Achieving optimal fan performance

Improper fan design can waste energy, produce excessive noise, and lead to increased system downtime. Design correctly–the first time around.

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Unusually high operating costs for commercial and industrial air handling systems are often caused by inefficient fan operation that, in turn, can be due to improper fan selection or poor system design. Improper fan selection can result in a fan that is too large or too small, or a fan that is running at a higher speed than necessary for the application.

The consequence of this is wasted energy consumption and expense, excessive airflow noise, and increased maintenance and system downtime. Flawed system design, such as improper fan inlet or outlet airflow conditions and sharp-cornered turns in duct systems, also cause these same problems. Fortunately, these problems can be avoided by taking advantage of the following guidance on fan and fan inlet and outlet design, which is based on decades of laboratory testing and field experience.

Simple fan selection process

The fan and air system market is mature, which means that there is a wide variety of fans to fill market niches that have specific performance requirements. For example, centrifugal fans with narrow blades operating at high speeds are suited for systems requiring low-volume flow rate and high pressure. Propeller fans are generally used to move air against low pressures from one open space to another. Material handling systems often use fans with radial bladed impellers. With this rudimentary knowledge of the fan selection process, the range of available fan sizes and models can be significantly narrowed to a much smaller range that best fits the system's requirements. A system designer can then select a fan specifically suited for the job, which achieves a balance of maximizing efficiency and minimizing cost.

Fan nomenclature is important because subtle differences in terminology can have big impacts on calculations. For example, fan static pressure and static pressure are often used interchangeably, but they're not the same thing. The terminology used in this article is based on ANSI/AMCA Standard 210-07 ANSI/ASHRAE 51-07: Laboratory Methods of Testing Fans for Certified Aerodynamic Performance Rating.

Typical selection parameters

For a simple system, four basic parameters are required to select a fan. These parameters are:

- Installation Type
- Air density at the fan inlet, ρ_f
- Desired airflow rate, Q_f
- Pressure required to move the airflow rate desired, P_{sf} or P_{tf}

With these parameters in hand, a fan manufacturer should be able to provide several selections of fan types and sizes; each type and size will have a unique operating speed and power requirement that meets the performance required. Given fans of equal maintenance requirements, the best fan selection becomes an economic calculation that considers lifecycle costs, that is, the best selection will have the lowest sum of initial cost plus future power costs, discounted to today.

Installation Type

There are four basic types of inlet and outlet ducting arrangements for fans. In fan terminology, these arrangements are called Installation Types (A, B, C, and D) as defined by Air Movement and Control Assn. International (AMCA).

An important point to remember is that ducting arrangements affect the performance of the fan. Fan manufacturers understand this, so when fans are tested for rating, they are tested in an Installation Type that is commonly used for the fan in question. Installation Type is among the parameters commonly found in fan catalogs and sizing/selection software.

Density, peanuts, and stones

Fans are constant volume machines. That is, regardless of the density of the gas the fan is handling, flow rate remains constant. Pressure and power requirements, however, change proportionally with the density of the gas (typically air) being moved. An analogy would be a person shoveling either peanuts or stones. Regardless of the material, the shovel holds the same volume (flow rate), but stones weigh much more than peanuts (pressure), and they take much more effort (power) to move.

Density is specified at the fan inlet and is usually expressed as one of the following symbols:

$\rho_1 = \rho_f$

Electronic fan selection programs are changing the fan catalog landscape, but at one time, nearly all fan catalogs contained fan performance “corrected” to standard air. This allows easier comparison of fan performance among models, sizes, and manufacturers.

Flow rate

Most frequently, a simple volume flow rate is provided as a specification.

$Q_1 = Q_f$

Sometimes, instead of volume flow rate, mass flow rate will be specified. Fan airflow rate is the mass flow rate divided by density at the fan inlet.

$Q_f = m / \rho_f$

In some cases, standard airflow rate (Q_s) is specified, but this is just another way of specifying mass flow rate. If ρ_s is standard air density (0.075 lbm/ft^3), then:

$Q_f = Q_s \, \rho_s / \rho_f$

Fan pressure

As stated above, one of two types of fan pressure is typically specified when selecting a fan. Fan static pressure:

P_{sf}

Or, fan total pressure:

P_{tf}

Fan total pressure is the difference between the total pressures at the fan inlet and outlet.

$P_{tf} = P_{t2} - P_{t1}$

Fan static pressure is:

$P_{sf} = P_{tf} - P_{vf}$

Where P_{vf} is fan velocity pressure. But this is *not* the standard equation most engineers are familiar with ($P_s = P_t - P_v$) because fan velocity pressure is the velocity pressure based on the average velocity at the fan's outlet, or

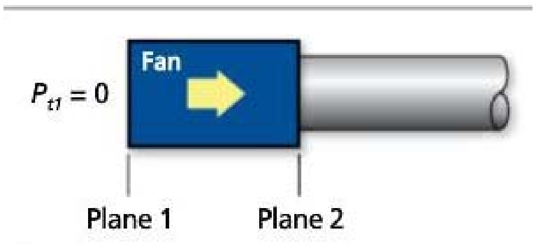
$P_{vf} = P_{v2}$

Which means that fan static pressure is the fan total pressure minus the velocity pressure at the fan's outlet, or

$P_{sf} = P_{t2} - P_{t1} - P_{v2}$

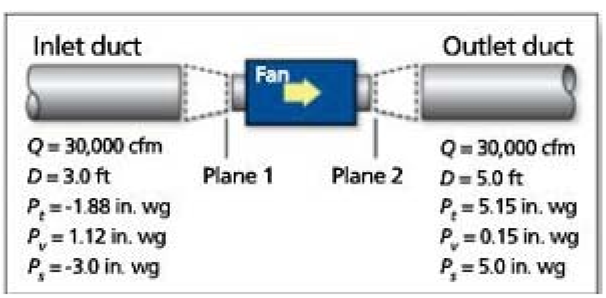
In Installation Type B, where the system duct size matches the fan outlet (see Figure 1), the inlet is open to atmosphere and we have the following calculation of fan static pressure:

$P_{sf} = P_{t2} - P_{v2} = P_{s2}$



Fan static pressure is simply the losses in the system downstream of the fan, including the system's exit loss to atmosphere. This relationship may have led to the common practice of specifying fan static pressure instead of fan total pressure for fans installed in Installation Type B. Note that architectural louver ratings, when tested in accordance with AMCA 500-L, include exit losses from the building.

Let's now consider a case where the inlet and outlet duct diameters are known and they don't necessarily match the fan's inlet or outlet diameters, as shown in Installation Type D in Figure 2.



Note that at this point in the design process it's impossible to know whether the inlet or outlet duct areas match the fan's inlet or outlet, because in this hypothetical situation we are still in the process of selecting the fan. Because the duct must transition to the fan, the static pressures will change over the distance of the duct to the fan. This is a result of the velocity and velocity pressure changing due to the increasing or decreasing area of the transition.

What will not change, neglecting the skin friction of the transition, is total pressure. From this we can determine the required fan total pressure.

$P_{tf} = P_{t2} - P_{t1}$ = 5.15 in. wg - (-1.88 in. wg) = 7.03 in. wg

So what is the required fan static pressure for our example? We won't know until the fan is selected and we know the fan's outlet area. Then:

P_{sf} = 7.03 in. wg - P_{v2}

Fan selection

In the AMCA Certified Ratings Program, a great deal of effort is put into standardizing how fan data are presented to the customer. The data provided allow customers to find the most efficient fan for their system. Therefore, it is the customer's responsibility to understand that data and how it applies to their system. The following data are required by the AMCA Certified Ratings Program:

Installation Type: The manufacturer will state which Installation Type was modeled when the fan was tested. If the fan is installed in an Installation Type that does not match how the fan was tested, the resulting change in performance is a type of system effect (described in the next section of this article).

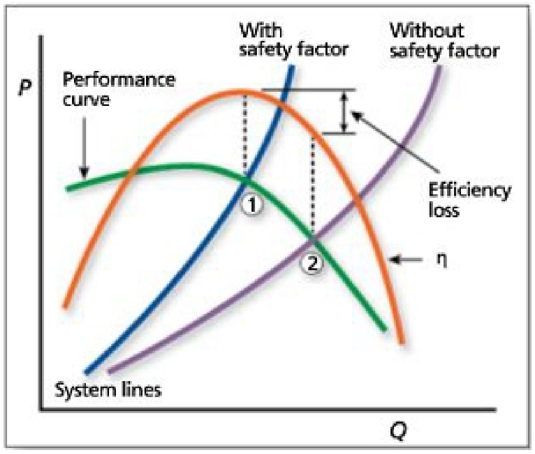
Pressure: For fans installed in ducted systems, either P_s or P_t may be cataloged. For nonducted fans, Ps is usually cataloged.

Accessories: Most fans are rated without including the effect of accessories. If an accessory is in the airstream, the effect of the accessory on fan performance needs to be taken into account. The fan manufacturer should be able to provide these data. Accessories that can affect fan performance are belt guards (if near the fan inlet) and backdraft dampers.

Drive or transmission losses: Depending on the fan's arrangement, the efficiency impacts of pulleys, sheaves, and other components may or may not be included in the catalog. When not included, and when necessitated by the fan's arrangement, these losses need to be added to the cataloged power.

Safety factor

A [U.S. Dept. of Energy]() study, Improving System Performance, from 1997 showed that almost 60% of fans in building systems were oversized. A major contributing factor to this phenomenon is the addition of safety factors. System designers sometimes compensate for uncertainties in the system design by adding a safety factor to the required performance of the fan.



If a safety factor is included in the fan selection, it means that the fan will not operate at its best efficiency point (see Point 1 in Figure 3) but will instead operate at Point 2. It is possible for Point 2 to generate more noise and, for some types of fans, be an unstable operating point.

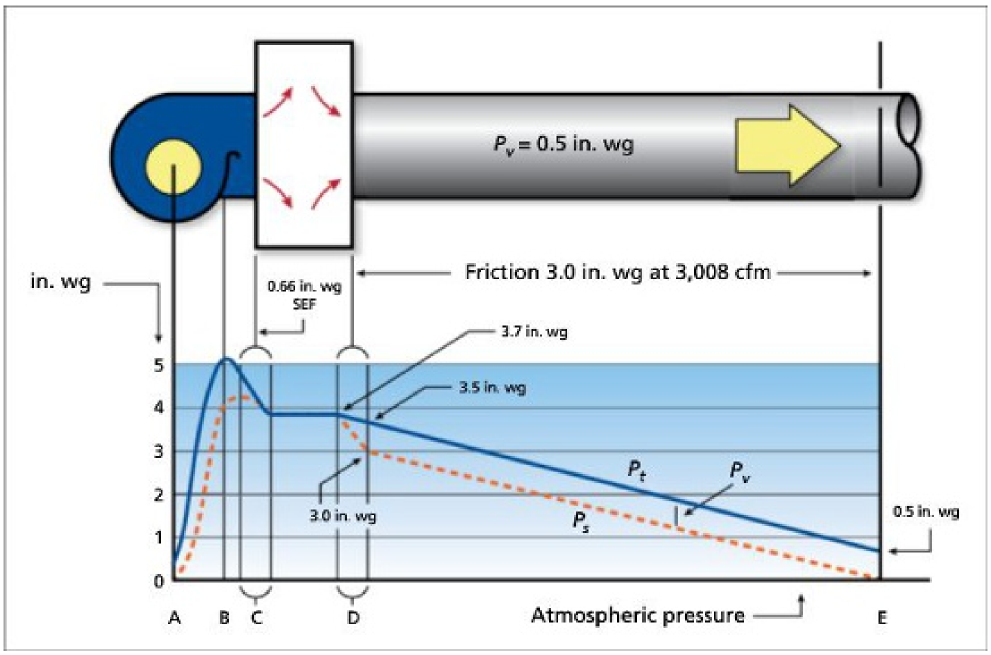
The use of safety factors also leads to oversizing fans to overcome the assumed higher pressure requirements, which results in increased capital costs and subsequent costs of adjusting down the fan performance to match the system.

System effect

The above overview of fan selection results in an ideal fan size and speed for ideal conditions. But in the field, actual conditions are often less than ideal. Obstacles to duct runs lead to sharp turns or changes in elevation, and then another correction to resume the planned path. Or, there might not be enough room for the ideal length of inlet or outlet duct to establish fully developed airflow. The results of less-than-ideal fan conditions are summarily called system effect.

In order for a fan to achieve its rated performance, that is, the stated performance from the manufacturer used to select the fan, the airflow at the inlet must be fully developed, symmetrical, and free from swirl. Ducting on the outlet needs to be designed such that the asymmetrical flow profile from the fan is allowed to diffuse and approach fully developed flow. The effect on system performance when these conditions are not met is called system effect.

On the fan's inlet side, AMCA Publication 201 recommends that elbows near the fan's inlet be located at least three duct diameters upstream of the fan, while acknowledging that elbows can cause system effect when they are located up to five diameters upstream.



On the fan's outlet side, AMCA Publication 201 introduces the term "effective duct length." Effective duct length is 2.5 duct diameters when duct velocities are 2500 fpm or less, with one duct diameter added for each additional 1000 fpm. A centrifugal fan needs full effective duct length on its outlet to avoid system effect, while a vaneaxial fan needs 50% effective duct length. Fully developed flow is not needed to avoid system effect.

Essentially, duct turns or fittings change the performance of the fan and system when they're located close to the fan, and the more fittings there are, the worse the problem becomes.

Paying for system effect

When system effect can't be avoided, AMCA Publication 201 provides means of calculating its magnitude. Knowing the magnitude can give engineers the ability to calculate the trade-off between the cost of proper ducting versus operating cost, and reduce the uncertainty of how the fan and system will work together in order for the engineer to hit that peak efficiency sweet spot even when there is system effect. The following example is taken directly from Publication 201.

In our example system, there is a short outlet duct on a centrifugal fan followed by a plenum chamber with cross-sectional area more than 10 times larger than the area of the duct.

The required system pressure is calculated as follows:

Plane	Description	Pressure
C-D	Outlet duct on fan as tested	0 in. wg
D	P_v loss (also P_t loss) as a result of air velocity decrease P_s does not change from duct to plenum at D	0 in. wg
E	Contraction loss – plenum to duct	0.2 in. wg
E	P_s energy required to create velocity at E	0.5 in. wg
E-F	Duct friction at Q = 3,008 cfm	3.0 in. wg
Required fan P_s		**3.7 in. wg**

If we change the system as shown in Table 2 and Figure 4, where the fan now discharges directly the plenum, we've created a system effect.

Notice that Table 2 includes a loss for system effect, which was calculated in accordance with AMCA Publication 201.

Plane	Description	Pressure
B-C	System effect	0.6 in. wg
B-C	P_v loss (also P_t loss) as a result of air velocity decrease P_s does not change from duct to plenum at C.	0 in. wg
D	Contraction loss – plenum to duct	0.2 in. wg
D	P_s energy required to create velocity at D	0.5 in. wg
D-E	Duct friction at Q = 3,008 cfm	3.0 in. wg
Required fan P_s		**4.3 in. wg**

Beyond the details of the pressure calculation, what's striking about this simple example of system effect is the power required to overcome it. If we assume that the same fan can be used in both of the above systems, and that both operating points can be attained with a change in speed only, the increase in power consumption to overcome the calculated system effect is 25% of the original system's power.

How did we calculate that? Fan pressure changes are proportional to the square of the change in speed. Looking at this relationship in reverse:

$(P_{sB} / P_{sA})^{1/2} = (N_B / N_A)$

$(4.3 / 3.7)^{1/2} = 1.08$

Changes in fan power requirements are proportional to the cubed change in speed.

$(N_B / N_A)^3 = (H_B / H_A)$

$(1.08)^3 = 1.25$

Obviously, it's important to keep in mind that there's an economic trade-off between increased fan power costs due to shrinking mechanical rooms and increasing income from revenue-generating floor space.

When leveraging fan speed to overcome system effect, it is necessary to be aware of the real dangers in doing so.

Fans have structural elements that are affected by fan speed. These elements can fail catastrophically if the maximum safe speed of the fan is exceeded. Before increasing the fan's speed, contact the manufacturer to determine that the fan's maximum operating speed is not exceeded.

In addition, as shown previously, even relatively small increases in speed have large penalties in power. Before increasing speed, first calculate the new power required and ensure that the fan's motor and electric service have enough capacity.

Finally, when fan speed is increased, expect the fan to be louder, have increased noise from the air system's fittings, and have more duct and fitting leakage.

Conclusion

Properly sizing and selecting fans during the design phase are important first steps leading to high-performance air systems that have the lowest lifecycle cost of ownership. Providing proper inlet and outlet conditions also is critical, which means designing and specifying straight runs of inlet and outlet ducts, and seeing them through in the field during construction. Following these steps will ensure that fan systems will meet the system performance requirements with minimal energy consumption and noise, and with longest service life.

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More on system effect

When conducting the research testing for system effect, it's difficult, if not impossible, to separate the pressure drop of the fitting causing the system effect from the system effect that it causes. So, when using AMCA Publication 201, the calculated system effect will include the actual system effect plus the actual pressure drop of the fitting. This means that the system effect calculation may overstate the system's pressure drop if the user does not first subtract the fitting causing the system effect from his system loss calculation prior to adding it back in the form of system effect.

Another thing to keep in mind is that when we talk about system effect on the fan's inlet, we're really talking about a change in performance of the fan When we're talking about system effect on the fan's outlet, we're talking about an unusually high pressure drop through the fitting on the fan's outlet due to the fact that the very uneven velocity profile on the fan's outlet, with very high local velocities, causes a higher than normal pressure drop through the fitting.