A Novel Method for Resetting Duct Static Pressure for Variable Air Volume Systems

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In the past, variable air volume (VAV) supply air systems have consisted of pneumatically actuated mechanical inlet vanes or discharge dampers which maintain constant duct static pressure. Today, mechanical vanes or dampers are being replaced with variable speed drives (VSD), which greatly reduce fan motor power at low flow rates. However, the past practice of maintaining constant duct static pressure has continued even with the advent of more sophisticated direct digital control (DDC). For buildings equipped with DDC, a control strategy is presented which allows the duct static pressure to vary such that minimum fan motor power is consumed at all flow rates. Emphasized is a simple low cost method in which VAV systems without DDC also can vary duct static pressure. This method requires just two inputs, static and velocity pressure, as a basis of resetting the duct static pressure. Applying this technique to a VAV system equipped with VSD's, fan energy savings of 50% were realized at times of low flow rate. In addition to enhanced energy savings, fan speed was further reduced thus increasing equipment life and reducing noise levels in the space. Lastly, a control strategy to minimize energy use of all HVAC equipment combined is discussed.

Introduction

The basis a variable air volume (VAV) supply air distribution system is a central air handling unit (AHU) serving multiple terminal box units. Flow control is accomplished by varying the pressure drop in the terminal box units by opening and closing a damper. The function of the AHU is to provide the source of air and a static pressure for the terminal boxes.

Control schemes for VAV systems have traditionally separated terminal box controls from AHU controls. In the traditional scheme, terminal box units are most commonly controlled by wall thermostats placed within the conditioned space. AHU's regulate air flow such that a constant static pressure is maintained at some predetermined location within the ductwork. The limitation of this control scheme is the likely possibility that none of the terminal box unit dampers are fully open. In such a case, the AHU is forced to provide excess static pressure which is then dissipated by the partial closure to the terminal boxes. The ideal control scheme would provide the minimum static pressure from the AHU such that at least one terminal box is fully open at all times. This would lead to lower fan energy use, less leakage in the ductwork and better control of those boxes with reduced flow rates. Such a control scheme would require that the terminal box controls and AHU controls be integrated.

This paper discusses two methods to integrate terminal box controls and AHU controls together. The goal is to minimize fan energy use while maintaining comfort. The first method makes use of the ability to monitor the operation of each and every terminal box unit. This method provides the greatest potential for fan energy savings but is limit to VAV systems with advanced monitoring capabilities. Because the majority of buildings with VAV systems today can not monitor every terminal box operation, a second method is presented which requires two simple pressure measurements.

Pneumatic Controls and Mechanical Inlet Vanes or Dampers

In the past, pneumatic HVAC controls have been designed to operate independently. Each HVAC component, whether it is a chiller, fan motor or outside air damper, is controlled independently without integration. A reason for this is the high cost and complexity involved in the integration of pneumatic controls. In addition to the limitation of pneumatic controls, AHU's of the past have operated at a fixed speed with inlet vanes or discharge dampers to control air flow. Both are inefficient means of controlling air flow. This again was due to cost constraints. The consequence of limited pneumatic controls and mechanical vanes or dampers are VAV systems in the past which are not as efficient as is economically possible today.

As mentioned earlier, AHU controls maintain a constant duct static pressure at some predetermined location. This control scheme often leads to higher duct static pressures and higher pressure drops across terminal units then required. Although lowering duct static pressure would reduce pressure drop across terminal units, pressure drops across the inlet vanes or discharge dampers would be increased. The outcome of these factors is the pneumatically controlled VAV system with inlet vanes or discharge dampers which we are familiar with today. Not only were HVAC designers of the past faced with high costs of integrating pneumatic controls, they were also faced with mechanical flow control which could not take advantage of more sophisticated controls anyway.

The conclusion is that VAV control strategy of the past was based on the available cost effective technology of the time. It has been the authors experience that far too many building engineers, operators, architects and even HVAC design engineers still cling to idea of maintaining constant static pressure in the duct. Most of the above mentioned would (and have) argued that static pressure must be maintained in order to operate a VAV system. This misunderstanding is further complicated with the mention of pressure dependant and pressure independent terminal box control. Evidence of this is clearly seen in new buildings equipped with both DDC and variable speed drives, yet still operate with constant duct static control.

DDC Controls and VSD'S

Today, pneumatic controls are being replaced with direct digital control (DDC). With DDC, all controls including terminal box units and AHU's, are tied to a central computer. This now opens the opportunity to integrate terminal box unit control and AHU control which is the main subject of this paper. As eluded above, integrated controls require more efficient flow control at the AHU to realize the full potential. Reduced costs of VSD's now allow for the efficient fan flow control needed.

DDC Controls and VSD'S: Application

Warren and Norford 1993 have applied DDC controls to integrate VAV terminal boxes and AHU's. The terminal boxes were able to send an alarm when unable to provide the required air flow to the space. An alarm indicates that the static pressure in the duct is too low. Once every minute, the terminal boxes were poled to determine how many were in alarm status. If more than two or three terminal boxes were in alarm, the duct static was incremented upward until two or three boxes were in alarm. If less than two or three terminal boxes were in alarm, the duct static pressure was incremented downward until two or three boxes were in alarm. Limitations of how fast the duct static pressure could change from one minute to the next ensured stable operation. Although two or three boxes are in alarm status at all times, loss of comfort control was not a problem. Fan energy savings of 19% to 42% compared to maintaining a constant duct static pressure were reported.

The technique used by Warren and Norford ensures that the duct static pressure is kept as low as possible while maintaining comfort. In both cases, the potential fan energy savings due to the reduced static pressure was realized with the use of VSD's. Also noted was the potential for reduced duct leakage and reheat energy due to the lower duct static pressures.

A similar technique was tried by the authors in a new building equipped with DDC control and VSD's in the Washington D.C. area. In this case, the terminal box damper positions could be monitored directly. During a cold winter day, the duct static pressure was manually reset until one terminal box was fully open. The duct static pressure was reduced from 1.5" to 0.75". Fan energy savings of 50% were estimated. Automating the static reset strategy is now underway.

Pneumatic Controls with VSD'S: Theory

Unfortunately, the majority of buildings today are equipped with pneumatic terminal box controls which can not be monitored remotely. With a minimal cost, it is possible to monitor all the terminal boxes as a whole. Unlike DDC which can monitor each and every terminal box unit separately, the technique described below can provide insight to the average terminal box position for all terminal boxes serve by the AHU.

As air flows though a duct, the static pressure loss from point A to B varies with the square of the velocity. If point B represents the exit of a supply duct, the static pressure loss from point A to B is simply the static pressure at point A. As air flows through the same duct, the velocity pressure at point A will also vary with the square of velocity. If point A represents the exit of a supply fan, then the quotient of the static pressure (Ps) over velocity pressure (Pv) will remain constant at any flow rate. For example, if the fan flow rate doubles, both the static pressure at the fan exit and the velocity pressure at the fan exit will increase by a factor of four. So the quotient Ps/Pv will remain the same for any flow rate. The value of Ps/Pv is then independent of flow rate and in fact describes the resistance of air flow through the duct.

In VAV systems, the terminal box units are equipped with dampers. The dampers modulate air flow by altering the resistance to air flow. So as the damper opens and closes, the value of Ps/Pv will change to reflect the new resistance to air flow for the duct and damper combined. If the value of Ps/Pv is correlated for each damper position, damper position can then be determined later by measuring Ps/Pv. Although this could be done for every terminal box unit, the number of pressure measurements would prove to be very costly. By measuring Ps and Pv in the main duct of a VAV system, the average damper position for all the terminal boxes can be correlated and later determined. With this information, duct static pressure can be reset to ensure the terminal boxes operate at the same position (on average) for any flow rate.

In Figure 1, three lines of constant Ps/Pv are shown for a theoretical VAV system. Static and velocity pressures are assumed to be located in the main duct before any branches. Line A represents the condition when all the terminal boxes are fully open. Line C represents the condition when all the terminal boxes are at their minimum position. Line B represents the condition when all the terminal boxes are 1/2 open. Because lines A and C represent the two extremes, every box position is known due to the fact that they are all open or in the minimum position. Line B only represents the average terminal box position. This is because many combinations of box positions can lead to an average position of 1/2 open.

In Figure 1, it is seen that for a given flow rate represented by constant velocity pressure, the static pressure in case C is always higher than case A. So operating the VAV system along line A minimizes static pressure and thus fan energy. However, following line A would not allow diversity between terminal boxes since all would be forced open. This would lead to lose of comfort control. Following line C would also lead to loss of comfort control because all terminal boxes would be forced to the minimum position. Line C would also represent the highest static pressure and thus highest fan energy use. Line B represents a condition which assures that the average box position is 1/2 open by definition. Some boxes would be open more than 1/2 and others would be open less than 1/2. Following line B, it is possible that some boxes would be fully open and still require more air flow. In such a case, either the pressure drop to and from the fully open box would needs to be reduced or a line closer to line C in Figure 1 would need to be chosen.

With the DDC technique, duct static pressure was reset to ensure at least one box was fully open. Using Ps/Pv, duct static pressure can be reset to ensure the average box position is 1/2 open. This technique in general will not offer the same fan energy savings as the DDC technique. For older VAV systems which can not monitor each and every terminal box, this second technique can be a low cost alternative. It is noted once more that the potential fan energy savings associated with lower static pressure can only be realized with the application of VSD's at the AHU.

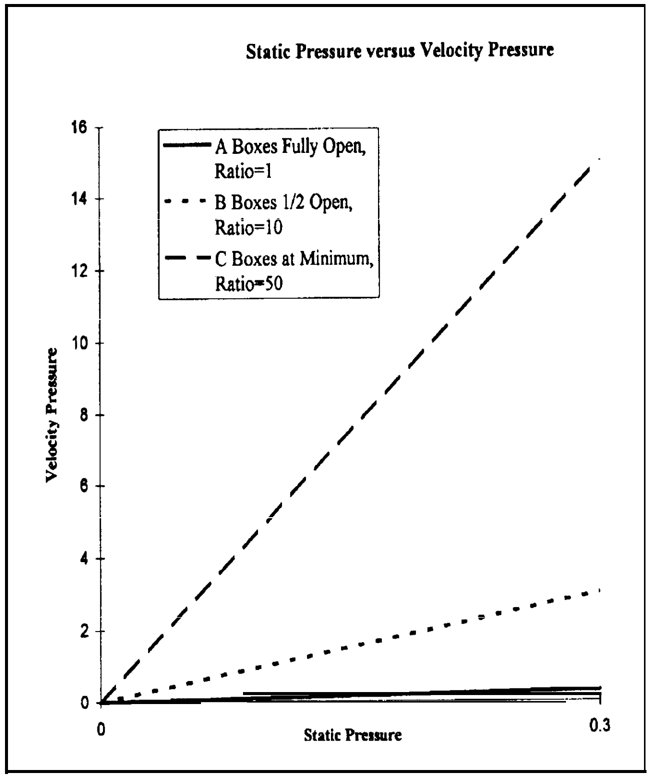

Figure 1. Static and Velocity Pressure at the Fan Outlet of an AHU[1]

Pneumatic Controls with VSD'S: Application

The above control scheme has been applied to an AHU. The particular AHU is one of 33 floor by floor units. Each AHU serves pneumatically controlled VAV terminal box units. Terminal boxes are not tied into any central energy management system, EMS. All AHU have been recently upgraded with VSD's and are remotely controlled by an EMS system. The AHU's originally maintained constant duct static pressure year round. The controls were then modified to maintain a constant value of Ps/Pv which was predetermined.

Hardware changes included two pressure transducers at a cost of $150 each. Static and total pressures were measured approximately 4 duct diameters down stream from the AHU. Total pressure was measured with a single traverse copper tube passing through the center of the duct. Velocity pressure was then determinedly subtracting the static pressure from the total pressure. Both pressure transducers were tied to the EMS with available analog input ports. Total cost of the hardware was a little over $300 dollars and was installed by the building engineering staff. It was fortunate that two available EMS points could be used with no additional cost. Software changes were also done in house with about one hour of effort. The most difficult aspect was choosing the value of Ps/Pv. Because the installation took place in winter, it was assumed that the boxes were close to their minimum positions. (This is not necessarily true if the supply air temperature is reset in the winter as was the case for this building.) With the system operating as usual, the static pressure at the AHU outlet was 2.08” and the velocity pressure was 0.086”. There for Ps/Pv was about 24. By disengaging the pneumatic line feeding the wall thermostats, the terminal boxes were forced to their normally open position. At this condition, Ps/Pv was close to unity. Based on the high value of 24 with the boxes in the minimum position and a low of 1 with the boxes open, the desired value of Ps/Pv was chosen to be 10. It was felt that a value of 10 represented the case in which the average terminal box position was a little more than 1/2 open.

With the chosen value of 10, software code was added to the EMS. Duct static pressure is not allowed to change by more than 0.1” every 1 minute to ensure stable operation. The building operator has experienced no problems since the installation. In fact, the building engineer has noticed that the number of comfort complaint calls has lowered since the installation.

Based on short term watt meter readings, fan energy savings of $350/Year are being realized when compared to the original control method. The $350 savings translates to a 14 month simple payback. If EMS points had to added at a cost of $400 each, the payback would then be 3.14 years. This does include additional cost reductions from reduced maintenance costs associated with lower fan speeds. For this application, fan speed was reduced from 40 Hz to 25 Hz in the winter mode. If realized year round, motor and fan bearing life would be increased by 60% due to the fact that bearing life is inversely proportional to motor speed.

As seen, the use of Ps/Pv does still rely on EMS systems to monitor Ps and Pv. However, the number of points is much lower than monitoring all terminal box units separately. For pneumatic systems with no central EMS at all, it would be possible build a control unit which could operate independently with two pressure inputs (Ps and Pv) and an output to reset duct static pressure.

Other Control Issues

The same procedures applied to air handling systems can be applied to water systems. In practice, if the position of chilled water coil values are known, the speed of a variable speed pump can be reset to ensure that the values are as open as possible, thus reducing pumping energy. Just as with air distribution systems, either DDC or Ps/Pv can be used to monitor each value position or the average valve positions.

The last step in the process would be the integration of all air systems, water systems, heating plants and chilling plants. For a single duct VAV system with electric terminal reheat and water side economizing, the choices available to the operator are supply air temperature and weather or not to use free cooling. If the AH Us are equipped with VSD’s and static pressure reset is employed, it is usually best to operate with the lowest supply air temperatures practical in the summer months. This is because the fan partload power can drop with nearly the cube of the percent flow rate while the chiller power will drop only linearly with the temperature lift. For example, a 500 sqft/ton chiller rated at .65 kW/ton has a design power density of 1.3 Watts/sqft. A 1.2 CFM/sqft fan with 3” water static at the fan and 70% motor/fan efficiency will have a design power density of .61 Watts/sqft. Using partload performance for a York centrifugal chiller, at 70% design cooling load, the chiller power density, with reduced condenser temperature, would be 60% of design or 0.78 Watts/sqft. If the fan power drops with the square of percent flow, the fan power density would be 49% of design, or 0.30 Watts/sqft. Now, if the supply air temperature is raised 5F, the chiller power density would (roughly) decrease linearly with the decrease temperature lift. So the new chiller power density would be 0.78 X (75-50)/(75-45) which equals 0.65 Watts/sqft. The air flow rate however would increase 31%, thus increasing the fan power density to 0.51 Watts/sqft. So the total power density of the chiller and fan at 70% design cooling load would be 1.08 Watts/sqft with the low operating temperature (45F chiller water and 55F supply air), but would be 1.16 Watts/sqft at the higher operating temperature (50F chiller water and 60F supply air). Such an analysis would need to performed for at each operating condition, but in general, if the chiller is in operating, the supply air temperature should be as low as practical to minimize overall chiller/fan energy use. For the building operator, this means operating the chiller with say 44F chilled water and resetting the duct static pressure based on terminal box position as described earlier.

In the winter months, free cooling is available. In this case, the dominant energy use is neither the fans or the chiller, but rather the heat or reheat energy. Although installed reheat power density will vary with regional design weather, installed reheat power density may be 2 to 6 times higher than the installed fan power density. For this reason, it is desirable to reduce heat or reheat energy at the expense of higher fan energy. For a given AHU serving both interior and exterior space, only one supply air temperature is available. Reducing (or eliminating if possible) reheat is accomplished by resetting the supply air temperature as high as possible while still providing enough cooling capacity for the interior spaces. This can be done by monitoring every each box position, in the case of DDC. In this case, the strategy is to set the supply air temperature based on minimum box position. By resetting the supply air temperature, the box with the least air flow should be kept just above its minimum position. This will ensure that the reheat coils are not activated. At the same time, the duct static pressure should be reset to keep the box with the maximum position as open as possible. Each time the supply air temperature is raised, the interior boxes will require more air flow to provide a given amount of cooling. This will in turn require a raise in the duct static pressure. At the end of this process, the supply air temperature will be as high as possible, the duct static pressure will be as high as possible and at least one exterior terminal boxes will be at its minimum position. Any further call for less cooling at the exterior box will cause the reheat coils to be activated.

The conclusion is that integrated terminal box control should be used to reset duct static pressure in the cooling months. The same integrated terminal box control should be used to reset supply air temperature in the heating, reheat months. In this way, total energy use by all equipment is minimized.

Conclusions

Monitoring the operation of VAV systems can lead to control strategies which minimize fan energy use and ensures comfort is controlled. The cost for new buildings with DDC control and VSD AHU's is merely the time to modify the controls. For buildings without DDC, a minimum use of pressure measurements can be used to monitor all the boxes on average. Fan energy can then be minimized. Monitoring the operation of VAV systems can also be used to reset supply air temperatures and reduce reheat energy.

Acknowledgments

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Endnote

By monitoring the static and velocity pressure at the fan outlet, the static pressure down the duct can be reset such that the VAV system operates along line A, B, C or any other line in between. Operating the VAV system to operate along line A would force all the terminal boxes open and thus have the lowest fan energy. Operating the VAV system to operate along line C would force all the terminal boxes to their minimum position and thus have the highest fan energy. Operating the VAV system to operate along line B would force all the average terminal box position to be 1/2 open and thus allow full control in the terminal boxes while reducing fan energy at partload conditions.

References

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