standby mode

An addendum (j) to ASHRAE Standard 62.1-2007 helps clarify the intent of the standard by adding the following to Section 8.3: "Systems shall be operated such that spaces are ventilated when they are expected to be occupied."

While the outdoor airflow delivered to the breathing zone (V_{bz}) can be reset as zone population (P_z) varies (per Section 6.2.7), the system must deliver at least the building-related (or "base") ventilation rate, R_a, whenever the zone is expected to be occupied [see "Minimum ventilation rate required in breathing zone (V_{bz})," p. 102].

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 is used to switch the zone to an "occupied standby" mode (see example in Table 31). In this mode, all or some of the lights in that zone can be shut off, the temperature setpoints can be raised or lowered by 1°F to 2°F (0.5°C to 1°C), and the minimum required outdoor airflow for that zone can be reduced, typically to the building-related (or "base") ventilation rate, R_a, required by ASHRAE Standard 62.1 [see "Minimum ventilation rate required in breathing zone (V_{bz})," p. 102]. The purpose of each of these actions is to save energy.

In addition, the minimum primary airflow setting of the VAV terminal serving that zone can often be lowered to avoid or reduce the need for reheat. The minimum primary airflow setting is typically selected to ensure proper ventilation or to increase air circulation for occupant comfort (see "Minimum primary airflow settings," p. 62). However, with no occupants and a reduced ventilation requirement, the minimum primary airflow setting can be lowered significantly, minimizing or avoiding reheat energy.

Table 31. 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)	60 cfm (28 L/s)
Minimum primary airflow setting	450 cfm (212 L/s)	225 cfm (106 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 ANSI/ASHRAE Standard 62.1-2007, 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, V_{bz} = 5 cfm/p x 0 people + 0.06 cfm/ft^2 x 1000 ft^2 = 60 cfm (2.5 L/s/p x 0 people + 0.3 L/s/m^2 x 93 m^2 = 28 L/s).

When the occupancy sensor indicates that the zone is again occupied, the zone is switched back to occupied mode.

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 "Zone is unoccupied," p. 4). But the system must still prevent the zones from getting too cold, perhaps 55°F (13°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 outdoor-air dampers 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.

In some cases, it may be important to control humidity, in addition to temperature, when the building is unoccupied. The BAS can monitor indoor humidity levels and take action if the humidity rises above a maximum limit (see "After-hours dehumidification," p. 125) or drops below a minimum limit.

Table 32 describes the typical functions of the different system components during the unoccupied mode.

Table 32. Coordination of equipment during unoccupied mode

Central air-handling unit	• Supply fan is off unless the zones require cooling or heating[1] • If the supply fan is on, modulates the cooling valve or heating valve to discharge air at the desired setpoint • Outdoor-air damper is closed, unless the zones require cooling and the condition of the outdoor air is suitable to provide free cooling (economizer) or if exhaust fans operate during the unoccupied mode (not recommended) • Central return fan (if equipped) is off, unless supply fan is on, and then it is modulated to maintain static pressure in the return-air plenum of the air-handling unit at the desired setpoint[2]
VAV terminal	• Varies primary airflow, cycles terminal fan (if included), and/or modulates (or stages) a local or remote heat source to maintain zone temperature at the unoccupied setpoint (cooling or heating)
Chilled-water plant	• Turns on chilled-water pumps when cold water is needed (if a variable-flow system, varies the speed of the pumps to maintain pressure in the chilled-water piping at the desired setpoint) • Turns on chillers and varies chiller capacity to cool water to the desired setpoint • If water-cooled, turns on condenser-water pumps when chillers need to operate and modulates cooling tower fans to cool condenser water to the desired setpoint
Hot-water plant (if included)	• Turns on hot-water pumps when hot water is needed (if a variable-flow system, varies the speed of the pumps to maintain pressure in the hot-water piping at the desired setpoint) • Turns on boilers and varies boiler capacity to heat water to the desired setpoint

1 If the system includes baseboard radiant heat, it could be used to maintain the unoccupied heating setpoint and the main supply fan could remain off. In addition, if fan-powered VAV terminals are equipped with heating coils, only the terminal fan and heating coil for the zone requiring heat would need to be activated, and the main supply fan could remain off.

2 If flow tracking is used, the central return fan modulates to maintain a fixed airflow differential from the supply fan airflow.

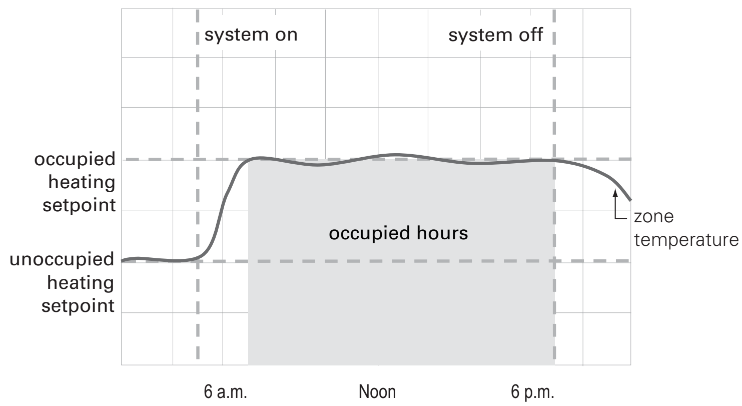
Figure 138. Zone temperature sensor with a timed override button

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 138). Typically, pressing this button directs the system to operate in the occupied mode for only a fixed period of time (three hours, for example). After this time period expires, the BAS automatically returns the zone to unoccupied mode.

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 purposes 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 enter the building (Figure 139). 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 139. 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 33 describes the typical functions of the different system components during the morning warm-up mode. Table 34 describes the typical functions of the different system components during the morning cool-down mode.

Table 33. Coordination of equipment during morning warm-up mode

Central air-handling unit	• Modulates the supply fan to maintain static pressure in supply duct at the desired setpoint, and shuts off when all zones have reached their occupied heating setpoints • Outdoor-air damper is closed[1] • Modulates the heating valve to discharge air at the desired setpoint[2] • Modulates the central return fan (if equipped) to maintain static pressure in the return-air plenum of the air-handling unit at the desired setpoint[3]
VAV terminal	• Opens the air-modulation damper until zone temperature reaches the occupied heating setpoint, then the damper closes[4]
Chilled-water plant	• Chilled-water pumps are off • Chillers are off • If water-cooled, condenser-water pumps and cooling tower fans are off
Hot-water plant (if included)	• Turns on hot-water pumps (if a variable-flow system, varies the speed of the pumps to maintain pressure in the hot-water piping at the desired setpoint) • Turns on boilers and varies boiler capacity to heat water to the desired setpoint

1 In some buildings, outdoor air may be brought into the building during the morning warm-up 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 central relief damper or relief fan should modulate to maintain indoor-to-outdoor static pressure difference at the desired setpoint.

2 If there is no source of heat in the central air-handling unit, morning warm-up can be accomplished with the heating coils in the VAV terminal units or with baseboard radiant heat within the zone:
- If the system includes baseboard heat within the zone, it could be used to warm up the zone prior to occupancy, and the central air-handling unit could remain off during this mode.
- If the system includes VAV reheat terminals, only the heating coil for those zones requiring morning warm-up would need to be activated.
- If the system includes fan-powered VAV terminals equipped with heating coils, only the terminal fan and heating coil for those zones requiring morning warm-up would need to be activated. The central air-handling unit could remain off.

3 If flow tracking is used, the central return fan modulates to maintain a fixed airflow differential from the supply fan airflow.

4 The air-modulation damper in the VAV terminal unit may be fully open, allowing for a fast warm-up, or it may modulate to achieve a more "controlled" warm-up and limit supply and return (if equipped) fan airflow to avoid potential motor overload.

Table 34. Coordination of equipment during morning cool-down mode

Central air-handling unit	• Modulates the supply fan to maintain static pressure in supply duct at the desired setpoint, and shuts off when all zones have reached their occupied cooling setpoints • Modulates the cooling valve to discharge air at the desired setpoint • Outdoor-air damper is closed, unless the condition of the outdoor air is suitable to provide free cooling (economizer)[1] • Modulates the central return fan (if equipped) to maintain static pressure in the return-air plenum of the air-handling unit at the desired setpoint[3]
VAV terminal	• Opens the air-modulation damper until zone temperature reaches occupied cooling setpoint, then the damper closes[2]
Chilled-water plant	• Turns on chilled-water pumps (if a variable-flow system, varies the speed of the pumps to maintain pressure in the chilled-water piping at the desired setpoint) • Turns on chillers and varies chiller capacity to cool water to the desired setpoint • If water-cooled, turns on condenser-water pumps when chillers need to operate and modulates cooling tower fans to cool condenser water to the desired setpoint
Hot-water plant (if included)	• Hot-water pumps are off • Boilers are off

1 In some buildings, outdoor air is brought into the building during the morning 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 central relief damper or relief fan should modulate to maintain indoor-to-outdoor static pressure difference at the desired setpoint.

2 The air-modulation damper in the VAV terminal unit may be fully open, allowing for a fast cool-down, or it may modulate to achieve a more "controlled" cool-down and limit supply and return (if equipped) fan airflow to avoid potential motor overload.

3 If flow tracking is used, the central return fan modulates to maintain a fixed airflow differential from the supply fan airflow.

In some climates, it may necessary to reduce the indoor humidity level prior to occupancy. This is often called "**humidity pull-down**." Similar to morning cool-down, this requires the HVAC system to start prior to occupancy, and operate long enough for the humidity inside the building to reach the desired occupied humidity setpoint by the time people enter the building. During the humidity pull-down mode, the supply-air dry-bulb temperature should be ramped down slowly to lower the indoor dew point temperature and avoid condensation of surfaces of the air distribution system.

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 simple solutions to minimize comfort complaints and avoid wasting:

- *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 138, p. 194) 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 or three hours, for example).

Using the timed override feature provides the opportunity to be more aggressive with time-of-day operating schedules. Avoid wasting energy by starting and stopping the HVAC system based on typical usage, not worst-case or once-a-year use. 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 (or a few) 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 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 to be created and maintained, group zones with similar usage patterns together and create one schedule for each group.

Chilled-water plant

For more information on the control of the chilled-water plant, refer to the Trane application manual titled *Chiller System Design and Control* (SYS-APM001-EN).

Control of the chillers, pumps, and cooling towers (in a water-cooled system) must be coordinated to provide chilled water to the VAV air-handling units when needed. The primary issues to address thorough chilled-water plant control include:

- When should the chilled-water plant be enabled or disabled?
- In a plant with multiple chillers, when a chiller must be turned on or off, which chiller should it be?
- If an attempt to turn on a chiller, pump, or cooling tower fails, or if there is a malfunction, what should be done next?
- How can the energy cost of operating the overall chilled-water system be minimized?

Hot-water plant

For more information on the control of the hot-water plant, refer to the 2008 *ASHRAE Handbook—HVAC Systems and Equipment* (Chapter 12) and *The Boiler Book* by Cleaver-Brooks (www.boilerspec.com).

Control of the boilers and pumps must be coordinated to provide hot water to the air-handling units, VAV terminals, and/or baseboard radiant heat when needed. The primary issues to address thorough hot-water plant control include:

- When should the hot-water plant be enabled or disabled?

- In a plant with multiple boilers, when a boiler must be turned on or off, which boiler should it be?
- If an attempt to turn on a boiler or pump fails, or if there is a malfunction, what should be done next?
- How can the energy cost of operating the overall hot-water system be minimized?

System optimization

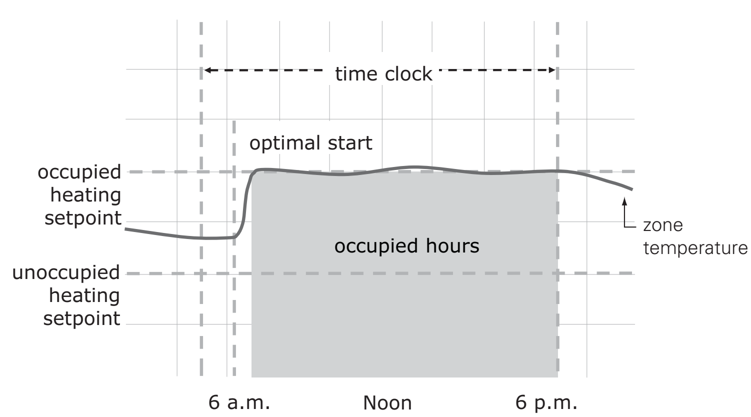
When a building automation system (BAS) provides system-level coordination of the various pieces of HVAC equipment, the next logical step is to optimize the control of that system. For this discussion, optimization is defined as minimizing the cost to operate the entire HVAC system, while still maintaining 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 that 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 BAS is used to determine the length of time required to bring each zone from its current temperature to the occupied setpoint temperature. Then, the BAS waits as long as possible before starting the system, so that the temperature in all zones reaches the occupied setpoint just in time for occupancy (Figure 140).

Figure 140. 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 ambient (outside) temperature.

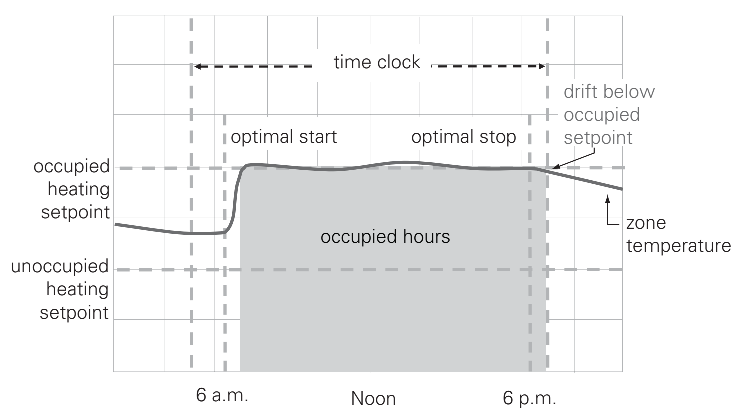
This strategy reduces the number of system operating hours and saves energy compared to maintaining the indoor temperatures at *occupied* setpoint, even though the building is still *unoccupied*.

Optimal stop

A related strategy is optimal stop. As mentioned previously, at the end of the occupied period, most systems are shut off and the indoor temperature is allowed to drift away from occupied setpoint. However, the building occupants may not mind if the indoor temperature drifts just a few degrees before they leave for the day.

Optimal stop uses the BAS to determine how early heating and cooling can be shut off for each zone, so that the indoor temperature drifts only a few degrees from occupied setpoint (Figure 141). In this case, only cooling and heating are shut off; the supply fan continues to operate and the outdoor-air damper remains open to continue ventilating the building.

Figure 141. Optimal stop



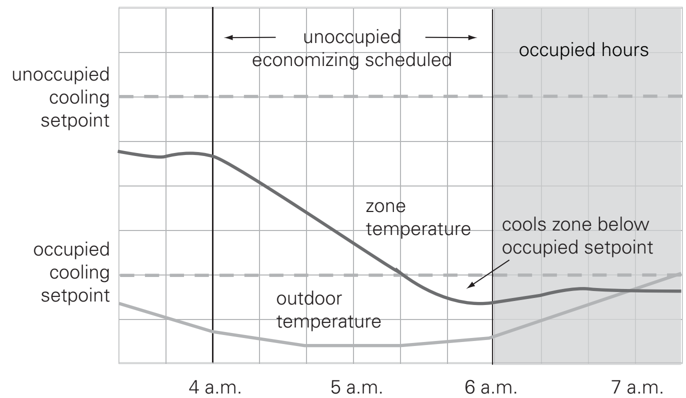
The optimal stop strategy also reduces the number of system operating hours, saving energy by allowing indoor temperatures to drift early.

Unoccupied “economizing”

During the unoccupied mode, the indoor temperature can sometimes drift up (warmer than the occupied cooling setpoint). This requires the system to enter the morning cool-down mode prior to occupancy.

However, if the outdoor dry-bulb temperature is cooler than the indoor temperature during these unoccupied hours, it is possible to use the outdoor air for “free” precooling. This strategy, often called night (or unoccupied) economizing, can save energy by precooling the building prior to the morning cool-down period (Figure 142).

Figure 142. Unoccupied economizing



Unoccupied economizing uses the BAS to determine if the condition of the outdoor air is suitable for “free cooling,” and determines which zones could benefit from precooling. Unoccupied economizing is allowed whenever the outdoor dry-bulb temperature is cooler than the current zone temperature by a defined margin, 15°F (8.3°C) for example. In addition, a humidity sensor can be added to prevent unoccupied economizing when the outdoor air is cool, but humid.

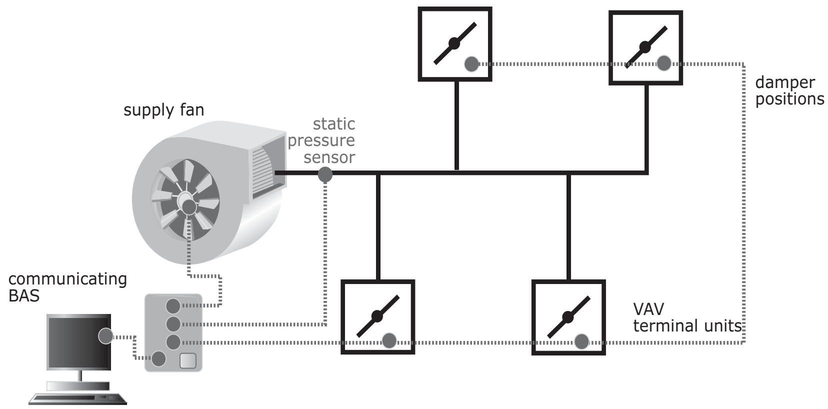
Fan-pressure optimization

For more information on using fan-pressure optimization in VAV systems, refer to the Trane *Engineers Newsletter* titled “VAV System Optimization: Critical Zone Reset” (ENEWS-20/2).

As described earlier in this chapter, the unit-level controller on the VAV air-handling unit varies the capacity of the supply fan to maintain the static pressure at some location in the supply duct at a desired setpoint.

When communicating controllers are used on the VAV terminal units, it is possible to optimize this static-pressure control function to minimize fan energy consumption. Each VAV unit controller knows the current position of its air-modulation damper. The BAS can continually poll the individual VAV terminal unit controllers, looking for the one with the most-open damper. The static-pressure setpoint is then reset so that at least one damper is nearly wide open. The result is that the supply fan generates only enough static pressure to push the required quantity of air through this “critical” (furthest open) VAV terminal unit. This concept is often called fan-pressure optimization (Figure 143).

Figure 143. Fan-pressure optimization



For VAV systems that use direct digital controls (DDC) on the VAV terminal units, fan-pressure optimization is required by ASHRAE Standard 90.1 (see "VAV fan control," p. 132).

This optimization strategy has several benefits. First, it results in less supply fan energy use at part-load conditions. A comparison of the three common fan control methods demonstrates the energy savings potential (Table 35). At full load (design airflow), all three control strategies are the same. However, at part-load conditions, fan-pressure optimization allows the supply fan to operate at a lower static pressure and, therefore, use less energy (Figure 144). For this example, the supply fan controlled using fan-pressure optimization uses only 43 percent of full-load power, compared to 55 percent with a non-optimized strategy that maintains a constant static pressure at a location two-thirds of the way down the supply duct.

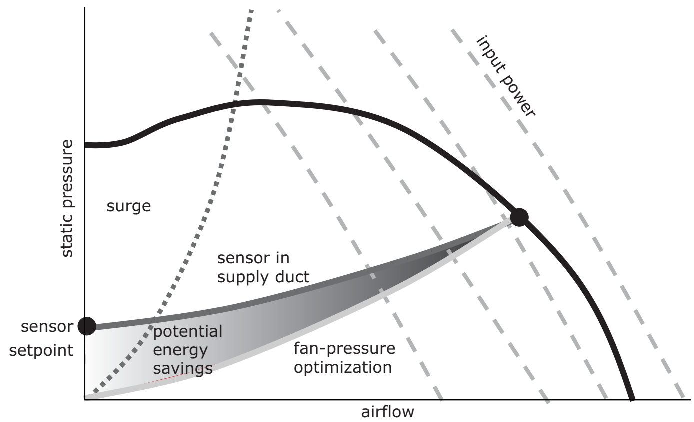
Table 35. Comparison of fan-capacity control methods

control method	airflow	fan static pressure	fan input power	% full-load power
full load	24,000 cfm (11.3 m^3/s)	2.7 in. H_2O (672 Pa)	22 hp (16.4 kW)	100%
part load				
fan outlet	18,000 cfm (8.5 m^3/s)	2.1 in. H_2O (523 Pa)	13 hp (9.7 kW)	60%
supply duct	18,000 cfm (8.5 m^3/s)	1.9 in. H_2O (473 Pa)	12 hp (8.9 kW)	55%
optimized	18,000 cfm (8.5 m^3/s)	1.5 in. H_2O (374 Pa)	9.5 hp (7.1 kW)	43%

A second benefit of fan-pressure optimization is lower noise levels. Because the supply fan does not need to generate as much static pressure, it generates less noise. In addition, with lower pressures in the supply ductwork, the dampers in the VAV terminals will be more open, resulting in less noise generated by the VAV terminals.

Another benefit is reduced risk of fan surge. This optimized control method allows the system to operate as if there is a static pressure sensor installed directly upstream of each VAV terminal unit. By keeping the worst-case damper nearly wide open at all times, the VAV system modulation curve is nearly identical to the design system resistance curve. While this results in energy savings, as previously mentioned, it also keeps the operating point of the supply fan further away from the surge region than the other fan capacity control methods (Figure 144).

Figure 144. Energy savings and reduced risk of fan surge



By making sure that every VAV terminal unit has just enough pressure to deliver the required airflow, this strategy ensures that no zones will be “starved” for air due to too little pressure in the upstream supply duct.

The fan-pressure optimization strategy is not only for the supply fan. It can also be used to control the fan in a dedicated outdoor-air system (see Figure 96, p. 124) or to control the central relief fan when floor-by-floor building pressure control is used (see Figure 133, p. 181).

Finally, fan-pressure optimization allows the static-pressure sensor to be located anywhere in the supply duct system. This presents the opportunity to have it factory-installed and -tested at the fan outlet inside the central air-handling unit. In this location, it can also serve as the duct high-pressure sensor, protecting the ductwork from damage in the event of a fire damper closing.

If the VAV terminal units include pressure-independent, communicating controls, the system-level communications are already in place, making fan-pressure optimization the lowest-cost, highest-energy-saving strategy for fan capacity control.

Impact of SAT reset on airside economizing

When the outdoor air is cooler than the supply-air-temperature setpoint, the water chillers are shut off, and the outdoor- and return-air dampers modulate to deliver the desired supply-air temperature (see “Airside economizer control,” p. 174). Raising this supply-air temperature setpoint allows the chillers to be shut off sooner and increases the number of hours of "modulated" economizer operation, when the economizer alone provides all the needed cooling.

However, increasing the supply-air temperature does increase fan energy. So the energy benefit of increasing the number of hours of "modulated" economizer operation must be weighed against the increased fan energy.

Supply-air-temperature reset

In a VAV system, it is tempting to raise the supply-air temperature (SAT) at part-load conditions in an attempt to save cooling or reheat energy. Increasing the supply-air temperature reduces cooling energy because: 1) it allows the control valve on the chilled water to modulate further closed, and 2) it improves the ability of the airside economizer to satisfy all or part of the cooling load (see sidebar), reducing the cooling load on the chiller plant. Finally, for those zones with very low cooling loads, raising the supply-air temperature increases air circulation and can avoid or minimize the use of reheat energy.

However, because the supply air is warmer, those zones that require cooling will need *more* air to offset the cooling load. This increases supply fan energy. And, warmer supply air means less dehumidification at the coil (in non-arid climates) and higher humidity levels in the zones (see “Resetting supply-air temperature,” p. 120).

Table 36 lists conditions that favor the use of SAT reset and conditions that reduce the potential to save energy using SAT reset.

Table 36. Conditions that favor, or don't favor, the use of SAT reset*

Conditions that favor using SAT reset	Conditions that reduce potential to save energy with SAT reset
• Mild climate with many hours when the outdoor dry-bulb temperature is below 70°F (21°C) • VAV terminal units that have minimum airflow settings higher than 30 percent of design airflow (increases likelihood that reheat will be needed to avoid overcooling the zones) • Efficient design of the air distribution system (low pressure loss causes less of a penalty for the higher airflows that result from raising SAT) • Systems with interior zones that have varying cooling loads (allowing them to be satisfied with a warmer supply-air temperature)	• Systems in which part-load dehumidification is a concern • Hot climate with few hours when the outdoor dry-bulb temperature is below 60°F (16°C) • Inefficient design of the air distribution system (high pressure loss increases penalty for the higher airflows that result from raising SAT) • Systems serving some zones that have nearly constant cooling loads (consider using a separate system for such zones, such as computer centers, so they do not force the system to operate at a colder supply-air temperature) • Efficient part-load fan modulation, such as a variable-speed drive

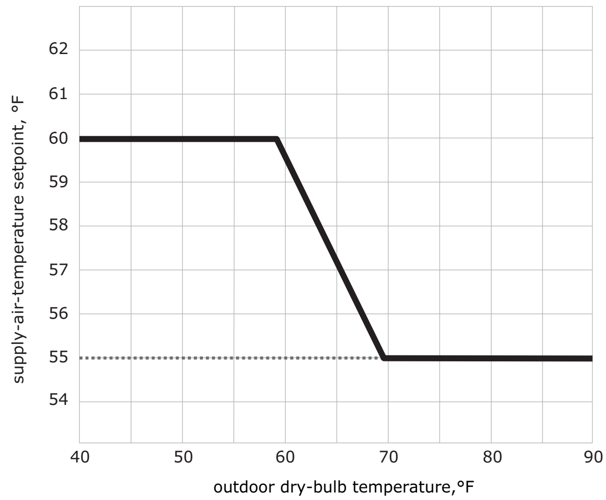
*This table is an excerpt from the California Energy Commission's *Advanced VAV System Design Guide*.

The SAT reset strategy should be designed to minimize overall system energy use, considering the trade–off between cooling, reheat, and supply fan energy. It should also minimize the negative impact on zone humidity levels. These competing issues are often best balanced by first reducing fan airflow, taking advantage of the significant energy savings from unloading the fan. Once fan airflow has been reduced somewhat, raise the supply-air temperature to minimize reheat energy and enhance the benefit of the airside economizer.

To accomplish this, the SAT setpoint is typically reset based on either 1) the changing outdoor temperature, or 2) by monitoring the cooling needs of the zones served by the air-handling unit.

Figure 145 shows an example of a SAT reset strategy based on the changing outdoor dry-bulb temperature. When the outdoor temperature is warmer than 70°F (21°C), no reset takes place and the SAT setpoint remains at the design value of 55°F (13°C). When it is this warm outside, the outdoor air provides little or no cooling benefit for economizing, and the cooling load in most zones is likely high enough that reheat is not required to prevent overcooling. In this region, the fan energy savings likely outweighs the impact on cooling energy. In addition, the colder (and drier) supply air allows the system to provide sufficiently dry air to the zones, improving part-load dehumidification.

Figure 145. Example SAT reset based on outdoor temperature



When the outdoor temperature is between 60°F and 70°F (16°C and 21°C), the supply-air temperature setpoint is reset at a 2-to-1 ratio (Figure 145). That is, for every 2°F (1.1°C) change in outdoor temperature, the setpoint is reset 1°F (0.6°C). In this range, SAT reset likely enhances the benefit provided by the economizer, and it is possible that some zone-level reheat can be avoided. Plus, the supply fan has likely been unloaded significantly already, so the increased airflow has less impact on fan energy.

Finally, when the outdoor temperature is colder than 60°F (16°C), no further reset occurs, and the SAT setpoint remains at 60°F (16°C). Limiting the amount of reset to 60°F (16°C) allows the system to satisfy the cooling loads in interior zones without needing to substantially oversize the VAV terminal units and ductwork serving those zones.

Alternatively, some systems reset the SAT setpoint based on the temperature in the zone that is most nearly at risk of overcooling, which would require activating local reheat. A building automation system monitors the temperature in all zones, finding the zone that is closest to heating setpoint temperature. Then, the SAT setpoint is reset to prevent this coldest zone from needing to activate reheat.

When considering using SAT reset in a VAV system:

- Analyze the system to determine if the savings in cooling and reheat energy will outweigh the increase in supply fan energy.
- If higher zone humidity levels are a concern, consider either 1) providing an outdoor dew point sensor to disable reset when it is humid outside, or 2) providing one or more zone humidity sensors to override the reset function if humidity in the zone exceeds a maximum limit.
- For interior zones that will likely have nearly constant cooling loads during occupied periods, calculate the design supply airflows for those zones based on a warmer-than-design supply-air temperature (60°F [16°C] rather than 55°F [13°C], using the example in Figure 145). This allows SAT

reset to be used during cooler weather, while still offsetting the cooling loads in the interior zones.

- Design the air distribution system for low pressure losses and use the fan-pressure optimization strategy (p. 200) to minimize the penalty of increased fan energy when the SAT is increased.

Ventilation optimization

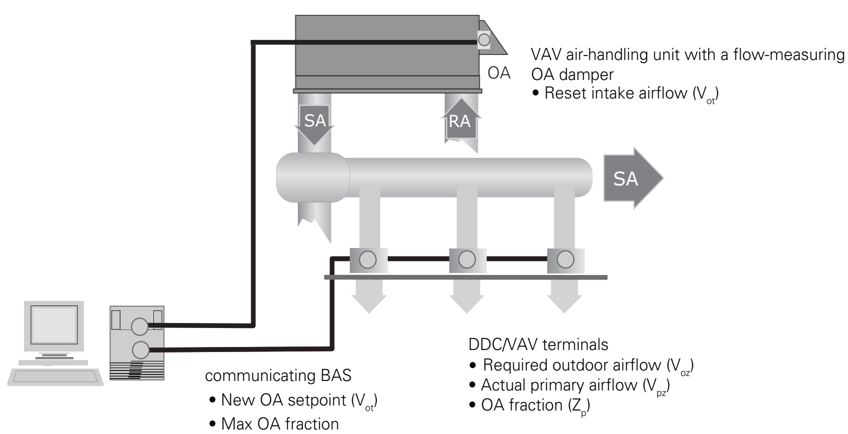
For more information on combining zone-level demand-controlled ventilation with system-level ventilation reset, refer to the Trane *Engineers Newsletter* titled "CO_2-Based Demand-Controlled Ventilation with ASHRAE Standard 62.1-2004" (ADM-APN017-EN) and the Trane *Engineers Newsletter Live* broadcast DVD titled "CO_2-Based Demand-Controlled Ventilation" (APP-CMC024-EN).

The ventilation control function of the zone-level VAV terminal units was discussed earlier in this chapter (see "Ventilation control," p. 184). In a multiple-zone VAV system, "zone-level" ventilation control must be coordinated to determine how to control the outdoor-air damper in the "system-level" air-handling unit. As mentioned in "Dynamic reset of intake airflow," p. 117, 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 whenever a zone is occupied.

- *Ventilation Reset: Resetting intake airflow in response to variations in ventilation efficiency*
  The system ventilation efficiency (E_v) of a multiple-zone VAV system changes as both zone airflows and system-level primary airflow change due to variations in building load. This "ventilation reset" strategy dynamically resets the system outdoor-air intake (V_{ot}) based on this changing efficiency.

  Implementing this strategy requires a communicating controller on each VAV terminal unit, a building automation system that can gather data from all the VAV controllers, and a method to measure and control outdoor airflow at the central air-handling unit (Figure 146). This is typically accomplished using a flow-measuring device in the outdoor air stream (Figure 88, p. 115).

Figure 146. Ventilation reset in a chilled-water VAV system



An energy-saving enhancement to the ventilation reset control strategy is for the BAS to enforce a maximum OA fraction (Figure 146), and increase the primary airflow to the "critical zone" (zone with the highest OA fraction) at low-load conditions. While this will activate reheat to avoid overcooling the zone, when it is hot or cold outside, the energy saved by avoiding the need to increase system-level intake airflow will likely outweigh the small amount of reheat needed for this one zone (or few zones).

The controller on each VAV terminal continuously monitors primary airflow (V_{pz}) being delivered to the zone. The DDC controller also knows the outdoor airflow required by the zone (V_{oz}). The BAS periodically gathers this data from all the VAV terminal units and dynamically solves the equations prescribed by ASHRAE Standard 62.1 to calculate the quantity of outdoor air that must be brought in through the system-level outdoor-air intake (V_{ot}) in order to satisfy the ventilation requirements of the individual zones [see "Calculating system intake airflow (V_{ot})," p. 105]. The BAS then sends this minimum outdoor airflow setpoint to the air-handling unit controller, which modulates the flow-measuring outdoor-air damper to maintain this new setpoint.

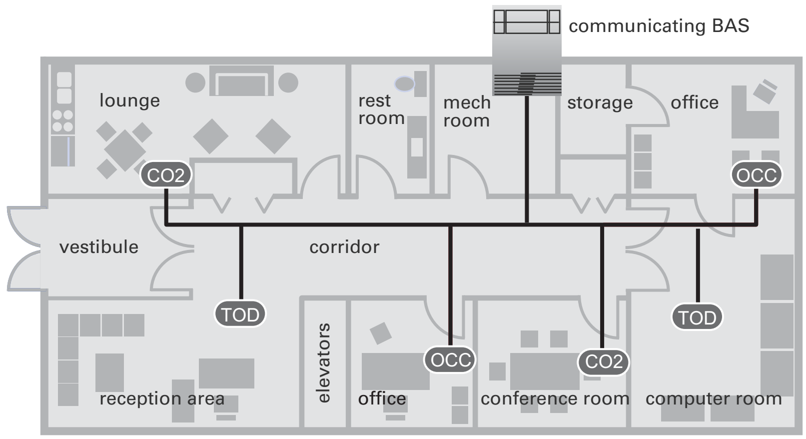
Ventilation reset ensures that all zones are properly ventilated at all operating conditions, while minimizing energy wasted by overventilating. In a DDC/VAV system, this strategy is fairly easy to implement because all of the necessary real-time information is already available digitally, so no new sensors are required. All of the equations are defined in an industry-wide standard (ASHRAE 62.1), and can be solved dynamically to calculate the outdoor-air intake flow required under the current operating conditions.

- *Demand-Controlled Ventilation: 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. Demand-controlled ventilation (DCV) is a zone-level control strategy that dynamically resets the outdoor airflow requirement for the zone based on changing population in that zone.

  As mentioned earlier in this chapter (see "Ventilation control," p. 184), time-of-day (TOD) schedules, occupancy sensors (OCC), and carbon dioxide (CO_2) sensors are commonly used to implement demand-controlled ventilation. A time-of-day schedule can be created in the BAS to indicate to the VAV terminal unit when each zone is occupied versus unoccupied, or to predict the actual number of people in a zone for any given hour. An occupancy sensor, such as a motion detector, can be used to detect the presence of people in a zone. A CO_2 sensor is used to monitor the concentration of carbon dioxide in the zone, which can be used as an indicator of the current per-person ventilation rate (cfm/person [m^3/s/person]).

In most VAV systems, the best approach often combines all three DCV approaches (TOD schedules, occupancy sensors, and CO_2 sensors) at the zone level, using each where it best fits, with ventilation reset at the system level (Figure 147 and Figure 148).

Figure 147. Ventilation optimization: DCV at the zone level



CO2 sensors increase both the installed cost of the system and the risk. These sensors need to be maintained and calibrated (or periodically replaced) in order 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.

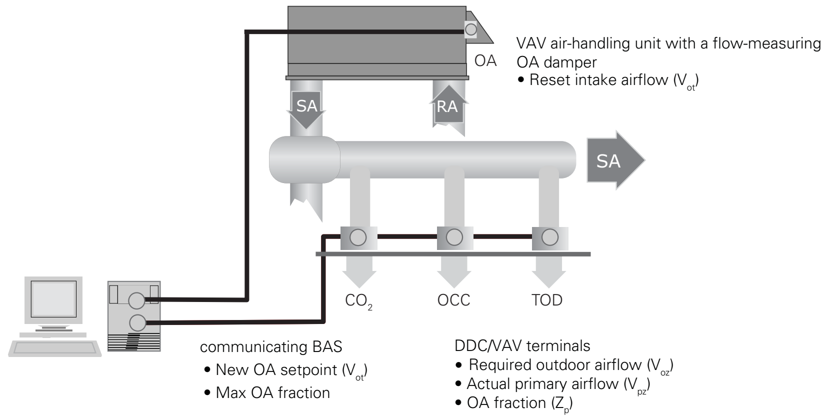
With this approach, CO_2 sensors are installed only in those zones that are densely occupied and experience widely varying patterns of occupancy (such as conference rooms, auditoriums, and gymnasiums). These zones are the best candidates for CO_2 sensors, and provide "the biggest bang for the buck." These sensors are used to reset the ventilation requirement for their respective zones (V_{oz}) based on measured CO_2.

Zones that are less-densely occupied (such as open plan office spaces) or do not experience significant variations in occupancy (such as private offices and many classrooms) are probably better suited for occupancy sensors. When the sensor indicates that the zone is unoccupied, the controller lowers the ventilation requirement for the zone.

Finally, zones that are sparsely populated or have predictable occupancy patterns may be best controlled using a time-of-day schedule. This schedule can either indicate when the zone will normally be occupied versus unoccupied, or can be used to vary the zone ventilation requirement based on anticipated population.

These various zone-level DCV strategies can be used to reset the ventilation requirement (V_{oz}) for their respective zones for current conditions. The BAS periodically gathers this data from all the VAV terminal units and uses the ventilation reset equations to determine how much outdoor air must be brought in at the central air-handling unit to satisfy all of the zones served (Figure 148).

Figure 148. Ventilation optimization: Ventilation reset at the system level



The ventilation optimization strategy can be used to earn the "Outdoor Air Delivery Monitoring" credit (Indoor Environmental Quality section) of LEED 2009, because it involves installing a CO_2 sensor in each densely occupied space and an outdoor airflow measuring device to measure intake airflow at the air-handling unit.

Besides being a cost-effective, reliable, and energy-efficient approach to ventilating a VAV system, combining zone-level DCV with system-level ventilation reset 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, avoids the periodic calibration and cleaning required to ensure proper sensor operation, and minimizes risk (see sidebar). For the other zones, occupancy sensors and/or time-of-day schedules are used to reduce ventilation.

Pump-pressure optimization

When variable-flow pumping is used in either the chilled-water or hot-water distribution system, some method is needed to control the pump capacity. Similar to the supply fan in a VAV system, the pump is often controlled to maintain a set pressure differential at some location in the piping system. In some systems, a differential pressure transducer is located between the inlet and the discharge of the pump, while in others it is located at the most distant point in the piping system.

An addendum (ak) to ASHRAE Standard 90.1-2007 requires pump-pressure optimization if differential pressure control is used to control a variable-speed pump and DDC controls are used.

Similar to the fan-pressure optimization strategy discussed previously (p. 200), it is possible to optimize this pressure control function to minimize pump energy consumption, if each air-handling unit or VAV terminal unit uses a communicating controller that knows the current position of its modulating valve. The BAS continually polls the individual controllers, looking for the valve that is the furthest open. The pump-pressure setpoint is then periodically reset so that at least one valve is nearly wide open. The result is that the pump generates only enough pressure to push the required quantity of water through this "critical" (furthest open) water valve. This concept is often called pump-pressure optimization or critical-valve reset.

This optimization strategy results in less pump energy use by allowing the pump to operate at a lower pressure at part-load conditions.

Chilled-water temperature reset

Raising the temperature of the water leaving the chiller decreases the energy used by the chiller. In a constant-flow pumping system, this will typically decrease overall system energy use, as long as control of indoor humidity is not compromised. As chilled-water temperature is increased, the temperature of the air leaving the cooling coil may increase above the desired setpoint, and indoor humidity levels can increase (see "Resetting supply-air temperature," p. 120).

In a variable-flow pumping system, however, raising the chilled-water temperature will often increase overall system energy use. Because the water is warmer, those VAV air-handling units that require cooling will need *more* water to offset the cooling load. This increases pumping energy.

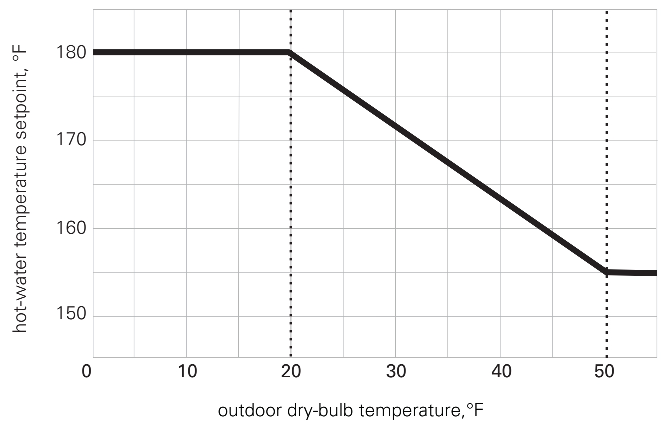
Hot-water temperature reset

Lowering the temperature of the water leaving the boiler decreases the energy used by the boiler. In a constant-flow pumping system, this will typically decrease overall system energy use. In a variable-flow pumping system, however, lowering the hot-water temperature will often increase overall system energy use. Because the water is not as hot, the heating coils will need *more* water to offset the heating load. This increases pumping energy.

Hot-water heating coils in VAV terminal units may be used for two purposes: 1) to provide *heated* supply air to offset heating loads in the zone, and 2) to *reheat* the cool supply air to avoid overcooling the zone at low cooling loads. When the zone requires heating, the hot-water coil heats the supply-air to a temperature that is significantly warmer than the zone temperature. When reheating, however, the coil warms the supply air to a temperature that is no higher than the zone temperature. The water temperature required for *reheating* can often be much lower than the temperature required for *heating*, and can be a good application for a hot-water temperature reset control strategy to save boiler energy. (For an example, see "Condenser heat recovery," p. 88.)

Figure 149 shows an example of a hot-water temperature reset strategy based on the changing outdoor dry-bulb temperature. When the outdoor temperature is colder than 20°F (-7°C), no reset takes place and the hot-water temperature setpoint remains at the design value of 180°F (82°C). When it is this cold outside, it is likely that many of the zones require the hotter water temperature for space heating.

Figure 149. Example hot-water temperature reset based on outdoor temperature



When the outdoor temperature is between 20°F and 50°F (-7°C and 10°C), the hot-water temperature setpoint is reset at a 1-to-1 ratio (Figure 149). That is, for every 1°F (0.6°C) change in outdoor temperature, the setpoint is reset 1°F (0.6°C). In this range, few zones are likely to require space heating, but are more likely to require reheat to avoid overcooling at minimum primary airflow. In this case, the heating coils in VAV terminal units can likely provide sufficient reheat with a lower water temperature, so hot-water temperature reset can be used to save boiler energy.

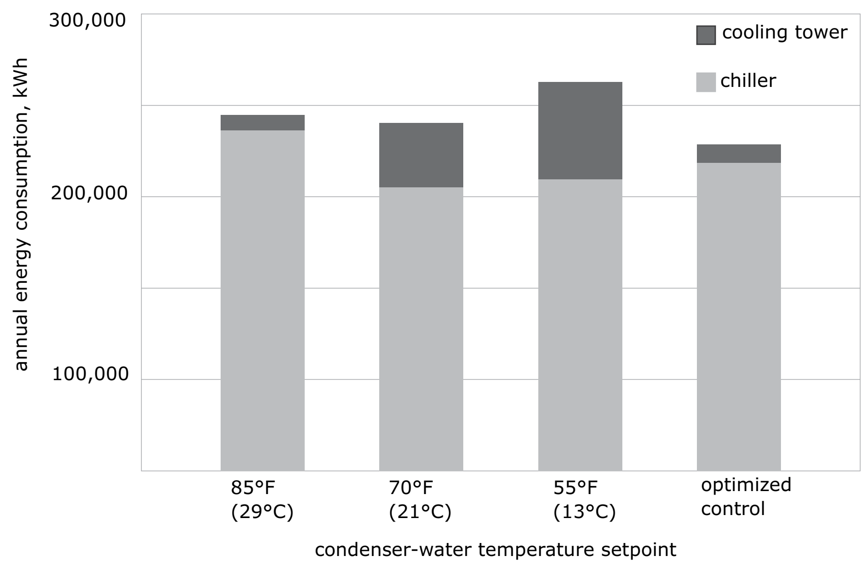
Finally, when the outdoor temperature is warmer than 50°F (10°C), no further reset occurs, and the hot-water temperature setpoint remains at 150°F (66°C).

Condenser-water temperature (chiller-tower) optimization

Depending on the system cooling load and outdoor conditions, cooling towers may have the ability to supply condenser water at a colder temperature than at design conditions. Lowering the temperature of the water leaving the cooling tower decreases the energy used by the chiller, but increases the energy used by the cooling-tower fans. The key to maximizing energy savings is knowing the relationship between cooling tower energy use and chiller energy use.

At design conditions, a chiller typically uses five to ten times more energy than a cooling tower. This would suggest that it might be beneficial to use more cooling-tower energy to save chiller energy. However, there is a point of diminishing return where the chiller energy savings is less than the additional energy used by the cooling tower. Figure 150 shows the combined annual energy consumption of a chiller and cooling tower in a system that is controlled to various condenser-water temperature fixed setpoints. (At near-design conditions, the cooling tower may not be able to supply the temperature requested, but it will supply the water at the coldest temperature possible.)

Figure 150. Optimized control of condenser-water temperature



The fourth column shows a system that uses the system-level controller to dynamically determine the optimal condenser-water temperature that minimizes the combined energy use of the chiller plus cooling tower fans.

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 on and off the HVAC system could also be used to turn on and off lights 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. 192), 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 ventilation requirement for that zone when it is unoccupied.
- A card access security system could be used to turn on lights, start the HVAC system, and increase ventilation 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 ventilation delivered to an individual theater based on the number of people that purchased tickets for the show.

This integration may also involve changing the operation of one or more components of the HVAC system to assist another system. For example, providing effective smoke control in larger buildings may involve enlisting the help of the fans in the air-handling unit. In this case, activation of a fire alarm causes the components of the HVAC system to perform smoke-control functions. While other smoke-control functions can be conceived, four common functions include:

- *Off*
  In this mode, both the supply fan and exhaust (or return) fan are turned off. In addition, both the outdoor- and exhaust-air dampers are closed, and cooling and heating are disabled.
- *Pressurize*
  In this mode, the outdoor-air damper is wide open, the supply fan is turned on and operates at 100 percent airflow, the VAV terminal units are fully open, and the exhaust (or return) fan is turned off. With only the supply fan operating (exhaust or return fan is off), the zones served by the central air-handling unit will become pressurized. This mode might be used to prevent smoke from migrating in from other areas of the facility during a fire.
- *Depressurize (exhaust)*
  In this mode, the outdoor-air damper is closed, the supply fan is turned off, cooling and heating are disabled, and the exhaust (or return) fan is turned on and operates at 100 percent airflow. With only the exhaust (or return) fan operating (supply fan is off), the zones served by the central air-handling unit will become depressurized. This mode might be used to clear the area of smoke from a recently extinguished fire or possibly to prevent smoke from migrating into other areas of the facility during a fire.
- *Purge*
  In this mode, the outdoor-air damper is wide open, the supply fan is turned on and operates at 100 percent airflow, the VAV terminal units are fully open, and the exhaust (or return) fan is turned on and operates at 100 percent airflow. This mode might be used to purge the smoke or stale air out of a building after a fire has been extinguished or to prevent smoke from migrating into other areas of the building.

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 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-mixing baffles. A device, located immediately downstream of the mixing box, that adds rotational energy and increases the velocity of the air stream, improving the mixing (blending) and minimizing temperature stratification.

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 airflow. Calculated by finding the single instance in time when the sum of the zone airflows is the greatest. This method is used for sizing a VAV supply fan that delivers a varying amount of air to the system.

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.

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.

CDQ™. Trane's Cool, Dry, Quiet technology. See series desiccant wheel.

chiller-tower optimization. A control strategy that uses the BAS to dynamically determine the optimal condenser-water temperature that minimizes the combined energy use of the water chiller plus cooling tower fans.

Coanda effect. Concept behind the operation of a linear slot diffuser. Air is discharged at a relatively high velocity along the surface of the ceiling, creating an area of low pressure that causes the supply air to hug the ceiling. As it travels along the ceiling, air from the occupied space is drawn into, and mixed with, the supply air stream. When the air settles to the occupied levels of the space, it has assumed an average temperature.

collection efficiency. Describes how well a particulate filter removes particles of various sizes from the air stream.

collision velocity. The speed at which moving air meets a wall or another air stream. When two air streams collide, the collision velocity is determined by adding the velocities of the two air streams at the point of collision.

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.

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-only terminal unit. The simplest type of single-duct VAV terminal unit. It has the capability of varying the airflow, but has no method to provide heating for the zone.

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.

critical valve reset. See pump-pressure optimization.

critical zone reset. See fan-pressure optimization.

Cv. Flow coefficient. Term used for the selection of fluid control valves (in I-P units).

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.

displacement ventilation. See thermal displacement ventilation.

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.

dual-duct terminal unit. A type of VAV terminal unit consisting of two airflow modulation devices, one for cool primary air and one for warm primary air. These units can be controlled to provide either a constant volume or variable volume of supply air to the zone.

dual-fan, dual-duct VAV system. A VAV system that consists of two air handlers, one that delivers cool primary air and one that delivers warm primary air. These two duct systems provide air to a dual-duct VAV terminal unit for each zone.

dust spot efficiency. A rating value, defined by ASHRAE Standard 52.1, that depicts the amount of atmospheric dust a filter captures.

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.

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.

face velocity. Velocity of the air as it passes through a device (airflow rate divided by the face area of the device).

fan array. A configuration that uses multiple fans, stacked to allow for parallel air paths.

fan outlet static pressure control. A method of VAV system static pressure control that mounts the static pressure sensor near the outlet of the main supply fan and maintains a constant static pressure at the sensor.

fan performance curve. A plot of a specific fan's airflow capacity at a given speed (rpm) versus the static pressure it generates.

fan-powered terminal unit. A type of single-duct VAV terminal unit that can provide heating to a zone by mixing warm plenum air with the cool primary air, using a small terminal fan.

fan-pressure optimization. An optimized method of VAV system static-pressure control, which uses the benefits of DDC control to continuously reset the static-pressure setpoint of the system so the VAV terminal requiring the highest inlet pressure is nearly fully open.

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.

flow tracking. Method of return-fan capacity control in which the return fan is controlled based on a fixed airflow differential from supply fan airflow.

flue gases. Exhaust gases from a boiler or gas-fired burner.

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.

HEPA. High-efficiency particulate air filter.

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).

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.

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.

Kv. Flow factor. Term used for the selection of fluid control valves (in SI units).

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-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.

nighttime economizing. See unoccupied economizing.

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, the outdoor airflow required can be reduced (typically to the building-related ventilation rate, R_a), and the minimum primary airflow setting of the VAV terminal can be lowered.

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.

optimal stop. An optimized system shutdown routine that determines how early heating and cooling can be shut off for each zone, so that the indoor temperature drifts only a few degrees from occupied setpoint by the end of the scheduled occupied period.

outdoor air. Air brought into the building from outside, either by a ventilation system or through openings provided for natural ventilation.

parallel fan-powered terminal unit. A fan-powered VAV terminal unit consisting of a primary airflow modulation device and a small, integral constant-volume fan packaged to provide parallel airflow paths.

PCO. Photocatalytic oxidation. An air cleaning technology that uses ultraviolet light shining on the surface of a catalyst to adsorb gaseous contaminants (such as VOCs) and inactivate biological contaminants.

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.

plenum fan. A fan assembly consisting of a single inlet impeller, mounted perpendicular to airflow, which pressurizes a plenum chamber in the air distribution system.

population diversity. The ratio of the actual system population to the sum of the peak zone populations.

pressure-dependent. VAV control method that uses the zone temperature sensor to directly control the position of the air-modulation damper. The actual airflow delivered to the zone is a by-product of this position and depends on the static pressure inside the duct at the inlet to the terminal unit.

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.

primary air. Conditioned air delivered by a central supply fan to a terminal unit.

psychrometric chart. A tool used to graphically display the properties of moist air.

pump-pressure optimization. An optimized method of variable-flow pump control, which uses the benefits of DDC control to continuously reset the pressure setpoint of the system so the hot-water control valve requiring the highest inlet pressure is nearly fully open.

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.

return air. Air that is removed from the conditioned space(s) and either recirculated or exhausted.

return-air grille. See grille.

riding the fan curve. A method of fan capacity modulation that involves no direct form of control, but simply allows the fan to react to the change in system static pressure and ride up and down it's performance curve.

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 fan-powered terminal unit. A fan-powered VAV terminal unit consisting of a primary airflow modulation device and a small, integral constant-volume fan packaged so that the airflow paths are in series. Provides a constant volume of supply air to the zone when operating.

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)

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.

static regain duct design method. Method of designing an air duct system that strives to maintain a fairly consistent static pressure throughout the entire duct. Recommended for sizing the supply ducts upstream of the terminal units in a VAV system. The design of a static regain duct system often requires the use of a computer program.

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-air-temperature reset. A control strategy that raises the temperature of the primary air leaving the rooftop unit at part load, in an effort to reduce overall system energy use.

supply duct static pressure control. Method of VAV system static pressure control that mounts the static pressure sensor somewhere in the supply duct system, allowing the supply fan to back down and lower the static pressure in the system under part-load conditions.

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.

surge. A condition of unstable fan operation where the air alternately flows backward and forward through the fan wheel, generating noise and vibration.

system resistance curve. A plot of the static pressure drop that the system (including the supply ductwork, duct fittings, terminal units, diffusers and grilles, coils, filters, dampers, etc.) creates over a range of airflows.

system ventilation efficiency (Ev). A measure of how efficiently the system distributes air from the outdoor-air intake to the breathing zone.

TAB. Test, adjust, and balance.

tempering. The process of adding sensible heat to the supply air at the VAV terminal unit to avoid overcooling the zone.

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.

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.

TR. Cold-spot thermal resistance ratio. A measure of thermal resistance of the air-handling unit casing.

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.

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 economizing. An energy-saving control strategy that attempts to precool the building, using cool outdoor air, prior to the morning cool-down period.

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.

UV-C. Ultraviolet radiation in the "C" wavelength band.

variable-air-volume (VAV) system. A type of air-conditioning system that varies the volume of constant-temperature air supplied to meet the changing load conditions of the zone.

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.

VAV box. See VAV terminal unit.

VAV reheat terminal unit. A type of single-duct VAV terminal unit that can provide heating using a small heating coil.

VAV system modulation curve. A curve that illustrates the VAV system fan static pressure requirement over the range of airflows.

VAV terminal unit. A sheet metal assembly used to vary the quantity of supply air delivered to the conditioned space.

ventilation. The intentional introduction of outdoor air into a zone through the use of the HVAC system in the building.

ventilation optimization. An optimized control strategy that combines various demand-controlled ventilation approaches (time-of-day schedules, occupancy sensors, and CO_2 sensors) at the zone level, with ventilation reset at the system level.

ventilation reset. A control strategy that attempts to dynamically reset the system outdoor-air intake based on changing system ventilation efficiency.

VOC. Volatile organic compound.

warm-up mode. See morning warm-up mode.

water chiller. A refrigerating machine used to transfer heat between fluids.

waterside economizer. A method of free cooling that diverts cool condenser water through a separate heat exchanger to precool the water returning to the water chillers.

Winterizer. An air-handling unit configuration that uses a combination of two different-sized units configured to allow the outdoor air to be introduced downstream of the cooling coil whenever the outdoor air is colder than 32°F (0°C), protecting the chilled-water cooling coil from freezing.

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.

References

Air-Conditioning and Refrigeration Institute (ARI). 2008. Air Terminals, ARI Standard 880-2008. Arlington, VA: ARI.

______. 2008. Method of Measuring Machinery Sound Within an Equipment Space, ARI Standard 575-2008. Arlington, VA: ARI.

______. 2001. Sound Rating of Ducted Air Moving and Conditioning Equipment, ARI Standard 260-2001. Arlington, VA: ARI.

______. 2001. Sound Rating of Large Outdoor Refrigerating and Air-Conditioning Equipment, ARI Standard 370-2001. Arlington, VA: ARI.

American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc. (ASHRAE). 2007. ASHRAE Handbook-Applications. Atlanta, GA: ASHRAE.

______. 2005. ASHRAE Handbook-Fundamentals. Atlanta, GA: ASHRAE.

______. 2008. ASHRAE Handbook-HVAC Systems and Equipment. Atlanta, GA: ASHRAE.

______. 1996. Cold Air Distribution System Design Guide. Atlanta, GA: ASHRAE.

______. 2002. Designer's Guide to Ceiling-Based Air Diffusion. Atlanta, GA: ASHRAE.

______. 1992. Gravimetric and Dust-Spot Procedures for Testing Air-Cleaning Devices Used in General Ventilation for Removing Particulate Matter, ASHRAE Standard 52.1-1992. Atlanta, GA: ASHRAE.

______. 2001. Humidity Control Design Guide for Commercial and Institutional Buildings. Atlanta, GA: ASHRAE.

______. 2007. Method of Testing General Ventilation Air-Cleaning Devices for Removal Efficiency by Particle Size, ASHRAE Standard 52.2-2007. Atlanta, GA: ASHRAE.

______. 2003. Selecting Outdoor, Return, and Relief Dampers for Air-Side Economizer Systems, ASHRAE Guideline 16-2003. Atlanta, GA: ASHRAE.

______. 2007. Standard 62.1-2007 User's Manual. Atlanta, GA: ASHRAE.

______. 2007. Ventilation for Acceptable Indoor Air Quality, ASHRAE Standard 62.1-2007. Atlanta, GA: ASHRAE.

References

ASHRAE and Illuminating Engineering Society of North America (IESNA). 2007. Energy Standard for Buildings Except Low-Rise Residential Buildings, BSR/ASHRAE/IESNA Standard 90.1-2007. Atlanta, GA: ASHRAE.

______. 2008. Standard 90.1-2007 User's Manual. Atlanta, GA: ASHRAE.

California Energy Commission. 2003. Advanced Variable Air Volume System Design Guide, 500-03-082-A-11 (October). http://www.energy.ca.gov/reports/2003-11-17_500-03-082_A-11.PDF

Callan, D., R. Bolin, and L. Molinini. 2004. "Optimization of air-handling systems for federal buildings," Engineered Systems (April): pp. 39-50.

Cleaver-Brooks. 2009. The Boiler Book. http://www.boilerspec.com

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

Institute of Electrical and Electronics Engineers (IEEE). 2006. Wireless Medium Access Control and Physical Layer Specifications for Low-Rate Wireless Personal Area Networks. IEEE Standard 802.15.4-2006. New York, NY: IEEE.

Institute of Environmental Sciences and Technology (IEST). 2006. HEPA and ULPA Filters. IEST-RP-CC001.4. Mt. Prospect, IL: IEST.

National Air Filtration Association (NAFA). 2007. NAFA Guide to Air Filtration, 4th Edition. Virginia Beach, VA: NAFA.

New Buildings Institute (NBI). 1998. Gas Boilers Advanced Design Guideline. Fair Oaks, CA: NBI. http://www.newbuildings.org

Sheet Metal and Air Conditioning Contractors National Association (SMACNA). 2006. HVAC Systems - Duct Design. Chantilly, VA: SMACNA.

Stanke, D. 2004. "Addendum 62n: Single-zone & Dedicated-OA Systems", ASHRAE Journal (October): pp. 12-20.

Stanke, D. 2005. "Addendum 62n: Single-Path Multiple-Zone System Design", ASHRAE Journal (January): pp. 28-35.

Stanke, D. 2005. "Standard 62.1-2004: Designing Dual-Path, Multiple-Zone Systems", ASHRAE Journal (May): pp. 20-30.

Stanke, D. 2006. "Standard 62.1-2004 System Operation: Dynamic Reset Options", ASHRAE Journal (December): pp. 18-32.

Trane. 2008. Using Ultraviolet Light to Control Microbial Growth in Buildings, CLCH-PRB014-EN. La Crosse, WI: Inland Printing Company.

______. 2009. Trane Catalytic Air Cleaning System, CLCH-PRB023-EN. La Crosse, WI: Inland Printing Company.

______. 2009. Direct-Drive Plenum Fans For Trane Climate Changer™ Air Handlers, CLCH-PRB021-EN. La Crosse, WI: Inland Printing Company.

______. 2004. Trane CDQ™ Desiccant Dehumidification, CLCH-PRB020-EN. La Crosse, WI: Inland Printing Company.

______. Bradway, B., Hallstrom, A., Stanke, D. and Bailey, N. 1998. Managing Building Moisture, SYS-AM-15. La Crosse, WI: Inland Printing Company.

______. Guckelberger, D. and Bradley, B. 2006. Acoustics in Air Conditioning, ISS-APM001-EN. La Crosse, WI: Inland Printing Company.

______. Murphy, J. and Bradley, B. 2002. Air-to-Air Energy Recovery in HVAC Systems, SYS-APM003-EN. La Crosse, WI: Inland Printing Company.

______. Murphy, J. and Bradley, B. 2002. Dehumidification in HVAC Systems, SYS-APM004-EN. La Crosse, WI: Inland Printing Company.

______. 1981. Variable-Air-Volume Duct Design, AM-SYS-6. La Crosse, WI: Inland Printing Company.

______. Schwedler, M. and Brunsvold, D. 1999. Absorption Chiller System Design, SYS-AM-13. La Crosse, WI: Inland Printing Company.

______. Hanson, S., Schwedler, M., and Bakkum, B. 2009. Chiller System Design and Control, SYS-APM001-EN. La Crosse, WI: Inland Printing Company.

______. Schwedler, M. and Brunsvold, D. 2003. Waterside Heat Recovery in HVAC Systems, SYS-APM005-EN. La Crosse, WI: Inland Printing Company.

______. 1987. Ice Storage Systems, SYS-AM-10. La Crosse, WI: Inland Printing Company.

______. 1988. Control of Ice Storage Systems, ICS-AM-4. La Crosse, WI: Inland Printing Company.

Trane. 2004. Air Conditioning Fans, TRG-TRC013-EN. Air Conditioning Clinic series. La Crosse, WI: Inland Printing Company.

_____. 2000. Cooling and Heating Load Estimation, TRG-TRC002-EN.

_____. 2001. Chilled-Water Systems, TRG-TRC016-EN.

_____. 2001. Fundamentals of HVAC Acoustics, TRG-TRC007-EN.

_____. 2001. VAV Systems, TRG-TRC014-EN.

_____. 2006. Ice Storage Systems, TRG-TRC019-EN.

Trane. Eppelheimer, D. and Bradley, B. 2000. "Cold Air Makes Good $ense." Trane Engineers Newsletter, 29-2.

______. Guckelberger, D. and Bradley, B. 2004. "Brushless DC Motors: Setting a New Standard for Efficiency." Trane Engineers Newsletter, 33-4.

______. Hsieh, C. and Bradley, B. 2003. "Green, Growing, Here to Stay: Energy and Environmental Initiatives." Trane Engineers Newsletter, 32-3.

______. Murphy, J. and Bradley, B. 2002. "Using CO_2 for Demand-Controlled Ventilation." Trane Engineers Newsletter, 31-3.

______. Murphy, J. and Bradley, B. 2005. "CO_2-Based Demand-Controlled Ventilation with ASHRAE Standard 62.1-2004." Trane Engineers Newsletter, 34-5.

______. Murphy, J. and Bradley, B. 2005. "Advances in Desiccant-Based Dehumidification." Trane Engineers Newsletter, 34-4.

______. Guckelberger, D. and Bradley, B. 2000. "Sound Ratings and ARI Standard 260." Trane Engineers Newsletter, 29-1.

______. Stanke, D. and Bradley, B. 1991. "VAV System Optimization: Critical Zone Reset." Trane Engineers Newsletter, 20-2.

______. Stanke, D. and Bradley, B. 2002. "Managing the Ins and Outs of Building Pressurization." Trane Engineers Newsletter, 31-2.

______. Stanke, D. and Bradley, B. 2006. "Keeping Cool with Outdoor Air: Airside Economizers." Trane Engineers Newsletter, 35-2.

______. Stanke, D. and Harshaw, J. 2008. "Potential ASHRAE Standard Conflicts: Indoor Air Quality and Energy Standards." Trane Engineers Newsletter, 37-4.

Trane. 2005. "ASHRAE Standard 62.1-2004: Ventilation Requirements," Engineers Newsletter Live satellite broadcast, APP-CMC023-EN (September 21, DVD). La Crosse, WI: AVS Group.

______. 2005. "CO_2-Based Demand-Controlled Ventilation," Engineers Newsletter Live satellite broadcast, APP-CMC024-EN (November 16, DVD). La Crosse, WI: AVS Group.

______. 2002. "Commercial Building Pressurization," Engineers Newsletter Live satellite broadcast, APP-APV013-EN (April 17, videocassette). La Crosse, WI: AVS Group.

______. 2006. "HVAC Systems and Airside Economizers," Engineers Newsletter Live satellite broadcast, APP-CMC026-EN (May 3, DVD). La Crosse, WI: AVS Group.

______. 2008. "ASHRAE Standards 62.1 and 90.1 and VAV Systems," Engineers Newsletter Live satellite broadcast, APP-CMC034-EN (November 11, DVD). La Crosse, WI: AVS Group.

U.S. Green Building Council (USGBC). 2009. Leadership in Energy and Environmental Design (LEED) Green Building Rating System. Washington, D.C.: USGBC. http://www.usgbc.com.

Warden, D. 1996. "Dual fan, dual duct systems: Better performance at a lower cost," ASHRAE Journal (January): pp. 36-41.

Warden, D. 2004. "Dual-fan, dual-duct goes to school," ASHRAE Journal (May): pp. 18-25.

Index

Trane
www.trane.com

For more information, contact your local Trane office or e-mail us at comfort@trane.com

Literature Order Number	SYS-APM008-EN
Date	September 2009
Supersedes	New