ENGINEERS NEWSLETTER

providing insights for today's HVAC system designer

The Threefold Challenge Of …

Ventilating Single-Duct VAV Systems

From the editor …
Inside this first issue of the Engineers Newsletter for 1998, you'll find a survey. It represents one of our resolutions for the new year: to discover how we can better tailor this publication to meet your needs. Please invest a few minutes to complete the questionnaire and return it to us promptly. We'd like to hear from you!

Many modern buildings rely on variable air volume (VAV) systems for heating, air conditioning and ventilation. The fact that they provide many zones of control at relatively low cost **and** save fan horsepower makes them a popular system choice. They also do an excellent job of indirectly controlling building humidity since the air handler delivers constant-temperature, low-dew-point air at all load conditions.

The simplest and most common system—single-duct VAV—comprises a central VAV air handler and several local VAV boxes, one for each space or thermal control zone. See Figure 1. The air handler delivers **primary air** (a mixture of first-pass outdoor and recirculated return air) to the VAV boxes through a single duct. Then each VAV box regulates the flow of primary air into the space to maintain the desired sensible temperature.

Providing proper ventilation with this simple multiple-space VAV system presents us with a three-part challenge … and ASHRAE Standard 62-1989, "Ventilation for Acceptable Indoor Air Quality," defines the standard of care we must exercise along the way:

- **Design** for adequate ventilation capacity.

  Standard 62 directs us to use a multiple-space equation (MSE) when calculating the required outdoor airflow. The MSE assures proper accounting of both first-pass and "unused" recirculated outdoor air.

- **Operate** for adequate ventilation at all loads.

  Standard 62 mandates that we provide each space with enough ventilation air to adequately dilute accumulated contaminants. Since the amount of air delivered varies with thermal load, the outdoor air damper must modulate to provide proper ventilation airflow at part load.

- **Control** to minimize the energy impact of adequate ventilation.

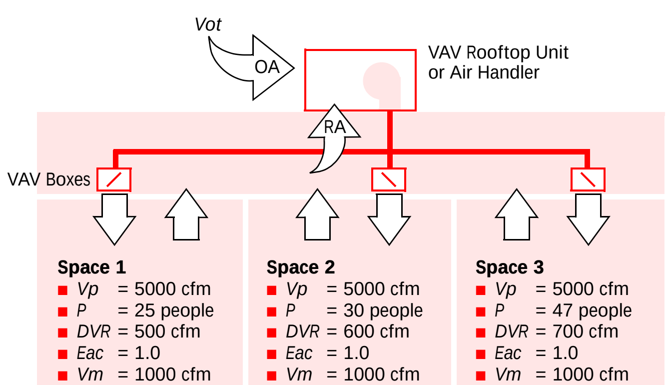
  Recognizing that outdoor airflow requirements vary with operating conditions, Standard 62-1989 advises use of a ventilation control scheme that assures proper outdoor airflow to every space at all load conditions.

Challenge 1: Design

Before we can find out required cooling and heating coil capacities, outdoor airflow must be determined. How much outdoor air is needed to ventilate a single-duct VAV system? By definition, a multiple-space system always includes spaces with differing ventilation needs. One space, the "critical space," receives precisely enough outdoor air for adequate ventilation; all others are overventilated. Consequently, air returning from the spaces includes "unused" outdoor air (suitable for ventilation) as well as "used" air (unsuitable for ventilation).

The amount of unused outdoor air in the exhaust air stream determines the efficiency of the ventilation system: the

Figure 1
Simple Three-Space, Single-Duct VAV System



higher the level of unused outdoor air, the less efficient the system. Accounting for **ventilation system efficiency** is critical to proper VAV system design. That's why Standard 62-1989 mandates a multiple-space equation for calculating the minimum **total outdoor airflow**, i.e. first-pass outdoor air introduced at the air handler, required for correct ventilation.

Space Ventilation. The first step is to find the design ventilation airflow, air change effectiveness and minimum primary airflow for each space based on its area, design population and design primary airflow. With that done, we can determine the worst-case ventilation fraction (at minimum primary airflow) for each space.

Design Ventilation Rate *(DVR)*. Using the Ventilation Rate Procedure defined in ASHRAE Standard 62-1989 to establish the *DVR* for each space, Figure 2, we find that:

- Space 1 is a 5,000-square-foot office area with five occupants per 1,000 square feet. Per Table 2 of the Standard, proper ventilation *(Vo)* means 20 cfm per person, so its *DVR* is 500 cfm: $DVR = 5 \times 5000 \div 1000 \times 20 = 500$ cfm.
- Space 2 is another office area that requires 20 cfm for each of 30 occupants. Its *DVR* is 600 cfm: $DVR = 30 \times 20 = 600$ cfm.
- Space 3 is an intermittently-used conference room that accommodates 75 people. Since the average occupancy is 47 and the per-person ventilation rate is 15 cfm, the *DVR* is 700 cfm: $DVR = 15 \times 47 = 700$ cfm.

Air Change Effectiveness *(Eac)*. This characteristic describes how well air supplied to the space mixes with air in the occupied zone. (Some air may pass directly from the supply-air discharge to the return-air intake.) Properly selecting diffusers and minimum primary airflow settings usually yields effective mixing, so let's assume complete mixing ($Eac = 1.0$) for each space in this example.

Minimum Primary Airflow *(Vm)*. In a single-duct VAV system, the primary airflow each space receives modulates in response to its thermal load, but never below a minimum amount when occupied. For this example, let's assume the minimum primary airflow must be at least 20 percent of design primary airflow ($Vm = 1000$ cfm) for each space.

Space Ventilation Fraction *(z)*. Each space receives the same fraction of ventilation air in its primary air stream, but requires a different fraction. Properly calculating the system's ventilation efficiency and first-pass outdoor airflow entails finding the worst-case (highest possible) ventilation fraction for each space. That condition occurs when the primary supply airflow is at its minimum

The Multiple-Space Equation ...

ASHRAE Standard 62-1989 requires that designers solve the multiple-space Equation 6-1 to determine the required ratio of first-pass outdoor airflow to primary (supply) airflow—i.e. $Y = X \div (1 + X – Z)$. With that ratio, we can then calculate the total outdoor airflow *(Vot)* required: $Vot = Y \times Vpt$.

Equation 6-1 can also be expressed in the following form (dubbed "MSE"). The divisor represents ventilation system efficiency.

MSE: $\boldsymbol{Vot = Von \div Ev}$

The variable names and definitions published here differ a bit from those in Standard 62-1989, allowing application to a broader range of VAV systems.

Other Equations

Von	=	$\sum DVR$	[1]
DVR	=	$P \times Vo$	[2]
Ev	=	$1 + X – Z$	[3]
X	=	$Von \div Vpt$	[4]
Vpt	=	$\sum Vp$	[5]
Z	=	largest *z*	[6]
z	=	$DVR \div (Eac \times Vp)$ $(DVR \div Eac) \div Vp$	[7] or

Equation Variables

DVR	=	space design ventilation requirement (cfm)
Eac	=	space air change effectiveness ratio (usually 1.0)
Ev	=	ventilation system efficiency (fractional)
P	=	population (persons)
Vm	=	minimum primary airflow (cfm)
Vo	=	space ventilation rate (cfm/*p*)
Von	=	total ventilation airflow (cfm)
Vot	=	total outdoor airflow (cfm)
Vp	=	space primary airflow (cfm); $Vp = Vd$ at design
Vpt	=	total primary airflow (cfm); $Vpt = Vdt$ at design
X	=	average ventilation fraction
Z	=	critical ventilation fraction
z	=	space ventilation fraction

Figure 2
Determining Total Outdoor Airflow With The MSE

		Space 1	Space 2	Space 3
Primary Airflow (cfm), *Vp*	=	5,000	5,000	5,000
Ventilation Rate (cfm), *DVR*	=	500	600	700
Minimum Primary Airflow (cfm), *Vm*	=	1,000	1,000	1,000
Ventilation Fraction, *zm*	=	0.50	0.60	0.70

$Vpt = DF \times \sum Vd = 10{,}000$ cfm
$Von = \sum DVR = 1{,}800$ cfm
$X = Von \div Vpt = 0.18$
$Z =$ largest $zm = 0.70$

Using the MSE ...
$Vot = Von \div (1 + X – Z)$
$= 1{,}800 \div (1 + 0.18 – 0.70)$
$= 3{,}750$ cfm

Design Tip ...

Establish a space minimum primary airflow that's 25 to 100 percent greater than the design ventilation rate. Using a Vm that equals the DVR (an all-too-common design mistake) means that the minimum primary airflow must consist entirely of first-pass outdoor air. This creates an added air-conditioning burden that's unnecessary and costly.

setting $(Vp = Vm)$. To calculate worst-case ventilation fraction zm for each space in Figure 2, divide the design ventilation rate by the minimum primary airflow; then correct for air change effectiveness: $zm = DVR \div (Eac \times Vm)$.

System Ventilation. Now, let's turn our attention to system-level needs. VAV systems require highest outdoor airflow at the lowest ventilation system efficiency. Lowest ventilation system efficiency occurs when the critical space requires minimum primary airflow (worst-case ventilation fraction) and total primary airflow Vpt is at design.

Using the space-related information determined above, we'll find worst-case ventilation fraction Z for the critical space and average ventilation fraction X; then lowest ventilation system efficiency Ev. Finally, we'll calculate worst-case (highest) minimum total outdoor airflow Vot using the MSE; see Table 1.

Critical Ventilation Fraction *(Z)*. The space with the highest ventilation fraction at minimum primary airflow becomes the ventilation-critical space for the system, defining the critical ventilation fraction (Z = largest zm). The critical space needs the highest fraction of outdoor air in its primary air stream. In our example, Space 3 has the highest ventilation fraction at minimum primary airflow, so the critical ventilation fraction becomes 0.70 (i.e. $Z = zm3 = 0.70$).

Average Ventilation Fraction *(X)*. Before we can find the lowest ventilation system efficiency, we need to determine the system's average ventilation fraction.

We'll begin by calculating the total ventilation airflow, Von, required for the system. That's simply a matter of adding the individual space design ventilation rates: $Von = \Sigma DVR = 1{,}800$ cfm.

Next, we'll determine total primary airflow Vpt when ventilation system efficiency is lowest. (This happens when the critical space is at minimum primary airflow and all other spaces are at or near design primary airflow.) For most VAV systems, lowest ventilation system efficiency occurs at design primary airflow, and it's usually determined by accounting for load diversity; i.e. $Vpt = Vdt = DF \times \Sigma Vd$, where load diversity factor DF is defined as Block Load ÷ Peak Load. For our example, let's assume a load diversity factor (DF) of 0.67 to find the total primary airflow at design: $Vpt = Vdt = DF \times \Sigma Vd = 0.67 \times 15{,}000 = 10{,}000$.

Now we can calculate average ventilation fraction X as the ratio of total ventilation airflow to total primary airflow: $X = Von \div Vpt$. For our example, average ventilation fraction $X = 1{,}800 \div 10{,}000 = 0.18$.

Ventilation System Efficiency *(Ev)*. We now have enough information to find the ventilation system efficiency. For single-duct VAV systems, Ev is defined in terms of average and critical ventilation fractions; i.e. $Ev = 1 + X - Z$. For this example, ventilation system efficiency is 0.48 (i.e. $Ev = 1 + 0.18 - 0.70$). In other words, the ventilation system uses only 48 percent of the first-pass outdoor air introduced for ventilation; the other 52 percent exhausts from the building without actually diluting contaminants.

Total Outdoor Airflow *(Vot)*. Finally, we can calculate the required worst-case total outdoor airflow, accounting for ventilation system efficiency: $Vot = Von \div Ev = 1800 \div 0.48 = 3{,}750$ cfm.

Notice that the percentage of **first-pass** outdoor air required in the primary air stream (3,750 ÷ 10,000 or 37 percent) is **less** than the percentage of ventilation airflow required by the critical space (70 percent). Is the critical space properly ventilated? Yes, because the unused outdoor air in the recirculated air stream provides the balance of the ventilation air needed in the critical space.

Table 1

System Load	Characteristic	Space 1	Space 2	Space 3	Outdoor Air, *Vot*, Required Per ASHRAE 62-1989
100%	Primary Airflow, *Vp*	5,000 cfm	4,000 cfm	1,000 cfm	37% or 3,750 cfm
	Ventilation Rate, *DVR*	500 cfm	600 cfm	700 cfm	
	Ventilation Fraction, *z*	0.100	0.150	0.700	
	Avg Ventilation Fraction		0.180		
70%	Primary Airflow, *Vp*	1,000 cfm	3,000 cfm	3,000 cfm	34% or 2,380 cfm
	Ventilation Rate, *DVR*	500 cfm	600 cfm	700 cfm	
	Ventilation Fraction, *z*	0.500	0.200	0.233	
	Avg Ventilation Fraction		0.257		
35%	Primary Airflow, *Vp*	1,000 cfm	1,500 cfm	1,000 cfm	63% or 2,210 cfm
	Ventilation Rate, *DVR*	500 cfm	600 cfm	700 cfm	
	Ventilation Fraction, *z*	0.500	0.400	0.700	
	Avg Ventilation Fraction		0.514		

Notice, too, that the percentage of **first-pass** outdoor air in the primary air stream (37 percent) is higher than average ventilation airflow (18 percent) required for the system. Is the system overventilated? No, because the critical space needs significantly higher ventilation airflow than the system average; the non-critical spaces are overventilated and return unused air to the air handler for recirculation.

Now, using worst-case total outdoor airflow *Vot,* the cooling and heating coils can be selected to accommodate both space and ventilation (outdoor air) loads.

Challenge 2: Operate

The design procedure just described identifies the worst-case total outdoor airflow. What happens to ventilation requirements during normal **operation**? Let's find out by examining our example single-duct VAV system at three load conditions: full (100 percent), 70 percent and 35 percent. See Table 2, Column A.

Full Load. At full load on a sunny summer afternoon, assume Space 1 requires design airflow ($Vp = 5{,}000$ cfm), Space 2 requires 80 percent of design airflow ($Vp = 4{,}000$ cfm) and Space 3 requires less than 20 percent of design airflow ($Vp = Vm = 1{,}000$ cfm). Full-load (design) ventilation system efficiency, calculated earlier ($Ev = 0.480$), results in a worst-case total outdoor airflow *(Vot)* of 3,750 cfm. For proper system ventilation at design, the air handler must provide primary air that contains 37.5 percent first-pass outdoor air.

70 Percent Load. On a cool summer morning, the load experienced by our example system may be only 70 percent. At this condition, assume Space 1 requires 20 percent of design airflow, while Spaces 2 and 3 each require 60 percent. To provide proper system ventilation at this part-load condition, we must determine these values:

- The **current** ventilation fraction for each space. In Space 1, $z = 500 \div 1{,}000 = 0.500$; in Space 2, $z = 600 \div 3{,}000 = 0.200$; and in Space 3, $z = 700 \div 3{,}000 = 0.233$.
- The **current** critical space ventilation fraction. In this case, Z = Space 1 $z = 0.500$.
- The **current** average ventilation fraction, i.e. $X = 1{,}800 \div 7{,}000 = 0.257$.

The system achieves a higher ventilation system efficiency at this load condition than at full load ($Ev = 1 + 0.257 - 0.500 = 0.757$). Since average ventilation fraction X increases from 0.180 to 0.257, less overventilation occurs in non-critical spaces so less unused outdoor air exhausts. The higher ventilation efficiency means that the system requires less total first-pass outdoor airflow ($Vot = 1{,}800 \div 0.757 = 2{,}380$ cfm) even though space design ventilation rates *(DVR)* remain unchanged.

For proper ventilation at 70 percent load, 34 percent of the total primary airflow delivered by the air handler must be first-pass outdoor air.

35 Percent Load. Now let's consider the 35 percent load of an autumn day. Assume Spaces 1 and 3 each require 20 percent of design airflow, while Space 2 requires 30 percent. As was true for the 70 percent load condition, we must account for the **current** ventilation system efficiency to assure proper ventilation.

This time, the ventilation fraction for Space 1 is $z = 500 \div 1{,}000 = 0.500$; for Space 2, it's $z = 600 \div 1{,}500 = 0.400$; and in Space 3, it's $z = 700 \div 1{,}000 = 0.700$. Since Space 3 has the highest ventilation fraction, critical space ventilation fraction $Z = 0.700$. Using this value and the current average ventilation fraction ($X = 1{,}800 \div 3{,}500 = 0.514$), we can calculate the ventilation system efficiency at this condition: $Ev = 1 + X - Z = 0.814$.

Table 2

System Load	Characteristic	Space 1	Space 2	Space 3	Outdoor Air, *Vot*, Required Per ASHRAE 62-1989	Outdoor Air, *Vot*, Delivered: Fixed Damper At "N"	Fixed Flow At Max	Ventilation Reset
					A	B	C	D
100%	Primary Airflow, *Vp*	5,000 cfm	4,000 cfm	1,000 cfm	37% or 3,750 cfm	18% or 1,800 cfm	37% or 3,750 cfm	37% or 3,750 cfm
	Ventilation Rate, *DVR*	500 cfm	600 cfm	700 cfm				
	Ventilation Fraction, *z*	0.100	0.150	0.700				
	Avg Ventilation Fraction, *X*		0.180					
70%	Primary Airflow, *Vp*	1,000 cfm	3,000 cfm	3,000 cfm	34% or 2,380 cfm	18% or 1,260 cfm	54% or 3,750 cfm	34% or 2,380 cfm
	Ventilation Rate, *DVR*	500 cfm	600 cfm	700 cfm				
	Ventilation Fraction, *z*	0.500	0.200	0.233				
	Avg Ventilation Fraction, *X*		0.257					
35%	Primary Airflow, *Vp*	1,000 cfm	1,500 cfm	1,000 cfm	63% or 2,210 cfm	18% or 630 cfm	100% or 3,500 cfm	63% or 2,210 cfm
	Ventilation Rate, *DVR*	500 cfm	600 cfm	700 cfm				
	Ventilation Fraction, *z*	0.500	0.400	0.700				
	Avg Ventilation Fraction, *X*		0.514					

Again, the system achieves greater ventilation system efficiency since the average ventilation fraction is high (X = 0.516). The result is less overventilation in non-critical spaces; therefore, less unused outdoor air in the exhaust.

Total first-pass outdoor airflow drops ($Vot = 1{,}800 \div 0.814 = 2{,}210$ cfm) even though space design ventilation rates *(DVR)* haven't changed.

For proper ventilation at 35 percent load, the primary air stream must include 63 percent first-pass outdoor airflow.

Fixed OA Damper. Traditionally, most VAV system designers set the outdoor air (OA) damper to a fixed position and allow actual outdoor air intake to vary with system primary airflow. Can the ventilation requirement be met with a fixed-position outdoor air damper?

Suppose we specify a minimum fixed OA damper position yielding a design outdoor airflow that reflects the average system ventilation requirement. In other words, we calculated average ventilation fraction X by summing the space ventilation rates *(Von)* and dividing by the design primary airflow *(Vpt).* We then set the OA damper position to provide this fraction of outdoor air at all load conditions. **This is a common design mistake!** Table 2, Column B, illustrates why …

Even though the system requires 37 percent (3,750 cfm) outdoor air at full load for proper ventilation, the fixed position of the OA damper delivers only 18 percent (1,800 cfm). At a 70 percent load, the system delivers 1,260 cfm (i.e. 18 percent of 7,000 cfm) while it needs 2,380 cfm. And at a 35 percent load, it delivers only 630 cfm from outside when 2,210 cfm is needed. In effect, fixing the OA damper to introduce 18 percent outdoor air at any primary airflow **underventilates** all spaces at all load conditions.

Fixed Outdoor Airflow. It takes the right **volume**, not percentage, of outdoor air to meet the ventilation requirement in ASHRAE Standard 62-1989. Can we achieve compliance by delivering a constant volume of outdoor air?

Suppose we specify a minimum fixed outdoor airflow that introduces the design outdoor air volume at all load conditions. In other words, we calculated ventilation system efficiency *(Ev)* and total outdoor airflow *(Vot)* at design, then provided the means to sense outdoor airflow and maintain it at the desired value. This strategy requires accurate sensing and control of outdoor airflow, and assumes a constant low ventilation system efficiency ($Ev < 1.0$) at both design and part-load conditions.

Table 2, Column C, shows the effect on our example system. Total outdoor airflow *(Vot)* is sensed and maintained at the design value (3,750 cfm), and the system is properly ventilated at full load. But when the load is 70 percent of design, the system delivers 3,750 cfm of outdoor air when it needs only 2,380 cfm. And, at 35 percent load, the system delivers 3,500 cfm of outdoor air when it requires only 2,210 cfm. (Since primary airflow is less than 3,750 cfm, all primary air is first-pass outdoor air.) In short, maintaining total outdoor airflow *Vot* at the worst-case condition yields proper ventilation at full load, but **excessive** (and costly!) **ventilation** at all part-load conditions.

This begs another obvious question: Can the total outdoor airflow setpoint be reset at part-load conditions to match actual ventilation requirements? The answer lies in **control**.

Challenge 3: Control

The previous **design** procedure yielded worst-case total outdoor airflow. But as we've seen, ventilation system efficiency goes up during normal **operation** since multiple-space systems exhaust less unused air at part load. That means we can meet part-load ventilation requirements with less total outdoor airflow. As designers, we're challenged to devise a system that **controls** ventilation at part load, assuring proper ventilation without overventilating and unnecessarily increasing operating costs.

ASHRAE Requirements. ASHRAE Standard 62-1989 states that *"… when mechanical ventilation is used, provision for airflow measurement should be included,"* and that for VAV systems, *"… provision shall be made to maintain acceptable indoor air quality throughout the occupied zone."* Here, we can interpret that "provision" for airflow measurement is an outdoor airflow sensor and that "provision" for maintaining acceptable IAQ is a control scheme that delivers proper ventilation to all spaces at all loads.

With regard to multiple-space system ventilation, ASHRAE Standard 62-1989 also states: *"When more than one space is served by a common supply system … the system outdoor air quantity shall then be determined using Equation 6.1"* and the multiple-space equation (MSE).

In other words, proper ventilation for multiple-space VAV systems must be determined using the MSE; outdoor airflow measurement and control are encouraged. It follows that a control scheme that senses actual space airflow and solves the MSE dynamically to reset

Design Tip …

Many indirect (e.g. temperature-based) airflow-sensing schemes are unable to meet the 10 percent accuracy criterion typical of air balancing requirements. Direct-sensing methods such as independent airflow-monitoring stations can be costly (require field-installation and -calibration) or difficult to apply (short outdoor air ducts often limit their use). Several air handler manufacturers offer an alternative: factory-installed and -calibrated airflow sensors that require very little straight duct, reduce installation costs and sense accurately over a wide range of temperatures and airflows.

the total outdoor airflow setpoint *(Vot)* will satisfy Standard 62's ventilation requirements. The term coined for this control scheme is **ventilation reset**.

Basic Ventilation Reset. Though space ventilation requirements remain constant, our example load conditions illustrate that decreased primary airflow leads to increased ventilation system efficiency. Since ventilation system efficiency increases at part load, the total first-pass outdoor airflow required to properly ventilate all spaces drops (from worst-case) as the load decreases.

Ventilation reset control takes advantage of increased ventilation system efficiency at part load to reduce the system's operating cost. A basic ventilation reset control scheme senses airflow in each space and uses these values to calculate the current required airflow setpoint (i.e. solves the MSE to find *Vot*). The control system then senses and modulates outdoor airflow to maintain the new setpoint.

Implementation. Practicing ventilation reset in a single-duct VAV application requires communicating controls throughout the system. Specifically, it requires VAV terminal units with pressure-independent (DDC/VAV) controllers, a building automation system (BAS) with simple equation-solving capability, and an air handler with a DDC controller and the means for sensing outdoor airflow and maintaining it at a setpoint value. See Figure 3.

In operation, individual DDC/VAV controllers sense the primary airflow *(Vp)* in the spaces they serve. Each controller includes a setting for its design ventilation requirement *(DVR)*. Using sensed primary airflow, each DDC/VAV controller (or the BAS) calculates the current space ventilation fraction: $z = DVR \div Vp$ (if air change effectiveness, *Eac,* is 1.0). Each controller must also include a **minimum airflow setpoint** *(Vm).* That value determines the required worst-case outdoor airflow and is used to size the cooling coil during the design process.

The BAS continuously polls each DDC/VAV controller to collect ventilation data from every space, including primary airflow *Vp,* design ventilation requirement *DVR,* and ventilation fraction *z*. It then calculates total primary airflow $(Vpt = \sum Vp)$, total design ventilation airflow $(Von = \sum DVR)$, and the critical space ventilation fraction $(Z = \text{largest } z)$. Using these values, it finds the average ventilation fraction $(X = Von \div Vpt)$, then the ventilation system efficiency $(Ev = 1 + X - Z)$, and finally, a new outdoor airflow setpoint $(Vot = Von \div Ev)$ that it sends to the air handler controller.

When the air handler's DDC controller receives this value, it modulates the OA damper as required to maintain outdoor airflow at the new setpoint.

What's It Worth? Ventilation reset control saves operating costs, but may increase the first cost of the system. Can it be justified?

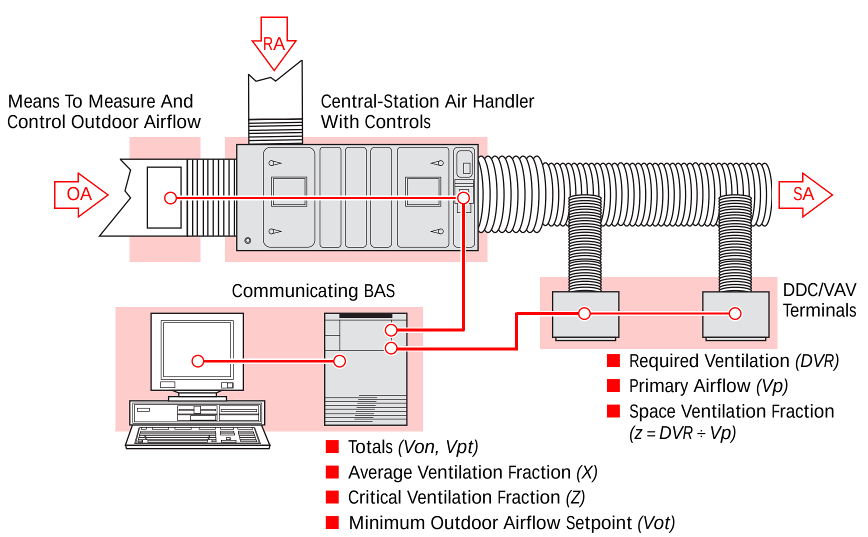
We used the TRACE® building energy and economic analysis software to model a three-story building with a single-duct VAV system. We compared the HVAC operating costs of three different ventilation schemes in four geographic locations:

- The simple, traditional fixed-damper approach, which underventilates at all loads and doesn't meet the ventilation requirements of ASHRAE Standard 62-1989.
- Controlled outdoor airflow fixed at the worst-case value, which overventilates and exceeds ASHRAE requirements. And …
- Ventilation reset, which meets ASHRAE requirements without overventilation (Table 2, Column D).

As Figure 4 shows, improper ventilation costs the least to operate, worst-case ventilation costs the most to operate, and properly controlled (reset) ventilation approaches the low cost of improper ventilation. Notice, too, that the value of ventilation reset changes dramatically with geographic location.

Enhanced Ventilation Reset. With the equipment, controls and information-processing system for basic ventilation reset in place, several

Figure 3
Air Handling System With Ventilation Reset



advanced control schemes become feasible. A brief discussion of them follows.

Critical Minimum Reset. During normal operation, the primary airflow to the critical space **significantly** impacts ventilation system efficiency and required total outdoor airflow. Why?

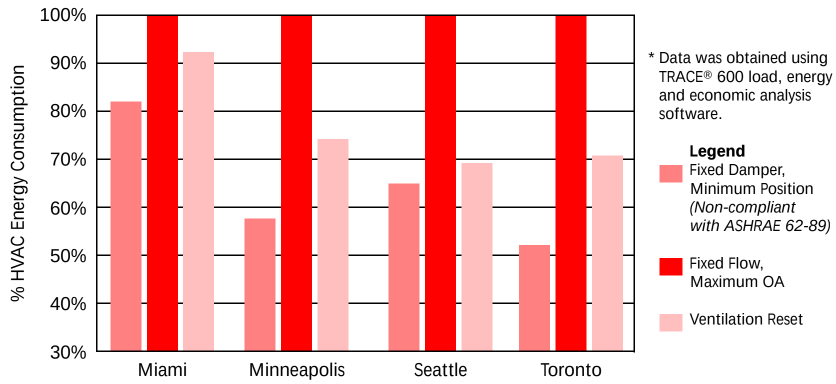
Recall that efficiency depends on the critical ventilation fraction ($Ev = 1 + X - Z$) and that this fraction, in turn, depends on the critical space primary airflow ($Z = z = DVR \div Vp$). So, raising the critical space primary airflow lowers the critical ventilation fraction which, in turn, raises ventilation system efficiency.

Primary airflow to the critical space can be increased simply by resetting the minimum primary airflow setting *(Vm)* upward. Of course, since the space thermostat requires less than the new minimum primary airflow, the supply air to the space must be tempered (reheated) to assure thermal comfort. Usually, the operating-cost savings of reduced outdoor airflow dwarfs the operating-cost penalty of tempering.

Predicted Occupancy Reset. Single-duct VAV systems often include a mixture of occupied and unoccupied spaces, all served by the same air handler. Many spaces exhibit predictable occupancy patterns … patterns so predictable that the BAS can be programmed with an occupancy schedule.

An unoccupied space typically needs no ventilation or primary airflow. Scheduling one or more spaces as “unoccupied” decreases total ventilation airflow *(Von)* slightly and total primary airflow *(Vpt)* more significantly. Consequently, the

Figure 4
A Comparison Of HVAC Operation*



* Data was obtained using TRACE® 600 load, energy and economic analysis software.

average ventilation fraction *(X)* increases which, in turn, increases ventilation system effectiveness *(Ev)* and decreases total outdoor airflow *(Vot)*. Operating-cost savings related to outdoor air preconditioning can be significant, especially for spaces with high occupant density and predictable patterns like churches and classrooms.

Estimated Occupancy Reset. Sensing or estimating actual occupancy can improve the “predicted occupancy reset” scheme just described. An accurate estimate of the current population *(P)* in a space can be used to reset the “design” ventilation rate to a lower level; recall that $DVR = P \times Vo$. An office designed for 30 people ($DVR = 30 \times 20 = 600$ cfm) but occupied by 10 requires only one third of the design ventilation rate: $DVR' = 10 \times 20 = 200$ cfm. Reducing the design ventilation rate in one or more spaces decreases the total ventilation airflow. It also yields a lower average ventilation fraction, a slightly lower ventilation system efficiency and a significant decrease in total outdoor airflow.

As we’ve already illustrated, operating-cost savings can be significant at partial occupancy, especially for spaces with highly variable populations such as theaters and airport terminals.

In A Nutshell …

- You **can** design single-duct VAV systems that meet ASHRAE Standard 62-1989 requirements for proper ventilation. Just be sure to use the multiple-space equation.
- Supply airflow varies during normal operation, so proper ventilation means maintaining the required volume. No more fixed dampers!
- Well-thought-out control schemes can minimize the operating-cost impact of proper ventilation at part load. Make sure your control schemes dynamically solve the multiple-space equation for current operating conditions. ■

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If you’d like to comment on this article, send a note to The Trane Company, Engineers Newsletter Editor, 3600 Pammel Creek Road, La Crosse WI 54601 or visit Trane on the Internet at www.trane.com.

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Only As Good As The Sum Of Its Parts

VariTrane® Air Valve. Truly the best modulation device available, this rugged valve is the only device on the market that provides **linear** airflow control. And that ultimately means improved comfort.

Flow Ring. This patented device accurately measures airflow regardless of inlet duct conditions. Our flow ring's patented design provides multiple flow-sensing points in multiple axes. So, whether the inlet duct enters the side, top or works its way around a construction beam, the flow ring will precisely detect the rate of airflow.

ARI 880-Certified. All Trane terminal units are certified by the Air Conditioning and Refrigeration Institute. That's the only way to assure that the performance you expect is the performance you get.

UL- And CUL-Approved. It's essential to safeguard the occupants of your building as well as your business. UL and CUL approval of Trane terminals units signify that these products conform to industry-established safety standards.

Leadership In …
Innovative Design

The Trane air valve design targets **total** comfort. And that's more than a comfortable environment for building occupants. It also means a quiet, efficient product that's rugged and reliable.

Leadership In …
Indoor Air Quality

Indoor air quality is of major concern today. To help prevent the terminal units from becoming a source of contaminants, we offer the most complete line of IAQ options of any terminal unit available. Our insulation options include standard matte-faced insulation, optional foil-faced insulation to meet NFPA 90A, UL 181 and bacteriological standard ASTM C665 requirements, and metal double-wall construction for unsurpassed durability and easy cleanup.

Leadership In …
Controls

Direct Digital Controls. Direct digital control of VAV units, DDC/VAV, is key to providing intelligent, responsive building management. A single, twisted-wire pair links the factory-mounted VAV microprocessor controls to Tracer® building management products. Trane provides more DDC/VAV terminals than anyone else in the world!

Advanced Controller. Designed **specifically** for VariTrane terminals, Trane DDC/VAV controller is incredibly reliable: its circuit-board failure rate is less than one percent! System layout options, control flexibility and user-friendly access further enhance the DDC/VAV control capabilities of our terminals.

The VariTrane product line also includes the most complete offering of pneumatic and electronic controls in the HVAC industry.

Putting It All Together …
Systems That Work!

A control strategy is only as accurate and reliable as the components and people underlying it. Our patented air valve and complete line of controls, alone, didn't earn Trane its leadership status. It meant combining these assets with the highest quality components and the best sales and service staff in the VAV industry.

For more information about Trane VAV terminals … or about any of our broad line of products, systems and services … contact your local Trane sales engineer.

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