Variable Speed Drive Volumetric Tracking (VSDVT) for Airflow Control in Variable Air Volume (VAV) Systems

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ABSTRACT

An airflow control method has been developed for variable air volume (VAV) systems. This airflow control method is named VSD volumetric tracking (VSDVT) since both the supply and return airflows are determined using signals of the variable speed drives (VSD) instead of the flow stations. Its performance is studied and compared with the fan tracking (FT) method using model simulations. The VSDVT maintains a constant building pressure and the required outside airflow under all load conditions, reduces the annual return air fan energy by up to 50%, and the annual supply air fan energy by up to 30%. This paper presents the VSDVT method, the system models, and the simulation results.

INTRODUCTION

Airflow control of VAV systems has been an important design and research subject since the VAV system was introduced. An airflow control method should: (1) ensure the airflow to each space or zone; (2) control outside air intake properly; and (3) maintain the positive building pressure. Several methods have been developed to ensure the air delivery to each space or zone. These methods include the static pressure control method and the damper position control method. The static pressure control method maintains a sufficient static pressure in the main duct (often 2/3 down stream of the main duct) to ensure the required airflow to each space or zone. Its performance has been proven to be reliable over many years of use. The fan power is lower under partial load conditions. The static pressure reset can further decrease the fan energy under the partial load conditions [1, 2, and 3]. The optimal reset of the static pressure is critical for minimizing the supply fan energy.

Hartman [4] proposed modulating the fan speed to maintain at least one terminal box damper full open for modern DDC systems, where the AHU controller can access all terminal box information directly. When properly designed and maintained, this method consumes minimal fan energy. Recently, Wei et al [5] improved this method by integrating it with the static pressure reset techniques to prevent malfunctioning under several typical building operating problems. Unfortunately, this method can not be used in buildings which have pneumatic terminal box controllers. In this paper, an optimal static pressure reset schedule is used.

The typical building pressure control methods include the fan tracking (FT), the direct building pressure control (DBP), and the volumetric tracking (VT). The FT method sets the return fan speed at or slightly lagging the supply fan speed. Under partial load conditions, the building pressure decreases as the total airflow decreases [6, 7, 8, 9, and 10] since the return air fan draws relatively larger amount of air from the building. The FT does not ensure constant or positive building pressure under partial load conditions.

The DBP method modulates the return air fan speed to maintain the set point of the pressure difference across the building envelope. This can be problematic in a large building, where it is difficult to measure the pressure difference properly since the variation of the pressure difference across the entire envelope is far larger than the set point. Hence, if the pressure difference across the envelope cannot be measured accurately, this method should not be used.

The VT method measures both the supply and return airflows using flow stations. The return air speed is modulated to maintain the required difference of the supply and return airflows. This method is very effective when the equipment location and space are available for the airflow stations [11, 12]. However, accurate measurement of airflow has proved difficult if not impossible for most of the systems due to lack of the appropriate length of straight ductwork. Therefore, this method has limited practical value [9, 13]. In this study, the volumetric tracking is used. The VSD signals and the fan pressure head are used to determine the airflow rates.

Outside air intake control often uses one of the following methods: the fixed damper position, the direct method [14], the plenum-pressure control, and the CO_2 demand control. In many buildings, it is a common practice to set both the outside and the return air dampers to fixed positions to control the outside air. The damper positions are determined during system testing and balancing process. The outside air intake is either higher or lower than the required airflow when the total airflow is different with the testing condition.

The direct method measures the outside airflow directly using a flow station. The controller modulates the outside air damper to maintain the required outside airflow. A minimum outside air duct or a fan is often required [15, 16, and 17] in order to measure the airflow accurately. The direct method provides good outside air control when the outside airflow can be measured accurately. Unfortunately, in many buildings, the air leakage of the maximum outside air damper may by-pass the flow station and therefore, reduces the accuracy of the outside air measurement.

The plenum-pressure control method maintains the mixed air chamber pressure at a required level by adjusting the return air fan speed or the return air damper [13]. The outside air damper serves as the flow meter. Since the damper has to be modulated during the economizer cycle, the hysteresis of the air damper can cause significant airflow variation [18]. The outside airflow may vary significantly even the plenum static pressure is controlled properly since the pressure drop across the outside air filter increases as the dirt built up.

The CO_2 demand control method maintains the representative carbon dioxide at its set point by modulating the outside air damper. This method provides reliable outside air intake control for typical occupancy [10, 19, 20, 21, and 22]. In this study, the CO_2 demand control method is used. An improvement is made to prevent the outside air backflow through the release duct building pressure when the mechanical exhaust is higher than the occupancy fresh air requirement.

In this study, a new airflow control method is developed for VAV systems. This method implements the volumetric tracking using the VSD speed signals and the fan pressure heads instead of airflow stations, controls the outside air intake using the improved CO_2 demand control techniques, and controls the supply air fan using an optimal static pressure reset schedule. Since the volumetric tracking is implemented using the VSD signals, this method is, called the VSD volumetric tracking (VSDVT). This paper presents the principals of the VSDVT method and studies its performance using simulations.

VSDVT METHOD

Figure 1 presents the airflow control schematic of the new VSDVT method. The physical (hard) input signals include the supply and return fan heads, the supply and return air static pressures, the return air temperature, the mixed air temperature, the outside air temperature, the return air or the critical zone CO_2 concentration. The output signals include the supply fan VSD speed, the return fan VSD speed, the outside air damper position, and the return and release air damper positions.

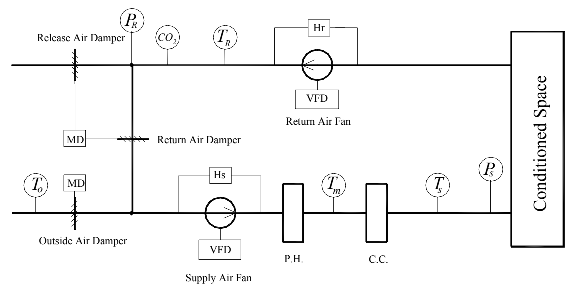


Figure 1: Airflow Control Schematic of the VSDVT Method

The VSDVT has four control loops: the supply fan speed, the return fan speed, the return air damper, and the outside air damper. For the supply fan speed loop, the controlled variable is the supply air static pressure. The controlled device is the VSD of the supply fan. This control loop maintains the set point of the supply air static pressure by modulating the supply fan VSD speed.

To minimize the supply fan energy, the supply air static pressure is reset to maintain the duct system resistance unchanged under the partial load conditions. According to the fluid dynamic theory (assuming turbulent flow), the pressure loss is proportional to the square of the flow rate. Therefore, to maintain the duct resistance under the design conditions, the static pressure should be reset proportional to the square of the airflow rate under partial load conditions. If the duct resistance is kept unchanged, the flow ratio equals the fan speed ratio according to the fan law. Therefore, the optimal static pressure reset schedule is expressed by equation 1.

$$p_s = c + p_d \left(\frac{\omega}{\omega_d} \right)^2 \tag{1}$$

A positive constant c is added to the optimal reset schedule to stabilize the control loop. Since the reset schedule uses the control output reset as the controlled variable, the fan speed will reduce to zero without adding the constant c. Strictly speaking, the optimal reset schedule applies to the uniform zone load pattern only. If the variation of the zone load ratios is small, a small correction factor can be used. Otherwise, a larger correction may be required.

Equation 1 may result in a static pressure set point higher than the design value when the VSD speed is high, or result in an unrealistic low set point when the VSD speed is low. Therefore, neither of these high and low static pressure set points should be used. The high set point causes the excessive fan energy. The low set point may not be able to ensure the air delivery to each zone. Since the flow becomes laminar at lower flow rate while equation 1 only applies to the turbulent flow, a low limit should be selected based on the minimum static pressure requirement of terminal box, the duct layout, and other information. If the calculated set point is higher than the design value, the design value should be used. If the calculated value is less than the low limit, the low limit value should be used.

For the return fan speed loop, the controlled variable is the return airflow rate. The controlled device is the return air VFD. The controlled loop output is the return fan VFD speed. The return airflow set point equals the difference of the supply airflow and the building exhaust and air exfiltration.

$$Q_{r,s} = Q_s - Q_{ex} - Q_{\inf} \tag{2}$$

The supply airflow can be calculated using equation 3 based on the supply fan VSD speed and the fan head (See Appendix A for details).

$$Q_s = \frac{\left(-a_1 \pm \sqrt{a_1^2 - 4a_2\left(a_0 - \frac{H_s}{\omega_s^2}\right)}\right)\omega_s}{2a_2} \tag{3}$$

The exhaust airflow is treated as a constant since it depends on the building envelope only. Therefore, the tighter the building envelope is, the smaller the value.

The return airflow is calculated using equation 4 according to the return fan VSD speed and the return fan head.

$$Q_r = \frac{\left(-b_1 \pm \sqrt{b_1^2 - 4b_2\left(b_0 - \frac{H_r}{\omega_r^2}\right)}\right)\omega_r}{2b_2} \tag{4}$$

The control loop modulates the return fan speed to maintain the return airflow set point.

For the outside air damper loop, the controlled variables are the return air static pressure and the return air or the critical zone CO_2 concentration when the economizer is not activated, or the mixed air temperature when the economizer is activated. The controlled device is the outside air damper. The set point of the CO_2 concentration should be predetermined using engineering principals. The set point of the return air static pressure is zero. The controller modulates the outside air damper to maintain both the CO_2 and the return air pressure set points only when the return air damper is in its maximum open position. If the return air static pressure is lower than the set point, the controller increases the outside air damper openness regardless of the CO_2 concentration. This prevents the negative building pressure when the fresh air requirement of occupants is less than the mechanical exhaust and the exfiltration. When the economizer is activated, the controller modulates the outside air damper to maintain the mixed air temperature set point.

For the return air damper loop, the controlled variable is the return air or the critical zone CO_2 concentration when the economizer is not activated, or the mixed air temperature when the economizer is activated. The controlled device is the return air and the release air dampers. The release and the return air dampers are interlinked. When the release air damper is in the minimum position, the return air damper is in the maximum position. The return air damper loop is activated only when the outside air damper is in the full open position. The controller decreases the return air damper openness if the CO_2 concentration is higher than the set point, or if the mixed air temperature is higher than the cold deck set point during the economizer cycle. Otherwise, the controller increases the return air damper openness.

The VSDVT method minimizes the fan energy using the optimal static pressure reset and decoupling the outside and return air dampers; implements the volumetric tracking using the VSD speeds and the fan heads; and uses CO_2 demand control to minimize outside air intake. The mathematical and physical models of the VSDVT are presented in the next section. Its performance are simulated and compared with the FT method.

PERFORMANCE ANALYSIS

This section compares the VSDVT with the FT using simulations. The impact of each key control measure is also investigated. Both the system modeling and simulation results are presented in this section.

Airflow Models

Figure 2 presents the airflow schematic of the VAV systems. The airflow system is divided into fifteen segments. Each segment can be expressed using the flow resistance factor (S), which is defined as the ratio of the pressure loss and the square of the airflow rate through the segment. The flow resistance factor represents the hydraulic characteristics of the ductwork, coils, filters, and fittings.

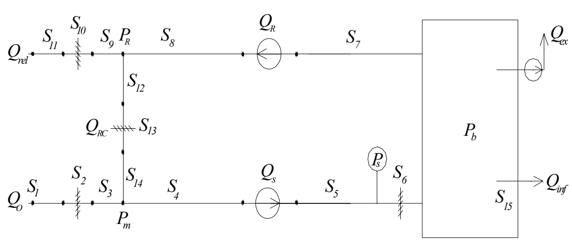


Figure 2: Airflow Schematic of VAV Systems

The resistance factors are constants except for the segments 2, 6, 10, and 13. The values of the resistance factors are calculated using equation 5 based on the design information (See Appendix B for details).

$$S_i = \frac{\Delta P_i}{Q_i^2} \tag{5}$$

The flow resistance factors of the segments 2, 10, and 13 depend on both the damper position (d) and the minimum flow resistance factor. The values of the flow resistance factors are calculated using Equation 6.

$$S_i = \frac{S_{i,0}}{f^2}, i = 2,10,13 \tag{6}$$

The minimum resistance factor is calculated using equation 5 and the design information. The correction factor f depends on both the damper position and the type of the damper. According to SMACNA [23], the correction factor is expressed using equation 7.

$$f = \begin{cases} 0.0252 - 0.1045d + 1.0618d^2 & Opposed \\ -0.0164 + 0.6452d + 0.377d^2 & Parallel \end{cases} \tag{7}$$

The airflow loop has three joints. Therefore, three airflow balance equations form the framework of the VAV system airflow models.

$$Q_s - Q_r = Q_{ex} + Q_{\inf} \tag{8}$$

$$Q_s = Q_{rc} + Q_o \tag{9}$$

$$Q_r = Q_{rc} + Q_{rel} \tag{10}$$

Where:

$$Q_{ex} = C \tag{11}$$

$$Q_{\inf} = \begin{cases} \sqrt{\dfrac{p_b}{S_{15}}} & p_b \ge 0 \\ -\sqrt{\dfrac{|p_b|}{S_{15}}} & p_b < 0 \end{cases} \tag{12}$$

$$Q_{rel} = \begin{cases} \sqrt{\dfrac{p_r}{S_9 + S_{10} + S_{11}}} & p_r > 0 \\ -\sqrt{\dfrac{|p_r|}{S_9 + S_{10} + S_{11}}} & p_r < 0 \end{cases} \tag{13}$$

$$Q_o = \begin{cases} \sqrt{\dfrac{p_m}{S_1 + S_2 + S_3}} & p_m < 0 \\ -\sqrt{\dfrac{|p_m|}{S_1 + S_2 + S_3}} & p_m > 0 \end{cases} \tag{14}$$

$$Q_{rc} = \sqrt{\frac{p_r - p_m}{S_{12} + S_{13} + S_{14}}} \tag{15}$$

Both the supply and return airflows are indirectly determined by balancing fan head and system pressure loss.

$$\omega_s^2 \sum_{i=0}^{2} a_i \left(\frac{Q_s}{\omega_s} \right)^i = |p_m| + (S_4 + S_5) Q_s^2 + p_s \qquad (16)$$

$$\omega_r^2 \sum_{i=0}^{2} b_i \left(\frac{Q_r}{\omega_r} \right)^i = p_p + (S_7 + S_8) Q_r^2 + p_r \qquad (17)$$

The supply and the return fan powers are calculated using equations 18 and 19, respectively, according to the fan heads, the airflow rates, and the fan efficiencies.

$$E_s = c \frac{Q_s H_s}{\eta_s} \qquad (18)$$

$$E_r = c \frac{Q_r H_r}{\eta_r} \qquad (19)$$

Where:

$$\eta_s = e_0 + e_1 \left(\frac{Q_s}{\omega_s} \right) + e_2 \left(\frac{Q_s}{\omega_s} \right)^2 \qquad (20)$$

$$\eta_r = f_0 + f_1 \left(\frac{Q_r}{\omega_r} \right) + f_2 \left(\frac{Q_r}{\omega_r} \right)^2 \qquad (21)$$

Equations 5 to 21 consist of the airflow models of the VAV systems. The equation set has a total 20 equations with 26 variables and is therefore over-determined. Six more conditions must be added in order to simulate the system performance. These six additional conditions allow define the specific system inputs and the operation and control schedules when performance simulation is conducted.

Performance Simulation

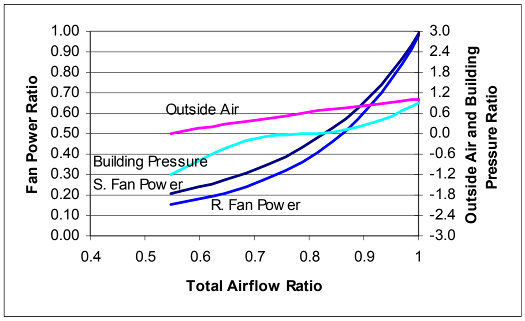
Eight simulations were performed to study the VSDVT performance. The study uses the FT method as the base case. The full occupancy is assumed in this study. The supply air fan speed and the supply air static pressure are the primary inputs for all the simulations.

The first two cases are labeled Fan Tracking-1 and the Fan Tracking-2. In both cases, the return air fan speed equals the supply air fan speed. In the FT-1 case, the return, release, and outside air damper positions are selected to provide the required minimum airflow when the supply fan provides 100% design airflow to the building. In the FT-2 cases, the damper positions are selected to provide the required minimum outside airflow when the supply fan provides 60% of the design airflow to the building. The outside air, the return, and the release air dampers are fixed at the initial condition regardless the load conditions. Constant static pressure set point is used for both cases.

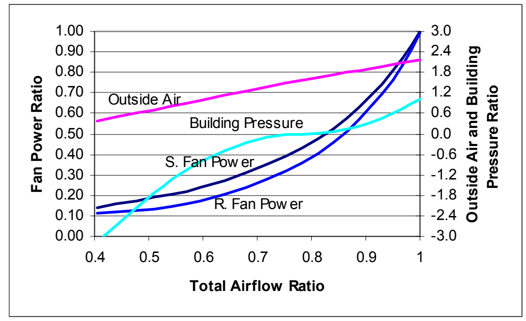
The third and the fourth cases add the Static Pressure Reset (SPR) in the FT-1 and FT-2 cases. The SPR resets the static pressure proportionally to the square of the airflow. The correction constant is selected as zero in this study. The fifth case adds the CO_2 demand control (DC) in the FT-1 case. The sixth case adds both the SPR and DC into the FT-2 case. The seventh case uses both the volume tracking (VT) and the DC. The eighth case adds the SPR to the seventh case. The eighth case represents the performance of the VSDVT method. The first two cases represent the base cases.

The simulations were performed based on an existing AHU. The design supply and return airflows are 16.5 m^3/s and 14.2 m^3/s (35,000 and 30,000 cfm), respectively. The minimum outside air intake is 3.8 m^3/s or 8,000 cfm. The sum of the mechanical exhaust and air exfiltration is 2.4 m^3/s or 5,000 cfm. The design fan heads are 1,375 Pa (5.5 inH_2O) for the supply fan, and 525 Pa (2.1 inH_2O) for the return fan. The static pressure set point is 500 Pa or 2 inH_2O. The detailed system design information is attached in Appendix B. Figure 3 presents the simulated outside air intake ratio, the building pressure ratio, and the supply and return fan power ratios.

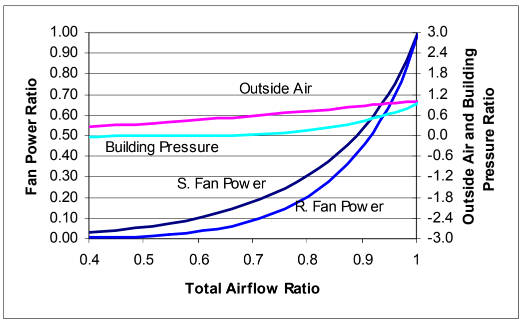
The simulation results show that the outside air intake decreases as the total airflow decreases when the FT method is used (Figures 3a and 3b). The FT-1 provides less outside air to the building under the partial load conditions. The FT-2 provides more outside air to the space when the total airflow is higher than 60% of the design airflow. In both FT-1 and FT-2 cases, the building pressure decreases from the design value to negative as the total airflow decreases. When the supply airflow is less than 54%, the return air is released from the outside air duct to the outside in the FT-1 due to the over-pressurization of the return fan. The FT method is prone to IAQ problems or high thermal energy consumption, and building pressure control problems.



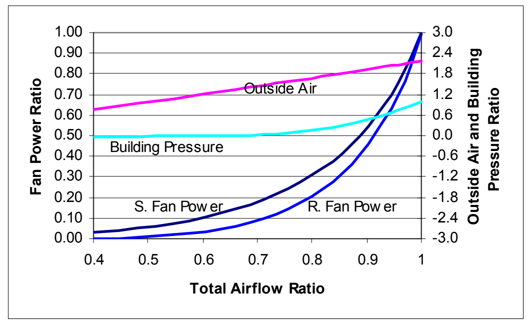
(a) FT-1



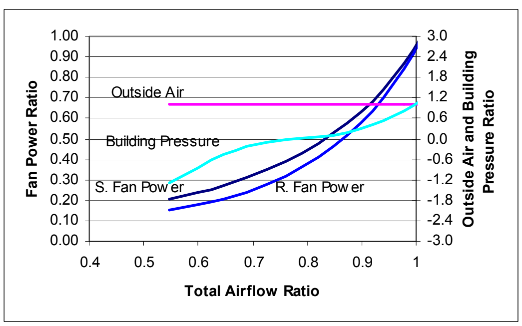
(b) FT-2



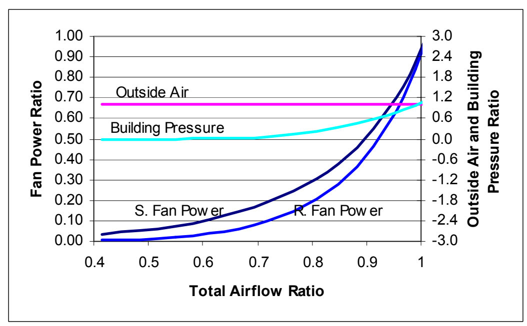
(c) FT-1 & SPR



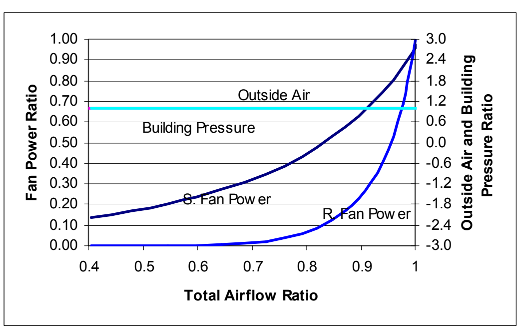
(d) FT-2 & SPR



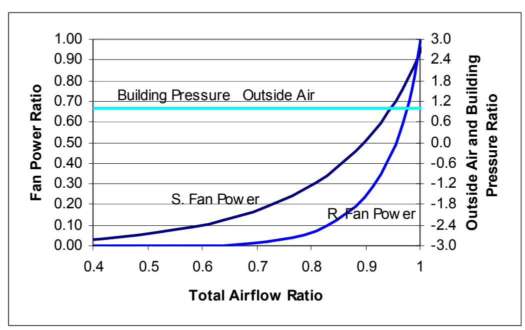
(e) FT & DC



(f) FT & DC & SPR



(g) VT & DC



(h) VSDVT

Figure 3: Simulated Outside Air Intake Ratios, Building Pressure Ratios, Supply and Return Fan Power Ratios

Figures 3c and 3d present the results of the FT with SPR. The static pressure reset improves the building pressure and the outside air controls, and decreases the fan power. The minimum building pressure ratio is improved from –3.2 to 0. The outside air intake ratios are relatively close to the design value. For example, the outside air intake ratio is decreased from 2.18 to 1.89 from the FT-2 case under the full load condition. The fan power is much lower. For example, the fan power ratio is decreased from 30% to 10% when the airflow ratio is 60%.

Figures 3e presents the results of the FT with the DC. The DC maintains the constant outside air intake at design level and has no impact on the building pressure control. Since the improved DC method minimizes the resistance of the return and the outside air dampers, the fan energy is lower than the FT method. For example, the fan power is approximately 5% lower under the full load conditions. It is important to point out that the proper outside air intake does not ensure the positive building pressure in a VAV system.

Figure 3f presents the results of the FT with both the DC and the SPR. The DC maintains the constant outside air intake and decreases the supply fan power. The SPR decreases both the supply and the return air fan power, and improves the building pressure control. Again, the proper outside air intake does not ensure the proper building pressure.

Figure 3g presents the results of the VT with the DC. Both the outside air intake and the building pressure are controlled at the required level. The return air fan power ratio is significantly lower than the supply air fan power ratio since the return airflow ratio is lower than the supply airflow ratio under partial load conditions.

Figure 3h presents the results of the VSDVT method. The VSDVT maintains both the building pressure and outside air intake properly with the minimum thermal and fan energy. Comparing with FT-2, the VSDVT decreases the outside air intake ratio from 2.18 to 1 under the peak load conditions.

Figure 4 presents the simulated the VSDVT fan power savings against the FT method. The fan power savings is expressed as the ratio of the power savings over the design fan power. The maximum fan power savings is 37% for the return fan and 17% for the supply fan. The annual average energy savings ratio could be up to 25% for the return fan and 15% for the supply fan since the total airflow varies between 0.7 to 0.95 most of the time for typical VAV systems. The VSDVT could decrease the annual fan energy by up to 50% for the return air fan and 30% for the supply air fan if the annual average fan power is 50% of the design value using the FT method.

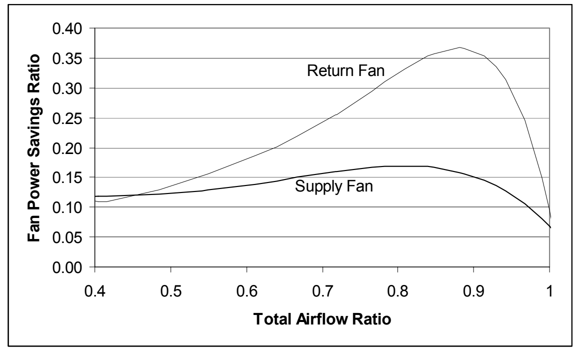


Figure 4: Potential Fan Power Savings of the VSDVT Method

CONCLUSIONS

The VSDVT method has been developed for the airflow control in VAV systems. This method can be used for typical AHUs, which have programmable controllers, since the volumetric tracking is implemented using the VSD speed signals and the fan heads.

The VSDVT ensures the required outside air intake and the positive building pressure under all load conditions with minimal thermal and fan energy. The annual fan energy savings is up to 50% for the return fan and 30% for the supply fan. Under the full load condition, the VSDVT reduces the outside air intake by 1.2 times of the design value comparing with the FT control.

NOMENCLATURE

a Regression constants for supply air fan
b Regression constants for return air fan
c Unit conversion factor, static pressure correction factor
d Damper position (0 to 1)
e Regression coefficient of supply air fan efficiency versus the airflow under full speed
f Correction factor, regression coefficient of return air fan efficiency versus the airflow under full speed
H Fan head (Pa or inH_2O)
P Pressure (Pa or inH_2O)
Q Airflow rate (l/s or cfm)
S Flow resistance factor

Δp Pressure loss (Pa or inH_2O)
ω Fan speed (RPM)

Subscripts
b Building
d Design
ex Exhaust
inf Infiltration/exfiltration
m Mixed
o Outside air
r Return
rc Re-circulated
rel Release
s Supply, set point

REFERENCES

1. Warren M. and Norford L. K., 1993. "*Integrating VAV Zone Requirement with Supply Fan Operation*," Journal of ASHRAE, Vol. 35, no. 35, pp 43-46.
2. Rose R. J. and Kopko W. L., 1994. "*A Novel Method for Resetting Duct Static Pressure for Variable Air Volume Systems*." Proceedings of ACEEE 1994 Summer Study on Energy Efficiency in Buildings, Vol. 5, pp. 219-223.
3. Liu M., Zhu Y., Claridge D.E., and White E., 1997. "*Impact of Static Pressure Set Level on HVAC Energy Consumption and Indoor Conditions.*" ASHRAE Transactions 1997, Vol. 103, pt. 2.
4. Hartman T., 1989. "*TRAV-A new HVAC Concept*," Journal of Heating/Piping/Air Conditioning, July 1989, pp. 69-73.
5. Wei G., Liu M., and Claridge D. E., 2002. "*Integrated Damper Position and Static Pressure Reset for the VAV Supply Air Fan Control*." Submitted to ASHRAE.
6. Gardner F, T., 1988. "*Part Load Ventilation Deficiencies in VAV Systems*." Journal of Heating/Piping/Air Conditioning, February 1988, pp 89-100.
7. Mumma S. A. and Wong Y. M., 1990. "*Analytical Evaluation of Outdoor Airflow Rate Variation vs. Supply Airflow Rate Variation Invariable-Air-Volume Systems When the Outdoor Air Damper Position Is Fixed.*" ASHRAE Transactions 1990, Vol. 96. Part 1, 1197-1208.
8. Solberg D. W., 1990. "*Measurement for the Control of Fresh Air Intake*." ASHRAE Journal, January 1990, Vol. 32, no.1, 46-51.
9. Elovitz D. M., 1995. "*Minimum Outside Air Control Methods for VAV Systems*." ASHRAE Transactions 1995, Vol. 101, Part 2, 613-618.
10. Mudarri D., Hall J. D., and Werling E., 1996. "*Energy Cost and IAQ Performance of Ventilation Systems and Controls*." IAQ 96, Paths to Better Building Environments, Baltimore, October 1996.
11. Kettler J. P., 1995. "*Minimum Ventilation Control for VAV Systems-Fan Tracking vs. Workable Solutions.*" ASHRAE Transactions 1995, Vol.101, Part 2, 625-630.
12. Avery G., "*The Instability of VAV Systems*." Journal of Heating/Piping/Air Conditioning, 1992, 64(2): 47-50.
13. Cohen T., 1994. "*Providing Constant Ventilation in Variable Air Volume Systems*." ASHRAE Journal, May 1994, pp 38-40.
14. Atkinson G. V., 1986. "*VAV System Volume Control Using Electronic Strategies*." ASHRAE Transactions 1986, Part 2B, 46-57.
15. Janu G. J., Wenger J. D., and Nesler C. G., 1995. "*Outside Air Flow Control for VAV Systems*." ASHRAE Journal, April 1995, Vol. 37, no. 4, 62-68.
16. Kettler J. P., 1998. "*Controlling Minimum Ventilation Volume in VAV Systems*." ASHRAE Journal, May 1998, Vol. 40, no. 5, 45-50.
17. Mumma S. A., 2001. "*Dedicated OA Systems*." ASHRAE IAQ Applications, Winter 2001, Vol. 2., No. 1, 20-22.
18. Krakow K. I., 1998. "*Reduction of Hysteresis in PI-Controlled Systems*." ASHRAE Transaction 1998, Vol. 104, pt. 1.
19. Smith R. B., 1987. "*Variable Air Volume Building Ventilation Control System*." Proceedings of ASHRAE Conference IAQ 87, Arlington, Virginia, May 1987, 349-357.
20. Meckler M., 1994. "*Demand-Control Ventilation Strategies for Acceptable IAQ*." Journal of Heating/Piping/Air Conditioning, May 1994, pp. 71-74.
21. Carpenter S. C., 1996. "*Energy and IAQ Impacts of CO_2-Based Demand-Control Ventilation*." ASHRAE Transactions 1996, Vol. 102, part 2.
22. Schell M., and Int-Hout D., 2001. "*Demand Controlled Ventilation Using CO_2*." ASHRAE Journal, February 2001, vol. 43, no. 2, 18-29.
23. SMACNA, 1993. "*HVAC Systems Testing, Adjusting & Balancing*," Sheet Metal and Air Conditioning Contractors National Association, Inc. Chantilly, VA.

APPENDIX A

Equation 3 is deduced based on the fan curve and the fan law. The procedure is listed below.

Assuming the fan curve can be expressed using a second order polynomial equation under the full speed:

$$H = a_0 + a_1 Q + a_2 Q^2 \quad \text{(a-1)}$$

If the fan is running under partial speed, the fan head and the airflow is correlated using equation a-2 according to the fan law.

$$H_\omega = \omega^2 (a_0 + a_1 Q/\omega + a_2 (Q/\omega)^2) \quad \text{(a-2)}$$

When both the fan head and the fan speed are given, the fan airflow is deduced as:

$$Q = \frac{\left(-a_1 \pm \sqrt{a_1^2 - 4a_2\left(a_0 - \frac{H_\omega}{\omega^2}\right)}\right)\omega}{2a_2} \quad \text{(a-3)}$$

APPENDIX B

The design pressure loss and the airflow rates are listed in the table below for each airflow loop section.

Table 1: Design Pressure Loss and Airflow Rates for Each Airflow Loop Section.

Segment	Pressure Loss (Pa/inH_2O)	Airflow ($m^3/s/cfm$)	Notes
1	12.5/0.05	16.5/35,000	Outside air ductwork
2	25/0.10	16.5/35,000	Outside air damper
3	12.5/0.05	16.5/35,000	Ductwork
4 and 5	825/3.3	16.5/35,000	Ductwork, filter, and coils
6	500/2.0	16.5/35,000	Down stream of the static pressure sensor
7 and 8	475/1.9	14.2/30,000	Return air duct
9	12.5/0.05	14.2/30,000	Release air ductwork-1
10	25/0.1	14.2/30,000	Release air damper
11	12.5/0.05	14.2/30,000	Release air ductwork-2
12	12.5/0.05	14.2/30,000	Re-circulate air ductwork-1
13	25/0.1	14.2/30,000	Re-circulate air damper
14	12.5/0.05	14.2/30,000	Re-circulate air ductwork-2
15	12.5/0.05	2.45,000	Building envelope