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A CONTROL SYSTEM THAT PREVENTS AIR FROM ENTERING AN AIR-HANDLING UNIT THROUGH THE EXHAUST AIR

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A CONTROL SYSTEM THAT PREVENTS AIR FROM ENTERING AN AIR-HANDLING UNIT THROUGH THE EXHAUST AIR DAMPER

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Synopsis

Traditional air-handling unit (AHU) control systems link the position of the exhaust air damper, recirculation air damper, and outdoor air damper. Tests at the National Institute of Standards and Technology (NIST) on a variable-air-volume (VAV) AHU have shown that air can enter the AHU through the exhaust air damper. This can negatively impact indoor air quality if the exhaust air duct is located near a pollution source.

This paper presents a new control system for variable air volume AHU's that use volume matching to control the return fan. The new control system links only the position of the exhaust air damper and recirculation air damper. During occupied times, the outdoor air damper is in the fully open position.

Simulation and laboratory results are presented to compare the new control system and a traditional control system. Several cases are simulated to examine the effect of damper sizing and system load on airflow in AHU's. The simulations demonstrate that the new control system can prevent air from entering the AHU through the exhaust air damper for conditions that the traditional control system cannot. A case demonstrating the limits of the new control system to prevent this phenomenon is included in the simulation results. The laboratory results provide further evidence that the new control system prevents air from entering the AHU through the exhaust air damper for conditions that cause the phenomenon with the traditional control system.

Introduction

The primary function of an AHU is to provide conditioned air to various rooms in a building. An AHU typically has dampers that are used to control the amount of outdoor air that enters the system, the amount of air exhausted from the system, and the amount of return air from the rooms that is recirculated through the system. Damper control is generally influenced by two main factors, namely, the need to provide ample outdoor air to meet indoor air quality (IAQ) standards, and a desire to conserve energy by limiting heating and cooling in the AHU coils. AHU's are designed with the intent that outdoor air enter the system only through the outdoor air duct and dampers. However, tests at NIST have shown that air can also enter an AHU through the exhaust air damper. Air entering the AHU through the exhaust air damper can have a negative impact on IAQ if, for example, the exhaust air damper is located near a pollution source such as a truck loading dock. Also, in some AHU's, prefilters and/or preheat coils are placed in the outdoor air duct. Air entering the AHU through the exhaust air damper bypasses these components. If the prefilter is bypassed, IAQ may suffer. If the preheat coil is bypassed and the air is very cold, the coils in the AHU could be damaged or the freeze protection thermostat may cause the entire AHU to shut down.

The objective of this paper is to describe a new AHU control system that helps prevent air from entering a VAV AHU through the exhaust air damper. The new control system may also save energy by reducing fan power under certain operating conditions.

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The basic components and traditional control strategies that influence airflow in a VAV AHU are described first. The new control system is then introduced. Equations governing airflow in the AHU are then presented, followed by results comparing the traditional and new control systems. This is followed by experimental results comparing the two control systems. Conclusions of this study are then presented.

Description of Traditional Air-Handling Unit

Figure 1 is a schematic of a VAV AHU. The primary function of an AHU is to provide conditioned air to various rooms. Also, AHU's are usually controlled to maintain a positive air pressure in the building relative to the outdoors. This helps prevent unconditioned air from entering the building through cracks in the building structure.

The supply fan is controlled to maintain a static pressure in the supply duct at a setpoint value. A common strategy for controlling the return fan is to maintain a constant difference between the supply airflow rate and return airflow rate. This strategy is called volume matching. With a volume matching strategy, the difference in the volume of air entering and exiting the AHU through the exhaust and outdoor air ducts must equal the difference in the supply and return airflow rates.

VAV AHU's are usually controlled to maintain a constant discharge air temperature. This is accomplished by controlling the cooling coil, heating coil, and dampers to provide the desired discharge air temperature in the supply duct.

Today's AHU's link the position of the exhaust air damper, recirculation air damper, and

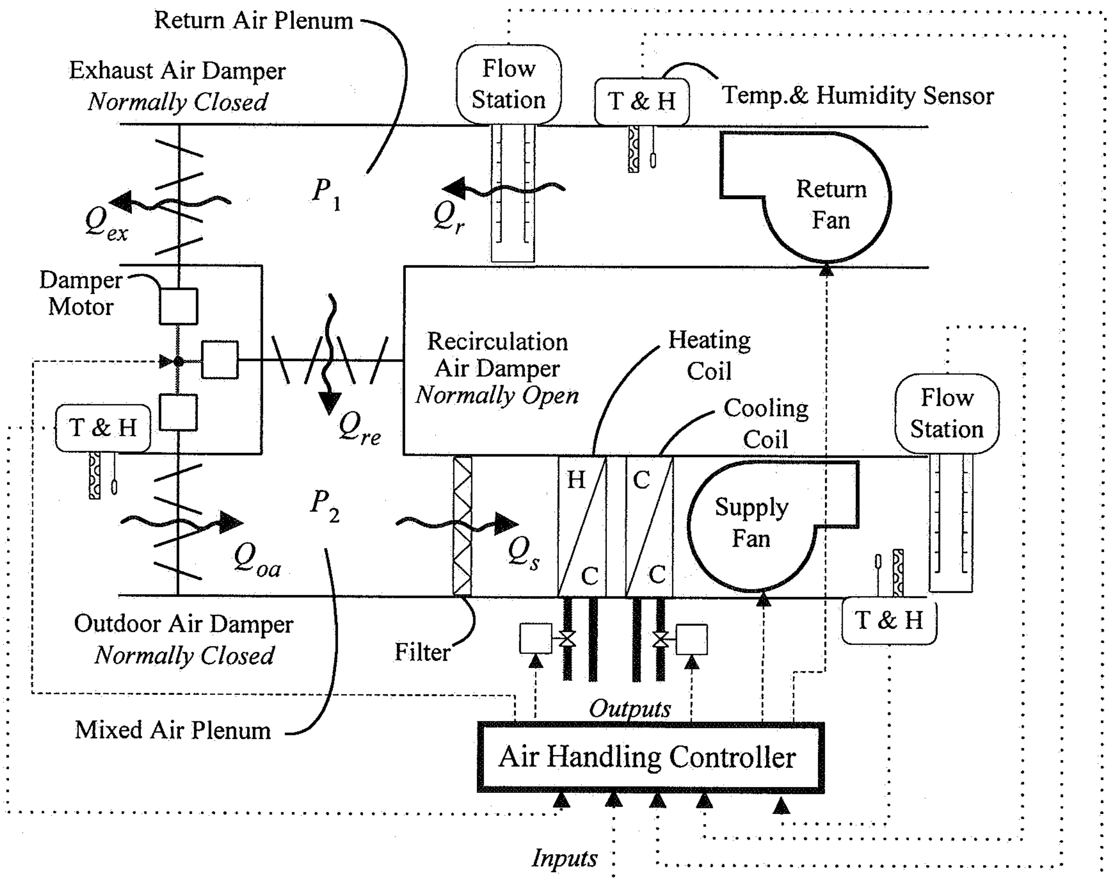

Figure 1 Schematic of air-handling unit.

outdoor air damper. The exhaust air damper and outdoor air damper are normally closed, and the recirculation air damper is normally open. As the exhaust and outdoor air damper begin to open, the recirculation air damper begins to close. Either mechanical linkage or the control system maintains the relationship between the three dampers. For traditional AHU's, the following equations describe the relationship between damper positions:

$$\theta_{re} = 1 - \theta_{ex} \tag{1}$$

$$\theta_{out} = 1 - \theta_{re} \tag{2}$$

where θ_{re} is the fraction of fully open position of the recirculation air damper, θ_{ex} is the fraction of fully open position of the exhaust air damper, and θ_{out} is the fraction of fully open position of the outdoor air damper.

New Air-Handling Unit Control System

The new AHU control system links the position of only the exhaust air damper and the recirculation air damper using the relationship in Equation 1. During occupied times, the outdoor air damper remains 100 percent open, i.e., $\theta_{out} = 1$.

Modeling of Airflow in Air-Handling Unit

This section reviews equations for modeling airflow in AHU's. The equations are based on the conservation of mass and energy. Assuming the density of air is constant throughout the system, conservation of mass applied at the mixed air plenum and return air plenum yields

$$Q_r = Q_{ex} + Q_{re} \tag{3}$$

$$Q_s = Q_{oa} + Q_{re} \tag{4}$$

where Q_r is the return airflow rate, Q_{ex} is the flow rate of air exiting the AHU through the exhaust air damper, Q_{re} is the recirculation airflow rate, Q_s is the supply airflow rate, and Q_{oa} is the flow rate of air entering the AHU through the outdoor air damper. Airflow rates just defined are shown in Figure 1. The airflow rates are related to air velocities and damper areas by

$$Q_{ex} = V_{ex} A_{ex} \tag{5}$$

$$Q_{oa} = V_{oa} A_{oa} \tag{6}$$

$$Q_{re} = V_{re} A_{re} \tag{7}$$

where A_{ex} is the area of the exhaust air damper, A_{oa} is the area of the outdoor air damper, A_{re} is the area of the recirculation air damper, V_{ex} is the velocity of air leaving the AHU through the exhaust air damper, V_{oa} is the velocity of air entering the AHU through the outdoor air damper, and V_{re} is the velocity of air going from the return air plenum to the mixed air plenum through the recirculation air damper.

If the total pressure in the return air plenum (P_1) is greater than atmospheric pressure (P_a), then return air will exit the AHU through the exhaust air damper. The energy equation for return air exiting the AHU through the exhaust air damper is

$$\frac{P_1}{\rho} + \frac{V_{ex}^2}{2} = \frac{P_a}{\rho} + \left(C_{exd} + C_{exit} + C_{screen}\right) \frac{V_{ex}^2}{2} \tag{8}$$

where P_1 is the static pressure in the return air plenum, ρ is the density of air, P_a is the atmospheric pressure, C_{exd} is the loss coefficient for the exhaust air damper, C_{exit} is the exit loss coefficient, and C_{screen} is the loss coefficient of a screen that is placed in the exhaust duct to prevent birds and other small animals from accessing the AHU ductwork. Other dynamic losses due to the ductwork are neglected. A damper loss coefficient is a function of damper position. For opposed and parallel blade dampers, the loss coefficient can be estimated from

$$C_{damper} = a_0 \, e^{a_1\theta + a_2\theta^2} \tag{9}$$

where a_0, a_1, and a_2 are constants that are determined from nonlinear regression, and θ is the fraction that the damper is fully open, e.g., if a damper is half-way open, then θ is 0.5.

If the atmospheric pressure (P_a) is greater than the total pressure in the return air plenum (P_1), then air will enter the AHU through the exhaust air damper. The energy equation for outside air entering the AHU through the exhaust air damper is

$$\frac{P_a}{\rho} = \frac{P_1}{\rho} + \frac{V_{ex}^2}{2} + \left(C_{exd} + C_{en} + C_{screen}\right)\frac{V_{ex}^2}{2} \tag{10}$$

where C_{en} is the entrance loss coefficient.

The energy equation for air entering the AHU through the outdoor air damper is

$$\frac{P_a}{\rho} = \frac{P_2}{\rho} + \frac{V_{oa}^2}{2} + \left(C_{oad} + C_{en} + C_{screen}\right)\frac{V_{oa}^2}{2} \tag{11}$$

where P_2 is the static pressure in the mixed air plenum and C_{oad} is the loss coefficient for the outdoor air damper. The energy equation for airflow from the return air plenum to the mixed air plenum is

$$\frac{P_1}{\rho} = \frac{P_2}{\rho} + C_{red}\frac{V_{re}^2}{2} \tag{12}$$

where C_{red} is the loss coefficient for the recirculation air damper.

The equations used in the AHU airflow analysis are idealized. However, it is believed that this level of analysis does capture the major physical characteristics of interest in this study.

Simulations

Simulations were performed to compare the traditional and new control systems. A simulation program [3] was used to solve the system of equations defined in the previous section. Four cases are compared. The base case simulation has the following characteristics:

Damper Geometry:	$A_{oa} = 2.5$ m^2	$A_{ex} = 2.0$ m^2	$A_{re} = 2.0$ m^2
Minor Losses:	$C_{en} = 0.5$	$C_{exit} = 1.0$	$C_{screen} = 0.32$
Airflow Rates:	$Q_s = 5.0$ m^3/s	$Q_r = Q_s - 1.0$ m^3/s	
Air Properties:	$P_a = 101.4$ kPa	$\rho = 1.202$ kg/m^3	

Additionally, the constants a_0, a_1, and a_2 were obtained by performing a nonlinear regression to data for opposed blade dampers having a ratio of the sum of the damper blade lengths to duct perimeter equal to 0.5 [1,2]. Damper sizes and minor losses were selected based on published guidelines [2].

Figure 2 shows the flow rate of air through the exhaust air damper for the base case simulation for both the traditional and new control systems. Negative values of the exhaust airflow rate indicate that air is entering the AHU through the exhaust air damper. For the traditional control system, outdoor air enters the AHU through the exhaust air damper when it is less than 30 percent open. For the new control system, outside air does not enter the AHU through the exhaust air damper. By opening the outdoor air damper, the flow resistance through the outdoor air duct is reduced and the pressure in the mixing plenum is raised. This raises the pressure in the return plenum to the point where it is greater than atmospheric pressure, thus preventing the reverse airflow phenomenon. Note that as the exhaust air damper approaches the fully open position, the curves in Figure 2 converge. This desirable (albeit expected) result is seen in all ensuing comparisons of the two control systems.

Figure 3 shows the fraction of outdoor air to supply air versus exhaust air damper position for the two control systems. The outdoor air quantity includes air entering the AHU through the exhaust air damper. When the exhaust air damper position is less than 30 percent open, the fraction of outdoor air for the traditional control system remains constant. For this range of damper positions, the flow rate of outdoor air entering AHU through both the exhaust and outdoor air dampers equals the difference between the supply and return airflow rates, i.e., 1.0 m^3/s. Because $Q_s = 5.0$ m^3/s, the fraction of outdoor air to supply air is 0.2.

Figure 4 shows the exhaust airflow rates for the control systems when the area of the recirculating air damper is changed from 2.0 m^2 to 0.8 m^2. Notice that when the exhaust air damper is near the fully closed position, the traditional control system allows air to enter the AHU through the exhaust air damper whereas the new control system does not.

By comparing the results for the traditional control system in Figures 2 and 4, it is seen that the transition point for reverse airflow in Figure 4 occurs when the exhaust air damper is more fully closed. The smaller recirculation damper area corresponding to the results in Figure 4 causes the air velocity between the return air plenum and mixed air plenum to increase for a given airflow rate. Equation 12 shows that increasing the recirculation air velocity increases the pressure difference between the return and mixed air plenums. Thus, increasing this pressure difference helps prevent reverse airflow through the exhaust air damper. However, there is an energy penalty associated with increasing this pressure difference: the fan power will increase.

Figure 5 shows the exhaust airflow rates for the control systems when the supply airflow rate is reduced from 5.0 m^3/s to 3.0 m^3/s. Again, the traditional control system allows outdoor air to enter the AHU through the exhaust air damper and the new control system does not.

Comparing the results for the traditional control system in Figures 2 and 5, it is seen that the transition point for reverse airflow in Figure 5 occurs when the exhaust air damper is more fully open than in Figure 2. Because the airflow difference between the supply and return air ducts is constant, for a given damper position, a higher percentage of the supply air is outdoor air when the supply airflow rate is reduced. This means there is a larger driving potential in the system to draw air into the AHU through the exhaust air damper. Thus, it is expected that the reverse airflow problem occurs more often at low load conditions.

Figure 6 demonstrates a case in which the new control system fails to prevent the reverse airflow problem. This is a more extreme case of the low load example shown in Figure 5. In this case, the supply airflow rate was reduced to 1.5 m^3/s and all other parameters were set to their base case values. Both curves in Figure 6 represent results obtained using the new control system. The curve labeled $A_{re} = 2.0$ m^2 (base case value) shows that the new control system is unable to prevent the reverse airflow phenomenon when the exhaust air damper is nearly fully closed. The second curve shows that by reducing the area of the recirculation air damper (by disabling and closing one or more blades), air is prevented from entering the AHU through the exhaust air damper.

An alternative to reducing the area of the recirculation air damper is to limit its maximum open position. The failed case corresponding to the curve labeled $A_{re} = 2.0$ m^2 in Figure 6 is shown as the curve labeled $\theta_{re} = 1 - \theta_{ex}$ in Figure 7. The curve labeled $\theta_{re} = 0.5\ (1 - \theta_{ex})$ corresponds to the new control system with maximum open position of the recirculation air damper limited to 50 percent. Thus, by limiting the maximum open position of the recirculation air damper, air is prevented from entering the AHU through the exhaust air damper.

The results in Figures 6 and 7 are presented to demonstrate that the new control system does not eliminate the possibility of outdoor air entering the AHU through the exhaust air

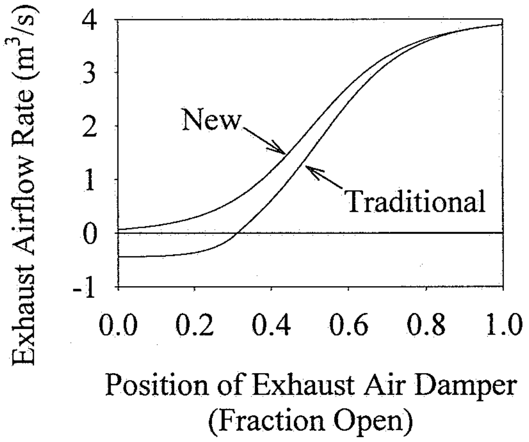

Figure 2 Exhaust airflow rate for base case.

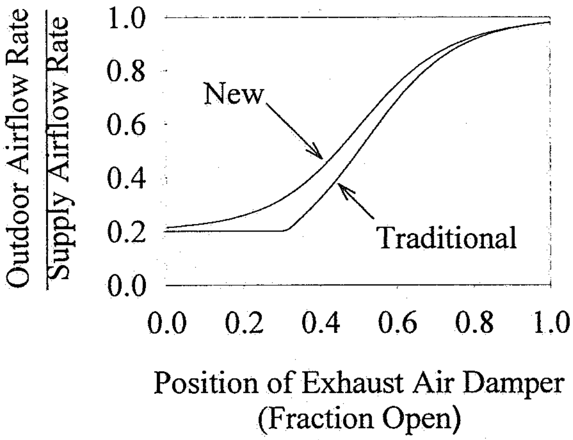

Figure 3 Fraction outdoor air for base case.

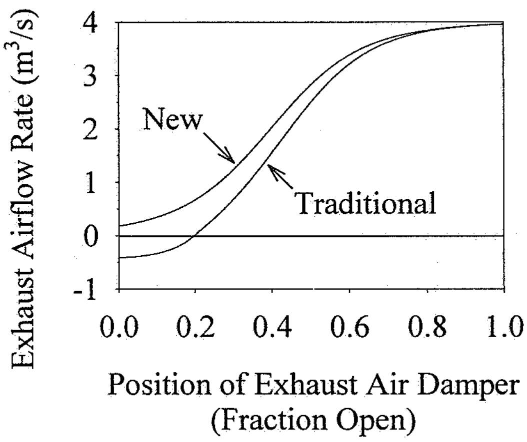

Figure 4 Modified base case: $A_{re} = 0.8$ m^2.

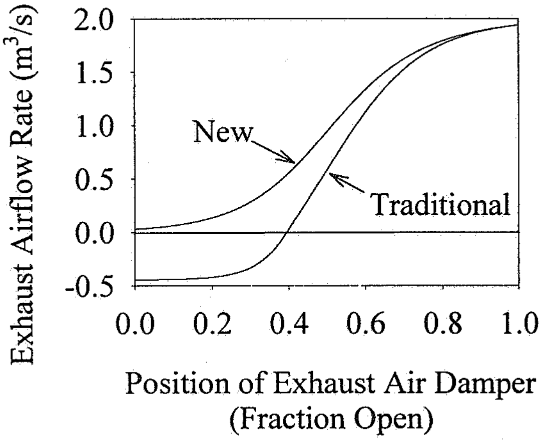

Figure 5 Modified base case: $Q_s = 3.0$ m^3/s.

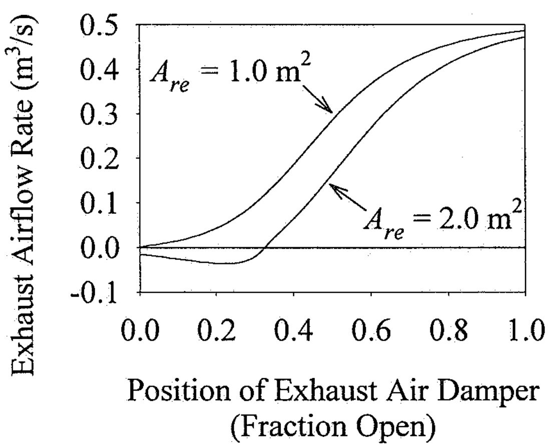

Figure 6 New control system when $Q_s = 1.5$ m^3/s and $Q_r = 0.5$ m^3/s.

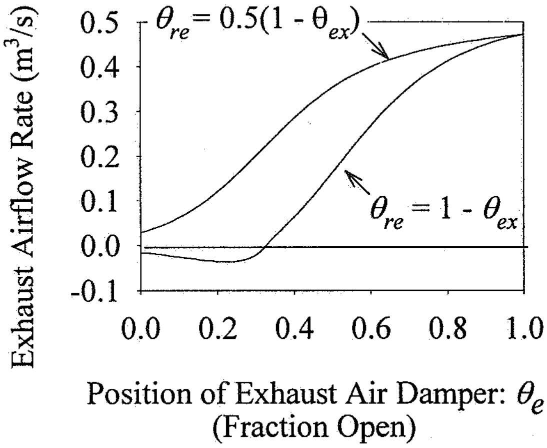

Figure 7 New control system when $Q_s = 1.5$ m^3/s and $Q_r = 0.5$ m^3/s.

damper, it simply reduces the likelihood of occurrence of the phenomenon. Under extreme conditions, it may be necessary to modify the flow resistance through the recirculation air damper to alleviate the problem.

Laboratory Tests

To validate the simulation results, the control systems were implemented in the NIST laboratory AHU. The outdoor and recirculation air dampers of the laboratory AHU have a parallel blade geometry and the exhaust air damper has an opposed blade geometry. Each damper has a separate actuator. Airflow measurements were obtained using airflow stations and differential pressure transducers. The airflow stations yield an average velocity pressure for a cross section of the duct. The differential pressure transducers are accurate to ± 0.05 percent of the reading + 0.001 percent of full scale [4].

To evaluate the error associated with the airflow measurements, the differential pressure measurements obtained with the airflow stations were compared with those obtained using a Pitot tube traverse. The recirculation airflow station error ranges from 6 to 7 percent. The error associated with the return airflow station is difficult to determine because the Pitot tube traverse measurements are not reliable. This is probably due to the presence of abrupt changes in duct geometry directly upstream and downstream of the airflow station. Thus, an alternative comparison involving the recirculation and return airflow station measurements was undertaken. This comparison consisted of sealing the exhaust air damper and measuring the recirculation and return airflow rates over a range of damper positions and fan speeds. The mean value of the disagreement in the measurements was 6.9 percent. For all conditions, the recirculation airflow rate was higher than that of the return airflow rate. Thus, in the results that follow, return airflow rates are corrected by multiplying the airflow rate by 1.069. Details of the assessment of measurement errors are available [5].

The exhaust airflow rates for the traditional and new control systems are plotted in Figure 8 as a function of the normalized control signal to the exhaust air damper, where normalized control signals of 0 and 1 indicate that the damper is commanded to the fully closed and fully open positions, respectively. The dashed (traditional) and solid (new) curves in Figure 8 are fourth order polynomials that were fit to the data using least-squares regression. For the experimental results, the supply airflow rate was nearly constant and approximately equal to Q_S = 1.38 m^3/s. The return fan was controlled to maintain the flow difference between the supply and return air ducts at 0.57 m^3/s.

The curves in Figure 8 are very similar to those presented in the simulation results. Although the data for the new control system indicates that airflow through the exhaust air damper becomes negative as the exhaust air damper is closed, no evidence of reverse airflow through the exhaust air damper was observed when smoke tests were performed. Smoke tests did confirm that air is drawn into the AHU when the traditional control system is used and the exhaust air damper approaches the fully closed position.

The experimental results lend credence to the simulation results obtained for the two control systems. The new control system is easy to implement and will reduce the range of conditions for which air can be drawn into the AHU through the exhaust air damper. However, it does not eliminate the possibility of this phenomenon. Thus, it may be necessary to take additional measures if the problem persists.

Conclusions

The objective of this paper was to describe a new VAV AHU control system that helps prevent air from entering an AHU through the exhaust air damper. Simulation and experimental results presented in this paper demonstrate that air can enter an AHU through the exhaust air damper for a traditional control system that links the positions of the outdoor air damper, exhaust air damper, and recirculation air damper.

The new control system links only the positions of the exhaust and recirculation air dampers and was developed for AHU's that use a volume matching control strategy for the return fan. For the new control system, the outdoor air damper is fully open during occupied

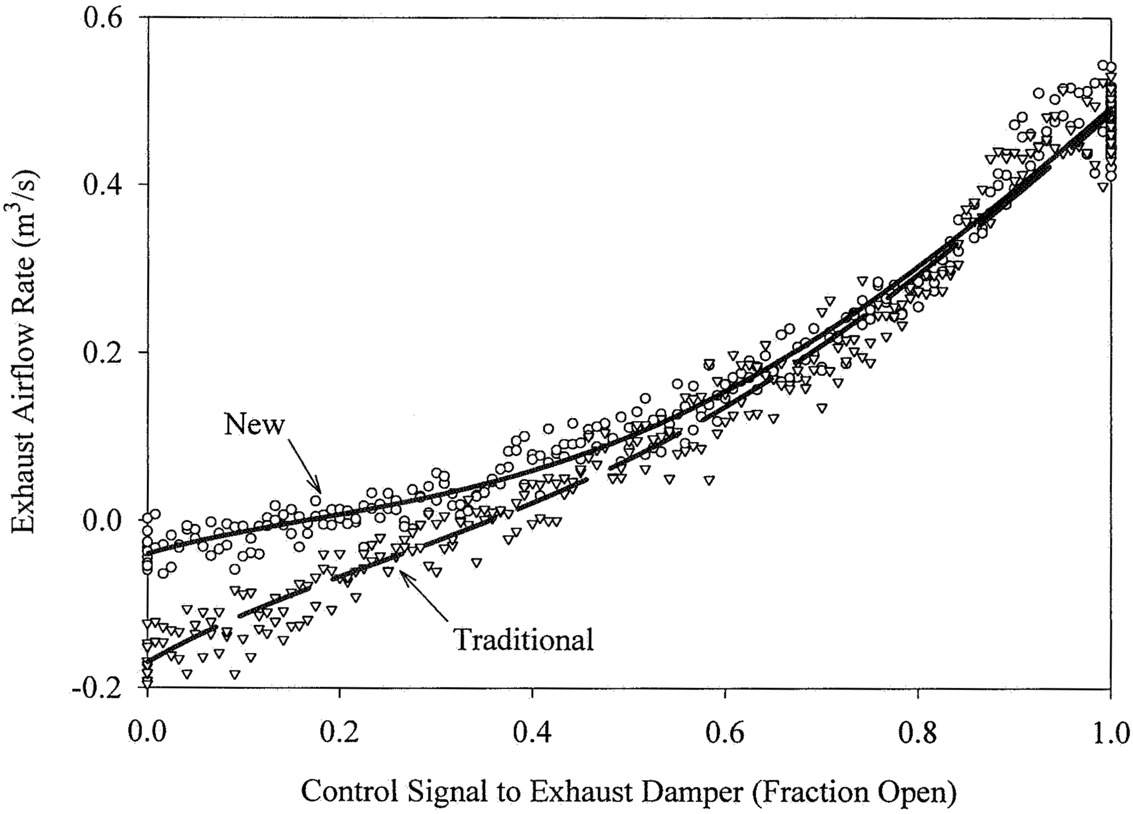

Figure 8 Results from NIST Air-Handling Unit Laboratory.

times. Simulation and experimental results demonstrated that the new control system helps prevent the reverse flow phenomenon, although the potential for the problem is not eliminated. The new control system can fail if the recirculation air damper does not provide a sufficiently large flow resistance. In this situation it is necessary to disable one or more of the recirculation air damper blades or to limit the maximum open position of the damper. Finally, the new control system is easy to implement and will save energy by reducing the fan power.

Acknowledgments

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