

Controlling Stack Effect in Ventilation Duct Risers Promotes Energy Efficiency and IAQ

Central ventilation and toilet exhaust risers are designed for the purpose of providing mechanically controlled ventilation, and to protect against poor IAQ. In high rise buildings (buildings over four stories); exhaust ventilation risers and subsequent fans are often dramatically affected by environmental factors such as stack pressure. In today's energy efficient construction, stack pressure has a greater effect on a system's ability to regulate indoor air quality, and actually detracts from a building's energy efficiency.

The central duct riser used for air exhaust and/or ventilation air distribution in tall buildings, is a main focus of building designers and engineers who look to improve on both energy efficiency and indoor air quality. Maintaining proper airflow rates in duct risers is the key for both indoor air quality and energy efficiency assurance. However, difficulty in system balancing, poor maintenance practices, and fluctuations in system pressure due to stack effect makes this very problematic, especially when coupled with the need to minimize energy consumption.

One challenge designers face is how to minimize the effect stack pressure has on a particular system, while minimizing fan motor power consumption. To combat seasonal fluctuations in system pressure, designers can either increase fan induced duct pressure or find a means to modulate the opening at each intake point. In the absence of either solution, these seasonal pressure variations will result in over or under ventilation, increased thermal load on the building, and fluctuations in sound levels at the intake points. The following discusses how stack pressure is determined, its effect on vertical riser system performance, and what can be done to overcome this ever-present condition.

Energy Efficiency and IAQ vs. Stack Effect

The difficulty in maintaining proper ventilation system riser airflow balance in areas with large shifts in climatic conditions is mainly due to stack effect. Stack, or hydrostatic pressure, is created when differences in air temperature, altitude, and vertical distribution of air from indoor and outdoor conditions exist. As discussed in ASHRAE Fundamentals Chapter 26, "stack pressure differences are positive when the

building is pressurized relative to outdoors, which causes flow out of the building. Therefore, in the absence of other driving forces, when the indoor air is warmer than outdoors, the base of the building is depressurized and the top is pressurized relative to outdoors; when the indoor air is cooler than outdoors, the reverse is true."

Stack effect is unavoidable, and increases as the building height increases, and as temperature differences between inside and outside air also increase. The level of stack pressure within vertical chases in a compartmentalized building (multiple floors) is also affected by the level of air tightness between floors, and with the exterior walls. Therefore, the taller the building, the larger the difference in indoor and outdoor temperatures, and energy efficient construction, all contribute to wider differences in pressures found within elevator shafts, stairwells, and exhaust risers.

Determining Stack Pressure

Since vertical duct risers penetrate the floors of compartmentalized buildings and provide an open vertical chase throughout the length of the duct itself, usually the height of the building, stack pressure within these ducts can be easily calculated using the following formula:

$$\Delta P_s = C_1 \cdot g \cdot \rho \cdot (T_1 - T_0) / T_1 \cdot H$$

Where

P _s	=	stack pressure, in. of water
C ₁	=	unit conversion factor = 0.00598 (in. of water) x ft x s ² /lbm
g	=	gravitational constant, 32.2 ft/s ²
ρ	=	indoor or outdoor air density
T ₀	=	outdoor air temperature (R)
T ₁	=	indoor air temperature (R)
H	=	Height (ft.)

A simple rule of thumb can be derived from the same formula as follows:

$$\Delta P_s = .0000274 \text{ IN W.G. per ft} \times (TF_1 - TF_0)$$

Example: A 200 ft building in Chicago, 0° F winter design condition, 70° F indoor temp

$$\begin{aligned} \Delta P_s &= .0000274 \times 200 (70^\circ \text{ F} - 0^\circ \text{ F}) \\ \Delta P_s &= .00548 (70) \\ \Delta P_s &= 0.38 \text{ IN W.G.} \end{aligned}$$

If the duct riser extends only partially throughout the building height, apply the same formula for only the length of the duct riser. To determine the stack pressure at each intake point, also apply the same formula for the length of each point to the fan. This assumes that the fan at the top of the riser is capable of handling the increase in pressure and resulting increase in flow, and the neutral pressure point is at or above the fan itself.

If the increase in stack pressure and flow result in conditions beyond a fan's performance capability, the neutral pressure point would be lower than the fan at a point within the duct riser itself. This would actually result in positive pressure, or outflow of air above the neutral pressure point at the top of the riser, and completely eliminate the ventilation performance in those areas. This is often identified as a cause of poor indoor air quality.

Changes in Airflow Caused by Stack Pressure

The increase in pressure from stack effect results in increased flow through the duct riser. The flow at each floors' intake points varies as a square root of the difference in pressure through the opening. Assuming that the fan can effectively remove this increase in flow, the percentage of change in flow at each intake point is taken as follows:

$$Q_1 = Q_0 \sqrt{\frac{\Delta P_0 + \Delta P_1}{\Delta P_0}}$$

Where

Q 0	= Flow at design ΔP_0 in the absence of stack pressure
Q 1	= New Flow under stack pressure conditions
ΔP_0	= Pressure in the absence of stack
ΔP_1	= Pressure including stack

Example: Flow at an exhaust grille located on the first floor is 100 cfm, ΔP_0 at .10 in w.g., the increase in flow as a result of increase in stack pressure:

$$Q_1 = 100 \sqrt{\frac{.10 + .38}{.10}} = 100 \sqrt{4.8} = 100 \times 2.2 = 220 \text{ cfm}$$

Based on our original example of a 200 ft. building, and assuming design of 100 cfm per floor with a total of 14 floors, the total system airflow would increase

from 1400 to 2286 cfm. This represents an increase of 63.2% in total flow, or 886 cfm of unwanted ventilation and additional load on the building!

Balancing Airflows in the Presence of Stack Pressure

Ultimately, the effect stack pressure has within a building relates to the amount of unwanted infiltration of unconditioned air and/or exfiltration of conditioned air. This unwanted movement of air relates to increased thermal load on the building and uncontrolled energy consumption. Since the mechanical ventilation riser is a contributor to overall building pressure buoyancy, not to mention proper regulation of ventilation for IAQ, it is important to recognize that proper balancing and regulation of these systems has a significant effect on energy consumption.

One technique to minimize the effect stack pressure has on exhaust ventilation system prescribed airflows is by increasing the internal duct pressure created by the fan. The greater the internal duct pressure, the less effect stack pressure can have on the system. However, increased pressure also relates to increased fan motor BHP and relative energy consumption in watts. To determine the increase in pressure necessary to overcome stack pressure within a tolerance of +/-10% in a balanced static system, the following formula can be applied:

$$\frac{Q_1}{Q_0} = 1.1 = \sqrt{\frac{\Delta P_0 + \Delta P_1}{\Delta P_0}}$$

Squaring both sides to solve

$$\begin{aligned} \text{for } \Delta P_0: \quad 1.21 &= \frac{\Delta P_0 + \Delta P_1}{\Delta P_0} \\ \Delta P_0 &= \frac{\Delta P_1}{0.21} = 4.76 \end{aligned}$$

(times the increase in stack effect pressure),

Where Tolerance factor

$$Q_1 = Q_0 \pm 10\%$$

Simply stated, the pressure drop at each grille for static balancing must be 4.76 times the anticipated stack effect at each respective intake point to maintain the airflow within 10% of design values. This is true for all the grilles regardless of elevation within the building. In the absence of stack effect, the formula does not apply. When applying the increase in pressure factor of 4.76 to our example, and given the original stack pressure of .38 at the first floor grille, the fan must now operate at a level to ensure 1.81 Ps in. w.g. at this same grille. The increase in necessary pressure will not only result in excessive energy consumption, but excessive noise generated at each grille as well.

Motor/Fan Performance and Energy Penalty to Overcome Stack Pressure

After solving for the increase in pressure necessary to maintain balanced flows, simple fan laws can be applied to determine the required increase in fan rpm. Fan laws show that pressure is proportional to the square of the rpm.

$$SP_1 / SP_2 = (RPM_1 / RPM_2)^2$$

Therefore using the previous example, the increase in rpm can be determined as follows. Assuming the original fan rpm is 1000 and pressure to achieve design airflows in the absence of stack pressure is 0.22, derived from 0.10 at each grille and 0.12 to account for duct loss:

$$RPM_2 = RPM_1 \sqrt{\frac{SP_2}{SP_1}} = 1000 \sqrt{\frac{1.91}{0.22}} = 2950rpm$$

The result is an increase of almost three times the original rpm design in order to prevent changes in airflow due to stack pressure effect. When applying this increase to energy consumption of fan motors, the increase varies with the cube of the rpm.

$$HP_1 / HP_2 = (RPM_1 / RPM_2)^3$$

Following the previous example:

$$HP_2 / HP_1 = (2950/1000)^3$$

$$HP_2 / HP_1 = 25.7$$

Therefore, the final result is a 25X+ increase in power consumption to operate a fan at the higher pressure required to ensure proper system balance in the presence of stack effect.

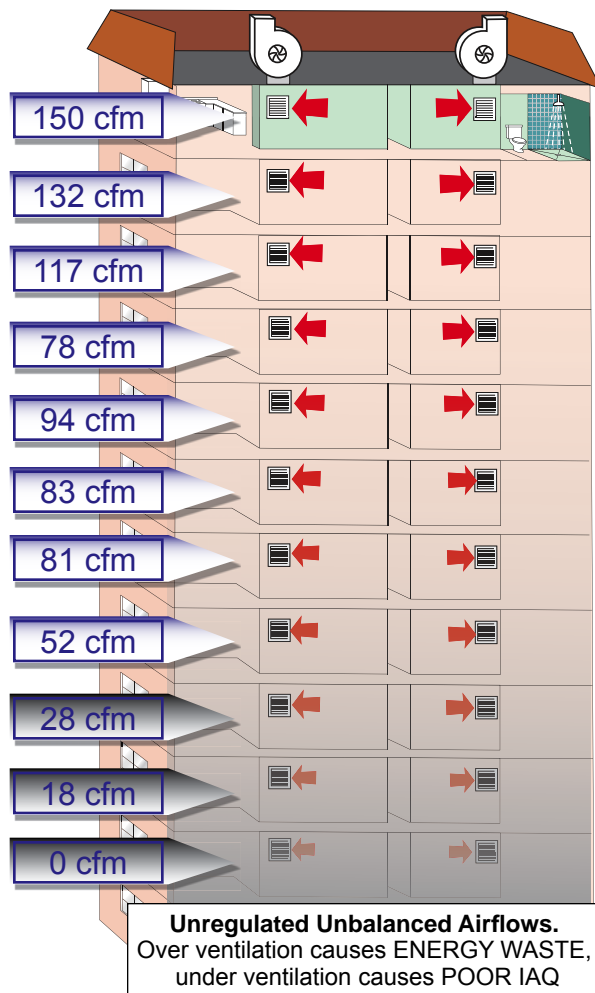
Analysis of Balancing and Control of Static Riser Systems

Through analysis of traditional central duct riser system designs, and factors that effect overall airflow performance, it is determined that excessive energy consumption will increase as stack pressure increases. Since statically controlled systems have no means of adjusting to fluctuations in stack pressure, the amount of excessive energy consumed will either come in the form of additional thermal load on the building and resulting increase in heating costs, or from increased fan power to control the flows at higher pressure.

The other negative factors associated with statically controlled risers are excessive noise and duct leakage created by high fan pressures, or the potential for under ventilating portions of the building. Either scenario can result in an unsuitable environment for the buildings occupants.

The only solution to dealing with stack pressure effect on vertical risers is to monitor the pressure at each intake point into the riser and modulate the opening to regulate flow in response to these changes. This will allow the use of lower pressure fans for energy

savings, and prevent stack pressure from effecting flow rates and resulting over and under ventilation. Unfortunately, most modulating dampers in the market today are designed using pitot tube pressure sensing devices and electric drive motors and controllers to actuate a damper for flow control. Using one of these devices on every intake point in a riser is often more costly than years of energy penalties on systems without them.



The Physical Challenge of Test and Balance

Balancing and commissioning of a ventilation riser is usually considered difficult and tedious. Low airflows through small, often inaccessible, sidewall mounted registers located on multiple floors, is challenging to any test and balance contractor. It requires special instrumentation and many man hours for typical riser systems. Even with modulating duct openings, the balancer's job is compounded by fine-tuning controllers to specified set points before and after airflow measurements.

Besides the physical constraints of balancing vertical risers, the time of year and stage of construction dramatically affect the measurement readings the contractor will record. This goes back to stack pressure effects on the system.

The American ALDES - CAR Constant Air Regulator Solution

The ultimate solution to insure proper system balancing and airflow regulation is the CAR constant airflow regulator. The model CAR is a factory calibrated passive airflow regulator that eliminates the need for balancing airflows at the grilles. It does not require any external power since it automatically adjusts to the proper airflows in response to duct inlet pressure.

The CAR operates by sensing pressure through metering orifices in the face of the device, and across a modulating element located in the throat of the mechanism. As pressure increases, the regulator senses this change and automatically modulates the element to control the free area through the duct opening.

Best of all, the CAR will maintain the prescribed airflows as it adjusts to changes in pressure caused by stack effect. This eliminates over ventilation caused by the exhaust riser, saving energy. The use of CARs also allows fan operation at the lowest pressure level possible without sacrificing airflow performance, which saves fan energy consumption. Finally, they eliminate under ventilation caused by imbalances of the exhaust system, protecting against poor indoor air quality.

The CAR constant airflow regulators are employed in thousands of buildings in the U.S. and around the world. This well proven technology has existed for over 20 years, and was developed to minimize fan energy use in the late 1970's. Today, they serve as a simple solution to indoor air quality ventilation regulation and energy savings. The CAR by American ALDES continues to lead the industry in economical passive airflow control regulation. Consult the factory, or an American ALDES certified representative to discuss how CAR's can save money, energy, and protect any building against poor ventilation control.

TYPICAL SPECIFICATIONS

Model CAR Constant Airflow Regulators by American ALDES Ventilation Corporation, Bradenton, Florida, shall solely operate on duct pressure and require no external power supply. Each regulator shall be pre-set and factory calibrated requiring no field adjustment to the airflows as indicated on the schedule, and shall be rated for use in air temperatures ranging from -25 to 140°F (-32 to 60°C).

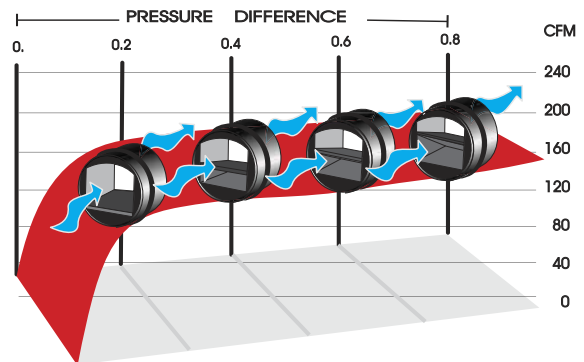
Constant airflow regulators shall be capable of maintaining constant airflow within +/-10% of scheduled flow rates (15% for units 50 cfm or less), within the operating range of 0.2 to 0.8 in. w.g. differential pressure, or 0.6 to 2.4 in. w.g. on high-pressure models. Sound power levels shall not exceed those for each size and cfm rating as scheduled.

Regulators shall be provided as an assembly consisting of a flame resistant plastic body with self-inflating silicon element housed within a .75mm galvanized steel sleeve or flanged plate for mounting in either round or rectangular duct. Each round sleeve must be fitted with a brush gasket to assure perimeter air tightness with the interior surface of the duct.

All constant airflow regulators will require no maintenance and must be warranted for a period of no less than five years. Constant airflow regulators shall be installed in tight ducting systems in accordance with all applicable codes and manufacturer's instructions.

HOW THE CAR-II WORKS

Constant airflow is achieved by controlling the free area through the device. At minimum static pressure, the aero-wing is parallel to the air stream. As the static pressure increases, the aero-wing lifts, thereby reducing the amount of free area through the regulator. At the same time, the higher static pressure increases the air velocity resulting in CONSTANT AIRFLOW. This occurs regardless of pressure differences in the range of 0.2 to 0.8 in. w.g. (50 to 200 Pa). The air velocity in the duct is in the range of 60 to 700 ft/min. (0.3 to 3.5 m/s).



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