Consulting-Specifying Engineer

Air-Handling Design: A Balancing Act

Engineers must consider testing, adjusting and balancing requirements when designing air-handling systems

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Air-handling-system testing, adjusting and balancing (TAB) is essential for ensuring systems meet designer specifications. And TAB results must be repeatable to ensure systems are both functional and reliable, according to the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Fundamentals Handbook. Unfortunately, the TAB procedure often degenerates into a process in which engineers, mechanical contractors, fan manufacturers and TAB contractors blame each other for system deficiencies.

Engineers must consider TAB procedures when designing systems. One of the biggest challenges specifiers face stems from the difficulty of measuring airflow in main ductwork and matching fans to system curves. Understanding several concepts helps designers specify air-handling systems capable of being properly tested, adjusted and balanced.

Fluid-flow equations

In order to design a good air system, engineers must be familiar with a couple of fundamental principles of fluid mechanics.

The law of conservation of mass provides a simple equation to measure the volumetric flow rate of air in a duct. The law states that mass within a system remains constant with time.

Consider the duct shown in Figure 1. The conservation of mass indicates that the mass flow rate ($\dot{m}$) at point 1 must equal the mass flow rate at point 2. Mass flow rate is the product of air density (ρ) times air velocity (V) times duct area (A). Since the mass flow rate at point 1 equals the mass flow rate at point 2:

$$\dot{m}_1 = \dot{m}_2$$

and

$$\rho_1 V_1 A_1 = \rho_2 V_2 A_2$$

If the airflow is assumed to be incompressible:

$$\rho_1 = \rho_2$$

and volumetric flow rate (Q) can be calculated as (Equation 1):

$$Q = V_1 A_1 = V_2 A_2$$

Equation 1 is known as the continuity equation. It provides a means of deriving air velocity at any point in the duct system if velocity is known at any other point.

For example, in Figure 1, if V_1=1,000 feet per minute (fpm), A_1=2 square feet, A_2=1 square foot, then:

$$V_2 = \frac{Q}{A_2} = \frac{(1{,}000\ fpm)(2\ square\ feet)}{1\ square\ foot} = 2{,}000\ fpm$$

The continuity equation also predicts volumetric flow rate, measured in cubic feet per minute (cfm), will be constant in a duct system despite changes in its cross-sectional area.

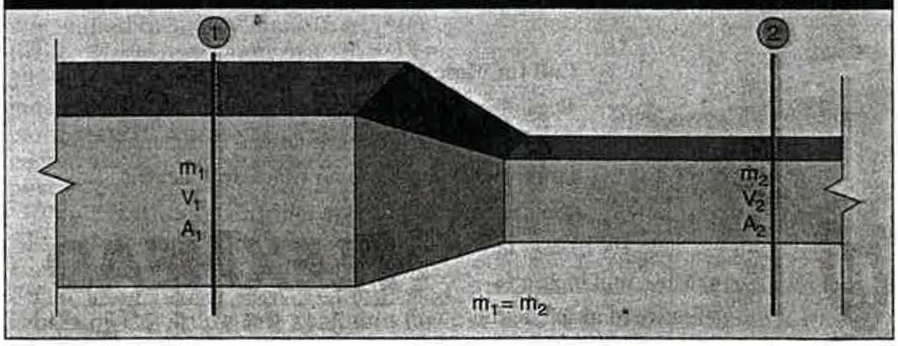
A second useful equation (Equation 2) describes the relationship between total pressure (TP), velocity pressure (VP) and static pressure (SP):

$$TP = VP + SP$$

Static pressure, or potential energy of the air, is exerted perpendicularly on the duct. Velocity pressure, or the air's kinetic energy, is exerted in the direction of airflow.

The first law of thermodynamics provides another important equation, proving that energy is conserved in every process and is nei-

FIGURE 1—**CONSERVATION OF MASS**



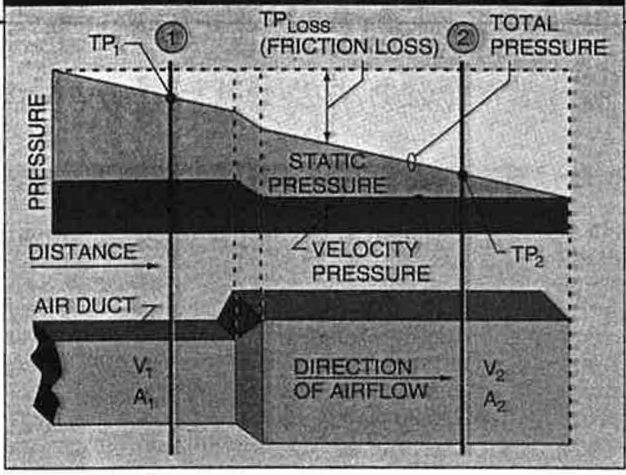
Although the above duct changes size, the law of conservation of mass indicates that mass flow rate will remain consistent.

ther created nor destroyed. Applying this dictum to the ductwork shown in Figure 2 reveals the following relationship (Equation 3):

$$TP_2 = TP_1 + (TP_{Loss})_{1-2}$$

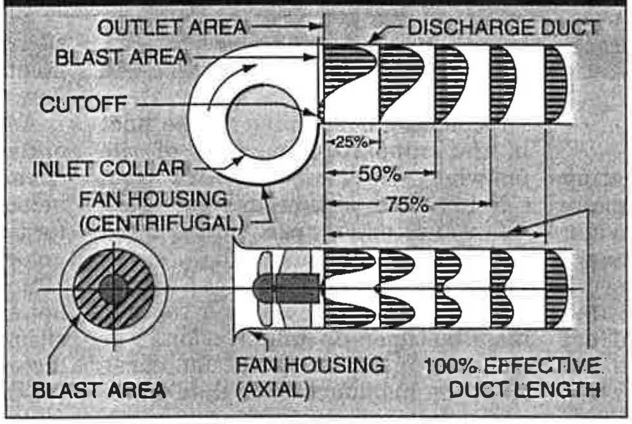
where TP_1 equals total pressure at point 1, TP_2 equals total pressure at point 2 and $(TP_{Loss})_{1-2}$ equals total pressure drop from point 1 to point 2.

FIGURE 2—DUCTWORK PRESSURE RELATIONSHIPS



Although velocity pressure remains constant, duct friction causes total pressure to drop over distance.

FIGURE 3—FAN AIR-VELOCITY PROFILES



Fan construction has a strong effect on velocity profiles.

Pressure loss between points 1 and 2 is caused by friction and dynamic losses. Frictional losses occur throughout the duct system and, according to the ASHRAE Fundamentals Handbook, are "due to fluid viscosity and are a result of momentum exchange between molecules in laminar flow and between particles moving at different velocities in turbulent flow." These losses can be derived from the Darcy-Weisbach equation or estimated from tables contained in the ASHRAE Fundamentals Handbook.

Dynamic losses result from flow disturbances caused by fittings that change the airflow path's direction or area, according to the ASHRAE Fundamentals Handbook. This publication provides tables and equations to estimate the contribution of dynamic losses in a duct system.

For this article, Equations 2 and 3 indicate that total pressure in a duct decreases in the direction of airflow, while static and velocity pressure can rise and fall. The energy level in the duct is given by the total pressure, and losses in the system are based on losses in total pressure.

These concepts and equations provide the foundation for several fundamental ideas that must be incorporated when designing air systems that can be properly balanced.

Uniform velocity profile

The continuity equation indicates that a system's air volume can be measured if its velocity is known. However, an accurate measurement is possible only if a smooth velocity profile exists within the air stream. If the velocity profile is irregular, more measurements are required and the accuracy will be reduced.

A number of air-system components can upset velocity-profile uniformity, including branch takeoffs, elbows, fittings and transitions that redirect airflow. When air flows around an elbow, air on the outside radius is deflected around the turn, while air on the inside radius follows a straight path and bumps into the air on the outer path. This phenomena causes air to swirl, distorting the profile. Consequently, accurate flow readings cannot be made unless the measuring device (usually a pitot tube) is located several duct diameters downstream and upstream from elbows and other transition pieces.

Air measurement on the discharge of fans also must take air-velocity profile changes into account. Figure 3 represents velocity profiles of a centrifugal and axial fan respectively. The velocity pressure at the fan's discharge is nonuniform; it takes several feet for the air to acquire a smooth profile. The ASHRAE Fundamentals Handbook provides several formulas to calculate the distance from the fan where the velocity profile will recover to 100 percent.

It is also advisable to take several readings in a duct to get a more accurate reading of the airflow. ASHRAE Standard 111 and the ASHRAE Fundamentals Handbook recommend traverse-point locations for accurate airflow measurements in circular and rectangular ductwork.

Fan selection

The concept of total, static and velocity pressures often are misapplied when fans are selected, and "system-effect" factors often are ignored. These issues can lead to several arguments in the field between designers, contractors and fan manufacturers.

Fans are tested according to American National Standards Institute (ANSI)/ASHRAE Standard 51 and ANSI/American Mechanical Contractors Association Standard 210, "Laboratory Methods of Testing Fans for Ratings." Sometimes designers specify fans based on "total static pressure." This practice leads to confusion because no official publications recognizes the term. The only term that matters when selecting a fan is total pressure.

System designers may confuse ductwork static pressure with fan static pressure. Duct and fan static pressures are not related. As Equation 2 indicates, total pressure in the ductwork is the sum of velocity and static pressure. Since the continuity equation indicates that air velocity is inversely proportional to the duct's cross-sectional area, velocity pressure increases and decreases based on ductwork size. Also, because conservation of energy dictates that total pressure remains the same (neglecting friction and dynamic losses), static pressure will increase

Nonuniform airflow into the inlet can reduce fan capacity

and decrease to offset velocity-pressure changes.

Thus, ductwork static- and velocity-pressure changes largely result from the duct's size. The correct method to size fans is to calculate the sum of the total pressure drop through the supply duct and add this total to the total pressure drop through the return-air ductwork.

Changes in the air's velocity and static pressure as it goes through the supply and return ductwork, is irrelevant from the viewpoint of the fan and its rating. Confusion can result when total pressure drop due to frictional and dynamic losses occurs in a duct with a constant cross-sectional area. In this case, total pressure drop equals loss in static pressure, because velocity pressure remains the same.

Obtaining Accurate Measurements

Pitot tubes—the most common devices for measuring airflow in a duct—become less accurate at lower velocities and can become unreliable below 800 feet per minute. Designers should recognize the limitations of these devices in low-pressure ductwork, in which air velocity can approach these levels.

Similarly, designers must be careful when sizing variable air-volume (VAV) boxes because controlling them is difficult if inlet pressure drops below a minimum value. Most VAV boxes are pressure-independent, with thermostats regulating their actuators to maintain required zone temperatures. Also, dampers are regulated as inlet pressure fluctuates because of system-pressure changes. Boxes cannot meet minimum air-supply requirements if their controllers cannot make accurate pressure readings.

System-effect factors

Misunderstanding or neglecting system-effect factors has led to many arguments in the field between designers, contractors and fan manufacturers. System-effect factors encountered at the fan's inlet and discharge can dramatically affect fan performance. To compound the problem, system-effect factors are difficult to measure in the field. However, ASHRAE does supply some guidelines on how to incorporate system-effect factors for many common types of duct fittings.

Several factors can reduce fan capacity, including:

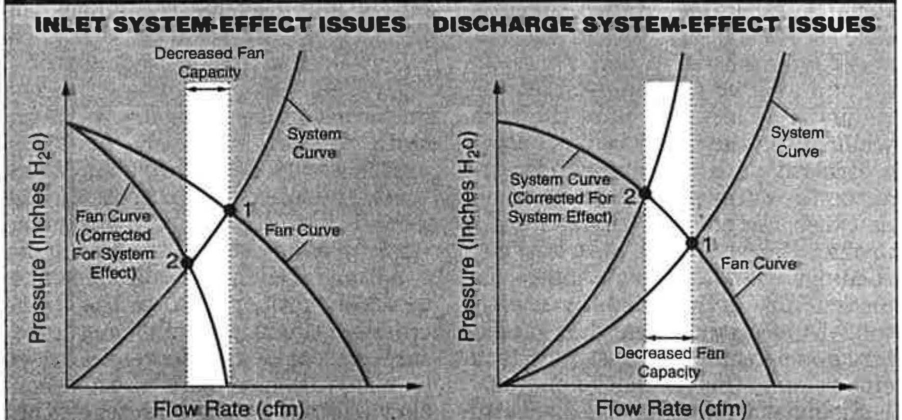
- Incoming air spinning at the inlet in the opposite direction of the fan wheel.
- Nonuniform airflow into the inlet.
- A restricted air space between the fan inlet and the duct wall.

These factors cause fan-curve characteristics to differ from those in manufacturer documentation. For example, in Figure 4, the fan curve from the manufacturer's data intersects the system curve at point 1. System-effect factors at the fan's inlet derate fan performance, creating a new fan curve. The actual intersection of fan and system curves is at point 2, resulting in lower volumetric flow rates delivered by the fan.

As mentioned earlier, the ideal fan outlet should extend for several duct diameters to allow for a smooth velocity profile to develop, and allow for the conversion of fan energy from velocity pressure to static pressure. Unfortunately, the space often does not exist to incorporate smooth transitional ductwork, so system-effect factors must be taken into account in fan-discharge specifications.

System-effect factors present on the fan discharge do not affect published theoretical fan curves, but system curves will shift upward. As indicated in Figure 6, the theoretical point of operation will be at point 1. If system-effect factors are present, the system curve moves up and the actual point of fan operation will be at point 2.

FIGURE 4–POTENTIAL SYSTEM EFFECTS ON FAN CAPACITY



System effects on both inlet (left) and discharge (right) sides need to be considered when specifying ductwork to ensure accurate testing measurements can be taken.

Designing for TAB success

System designers play an important part in the success of TAB efforts. First, volumetric flow-rate measurement can be improved if designers incorporate space in the ductwork where a smooth velocity profile can develop. Second, designers must remember that fans should be sized based on total pressure, not static pressure. And, finally, designers must consider system-effect factors that can distort results delivered by TAB contractors and imply that fans are not meeting their published ratings. □