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Understanding Single-Zone VAV Systems

Single-zone variable-air-volume (VAV) is not a new concept, but due to new energy code requirements and greater attention to reducing energy use, it is being applied more frequently. This EN will review these new requirements, discuss the benefits and challenges of single-zone VAV systems, and identify common applications for this system.

A conventional, single-zone constant-volume system uses a temperature sensor in the zone to vary cooling or heating capacity, while the supply fan delivers a constant quantity of air whenever the system is operating.

In a single-zone VAV system, however, the temperature sensor in the zone is used to vary the cooling or heating capacity *and* the airflow delivered by the supply fan to maintain supply-air temperature at a desired setpoint (Figure 1).

Traditionally, single-zone VAV has been used for larger, densely occupied zones that have variable cooling loads. Common examples include gymnasiums, cafeterias, lecture halls, auditoriums, large meeting rooms, churches, and arenas. Therefore, it has been available primarily in larger air-handling units and large, packaged rooftop equipment. This type of equipment has long been available with variable-speed fans and cooling/heating that can be staged or modulated to control discharge-air temperature.

However, due to increased focus on reducing energy use, single-zone VAV is starting to be used more often in K-12 classrooms, retail stores, dormitories, and even offices. As its popularity has increased for these smaller zones, this functionality is beginning to be offered in smaller equipment, such as small

packaged rooftop units or direct expansion (DX) split systems, fan-coils, classroom unit ventilators, and water-source heat pumps.

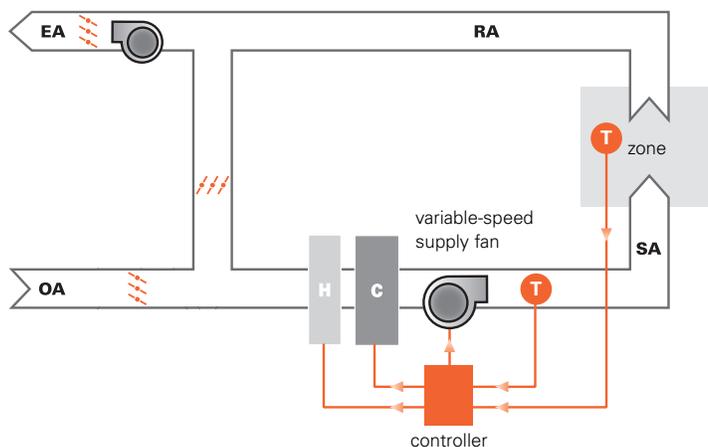
Requirements of ASHRAE Standard 90.1

Requirements for VAV control in single-zone systems were added to the 2010 edition of ANSI/ASHRAE/IESNA Standard 90.1, *Energy Standard for Buildings Except Low-Rise Residential Buildings*.¹ This is especially significant because the U.S. Department of Energy has mandated that each state must update its commercial building code to meet or exceed Standard 90.1-2010 by Oct. 18, 2013.

These requirements are included in the mandatory provisions of Section 6.4. This means that projects must comply with the requirements for single-zone VAV control, regardless of whether the Simplified Approach (by reference from Section 6.3), prescriptive path (Section 6.5), or Energy Cost Budget (Section 11) is used for compliance.

Section 6.4.3.10 (see inset, p.2) includes two parts: the first addresses systems that use chilled water for cooling, while the second part addresses systems that use direct expansion.

Figure 1. Single-zone VAV system



Section 6.4.3.10 (“Single Zone Variable-Air-Volume Controls”) of ASHRAE Standard 90.1-2010.

HVAC systems shall have variable airflow controls as follows:

(a) Air-handling and fan-coil units with chilled-water cooling coils and supply fans with motors greater than or equal to 5 hp shall have their supply fans controlled by two-speed motors or variable-speed drives. At cooling demands less than or equal to 50%, the supply fan controls shall be able to reduce the airflow to no greater than the larger of the following:

- One-half of the full fan speed, or
- The volume of outdoor air required to meet the ventilation requirements of Standard 62.1.

(b) Effective January 1, 2012, all air-conditioning equipment and air-handling units with direct expansion cooling and a cooling capacity at AHRI conditions greater than or equal to 110,000 Btu/h that serve single zones shall have their supply fans controlled by two-speed motors or variable-speed drives. At cooling demands less than or equal to 50%, the supply fan controls shall be able to reduce the airflow to no greater than the larger of the following:

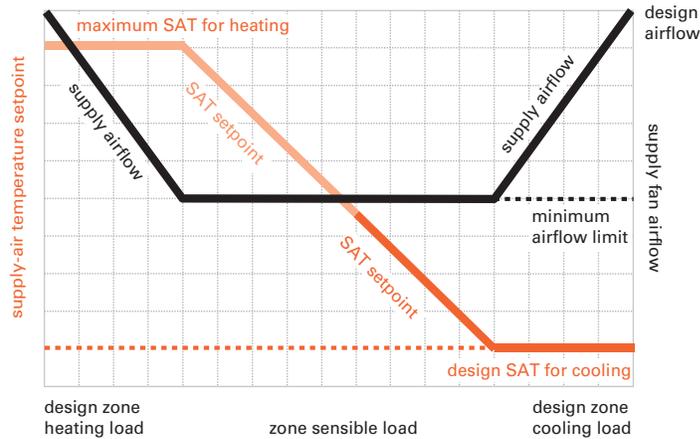
- Two-thirds of the full fan speed, or
- The volume of outdoor air required to meet the ventilation requirements of Standard 62.1.

*Note from the author: In part (a) above, the phrase, “At cooling demands less than or equal to 50%...” may be confusing, so the Standard 90.1 User’s Manual clarifies by stating the following:*²

“The term ‘cooling demand’ refers to the zone sensible cooling load. That is, when the zone sensible cooling load decreases to 50% of the design sensible cooling load for the zone, the supply fan controls shall have reduced airflow to the threshold described.”

In addition, the User’s Manual clarifies that the supply fan can be controlled by either a two-speed motor, an electronically commutated motor (ECM), or a variable-frequency drive (VFD).

Figure 2. Example SZVAV control with variable-speed fan



System Operation

Figure 2 depicts an example control sequence for a single-zone VAV system that uses a variable-speed fan. (Note that control can vary by manufacturer, and there are nuances that depend on whether chilled water or DX is used for cooling and whether the type of heater used can support variable airflow.)

When the zone is at design sensible cooling load (right-hand side of the chart), this system delivers maximum supply airflow at the design supply-air temperature (SAT) for cooling (e.g., 55°F). As the zone cooling load decreases, supply airflow is reduced as needed to maintain the desired temperature in the zone. This is accomplished by varying the speed of the fan motor. Cooling capacity is then staged or modulated to maintain SAT at the same design setpoint. If the system has an airside economizer, the economizer may provide all or part of the cooling needed to achieve the SAT setpoint.

Eventually, the zone sensible cooling load decreases to the point where supply airflow reaches a minimum limit (discussed on p.3). As the cooling load continues to decrease, the fan remains at minimum airflow, but now the SAT setpoint is gradually reset upward to avoid overcooling the zone.

Depending on the outdoor conditions, as the SAT setpoint increases, eventually no mechanical cooling is needed to make this temperature, so the compressors are turned off. And even the airside economizer modulates back to bring in only the minimum outdoor airflow required for ventilation. When that happens, and the load continues to decrease, the zone temperature begins to drop below the zone cooling setpoint, into the deadband between cooling and heating setpoints. The fan continues to operate at minimum airflow, with no compressors or heaters operating, and the zone temperature is allowed to float within this deadband.

Now consider what happens when the zone eventually requires heating—that is, when the zone temperature drops to the heating setpoint. The fan continues to operate at minimum airflow and the SAT setpoint is reset even further upward. Heating capacity is staged or modulated to maintain this supply-air temperature.

Eventually, the zone heating load may increase to the point where the SAT reaches a pre-set maximum limit (also discussed on p.3). As the zone heating load continues to increase, supply airflow is again increased, while heating capacity is staged or modulated to maintain the SAT at this maximum limit.

Minimum airflow. The minimum airflow limit (Figure 2) might be determined by how far the variable-frequency drive (VFD) or electronically commutated motor (ECM) can be turned down. This will be equipment specific, and might be to ensure motor reliability, or in the case of DX equipment, to ensure reliability of the refrigeration system. Alternatively, for some applications this limit might need to be high enough to meet some air distribution requirement.

Standard 90.1 (see inset, p.2) requires this minimum airflow to be no higher than one-half of design airflow for chilled-water equipment, or two-thirds of design airflow for DX. But if the outdoor airflow required to meet Standard 62.1 is higher than this minimum threshold, the higher ventilation-related airflow can be used to avoid underventilating the zone.

Maximum supply temperature when heating. The maximum supply-air temperature (Figure 2) might be determined by the equipment manufacturer, for safety or reliability reasons. Alternatively, for some applications this temperature might be limited to minimize stratification when supplying hot air from overhead diffusers.

When both supply-air diffusers and return grilles are mounted in the ceiling, and the supply air is hot, the buoyancy of this hot air causes some of it to stay up near the ceiling, and bypass from the supply diffusers to the return grilles. This bypass can cause comfort problems, and impacts

ventilation performance, requiring more outdoor air to be brought in through the system intake.

One way to limit or avoid this effect is to deliver the air at a “not-so-hot” temperature during heating. In this case, the maximum supply temperature for heating might be set to 15°F above the zone setpoint.

Constant airflow when heating. For some equipment, the type of heater available may not be able to accommodate variable airflow. In that case, the control sequence will vary airflow when cooling, but then increase to some constant airflow whenever the heater is activated (Figure 3).

When the zone temperature drops to heating setpoint, the fan ramps up to full speed, and the heater is cycled on or staged to warm up the zone. Once the zone temperature rises back up to the deadband between heating and cooling setpoints, the heater turns off and the fan again backs down to minimum speed.

Two-speed fan control. Standard 90.1 states that the supply fan can use either a variable-speed or two-speed motor. The most common implementation of a two-speed motor is probably a DX unit that has either two compressors or one two-stage compressor. This allows the equipment to be used with a conventional thermostat (with COOL1/COOL2 control).

Figure 4 depicts an example control sequence when using a two-speed fan. When the thermostat calls for both stages of cooling, the supply fan operates at high speed with both compressors on. When only one stage of cooling is needed, the fan turns down to operate at low speed with only one compressor on. Then when the thermostat calls for no cooling, the fan continues to operate at low speed with both compressors shut off.

Finally, if the thermostat calls for heat, the fan operates at high speed and the heater turns on. When the zone warms up, the thermostat shuts off the heater and the fan returns to operate at low speed.

Like with variable-speed fan control, the minimum airflow at low fan speed might be determined by equipment design, for safety or reliability reasons. But Standard 90.1 requires this minimum airflow to be no higher than two-thirds of design airflow for DX equipment or one-half of design airflow for chilled water.

Ventilation Control

Another important aspect of system control involves ensuring proper zone ventilation at all operating conditions.

With a constant-speed fan, proper ventilation is achieved by setting the position of the outdoor-air damper during startup and balancing. But with a variable-speed fan, it is no longer that simple.

Figure 3. Example SZVAV control with variable-speed fan and constant airflow when heating

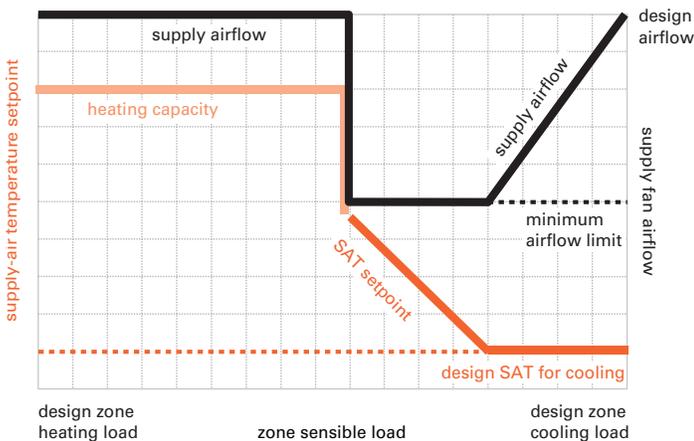
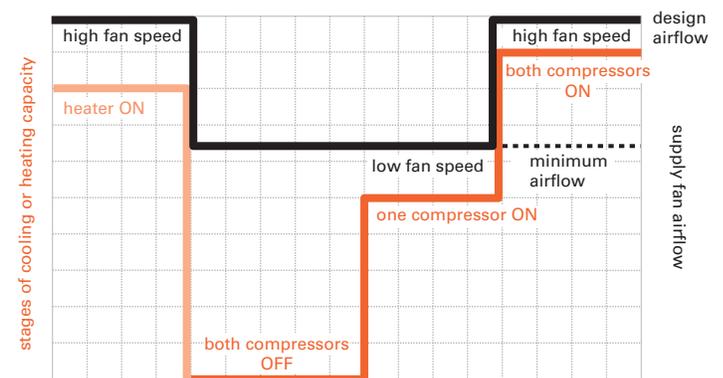


Figure 4. Example SZVAV control with two-speed fan



To demonstrate, Figure 5 depicts the static pressure as air moves through the various components of simple, single-zone VAV system.

The supply fan must create a high enough pressure at the fan discharge (A) to overcome the pressure losses associated with pushing the air through the supply ductwork and diffusers, and into the zone.

In addition, the fan must create a low enough pressure at its inlet (B) to overcome pressure losses associated with drawing the return air out of the zone and through the return-air path (which might include a return grille, ceiling plenum, and some ductwork), and then to draw the air through the return-air damper, filter, and coils inside the air-handling or rooftop unit.

For this example system, the mixing box is where outdoor air mixes with the recirculated air. Due to the pressure drop through the return-air path and damper, the static pressure inside this mixing box (C) is negative (lower than the pressure outside the building). This causes outdoor air to be drawn into this mixing box. During startup and balancing, the return damper is adjusted so that the pressure inside the mixing box is low enough to ensure that

the desired quantity of outdoor air enters through the outdoor-air damper.

In a single-zone VAV system, supply airflow is reduced during part-load operation. Pushing less air through the supply ductwork and diffusers results in less pressure loss, so the fan does not have to create as high a pressure at the fan discharge (D).

In the same manner, moving less air through the return-air path also results in less pressure loss. This causes the static pressure inside the mixing box (E) to increase. In other words, it is not as negative as at design airflow.

The static pressure inside the mixing box (E) has a direct effect on how much outdoor air enters the system. If the outdoor-air damper remains set at a fixed position, the quantity of outdoor air entering through this damper will decrease because the pressure inside the mixing box is not as negative. The result is that outdoor airflow will decrease as supply airflow is reduced.

To ensure that the same quantity (cfm) of outdoor air enters the system, the outdoor-air damper needs to be opened farther when supply airflow is reduced. This requires designing and programming a control sequence that was not

necessary with a traditional, constant-volume system.

To address this issue and ensure proper ventilation in a single-zone VAV system, consider one of the following approaches.

Two-position OA damper control.

For a system with two-speed fan control (Figure 4), two position setpoints can be used to control the outdoor-air damper.

When the fan is operated at high speed, the OA damper position is set to bring in the quantity of outdoor air required by code. Then, when the controller switches the fan to low speed, the OA damper is opened more, in order to bring in the same quantity (cfm) of outdoor air.

Note that these are minimum position setpoints. If the system includes an airside economizer, the OA damper may be opened farther when conditions are suitable for economizing.

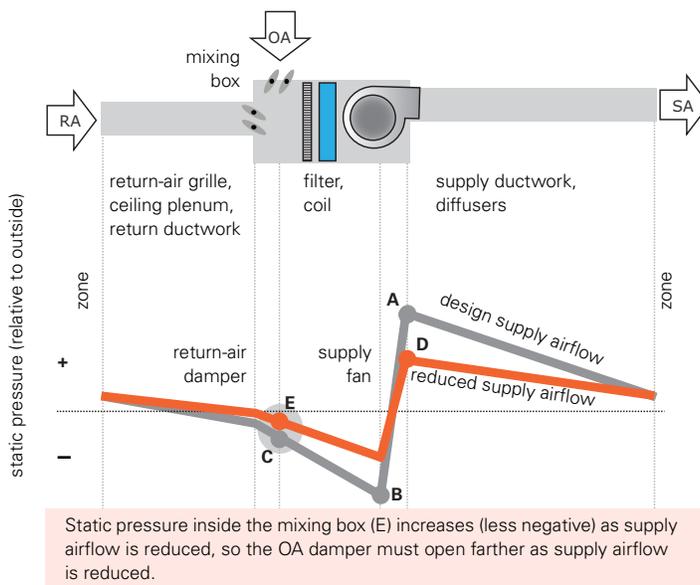
Proportional control of OA damper.

For a system with variable-speed fan control, one solution could be to modulate the position of the outdoor-air damper in proportion to the changing supply fan speed (Figure 6). This is a relatively inexpensive solution, so it is likely to be used in many of the smaller, single-zone VAV applications.

With the system operating at design supply airflow (maximum fan speed), the OA damper position is set to bring in the quantity of outdoor air required by code. Then with the system operating at minimum airflow (minimum fan speed), the OA damper is opened more, in order to bring in the same quantity of outdoor air.

During system operation, the controller modulates the position of the OA damper in proportion to the change in supply airflow, which is determined from the signal sent to the variable-speed drive on the fan. Again, this is the minimum damper position, so it may be opened farther when conditions are suitable for economizing.

Figure 5. Need for ventilation control in SZVAV system



This method is not perfectly accurate over the entire range of airflows, since damper performance is nonlinear. And it does not account for outside influences like wind or stack effect. Therefore, it does result in some over-ventilation in the middle of the fan speed range, but much less over-ventilation than if a fixed-position damper were used (see inset).

One way to minimize this inaccuracy is to add a third damper position setpoint for some intermediate fan speed (Figure 6). This ensures proper ventilation at this intermediate speed, and minimizes any over-ventilation by shrinking the ranges between setpoints.

Flow-measuring OA damper.

Another method for controlling ventilation is to 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 an airflow measurement station or a flow-measuring damper (Figure 7).

This method is more accurate over the range of airflow, and can respond to pressure fluctuations caused by wind or stack effect. This approach 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, so it is more likely to be used in larger systems.

Dedicated OA delivered directly to each zone. The three previous approaches are suitable for a piece of equipment that has a mixing box and outdoor-air damper—such as a packaged rooftop unit, air-handling unit, or classroom unit ventilator.

But most models of fan-coils or water-source heat pumps just have a single inlet opening—no mixing box or outdoor-air damper. Ventilation is typically provided by a dedicated outdoor-air system.

In some applications, the dedicated OA system delivers the conditioned outdoor air to the inlet side of each local heat pump or fan-coil (Figure 8).

For a horizontal-style unit, the air might be ducted to discharge into the open ceiling plenum, near the intake of the unit. For a vertical-style unit, it might be ducted to a closet where the unit is installed. In either case, the outdoor air mixes with recirculated air in the plenum or closet before being drawn in through the intake of the heat pump or fan-coil.

As described earlier (Figure 5), if the local unit has a two-speed or variable-speed fan, the static pressure at its inlet is not as negative when fan speed

Fixed-position OA damper and SZVAV

ASHRAE Standard 62.1-2010, Section 5.3, states: ³

“The system shall be designed to maintain no less than the minimum outdoor airflow as required by Section 6 under any load condition. Note: Variable Air Volume (VAV) systems with fixed outdoor air damper positions must comply with this requirement at minimum system primary airflow.”

This means that the OA damper position must be set to bring in the minimum required outdoor airflow when the fan is operating at minimum speed, which results in significant over-ventilation and increased energy use when the fan operates at higher speeds (Figure 6).

is reduced. Therefore, outdoor airflow will decrease as supply airflow is reduced.

An added challenge is that the outdoor air is delivered by a separate fan, and the pressure inside the ventilation ductwork is influenced by the changing operation of all the local fans served by the ventilation system.

With this configuration, to ensure that the required quantity of outdoor air is delivered, a pressure-independent VAV terminal should be installed in the ventilation duct for each zone (Figure 8). As the local fan speed changes, this damper modulates to maintain the same outdoor airflow, regardless of fan speed.

Figure 6. Proportional control of OA damper

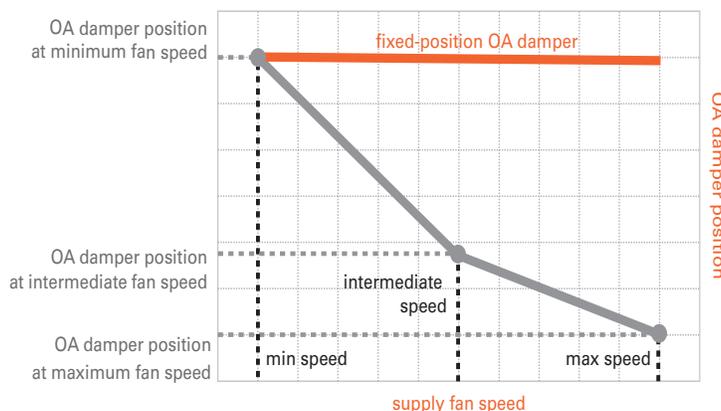


Figure 7. Flow-measuring OA damper



Alternatively, consider delivering the conditioned outdoor air directly to each zone, rather than to the inlet of the fan-coils or heat pumps (Figure 9). In this case, the local units condition only recirculated air.

Because the outdoor air is not distributed through the local fan, that fan can operate with a two-speed or variable-speed motor, without impacting the quantity of outdoor air delivered to the zone.

This “direct-to-the-zone” approach also affords the opportunity to deliver the outdoor air at a cold temperature, rather than reheated to neutral. This has several installed cost and energy-saving benefits.^{4,5}

CO₂-based DCV. Single-zone VAV systems are often used for densely occupied zones with varying population. This typically makes them good candidates for using CO₂-based demand-controlled ventilation (DCV).

With this approach, a sensor measures the concentration of CO₂ in the zone, and this concentration is then used by the controller to reset outdoor airflow as population changes. This saves energy by not over-ventilating the zone during periods of partial occupancy.

While Standard 62.1 allows the use of DCV, it also includes a few requirements to make it work:³

“The breathing zone outdoor airflow shall be reset in response to current occupancy and shall be no less than the building component ($R_a \times A_z$) of the zone.” (Section 6.2.7.1.2)

“Systems shall be operated such that spaces are ventilated in accordance with Section 6 when they are expected to be occupied.” (Section 8.3)

While the people component of the ventilation rate ($R_p \times P_z$) can be varied, Standard 62.1 states that the controller cannot reduce outdoor airflow any

lower than the building component of the ventilation rate ($R_a \times A_z$) for that zone.

Section 8.3 clarifies that this applies whenever that zone is “expected” to be occupied. That is, the standard is not requiring this building component of the ventilation rate to be delivered to the zone 24 hours a day, 7 days a week. Rather, it only requires this during the normally scheduled occupied period.

Because the mixing box pressure varies as supply airflow changes (Figure 5), DCV must be combined with either proportional damper control or a flow-measuring damper in order to prevent outdoor airflow from dropping below this minimum threshold.

A previous EN and the *Standard 62.1 User’s Manual* both describe a proportional control sequence that can be used to implement DCV in a single-zone system.^{6,7}

Potential Benefits of SZVAV

Compared to a constant-volume system, typical benefits of single-zone VAV include lower energy use, better dehumidification at part load, and less fan-generated noise at reduced speeds.

Lower energy use. Compared to a constant-volume system, a single-zone VAV system can result in significant fan energy savings at part-load conditions.

Figure 10 shows the potential energy savings of using two-speed or variable-speed fan control in an example single-story K-12 school. TRACE™ 700 was used to model the building with a packaged rooftop unit serving each zone.⁸ The baseline building uses conventional, constant-volume rooftops.

Figure 8. Conditioned OA delivered to inlet of each local unit

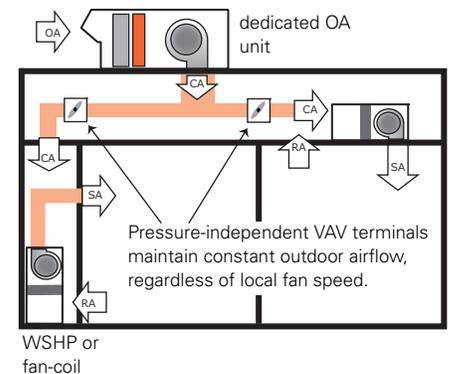
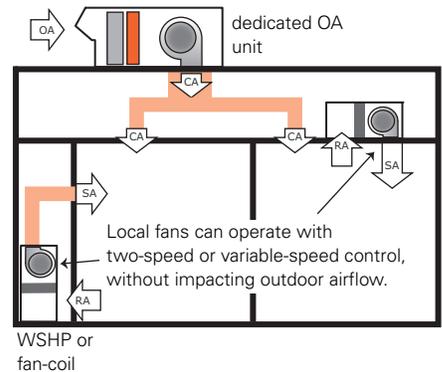


Figure 9. Conditioned OA delivered directly to each zone



For this example, two-speed fan control (constant fan speed when heating) reduced HVAC energy use by about 25 percent, while variable-speed fan control (capable of variable fan speed when heating) reduced HVAC energy use by about 30 percent.

Of course, the impact of two-speed or variable-speed fan control on the energy use of a specific building depends on climate, building layout, and operating hours. For this example, high internal loads in the classrooms resulted in systems operating in cooling mode for most occupied hours (even in Minneapolis). Operation of the heater was mostly limited to morning warm-up mode.

For applications with more occupied hours of operation in the heating mode, the difference in energy use between two-speed and variable-speed fan control will likely be larger, since the two-speed fan operates at high speed whenever the heater is on.

Better dehumidification at part load. Compared to a constant-volume system, a single-zone VAV system improves dehumidification performance because it continues to deliver cool, dry air at part-load conditions.

To demonstrate this benefit, Figure 11 compares the performance of two systems serving an example classroom in Jacksonville, Florida.⁹ At cooling design conditions, this system mixes 450 cfm of outdoor air with 1050 cfm of recirculated air, then delivers the resulting 1500 cfm of supply air to the classroom.

At the part-load, peak dew point condition, it is not as hot outside, but more humid. The system with a constant-speed fan (CV) continues to supply a constant volume of air to the zone (1500 cfm). Meanwhile, as the sensible cooling load in the zone decreases, the compressor cycles on and off, resulting in warmer air delivered to the zone—63°F for this example load condition.

Although delivering this warmer air prevents over-cooling the zone, the cycling compressor results in a warmer average coil temperature and less moisture removed. This 63°F air has not been dehumidified as much as if it was cooled down to 55°F. This warmer, wetter supply air causes zone relative humidity to rise to 67 percent at this part-load condition.

In contrast, at part-load conditions, as the sensible cooling load in the zone decreases, the system with a variable-speed fan (SZVAV) responds by first

reducing airflow delivered to the zone, while maintaining a constant supply-air temperature. At this same example part-load condition, supply airflow is reduced from 1500 cfm to 900 cfm, while the system still delivers the air at 55°F.

Reducing the airflow allows the system to continue supplying the air at a cool temperature, so the coil removes more moisture. At this part-load condition, because the supply air is still cool and dry, the relative humidity in the classroom rises to only 57 percent with the variable-speed fan, compared to 67 percent with the constant-speed fan (Figure 11).

In this comparison, for the variable-speed fan system, the load on the cooling coil is 4 tons and the fan moves 900 cfm. So achieving this lower humidity level requires a bit more compressor energy, while benefitting from much lower fan energy (Table 1).

As another point of comparison, on a mild rainy day, the variable-speed fan results in much better dehumidification in this same example classroom—60 percent RH, compared to 73 percent RH with a constant-speed fan (Table 1).

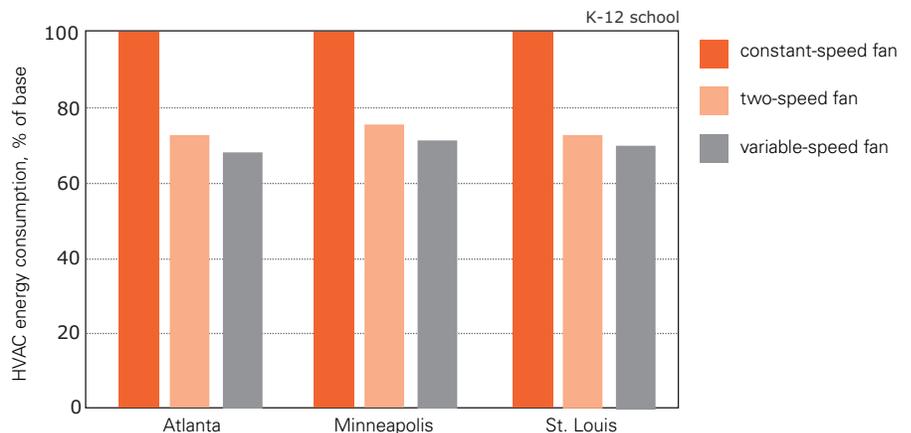
Another method for improving part-load dehumidification could be to add hot gas reheat (HGRH). In order to limit space relative humidity to 60 percent, a constant-volume system will over-cool the air to further dehumidify, and then reheat it to prevent overcooling the zone. This requires more compressor energy, along with more fan energy (Table 1).

For many applications, the variable-speed fan may do an adequate job of limiting indoor humidity levels, so that the system may not require any other dehumidification enhancements—such as hot gas reheat.

However, if a project requires lower humidity levels than a single-zone VAV system can achieve, some types of equipment can be equipped with hot gas reheat also. In that case, the variable-speed fan may limit humidity most of the time, but the reheat can be used if needed.

In the case of fan-coils or water-source heat pumps, a more efficient approach would typically be to use the dedicated outdoor-air system to dehumidify the outdoor air centrally, rather than equip each local unit with reheat.^{4,5}

Figure 10. Example energy savings from two-speed or variable-speed fan control



Less fan-generated noise at reduced fan speeds.

Air distribution equipment, including fans, ductwork, and diffusers, can have a significant impact on background sound levels in occupied spaces.

Because the fan operates at reduced speed, a single-zone VAV system benefits from less fan-generated noise at part-load conditions. The actual sound level depends on the type of fan, operating conditions, and installation. But, in general, as fan speed decreases, the sound level will also decrease.

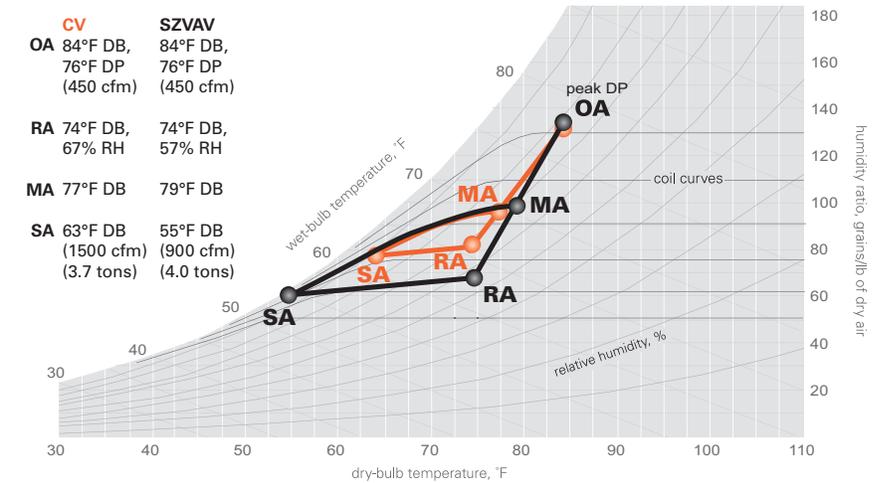
Application Considerations

In order to realize these benefits, the specific application must be well-suited to using single-zone VAV. Following are some of the challenges or considerations when applying this type of system.

Loads throughout a large zone must be fairly uniform. This is a single-zone system, responding to a single temperature sensor, so it does not have the capability to satisfy simultaneous heating and cooling requirements.

If the loads throughout a large zone are not fairly uniform, it may result in undesirable temperature variations in some areas of the zone that are farther away from the temperature sensor.

Figure 11. Improved dehumidification at part load for an example classroom in Jacksonville, FL



Design air distribution system to accommodate variable airflow. Use diffusers that will provide proper air distribution at reduced airflows. Diffusers that are appropriate for conventional VAV systems are probably best suited.

Keeping duct runs as short and symmetric as possible is good general guidance. Systems with longer or asymmetric duct runs do not necessarily prohibit the use of single-zone VAV, but they are more susceptible to unequal airflows between diffusers as the supply fan modulates.

Ensure proper ventilation as supply airflow changes. For smaller systems, two-position or proportional damper control is more likely to be used, while larger systems may use an airflow-measurement device, such as a flow-measuring damper.

When a dedicated outdoor-air system is used, consider delivering the conditioned outdoor air directly to each zone, rather than to the inlet of the local equipment.

If implementing demand-controlled ventilation, be sure to provide some method to vary relief (exhaust) airflow to prevent negative building pressure when DCV reduces the intake airflow. Likewise, if a zone has some type of minimum exhaust requirement—such as a restroom, kitchen, or locker room—DCV should not be allowed to reduce intake airflow any lower than the required exhaust, unless the zone receives makeup air from some other source.

Table 1. Part-load dehumidification comparison: CV versus SZVAV

	constant-speed fan	variable-speed fan	constant-speed fan with HGRH
peak DPT (84°F DBT, 76°F DPT)			
zone humidity, %RH	67%	57%	
cooling load, tons	3.7	4.0	
fan airflow, cfm	1500	900	
mild, rainy (70°F DBT, 69°F WBT)			
zone humidity, %RH	73%	60%	60%
cooling load, tons	1.6	1.9	2.4
fan airflow, cfm	1500	750	1500

Consider system operation when heating. ASHRAE 90.1 addresses single-zone VAV during cooling operation. If variable airflow is also desired during heating operation, the equipment will likely require some type of modulating heat, such as a modulating gas heater or a hot-water heating coil.

Otherwise, if the equipment has on/off gas or electric heat, the manufacturer may require it to operate at full airflow whenever the heater is activated (Figure 3). Therefore, check with the manufacturer when specifying or purchasing single-zone VAV equipment.

Variable-speed fan control requires a zone temperature sensor. A conventional thermostat (with COOL1/COOL2 control) can typically be used for two-speed fan control, but variable-speed control requires a zone temperature sensor connected to the unit controller.

This is important to communicate when specifying or purchasing single-zone VAV equipment. If replacing an older piece of equipment (controlled by a conventional thermostat) with a new, variable-speed fan unit, this will likely require replacing the thermostat with a zone sensor. Wireless zone sensors make this much easier, especially for those projects where pulling wires may be difficult.

By John Murphy, applications engineer, and Beth Bakkum, information designer, Trane. You can find this and previous issues of the Engineers Newsletter at www.trane.com/engineersnewsletter. To comment, e-mail us at comfort@trane.com.

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