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DEVELOPMENT OF A FAN AIRFLOW STATION FOR AIRFLOW CONTROL IN VAV SYSTEMS

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ABSTRACT

A fan airflow station has been developed by the authors. This station is used to determine fan airflow using fan speed and fan head as inputs. These inputs can be measured accurately in variable air volume (VAV) systems, so they can be used as a cost-effective control monitoring method for heating, ventilation and air conditioning (HVAC) systems. Theoretical models have been developed for the flow station using both second and third order relationships between the airflow and the inputs. The theoretical model has been experimentally tested and excellent agreement between the model and the experimental values was found. The second order model was within 1.52% of the experimental values, and the third order model was within 1.71%. The third order model is much more complex than the second order model, and offers no advantages.

INTRODUCTION

With the energy crisis of the 1970’s a push was made by engineers to design more energy efficient HVAC systems. It was during this time that variable air volume (VAV) systems began to be used commonly, rather than constant air volume (CAV). By modulating the supply airflow these systems supply less cold air to rooms which do not need as much cooling, thus saving reheat energy. With several terminal boxes modulating to satisfy cooling loads, the overall systems had a varying flow requirement. There were several ways to accommodate this requirement. Simply adding flow resistance in the ductwork would reduce the flow. However by varying the speed of the fan, or by deploying inlet guide vanes, fan energy could also be saved.

The more efficient of these solutions is to vary the fan speed [1, 2]. This is typically carried out using a variable frequency drive (VFD), a device which modulates the frequency of the alternating current electrical signal. However, CAV systems, though less energy efficient, have an advantage in that they are easily controlled. After the initial testing and balancing the building can have the proper outside airflow rate established, and the proper difference in the supply and return airflow rates, thus ensuring proper building pressurization.

Building pressurization is required to eliminate infiltration [3]. Infiltration is uncontrolled airflow into a building, and it can have many negative effects on the building. The most important is that infiltration brings moisture into the building during the cooling season. This moisture can condense, causing damage to the building structure, to insulation, and promoting growth of mold and mildew [4]. Other detriments of infiltration can include local thermal comfort problems (in exterior zones, particularly during very cold weather) and introduction of air pollutants into the building. Conversely, if ventilation air enters the building in a controlled fashion it can be drawn from an area isolated from local sources of pollution, then filtered and dehumidified before it enters the occupied space of the building.

To ensure that there is no infiltration, the mechanical system should supply more air to the building than it returns and exhausts. This difference between supply flow rate and combined exhaust and return flow rates is the exfiltration flow rate.

\[ Q_{ef} = Q_s - (Q_{eb} + Q_r) \]  

The exhaust amount is normally constant, and the desired exfiltration flow rate is constant. The situation becomes complicated when the supply and return amounts vary.

Three building pressurization control methods have been developed over the last 30 years. These methods are fan tracking, direct building pressure control, and volumetric tracking. The fan-tracking method (sometimes referred to as VFD proportional drive-slaving) involves a link between the
VFDs controlling the supply and return fans. The supply fan is normally controlled to keep a constant static pressure at some remote point in the ductwork. When less supply air is needed, the supply fan slows down and the return fan also slows down.

The problem is that the system flow resistance characteristics change on the supply side (the terminal box dampers), but do not change on the return side. This means that the supply fan is running at a different speed and is encountering different pressure than at design conditions, so that flow is not proportional to speed. The return fan, however, is operating at a fixed flow resistance, so that a link between the speeds of the return fan and supply fan will not maintain correct building pressure except at the point for which the system is balanced [5-7].

The direct building pressure control method measures the building pressure with reference to outdoor pressure, and uses this as a control input for the return fan. If the building is under-pressurized the return fan slows down so that less return air is drawn back from the space, and less air is sent to the relief air duct. In theory this is an excellent solution, but in practice the problem comes in the measurement of building pressure. It is very difficult to accurately measure such small static pressure differences as are required for building pressurization (typically 0.05 in. w.g. (0 Pa) to 0.10 in. w.g. (25 Pa) depending on climate [8]). The sensors need to be calibrated, and anecdotal evidence suggests that very few operations managers know the location of the pressure sensor. If the pressure sensor is subject to wind then it won’t serve its purpose. Also, the pressure may vary throughout the building, particularly in a multi-story building. This method, using existing technology, is extremely difficult to implement.

Volumetric tracking is a control method that can be effective under certain circumstances [6, 9-10]. It involves putting a flow station in the main supply duct and another in the main return duct. The return fan speed is controlled by comparing the flow rates in the supply and return ducts. The fan is modulated to keep this difference at a constant setpoint, ensuring a constant exfiltration rate (assuming exhaust as a constant). The problem with this method lies in the flow station measurements. For accuracy within 5 to 10% a flow station normally needs a straight, unimpeded duct run for 6-10 diameters upstream and several downstream of it [11]. There are very few systems that have such duct runs in the main supply and return ducts.

Therefore, to properly control building pressurization a fan airflow station that can accurately determine airflow across a fan provides a viable solution.

This paper presents an experimental verification of a fan airflow station model proposed by Liu [12]. The fan airflow station uses fan speed and fan head as inputs, with a second-order relationship to fan airflow. An extension of this fan airflow station is also presented in this paper, using an assumption of a third-order relationship between fan head, fan speed, and flow rate. The experimental results are compared to the second order and third order models to assess the validity of the fan airflow station. The fundamental theory, experimental equipment and procedures, and experimental results are discussed in detail in this paper.

### THEORY

The fan airflow station includes a differential pressure transducer and fan speed transducer. The fan speed transducer may be replaced by the control system’s command to the VFD. The fan airflow station can be implemented using typical Energy Management and Control Systems (EMCS). Figure 1 presents a schematic diagram of the airflow measurement and control in a typical VAV system using fan airflow stations. The supply airflow is determined using the measured fan head and the VFD speed. The return airflow setpoint is then determined by taking the difference between the supply airflow and the sum of the building exhaust and exfiltration flow rates. The return airflow rate is calculated using the measured return fan head and speed. The return fan speed is modulated to maintain the return airflow setpoint.

The fan airflow station is developed based on the following theories. The fan head can be regressed as a function of the airflow using the design fan curve. Equation (2) is a typical polynomial regression curve.

\[
H_d = \sum_{i=0}^{6} a_i Q_d^i
\]

where \(H_d\) and \(Q_d\) are the fan head and airflow rates for the fan at 100% speed.

![Diagram of Airflow in VAV Systems using Fan Airflow Stations](image)

Figure 1. Diagram of Airflow in VAV Systems using Fan Airflow Stations

When the fan is operating at reduced speed, the fan head and airflows can be determined with Eq. (3), based on the fan laws.

\[
H = \omega^2 \sum_{i=0}^{6} a_i \left( \frac{Q}{\omega} \right)^i
\]

where \(\omega = \frac{N}{N_d}\), \(H = \omega^2 H_d\), and \(Q = \omega Q_d\).

When a third order polynomial regression equation is used to represent the design fan curve, then fan airflow can be predicted using the measured fan head and the fan speed with Eq. (4). The regression coefficients are labeled \(a_i\),
\[ Q = \sqrt{\frac{-q + \sqrt{q^2 + \frac{4p^3}{27}}}{2}} + \sqrt{\frac{-q - \sqrt{q^2 + \frac{4p^3}{27}}}{2} - \frac{a_2}{3a_3}} \]  

(4)

where \( q = \frac{2a_3^3}{27a_2^2} - \frac{a_2a_1}{3a_3^2} + \frac{a_0a_2^2}{a_3} - H \) and \( p = \frac{a_1}{a_3} - \frac{a_2^3}{3a_3^2} \).

Using a second order polynomial regression of the design fan curve, as developed by Liu [15], the fan airflow can be predicted using the measured fan head and fan speed with Eq. (5).

\[
Q = \frac{-a_1 - \sqrt{a_1^2 - 4a_2\left(a_0 - \frac{H}{\omega^2}\right)}}{2a_2} \sqrt[3]{\omega}
\]  

(5)

EXPERIMENTAL METHOD AND SETUP

The objective of the experimental portion of this research was to verify that the theory holds true. The experiments were carried out on an actual air handling unit (AHU) installed in a building systems laboratory. The AHU, shown in Fig. 2, has two sets of heating and cooling coils and a bank of filters (a low efficiency pre-filter and a high efficiency final filter) in the 13' (4 m) length between the flow hood and the supply fan.

The experiments conducted included:

a) Fan speed/EMCS calibration
b) Generation of a design fan curve
c) Fan airflow station verification

The first of these experiments was to determine the correlation between the EMCS command to the VFD and the actual fan speed, since the VFD speed will eventually be represented by the EMCS command. The EMCS command was set in 5% increments from 100% down to 5%, then back up to 100%. The fan speed was measured using the digital laser tachometer.

The design fan curve is defined as the correlation between fan head and fan airflow when the fan is running at its maximum speed. For experiment b), the fan speed was controlled at 3,450 rpm. The airflow was increased from 250 cfm to 2,500 cfm (0.118 to 1.18 m³/s) by modulating terminal box dampers and the flow resistance device installed for this study. The airflow through the fan was measured using the flow hood. The fan head (the differential pressure measured across the fan) was measured with a digital pressure sensor. For this and all parts of the study each flow and pressure measurement was taken over a period of time, and time-averaged. Corresponding flow and pressure measurements were always made simultaneously (over the same time period). A total of 50 measurements were taken in this portion of the study.

For part c) of this study—verification of the fan airflow station model—220 tests were conducted. These included 20 fan speeds (5% to 100%) for each of 11 flow resistance configurations. At full speed, the flow rates achieved with these resistances ranged from 115 to 2,200 cfm (0.054 to 1.04 m³/s). In each test, the EMCS command, the fan head, and the airflow were measured and recorded. The fan speed was calculated using the EMCS command and the previously mentioned correlation factor. The fan head and fan speed were then used as inputs to the fan airflow station model, and the calculated fan airflow was compared to the measured fan airflow to verify the theoretical model.

Finally, error analysis was conducted. The flow hood error—7 cfm (0.0033 m³/s)—constitutes the maximum airflow measurement error. The maximum fan airflow station error is determined to be 2 cfm (9.43 x 10⁻⁵ m³/s) (See appendix for calculation details).
RESULTS

a) Fan speed/EMCS calibration

The correlation of actual fan speed with the EMCS command to the VFD is plotted in Fig. 3. The EMCS command varies from 5% to 100% while the fan speed varies from 500 rpm to 3400 rpm. At each point on the graph there are two data (one on top of the other): one from the increasing fan speed series, and one from the decreasing. These data show a very reliable linear relationship between the EMCS and the actual fan speed. There is no measurable hysteresis on this system. The regression line gives a value to relate EMCS speed with actual fan speed. When the EMCS command is 0, the VFD output is 7 Hz, and the fan speed is 316 rpm.

b) Generation of a design fan curve

The design fan curve that was generated is shown in Fig. 4. The third-order polynomial regression fits the data very well. The data in the higher range were taken by opening and closing dampers in the distribution system. Quite a few of the 50 data are in this range. In the lower flow ranges it became difficult to create sufficient flow resistance because of duct leakage in the distribution system. For the lowest flow rates the supply duct was covered over with a board and sealed with putty, and for increasing flow rates the putty was removed and the board was shifted slightly.

\[ y = 30.758x + 315.68 \]
\[ R^2 = 0.9998 \]

Figure 3. Fan Speed as a Function of EMCS Command

\[ y = 30.758x + 315.68 \]
\[ R^2 = 0.9998 \]

Figure 4. Measured Fan Curve at Full Speed

\[ y = -8.86E-07x^2 + 7.55E-04x + 5.79E+00 \]
\[ R^2 = 0.98E-01 \]

Figure 5. Measured Airflow and Fan Head Data

Focusing on normal operating conditions—fans operating in the stable region, and from 25 to 100% of the design flow rate—leaves a total of 47 data in this study. Figure 6b compares the stable operation zone measured data to the fan airflow station modeled data. The fan airflow station values match the directly measured values very closely. A linear regression of the data with the regression forced through zero, gives an error of 1.71% for the cubic model, and 1.52% for the quadratic model. When the regression is not forced through zero, the y-intercept values are less than 6 cfm (0.003 m³/s), which is lower than the maximum measurement error. Thus the 2nd and 3rd order models give nearly identical accuracy.

The data generated in the experiment appear to be highly repeatable. There are no drastic outliers, or unexpected results within the normal fan operation range. The very close agreement between measured and predicted values shows that observed amongst many of the data in the low flow range. These anomalous data are the result of the tests involving a very high flow resistance. This means that they fall within the unstable region of the fan curve. They are used to determine where the start of the stable operating range occurs. Those data which fall outside the stable operating range are ignored in this study.
the airflow can be accurately predicted using fan speed and fan head as inputs. The proposed method has excellent potential for accurate determination of airflow in actual systems because the fan speed and fan head measurement methods in actual systems are very similar to those in the experiment.

CONCLUSIONS

The fan airflow station model values match the direct measured airflows well and show excellent repeatability. This leads to the conclusion that the fan airflow station model is valid.

A comparison of the results using the quadratic and cubic models shows no advantage to the cubic model. The cubic model is difficult to implement due to its complicated solution. Since the fan curve can be represented accurately in the normal operating range using the second order regression, this model should be used.

Because fan airflow can be measured accurately using fan speed and fan head, the fan airflow station provides an accurate and cost effective means of airflow measurement in HVAC systems. A direct and important application of this model is building pressurization control.

These experiments were conducted using a direct drive plenum fan under controlled test conditions. Further experiments could examine the effects of pressure transducer location, use of manufacturer’s fan curves rather than in situ data, and the performance of the fan airflow station model on belt driven fans.

NOMENCLATURE

\( H \) – fan head
\( N \) – fan speed
\( Q \) – airflow rate
\( a_i \) – polynomial regression coefficients
\( \omega \) – fan speed ratio

Subscripts

\( d \) – design, at 100% fan speed
\( r \) – return
\( s \) – supply
\( xf \) – exfiltration
\( xh \) – exhaust

APPENDIX: ERROR ANALYSIS

The fan airflow station use the measured fan speed and fan head to predict airflow, using the following equation:

\[
Q = \frac{-a_1 \pm \sqrt{a_1^2 - 4a_2(a_0 - \frac{H}{\omega^2})}}{2a_2} \quad (A-1)
\]

The maximum airflow prediction error depends on the accuracy of both the fan head and the fan speed measurement.

\[
\delta Q = \sqrt{\left(\frac{\partial Q}{\partial \omega}\delta \omega\right)^2 + \left(\frac{\partial Q}{\partial H}\delta H\right)^2} \quad (A-2)
\]

By introducing Eq. (A-1) into Eq. (A-2), the maximum airflow projection error becomes:

\[
\delta Q = \sqrt{\left((2a_1 Q + a_2 \omega)\delta H\right)^2 + \left(\frac{a_1 - 8a_2}{2(2a_1 Q + a_2 \omega)} - \frac{1}{2}\right)^2} \quad (A-3)
\]

The maximum measurement errors of fan speed and fan head are 1 rpm and 0.005 in.w.g. (1.2 Pa), respectively. The regression coefficients \( a_1 \) and \( a_2 \) are -0.0000009 and 0.0008, respectively. Figure A-1 shows the maximum predicted airflow error as a function of airflow and fan speed. The maximum error decreases from 2 cfm (9.4 x 10^{-4} m³/s) to 0.1 cfm (4.7 x 10^{-5} m³/s) when the airflow increases from 500 cfm (0.24 m³/s) to 2500 cfm (1.2 m³/s). The maximum relative error is less than 0.5%.
Figure A-1. Maximum Projected Airflow Error versus the Airflow and Fan Speed

REFERENCES