

LABORATORY TESTS OF THE NONLINEARITY OF OUTDOOR AIR PERCENTAGE AS A FUNCTION OF DAMPER POSITION AND INDUCED INLET PRESSURE

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ABSTRACT

The effects of damper position and inlet pressure on the ratio of outside to total supply air were tested in an HVAC laboratory. It was found that the ratio varies nonlinearly with the damper position at low outdoor airflows. With induced pressure at the inlet to the main air-handling unit (simulating a windy outdoor air condition), significantly more outside air was taken in than the damper position might suggest. The relationship between flow and damper actuator signal or position is key to the proper control of the outside air damper.

INTRODUCTION

Recent concerns about airborne pollutants have placed much emphasis on indoor air quality, particularly in forced-circulation systems. The simplest way to ensure acceptable air quality (in unpolluted areas, of course) is to bring 100% outside air into a building. The cost of conditioning this air, on the other hand, makes this a rather expensive option, especially in extreme weather conditions. A more typical control scheme is to bring in the minimum amount of outside air necessary to satisfy local code requirements (or ANSI/ASHRAE 62-1989), except when ambient conditions permit the use of the economizer mode.

The process of bringing outside air into a building is relatively simple. Dampers are modulated to either recirculate return air back into the main air-handling unit(s) or to expel it to the outdoors. In the latter case, fresh outside air is then drawn into the building to make up for the exhausted air.

In the case of minimum outside air intake, it is fairly well established that the ratio of fresh air to total building supply air will not be a linear function of the damper position, since the outside airflow will increase rapidly as soon as the damper begins to open yet will not show much change in flow once the damper is close to fully open. It is also generally assumed that this ratio will remain a constant for a given damper position, regardless of the total supply airflow rate. However, the test results presented here show that this may not always be the case.

TEST FACILITY

The experiments were performed at an HVAC systems and controls laboratory. This facility consists of a single-deck variable-air-volume system that can supply 12,000 ACFM to four individual zones (see Figure 1). The main air-handling unit (AHU-1) is a front vertical discharge system, and the return fan is an in-line axial fan. Both fan speeds are controlled using variable-frequency drives.

The temperature and humidity of the simulated outside air are maintained by the outside air-conditioning station (OACS) located upstream of AHU-1. The OACS is also used to control the total pressure at the inlet to AHU-1.

Return air can either be exhausted from the building or recirculated back into AHU-1. Both the recirculated and outside air dampers on AHU-1 are linked and are modulated by an electronic actuator. The freeze protection and relief dampers are also controlled from the same signal that is used to control the outside air damper. The damper angular position is linear with respect to the control voltage. Both the recirculated and outside air dampers are of parallel-blade configuration.

All data acquisition and control in the HVAC laboratory is handled by a network of dataloggers and computers. More than 250 channels are monitored every 10 seconds, while concurrent analog control signals are generated by a combination of timed digital outputs and pulse-width-modulator signal generators.

Duct static and total pressures are recorded at several locations. The static pressures are measured using differential pressure transducers attached to four taps located on the duct walls. The total pressures are measured using pitot tubes located in the center of the airstream.

Air temperatures are measured using platinum RTDs, each packaged with individually calibrated transmitters. The sensor lengths are 12 or 24 feet, depending on the duct size, and are mounted such that they sense the average duct air temperature. These sensors have a time constant of approximately one minute due to their metal sheath and a rated accuracy of 0.1°F.

Air velocities are measured with a combination of

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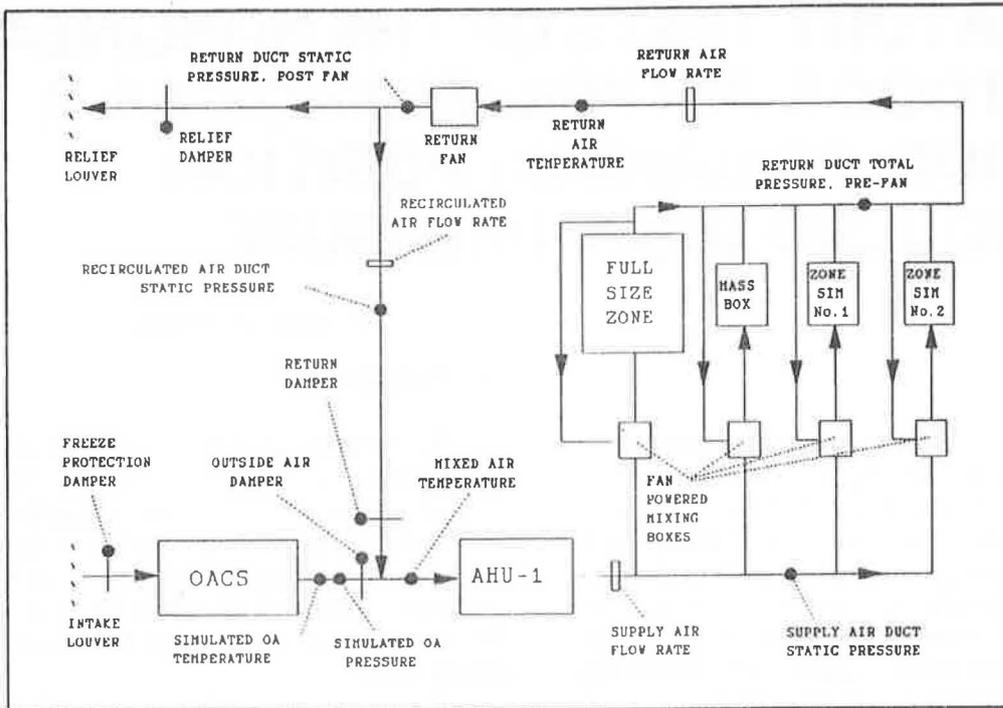


Figure 1 HVAC laboratory air system schematic (showing placement of sensors).

averaging pitot tube arrays and differential pressure transducers. The volumetric airflows are then calculated by multiplying the duct area by the air velocity after making corrections for the atmospheric pressure and the air temperature at the location of the airflow station. The conversion equation from differential pressure to airflow is given by

$$V_{air} = A_{duct} \sqrt{\frac{2 \Delta P}{\rho}} \quad (1)$$

where

- V_{air} = volumetric flow rate of air,
- A_{duct} = area of duct,
- ΔP = pressure differential across the pitot tube array, and
- ρ = density of air (a function of the atmospheric pressure and air temperature).

The airflow stations are periodically recalibrated by comparing pitot tube traverses to the output from the pressure transducers. The pressure transducers have been shown to be quite accurate within their specified pressure range, with any significant error coming solely from sensor drift over time.

Measurement errors could arise from an uneven velocity profile in the duct. The pressure differential from the higher airspeeds would tend to dominate, and the overall reading would be too high. An additional source of error is the decrease in effective duct area due to the

pitot tube grid hardware. This hardware can occupy as much as 20% of the total duct cross section. The airflow stations were found to have poor accuracy and slightly nonlinear response. While we are able to compensate for much of this intrinsic error in the computer-based conversion equations, the accuracy is still only good to within approximately 7% to 10% of the total volumetric airflow reading.

There are 11 airflow stations in the lab, one of which is located downstream of AHU-1 (which measures the total supply airflow rate) and one in the recirculated air duct. The amount of outside air drawn into AHU-1 is the difference between these two airflows.

A redundancy check is made on the ratio of outside air to total supply air by using an energy balance based on the temperatures of the return, mixed, and outside airstreams. The ratio of outside air to total supply air, based on the energy balance approach, is given by

$$\% \text{ OA} = 100 \times \frac{T_{ma} - T_{ra}}{T_{oa} - T_{ra}} \quad (2)$$

where

- $\% \text{ OA}$ = ratio of outside airflow to total supply airflow,
- T_{oa} = outside air temperature,
- T_{ma} = mixed air temperature, and
- T_{ra} = return air temperature.

TEST PROGRAM

Sixteen separate tests were performed. The goal of each test was to quantify the ratio of outside air to total supply air for different supply airflow rates. During each test, the computers continuously increased the supply fan speed from 20% to 100% full speed at a rate of 2% per minute.

The chilled-water system was in operation during these tests to ensure a relatively large temperature differential between the outside and return airstreams. Warm outside air was brought in unconditioned, while the air-handling unit and individual load coils in the zones were used to lower the return air temperature.

The 16 tests arose from various combinations of experimental parameters that were fixed for the duration of each test. We wanted to examine the effects of (1) different outside air damper positions, (2) a positive pressure on the outside air inlet louvers, and (3) the return fan speed control. The experimental parameters were as follows:

- Return fan speed control

Two methods were used to control the return fan speed. The first involved a simple tracking scheme, where the return fan operated at 5% *full scale* less than the supply fan; i.e., if the supply fan was at 80% of its maximum 900 rpm, then the return fan was at 75% of its maximum rotational speed of 1,500 rpm. The second method had the return fan operating so as to maintain a return duct static pressure of -0.1-in. equivalent water pressure, measured between the return risers from zone simulators 1 and 2.

- OACS fan control

Normal operation of the OACS fan is such that atmospheric pressure is maintained at the inlet to AHU-1. When using this scheme, the fan is necessary only to overcome the pressure drop associated with the dampers and coils in the OACS. Half the tests were performed this way. During the other tests, the OACS fan maintained a 0.16-in. water column equivalent pressure at the inlet of AHU-1. This pressure corresponds to that produced by a steady wind of 20 mph impinging on the outside air intake louvers of AHU-1.

- Outside air/recirculated air damper position

A different test was performed for each of several settings of this damper. The settings used were 10%, 20%, 30%, and 40% full open.

RESULTS

Data were taken every 10 seconds, then averaged over 60 seconds and written to a data file for later analysis. Each test took approximately one hour to

perform—40 minutes to ramp the fan speed up and 20 minutes to let the system air temperatures stabilize for a new damper position and induced inlet louver air pressure.

The OACS fan speed was controlled such that the induced inlet air pressure at AHU-1 was maintained at the desired setpoint (0 in. or 0.16 in. w.c.) with a standard deviation of ± 0.005 in. water column, while the typical maximum deviation was no more than 0.015 in. water column.

The ratios of outside air to total supply air, as calculated by the airflow rates and from the temperatures (Equation 2), agreed fairly well throughout all the tests; a slight discrepancy between the two was noted, typically 4% full scale but sometimes as much as 10% at low supply airflows when the OA percentage was quite high. This error most likely arises from airflow station calibrations.

Figure 2 shows the airflow readings from the data set when the return fan was controlling the return duct static pressure, the AHU-1 inlet louver damper pressure was at 0 in. w.c., and the OA damper was at 10% full open. Figure 3 shows the duct air pressures for the same test.

Figures 4 through 7 show the calculated airflow as a function of the total supply airflow. The calculated airflow is based upon the temperature energy balance from Equation 2 because the accuracy of the airflow stations suffers at very low flows.

A slight periodic variation in the outside air percentage is noticed in some of the data sets. This is due to the consistent cycling of the chiller used to maintain a large temperature difference between the outside and return airstreams.

No provisions were made for actual outside air pressures against the *relief* dampers. This louver empties to an alleyway, however, and faces east (away from the prevailing winds), so it is assumed that actual outside induced pressures would not affect the experiments significantly.

Figure 4 shows the results using a zero inlet damper pressure and the return fan tracking the supply fan. The airflow is increased from 3,000 to about 11,000 ACFM (roughly 20% to 100% full-scale output). The return airflow rate during these tests matched the supply airflow quite well, demonstrating that the system is in good balance using this technique. The outside/total airflow ratio does not increase at the same rate as the increase in damper position and, in fact, remains fairly constant for each of the four tests. At 10% damper position, the ratio stays between 10% and 15%, while at 40% damper position, the ratio is between 16% and 23%.

Figure 5 gives the same data as in Figure 4, except with a positive inlet pressure at AHU-1. Comparison of these two figures shows an obvious difference in the behavior of the airflows. The actual outside airflow rate appears to remain basically constant over the entire range of supply airflow rates, despite the fact that the recir-

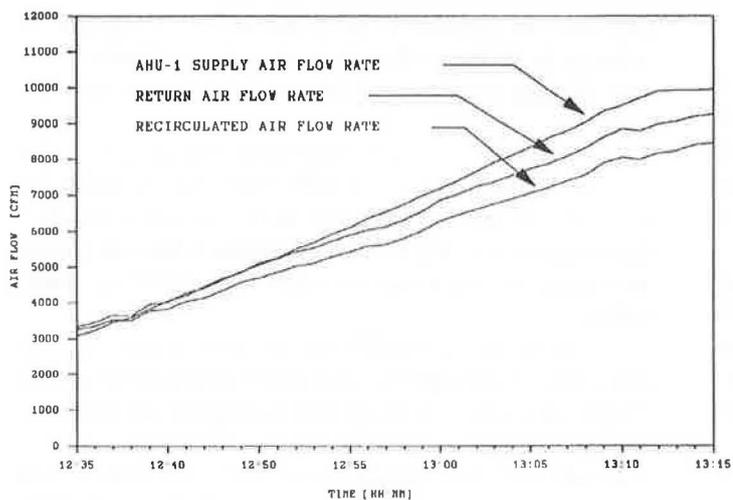


Figure 2 Typical airflows.

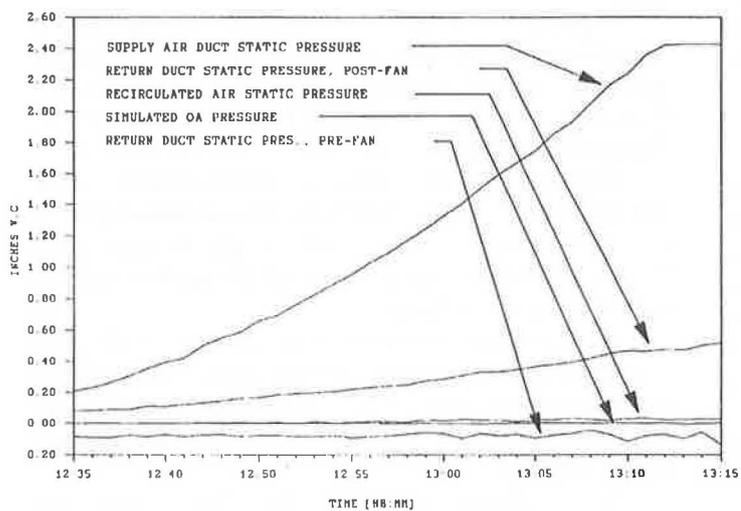


Figure 3 Typical air pressures.

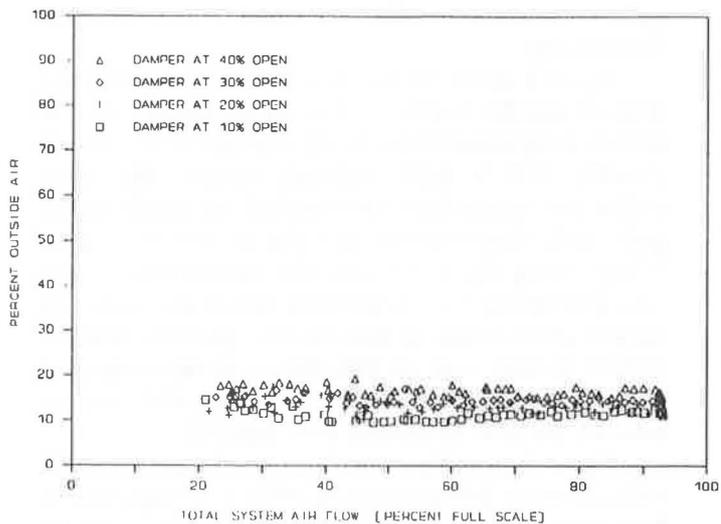


Figure 4 Outside air percentage with return fan tracking supply fan and atmospheric pressure at inlet louver of AHU-1.

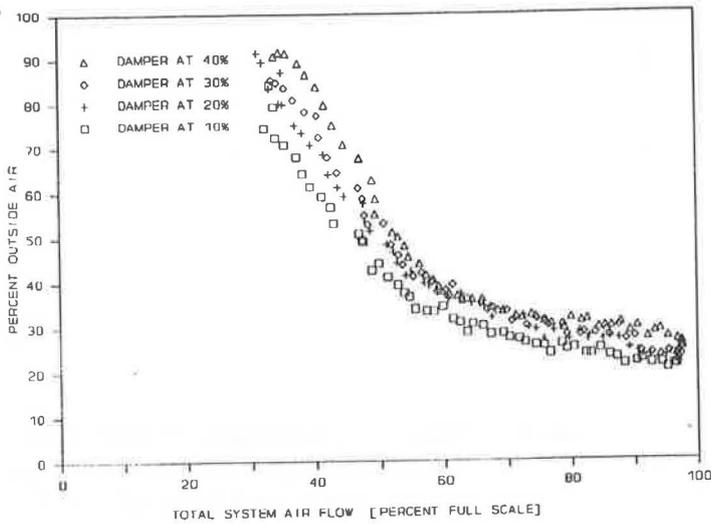


Figure 5 Outside air percentage with return fan tracking supply fan and positive pressure at inlet louver of AHU-1.

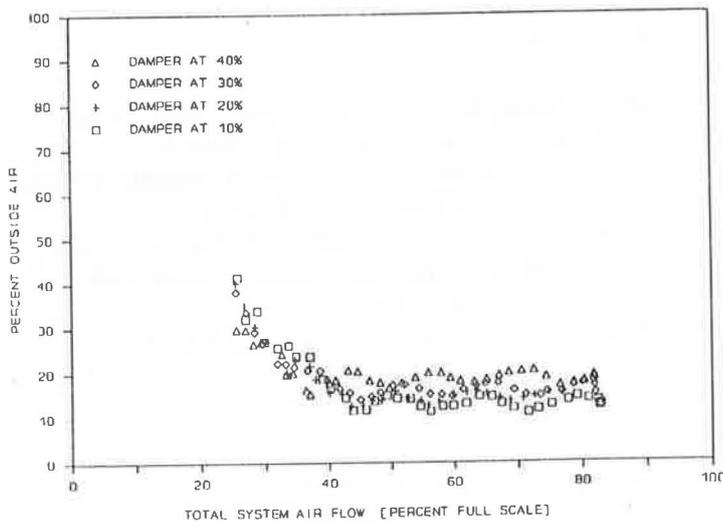


Figure 6 Outside air percentage with return fan controlling return duct static pressure and atmospheric pressure at inlet louver of AHU-1.

culated air duct static pressure is noticeably higher than the induced outside air pressure. As a result, at low demand for building air (i.e., low total airflow) the ratio of outside air increases dramatically. This is the case for all four tests. As in the zero inlet pressure tests, the overall OA ratio does not vary much with damper position; rather, the main cause of variation of this ratio is changes in the total airflow rate.

Figure 6 shows the effect of increasing the supply airflow rate when the induced inlet pressure is at atmospheric and the return fan is controlling to a negative pressure in the return duct. Here the OA percentage at low supply airflow is roughly twice the percentage at maximum supply flow.

Figure 7 gives the experimental data for a positive inlet pressure and a return fan controlling to a negative return duct pressure. During these tests, the OA percentage appears to *increase* as the supply airflow rate increases, albeit very slightly. This increase falls within the range of experimental error, however, and should not be viewed as a true effect.

CONCLUSIONS

For all the tests, it was noticed that the ratio of outdoor air to total supply air did not vary much for a given particular fan speed, despite damper positions between 10% and 40% full open. An interesting result is that the OA percentage remains relatively constant over the entire range of fan speeds when (1) there is an atmospheric pressure at the inlet to AHU-1 and the return fan tracks the supply fan, and (2) there is positive pressure at the inlet to AHU-1 and the return fan is controlling to a negative return duct pressure. For the other two cases, the OA percentage was greater (often dramatically so) at lower supply airflow rates. This is an important finding, since air handlers are usually operated in the range of 10% to 30% outdoor air.

Of particular interest is the case of positive inlet pressure and the tracking return fan (Figure 5). Since the recirculated air duct static pressure is higher than the induced inlet pressure, one might expect the OA percentage to be very low, yet the airflow data show just the

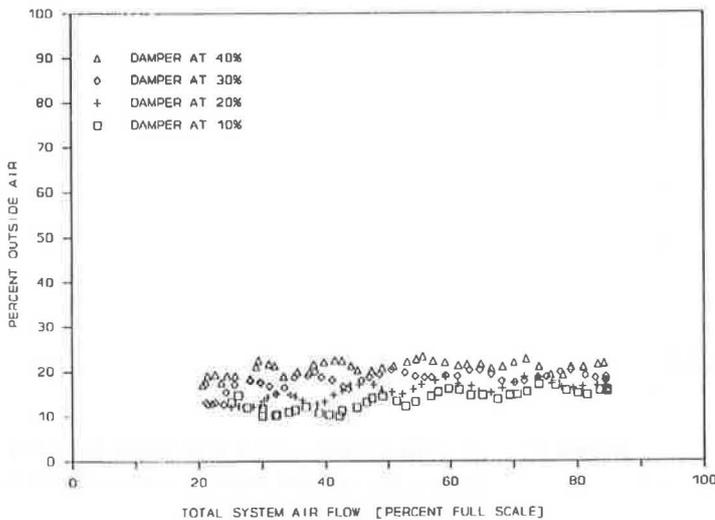


Figure 7 Outside air percentage with return fan controlling return duct static pressure and positive pressure at inlet lower of AHU-1.

opposite. Also during these tests, the balance of the system is thrown off, with the return airflow rate much less than the supply flow. It is believed that the elevated pressure in the recirculation duct is caused not by the return fan but rather by the induced pressure at the inlet to AHU-1. This would also cause an elevated pressure rise across the return fan for any given fan speed, thereby lowering the total flow rate as compared with the analogous data from the tests at zero induced pressure. This would also explain the overall rise in system pressure; the only occurrence of positive return duct pressures is during this series of tests.

In our system, which is fairly typical of current commercial construction, the amount of outside air entering the main air-handling unit varies nonlinearly with respect to the damper position. Energy management systems designed to minimize energy use while maintaining comfort conditions could not rely on the damper position as an accurate indicator of the outdoor air ratio.

Similarly, this ratio also depends on the relative pressure against the inlet louvers. A positive pressure similar to that caused by wind was shown to induce a much larger outdoor airflow than damper position might suggest. Design and placement of inlet dampers should consider prevailing wind directions and speeds. For example, a VAV system with reheat may be taking in more outside air than expected on cold days. This could cause large loads on the heating plant.

Finally, it was shown that the accurate measurement of outside air percentages is a difficult task, even under laboratory conditions using well-maintained equipment. This raises the possibility of inaccurate values of OA flow as they are measured in typical building HVAC systems from the use of single-point temperature references, misplaced sensors, etc. With the increased demand for such data arising from the new standards for fresh air in buildings, it becomes apparent that much more research is needed in this field.

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APPENDIX

DATA ACQUISITION AND CONTROL SYSTEM COMPONENTS

The equipment used in the data acquisition and control system is manufactured by Hewlett-Packard, including the following:

- HP3497A Data Acquisition and Control Unit (2)
- HP3456A Precision Multimeters (2)
- HP9000 Series Computers (6, various models)
- HP Shared Resource Management System (1)
- HP48000 RTU for OACS experimental control (1).

Sensors used include the following:

- Airflow - Air Monitor Company multiple pitot tube arrays calibrated with pitot tube traverses; MKS model 223BD pressure transducers used with arrays.
- Atmospheric Pressure (used in airflow calculations) - MKS model 122AA absolute pressure transducer.

Duct Air Pressures - MKS model 223BD pressure transducers.

Duct Air Temperatures - Omega Engineering platinum resistance temperature detectors (100-ohm nominal resistance); MAMAC Systems TE-211 platinum averaging RTD.

Actuators include the following:

Fan Speed - Allen-Bradley variable-frequency drives on all fans.

Damper Motion - Belimo SM24-SR electronic damper actuator.

Signal Generator - MAMAC Systems PW600 pulse width modulator.