

Fan Performance

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**AIR MOVEMENT AND CONTROL
ASSOCIATION INTERNATIONAL, INC.**

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Abstract

Unusually high operating costs are often caused by inefficient fan operation that, in turn, can be the result of improper fan selection or poor system design. Improper fan selection can often result in a fan that is too large, or a fan that is running at a higher speed than necessary for the application. The result of this is high energy costs, excessive airflow noise and increased maintenance requirements. Flawed system design can lead to high operating costs as a result of poor fan inlet or outlet airflow conditions, which can cause significant efficiency and airflow losses.

This paper discusses fan selection, methods of predicting pressure requirements to avoid poor system design, and ways to improve fan performance and energy efficiency with proper inlet and outlet ducting.

Fan Selection Process for a Simple System

The fan and air system market is mature. What that means to the process of fan selection is 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. Systems handling material often have radial bladed fans. Fans in residential applications are smaller and have single phase motors.

This case addresses a simple ventilation system in which the general fan type is already known. This means that with rudimentary knowledge in the hypothetical 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 select a fan, specifically suited for a job, which achieves a balance of maximizing efficiency and minimizing cost. However, the designer must understand how system design effects performance.

Fan Nomenclature and Specifications

Common industry nomenclature is important, and this section covers some of the more important terms when discussing fan performance.

As show in Figure 1, Plane 1 of the fan is the fan's inlet. Plane 2 is the fan's outlet.

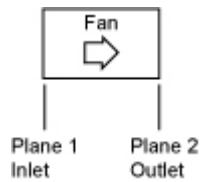


Figure 1

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

- Installation Type
- Density at the fan inlet, ρ
- Desired airflow rate, Q_r
- Pressure required to move the airflow rate desired, P_{st} or P_r

With these parameters in hand, a fan manufacturer should be able to provide several selections of fan types and sizes meeting the above requirements. 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 question, i.e., the best selection will have the lowest 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.

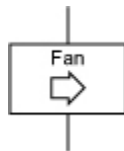


Figure 2 - Installation Type A: Free Inlet, Free Outlet



Figure 3 - Installation Type B: Free Inlet, Ducted Outlet



Figure 4 - Installation Type C: Ducted Inlet, Free Outlet

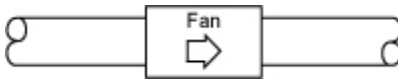


Figure 5 - Installation Type D: Ducted Inlet, Ducted Outlet

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 common for the fan in question.

Density

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 density. An analogy would be a man shoveling either peanuts or stones. Regardless of the material, the shovel holds the same volume, 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:

$$\rho_i = \rho$$

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

Flow Rate

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

$$Q_f = Q_i$$

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 another way of specifying mass flow rate. If ρ_s is standard air density (1.2 kg/m^3 or 0.075 lbm/ft^3), then:

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

Fan Pressure

As stated above, two types of pressure may be specified. Fan static pressure:

$$P_{sf}$$

Or, fan total pressure:

$$P_{tf}$$

Knowing when to use either depends on the system the fan is to be installed in. Two Installation Types will be examined here: B and D. But, before that, it is necessary to know exactly what P_{st} and P_{tf} mean.

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:

$$P_{vf} = P_{v2}$$

Which means:

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

This provides the system designer with the knowledge of how to specify fan pressure in Installation Types B and D.

In Installation Type B, where the system duct size matches the fan outlet, the inlet is open to atmosphere and we have the following:

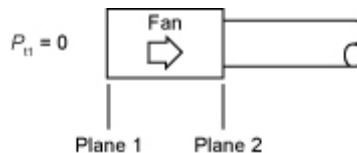


Figure 6

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

Fan static pressure is simply the losses in the system downstream of the fan, including the system's exit loss to atmosphere. Architectural louver ratings, when tested in accordance with AMCA 500-L, include exit losses. 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 D.

An article by Barrie Graham in *Heating, Piping and Air Conditioning* provides an excellent example of why fan total pressure should be specified for fans in Installation Type D.

Consider the fan installed in Figure 7.

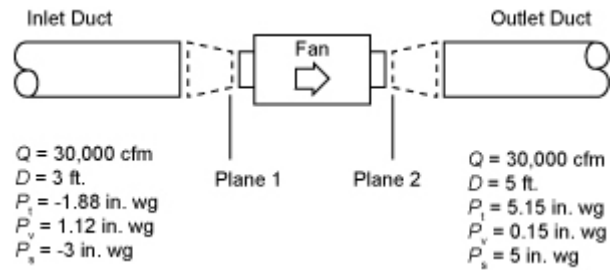


Figure 7

Note that neither the inlet nor outlet duct areas match the fan inlet or outlet, and transitions to and from the system's duct have not yet been designed because in this hypothetical situation we are still in the process of selecting the fan. Because the duct shown above must transition to the fan, the static pressures shown above 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.

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

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:

$$P_{sf} = 7.03 \text{ in. wg} - P_{v2}$$

System Effect

In order for a fan to achieve its rated performance, i.e., 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, as shown in Figure 8. The effect on fan performance when these conditions are not met is called System Effect.

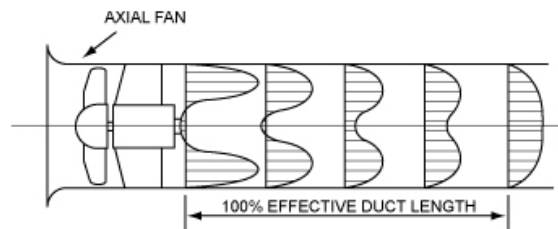


Figure 8

AMCA Publication 201 quantifies System Effect for a number of the more common causes, and offers recommendations for avoiding 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 100% of an Effective Duct Length on its outlet to avoid System Effect, while a vaneaxial fan needs 50% Effective Duct Length.

When System Effect can't be avoided, AMCA Publication 201 provides means of calculating its magnitude. The following two systems are examples taken directly from the publication.

In the first system, shown in Figure 9, 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.

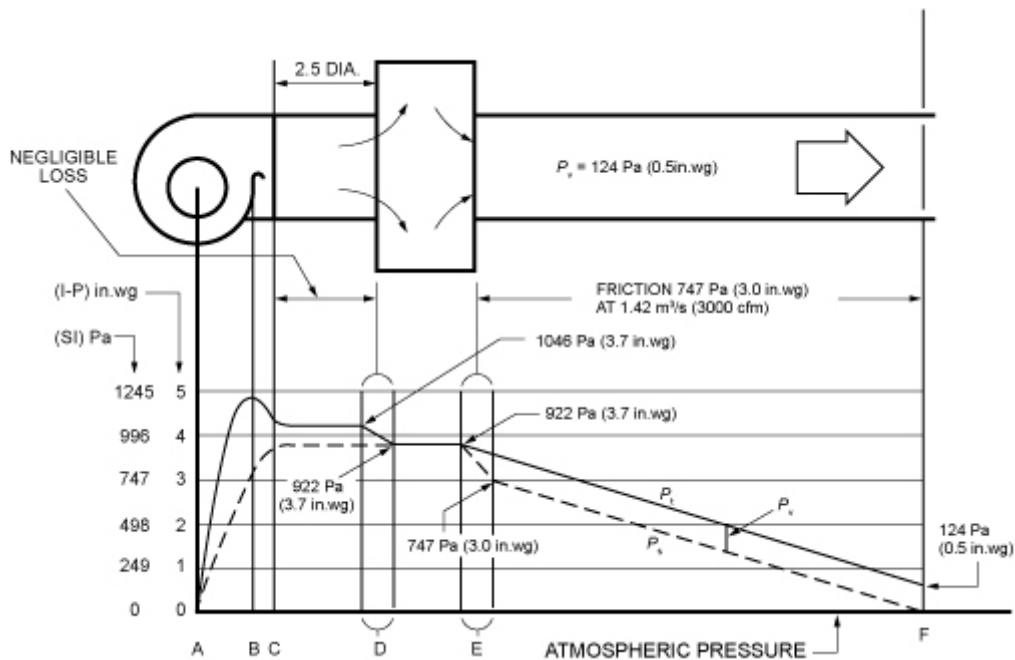


Figure 9

If we consider the pressure losses in the opposite direction of flow, the velocity in the duct from E to F is 14.4 m/s, equal to a velocity pressure of 124.5 Pa. At point F, the P_v is 124.5 Pa, the P_s is 0.0 Pa, and the P_t is 124.5 Pa. The friction of duct will cause a gradual increase in P_s and P_t back to point E. If the duct has a uniform cross-sectional area the P_v will be constant through this part of the system.

Since there is an energy loss of 49.8 Pa as a result of the abrupt contraction from the plenum to the duct, the P_t requirement in the plenum is 921.3 Pa, or $P_t = P_s$ (747 Pa) + P_v (124 Pa) + Contraction (49.8 Pa).

Air flowing across the plenum from D to E will have a relatively low velocity, P_v in the plenum will be 0.0 Pa, and then $P_s = P_t = 921.3$ Pa.

At point D, there is an abrupt expansion energy loss equal to the entire P_v in the duct discharging into the plenum. The outlet duct between the fan and the plenum is 2.5 equivalent diameters long, which is the same as used during the fan rating test. The P_s in the outlet duct is the same as the P_s as measured during the rating test.

		Pressure
C-D	Outlet duct on fan as tested	0 Pa
D	P_v loss (also P_t loss) as a result of air velocity decrease.	0 Pa
	P_s does not change from duct to plenum at D.	
E	Contraction loss – plenum to duct	49.8 Pa
E	P_s energy required to create velocity at E	124.5 Pa
E-F	Duct friction at $Q = 1.42$ m ³ /s	747 Pa
	Required Fan P_s	921.3 Pa

The second system, shown in Figure 10, is similar to the first, except the duct at the fan outlet has been omitted. The fan discharges directly into the plenum, which creates a System Effect.

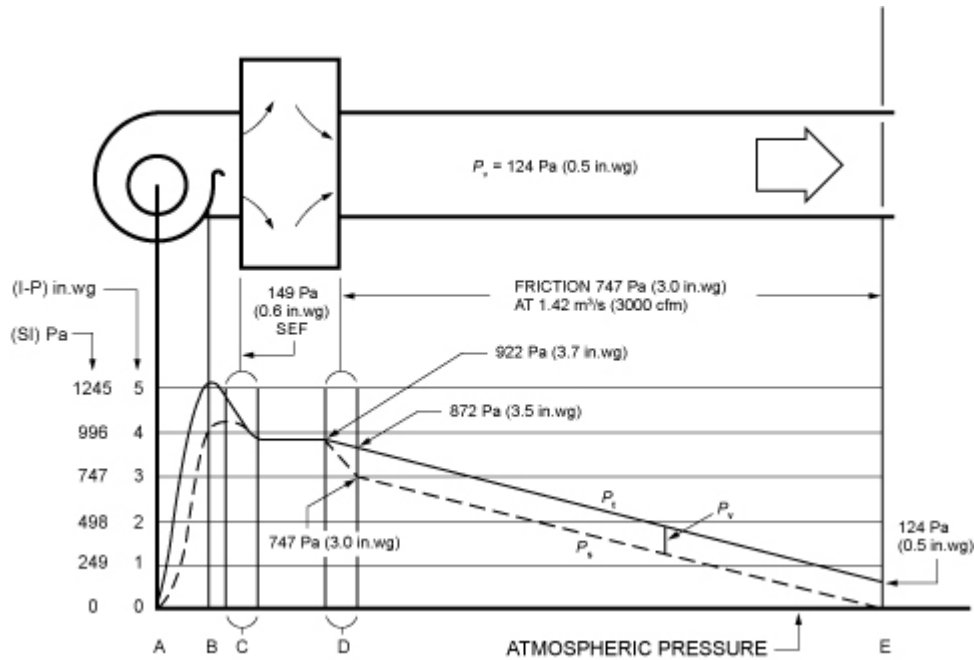


Figure 10

		Pressure
B-C	SEF	149.4 Pa
B-C	P_v loss (also P_1 loss) as a result of air velocity decrease.	0 Pa
	P_s does not change from duct to plenum at C.	
D	Contraction loss – plenum to duct	49.8 Pa
D	P_s energy required to create velocity at D	124.5 Pa
D-E	Duct friction at $Q = 1.42 \text{ m}^3/\text{s}$	747 Pa
	Required Fan P_s	1070.7 Pa

Notice that the above table includes a loss for System Effect. The System Effect was calculated as follows.

We assume in this case that Blast Area / Outlet Area = 0.6,

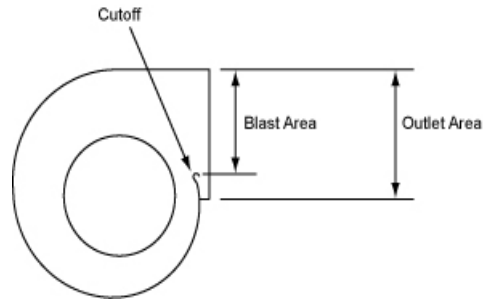


Figure 11

With this ratio, we find the applicable System Effect curve from Figure 8.3 of AMCA Publication 201.

	No Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	100% Effective Duct
$\frac{\text{Blast Area}}{\text{Outlet Area}}$	System Effect Curve				
0.4	P	R-S	U	W	—
0.5	P	R-S	U	W	—
0.6	R-S	S-T	U-V	W-X	—
0.7	S	U	W-X	—	—
0.8	T-U	V-W	X	—	—
0.9	V-W	W-X	—	—	—
1.0	—	—	—	—	—

Figure 12 - System Effect Curves for Outlet Ducts

Then, using curve R from Figure 7.1, we find the System Effect equal to 149.4 Pa.

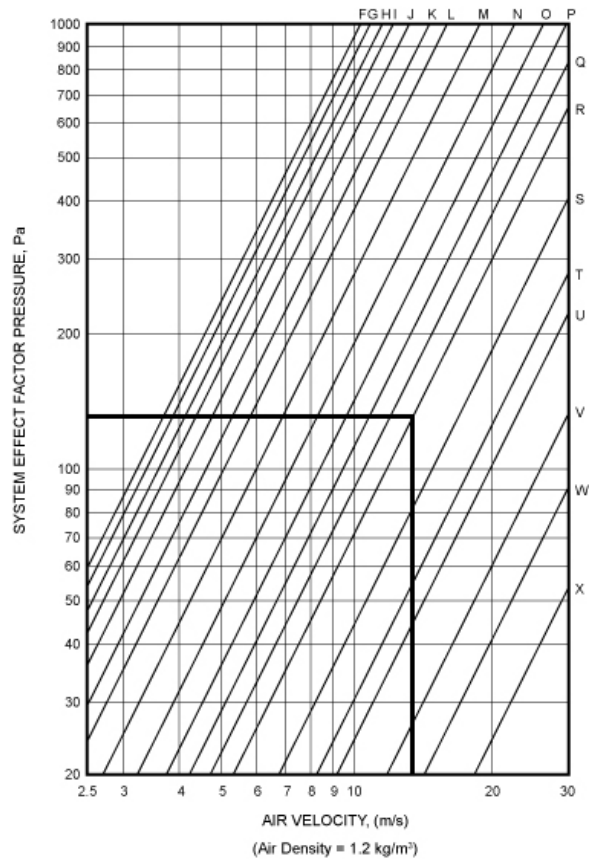


Figure 13

What is 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 our calculated System Effect is 25% of the original system's power.

This is calculated as follows: the Fan Law for pressure requires that the fan speed be increased 8% to overcome the additional 149 Pa of System Effect.

Fan pressure changes are proportional to the square of the change in speed. Looking at this in reverse:

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

$$(1071 / 922)^{1/2} = 1.08$$

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

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

$$(1.08)^3 = 1.25$$

It may be tempting from the reader's perspective to point out that, because the example system's pressure requirements were fairly low to start with, any System Effect added to the system will appear large. However, it should be noted that AMCA Publication 201 points to anecdotal evidence of airflow losses as high as 45% in poorly designed systems, and Bill Cory in *Fans and Systems* points out that flow rate reductions can be as high as 60%.

What should not be lost in this discussion is the economic trade-off between increased fan power costs due to shrinking mechanical rooms and increasing income from revenue-generating floor space.

Increasing Fan Speed

Before increasing the speed of a fan 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 show above, 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.

Catalogs

In the AMCA Certified Ratings Program, a great deal of effort is put into standardizing how fan data is presented to the customer. The data provided allows 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. See the above example.

Pressure

For fans installed in ducted systems, either P_s or P_t may be cataloged. For non-ducted fans, P_s is cataloged.

Accessories

Most fans are rated without including the effect of accessories. If an accessory is in the air stream, the effect of the accessory on fan performance needs to be taken into account. The fan manufacturer should be able to provide these data.

Drive or Transmission Losses

Depending on the fan's arrangement, these 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 United States Department of Energy study from 1997 showed that almost 60% of fans in building systems were oversized.² 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 (Point ① in Figure 14), but will instead operate at Point ②. It is possible for Point ② to generate more noise, and for some types of fans, Point ② can be an unstable operating point.

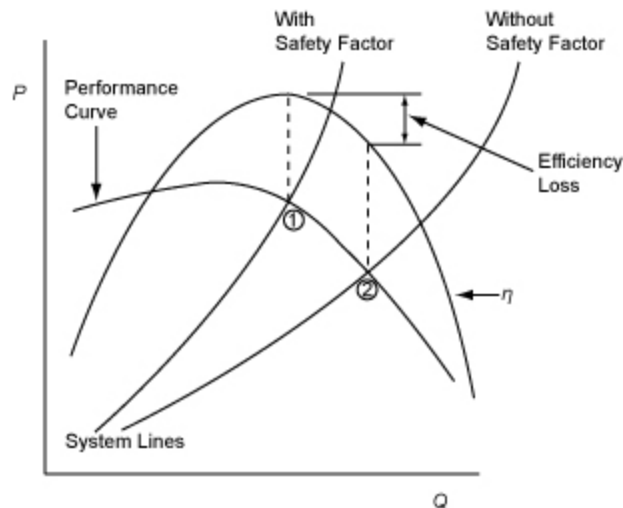


Figure 14

The use of safety factors also leads to over sizing 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.

Conclusion

The efficiency of the system is a combination of the efficiency components of the systems, which can be conceived abstractly as:

$$\eta_{\text{overall}} = \eta_f \times \eta_t \times \eta_m \times \eta_c$$

η_{overall} = overall fan efficiency

η_f = fan, $\eta_f = \eta_{\text{SE}} \times \eta_{\text{catalog}}$

η_t = transmission

η_m = motor

η_c = control system

η_{SE} = System Effect, zero System Effect means $\eta = 1.0$

η_{catalog} = catalog fan efficiency, affected by proper fan selection

Overall fan efficiency is the primary efficiency to consider when evaluating energy consumption rather than any single component. It is critical to consider the effect of the system on the fan when attempting to reduce fan energy consumption.

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