Controlling Minimum Ventilation Volume In VAV Systems

Technician adjusts air dampers in an outdoor AHU.

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Accurately controlling the amount of outdoor air brought into a building is a major factor in ensuring good indoor air quality (IAQ). However, controlling this minimum ventilation volume is difficult in variable-air-volume (VAV) systems.

While a number of proposed control systems work well in theory, they fall short in practice. Their primary failing is an unrealistic assumption of control accuracy, which can lead to gross errors in ventilation volume. This article reviews several options for controlling the minimum ventilation volume in a VAV system. It demonstrates the inaccuracies inherent in several widely-accepted control schemes, before proposing two methods that offer much greater accuracy.

VAV System Challenges

With constant-air-volume systems, controlling the minimum ventilation volume is relatively simple. Because the supply fan always provides the same amount of air, the minimum position of the outdoor air damper can be set so that it always supplies the specified minimum amount of outdoor air.

After the initial air-system balancing—assuming that the system is properly maintained and that no modifications are made—no further control is required to ensure that minimum ventilation requirements are met.

However, a problem arises when this same control logic is applied to VAV systems. For example, consider a large VAV system with design conditions of 100,000 cfm (47 195 L/s) of supply air and 15,000 cfm (7079 L/s) minimum of outdoor air. If the load falls to 60%, the supply fan backs off to 60% flow. This causes the outdoor air volume to fall to 60%, or 9,000 cfm (4247 L/s), and it no longer meets the design requirement of 15,000 cfm (7079 L/s) minimum.

So, how does one accurately control the minimum ventilation volume in VAV systems? Several options are available.

Fan-Tracking Systems

Some engineers believe the way to control minimum ventilation volume is to use return fans and a fan-tracking system.[1,2] The most accurate fan-tracking system is volumetric tracking. This control method attempts to maintain a fixed differential between the supply-fan and return-fan volumes.

The fixed differential is the minimum ventilation volume that the supply fan draws through the outside air dampers. *Table 1* shows the airflow design intent, using the previous example.

Figure 1 shows a simplified drawing of the airflow control components in a typical volumetric fan-tracking system. The key components are the airflow-measuring stations for the supply air (AFS-1) and return air (AFS-2). They measure the velocity pressure of the airflows, which then can be converted into air speed and air volume.

Their readings are sent to the airflow controller (AFC-1), which adjusts the return-fan volume with a variable-frequency drive (VFD) to maintain the fixed differential between the supply-fan and return-fan volumes.

For this example, the supply airflow is assumed to have an even-velocity profile of 4,000 fpm (20.32 m/s) at design conditions, while the return airflow has a velocity of 2,000 fpm (10.16 m/s). These velocities may be higher than average, and duct areas with a consistent velocity profile may be impossible to find in a typical system, but the intent here is to give the control system every possible advantage.

Now, check these same systems when the supply airflow is reduced to 60% of design. Keeping a constant 15,000 cfm (7079 L/s) differential between the supply airflow and return airflow would re-

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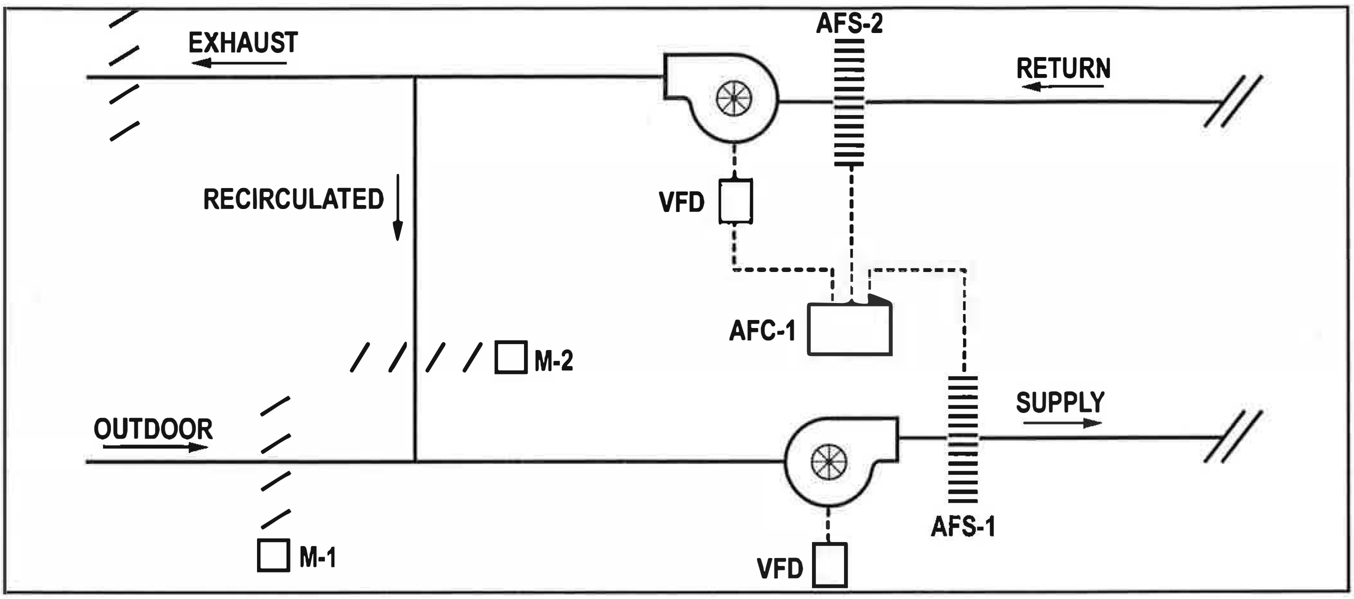

Figure 1: Volumetric fan-tracking system.

quire the return fan to operate at 53% of maximum flow, as calculated below:

$$\frac{60{,}000\text{ cfm supply air} - 15{,}000\text{ cfm ventilation air}}{85{,}000\text{ cfm design return air}}$$
$$= 53\%\text{ of design return-air volumetric flow rate*}$$

Assuming perfect measurement, but allowing an extremely low ±5% for possible transducer and control errors, *Tables 2* and *3* show the readings for the supply fan and return fan, respectively. Controlling within ±5% means that the system has to control velocity pressure within ±0.04 in. w.g. (10 Pa) on the supply fan and ±0.007 in. w.g. (1.7 Pa) on the return fan. These are very small differences—it would be difficult to find controls with this level of accuracy.

Now assume the system is trying to provide 60,000 cfm (28 317 L/s) of supply air, and the system has to account for a *minus* 5% error. That is 57,000 cfm (26 901 L/s) of supply air. Now, also assume that the system is trying to supply 45,000 cfm (21 238 L/s) of return air, and allow for a *plus* 5% error. That is 47,250 cfm (22 300 L/s) of return air. The difference between the supply air and return air volumes is the outside-air volume, which is only 9,750 cfm (4601 L/s). This is 35% below the required minimum of 15,000 cfm (7079 L/s).

Even worse, this 35% error is assuming perfectly accurate sensing of the airflow and controlling within 0.007 in. w.g. (1.7 Pa). When errors for the controls and velocity profiles are factored in, the very best that might be expected is ±50% error. This has been confirmed by Ted Cohen, past president of Associated Air Balance Contractors (AABC), and chairman of the ASHRAE Project Committee for *Guideline 1-1989R, The HVAC Commissioning Process*.

Table 4 shows data from an actual fan-tracking system. This is a typical unit on a project with approximately 100 supply/return fan systems. Electronic flow-measuring stations were in place in the supply and return ducts. In addition, a certified balancing contractor ran pitot-tube duct traverse readings to verify the accuracy of the measuring stations.

It took several days of fine-tuning and recalibrating the stations to achieve even this accuracy. Eventually, after changing the microchips in the stations, the readings were tuned in the software to match the "as-installed" conditions.

Note that at 100% airflow, even though all supply-air and return-air readings were within 3% of design, the pitot-tube-measured outside air volume of 1,805 cfm (852 L/s) was below design by more than 20%. Since the readings from the measuring stations were used as the control point, it should be expected those numbers would agree with the design intent. However, the pitot-tube readings are a better gauge for determining actual flows.

The numbers were similar at reduced flow (60% of design). With a 0.7% error on the measuring stations, the pitot-tube-measured outside air volume was off by 26%. This is not the kind of accuracy engineers need considering today's heightened awareness of IAQ.

Fan-tracking systems ignore the realities of control accuracy. Combining control inaccuracy with a system that measures two indirect variables to directly control a third is a recipe for disaster.

	Design Flow		Reduced Flow	
	(cfm)	(L/s)	(cfm)	(L/s)
Supply Air	100,000	47 195	60,000	28 317
Return Air	85,000	40 116	45,000	21 238
Ventilation Air	15,000	7079	15,000	7079

Table 1: Example of volumetric fan-tracking theory.

$$\text{*}\frac{28\ 317\text{ L/s supply air} - 7079\text{ L/s ventilation air}}{40\ 116\text{ L/s design return air}}$$

Moreover, the above analysis ignores instability problems associated with fan-tracking systems—a topic of several ASHRAE papers in recent years.[3,4] It is time for engineers to take a different approach to controlling minimum ventilation volume.

Measuring Outdoor Air Intake

Placing a measuring station in the outdoor air intake to measure the total outdoor airflow seems like a logical approach to measuring not only the minimum flow, but the amount of outdoor air entering the VAV system at all times. At least this would directly measure the value that is to be controlled. However, while this method works fine in theory, it too, fails in practice.

To accurately measure outdoor air, the system must first overcome the turbulence from air entering and changing directions through the outdoor air louvers. This can be accomplished using air straighteners or equivalent configured orifices.

Assuming the turbulence can be overcome, the challenge lies in finding control components to measure airflow with sufficient accuracy. Temperature extremes in outdoor airstreams usually preclude using electronic hot-wire sensors, even if they are temperature-compensated, because the compensation typically covers narrower bands than the summer/winter temperature extremes experienced in much of the United States.

A better solution is multiple-point, pitot-tube averaging probes, which work well with VAV boxes. However, it must be remembered that the inlet velocity to VAV boxes at design flow is in the 3,000 fpm (15.2 m/s) range. And by using amplifying pick-up probes, the differential pressure can be raised to almost 1 in. w.g. (249.0 Pa). Signal levels that start this high at design airflow are still easy to accurately read and control at reduced airflows.

However, to achieve that level of velocity pressure in an outdoor air measuring station, the outdoor air damper or flow station must be smaller to increase the outdoor air velocity to controllable levels. The velocity must then be reduced immediately beyond the measuring station because the velocity through a filter section cannot exceed 500 fpm (2.5 m/s). The air pressure drop through this restricted section imposes a tremendous energy penalty on the supply fan.

	Velocity		Velocity Pressure		Vel. Press. Error	
	(fpm)	(m/s)	(in. w.g.)	(Pa)	(in. w.g.)	(Pa)
Design Flow	4,000	20.32	0.997	248.33	0	0
60% Flow	2,400	12.19	0.359	89.42	0	0
+5% Error	2,520	12.80	0.396	98.64	+0.037	+9.22
-5% Error	2,280	11.58	0.324	80.70	-0.035	-8.72

Table 2: Supply-fan duct velocity.

	Velocity		Velocity Pressure		Vel. Pres. Error	
	(fpm)	(m/s)	(in. w.g.)	(Pa)	(in. w.g.)	(Pa)
Design Flow	2,000	10.16	0.2493	62.10	0	0
60% Flow	1,060	5.38	0.0700	17.44	0	0
+5% Error	1,113	5.65	0.0772	19.23	+0.0072	+1.79
-5% Error	1,007	5.12	0.0632	15.74	-0.0068	-1.70

Table 3: Return-fan duct velocity.

Measuring Device	Supply Air		Return Air		Outside Air		Error (%)
	(cfm)	(L/s)	(cfm)	(L/s)	(cfm)	(L/s)	
Design Intent	11,890 (100.0%)	5611 (100.0%)	9,630 (100.0%)	4545 (100.0%)	2,260 (100.0%)	1067 (100.0%)	0
Measuring Station	12,169 (102.3%)	5743 (102.3%)	9,918 (103.0%)	4681 (103.0%)	2,251 (99.6%)	1062 (99.6%)	-0.4
Pitot-Tube	11,700 (98.4%)	5522 (98.4%)	9,895 (102.8%)	4670 (102.8%)	1,805 (79.9%)	852 (79.9%)	-20.1

Table 4: CFM readings from an actual fan-tracking installation (100% airflow).

To reduce energy waste, measuring stations might be sized so the differential pressure across the amplifying probes is 0.6 in. w.g. (149 Pa) when the unit is supplying 100% outside air in the economizer mode, rather than the 0.2 in. w.g. (50 Pa) typical for air-handling unit mixing boxes. If the unit is selected for a 20% safety factor for future expansion, the differential pressure at design flow (80% of maximum) would be $(0.8)^2$ (0.6 in. w.g.) = 0.38 in. w.g. (94.7 Pa).

If the minimum ventilation volume is 15% of installed capacity, the differential pressure at the measuring stations would be $(0.15)^2$ (0.38) = 0.0085 in. w.g. (2.1 Pa). To control the system within 10% would require controlling velocity pressure within the incredible precision of ±0.0008 in. w.g. (0.2 Pa).

To help appreciate how small that pressure difference is, consider a real-life example. When a person is walking at two feet per second (0.6 m/s) in still air, the air is hitting that person's face at 0.0008 in. w.g. (0.2 Pa)—the same velocity pressure cited in the above example. Obviously, one would be justified in questioning whether such control precision is realistic, especially in an outdoor airstream affected by wind gusts.

Remember that this is also at the energy expense of shrinking the outdoor air inlet area to accelerate the air to a readable range. Given these two major drawbacks, it is not a suitable solution to use fixed-area-measuring stations sized to handle total outdoor airflow to control minimum ventilation volume on air-handling units with economizer control.

It should be noted that the previous statement refers to fixed-area measuring stations, designed to measure outdoor airflow from 100% down to minimum flow. Stations designed to measure minimum airflow only, or dynamic measuring stations are a different story. Dynamic airflow stations measure velocity pressure between the damper blades, then compare the current damper position and velocity pressure to accurately calculate airflow. These devices do not impose an energy penalty on the supply fan because the velocity between the blades increases as the damper modulates toward the closed position. Dynamic measuring stations should not be confused with full-sized, fixed-area measuring stations.

The previous examples demonstrate that VAV systems with fan tracking, or fixed-area full-sized outdoor air measuring stations, cannot accurately supply the required minimum ventilation volume. The following are two control methods that can achieve this.

Solution 1: Injection-Fan Method

Figure 2 shows a simplified drawing of a control system using a separate injection fan, which ensures that the minimum outdoor air is supplied. The injection fan runs whenever the system requires outdoor air. Its ductwork is sized so that velocity pressures are easily readable. An airflow-measuring station (AFS-1) adjusts the injection-fan volume to correct for wind effects or back pressures.

This system has numerous advantages over outdoor air measuring stations and the fan-tracking method. As described earlier, fixed-area-measuring stations do not accurately control outdoor airflow from 100% down to minimum flow, and they impose a heavy energy penalty on the supply fan.

The injection-fan method overcomes these disadvantages because it uses a parallel airflow path for the minimum ventilation airflow. This allows the air velocity to be raised to an easily readable and controllable range. This is a more appropriate application for a fixed area airflow station, particularly if the minimum airflow is to be reset.

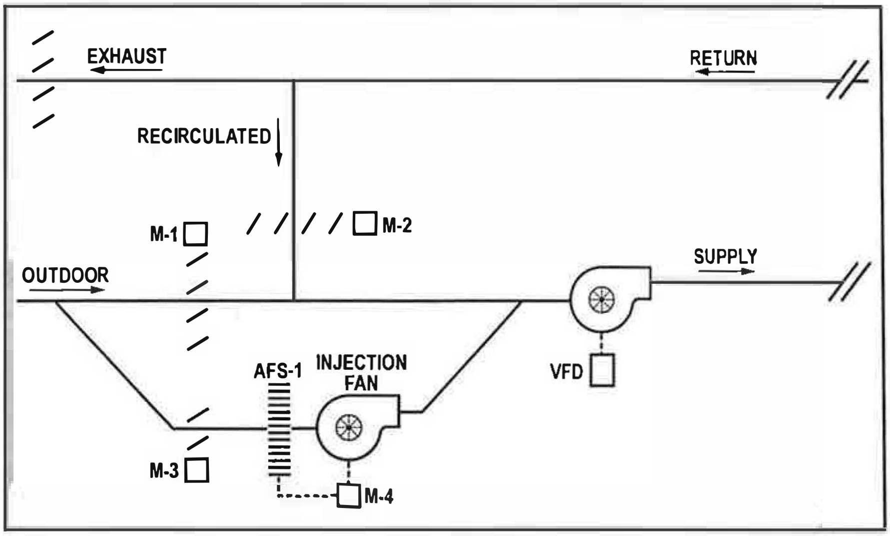

Figure 2: Injection-fan system.

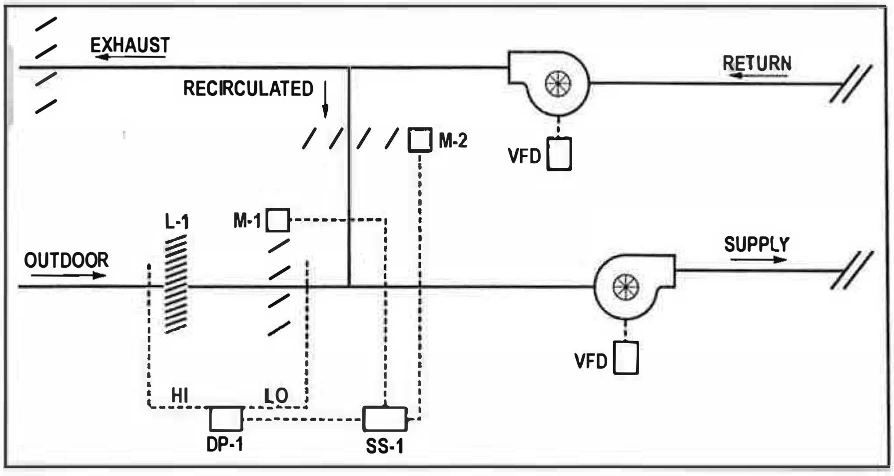

Figure 3: Plenum-pressure control system.

It was also demonstrated that the fan-tracking method—which attempts to control outdoor airflow indirectly by controlling the supply and return airflows—resulted in a small control error, but a large minimum-airflow error.

By contrast, the injection-fan method controls the minimum outdoor airflow directly, so that a small control error would result in only a small airflow error. Using the earlier example, the ±5% error in the 15,000 cfm (7079 L/s) of outdoor air would result in an error of only ±750 cfm (354 L/s)—not ±3,750 cfm (1769 L/s) that occurred with the fan-tracking method.

The injection-fan method improves minimum airflow control and eliminates the energy penalty imposed on the supply fan. Unfortunately, this method may require extra ductwork on new jobs, and can be difficult to retrofit to existing systems.

Solution 2: Plenum-Pressure Control

Figure 3 shows a simplified drawing of another method for controlling minimum ventilation volume that offers superior accuracy compared to full-sized, fixed-area measuring stations and the fan-tracking method. It is known as plenum-pressure control and was first proposed by Mumma and Wong.[5] Plenum-pressure control can be used on new or existing jobs for little additional cost over the simple economizer cycles found in constant-volume systems.

Here, a differential-pressure transducer (DP-1) and a signal selector (SS-1) have been added. A separate actuator is added onto the recirculated air damper, if one is not there already.

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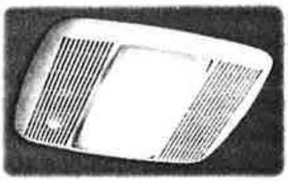

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The minimum position of the outdoor air damper is set so that—with the recirculated-air damper open and the supply fan delivering maximum airflow of 100,000 cfm (47 195 L/s)—the amount of outdoor air is 15,000 cfm (7079 L/s). This is the same process a balancing contractor would perform on a constant-volume system.

The combination of the louver and throttled damper is now treated as a fixed orifice for airflow control. DP-1 measures the pressure drop across the outdoor-air louver (L-1) and the outdoor air damper in this minimum position. On jobs where this method has been used, the settings have varied from 0.35 in. w.g. (87 Pa) to 0.15 in. w.g. (37 Pa)—well within controllable ranges.

As the supply fan reduces airflow, the differential pressure drops. The control system holds the outdoor air damper at the fixed minimum position and modulates the recirculated-air damper toward the closed position to maintain the required flow through the outdoor air damper. This ensures that the minimum ventilation volume is always supplied.

The energy penalty imposed by plenum-pressure control is minimal, especially when compared to the penalty imposed by a reduced-size, airflow-measuring station. At maximum airflow, it imposes no additional system pressure drop.

At reduced airflow, modulating the recirculated-air damper has minimal energy impact because the added resistance is only required to maintain the set point of the differential-pressure controller, typically in the range of 0.18 in. w.g. (44.8 Pa).

Proper operation of a plenum-pressure control system is dependent upon repeatable positioning of the outdoor-air damper. A damper with a fixed-minimum section is easier to control accurately. Repeatable positioning of a full outdoor air damper can also be obtained by the use of the new, electronic, over-the-shaft-type damper actuators.

While it is preferable to utilize electronic or direct-digital controls (DDC) in a plenum-pressure control system, it is not absolutely necessary. However, electric or pneumatic control systems require signal selectors, and recalibration may be required more often than with electronic controls. With a DDC system, the control of the recirculated-air damper can be programmed into the software, with positions and pressures trend-logged or monitored remotely.

In summary, plenum-pressure control has several advantages: it allows accurate measurement of minimum ventilation air; it provides readable pressure drops; its energy penalty is small; it can be used with systems with or without return fans, and its controls are very simple.

Systems with or without return fans; require separate controls to maintain building pressurization. Neither maintaining a minimum outdoor airflow rate nor "fan-tracking" assures building pressure control will be maintained.

Building pressure should be controlled by modulating a relief damper from a building pressure sensor, dampened to compensate for wind gusts. Return fans should be controlled to maintain either a fixed positive discharge pressure or a pressure reset by relief damper position. This eliminates wind blowing in through the relief damper.

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Conclusion

Before specifying fan-tracking systems or airflow-measuring stations, engineers should calculate the potential error in measuring airflows for a specific application. They should also calculate the potential energy penalty in accelerating the outdoor air to levels that can be measured and controlled at minimum flow.

Errors with fan-tracking systems can easily exceed ±25%. Measuring airflow from 100% down to minimum with fixed-area, flow-measuring stations generally results in velocity pressures that cannot be accurately calculated and controlled—despite what the computer readouts might say. Engineers cannot expect closer control. Moreover, with these kinds of errors, ventilation rates to meet *ANSI/ASHRAE Standard 62-1989, Ventilation for Acceptable Indoor Air Quality* cannot be maintained without use of energy wasting safety factors. The plenum-pressure control approach is gradually gaining acceptance. It has been recommended and used by Ted Cohen, past president of AABC and the author has used it on numerous systems.

The bottom line is this: fan-tracking systems and fixed-area, flow-measuring stations sized to measure total outdoor airflow do not work in practice as they do in theory, and it is time the engineering community recognizes this. The injection-fan method and plenum-pressure control are better solutions.

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