Measurement for the control of fresh air intake

The Standard 62-1989 recommendations for measuring and documenting outdoor air intake flow are discussed

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This article documents the key reasons why ASHRAE *Standard 62-1989*, "Ventilation for Acceptable Indoor Air Quality," recommends the measurement and documentation of outdoor air intake flow on constant volume and variable air volume (VAV) systems. The article also analyzes the most critical portion of the fan system control algorithm, the control of minimum outdoor air during mechanical refrigeration (cooling coil operation).

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The results presented clearly demonstrate that intake air cannot be controlled on either constant volume or VAV systems with a fixed-position minimum outdoor air damper (or even with a minimum position damper, reset by supply air flow, on VAV systems). The lack of control of minimum ventilation air is due to the system's operational characteristics and boundary conditions, such as wind and stack effect.

The recent industry focus on Indoor Air Quality (IAQ) has finally stimulated systematic engineering evaluation of the control of outside air intake rates in constant volume and VAV systems. Consulting engineers are keenly interested in providing proper control of fresh air intake for indoor air quality for two powerful reasons: the health and productivity of the occupants, and the avoidance of potential legal liability.

ASHRAE recommends the measurement of intake air flow in "mechanical ventilation" systems and requires the "documentation" of outdoor air control during actual "system operation" of the building. Efficient designs, based on the "6.1 Ventilation Rate Procedure" (ASHRAE 1989a), now require the measurement of outdoor air intake flow, for both constant volume and VAV systems, to optimize energy use as well as to verify outdoor air intake rates.

In the new *Standard 62-1989*, the provisions for the measurement of outdoor air intake and documentation are stated as follows:

Measurement—"When mechanical ventilation is used, provision for air flow measurement should be included."

Documentation—"Design criteria and assumptions shall be documented and should be made available for operation of the system within a reasonable time after installation."

The foreword of the standard states: "It must be recognized, however, that the conditions specified by this Standard must be achieved during the operation of buildings as well as in the design of buildings, if acceptable indoor air quality is to be achieved. To facilitate this, the Standard includes requirements for ventilation design documentation to be provided for system operation."

In 1988, ASHRAE awarded a research project TRP-590 to the University of Missouri-Rolla (ASHRAE 1988). Due to be completed in September 1990, this project studied dynamic VAV systems, in both economizer and cooling coil operation modes.

This article covers only the portion of control required for minimum outdoor air intake, during the operation of the cooling coils. (The economizer system dynamics are also part of the control of fresh air, but will not be examined here. The economizer cycle will be the subject of a follow-up research project.) For VAV systems, it is important to note that the control of outdoor air must be stable during the economizer cycle for proper building pressurization.

The following system analysis is based on conservative assumptions. The boundary conditions of stack effect and wind will be compounded in buildings taller than the "low-rise" equivalent used in this model. And geographically, the stack effect will be worse in Minneapolis, for example, than in Atlanta.

A key parameter in the following analysis is the mixed air plenum pressure, which will be more negative in supply/exhaust fan systems. The pressure will be less negative in supply/return fan systems, depending on the dampers selected. For this reason, the more conservative supply/return fan system was used as a model.

This research has also dramatically identified that damper selection is an area requiring additional investigation. Initial, unpublished test data (Ebtron 1985) on the operation of intake louver/damper systems indicates that the current ASHRAE source documents (Brown, Fellows 1957; Dickey, Coplan 1942) for the *Handbook of Fundamentals'* damper flow coefficients are not sufficient, and that the tables will require significant modifications.

Although the setpoint for the outdoor air intake may be greater than the flow required for indoor air quality, there is evidence to indicate that higher rates are required. A report published test data on the lack of outdoor air flow control for both constant volume and VAV systems (Persily, Grot 1985). The report's conclusion, based on data gathered by gas chromatography of a sulfur hexafluoride tracer, included the following:

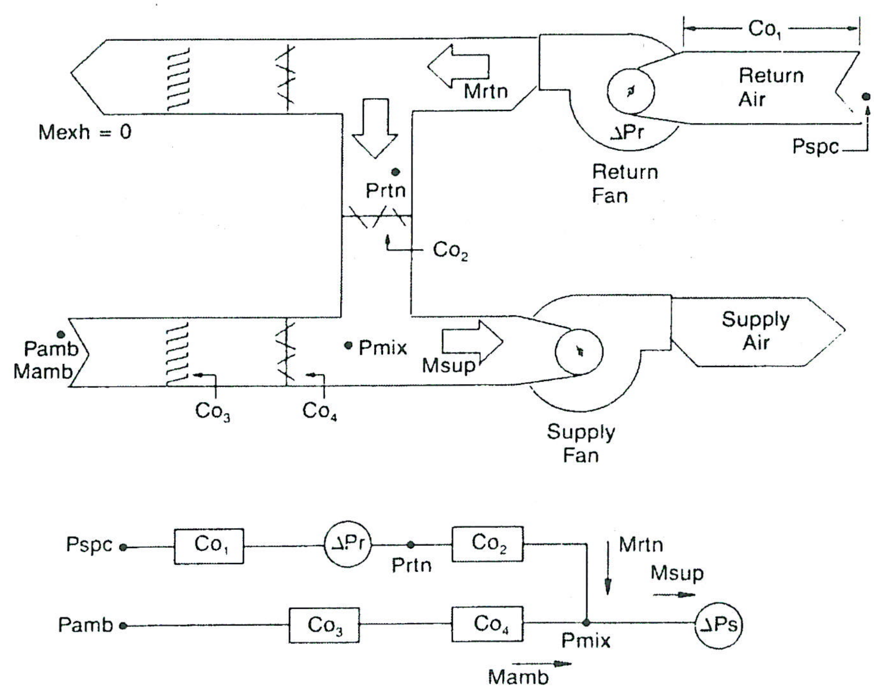
"The minimum ventilation rates were compared to minimum outdoor air intake levels suggested by ASHRAE, and we found that most of the buildings were operated very close to or below the [1985] ASHRAE recommendation. Two of the buildings were operated well below this recommended ventilation rate. Local variations in air distribution and problems of ventilation efficiency can lead to effective ventilation rates in specific areas of the building that are significantly lower than the average rate for the building."

The Persily report highlights the impact of air infiltration (or building leakage). Because of the relatively lower building leakage coefficients in modern designs, this variable needs to be quantified in any thorough building ventilation model.

Although it is beyond the scope of this article, tighter building designs can increase the need for air flow measurement to more adequately control building pressurization, as well as to meet IAQ standards.

Fan system hardware

For this article, a supply/return fan system model was used for both constant volume and VAV system analysis. These two-fan systems were analyzed when the cooling coil is operating with minimum outdoor air intake. The fan systems did not require exhaust at the air handling unit, during minimum outdoor intake, due to the operation of local exhaust fans (i.e., exhaust from toilets). The VAV system had the capability of flow variation down to 30 percent of design flow.



Flow Equations

(Pspc − Prtn) α $Co_1 \cdot M^2$ rtn
(Prtn − Pmix) α $Co_2 \cdot M^2$ rtn
(Pamb − Pmix) α (Co_3 + Co_4) M^2 amb

Abbreviations

Co_1 = Return Duct Flow Coef. = Constant
Co_2 = Return Damper Flow Coef. = Constant
Co_3 = Intake Louver Flow Coef. = Constant
Co_4 = Intake Damper Flow Coef. = f < Damper Angle

Operation

Msup Range (30% - 100% Flow)
Pspc = 0"
Pmix = − 0.40", when Msup at 100% Flow
Pamb = Function of Ambient Temp. & Wind

Mrtn = Mass Flow Return Air
Mamb = Mass Flow Ambient Air
Msup = Mass Flow Supply Air

Pspc = Space Pressure
Prtn = Pressure at Inlet to Return Damper
Pmix = Pressure in Mixed Air Plenum
Pamb = Ambient Pressure

Figure 1. Schematic of supply/return fan system model.

A typical 12,000 cfm fan system was chosen, with its fresh air intake located 50 ft below the top of a "low-rise building" (ASHRAE 1989b). The 60 × 72 in. intake louver was sized for a 0.15 in./wc pressure drop, at a face velocity of 400 fpm, according to ASHRAE recommendations (ASHRAE 1989c), limiting the possibilities of water penetration (Johnson Controls).

Two opposed blade dampers with 5.5-in. blades were used. The main intake damper opening was 60 × 60 in., and a minimum outside air damper opening of 60 × 12 in. These dampers were located near the intake louver and were sized to produce the same 400 fpm face velocity and to reduce the possibilities of water penetration (see *Figure 1*).

The mixed air plenum pressure at 100 percent supply flow was −0.40 in./wc, which corresponds to a 0.25 in./wc pressure drop across the larger intake damper at full flow. The selection of −0.40 in./wc for the initial value was based on both industry experience and on the required slope of the flow coefficient versus angular damper position for multi-bladed dampers, for both mechanical refrigeration and economizer cycles (Brown, Fellows 1957).

The upstream pressure of the return air damper decreased from an initial value of 0.15 in./wc, as the square of flow. Therefore, the full-flow pressure drop of the return air damper is 0.55 in./wc. An opposed blade return damper was selected at an 800 fpm full flow face velocity, due to the flow coefficient requirements for economizer operation, with its companion outdoor air mixing damper. The return damper flow coefficient at full flow was 17 (ASHRAE 1989d).

VAV pressure variations

Figure 2 shows the mixed air plenum as a function of supply air flow during the mechanical refrigeration cycle. The return air damper flow coefficient is held at a constant 17 (fixed position). The pressure in the mixed air plenum was calculated by taking the required pressure differential across the return air damper and subtracting the pressure upstream of the return damper:

Mixed air plenum pressure =
Required pressure differential across return air damper −
Inlet pressure to return air damper

The fresh air intake flow rate is a function of inlet pressure, outlet pressure, and the louver/damper/duct system flow coefficient. The outlet pressure is the mixed air plenum pressure. It is because of the mixed air plenum pressure variance that ASHRAE recommends direct air flow measurement of outside air in-outdoor air intake damper, the fresh air flow drops linearly with supply flow. Since most VAV fan systems operate predominantly at 65 percent flow and do not have intake damper reset, the outdoor air intake flow is reduced 35 percent during normal operation.

Figure 6 shows the lack of control when a fan system is setup at 25°F with no wind, and is operated at 85°F with no wind. The constant volume fan system operation point is on the vertical axis at 88 percent outdoor air intake flow, which corresponds to a 12 percent reduction in fresh air intake. On the VAV fan system, the stack effect is constant while the mixed air plenum pressure is variable. At 55 percent supply flow, there is no outside air intake, because there is no pressure differential across the intake damper. Resetting the intake damper is of little consequence because the pressure variation of the stack effect is much larger than the pressure variation of the mixed air plenum.

Figure 7 indicates the effects on the air handling system when the system is setup at 70°F with a 15 mph stagnation wind, and operated at 70°F with a 15 mph crosswind. The constant volume system operation point is on the vertical axis at 80 percent flow, corresponding to a 20 percent reduction in fresh air intake. The degradation of fresh air intake with supply air flow is more pronounced than by the stack effect conditions shown in *Figure 6*. Below 65 percent flow, there is no intake of outside air. Again, resetting the intake dampers is ineffective.

Figure 8 considers the setup conditions of 25°F with a 15 mph stagnation wind and operated at 85°F with a 15 mph crosswind. This combines the effects depicted on *Figure 6* and *Figure 7*. The constant volume system operation point is on the vertical axis at 68 percent flow, which corresponds to a 32 percent reduction in fresh air intake. The degradation of outdoor air intake with supply air flow is extremely significant. When a VAV system is operated under these conditions, there will be no outside air intake during a majority of the time. Below 80 percent of maximum flow, there is no intake of outside air. Yet again, resetting the intake damper is inconsequential.

Measurement technology

Direct measurement of outdoor air is at least three times more accurate than the value of intake air flow calculated by subtracting the return from the supply air flow. (This assumes that the error is a constant percent of flow.) This is true since the flow of the supply air is always greater than three times the outdoor air flow. Also, the error in the "calculated" (supply minus return) outdoor air flow will at times be even worse. This occurs when the supply and return flows combine to produce a low intake value.

The ASHRAE-recommended maximum face velocity for intake louvers is 400 fpm (ASHRAE 1989c). This corresponds to 800 fpm "free-area" velocity on an intake louver with 50 percent free area. Because most standard performance louvers have less than 50 percent free area, one can expect free-area velocities to be higher than 800 fpm, when using ASHRAE's recommended maximum.

In most HVAC systems, there is little room to increase duct velocity for velocity pressure measurement by reducing duct size due to the proximity of fan systems to exterior walls. There is also the necessity to avoid water carryover with induced higher velocities near the intake louver (Johnson Controls). Only the most expensive louvers (AMCA Certified for Air and Water Performance) will prevent

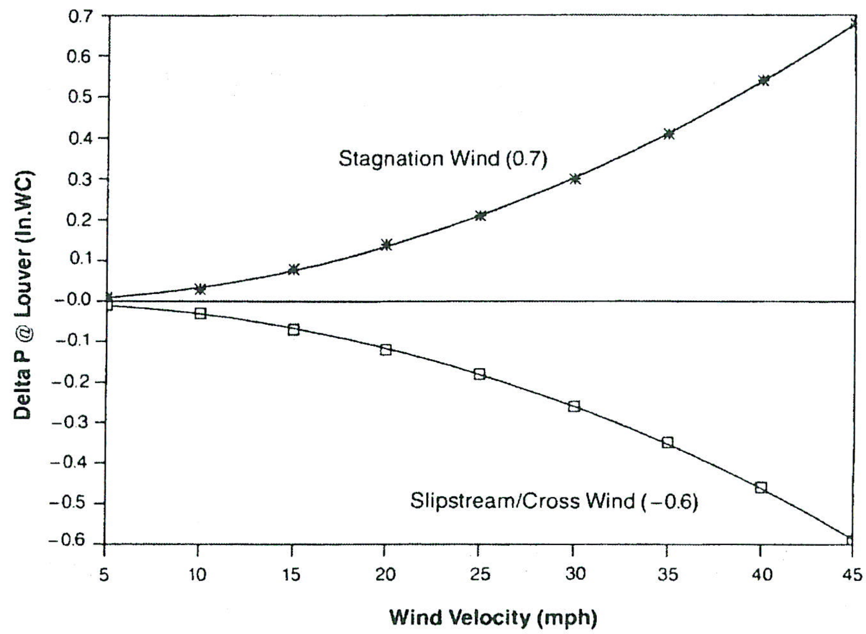


Figure 4. Intake louver pressure, comparing delta P at louver and wind velocity.

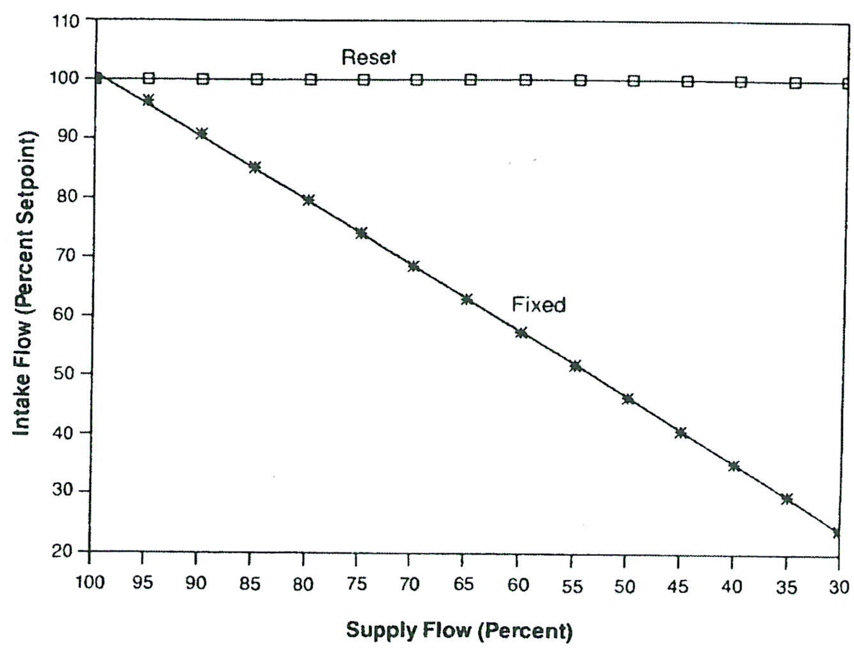


Figure 5. Variable mixed air plenum condition, comparing intake flow and supply flow.

take, or resetting the outdoor air damper as a function of supply flow:

"A velocity controller installed in the outside air intake ensures that dampers are open sufficiently to provide the minimum outside air flow desired, when total flow is reduced. A less costly method is to schedule a variable minimum position of the outside air damper from the duct static pressure sensor or the fan control operator, either of which reacts directly to system volume reduction." (ASHRAE 1987)

The calculations presented in this article show that the "less costly method" of resetting the outdoor air damper based on supply air flow is not adequate.

Stack and wind pressure effects

Figure 3 shows the pressure at the intake louver, for the fan system under analysis (for 50 ft elevation from neutral plane). The data were calculated by using the equation for stack effect (ASHRAE 1989e), which states pressures to temperature:

$$P_{se} = 0.192(p_a - p)(z_2 - z_1)$$

where,

P_{se} = stack effect, in. of water
p_a = density of atmospheric air, lbs/ft^3
p = density of air within ducts, lbs/ft^3
z = height in feet

Figure 4 shows how pressure at the outdoor air intake louver varies with wind conditions for a "low-rise building" (ASHRAE 1989b). When the wind is blowing directly into the "low-rise building" wall containing the air intake damper, part of the velocity pressure is converted to static pressure (ASHRAE 1989f). The maximum "local wind pressure coefficient" (Cp) for a stagnation wind, lower elevation on a "low-rise building", is 0.7. Therefore, the pressure at the intake louver is equal to 0.7 times the velocity pressure.

When the wind is blowing at right angles to the intake louver face, a cross-wind or slipstream condition exists that reduces the pressure. In this case, the Bernoulli equation (ASHRAE 1989g) also shows that the pressure is reduced by the velocity pressure. The maximum "local wind pressure coefficient" (Cp) for a lower elevation on a "low-rise building" is −0.6, at a slipstream wind of 30 degrees. Therefore, the pressure at the intake louver is decreased by an amount equal to 0.6 times the velocity pressure.

The location of the intake louver is 50 ft below the top of the building. There is no flow through the exhaust louver due to local exhaust and therefore exhaust louver interactions have been ignored.

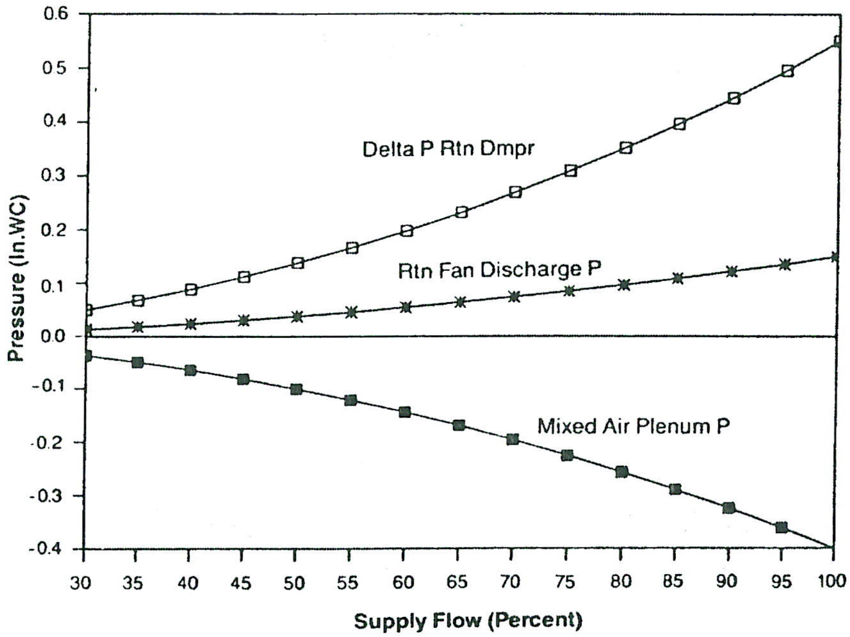


Figure 2. Return damper, inlet and outlet pressure.

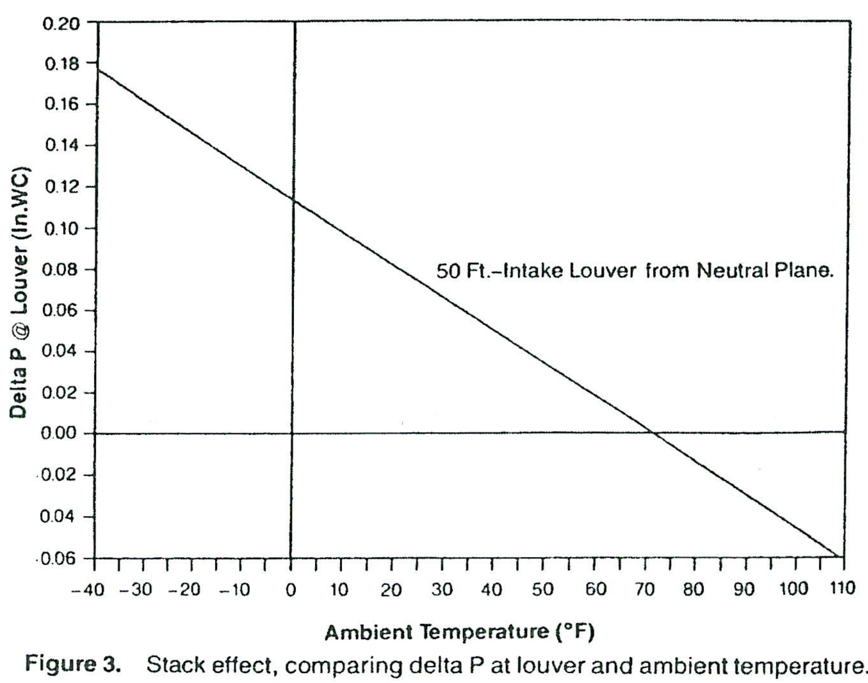


Figure 3. Stack effect, comparing delta P at louver and ambient temperature.

System operational effects

Figures 5, 6, 7 and *8* demonstrate the effects of mixed air plenum pressure variation, stack effect, and wind conditions on the control of minimum outdoor air intake during the operation of the cooling coil. Each figure is based on the system being set up at one environmental condition (temperature, wind, etc.) and operated at another.

There are two curves plotted on each figure. One curve is for a fixed intake damper position and the other is for the minimum outdoor air damper being reset from supply air flow. (The reset assumes that the damper can be accurately positioned to achieve the exact flow coefficient required.) The point of operation for constant volume systems is the point on the curve for 100 percent supply flow.

Figure 5 emulates a fan system that is set up and operated under the same conditions. It graphically shows why ASHRAE specifies resetting the outdoor air intake damper on VAV systems (ASHRAE 1987). Without resetting the water carryover at free-area velocities greater than 800 fpm (AMCA 1989) and are of little help in avoiding entrainment of wind-driven rain.

About 400 fpm is the highest velocity that an air flow measuring station would experience in typical air intake applications. At 400 fpm, the velocity pressure of air is 0.001 in./wc. Conventional pitot tube arrays are inappropriate (by the manufacturers' own recommendations) at flows under 600-700 fpm. They are sometimes used in this application, but their performance and/or the increased velocity's effect on water carryover can be easily projected.

Pitot tube arrays average several pressure points in the vector flow field, which varies as the square of the velocity.

The verifiable commercial performance of other air flow measurement technologies should be evaluated, before considering such techniques. These include air turbines, deflected air jets, and thermal anemometry.

Microprocessor-based and temperature-compensated thermal devices have been effective in measuring intake air flows, even in velocities below 100 fpm. The microprocessor-based thermal array eliminates the inherent inaccuracy present in pitot tube arrays. It also offers a much closer approximation to the "constant error as a percent of flow" assumption (presented earlier) which is inappropriate for pitot tube arrays. In thermal arrays, the microprocessor accurately calculates the velocity for each point measured instead of averaging nonlinear signals from all measured points.

By temperature-compensating the measurements from discretely calculated individual points across the duct face area, true duct-average volumes can be electronically determined in real-time. Because the flow signals are already linear and in electronic form, they can be accurately transmitted for use with all popular DDC controllers.

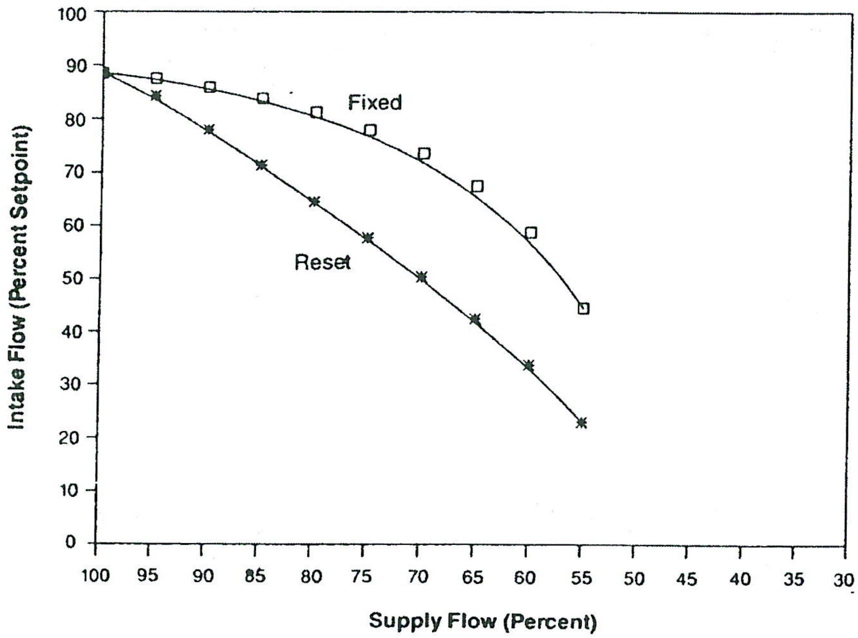


Figure 6. Stack effect, comparing intake flow and supply flow.

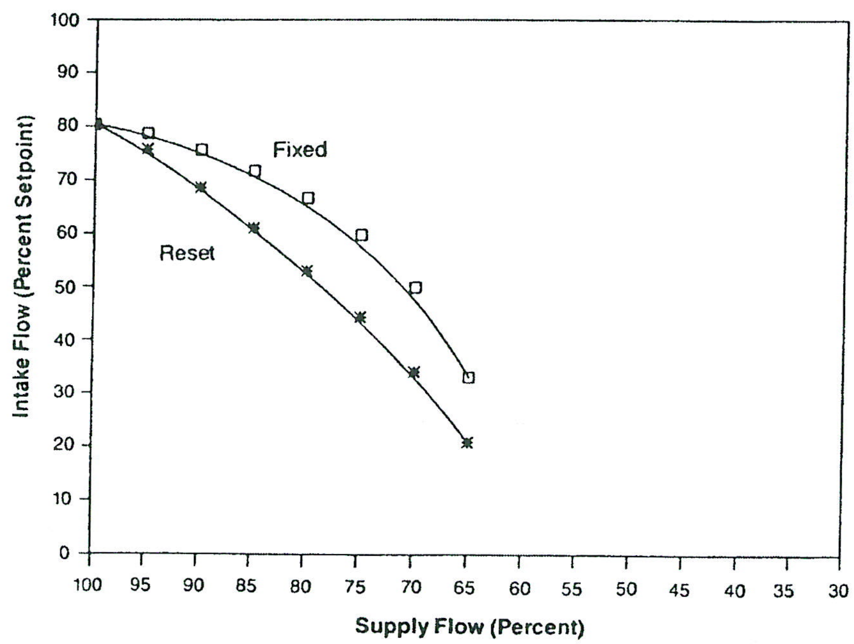


Figure 7. Wind effect, comparing intake flow and supply flow.

Conclusion

Standard 62-1989 recommends the measurement of air in both constant volume and variable air volume systems. This is due to the systems' operational pressure variations (VAV only) and due to the boundary effects of temperature and wind. Direct measurement of intake air flow should be used, because it is more accurate than "calculated" rates. Intake air measurement by microprocessor-based thermal arrays is one effective solution to meet the new indoor air quality standards of *Standard 62-1989*.

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References

Air Movement and Control Association Inc (AMCA). 1989. "Certified ratings program for louvers and dampers." *AMCA Test Standard 500-89*. Water Penetration and Air Performance.

ASHRAE. 1987. "Components and controls." *Handbook of HVAC Systems and Applications*. Controls, p 2.7.

ASHRAE. 1988. "Control of outside air and building pressurization in VAV systems." *Technical Research Project TRP-590*. From ASHRAE Technical Committee 9.1.

ASHRAE. 1989a. "Ventilation for acceptable indoor air quality." *Standard 62-1989*. June. (ISSN 1041-2336).

ASHRAE. 1989b. "Wind pressures on buildings, local wind pressure coefficients, low-rise buildings." *Handbook of Fundamentals*. pp. 14.4-14.5 (Holmes 1986).

ASHRAE. 1989c. "System component design veloc-

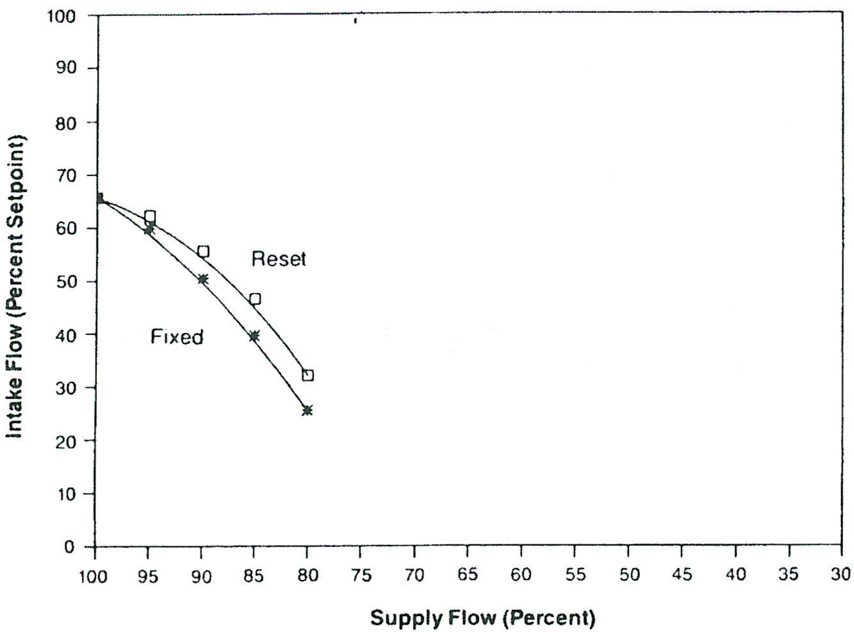

Figure 8. Stack and wind effects, comparing intake flow and supply flow.

ities." *Handbook of Fundamentals*, p. 32.15, Figure 11, Table 8.

ASHRAE. 1989d. "Local loss coefficients equation." *Handbook of Fundamentals*, p. 32.7, (34).

ASHRAE. 1989e. "Stack effect equation." *Handbook of Fundamentals*, p. 32.2, (11).

ASHRAE. 1989f. "Velocity pressure" *Handbook of Fundamentals*, p. 32.3, Equation #13.

ASHRAE. 1989g. "Bernoulli equation." *Handbook of Fundamentals*, p. 32.1, Equation #2.

Brown, Edward J., Fellows, Julian R. 1957. "Pressure losses and flow characteristics of multiple leaf dampers." ASHRAE Journal Section, *Heating, Piping & Air Conditioning*, August. [Source document for 1989 ASHRAE *Handbook of Fundamentals*, p. 32.47, #6-5 "Damper, rectangular, parallel blades." (Brown 1957) and #6-6 "Damper, rectangular, opposed blades," (Brown 1957)].

Dickey, P.S., Coplan, H.L. 1942. "A study of damper characteristics." *ASME Transactions*, February.

EBTRON Inc., 1989. Initial factory test data on louver/damper flow coefficients, based on 400 fpm louver face velocity.

Johnson Controls. *Engineering Data Book*, Section Db.5, Dampers, 0687, Outside Air Dampers. "Reducing the outside air damper with blank-off plates is not permitted in a close coupled arrangement since the velocity will increase in certain areas of the weather louver and snow and rain will be drawn into the outside air intake. The outside damper must, by necessity, be sized to match the louver size."

Persily, A.K., Grot, R.A. 1985. "Ventilation measurements in large office buildings." *ASHRAE Transactions*, Vol. 91, Pt. 2.