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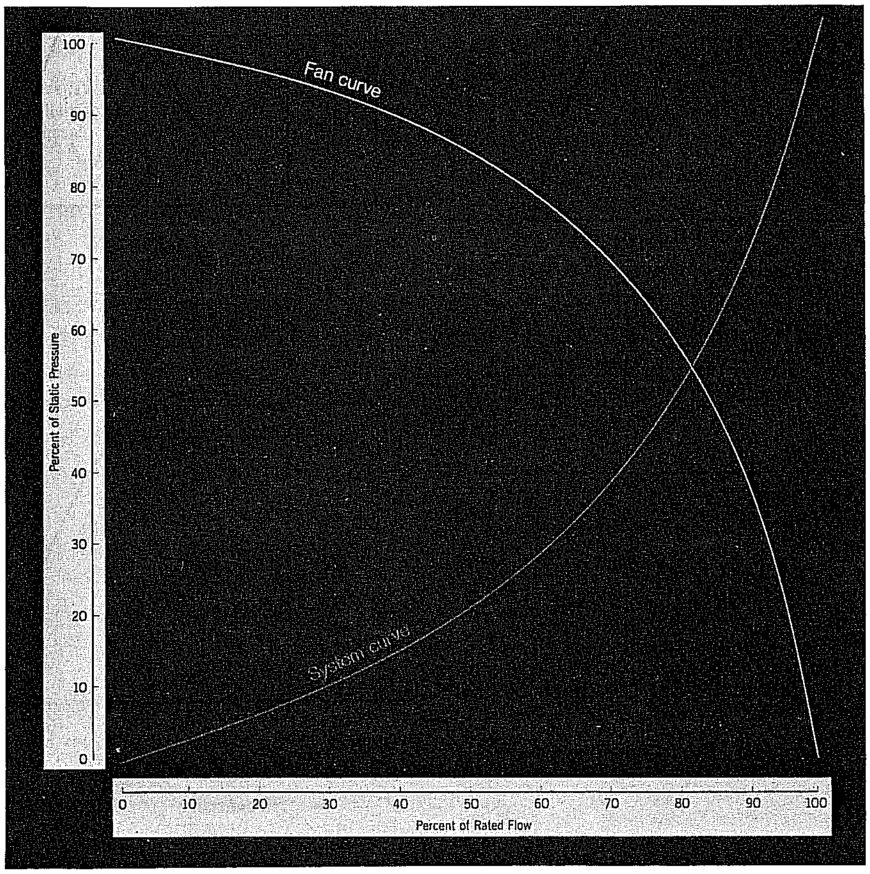
Analysis of fan/system characteristics and applications

Advanced technology and energy sensitivity in design call for more accurate methods of matching fan to system other than manufacturers' data coupled with safety factors. This discussion, based on actual tests, outlines a way to combine a system curve with a published fan curve for a true picture of system balance.



The word "system" and the phrase "systems approach" have been widely used in recent years to describe any number of concepts in engineering and architecture. At the risk of jumping onto a popular bandwagon, in this chapter "system" will be used in two different ways.

First, "system" as defined by the first meaning in *Webster* is "a regularly interacting or interdependent group of items forming a unified whole." It is in this context that systems engineering will be discussed. Second, to paraphrase the definition of system as defined in elementary thermodynamics, the term system "is used to designate any portion of matter that is separated from its surroundings by either real or imaginary boundaries." It is in this context that system will be used to analyze components or component subassemblies that operate within an overall engineered building system.

In the field of building environmental technology, HVAC systems are, in general, designed by a team of systems engineers. The approach is to select a grouping of manufactured products and integrate them together in such a manner that their respective interaction will achieve the design goals set for the overall system. However, over the years, manufacturing technology, taking advantage of the cost efficiencies of mass production versus field erection, has (at the cost of flexibility) resulted in prepackaging of various subsystems to be incorporated into an overall system, in many cases, in lieu of individual machinery components. It is the intent of this chapter to address the subjects of definition of boundaries in order to define the limits of such systems, and the resulting mathematical formulas that will improve the science currently employed by an overall systems designer. Each will be discussed in relationship to handling conditioned air.

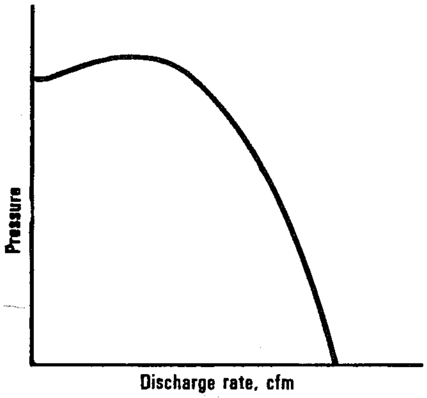


Fig. 53-1. A typical fan curve for a backward inclined or airfoil fan.

Fan curve important tool

Many years ago, manufacturers of air moving devices, blowers, developed an extensive grouping of variables affecting the performance of a blower when applied to an attached distribution system. A product's performance, as tested in accordance with industry testing standards, is registered as a fan curve. An example of a typical fan curve for a backward inclined or "airfoil" fan is shown in Fig. 53-1. Figure 53-2 illustrates the standard test method for measuring the respective inlet and outlet static and total pressures and volumetric flow rates. It is relevant to point out at this point that a fan is an assembly of components consisting of an inlet cone, wheel, shaft and bearing assembly, and scroll. The purpose of the test arrangement shown in Fig. 53-2 and the resulting curve and rating tables is to provide a standard method of rating the performance of one product compared to another. Members of the systems engineering profession have had only these data to assist in applying a fan to a system since the ratings were established; albeit standard test conditions are seldom, if ever, experienced in actual system installations.

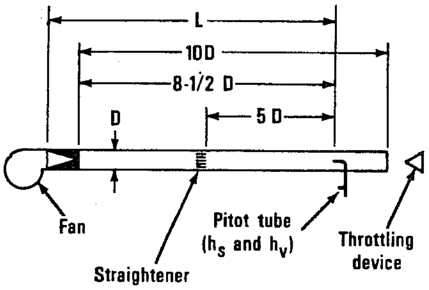


Fig. 53-2. The standard test method used to measure inlet and outlet static and total pressures and volumetric flow rates.

Understand fan laws

An additional consideration of background development are the so-called fan laws. Although much more extensive, all fan laws are based on three basic relationships:

- The discharge rate, cfm, varies directly as the speed.
- The total pressure increase varies directly as the square of the speed.
- The air power varies as the cube of the speed.

These laws are readily indentified by a vector analysis of a fan wheel if the system in which the fan is performing follows the turbulent flow characteristics of:

$$\Delta h = K_s(Q)^2 \qquad (1)$$

where

Δh = head loss,
K_s = system constant,
Q = discharge rate, cfm.

A systems designer, working with the only available information, applies a fan within an air system by using the manufacturer's comparative data in the form of fan curves (or tables), assumes the system curve is based on Eq. (1), plots the parabola against the respective fan curve, and thus identifies an anticipated operating point. This approach, however, has resulted in less than reliable results in a vast number of designs. Heretofore, multiple safety factors incorporated throughout design development have allowed approximation to continue. In recent years, however, with design refinements realized by computerized full load calculations, variable air volume systems wherein full load diversity is applied to fan and distribution system sizing, along with energy sensitivity in design, the profession is faced with the need for more accurate methods for matching a fan component to the distribution and conditioning components of an overall system. One method for assisting a system designer in improving the accuracy of application data accounts for system effects by assigning K-factor constants to various types of fan connection configurations.

Consider total fan system

A fan system is generally assumed to include, as stated previously, an inlet cone, wheel, and scroll as the devices that relate to characteristics; i.e., those devices that if dimensionally changed would result in a different characteristic curve. Thus, a system is defined by the boundaries encompassing these components.

It has also become rather common practice to include within the boundaries of a system variable inlet vanes. As vanes are closed, creating an additional pressure or energy loss at the inlet of a fan, the discharge rate decreases. Figure 53-3 shows a typical method of representing this phenomenon on a fan curve. If two operating points are selected,

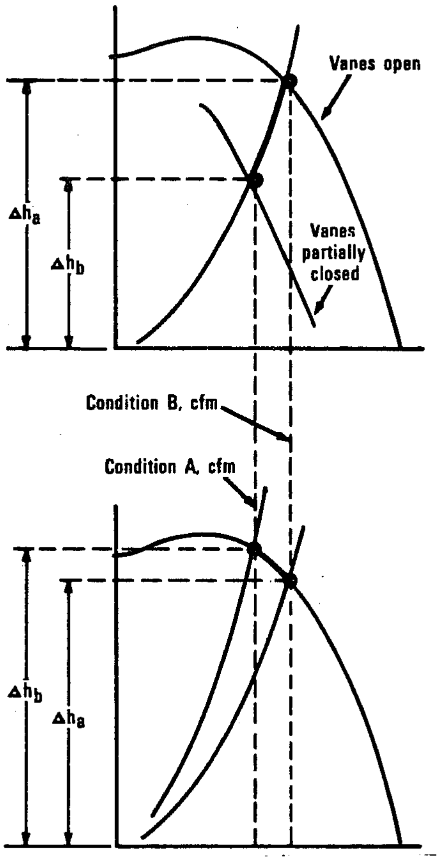
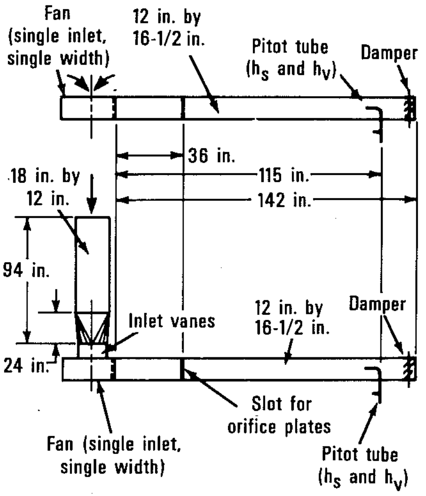


Fig. 53-3. A typical method of representing the decline in discharge rate when vanes are closed, creating an additional pressure or energy loss at the inlet of a fan. Referring to the bottom drawing, the curve moves upward and to the left when a damper is provided on the discharge side and the air flow rate is reduced from Condition A to B.

Conditions A and B, it is readily seen that as the inlet vanes close, both the discharge rate *and* the pressure rise decrease. If, on the other hand, a damper is provided on the discharge side of a fan, and the discharge rate is reduced from Condition A to B, this is generally represented as an increased constant K_S in Eq. (1), swinging the system curve upward to the left. In this case, the discharge rate decreases but the pressure rise increases. What explains the difference in these two relationships?



The basic hypothesis to explain this is simply where the boundaries of a system are defined. In the case of inlet vanes, a damper was included as part of the fan system, and in the other case, it was not. To substantiate this hypothesis, a test stand was constructed, admittedly with considerable liberties as concerns AMCA standard tests. The differences were necessitated by the need for changing the boundaries and obtaining consistent results. Figure 53-4 is a schematic diagram of the test configuration shown in the accompanying photo. The tests revealed that the inlet vane curve could be duplicated by positioning of the discharge damper and vice versa. To remove the inlet vanes from the fan section, the inlet pressure was measured in the center of the fan wheel. In either case, the fan system efficiency (assuming a good selection initially) decreased with reduced flow. However, this reduction in efficiency is less pronounced with the inlet vane system because of the directional nature of the vanes. This improved

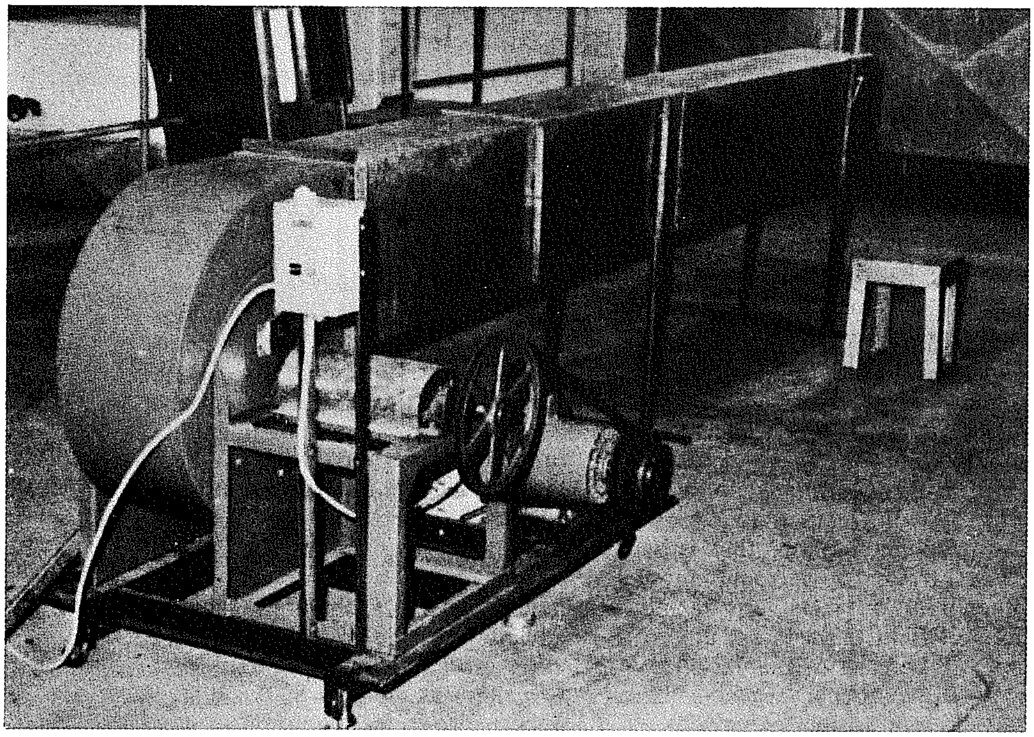


Fig. 53-4. A schematic diagram (above) and photograph of the test configuration.

reduced flow efficiency could likely be reproduced or even improved by fixed inlet vanes, which would still allow a designer to extend the boundaries of a fan system to include multiple distribution zones, each with different flow and pressure requirements. The control techniques would be identical to those employed for variable inlet vanes.

The use of inlet vanes for either initial balance of fan to system curve or for operational reduction of flow is not discouraged as a sound practice if it is applied with care. However, an application engineer should be careful in his decision to employ inlet vanes. The idea that reduced flow can be achieved with a more effective reduction in horsepower than possible by the same fan with discharge damper control is not always true. The basic reason is that because of the restrictive nature of inlet vane hardware at a very critical point in the fan system, the tests revealed a significant reduction in fan capacity with the vanes completely open. This restriction dropped the effective fan curve in the test apparatus by a pressure rise reduction of 15 percent with a discharge rate of 0.75 free delivery.

Many devices in air system

A composite air-handling system consists of multiple devices arranged in series with one another. Considering the simple system shown diagrammatically in Fig. 53-5, the devices are: return air inlet, return air duct, filters, cooling coil, fan, supply air duct, supply air grille, and the conditioned space.

In order to clearly understand the behavioral aspects of each device, the components are grouped into various systems by simply defining the respective boundaries. These boundaries as illustrated in Fig. 53-5 are: ducting and grilles, filters, coil, and fan.

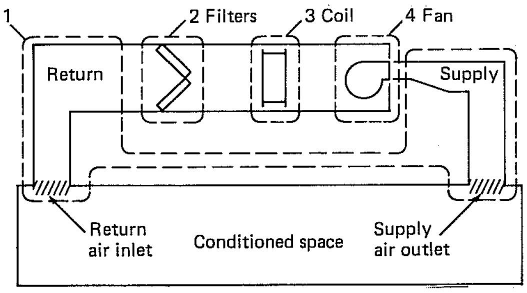


Fig. 53-5. A schematic diagram of the devices included in the boundaries of the test system.

The reason for grouping in this manner is that the pressure and flow rate characteristics of each device probably differ from those of the others.

The system including the ducting and grilles is found to be described quite accurately by Eq. (1). The source of this equation is the Darcy–Weisbach equation:

$$\Delta h = f(l/D)(V^2/2g) \qquad (2)$$

where

f = friction factor,
l = length,
D = diameter,
V = velocity,
g = gravitational constant.

In this equation, the friction factor is obtained from the Moody diagram and is relatively constant at the higher Reynolds numbers found in air-distribution systems.

This phenomenon of constant friction factor does not necessarily hold true for the flow rate friction loss characteristics of other systems, i.e., the coils and filters. The significance of this deviation from the form of Eq. (1) is becoming increasingly important in systems wherein the major contributors to head loss are these two components. This situation exists with even moderately efficient filtration and deep chilled water cooling coils. It is in this type of system that the classical system curve based on Eq. (1) and even application of the basic fan laws can be misleading during both design and balancing.

Cooling coil pressure drop

The published catalog data for pressure drops through cooling coils reveal that they do not behave in accordance with Eq. (1). The coils produced by one manufacturer were analyzed. The results obtained for wet coils were found to closely follow the relation:

$$\Delta h_c = K_c(Q)^{X_c} \qquad (3)$$

where

Δh_c = pressure drop through cooling coil,
K_c = coil constant.

The exponent, X ranged from 1.66 to 1.81 rather than the factor 2 as found in Eq. (1). For greater accuracy, the coil pressure drop should be expressed as:

$$\Delta h_c = K_{c_1}(Q)^{X_{c1}} + K_{c_2}(Q)^{X_{c2}}. \qquad (4)$$

However, the simpler form of the equation should suffice. Thus, if coil manufacturers would rate cooling coils simply by providing K_c values and X values for each series of coils, a designer could correct the available fan curve to account for coil pressure drop as discussed.

Filter air flow resistance

Filters manufactured of relatively high-resistance tightly woven media have come into common use in large building systems in recent years. These filters and cooling coils often represent the major air flow resistance elements. If a filter and its holding or mounting assembly are considered as the filter *system*, it is found that there are two distinct contributors to air flow resistance: configuration resistance and media resistance. In general, configuration resistance behaves in accordance with Eq. (1), and media resistance follows the laminar relationship of the Hagen Poiseuille law:

$$f = 64/Re \qquad (5)$$

where

f = friction factor,
Re = Reynolds number.

When combined with the Darcy–Weisbach equation, we obtain:

$$\Delta h = K_{f_1}(Q). \qquad (6)$$

If these two relationships were coupled, flow resistance for a filter system would be expressed by the equation:

$$\Delta h = K_{f_1}(Q) + K_{f_2}(Q)^2. \qquad (7)$$

However, with a reasonable degree of accuracy and within the discharge rate limits normally applied, the relationship for filters can be simplified to:

$$\Delta h_f = K_f(Q)^{X_f}. \qquad (8)$$

Again, a literature search of currently available filtration systems with efficiencies ranging from 95 to 35 percent revealed an exponent (X) value of from 1.49 to 1.70 with the higher exponent relating to filters in which configuration loss predominated over media loss.

Filter manufacturers do not catalog filter pressure drop versus flow characteristics for other than clean filters—a most unfortunate shortcoming from the standpoint of applying a filter to a system.

However, if a system engineer accepts a given pressure loss increase as the criterion for replacing filters, this fixed differential can be considered in matching a filter system to a fan system curve. Such a curve could be developed that takes these concepts into account and provides a more accurate picture.

Combining the data

The parabolic curve developed from Eq. (1) that represents the pressure loss characteristics found in duct systems (System 1, Fig. 53-5) is shown in Fig. 53-6. The significance in grouping the data in this manner is as follows:

- A fan system manufacturer could include in catalog data all aspects of a system provided as a product. For example, if a fan is furnished within a cabinet, the resultant losses can be represented as a depression in the fan curve. If the cooling coil is included, this depression in the curve can also be shown, and so forth.

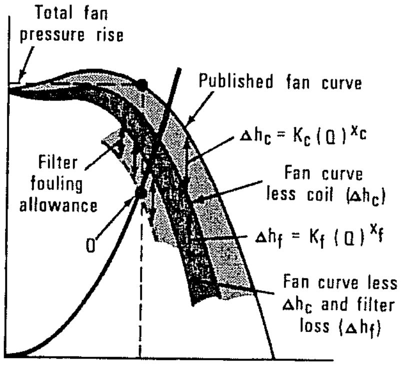


Fig. 53-6. A corrected fan curve that reflects the variable factors of coil pressure drop, filter loss, and fouling allowance.

Thus, for the so-called rooftop units that are currently so widely used, a manufacturer could publish an effective fan system curve to which a systems engineer need only apply the distribution curve to accurately identify the operating point.

- With this approach, the common fan laws could be applied more accurately to assist in system balancing, since the resulting external system follows the second power parabolic relationship stated in Eq. (1).

Consider energy use in design

The sensitivity to energy economics in systems design has created a major thrust to minimize fan power (and resulting energy) that has resulted in lower fan system pressures and an increasing use of variable volume systems. The use of lower system pressures creates situations wherein the major pressure losses are in the cooling coils and the filters, neither of which characterizes in accordance with Eq. (1), leading to error and misunderstanding both in the design and balance phases of system design. The use of variable volume systems results in the need for a clearer understanding of the behavior of the fan within the system—both for reasons of energy control and system performance under varying needs.

Into this latter void has come a groundswell of literature and products portraying the energy consumption and control advantages of controlled variable inlet vanes. In some applications, these devices are well applied; however, a designer should be aware of the limitations of such devices. For example, instead of comparing the reduced horsepower requirements for air flow of an inlet vaned fan at reduced flow to that at full design flow, a designer should first compare the full design flow horsepower of a fan with inlet vanes to the same fan without inlet vanes. In many cases (particularly with smaller fans), the variable inlet vane system may prove to consume more energy per year than the same fan with simple damper control. (However, well-designed fixed inlet vanes may prove *most* beneficial from an annual energy standpoint although these are very rare in the air conditioning products market.)

Standardize product ratings

Unfortunately, the industry rating systems for fan products have been developed for purposes of fair comparison of one product to another—not for the specific purpose of matching a fan to a distribution and conditioning system. The first effort at the latter is the ARI cabinet fan ratings, a system that should be extended to the entire field of manufactured subsystems.

Manufacturers of coils, filters, and other system components should standardize product data on flow versus pressure characteristics in terms of system constants (K) and exponents (X), so that an application or systems engineer could either develop a corrected fan curve to which the system curve could be applied, or an accurate system curve of the form:

$$\Delta h_s = K_c(Q)^{X_c} + K_f(Q)^{X_f} + K_d(Q)^2 \quad (9)$$

where

Δh_s = total system head loss,
K_c = cooling coil constant,
K_f = filter constant,
K_d = distribution system constant.

This equation could then be plotted against the published fan curve.

By combining this actual system curve, the published fan curve, and the mathematical relationship of Eq. (1) with the fan laws, the phenomenon of component matching and system balance would be better understood.