Ventilation and Fan Pressure Optimization for VAV Systems

Meeting today's ventilation and comfort requirements in multiple-space VAV systems can be challenging. Building heating and cooling loads are typically not static; therefore VAV systems need to be flexible to respond to the continually changing outdoor conditions and occupant activities. To optimize comfort and minimize energy usage in this dynamic environment, the HVAC control system must also be dynamic. An effective way to assure that the comfort and ventilation needs for all areas of the building are met is to continuously monitor system operation. The system then must then adjust control parameters as necessary to meet the needs for each specific operating condition. This type of system level control can accomplish comfort objectives while keeping operating energy costs in check. The key to optimizing a building's comfort system is factory-mounted, communicating controls on the HVAC equipment.

This Engineering Bulletin discusses how a Trane Integrated Comfort™ system can effectively control building ventilation and supply fan pressure for increased comfort and good indoor air quality (IAQ) while keeping system operating energy costs at the lowest possible levels.

Ventilation Reset Outdoor Air Control For VAV Systems

The Challenges of Ventilation Control

During operation, the typical VAV system delivers a mixture of outdoor air and recirculated air to the multiple spaces it serves. The volume of conditioned air that enters the space is controlled by a space temperature sensor. It's important to remember that the ventilation requirements for a space remain constant as the supply air varies with thermal load. The challenge is to maintain the proper amount of ventilation air to each individual space while varying the supply air to the spaces. To accomplish this, the percentage or 'richness' of the ventilation air in the total supply air must vary with system operating conditions.

ASHRAE Standard 62-1989, *Ventilation for Acceptable Indoor Air Quality*, is considered the "standard of care" for the design of ventilation systems for commercial buildings. Section 6.1 of the Standard details the **Ventilation Rate Procedure** as "a prescriptive method of compliance that relies on adequate dilution of indoor space contaminants to achieve acceptable indoor air quality (IAQ)."

Two ventilation rates must be considered to successfully apply the Ventilation Rate Procedure. First, consider the minimum ventilation rate required for each individual space and second, consider the total amount of outdoor air required at the air handier. Minimum ventilation to each space is based on the type of space (e.g. office, classroom, etc.) and the design occupancy of the space. But how does the designer determine the amount of outdoor air required at the air handler for the system? Traditionally, the sum of all the individual space ventilation requirements has been used as the system OA rate. This approach does not meet ASHRAE 62-89 and can result in zones being underventilated, even at full load conditions. The situation becomes even more severe when the system is operating at part load conditions.

The proper method is to use **Equation 6.1 of ASHRAE 62-89**. This equation allows the designer to determine the amount of outdoor air required at the air handier in multiple-space systems such as VAV systems. It uses the individual space ventilation requirements and allows the designer to take credit for "unused" ventilation air returning from the overventilated spaces in the system. This reduces the amount of outdoor air for an air handling system while adequately ventilating each zone. Equation 6.1 is the important link between multiple zone ventilation requirements and the central air handling unit outside airflow.

ASHRAE 62-89, Equation 6-1 is:

$$Y = X / (1 + X - Z)$$

Substituting gives us:

$$V_{ot} = V_{on} / (1 + X - Z)$$

WHERE:

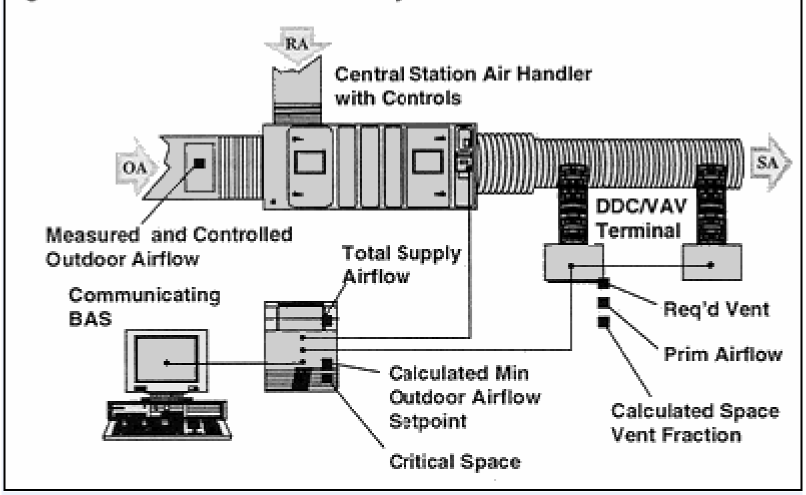
X =	V_{on}/V_{st} = uncorrected outdoor airflow fraction
Y =	V_{ot}/V_{st} = corrected outdoor airflow fraction
Z =	V_{oc}/V_{sc} = critical space ventilation fraction
F =	V_o/V_s = space ventilation fraction
V_{on} =	total ventilation airflow (sum V_o)
V_o =	space ventilation airflow
V_s =	space primary airflow
V_{oc} =	critical space ventilation airflow
V_{sc} =	critical space primary airflow
V_{st} =	total supply airflow (sum V_s)
V_{ot} =	required system outdoor airflow

How Does Ventilation Reset Work?

The DDC/VAV terminal units continuously monitor the amount of *delivered primary airflow to the space* (V_s) and communicates this information to the building automation system (BAS). The BAS system knows the *design space ventilation airflow* (V_o) requirements for each zone and can thus calculate the percentage of ventilation air in the supply air required at each VAV terminal unit. This is known as the *ventilation fraction (F)*. The largest space ventilation fraction (F) is determined to be the *critical space ventilation fraction (Z)*. The *total supply airflow* (V_{st}) for the air handling unit is determined by summing the *space primary airflow* (V_s) to each VAV terminal and adjusting for duct leakage. The *total ventilation airflow* (V_{on}) for the air handling unit is determined by summing the *space ventilation airflows* (V_o) required by each VAV terminal. Using this information, Equation 6.1 is then solved to dynamically determine the required amount of outdoor airflow that must be introduced into the system at the air handler and the BAS sets the outdoor airflow setpoint to that value. Recalculation of these values is repeated every 15 minutes.

The air handler, in turn, constantly monitors the amount of outdoor air entering the system and resets the system outdoor airflow damper in response to the current outdoor airflow setpoint as determined by the BAS. This assures that **all zones** are properly ventilated at **all load conditions** while **minimizing wasteful overventilation**. Figure 1 below graphically depicts the Ventilation Reset system.

Figure 1 - Ventilation Reset for VAV Systems



Ventilation Control Method Comparison
Figure 2 compares the various methods of controlling outdoor ventilation air. A simple three zone VAV system at three different load points is illustrated in this example. Notice that although the amount of supply air varies, the amount of ventilation for the zones remains constant. This results in the fraction, that is the amount of OA in the supply air, changing at each different operating condition.

Figure 2 - Outdoor Air Control Method Comparison

SYSTEM OPERATING CONDITION	OA REQ'D	OA DELIVERED			
	ASHRAE 62-1989. EQ. 6.1	FIXED DAMPER: MINIMUM OA METHOD	FIXED DAMPER: ASHRAE METHOD	FLOW CONTROL, MAX. OA	WITH VENT'LN RESET
Design Load: SA CFM= 3000 = 1000 +1000 +1000; OA CFM= 700 = 200 + 200 + 300; FRACTION = .20 .20 .30	25% or 750 CFM	~~23% or 700 CFM~~	67% or 2010 CFM	31% or 933 CFM	25% or 750 CFM
Part Load: SA CFM= 2200 = 800 + 800 + 600; OA CFM= 700 = 200 + 200 + 300; FRACTION = .25 .25 .50	39% or 856 CFM	~~23% or 506 CFM~~	67% or 1474 CFM	42% or 933 CFM	39% or 856 CFM
Part Load: SA CFM= 1400 = 500 + 500 + 400; OA CFM= 700 = 200 + 200 + 300; FRACTION = .40 .40 .75	67% or 933 CFM	~~23% or 322 CFM~~	67% or 933 CFM	67% or 933 CFM	67% or 933 CFM

Does not meet ASHRAE 62-89!

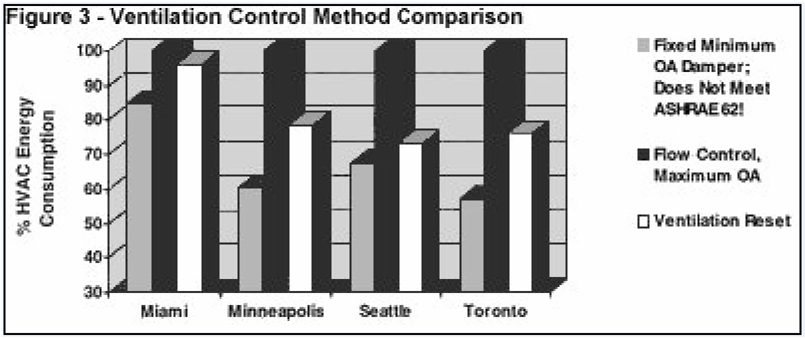
The first column illustrates the **Outdoor Air Required** at the air handler for these different operating conditions based on Equation 6.1 from ASHRAE 62-1989. It's important to note that the amount of ventilation air actually increases at part load as some of the zones throttle causing the system's ventilation demand to become unbalanced.

The next two columns depict the amount of OA that would be provided if a conventional fixed OA damper is used. The column labeled **Fixed Damper, Minimum OA Method** is shown for reference purposes only. IT DOES NOT MEET ASHRAE 62-89 and thus is not a viable OA control method. This method, which was typically used in the past, uses the sum of the individual space ventilation requirements to determine the amount of OA required at the air handler. The OA damper is then fixed at the position corresponding to that flow. This approach is no longer acceptable for systems serving multiple spaces, such as VAV systems, because it does not account for the OA needs for each zone and due to the airflow being a fixed percentage, it does not follow the building load typically resulting in a considerable shortage of OA at part load and/or unbalanced system conditions.

The **Fixed Damper, ASHRAE Method** shown in the next column, uses the same fixed damper methodology; however, the OA damper position is locked into a position representing the highest OA or "worst case" amount- which in this case is 67 percent. Using this worst case approach MEETS ASHRAE 62-89 by assuring that ALL zones are properly ventilated at ALL load conditions. However, as you can see, it can also result in extreme over-ventilation at other load conditions and thus unnecessarily drives up system operating energy costs.

The column labeled **Flow Control, Maximum OA** represents an important step toward flow-based ventilation control. By adding the ability to measure and reset the outdoor airflow through the use of a Trane Traq™ damper or an air monitoring station, we can now control to a constant OA airflow (cfm) to satisfy our ventilation requirements rather than just a fixed damper position. It allows the system to meet the OA requirements of ASHRAE 62 at all load conditions while significantly reducing the over-ventilation at the higher load points. The last column depicts the OA provided using the **Ventilation Reset** control strategy. Notice that by dynamically calculating the actual amount of OA required at each specific operating point, the exact amount of ventilation air required is brought into the system with NO wasteful over-ventilation. The result - a properly ventilated VAV system with the lowest possible operating cost.

A Real World Example
The example we used in Figure 2 to explain the different control methods was purposely kept quite simple and straightforward. You might ask: Is it realistic to expect these savings in real world situations in buildings with many zones? Does the location of the building affect the savings potential? In Figure 3 a typical 3-story, 60,000 square foot office building was modeled using TRACE 600® with various ventilation control methods for four different cities in North America. The system has 24 zones and uses a forward-curved fan with a variable-speed drive, a comparative-enthalpy economizer and series fan-powered VAV terminals. This example compares **total building HVAC energy** usage for each different control method.

Figure 3 - Ventilation Control Method Comparison


The middle bar serves as the baseline for this comparison. It represents the **Flow Control, Maximum OA** method in which the volume of OA (cfm) is controlled to the constant flow rate which properly ventilates the system at the worst case operating condition. (This condition was represented by 933 cfm in our previous example shown in Figure 2). The bar on the right, represents the system HVAC energy consumption using the **Ventilation Reset** method in which the OA volume is dynamically controlled to match the actual HVAC system operating conditions. System operating energy cost saving potential is of course, dependent on the building location, construction and operating profile, e.g., number of hours in cooling, heating, etc. Based on our assumptions, the HVAC energy savings determined by our TRACE 600® analysis ranged from approximately 5 percent in Miami to over 27 percent in Seattle!

The **Fixed Minimum OA Damper** method, depicted by the bar on the left, is shown for reference purposes only to represent the effect of traditional ventilation methods of the past on system energy costs. This method **does not comply with ASHRAE 62** and **is not viable**.

Hardware and Control Requirements

The Ventilation Reset control strategy requires the following equipment and control capabilities:

- The ability to measure/control the OA brought into an the air handler such as Traq™ dampers.
- VAV terminals with pressure independent DDC controls
- A building automation system capable of solving ASHRAE Equation 6.1 such as Trane Tracer Summit®.

Supply Fan Pressure Optimization

The Challenges of Supply Fan Pressure Control

An important benefit of a Variable Air Volume (VAV) system is the potential for reduced operating cost at part load conditions. The lower operating cost at part load is a result of the supply fan having to move less air and thus requiring less fan horsepower.

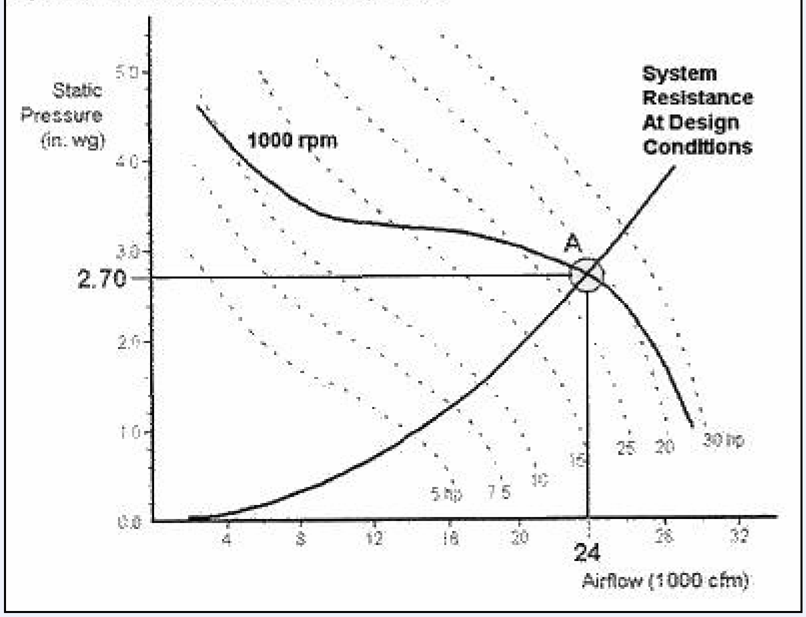
Traditionally, fan capacity control is accomplished through a pressure control loop consisting of a static pressure sensor, a pressure controller and a means of fan control (either inlet guide vanes or a variable speed drive). Based on an input from a duct pressure sensor located in the duct system, the pressure controller modulates the inlet guide vanes or fan speed to maintain a constant pressure at the sensor.

Various methods of fan pressure control are used today. Most reduce operating energy costs to some extent, however determining the static pressure sensor location and static pressure setpoint can be difficult. Typically, worst-case conditions are used which result in higher than necessary operating pressures (and thus costs). All other conditions or energy savings are the primary target resulting in some critical spaces becoming starved for air at some conditions. The challenge: **Control the VAV system to maximize part-load energy savings without sacrificing zone comfort.**

Fan Pressure Control Methods

To determine the effects of the various pressure control methods on fan energy, let's use a simple VAV system as an example. The example consists of a forward-curved (FC) fan with inlet guide vanes operating at design conditions of- 1000 rpm, 24,000 cfm assuming 2.70 in. wg system resistance. As shown in Figure 4, the fan operates at the intersection of the fan curve and the system resistance curve (point A) using approximately 23 brake horsepower (bhp). The following examples will look at three different methods of controlling the duct pressure at the part load operating point of 18,000 cfm.

Figure 4 - Example System at Design Load

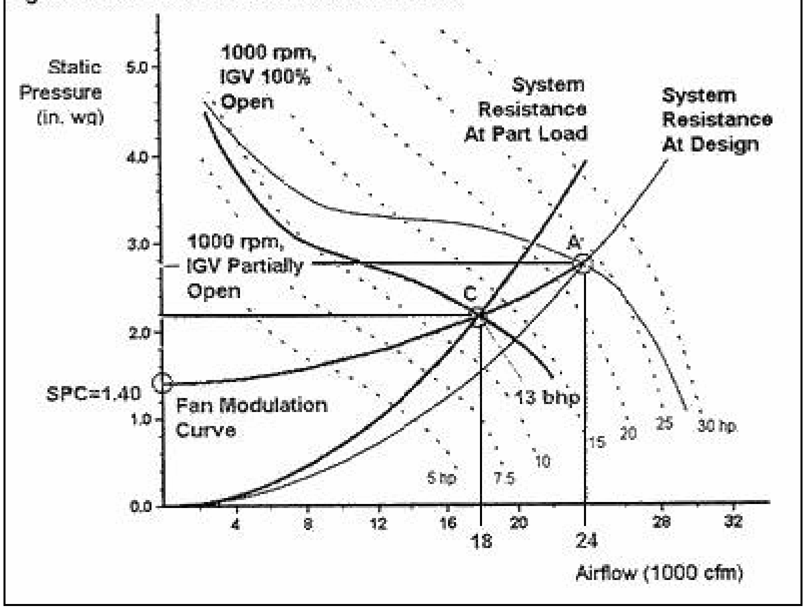


Fan Outlet Static Control

It is often convenient to mount the duct pressure sensor at the supply fan outlet and set the static pressure controller to maintain the static pressure required at design flow. The appeal of this method is that the pressure sensor can be easily installed in the factory assuring relatively high reliability because it can be factory tested. Reliable sensor operation entails proper location and signal routing. If fire dampers are included in the supply duct, sensing at the fan outlet assures that the sensor is on the fan side of the damper so the duct is protected from high pressures. Also, depending on the design and layout of the duct system, this method may eliminate the need for multiple duct-mounted sensors.

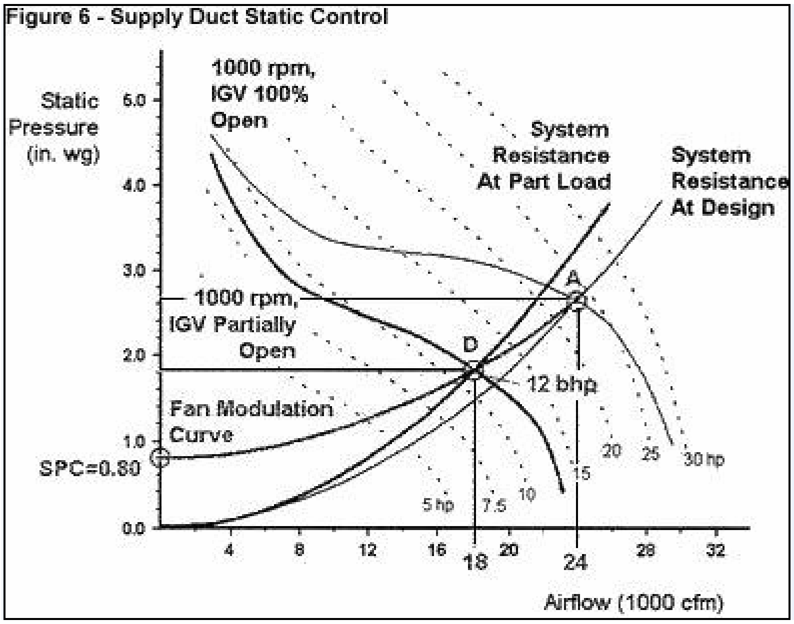
Going back to our example system with the pressure sensor located at the outlet of the fan, the static pressure controller (SPC) must be set to 1.4 in. wg so that 0.80 in. wg is available at the inlet of the VAV terminal to assure design airflow. Figure 5 shows the system resistance curves, fan modulation curve and new fan curve at part load flow. The inlet guide vanes are repositioned so that 18,000 cfm is delivered and the static pressure at the fan outlet is 2.2 in. wg. The fan now operates at the intersection of the fan modulation curve and the new fan curve (point C) and the fan brake horsepower is reduced to 13 bhp.

Figure 5 - Fan Outlet Static Pressure Control



Supply Duct Static Control
The most commonly recommended static pressure sensor location is two-thirds down the supply duct system. The sensor is field installed and the controller is set to maintain the pressure corresponding to that location in the duct system at design airflow conditions. Additional sensors may also be needed if fire dampers are installed in the system. Also, in larger systems with many terminal units, determining the best sensor location for all load conditions can be difficult - if not impossible. The field installation and adjustment of one or possibly several duct pressure sensors raises the first cost over the fan outlet method previously discussed; however, the system operating cost using this method is typically lower.

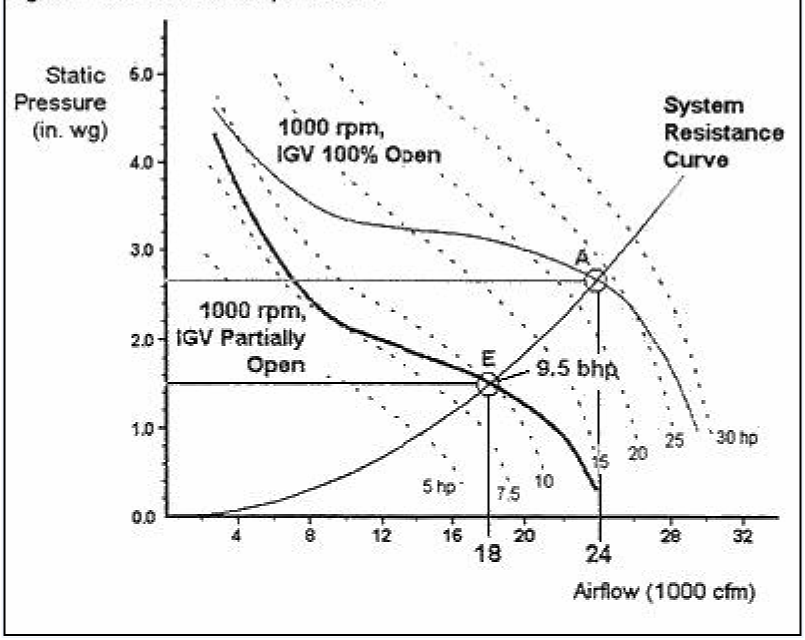
Figure 6 shows the system resistance curves, fan modulation curve and fan curves for our example. At our part load flow of 18,000 cfm, the inlet guide vanes are repositioned creating another new fan curve. The fan operates at the intersection of the new fan curve and the new fan modulation curve, indicated at point D. At this condition, the fan now only requires 12 bhp to deliver the 18,000 cfm.



Fan Pressure Optimization
Both of the preceding methods, fan outlet static control and supply duct pressure control, have advantages and disadvantages. Fan outlet pressure control is reliable with low first cost; however, the part load fan bph requirement is higher than other methods. Supply duct pressure control has a lower part load power requirement; however, it typically requires field installation and calibration of multiple sensors, thereby increasing first cost and possibly reducing reliability.

The Fan Pressure Optimization method combines the location-related benefits of fan outlet control with operating cost savings that exceed those of duct pressure control. A single static pressure sensor is located at the fan outlet and the static pressure controller adjusts the static pressure setpoint based on the **position of the VAV terminal dampers**. And because the pressure sensor is at the fan outlet, it can also serve as a duct high pressure sensor.

Figure 7 - Fan Pressure Optimization



Applying this control strategy to our example (Figure 7) with the inlet vanes positioned for 18,000 cfm, notice that the fan horsepower requirement drops to 9.5 bhp! This is the lowest possible pressure to satisfy the load condition. Supply fan operation is truly optimized. Shown below in Figure 8 is a comparison of the various fan control methods for our simple example.

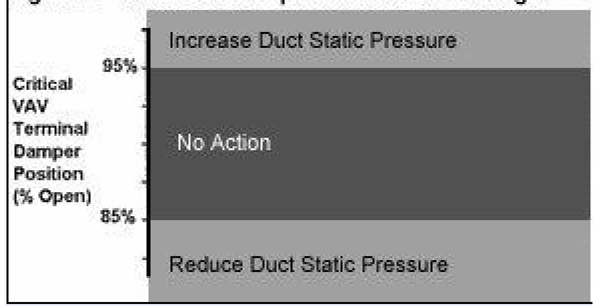
Figure 8 - Fan Control Method Comparison

Control Method	Airflow (cfm)	Fan Static (in. wg)	Brake HP	Brake HP Reduction
Design Condition	24,000	2.70	22.0	---
No Control	18,000	3.15	18.0	22%
Fan Outlet	18,000	2.13	13.0	43%
Duct Static	18,000	1.85	12.0	48%
Press. Optimization	18,000	1.52	9.5	59%

How Does Fan Pressure Optimization Work?

The DDC/VAV controllers know the position of their individual air valves (or dampers). The building automation system (BAS) continually polls the VAV terminal units looking for the most open VAV damper. The BAS adjusts (resets) the duct static pressure setpoint so that at least one VAV terminal, the terminal requiring the highest inlet pressure, is nearly wide open (Figure 9). The result is that the supply fan only generates enough pressure to get the required flow through the critical terminal unit. In addition to the obvious fan energy savings, by definition this method assures that zones cannot be starved for air. There are also significant acoustical benefits at part load by operating the fan and VAV terminals at the lowest possible duct pressure.

Figure 9 - Fan Pressure Optimization Control Logic



Fan Energy Savings Potential

The example we used to explain the different control methods was very simple. You might ask: Is it realistic to expect these savings in real-world situations? Figures 10 and 11 compare the energy consumption for the different control methods for a typical 3-story, 60,000 square foot office building using the various fan control methods in four different cities as modeled using as modeled using TRACE® 600. The system uses a forward-curved fan with variable-speed drive, a comparative-enthalpy economizer and series fan-powered VAV terminals. Figure 10 compares energy consumption for the supply fan only while Figure 11 relates the savings to the total building HVAC energy usage. Using Fan Outlet Pressure control as the baseline, the supply fan energy saving potential with fan pressure optimization ranges from 19 percent in Miami to almost 40 percent in Seattle! In terms of total HVAC system energy, fan pressure optimization can save from 2 percent to 4 percent depending on location.

Figure 10 - Fan Control Method Comparison Based on Supply Fan Energy

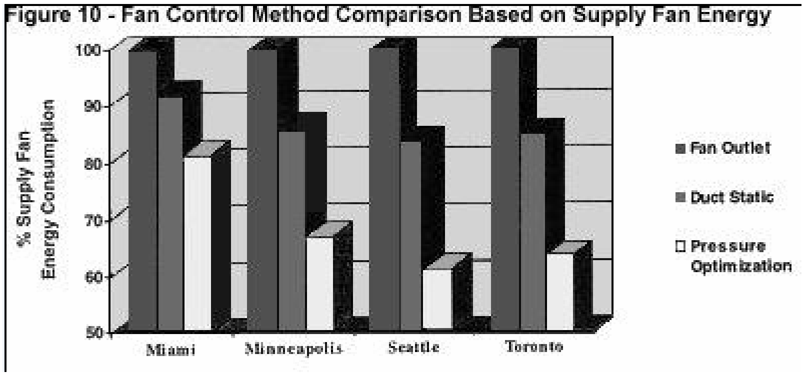
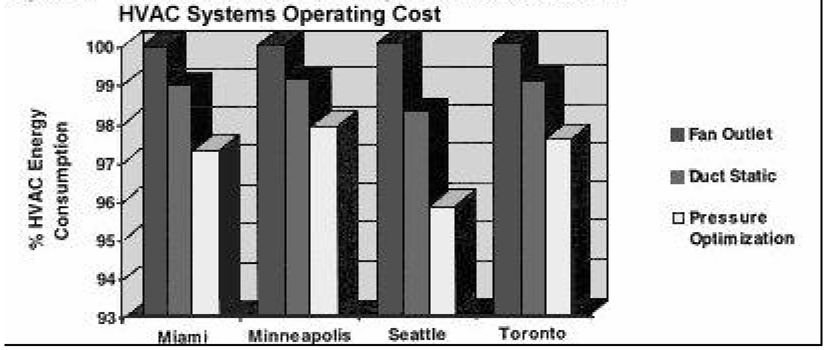


Figure 11 - Fan Control Method Comparison Based on Total HVAC Systems Operating Cost



Hardware and Control Requirements

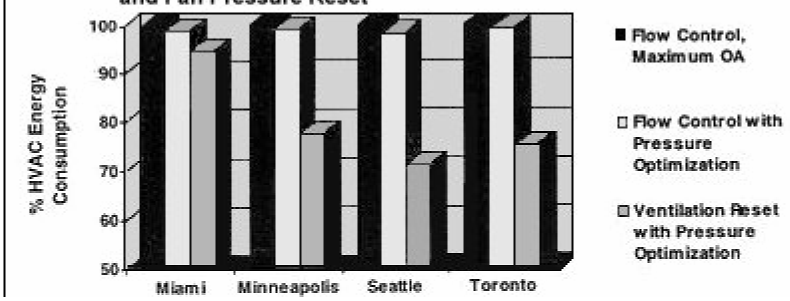
The Fan Pressure Optimization control strategy requires the following equipment and control capabilities:

- Air handling unit with inlet guide vanes or variable-speed drive.
- VAV terminals with pressure independent DDC controls.
- A building automation system, such as Tracer Summit®.

Why Not Use Both Control Strategies?

The key to these control strategies and the improved level of control and reduced operating costs is the communication capabilities of the Integrated Comfort™ system. The strategies discussed are not mutually exclusive. They can be combined for added savings and system control. Figure 12 shows the TRACE® 600 analysis using our sample 3-story office building comparing various outdoor air and fan pressure control methods. Although the savings vary by location, they are significant in all cases.

Figure 12 - HVAC Energy Comparison with Ventilation Reset and Fan Pressure Reset



Bar #1 (left) uses the traditional fixed-damper method with the position based on the average ventilation value for the system. This method **does not meet ASHRAE 62-89** and is shown only as a point of reference.

The equipment and controls required to implement these control strategies are not exotic nor proprietary; in fact most systems today already have the basic components and capabilities. The HVAC equipment and software requirements to implement both Ventilation Reset and Fan Pressure Optimization are:

1. The ability to measure and control outdoor air, such as Trane Traq™ dampers.
2. Communicating DDC controls on the air handler and VAV terminals
3. Supply fan flow control, either inlet guide vanes or a variable-speed drive
4. A building automation system, such as Tracer Summit®, capable of performing the ventilation calculations, specifically Equation 6.1 of ASHRAE 62-1989.

System Specifications

Ventilation Reset and Fan Pressure Optimization are not stand-alone equipment sequences but rather are ***system-level*** control strategies. Thus, it is necessary to specify them as a ***system***. Refer to the Trane BAS/ Microelectronic Controls Specification Manual, ICS-MNL-8 for detailed sequences of operation and points lists.

Conclusion

Ventilation Reset and Fan Pressure Optimization are great ways to minimize **system** energy consumption, provide occupant comfort and meet the ventilation requirements of ASHRAE 62-89. These system control strategies allow us to use microelectronic control technology to do more than just control systems the way pneumatic systems have in the past. It is a missed opportunity to apply new technology in old ways. The **"power of information"** allows us to control our indoor environment smarter for increased comfort and reduce energy costs.