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Dear Subscribers:

This is update #38 that should be placed within your copy of the Heat Transfer and Fluid Flow Data Books. This material was ordered and paid for via subscription.

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All of the pages in this update go into the Fluid Flow Volume. This update was written by M. Mohammed Shah, Consulting Engineer, Port Jefferson, NY.

The new material is a comprehensive revision of Section 409 on Fans. Information on the relative merits of axial and centrifugal fans, on fan noise and stability, and on fan capacity control is presented in an easily readable format. The section also describes the system requirements for air flow and pressure and the selection of fan drives and motors. The section concludes with a list of fan manufacturers to aid the designer in the fan selection process.

New and revised pages in this update for the Fluid Flow Volume.

Section	Remove and Discard these pages	Add the revised Pages Dated	Reason for Change
FF List of Contributors	Previous pages dated May '82	April '83	New Author
FF Table of Contents	Previous pages dated May '82	April '83	Revision
409.1 through 409.13	All pages	April '83	Revision

We are enclosing the Directory of Licensed Products, Feb. 1983 from the Air Movement and Control Association Inc. You may save it for reference at the end of Section 409.

Yes, call us if you have any questions or comments.

Sincerely,

Joseph O. Accrocco

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Technology Marketing Operation
Phone (518) 385-2577

JOA/11
Enc.

★Supersedes issue of December 1969

SYMBOLS AND SUBSCRIPTS

The definitions of symbols given here apply unless defined differently in the text for a particular equation. Similarly, the units given here apply unless different units have been given in the text for a particular equation. For dimensionless equations any consistent units, besides those given here, can be used.

Symbol	Quantity	Units
a	radius of generating circle of involute scroll	ft.
A	area	sq. ft.
A_2, A_f	rotor discharge area: axial flow: $A_2 = \frac{\pi(D_2^2 - D_1^2)}{4}$ centrifugal: $A_2 = \pi D_2 b$	sq. ft.
b	width	ft.
c	clearance	ft.
d	particle diameter	ft.
d_s	specific diameter, defined by Eq. (1-12)	$\text{ft.}\frac{(\text{in. WG}\dagger)^{1/4}}{(\text{ft.}^3/\text{min.})^{1/2}}$
D	diameter (in general)	ft.
D_1	impeller diameter at inlet for centrifugal fans, hub diameter for axial flow fans	ft.
D_2	impeller diameter at inlet for centrifugal fans, tip diameter for axial flow fans	ft.
D_c	characteristic diameter of fan	ft.
D_{eq}	equivalent diameter of rectangular duct, defined by Eq. (4-9)	ft.
D_o	diameter of inlet orifice	ft.
g	gravitational acceleration	ft./sec.^{2}
K_{se}	system effect loss coefficient, defined by Eq. (4-7)	dimensionless
l_c	chord length	ft.
L	length	ft.
L_{min}	minimum length of straight discharge duct needed to avoid system effects	ft.
M	Mach number	dimensionless
n	rotational speed	rpm
n_s	specific speed, defined by Eq. (1-11)	$\left(\frac{\text{ft}^3}{\text{min}}\right)^{1/2}\frac{1}{(\text{in. WG}\dagger)^{3/4}}$
N	number of blades	dimensionless
p	pressure	lb/sq. ft.
Δp	pressure difference, or pressure developed by fan	lb/sq. ft.
Δp^*	pressure difference, or pressure developed by fan	in. WG†
Δp^*_{fse}	loss of pressure due to fan system effects	in. WG†
Δp^*_s	static pressure difference, or static pressure developed by fan	in. WG†
Δp^*_t	difference in total pressure, or total pressure developed by fan	in. WG†

†in. WG = 1 inch "water gauge"
= 1 in. of water

Symbol	Quantity	Units
P	power input to air	any unit
P_H	power input to air	horsepower
P_w	power input to air	watts
PWL	sound power level, defined by Eq. (6-1)	decibels (db)
PWL_s	specific sound power level	decibels (db)
Q	volumetric flow rate	cfm = cubic ft./min.
R_D	Reynolds number based on diameter	dimensionless
S	fan shape factor	dimensionless
S_1	diameter ratio D_1/D_2	dimensionless
SPL	sound pressure level, defined by Eq. (6-2)	decibels (db)
t	in Section 409.3, blade spacing	ft.
t	in Section 409.4, temperature of air	°F
u	blade velocity at any point	ft./sec.
u'	blade velocity at any point	ft./min.
v	relative fluid velocity	ft./sec.
v'	relative fluid velocity	ft./min.
V	absolute fluid velocity	ft./sec.
V'	absolute fluid velocity	ft./min.
w	width of scroll	ft.
α	is proportional to	dimensionless
α_e	expansion angle	degree, radian
β	blade angle, measured from tangential direction	degree radian
β^*	angle made by fluid with tangential direction	degree, radian
β_{co}	cutoff angle	degree, radian
γ	in Section 409.1, ratio of specific heats	dimensionless
γ	in Section 409.3, stagger angle	dimensionless
Δ	"change of"	dimensionless
η	efficiency	dimensionless
θ	angle	degree, radian
μ	$\frac{\text{mass flow rate of solids}}{\text{mass flow rate of gas}}$	dimensionless
ν	kinematic viscosity	ft²/sec.
ρ	density	lb/cubic ft.
ϕ	flow coefficient, defined by Eq. (1-6)	dimensionless
ψ	pressure coefficient, defined by Eq. (1-5)	dimensionless

SUBSCRIPTS

1	pertaining to inner blade edge (hub or blade inlet)
2	pertaining to blade outer edge, blade tip, or blade outlet
1,2,3	also used to distinguish between different fans, shape parameter, etc.
a	axial component of
b	blade
i	at fan inlet
m	at mean diameter as defined by Eq. (3-1)
r	radial component of
t	tangential component of; total

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I. FUNDAMENTAL CONCEPTS AND DEFINITIONS

A. Fan Defined

The word fan, as used here, applies to a device for delivering or exhausting a quantity of air or other gas with little change in pressure, by means of a rotating impeller. If the change in pressure is substantial, the device is known as a compressor. The term blower is often applied to low volume high pressure centrifugal fans. The word fan may mean only the impeller, or it may mean the impeller, casing and any other components that may be considered part of the unit, such as the fan shaft, guide vanes, inlet cone, outlet, and diffuser. Frequently the impeller is referred to as the fan wheel or rotor, and the fan casing as housing, shell, volute, scroll, or diffuser.

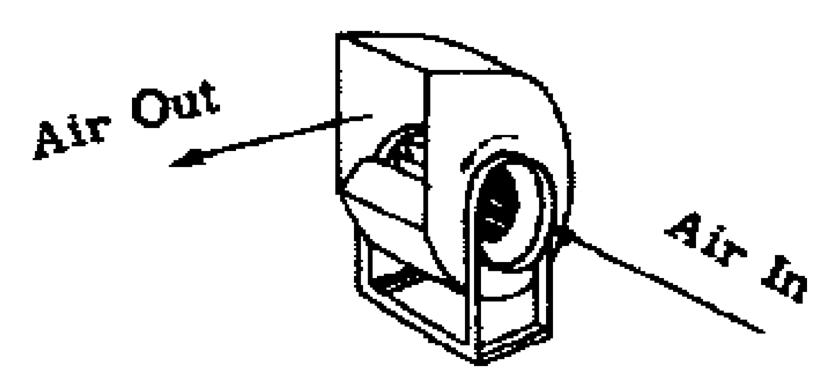
B. Types of Fans

The various types of fans are briefly described here. See [1.1] to [1.7] and Sections 409.2 and 409.3 for more information.

1. Centrifugal Fans

In centrifugal fans, compression occurs mainly due to centrifugal action. Motion of gas over the impeller occurs substantially in the radial direction. For this reason, they are also known as radial flow fans. Figure 1-1 shows a typical centrifugal fan. Also see Figure 2-1.

Figure 1-1. Typical Centrifugal Fan.



2. Axial Flow Fans

Axial flow fans are fans in which the motion of the fluid over the impeller is parallel to the impeller shaft. The three basic types of axial flow fans are the propeller, tubeaxial, and vaneaxial. The most common axial flow fans are the familiar desktop fans and ceiling fans. Figure 1-2 shows the three basic types of axial flow fans.

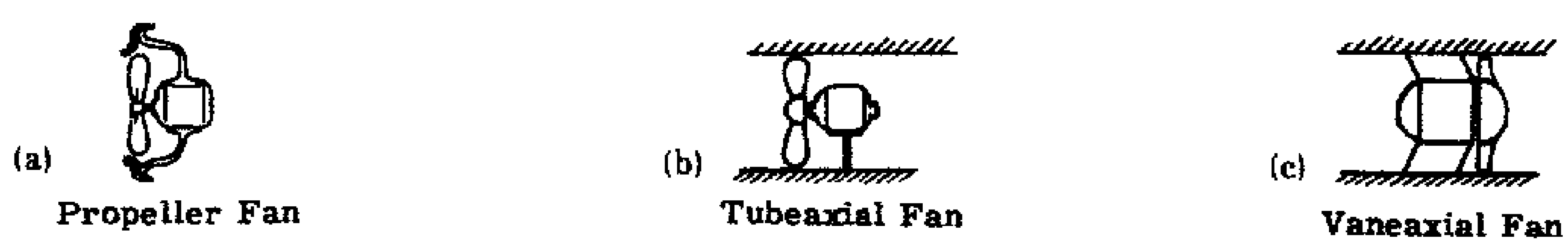


Figure 1-2. The Three Basic Types of Axial Flow Fans.

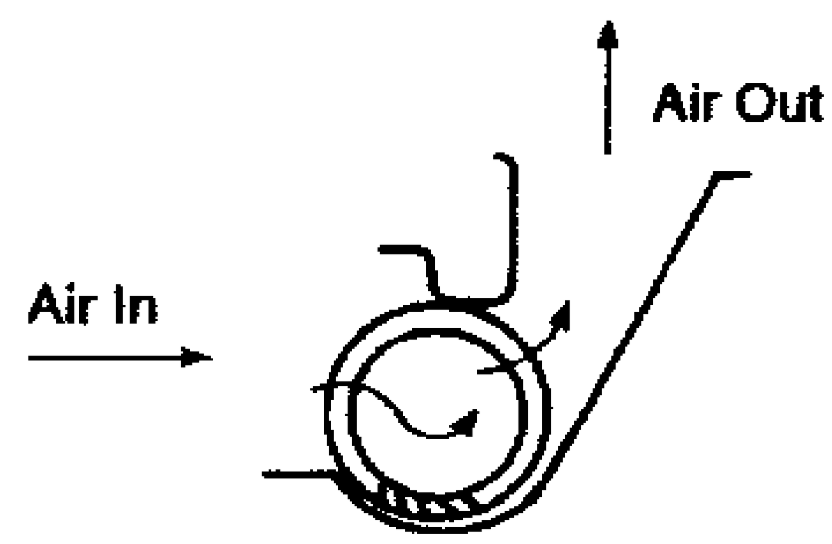
3. Mixed Flow Fans

The term mixed-flow fan is sometimes applied to radial-flow fans which receive their energy from the axial direction and whose blade shape gives the fluid an axial component of velocity while passing through the blades. The term is also sometimes applied to axial-flow fans in which the fluid passing through the impeller has a large radial component of velocity.

4. Cross Flow Fans

Cross-flow fans have impellers similar to those of centrifugal fans. Air enters tangential to the blade tip, comes out, and then passes over the blades on the other side before leaving the impeller. Most of the pressure produced is in the form of velocity pressure. Figure 1-3 shows a cross flow fan.

Figure 1-3. Cross-flow fan.



Cross-flow fans have been used in air-curtains, spin dryers, portable electric heaters, etc. but only rarely. They are not discussed any further here. Interested reader will find a comprehensive review in [1.1]. Ref. [1.8, 1.9, 1.10] report recent researchers on crossflow fans.

5. Fans of Special Design

Two widely used special fan designs are described here.

Tube-Centrifugal Fan

This fan has a radial flow impeller fitted in a cylindrical casing. Air leaves the impeller radially but then turns at right angle and discharges axially out of the fan. This fan is also known as inline centrifugal. Figure 1-4 shows this type of fan with common terminology.

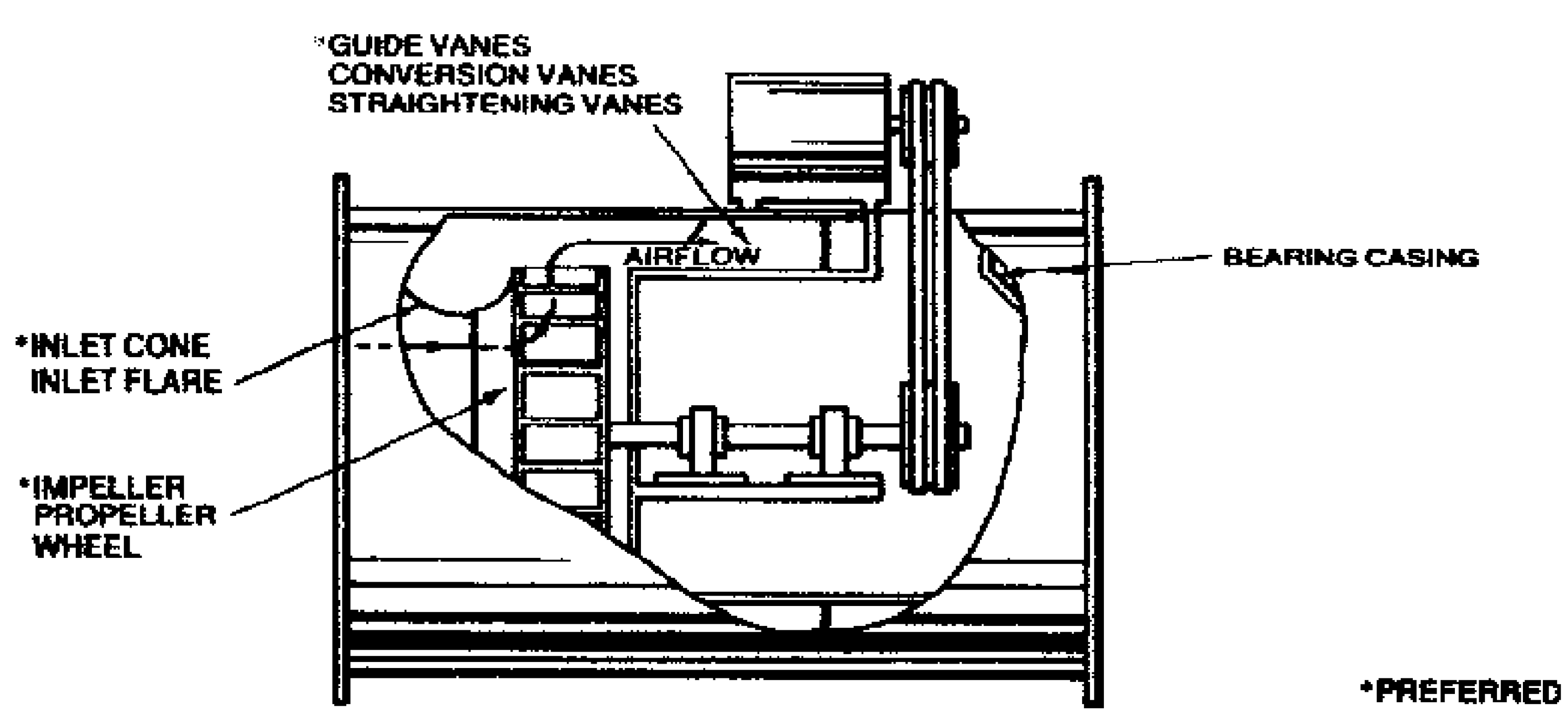


Figure 1-4. Tube-centrifugal fan and common terminology for its components.
(From AMCA Publication 201)

Power Roof Ventilators

These are widely used for exhaust ventilation and providing makeup air. Their impeller may be of centrifugal or propeller type. The centrifugal type have no housing in the conventional sense and discharge the air in a 360° pattern. Power roof ventilators are generally sold as packages which include a weather hood and hardware for easy installation. See the article by Wendes and Pannoke [1.11] for a description of the various designs of power roof ventilators. Figure 1-5 shows the two basic types.

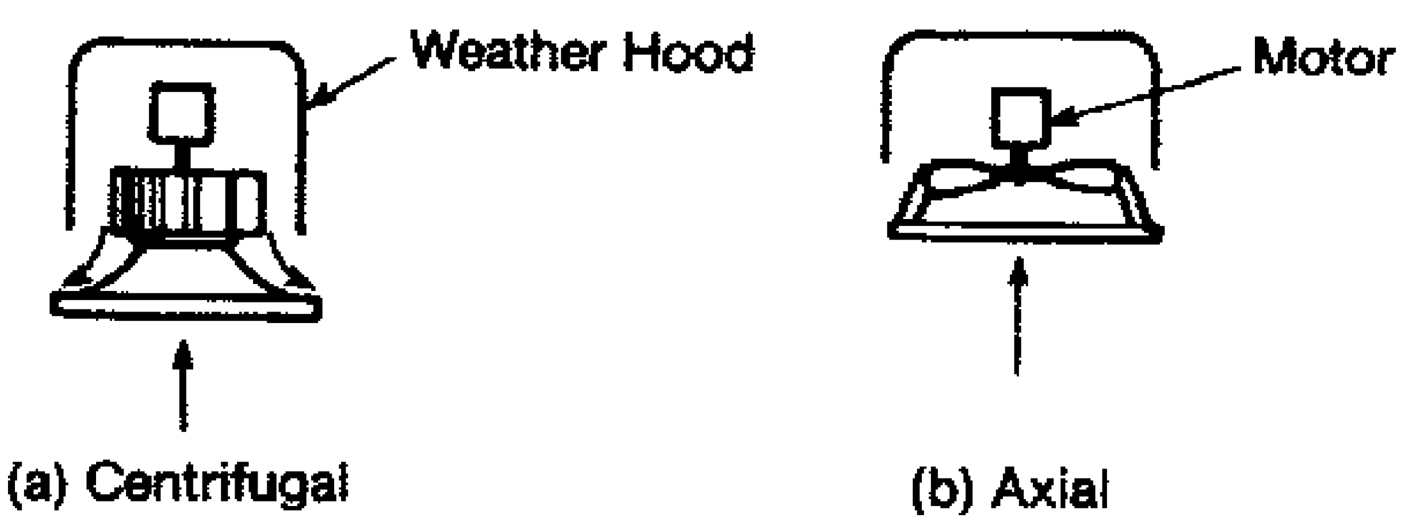
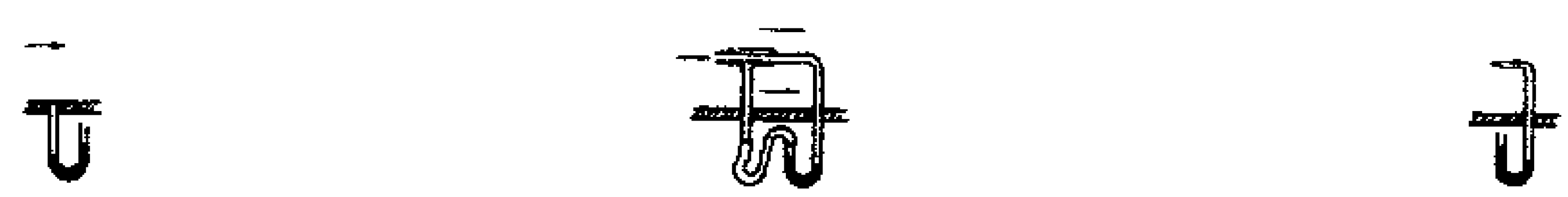


Figure 1-5. The Two Basic Types of Power Roof Ventilators.

C. Pressure

1. Static Pressure

In fan calculations, the static pressure, or static head, at a given point in a moving gas is defined by the height of a column of a fluid, usually water, which the pressure at that point will just sustain. Figure 1-6a shows how to obtain a measurement of static pressure.



(a) (b) (c)

Figure 1-6. Pressure Measurement.

2. Velocity Pressure

The velocity pressure or velocity head, at a given point in a moving gas is defined as the pressure corresponding to the kinetic energy of the gas moving at velocity, V. It is obtained by converting the velocity at that point into equivalent pressure by means of the expression $\Delta p_v^{*\prime} = V^2/2g$ from Bernoulli's equation. Here, $\Delta p_v^{*\prime}$ is expressed in feet of the gas flowing, V is velocity in ft/sec and g is gravitational acceleration in ft/sec^2. In fan calculations this pressure is usually converted into inches of water by means of the expression $\Delta p_v^* = (V'/4000)^2$, where V' is in ft/min, provided the gas is air at close to atmospheric pressure and a temperature of 70°F. A special pitot tube for measuring velocity pressure is shown in Figure 1-6b.

3. Total Pressure

The total pressure at a given point in a moving fluid is equal to the sum of the static and velocity pressures at that point. It is sometimes called impact pressure or head, because it is measured with a simple impact tube pointed against the direction of flow of the fluid as in Figure 1-6c. As the static and velocity pressures are inter-convertible, it is more meaningful to compare total pressures and total pressure losses. However, the use of static pressure for fan performance and pressure drop calculations is widespread.

D. Efficiency

The efficiency of a fan is defined as the ratio of useful power output to power input. The power output is commonly called air horsepower, and the power input is brake horsepower (bhp). The air horsepower is directly proportional to the product of the rate of air flow and the increase in air pressure. Since the gas has a total pressure which is made up of static and velocity components, efficiency is commonly specified in terms of static efficiency and total efficiency.

In those cases where static pressure is desired, there is little worth in total efficiency values. However, where velocity pressure is important in the service application, total efficiency is more important than static efficiency.

Static efficiency is more commonly used because it is easier to measure and because the static efficiency characteristics of some fans are more sharply defined than the total efficiency characteristics.

The static and total efficiency are given by

$$\text{Static efficiency } \eta_s = \frac{0.000157 Q \Delta p_s^*}{P_H} = \frac{0.118 \Delta p_s^* Q}{P_w} \tag{1-1}$$

$$\text{Total efficiency } \eta_t = \frac{0.000157 Q \Delta p_t^*}{P_H} = \frac{0.118 \Delta p_t^* Q}{P_w} \tag{1-2}$$

where P_H and P_W are actual power input to the fan in horsepower and watts respectively, Δp^* is expressed in inches of water, and Q is flow in cfm.

Losses occur due to leakages, fluid friction, and mechanical friction. Efficiencies associated with various losses can be defined. Generally, the total efficiency may be expressed by the following expression.

$$\eta_t = \eta_{vol} \eta_h \eta_m \tag{1-3}$$

where η_{vol} is the volumetric efficiency, η_h is the hydraulic efficiency, and η_m is the mechanical efficiency. These concepts are useful in fan development. For fan selection, only η_t and η_s are of interest.

E. Geometrically Similar Fans

Two fans are geometrically similar when all dimensions of the second fan are scaled up (or down) in equal proportion to the dimensions of the first fan to give a fan of the same shape but a different size. It is common manufacturing practice, when a fan shape has been obtained which will give desirable operating characteristics, for example, high peak efficiency, to construct a series of geometrically similar fans to handle different requirements of pressure and rate of flow. All the fans of this series, within wide limits of size and speed, will have the same peak efficiency and the same type of operating characteristics. The actual pressures, rates of flow, speeds, and horsepower requirements of the different fans will be related by the fan laws. These laws are explained in 409.5.

The amount of noise produced by similar fans can be calculated by a similar set of relationships if the fans are operating in similar surroundings. These noise laws are also explained in Section 409.6.

F. Performance Coefficients

Using the methods of dimensional analysis, the pressure developed by a fan system can be expressed in the form

$$\Delta p = \frac{\rho u_2^2}{g} f\left[\left(\frac{Q}{u_2' D_2^2}\right) \cdot \left(\frac{u_2 D_2}{\mu}\right) \cdot \left(\frac{u_1}{\frac{\gamma p_1 g}{\rho}}\right) \cdot \left(\frac{L_n}{D_2}\right) n = 1, 2, \ldots \right] \tag{1-4}$$

where $\frac{L_n}{D_2}$ is the ratio of a linear dimension of the fan (L_n) to the fan tip diameter D_2. Let

$$\psi = \frac{\Delta p}{\frac{\rho u_2^2}{2g}} \quad \text{Pressure coefficient} \tag{1-5}$$

$$\phi = \frac{Q}{u_2' A_2} = \frac{V'_{2m}}{u_2'} \quad \text{Flow coefficient} \tag{1-6}$$

$$M_1 = \frac{u_1}{\sqrt{\frac{\gamma p_1 g}{\rho}}} \quad \text{Mach number} \tag{1-7}$$

$$R_D = \frac{u_2 D_2}{\gamma} \quad \text{Reynolds number} \tag{1-8}$$

$$\frac{L_n}{D_2} = S_n \quad \text{Shape factor} \tag{1-9}$$

It should be noted that in some publications, ψ is defined such that it is half of the value given by Eq. (1-4). Hence the definition of ψ should be checked before using any data or formula. The equation above can then be rewritten as:

$$\psi = f(\phi, R_D, M_1, S_1, S_2 \ldots S_n) \tag{1-10}$$

For many applications ψ will be independent of R_D and M_1. Thus for a given geometric fan family ($S_1, S_2 \ldots S_n$ the same) ψ is dependent only on ϕ.

$$\psi = f(\phi)$$

Under these conditions, there is a one to one relationship between flow coefficient and pressure coefficient.

The relation $\psi = f(\phi)$ or $\frac{\Delta p}{\frac{\rho u_2^2}{2g}} = f\left[\frac{Q}{u_2' A_2}\right]$ also embodies all of the fan laws (see 409.5). As such it is often convenient to plot fan data in terms of ψ vs ϕ and η (static efficiency). Plots of this type are useful for:

1) Prorating known fan performance to different speed, size, or fluid medium within the limits of independence on Reynolds number or Mach number.

2) Comparing the performance of geometrically different fans, run at different conditions.

3) Correlating test information.

4) Identifying Reynolds Number and Mach number effects.

Pressure coefficient Ψ can be interpreted as the pressure developed in proportion to the dynamic head of the blade tip.

Flow coefficient ϕ is the ratio of the average velocity normal to the rotor discharge area to the rotor tip speed.

For a given operating point, the relation $\Psi = f(\phi)$ implies that with ϕ, Ψ, η, fixed:

1) Pressure ~ rpm^2
   ~ density

2) Cfm ~ rpm independent of density

Figure 1-7 shows a comparison of centrifugal fans with different kinds of blades in terms of ϕ and Ψ.

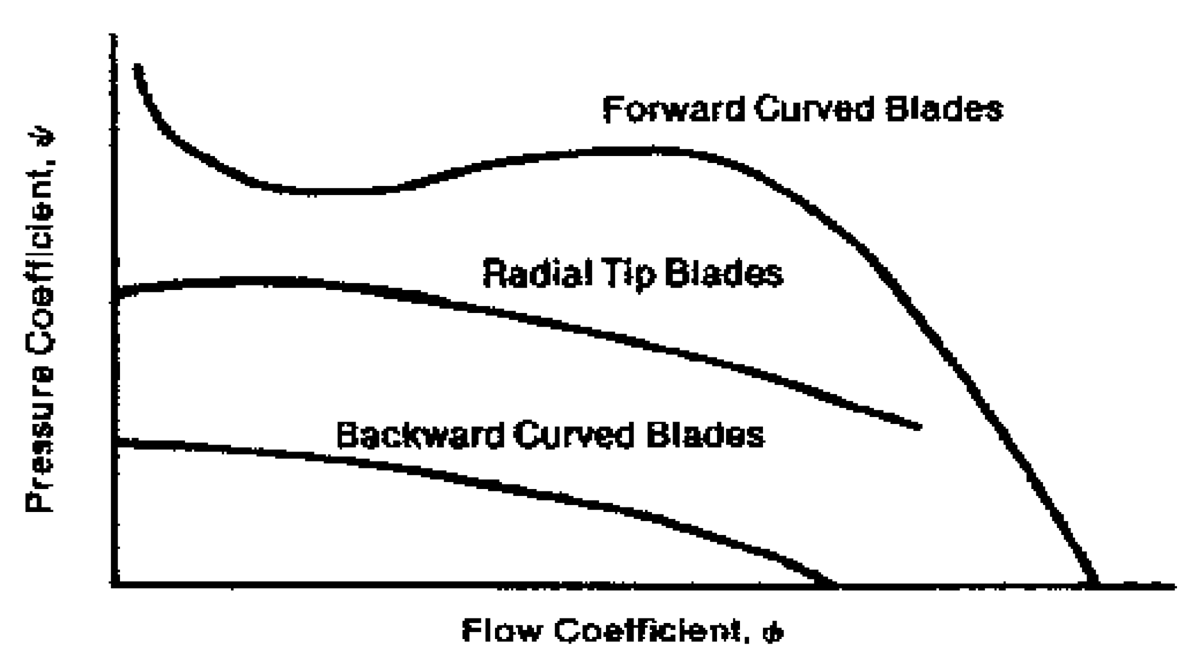


Figure 1-7.

G. Specific Speed and Specific Diameter

Another commonly used performance coefficient is the specific speed, n_s. This is defined as the speed of a geometrically similar fan, forcing one cubic foot of air per minute against a head of one inch of water.

$$n_s = \frac{nQ^{1/2}}{\Delta p^{*3/4}} \left(\frac{\rho}{0.075} \right)^{3/4} \tag{1-11}$$

Specific diameter d_s is defined by the following equation:

$$d_s = D_c \frac{\Delta p^{*1/4}}{Q^{1/2}} \left(\frac{0.075}{\rho} \right)^{1/4} \tag{1-12}$$

where D_c is a characteristic diameter of the fan. For units to use in equation (1-11) and (1-12) see Part 2, Symbols. Plots of specific diameter and efficiency against specific speed are sometimes used for comparing different types and designs of fans. Figure 1-8 is an example. While these parameters have the merit of more compact presentation of performance data, manufacturers rarely publish their data in this form. Hence these parameters are rarely useful in fan selection. These can be very useful in fan design, as will be seen in Section 409.3.

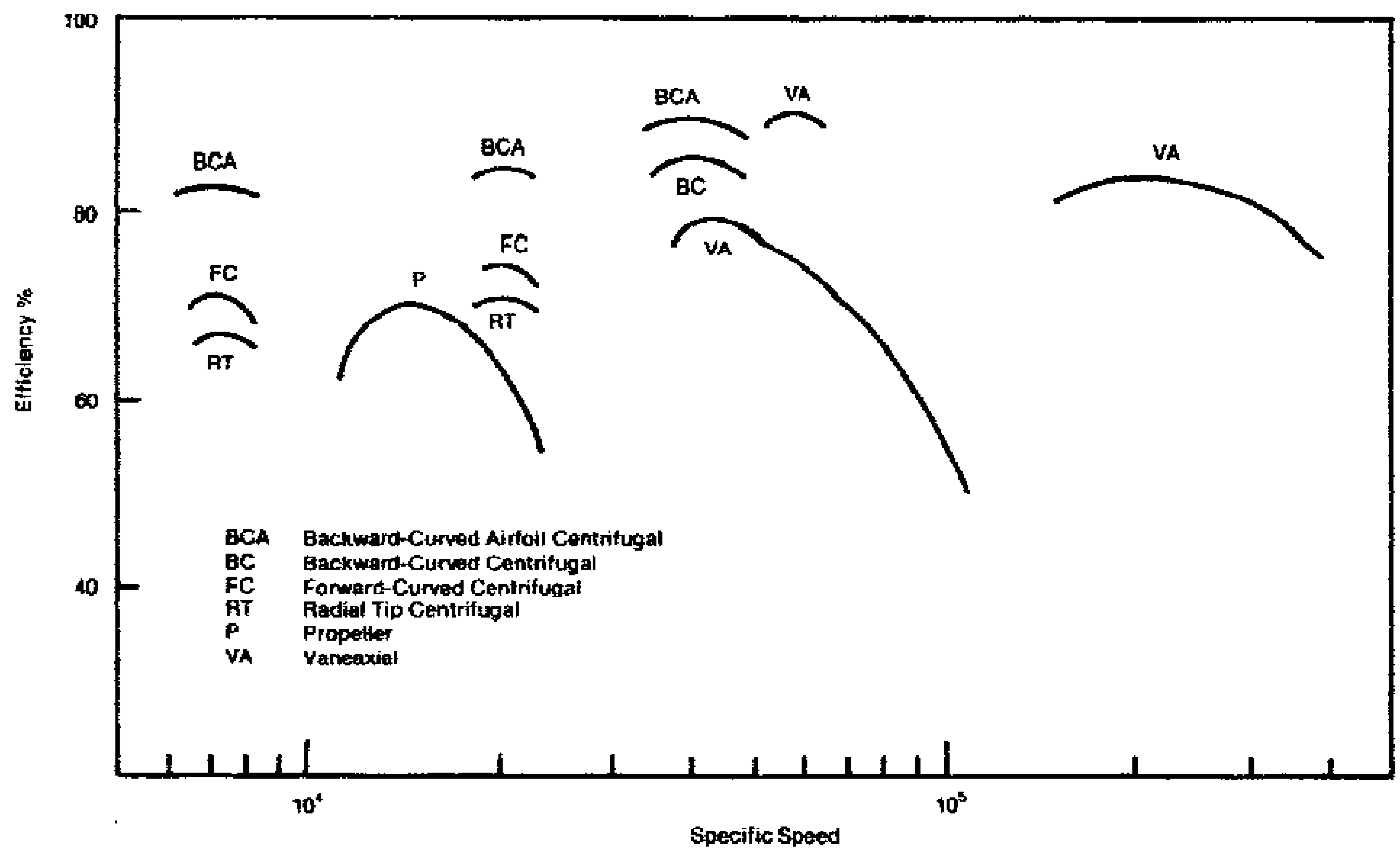


Figure 1-8. Efficiencies of Various Types of Fans as Functions of Specific Speed.

The definitions of specific speed and specific diameter used in some publications are different from those given here. Hence before using any data in terms of these parameters, the definitions used should be carefully checked.

H. System Resistance and System Curve

The system through which the fan is required to move air offers resistance to flow which causes pressure drop. The fan has to develop sufficient pressure to overcome this pressure drop. The resistance to flow is due to frictional effects and due to dynamic effects. In turbulent flows, resistance to flow is approximately proportional to the square of volumetric flow rate. Hence a system resistence R may be defined by the following equation:

$$\Delta p^* \propto Q^2 R \tag{1-13}$$

The flow in most practical systems is turbulent.

A system curve is a plot of pressure drop in the system at various flow rates. If the pressure drop at a particular flow rate is known, the pressure drop at other flow rates can usually be calculated by the above relation. System curves are very valuable in fan selection and analyzing the fan-system relation over the range of operating conditions.

I. ELEMENTS OF FAN

Figure 2-1 shows the elements of a typical centrifugal fan and the terms by which they are called. This figure shows the most common design of centrifugal fans in which the impeller is enclosed by a housing which discharges the air at right angles to the air inlet direction. There are designs in which there is no housing, for example in power-roof ventilators. In tube-centrifugal fans, the impeller is enclosed in a tube and the discharge is in the axial direction, as shown in Figure 1-4.

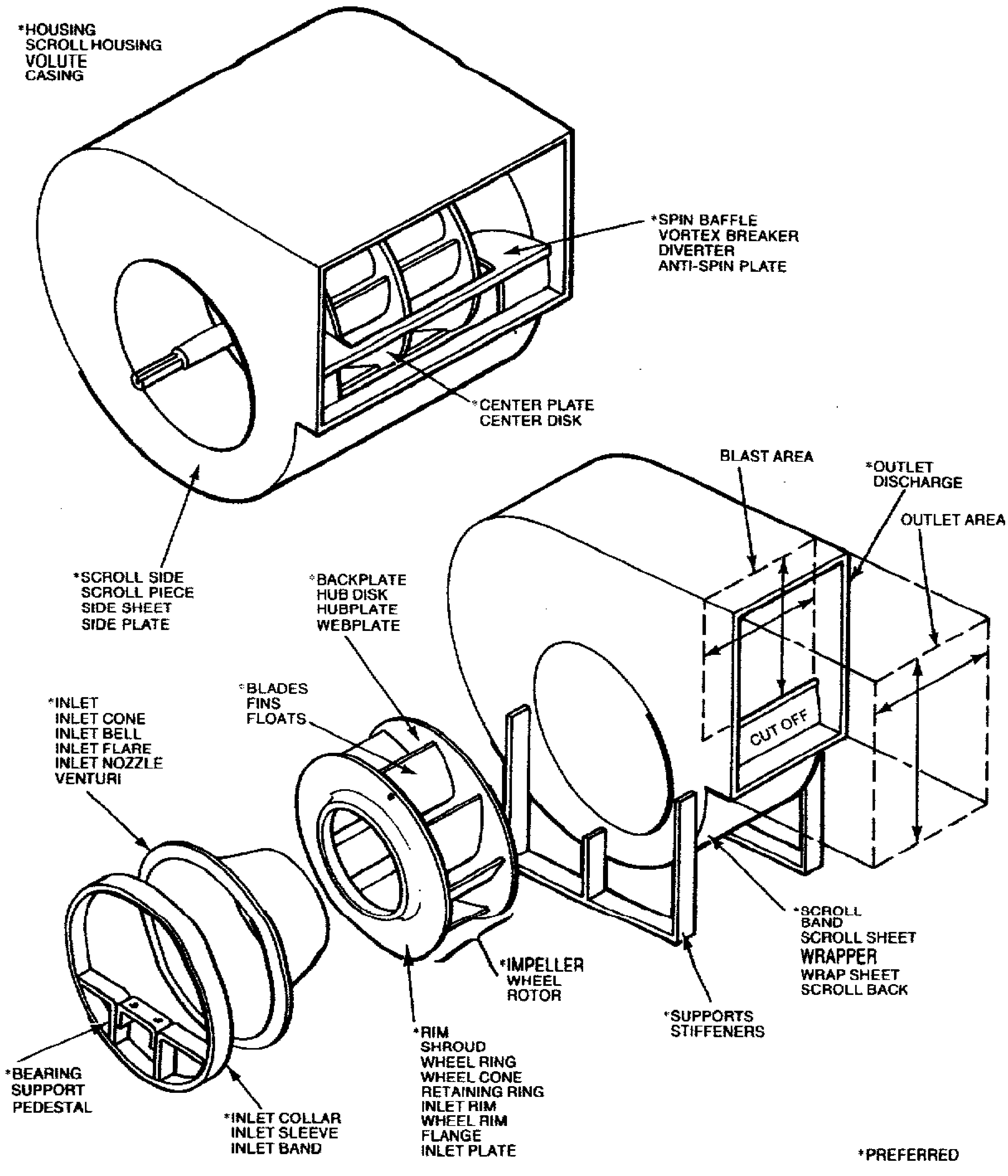


Figure 2-1. Components of Centrifugal Fan with Common Terminology. From AMCA Publication 201[2.1].

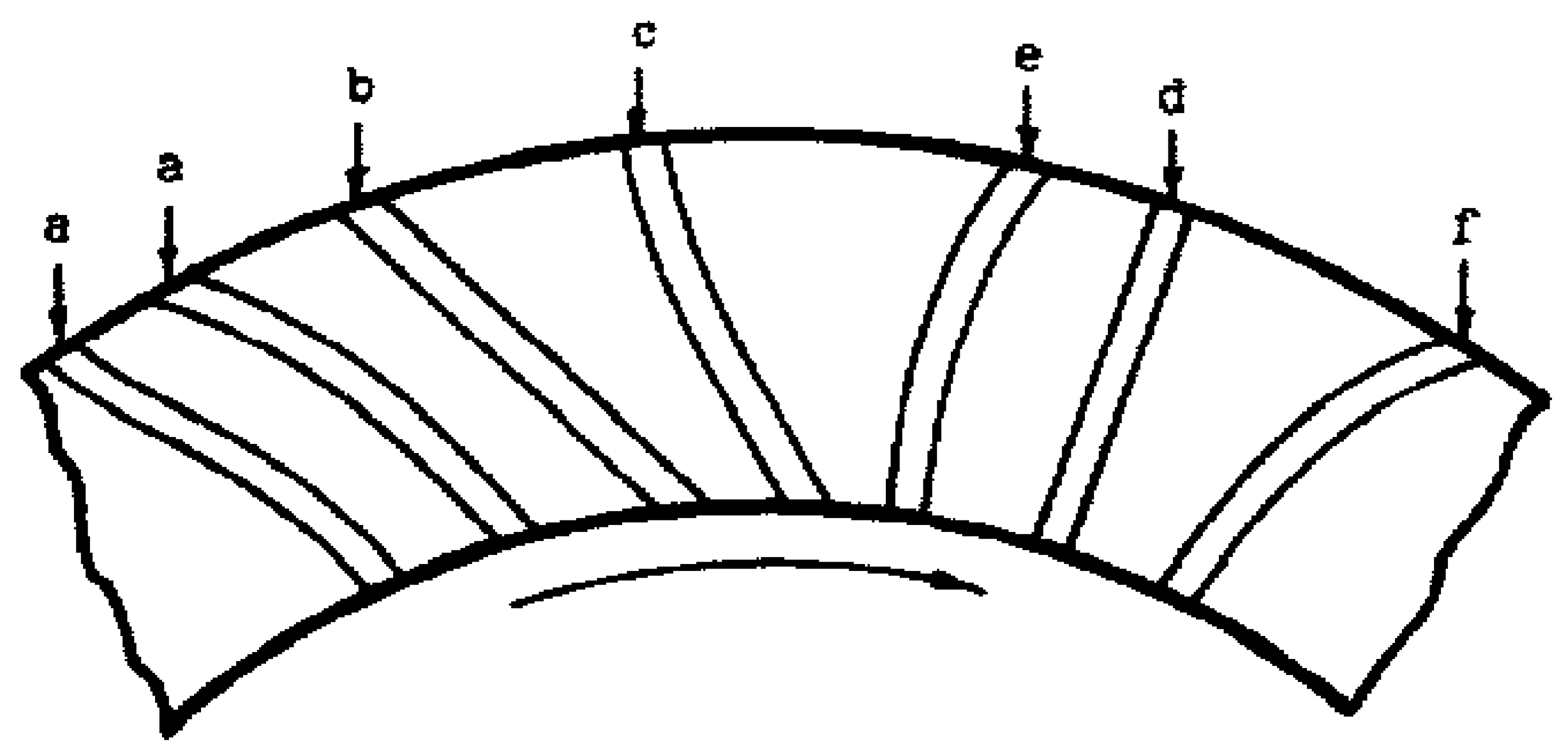
II. VARIOUS FAN CLASSIFICATIONS

Centrifugal fans may be divided into groups according to their blade shape. drive arrangement, type of inlet, type of discharge, and their pressure-flow range. These classifications are discussed in the following.

A. Blade Shape

The blade shapes of centrifugal fans may be divided into four general groups, as shown in Figure 2-2.

Figure 2-2. Centrifugal Fan Blade Shapes.



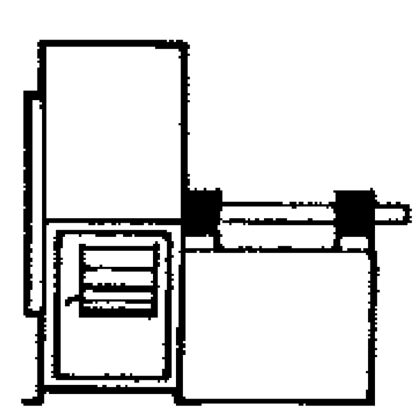
1. Backward-inclined (a) (b)
2. Backward-inclined radial-tip (c)
3. Radial (d)
4. Forward-inclined (e) (f)

For fan wheels of the same size and speed, the alphabetical sequence in Figure 2-2 parallels a trend toward increasing number of blades, inlet diameter, axial width, static pressure, capacity, turbulence, horsepower requirements, and decreasing stability and efficiency.

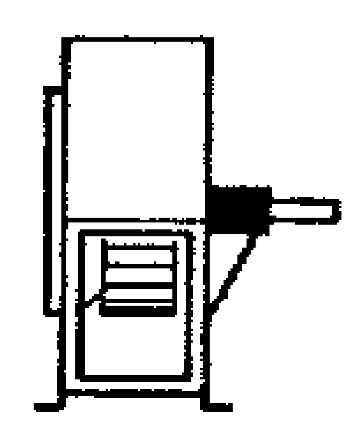
B. Drive Arrangement

Fan wheels are usually driven by electric motors because of the high efficiency and smooth operating characteristics of motors. Occasionally, however, they are driven by other means, such as gasoline or diesel engines or steam turbines. The fan wheel may be mounted directly on the driver or may be driven through a rigid, flexible, or fluid coupling or a belt drive. The direct drive is preferred whenever space and speed limitations permit. Air Moving and Conditioning Association (AMCA), in its Standard 2402, has given sixteen standard drive arrangements which are shown in Figure 2-3. These have been generally adopted by the manufacturers.

SW—Single Width DW—Double Width
SI —Single Inlet DI —Double Inlet
Arrangements 1, 3, 7 and 8 are also available with bearings mounted on pedestals or base set independent of the fan housing.

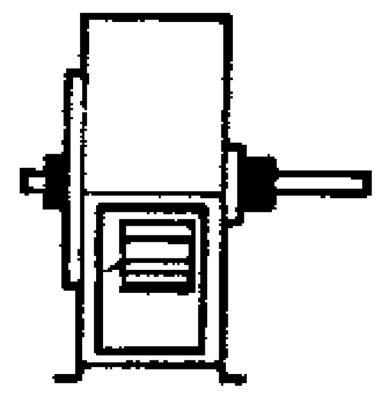


ARR. 1 SWSI For belt drive or direct connection. Impeller overhung. Two bearings on base.

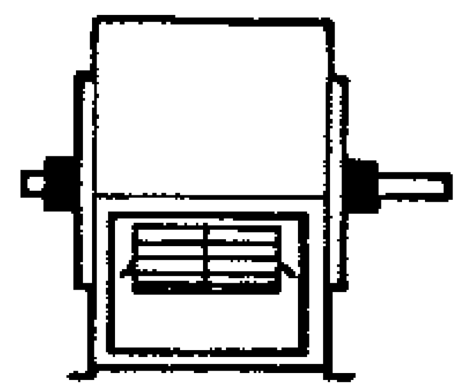


ARR. 2 SWSI For belt drive or direct connection. Impeller overhung. Bearings in bracket supported by fan housing.

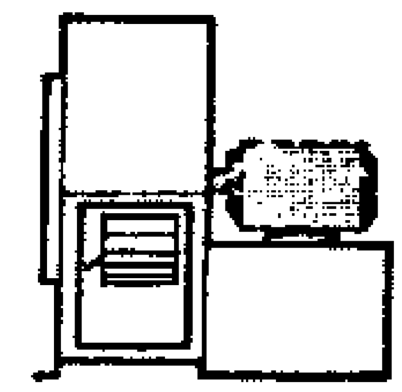
Figure 2-3a. Standard Drive Arragnements per AMCA Standard 2404-78.



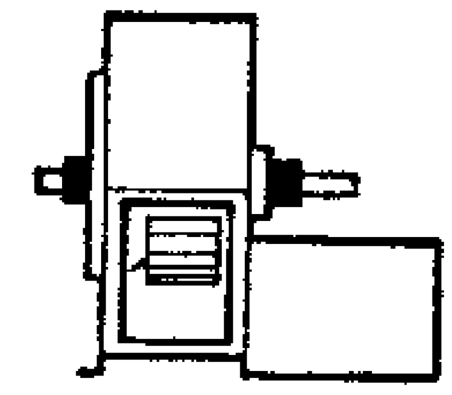
ARR. 3 SWSI For belt drive or direct connection. One bearing on each side and supported by fan housing.



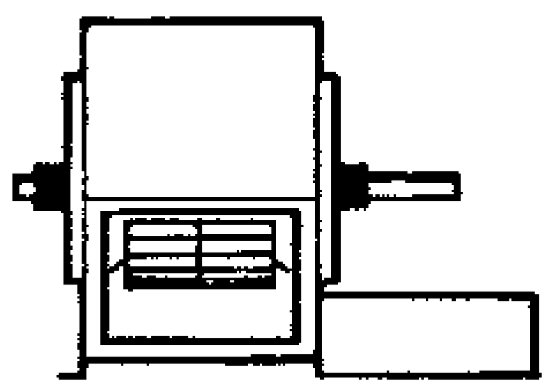
ARR. 3 DWDI For belt drive or direct connection. One Bearing on each side and supported by fan housing.



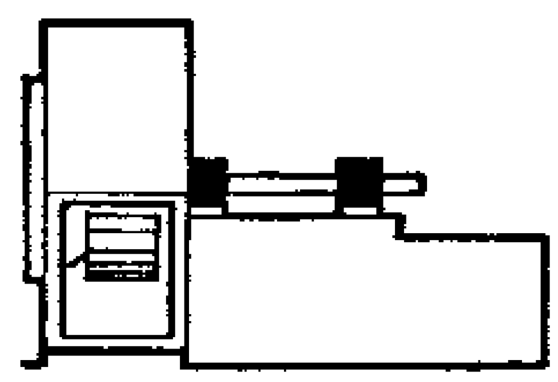
ARR. 4 SWSI For direct drive. Impeller overhung on prime mover shaft. No bearings on fan. Prime-mover base mounted or integrally directly connected.



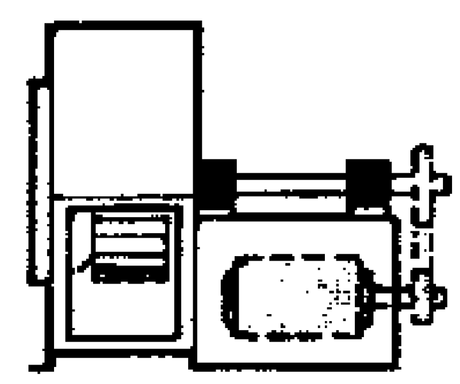
ARR. 7 SWSI For belt drive or direct connection. Arrangement 3 plus base for prime mover.



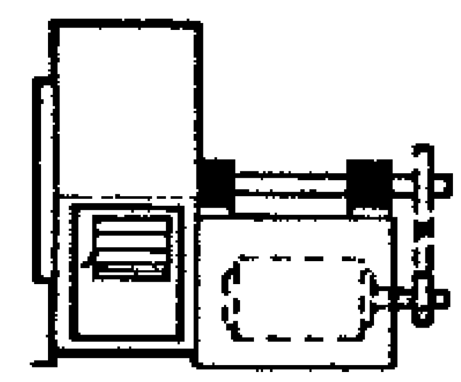
ASRR. 7 DWDI For belt drive or direct connection. Arrangement 3 plus base for prime mover.



ARR. 8 SWSI For belt drive or direct connection. Arrangement 1 plus extended base for prime mover.

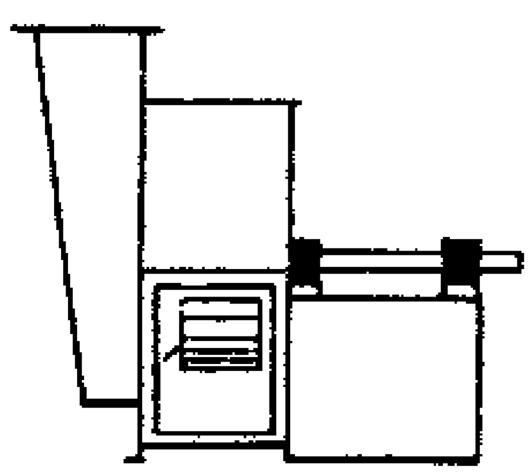


ARR. 9 SWSI For belt drive. Impeller overhung, two bearings, with prime mover outside base.

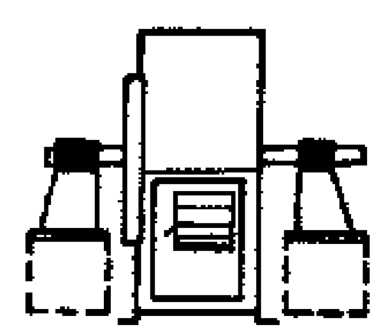


ARR. 10 SWSI For belt drive. Impeller overhung, two bearings, with prime mover inside base.

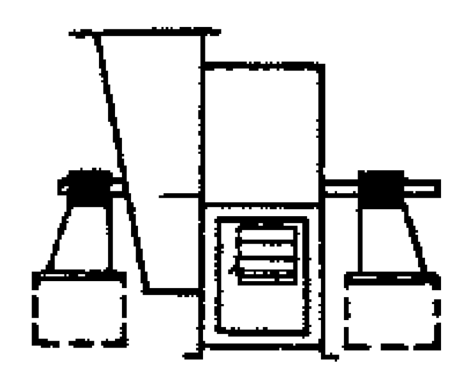
Figure 2-3a. Standard Drive Arragnements per AMCA Standard 2404-78. (Cont'd)



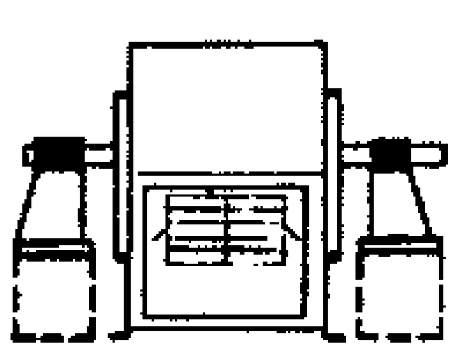
ARR. 1 SWSI WITH INLET BOX For belt drive or direct connection. Impeller overhung, two bearings, on base. Inlet box may be self-supporting.



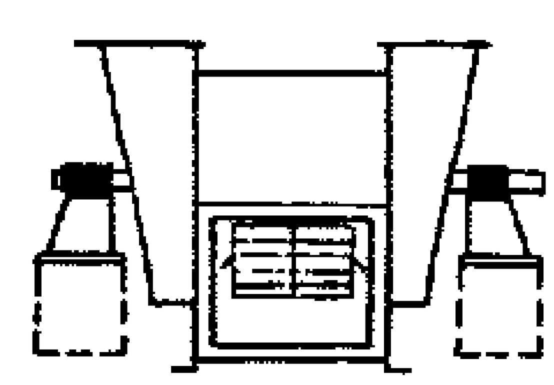
ARR. 3 SWSI WITH INDEPENDENT PEDESTAL For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals.



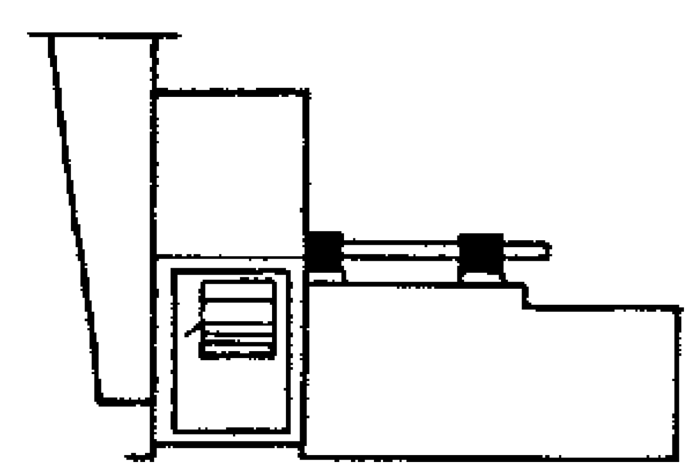
ARR. 3 SWSI WITH INLET BOX AND INDEPENDENT PEDESTAL For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals with shaft extending through inlet box.



ARR. 3 DWDI WITH INDEPENDENT PEDESTAL For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals.



ARR. 3 DWDI WITH INLET BOX AND INDEPENDENT PEDESTALS For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals with shaft extending through inlet box.



ARR. 3 SWSI WITH INLET BOX For belt drive or direct connection. Impeller overhung, two bearings on base plus extended base for prime mover. Inlet box may be self-supporting.

Figure 2-3b. Additional drive arrangements per AMCA Standard 2404-78.

C. Type of Inlet

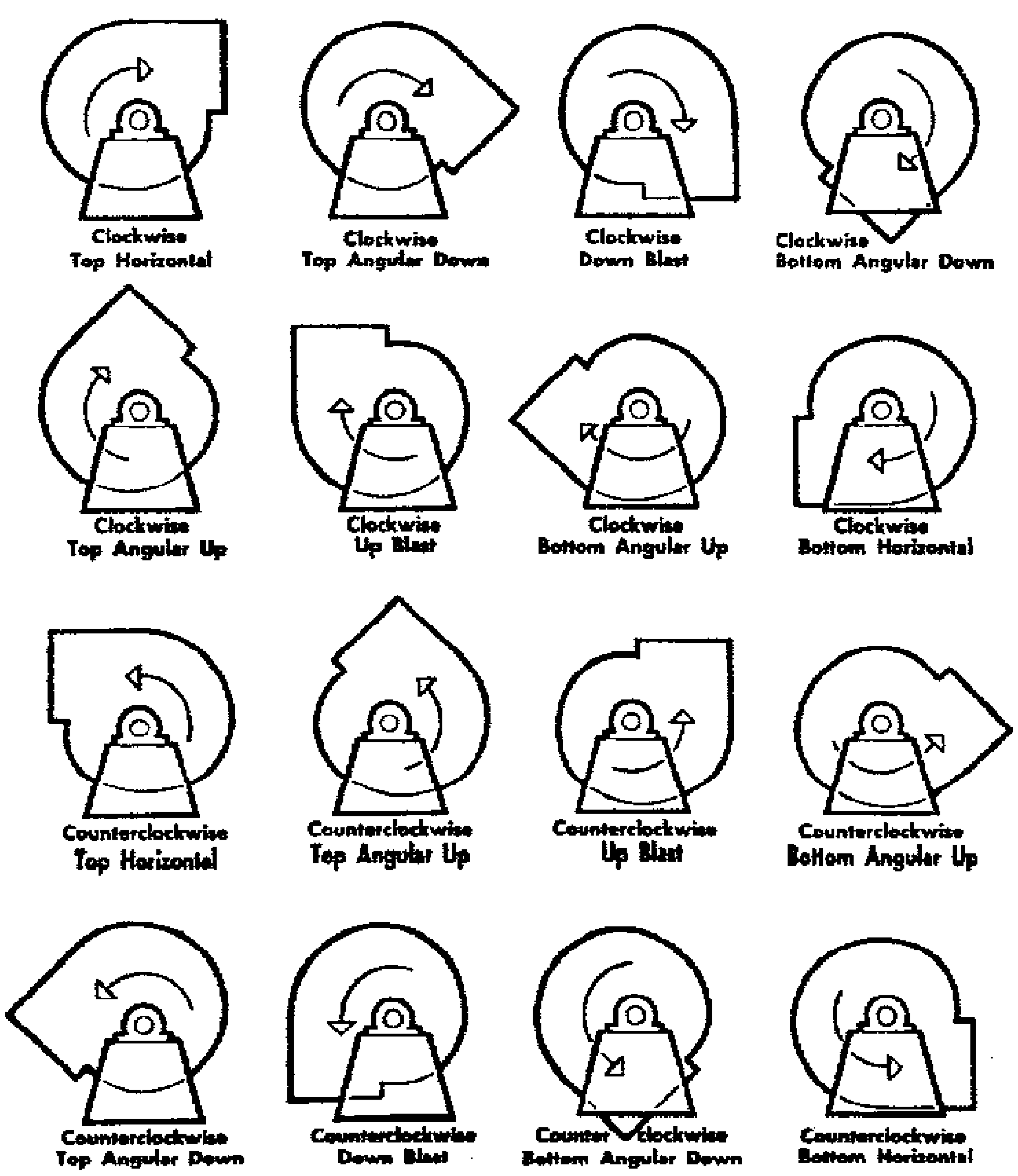
The three principal types of inlet are single-width, single-inlet; single-width, double-inlet; and double-width, double-inlet. The single-width, single-inlet construction is shown in Figure 1-1.

To obtain double-inlet construction with the same fan, the air is admittedd axially from both ends of the shaft, and a material improvement in static pressure capacity and efficiency is obtained. The double-width, double-inlet construction is obtained by placing two single-width, single-inlet fans back to back in a single casing.

D. Type of Discharge

With most centrifugal fans, a scroll casing is provided, as shown in Figure 1-1 of Section 409.1. Its main functions are, first, to collect the individual air streams from the blades and direct them to a single outlet and, second, to convert some of the velocity pressure of the gas leaving the blades into static pressure. The conversion is most effective with forward-inclined and radial blades since the gases leaving the impellers of these fans have relatively high velocities. The gases from backward-inclined blades, on the other hand, have relatively low velocities so that scrolls add little to their performance.

The demands of different installations may require different directions for the discharge opening relative to the inlet opening. Sixteen different directions have been standardized by AMCA, through its standard 2406, as shown in Figure 2-4.



Direction of rotation is determined from drive side of fan.

On single inlet fans, drive side is always condidered as the side opposite fan inlet.

On double inlet fans with drives on both sides, drive side is that with the higher powered drive unit.

Direction of discharge is determined in accordane with diagrams. Angle of discharge is referred to the horizontal axis of fan and designated in degrees above or below such standard reference axis.

For fan inverted for ceiling suspension, or side wall mounting, direction or rotation and discharge is determined when fan is resting on floor.

Figure 2-4. Designations for Rotation and Discharge of Centrifugal Fans According to AMCA Standard 2404-66.

In occasional installations the impeller will be encircled by a diffuser consisting of a series of curved stationary blades that convert some of the velocity pressure into static pressure. The cost and space requirements of such diffusers seldom justify their use.

For installations in which there is not room for a scroll or diffuser, or the gas is discharged into a large space, the scroll casing is frequently eliminated. This arrangement is usually called radial or circumferential discharge. Motor and generator cooling applications are usually of this type.

E. Classification According to Flow-Pressure Capability

AMCA Standard 2408 has divided each type of centrifugal fan into three classes according to their pressure-flow capabilities. Class I fans develop the least pressure while Class III fans develop the highest pressure. The pressure capability of Class II fans is intermediate between that of Class I and Class III fans. Figure 2-5 shows the classification for single width backward inclined and airfoil blades. AMCA 2408 also gives similar graphs for other designs of centrifugal fans.

These classifications are widely accepted and used in the industry.

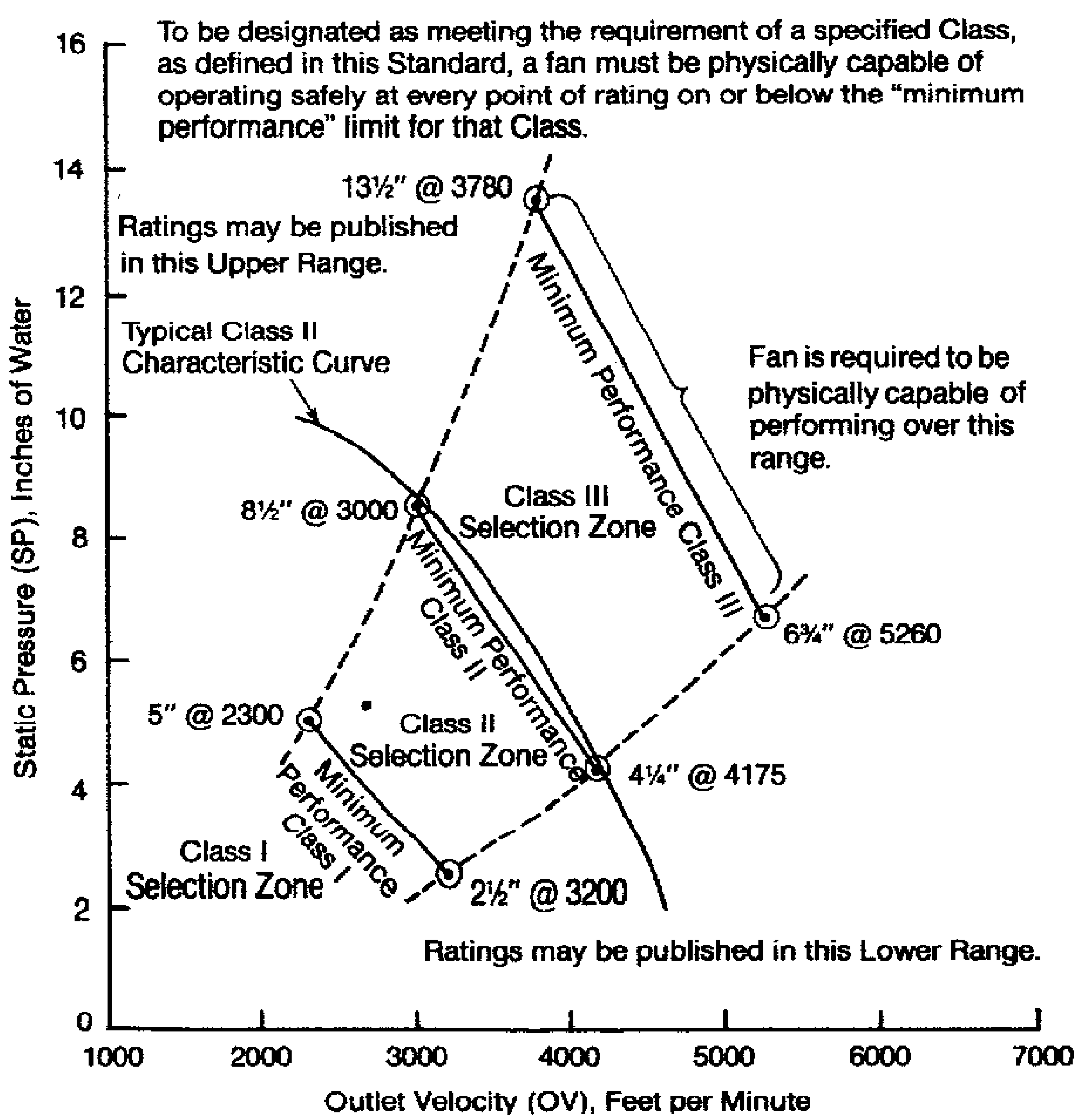
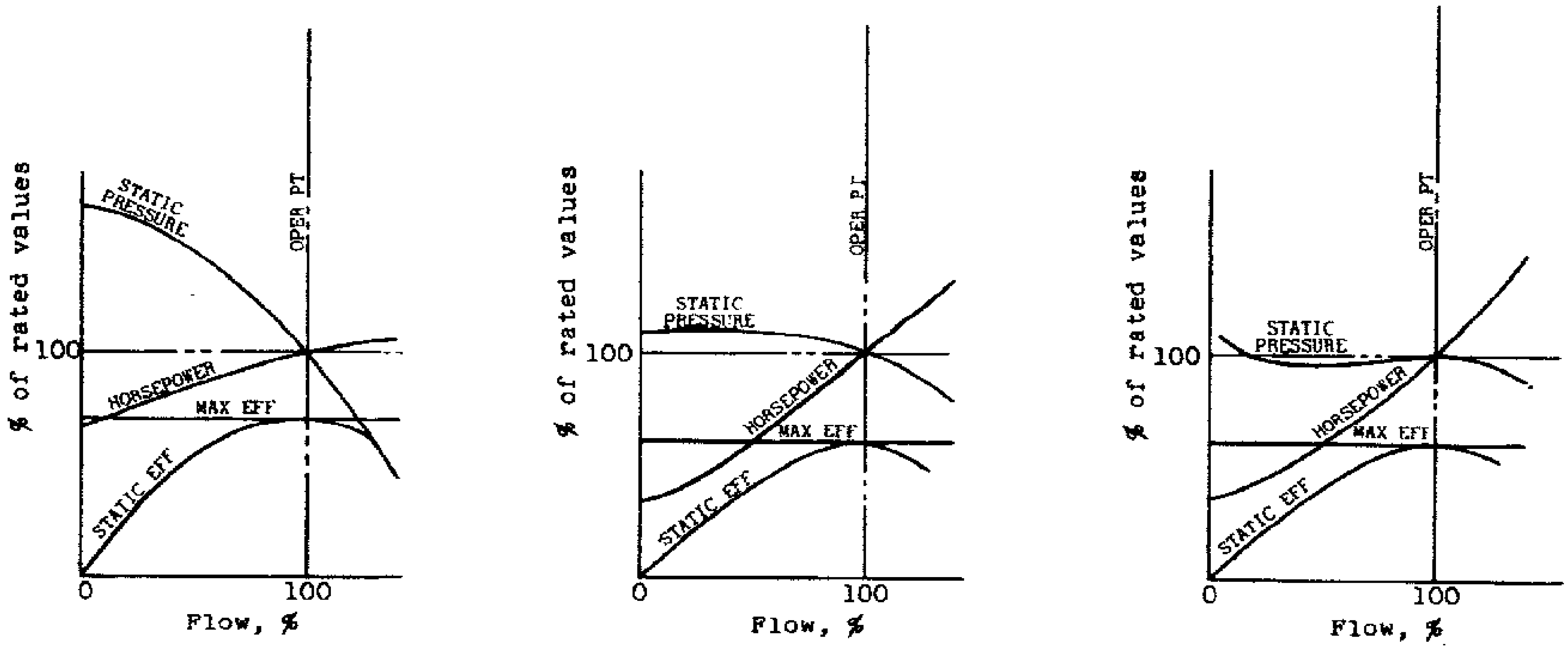


Figure 2-5. Classification of Single Width Backward-Inclined and Airfoil Blade Ventilating Centrifugal Fans According to AMCA Standard 2408.

III. PERFORMANCE CHARACTERISTICS

The relative performance of three main types of radial flow fans are shown in Figure 2-6. These curves do not show the behavior of any particular fan but rather the average characteristics of fans of these types.



(a) Backward Curved Blades (b) Radial Tip Blades (c) Forward Curved Blades

Figure 2-6. Characteristics of Radial-Flow Fans with Scroll.
(Caution: The rated pressure will vary with the type of blade, for the same size and speed)

On these curves are shown static pressure, static efficiency, and fan horsepower, plotted as a function of flow at constant speed. These quantities are plotted as percentages of their rated value (except efficiency, which is already dimensionless) in order to show the characteristics of a series of fans, rather than just those of the particular fan tested. These curves are discussed in the following. Also see [2.2].

A. Backward-Inclined Blades

1. Pressure

The pressure curve of Figure 2-6a shows that, starting with the no-delivery point at left with higher than design pressure, the curve passes through the design point (taken at maximum static efficiency) to the point of free delivery at far right. Aside from a slight rise that sometimes occurs near the point of no-delivery, the curve falls steadily without the dips that frequently occur in the curves of forward-inclined blades and axial flow fans. This pressure characteristic of backward-inclined blades results in a stability of performance that makes these blades (and radial-tip blades with similar characteristics best suited for fluctuating systems and parallel operation.

2. Fan Horsepower

The peak value of brake horsepower occurs in the operating range of the fan not far from the point of maximum efficiency. When the driving motor is selected with a power rating equal to this peak value, overloading of the motor is impossible. Thus, the backward-inclined radial fan is said to be of the non-overloading type, a very desirable characteristic.

3. Efficiency

The point of design is usually taken at the point of maximum static efficiency. Eighty percent static efficiencies are common in large, well-designed units. Efficiencies up to 91 percent have been achieved in large centrifugal fans with airfoil blades.

B. Forward-Inclined Blades

1. Pressure

The region of decreasing pressure with decreasing flow, apparent in the pressure curve of the forward-inclined blade fan (Figure 2-6c) may cause unstable operation when this type of fan is operated in parallel with similar fans or on fluctuating systems. Even in other applications, precautions must be taken so that the fan does not operate in the unstable range of the pressure curve.

2. Fan Horsepower

With increasing volume the fan horsepower increases rapidly, giving a maximum horsepower at maximum rate of flow. This maximum horsepower is frequently twice the horsepower at maximum efficiency. Since radial-flow fans are seldom operated at maximum flow (axial-flow fans operating more efficiently here), this means that power requirements are much larger beyond, than within, the operating range. Thus forward-bladed fans are said to have overloading characteristics, a great disadvantage for some applications.

With units having small horsepower requirements, a motor is selected capable of supplying the load at free delivery. With large units, however, this procedure is uneconomical and a motor is selected only slightly larger than needed to supply power requirements under operating conditions, provision being made in the installation design to prevent operation at too low static pressures.

Thus the operating range of the forward-inclined blade fan, limited at high flows by increasing power requirements and at low flows by unstable operation, is much narrower than that of the backward-blade fan.

3. Efficiency

The static efficiency curve has its maximum at the point of design. Maximum efficiencies for large well-designed fans with suitable scrolls may be as high as 77 percent as compared with 91 percent for backward-inclined blade fans. Efficiencies of forward-curved fans often range from 50 to 60 percent when operating at the design point.

C. Radial Blades

The performance characteristics of radial blade fans (Figure 2-6b) are, in general, intermediate between those of the backward-inclined and forward-inclined blade fans. With a properly designed inlet, the efficiencies can be about as high as for the forward-curved fan. However, the simple radial blade, with no curvature at inlet, which is used on electric machine applications for the same performance in either direction of rotation, has very low efficiency due to the shock loss at entrance.

IV. IMPELLER DESIGN

A. General

Some important factors in impeller design are briefly discussed in the following. However, the information given here is not comprehensive. Those interested in detailed information will find [1.1] particularly useful and it also contains extensive references to other sources of information. Also see [2.3] and [2.4].

B. Forward, Radial or Backward

Table 2-1 summarizes the main operating characteristics of the three major types of centrifugal fans. This table, together with the performance characteristics shown in Figure 2-3 and the discussion should give a clear indication of the fan which is best for a given application.

Table 2-1
Effect of Blade Shape on Centrifugal Fan Operation

Characteristic	Backward-Inclined	Radial	Forward-Inclined
Number of blades	Smallest	Medium	Largest
Size for same pressure and capacity	Largest	Medium	Smallest
Volume handled for same size and speed	Smallest	Medium	Highest
Range of produced volume of flow	Widest	Medium	Narrowest
Pressure for same size and speed	Lowest	Medium	Highest
Efficiency (1)	Highest	Medium	Lowest
Usual operating speed	Highest	Medium	Lowest
Overloading characteristics	Non-overloading	Usually non-overloading	Will overload under some conditions
Reversibility	Not reversible	Reversible (with straight blades)	Not reversible
Suitability for parallel operation	Best	Medium	Worst
Suitability for direct motor drive	Best	Medium	Worst
Self-cleaning characteristics	Nonself-cleaning	Self-cleaning	Nonself-cleaning
Cost for same impeller size	Higher	Lowest (with straight blades)	Higher

(1) Radial and forward-inclined have scrolls.

C. Blade Profile

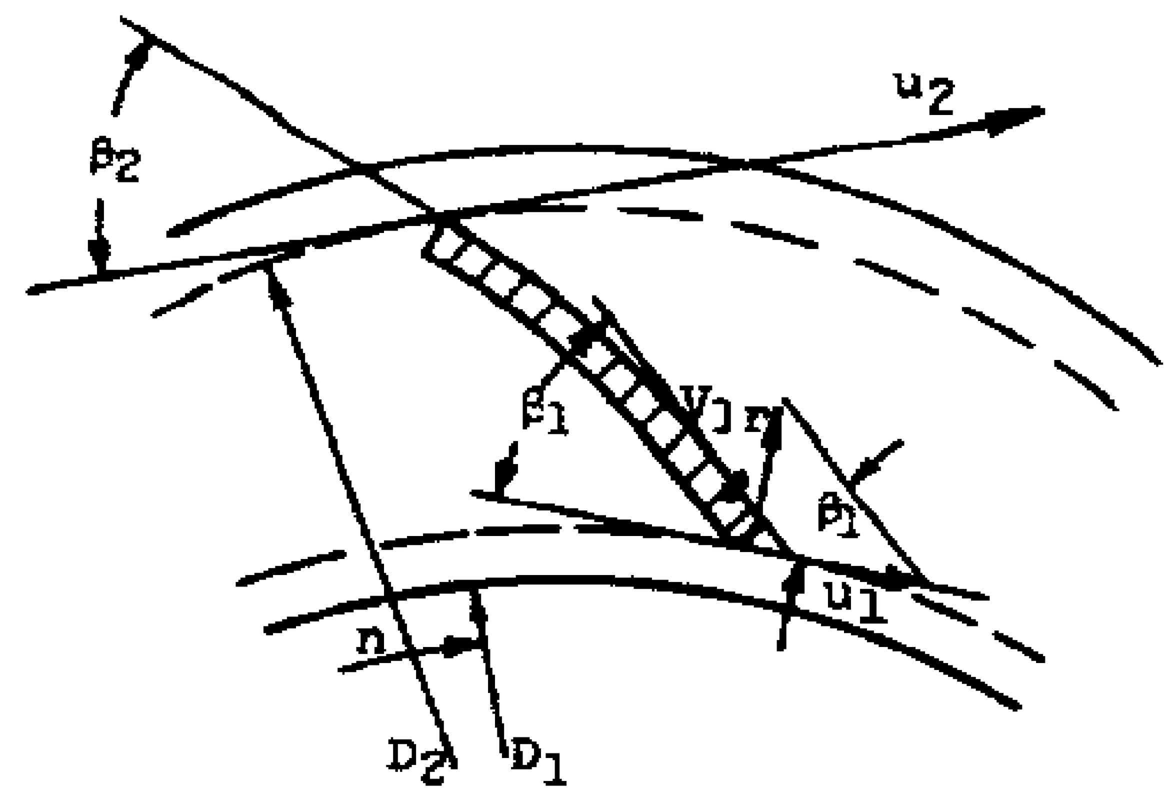
For forward-inclined and radial blades, only single-thickness profiles are used. For backward-inclined blades, both single-thickness and airfoil profiles are in use. Properly designed airfoil blades significantly reduce aerodynamic losses. Efficiencies up to 92% have been achieved in large fans using airfoil blades. However, airfoil blades are more expensive and may not always be economically justifiable in small fans where the gain in efficiency is generally modest.

D. Blade Inlet Angle

From Figure 2-7, it is clear that the angle of entrance of air is given by:

$$\tan(\beta_1^*) = \frac{V_{1r}}{u_1} \tag{2-1}$$

Figure 2-7. Blade Inlet and Outlet Angles and Fluid Entrance Angle.



The angle of incidence then is given by:

$$i = \beta_1 - \beta_1^* \tag{2-2}$$

According to [1.2], experiments have shown that optimum i for heavily backward curved blades is −3 to 5°. But for heavily forward-curved blades, the optimum value of i is much more than 0°.

A theoretical derivation by Eck [1.1] (page 93) shows that the optimum value of β_1 is 35.26° in the absence of pre-rotation. If air is pre-rotated in the direction of impeller, optimum β_1 increases. Pre-rotation in the reverse direction reduces β_1. Curvature of entrance reduces optimum β_1 below 32.5°.

E. Blade Exit Angle

The angle β_2 is the angle between the blade at exit and the tangent to the impeller circumference at this point. This angle has a profound effect on the pressure developed. Theoretical calculations on an ideal impeller show that the total head developed increases with increasing β_2, but at the same time the static pressure portion of total head decreases. Maximum total head is developed by extremely forward-curved blades with the static head being zero. Radial blades ($\beta_2 = 90°$) develop half of their total pressure in the form of static pressure. Figure 2-8 shows the variation of theoretical head with β_2.

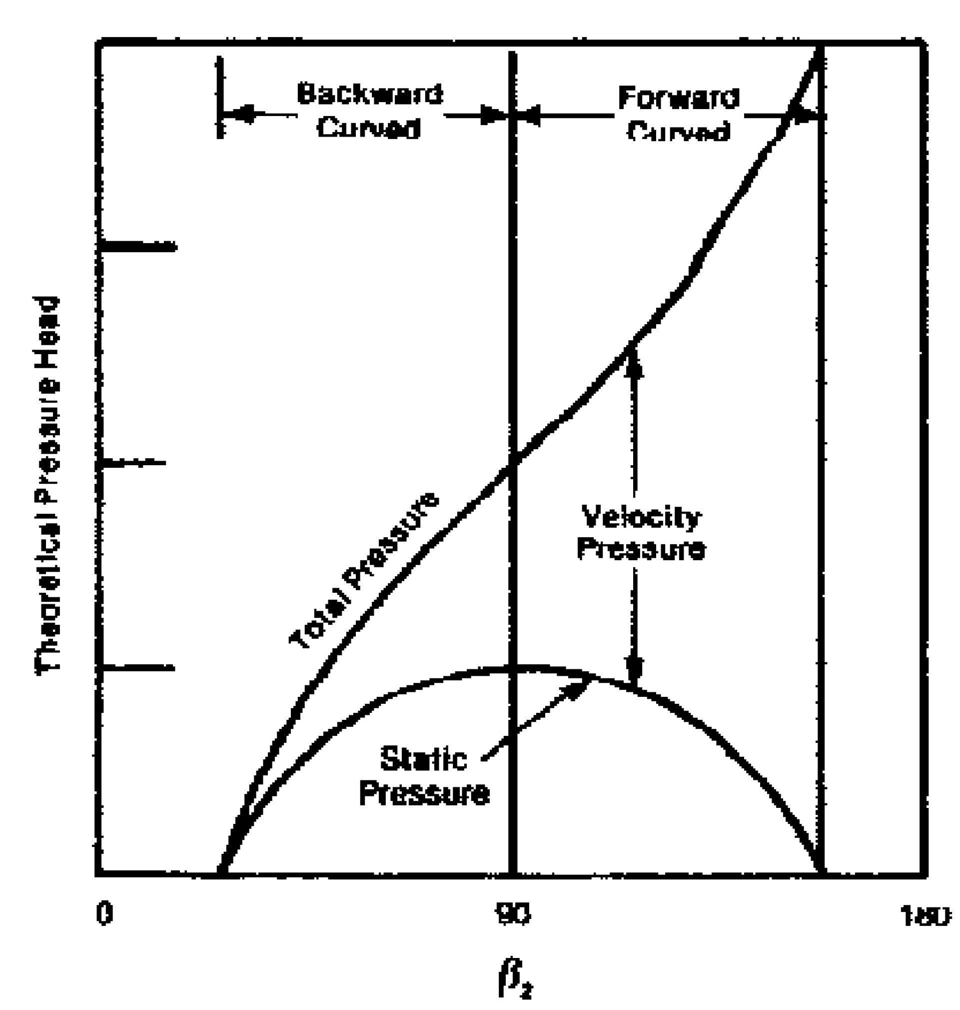


Figure 2-8. Effect of Blade Exit Angle on Theoretical Pressure Developed by Centrifugal Fans [1.2].

F. Number of Blades

The number of blades has two effects. If too many are used the surface friction loss increases and there is too much restriction of the inlet air flow to the blades. If too great spacing is allowed, losses occur due to vortex formation.

Eck [1.1] (page 51) has given an approximate formula for determining number of blades:

$$N = 8.5\frac{\sin\beta_2}{1 - D_1/D_2} \tag{2-3}$$

This formula is intended to give an indication of the number of blades needed for radial impellers, the author stating that the optimum number of blades can only be determined experimentally.

According to Cherkassky [1.2] (page 50), experiments have shown that the optimum number of blades is obtained by spacing them at about half the blade length.

G. Ratio of Inner Diameter to Outer Diameter

For a fan with given outer diameter and speed, higher pressures are produced by longer blades. However, as the inner diameter is reduced, the flow becomes restricted. Thus, the inner diameter should not be made so small that more head is lost at entrance due to the restriction of flow, than is gained by the increased blade length.

The effect of diameter ratio on efficiency is generally small. Good efficiency is generally obtained within $0.3 < D_1/D_2 < 0.8$. Best efficiency is generally obtained with $D_1/D_2 = 0.6$ to 0.7 [1.2] (page 51).

A theoretical deviation by Eck [1.1] (page 95), shows that the optimum value of D_1/D_2 is given by:

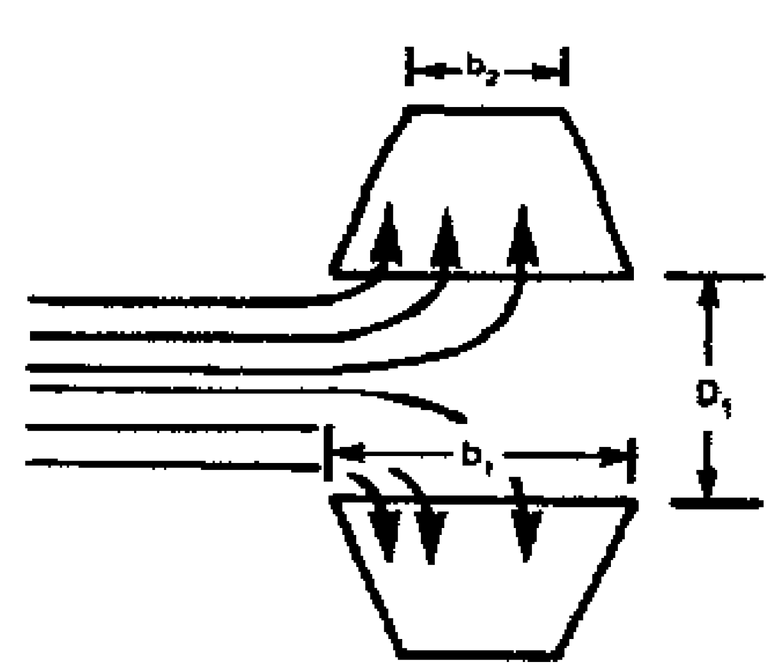
$$\frac{D_1}{D_2} = 1.194\,\phi^{1/3} \tag{2-4}$$

H. Blade Width

The width of blade at entrance (b_1) is usuually chosen such that the velocity upstream of the blades equals the velocity through the blades. The ratio of entrance area to the entrance area at the inner blade edge is (see Figure 2-9):

$$\frac{A_i}{A_1} = \frac{\pi D_i^2/4}{\pi D_1 b_1} = \frac{D_1}{4b_1} \tag{2-5}$$

Figure 2-9. Blade Width.



If $A_i = A_1$, it follows that:

$$b_1 = \frac{D_1}{4} \tag{2-6}$$

According to Cherkassky [1.2] (page 51) experience has shown that best results in some cases are obtained with somewhat larger values of b_1. He recommends that b_1 be obtained by the following formula:

$$b_1 = (1 \text{ to } 1.5)D_1/4 \tag{2-7}$$

The width of blades at exit, b_2, is usually determined by the following relation:

$$b_2 = b_1 D_1/D_2 \tag{2-8}$$

For ease of manufacture, $b_1 = b_2$ is sometimes used. This results in 2 to 3% loss in efficiency [1.2] (page 51).

V. FAN INLET DESIGN

Conditions at the inlet to the fan are very important for satisfactory operation. If careful attention to streamlining is neglected, poor efficiency and noisy operation will result. The addition of an inlet orifice, properly dimensioned to direct the flow smoothly into the fan, greatly improves performance.

The design of inlets may be as important as the design of the fan itself. This is especially true in the design of a fan by scaling up or down from a similar fan. Inlet conditions should be duplicated, in the chosen scale ratio, as closely as possible.

Proper design of inlets leads to not only good performance characteristics, but also minimum noise level. Inlet design should be chosen both to minimize resistance to flow and to improve the flow lines so that eddies (flow separation from the surface) and entrance shock do not occur.

Inlets and inlet conditions not purposely constructed to provide good flow are so varied in character that very little can be said about them even qualitatively. Each individual case deserves careful consideration. Smoke tests are useful in this connection.

A. Shape of Inlets

The type of fan blade and shape of inlet to be used for a given application will depend upon the relative importance of high efficiency and first cost for that application. For instance, optimum performance can be obtained by using a blade shape as shown in Figure 2-10 (e); however, cost considerations may dictate the use of an inlet as shown in Figure 2-10(c), 2-10(d), or even 2-10(b). An inlet as shown in Figure 2-10(a) should be avoided since poor efficiency and noise will result.

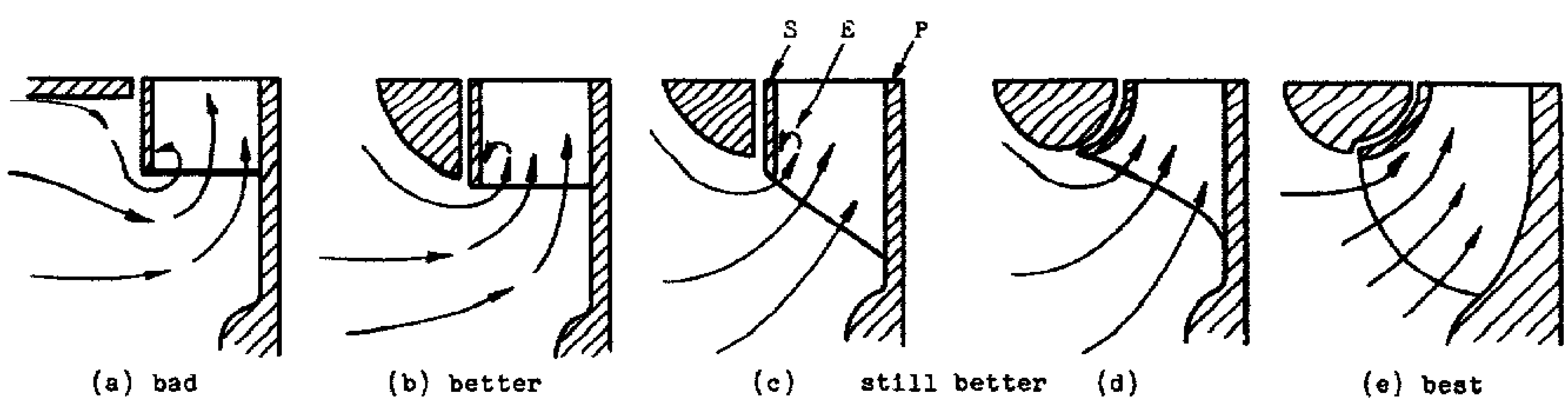


Figure 2-10.

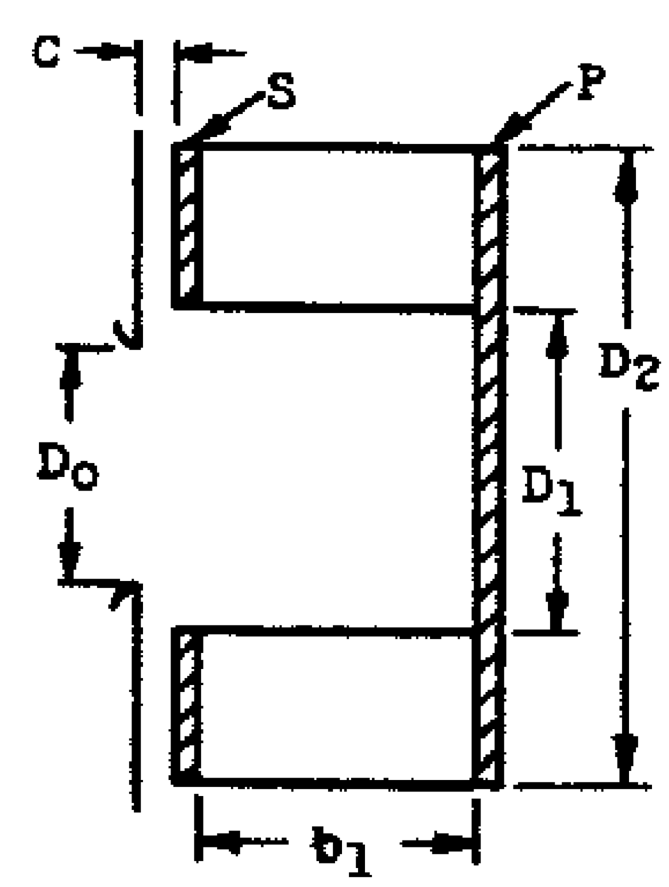
The use of a beveled blade shape, as shown in Figure 2-10(c), compensates for the axial momentum of the air entering the fan which tends to carry the air toward the backplate, P. The increased depth of this blade also increases the developed pressure. Eddies, E, may still exist near the shroud, S, however, unless the inlet surface is faired smoothly to a radial direction, as shown in Figure 2-10(d). (Ref [2.5], page 23). Blade shape, 2-10(d), is highly desirable because the flow changes gradually from axial to radial.

B. Size of Inlet

The area of flow at entrances should not be made so small that appreciable head is lost in this region. Sharp bends and restrictions should be avoided.

Design of inlet orifices depends upon the D_0/D_2 ratio of the blades and upon the blade length b_1. In any case, it is highly desirable that $(\pi D_0^2/4)$ be greater than $(\pi D_1 b_1)$ [2.6]. See Figure 2-11 for notation. This prevents flow separation from shroud, S, and crowding of the flow toward the backplate, P. This requirement results in narrow blades, i.e., small b_1, but the volume output per inch of blade length is high.

Figure 2-11.



For the case where $D_0 = D_1$ the requirement becomes: $D_1 > 4b_1$.

Less conservative ratios are recommended ([2.6] page 70-75) as follows:

	Backward turned blades	Radial blades	Forward turned blades
D_0/D_2	0.60-0.75 large volume fans 0.2-0.5 pressure fans	0.70-0.80 mechanical draft fans 0.2-0.5 pressure blowers	0.80-0.95
Recommended b_1/D_0	0.40	–	–
Upper limit b_1/D_0	0.55 large volume fans	0.55	0.65

C. Clearance

Clearance between inlet orifice and the impeller shroud (C in Figure 2-11) should be as small as manufacturing methods will allow, as this clearance causes leakage of discharge air back to the inlet. This results in loss of volumetric efficiency. A clearance of 0.5% of fan diameter is easily achievable in practice. The higher the fan pressure head, the greater is the importance of minimizing clearance. Eck recommends that for D_1/D_2 of 0.6 or less, use of a labyrinth seal should be considered (Ref. 1, page 82).

In fans with forward-curved blades, one may keep a large clearance to provide recirculation which will improve stability during discharge throttling. However this will cause deterioration in efficiency.

D. Inlet Control of Velocity and Direction

Twisted radial turning vanes (stationary) located close to the fan in the inlet stream have been used to give the air an introductory rotation in the direction of blade rotation prior to entrance into the blade. Shock of the inlet edges is thereby reduced, resulting in a net increase in efficiency, and general performance is improved.

If the inlet guide vanes are turned backward (against rotation) more energy is imparted by the impeller to the fluid and an increase in pressure or capacity or both may result, but at a loss in efficiency. Inlet flow conditions may be predetermined by the use of adjustable inlet guide vanes which can be adjusted to give high efficiency over a wide range of flow. If the entering air has a component of rotation in the direclion of fan rotation, it will decrease the relative velocity of the air and fan blades. If it is rotating against the fan, it increases the relative velocity. For this reason, the operating characteristics of a fan rotating at a given speed can be varied widely by giving the entering air a component of rotation. Variable inlet guide vanes (VIV) are discussed in Section 409.9.

Badly designed inlet duct connections can cause deterioration in fan performance due to system effects (Section 409.4).

VI. FAN SCROLLS

A. Purpose of Scrolls

The fan casing serves two purposes. First, it collects the air discharged by the impeller and delivers it to one or more outlets as needed. Second, it converts the velocity pressure developed by the impeller to static pressure. As is seen in Figure 2-8, a significant portion of the total pressure developed by most impellers is in the form of velocity pressure. Hence this conversion function is of much importance, especially for forward-curved blade impellers. The collection of air and conversion from velocity to static pressure should be carried out with minimum losses.

B. Types of Scrolls

The casing most commonly used is involute shaped with straight sides. See Figure 2-12(a). The annular gap between the impeller and casing increases as the casing exit is approached. Usually, the casing has a single outlet. However, multiple outlets are sometimes used if supply to more than one duct is needed. Figure 2-12(b) shows a casing with two outlets. Casings with multiple outlets mainfolded together to a single outlet have been found to improve efficiency and are sometimes used (Ref. [1.1] page 203).

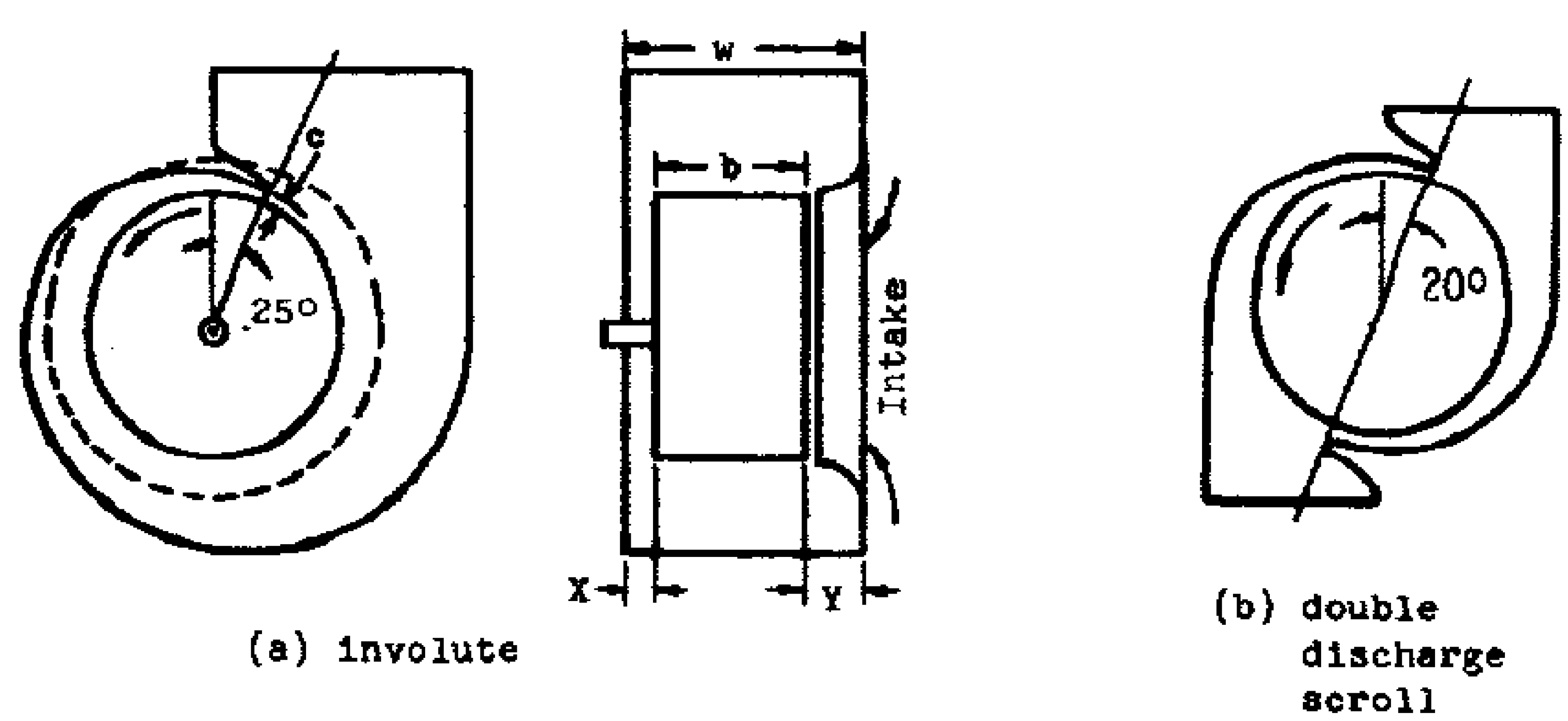
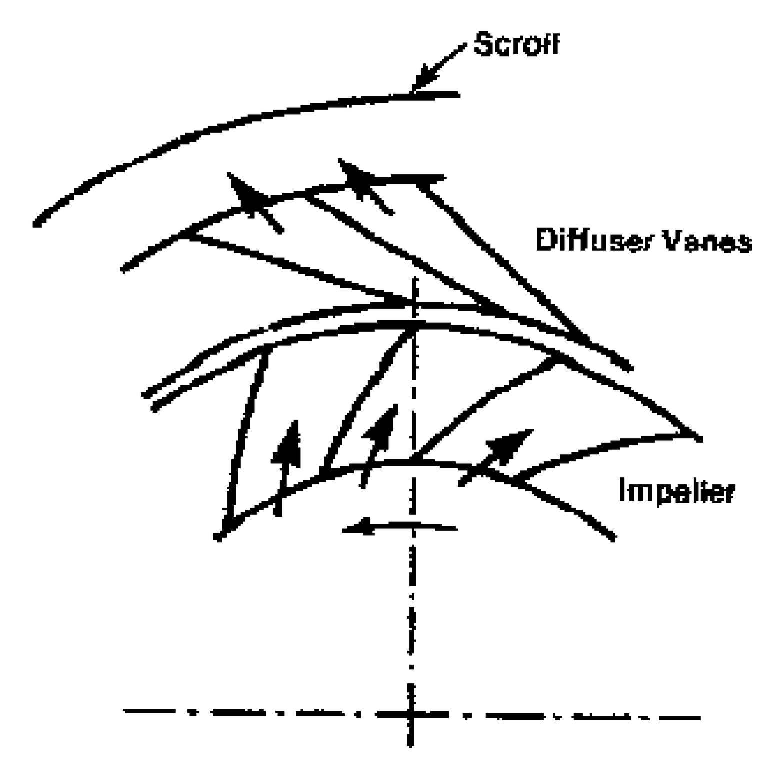


Figure 2-12.

Some fan casings include diffusion vanes (guide vanes) around the impeller as shown in Figure 2-13. Air discharged by the impeller expands as it flows through the diffusion vanes, thus converting kinetic energy to potential energy. For efficient conversion, the vanes should ideally have the shape of a logarithamic spiral. However, straight vanes are usually used because they are easier to fabricate (Ref. 1.1 page 187). Diffusion vanes are rarely used in single stage fans.

Figure 2-13. Fan with Diffuser Vanes.



C. Design of Involute Casings for Scrolls

The periphery of a scroll is usually an involute curve generated from a circle. An involute is the curve generated by the end of a string (b-c in Figure 2-14) which is kept taut while being unwound from a circle, called the generating circle (radius a in Figure 2-14). The equations of the curve are:

$$x = a(\cos\theta + \theta\sin\theta) \tag{2-9}$$

$$y = a(\sin\theta - \theta\cos\theta) \tag{2-10}$$

where θ = the angle, measured from the point of intersection of the involute and the generating circle radius, in radians
a = the radius of the generating circle

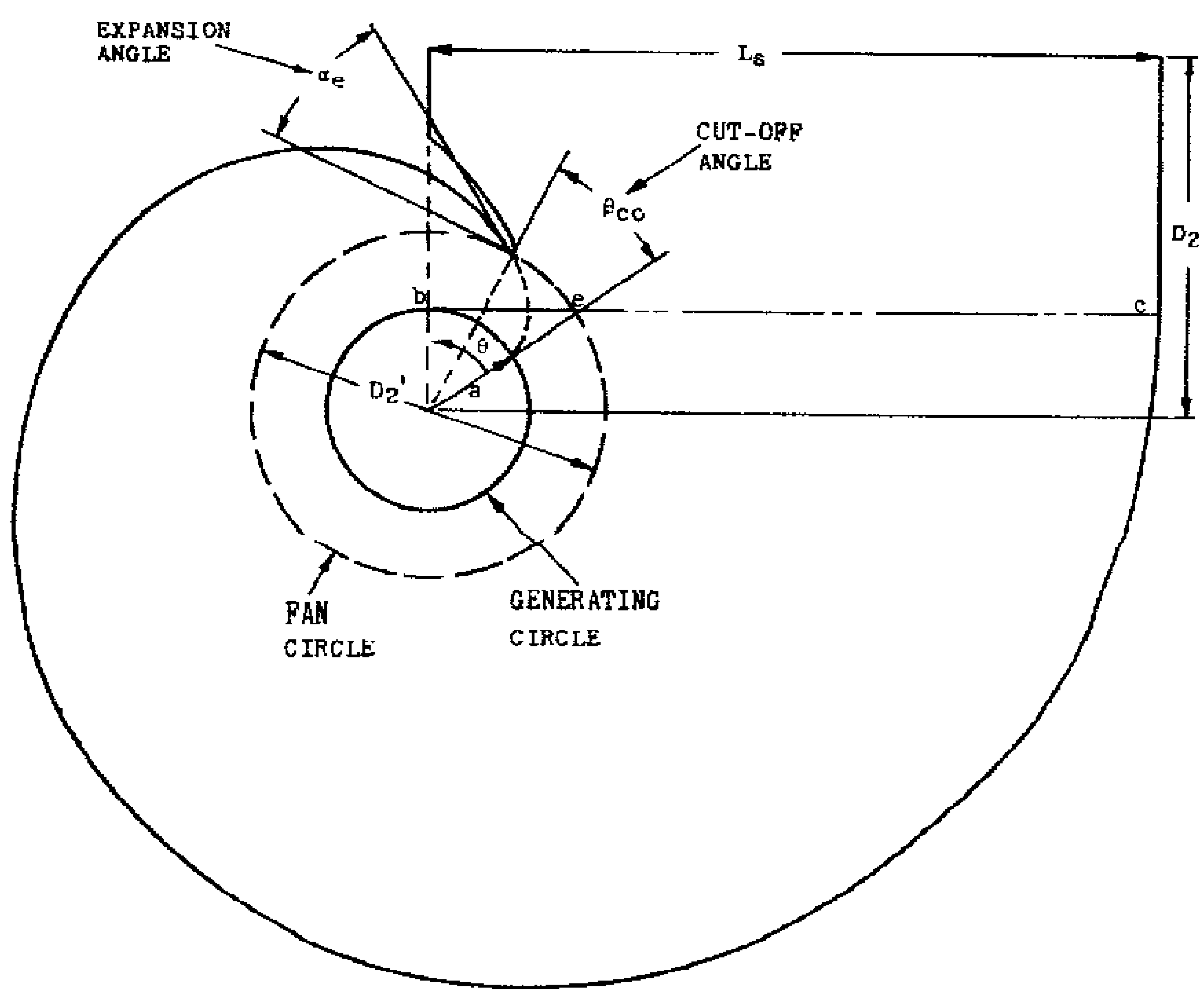


Figure 2-14. Design of Involute Casing.

These equations permit a point by point calculation of the involute when the radius of the generating circle, a, is known. The size and shape of a fan scroll depend upon the radius of the base circle, a, the diameter of the fan impeller plus clearance (fan circle), D_2', (clearance requirements are discussed later), the width of the scroll opening, L_s, the expansion angle, α_e (see section 406), and the cutoff angle, β_{co}. These quantities are all shown in Figure 2-14.

The relationships between the principal parameters of the scroll are:

$$a = (D_2'/2)\sin \alpha_e \tag{2-11}$$

$$L_s = a\,(2\pi - \beta_{co} + \text{ctn}\,\alpha_e) = \text{scroll opening} \tag{2-12}$$

The construction of an involute may be accomplished as follows. The radius of the generating circle, a, is determined by the expansion angle, α_e, and the diameter of the fan plus clearance, D_2' (see equations). The involute can then be constructed about the generating circle by calculation of equations (2-9) and (2-10). The cutoff angle, β_{co}, is measured from $\theta = 0$ to a radial line drawn through the intersection of the involute and the fan circle. A line drawn through the intersection, e, of the fan circle and the radius at $r = 0$, and tangent to the generating circle, a, intersects the involute at the termination point, c, of the involute. This line determines, L_s, the scroll length at the exit.

The straight portion of the exhaust should extend a length at least equal to the fan diameter, D_2, as shown in Figure 2-14 for best performance [2.7]. Experience has shown, however, that this extension can be materially reduced without serious loss in performance.

1. Clearance and Cutoff Design

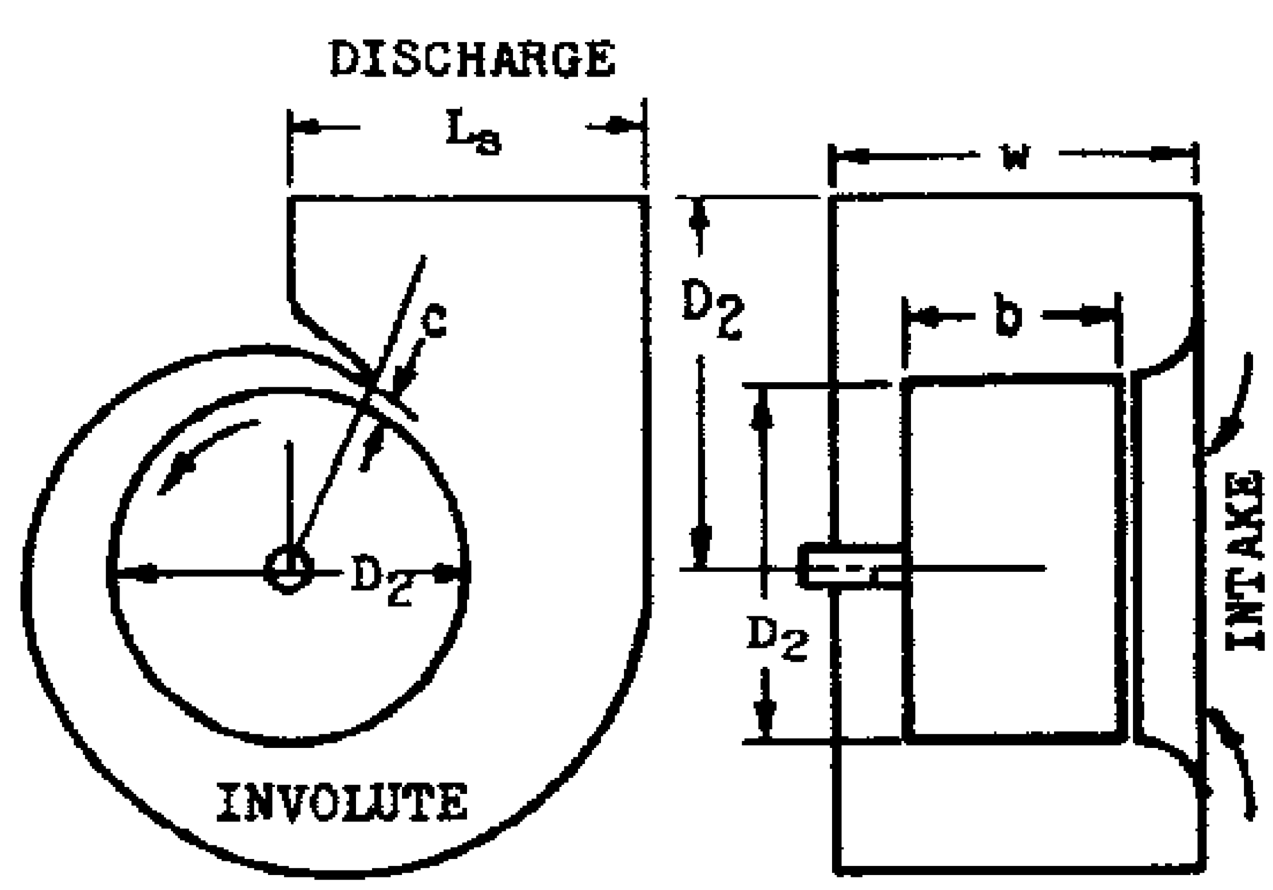
The amount of clearance over the cutoff is about 20% of the impeller diameter. Clearance as small as 5% may be used (Ref. [1.3] page 211). If the cutoff lip is too close to the impeller, noisy operation will result.

2. Casing Width and Impeller Location

The casing width, w, should be approximately 1.6 the blade width, b, except for backward leaning blades where a ratio of as high as 2.2 has been found best. If w is less than this value, the fan capacity drops below its optimum value.

The fan should be located closer to the backplate than to the inlet opening as shown in Figure 2- 15. By this arrangement better performance due to a better distribution of the air over the axial length of the blade is obtained. (Ref. [2.7] page 13 and Ref. [2.8] page 144.)

Figure 2-15. Casing Width.



3. Space Limitations

Experience has shown that small deviations from the true involute shape which might be necessary in order to fit the scroll in the space available will not have any noticeable effect on fan performance.

4. Performance

The effect of this type of scroll on the performance of a single radial tip impeller is shown in Figure 2-16. The curve shows the unfavorable effect on performance of an excessive reduction in outlet opening whether by reduction in α_e, β or w or all three combined.

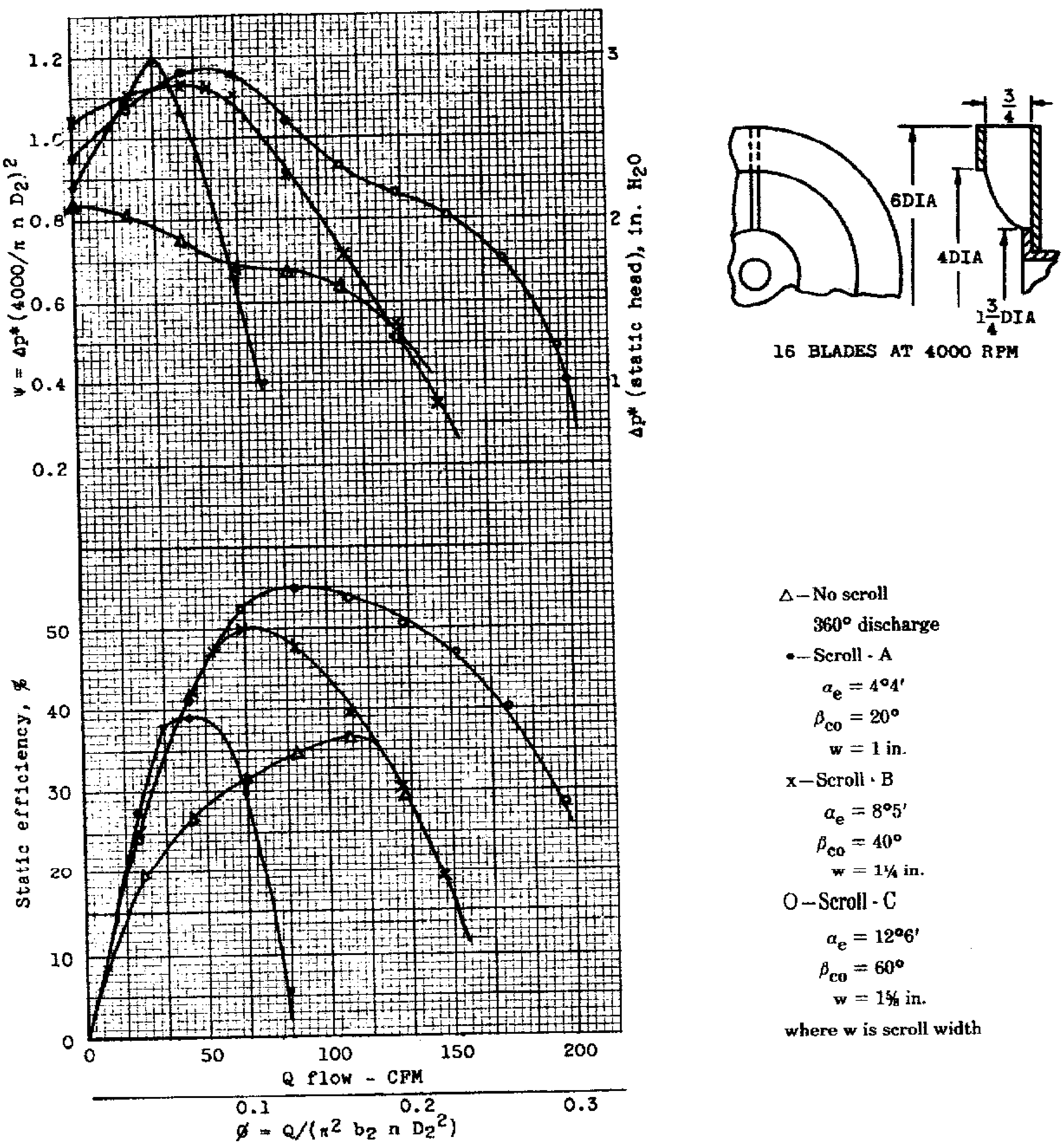
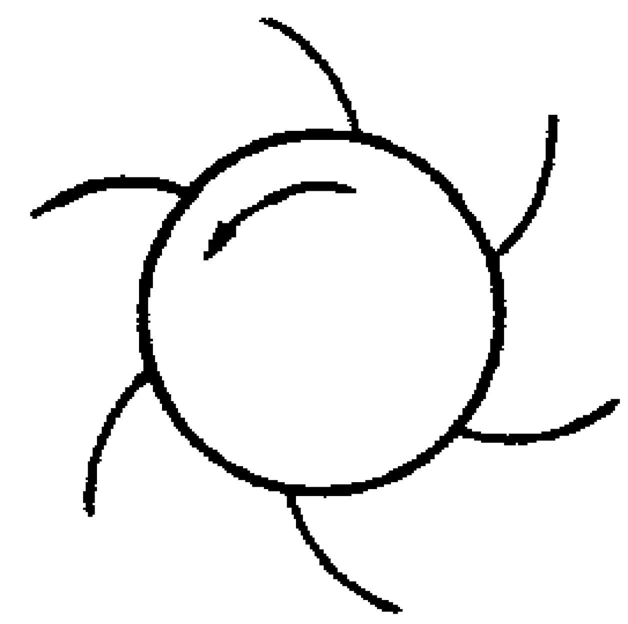


Figure 2-16. Effect of Scroll on Radial Fan Performance.

VII. DIFFUSERS FOR FANS WITHOUT SCROLL

A multivane diffuser, as shown in Figure 2-17 is used if it is desired to effect recovery of velocity head but it is not desired to direct the air leaving the blades in only one direction. Where direction of the air is desired, scrolls are used; they are discussed later in this section.

Figure 2-17. Multivane Diffuser.



Equations for design of multivane diffusers (Figure 2-18) are as follows: (Development of these equations may be found in Ref. [2.7].

$$A_d = A_F \frac{\phi u_2}{V_{t_3}} \tag{2-13}$$

and

$$A_{d_N} = \frac{A_d}{N_d} \tag{2-14}$$

where A_d = cross sectional area, measured perpendicular to the direction of relative flow between the diffuser blades at the diffuser exit
A_F = fan exit area, $\pi D_2 b$, where b = blade width
A_{d_N} = discharge area between each pair of blades
V_{t_3} = total flow velocity of air leaving the diffuser perpendicular to A_d
u_2 = tangential component of velocity of air leaving fan
ϕ = flow coefficient
N_d = number of diffuser vanes

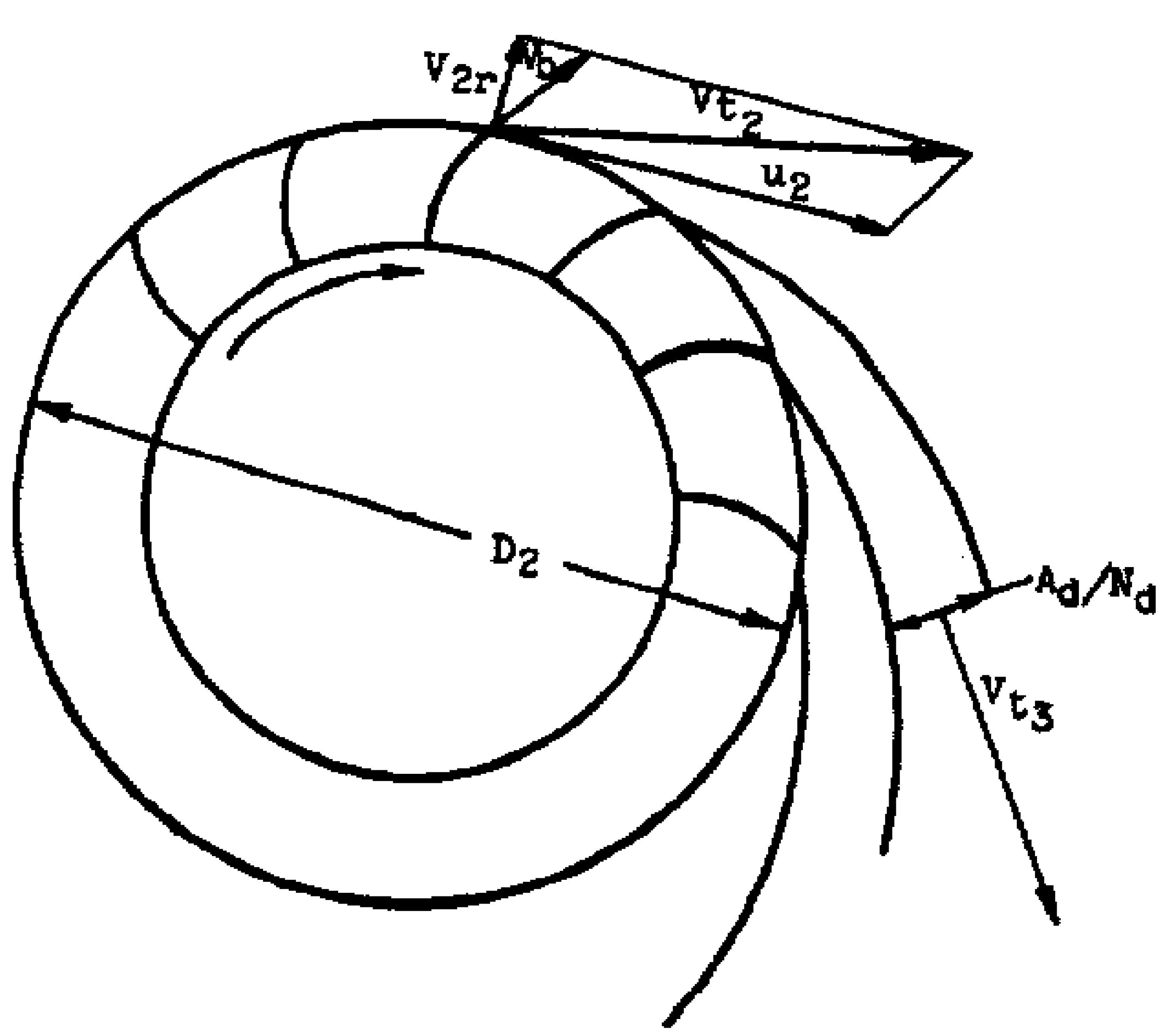


Figure 2-18.

In many cases the radial component of velocity, V_{2r}, of air leaving the impeller is small enough compared to the impeller tangential velocity, u_2, so that V_{t_2} and u_2 are nearly equal. In such cases, a velocity ratio of $V_{t_3}/V_{t_2} = 0.70$ is a typical choice so that:

$V_{t_3} = 0.70\ V_{t_2} = 0.70\ u_2$, and substituting in equation (2-13)

$$A_d = 1.43\,\phi\,A_F \tag{2-15}$$

The shape of the diffuser blades is an involute curve about the fan center. For six or more blades a fixed radius curve is an adequate approximation. The blade angle at the diffuser should be parallel to the velocity vector V_{t_2}; it therefore depends on the flow (or the flow coefficient, ϕ) expected, which in turn is affected by the flow resistance of the systems in which the fan operates.

VIII. FANS WITHOUT SCROLLS OR DIFFUSERS AS USED IN SMALL MOTORS

A. Blade Width and Shroud Depth Interdependency

Blade axial width, b, and shroud radial depth are interdependent and must therefore be considered together. There is an optimum fan blade depth and shroud depth for each load restriction. There is also an optimum blade width, b, and blade inlet area $A_1 = (\pi b D_1)$ for each shroud depth. If b and A_1 are too small compared to the shroud inlet area for axial flow to the impeller they will offer excessive restriction and use up an excessive amount of the static pressure produced by the fan. If b and A_1 (or more specifically, D_1) are too large compared with diameter at the shroud inlet, the radial component of flow at the blade inlet periphery will be too small and excessive impact loss will result [2.9].

Figure 2-19(a) shows the relative proportions for high flow, low pressure work. Figure 2-19(b) shows the relative proportions for medium flow, medium pressure work. Figure 2-19(c) shows the relative proportions for high pressure, low flow work [2.9].

Where the blades extend inward nearly to the center of the fan Figure 2-19(c), shroud depth rather than blade depth is the important dimension to consider. Its effect is illustrated, for narrow blade fans, in Figure 2-20 [2.9].

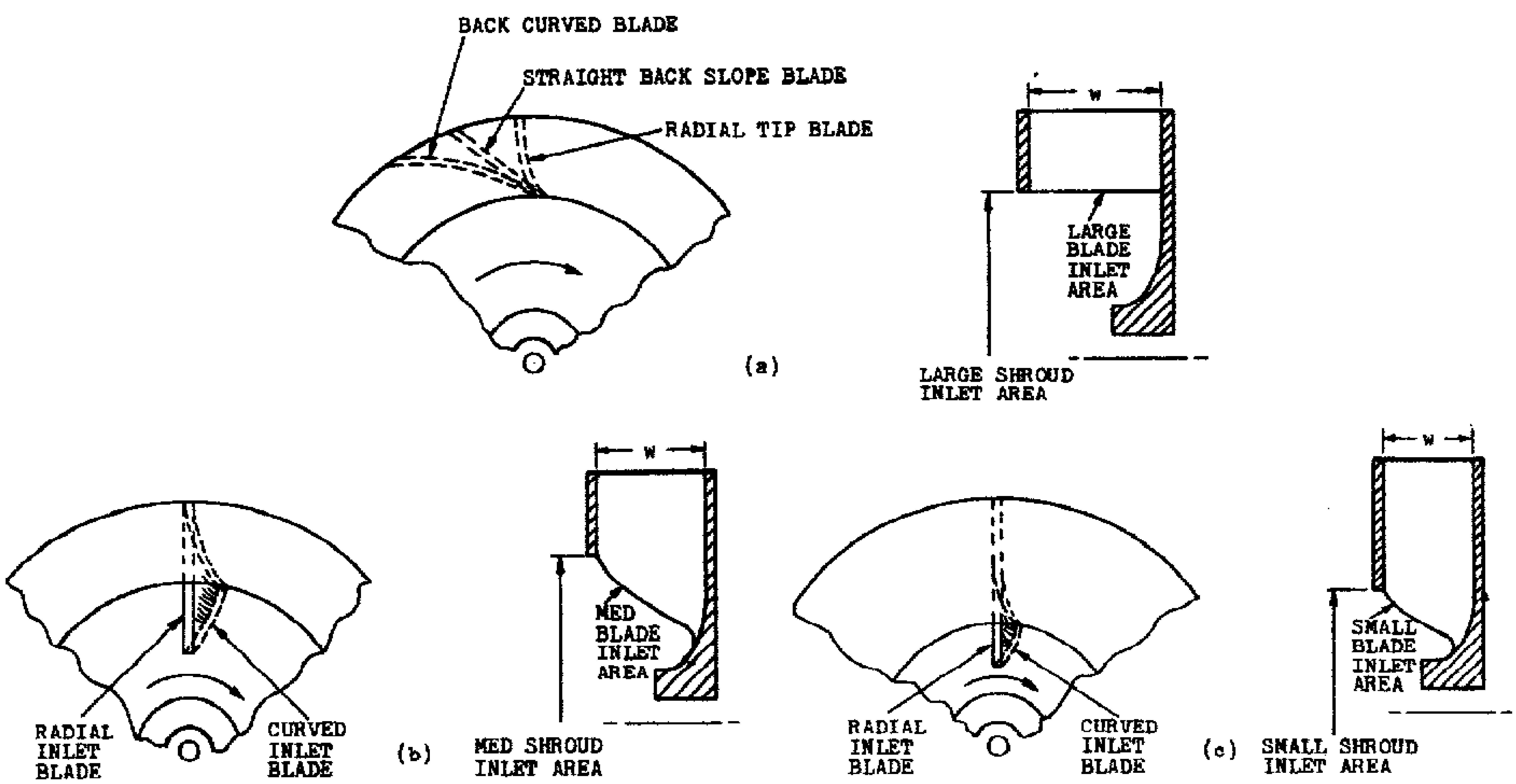
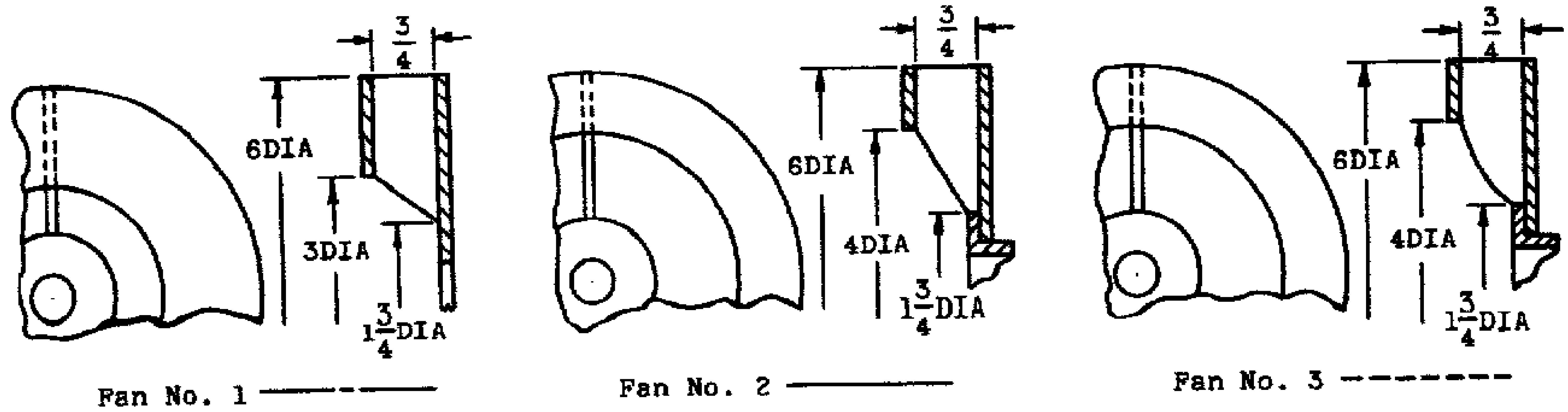


Figure 2-19. Fans Without Scrolls or Diffusers.



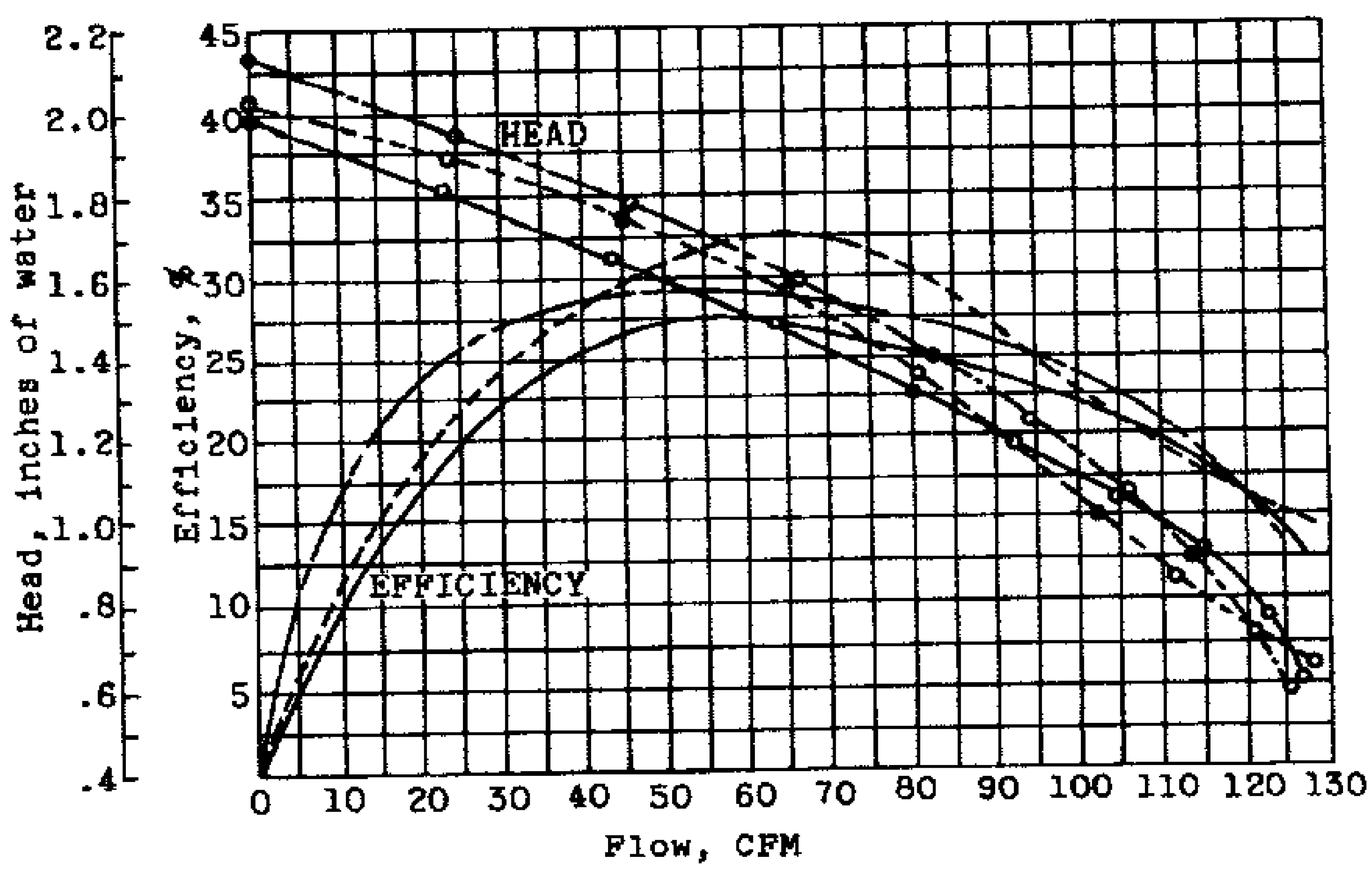
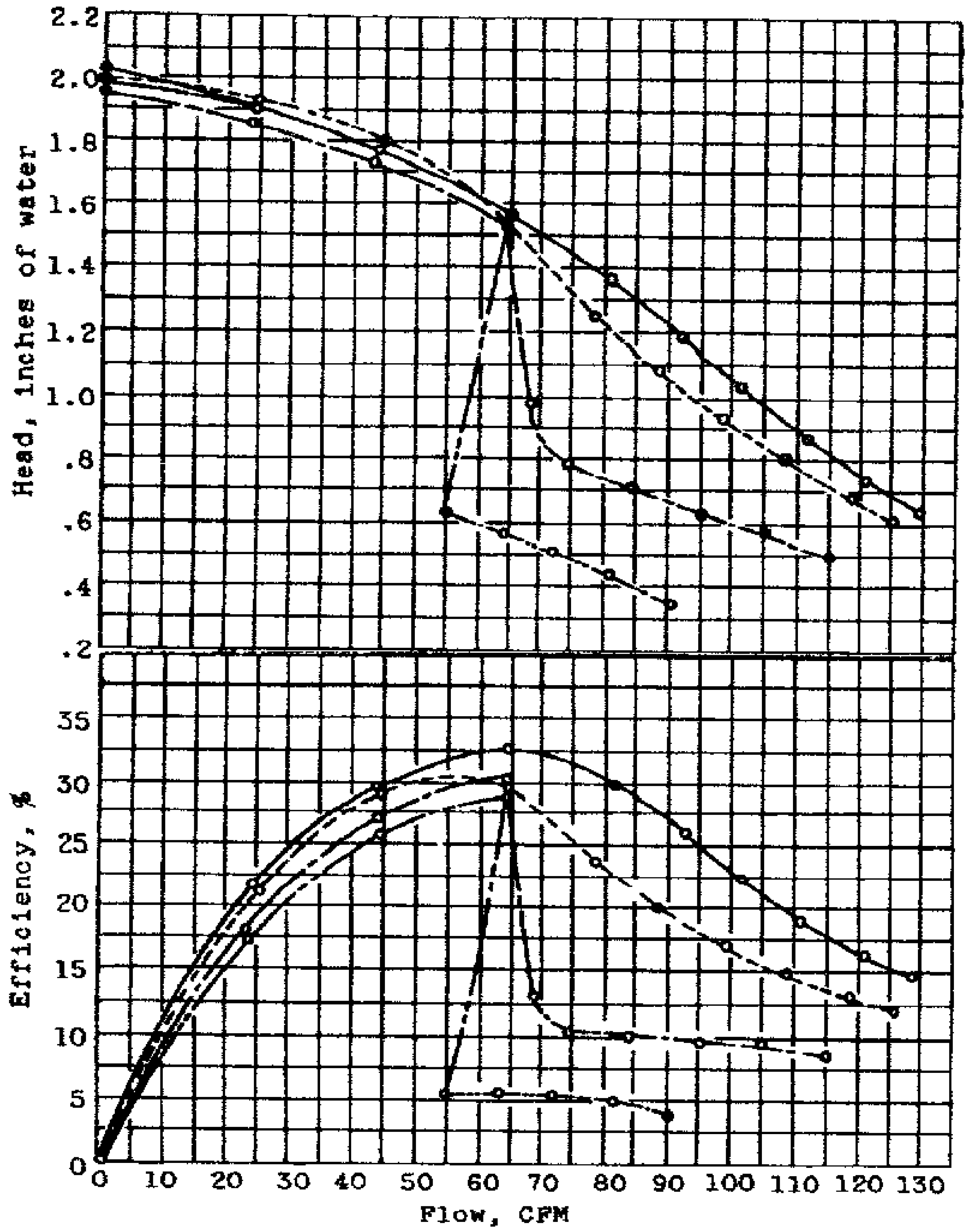


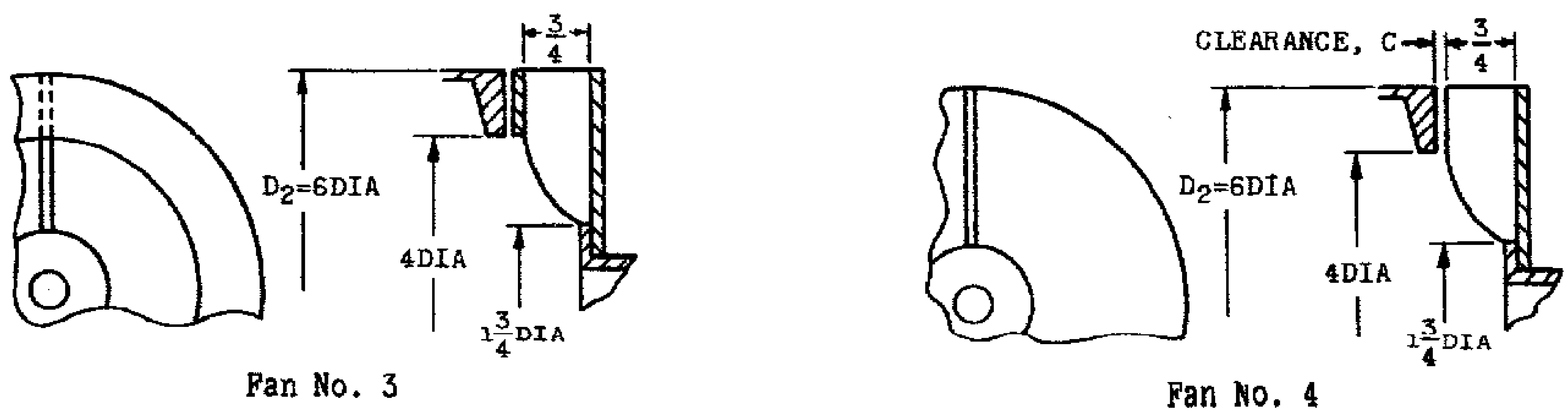
Figure 2-20. Radial Fan Blade Depth Comparison (Narrow – 16 Blades – 4000 RPM).

B. Clearances

Stationary shrouds must have close clearances (less than about 0.4 percent of outer blade diameter) with the blades; pressure outputs and flow, Q, both suffer from too large clearances. See Figure 2-21 for comparison of narrow blades.

Considerations of economy of manufacture or convenience of assembly sometimes make it desirable to permit large running clearances resulting in a relatively poor seal between the exhaust and inlet side of the fan. When designing a fan for such conditions, provision should be made for greater air flow through the impeller to allow for the recirculation. As an approximation one may use an increase of flow of approximately 25 percent; then check the final design by calculating the usual air flow by the method outlined in Ref. [2.10] then make any necessary final adjustments in the fan design.





——— No.3 minimum clearance from orifice
— — — No. 4 minimum clearance from orifice
—— – —— No. 4 ⅛ in. clearance from orifice
——·—— No. 4 ¼ in. clearance from orifice

Figure 2-21. Clearance Comparison Using Orifice as Shroud (Narrow—16 Blade—4000 RPM).

IX. OPERATION AND MAINTENANCE

Proper operation and maintenance is essential for satisfactory service. [2.11] through [2.18] provide useful information on this subject.

I. TYPES OF FANS

Axial flow fans are of three basic types namely propeller, tubeaxial and vaneaxial. See Figure 1-2.

A. Propeller Fan

The propeller fan is an axial-flow fan which either has no housing, as, for example, the common office or desk fan, or has a simple ring mounting consisting of a plate with a hole in it. An example of the latter is a kitchen ventilating fan. This fan is especially suited to handling large volumes of flow at low heads, not exceeding one inch of water.

B. Tubeaxial Fan

The tubeaxial fan is an axial-flow fan in a tubular housing without stationary guide vanes. It can be used at heads somewhat higher than those suitable for propeller fans.

C. Vaneaxial Fan

The vaneaxial fan is an axial-flow fan mounted in a tubular housing with stationary guide vanes, either before or after the fan wheel. The vanes, which may be either fixed or adjustable, direct the air flow to reduce the circulation and thus increase useful head and efficiency. The vaneaxial fan is used for higher pressures, while still higher heads can be developed by using several stages.

II. PERFORMANCE CHARACTERISTICS

Performance curves for a typical axial-flow fan are shown in Figure 3-1. These curves show static pressure, static efficiency and fan horsepower plotted as a function of rate of flow at constant speed. As in the section on centrifugal fans, these curves are plotted as percentages of their rated value.

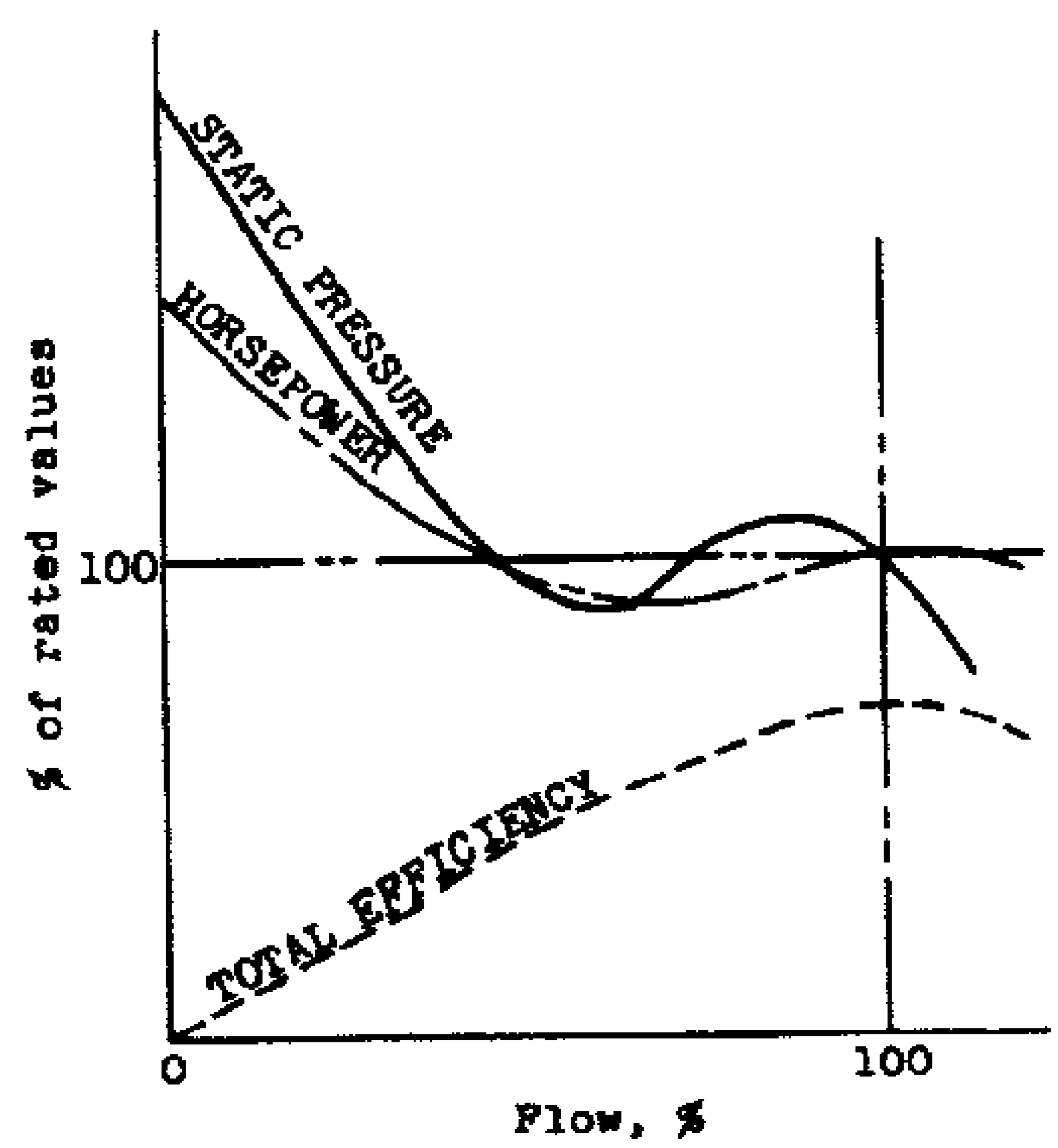


Figure 3-1. Characteristics of a Propeller Fan.

A. Pressure

The pressure characteristics of an axial-flow fan are very similar to those of a forward-inclined centrifugal fan. The no-delivery pressure is high and drops rapidly with increasing capacity until a point at about 50 to 80 percent of rated capacity is reached. At this position a point of inflection, and on some fans a small rise in pressure, occurs. With further increase in flow the pressure drops off steeply until the free delivery point is reached. The inflection point, sometimes called the breakdown point, represents a transition from the condition of flow at low capacity, in which radial components of velocity are appreciable, to the condition at higher capacity, in which the flow is almost entirely axial. The breakdown at low flows is caused by stalling at the blade entrance. Operation of the fan in this region is likely to be unstable with pulsation in the flow.

B. Fan Horsepower

Figure 3-1 shows that the fan horsepower is a maximum at no flow, decreasing with increasing flow while in the unstable region. As the flow becomes stable the horsepower increases to a value less than that for zero flow and then again decreases with larger flows. The slope of the decreasing portions of the curve varies widely among different designs. With some fans, it is practically horizontal, while with others it is quite steep. Fans with steep characteristics require much larger motors than for rated conditions.

C. Efficiency

Axial-flow fans are usually used for the development of velocity head, hence the total efficiency rather than the static efficiency is commonly taken as a basis for rating. Total efficiencies of up to 70 percent may be attained with small enclosed units, however efficiencies of 50-60 percent are more common. Large vaneaxial units frequently attain 85 to 90 percent. Open, household type, fan efficiencies range from 40 percent for small fans to 50 percent for large exhaust fans.

III. FAN DESIGN

Some factors involved in fan design are briefly discussed here and some design formulas are given. For detailed information on fan design, see [1.1] to [1.4] and [3.1]

A. Definitions

The various terms involved in fan design are first defined. Figure 3-2 shows a cylindrical section through blades developed on a plane.

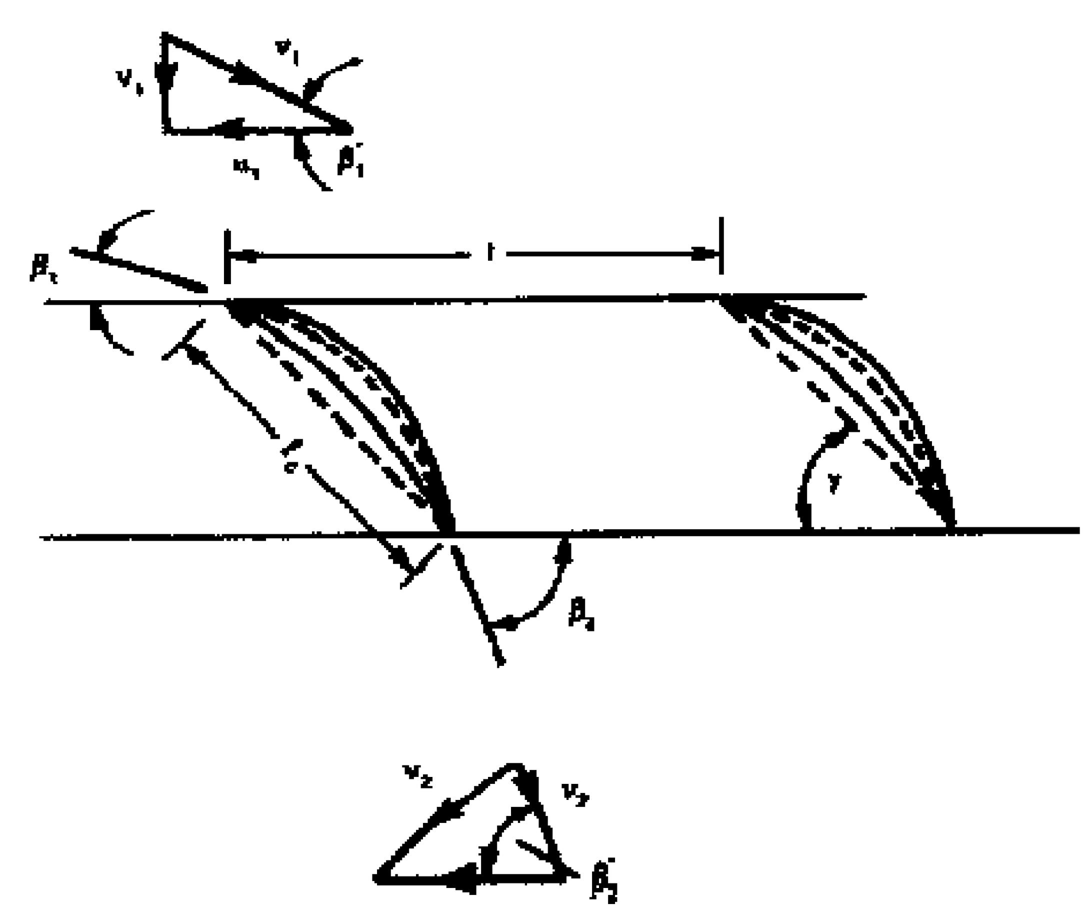


Figure 3-2. Blade Cascade of an Axial Flow Fan with Various Definitions.

β_1 is the angle between the tangent to the blade mean thickness line at the blade leading edge and a reference line drawn tangent to the hub at that point in the direction of rotation. Similarly, β_2 is the blade angle at the trailing edge. The angle which the entering air stream makes with the tangent in the direction of rotation is β_1^* and β_2^* is the corresponding angle where the air stream leaves.

A straight line joining the leading and trailing edges is called the blade chord. Its length l_c is known as the chord length or blade width. The angle γ between the chord and the reference line is known as the blade setting, or stagger angle, or the pitch angle.

The angles and lengths shown in Figure 3-2 may change in the radial direction.

Figure 3-3 shows a front view of the fan. D_1 is the hub diameter and D_2 is the diameter of rotor at the blade tip. Design calculations for axial flow fans are often done using a mean diameter D_m defined as:

$$D_m^2 = \frac{D_1^2 + D_2^2}{2} \tag{3-1}$$

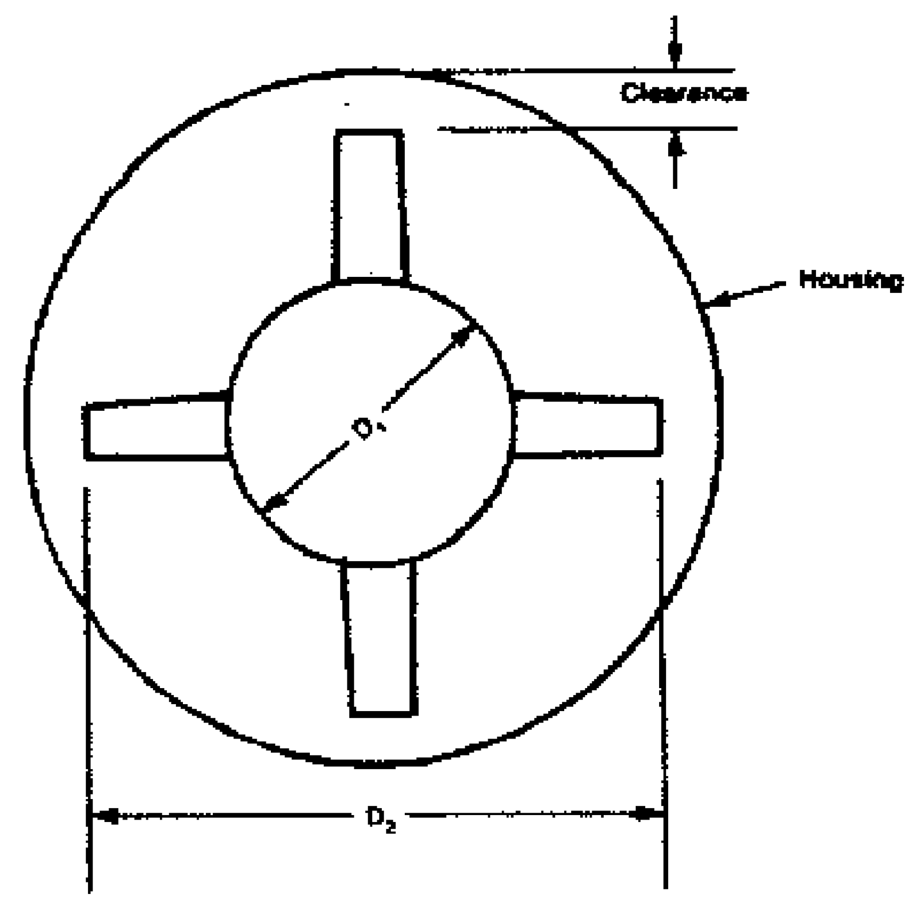


Figure 3-3. Axial Flow Fan with Four Blades.

B. Fan Speed

High fan speeds result in compact designs but also higher noise. If direct drive is to be used, the fan speed is limited to the synchronous speeds of motors. The choice has to be made based on experience and the relative importance of considerations such as noise and size.

C. Design Diameters

According to [1.3] (page 222), experience has shown that there is an optimum value of (D_1/D_2) for each value of specific speed n_S. The relation between n_S and optimum (D_1/D_2) based on their experience is given in graphical form in [1.3]. It can be expressed by the following equation:

$$S_1 = D_1/D_2 = 9460/n_S^{0.86} \tag{3-2}$$

Knowing the required flow in cfm, the total pressure to be developed in inches of water, and the speed in rpm, n_S can be calculated with Eq. (11). It should be noted that Eq. (3-2) is applicable only for n_S between 42,000 and 230,000. Extrapolatians should not be made.

According to [1.2] (page 212), values of (D_1/D_2) used in practice range from 0.4 to 0.8, with the higher values being used for higher pressure fans.

Applying the continuity equation, the following dimensionless equation is obtained for D_2 [1.2]:

$$D_2 = 0.74 \left[\frac{1}{S_1(1 - S_1^2)}\right]^{1/3} \left[\frac{Q}{kn}\right]^{1/3} \tag{3-3}$$

where k is the ratio of air velocity through the passage between hub and casing to the peripheral velocity at hub. This can be expressed as:

$$k = \frac{4Q}{\pi^2 n D_2^3 (1 - S_1^2)\, S_1} \tag{3-4}$$

This is a dimensionless equation and hence any consistent units can be used. For example take Q in cfm, n in rpm, and D_1 and D_2 in feet. The value of k is generally between 0.6 and 1 [1.2].

One way to use the above equations is to first assume a value of k between 0.6 and 1 and calculate D_1 and D_2 with Eq. (3-3) and (3-4). S_1 thus calculated can then be checked against Eq. (3-2). Also see [3.6].

D. Number of Blades and Blade Pitch

The pressure developed increases directly with the number of blades. The relation breaks down as the number of blades becomes large, since there is interference between blades and restriction of flow due to the cross-sectional area of the blades. The selection of the correct number of blades is largely a matter of experience. For low pressures, it is usually desirable to have as few blades as possible, two, three, or four being optimum depending on mechanical construction considerations. For higher pressures, using vaneaxial or tubeaxial construction, a large hub diameter is generally used, and a large number of blades, as many as 40.

Ref. [1.3] (page 224) gives the following approximate formula for determining the optimum number of blades N:

$$N = \frac{6D_1}{D_2 - D_1} \qquad (3\text{-}5)$$

The blade spacing t is obtained by dividing the circumference (πD) by the number of blades N. The value of t increases from hub to tip. Thus:

$$t = \frac{\pi D}{N} \qquad (3\text{-}6)$$

Another method of calculating the optimum blade pitch and therefore the optimum number of blades is provided by the analysis of Zweifel which leads to the following formula [1.1] (page 262):

$$\frac{l_c \mathrm{Sin}\gamma}{t} = 2.5\ \mathrm{Sin}^2\beta_2[\mathrm{Cot}\beta_1 - \mathrm{Cot}\beta_2] \qquad (3\text{-}7)$$

E. Blade Width (Chord Length)

The blade width l_c (generally known as chord length) may either increase, decrease, or remain constant with distance from the hub. Aerodynamic considerations require that blade width should increase from hub to tip. Structural considerations require the reverse. As a compromise, many designers use a constant blade width from hub to tip [1.3] (page 224). As the propeller fans develop only a small pressure rise, the forces involved are small and hence blades with width increasing away from the hub are often used. Due to the higher pressures and forces involved in vaneaxial or tubeaxial fans, the blade width often decreases from hub to tip.

As the blade spacing t increases from hub to tip, the ratio t/l_c also increases similarly. Ref. [1.3] (page 224) gives two correlations in graphical form for optimum t/l_c at the mean effective diameter D_m. These correlations can be expressed by the following equations (applicable range n_s from 40,000 to 250,000):

$$\left(\phi \frac{t}{l_c}\right)_m = 19 \times 10^{-6} n_S^{0.91} \qquad (3\text{-}8)$$

$$\left(\psi \frac{t}{l_c}\right)_m = 0.0064 n_S^{0.41} \qquad (3\text{-}9)$$

The subscript m indicates that ψ, ϕ, and t, are to be at the mean effective diameter, D_m which is defined by Eq. (3-1). The pressure and flow coefficients ψ and ϕ are calculated using Eq. (1-5) and (1-6) with the flow area based on D_m. This leads to the following dimensionless equations:

$$\psi_m = \frac{\Delta p_t}{\frac{\rho}{2g} (\pi D_m n)^2} \tag{3-10}$$

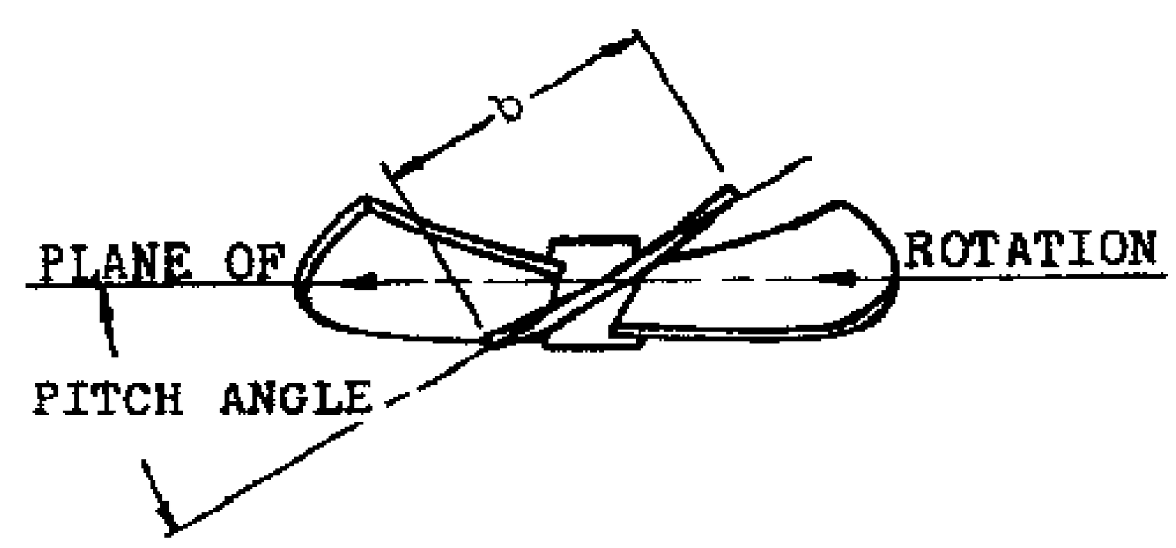
$$\phi_m = \frac{Q}{(\pi D_m n)\, A_2} \tag{3-11}$$

It should be remembered than n_s is calculated using total fan pressure in inches of water. The values of t/l_c from Eq. (3-8) and (3-9) will generally not be exactly equal. If the difference in calculated values is appreciable, it usually indicates that a new value of D_m should be tried. Eq. (3-8) and (3-9) are based on data on fans of conventional design without pre-rotation.

F. Stagger Angle (Blade Setting)

Stagger angle γ has been defined earlier and is shown in Figures 3-2 and 3-4. It is most commonly known as blade setting and is also sometimes called pitch angle. It varies with the pressure required and with the blade shape. Within limits, the greater the stagger angle the greater the pressure developed at a given speed, since the lift coefficient C_L of a section is roughly proportional to the sine of the angle of attack for small angles. As the angle increases, the point of aerodynamic stall is reached, and the instability shown in Figure 3-1 occurs. Optimum stagger angle can be calculated with Eq. (3-7) once l_c/t is known.

Figure 3-4. Pitch Angle (Stagger Angle) and Chord Length.



The stagger angle or "helix" angle should be varied from hub to tip in order to keep the pressure across the face of the fan constant, so that the lift coefficient times the width times the relative air velocity remains constant. The parts of the blade away from the hub are traveling faster than those near the hub and do not require as great a pitch angle to produce the same pressure. However, there is a limit to the pressure equilization that can be achieved by this method. With a small hub diameter, there will be a greater pressure at the tip than at the hub. This can lead to serious back flow if the pressure difference is large. Hence for tubeaxial and vaneaxial fans operating with larger pressure rises, the hub diameter has to be made larger.

The performance of the fan can be altered over a wide range by changing the blade setting (stagger angle). Flow reductions of up to 50 percent can usually be achieved with little loss of efficiency. Maximum flow variation usually possible by this method is 1 to 8. Some fans have manually adjustable blades while some have automatically adjustable blades which permit blade setting to be modulated during fan operation. Figure 3-5 shows performance data for a typical vaneaxial fan. Capacity control by this method is discussed in Section 409.9.

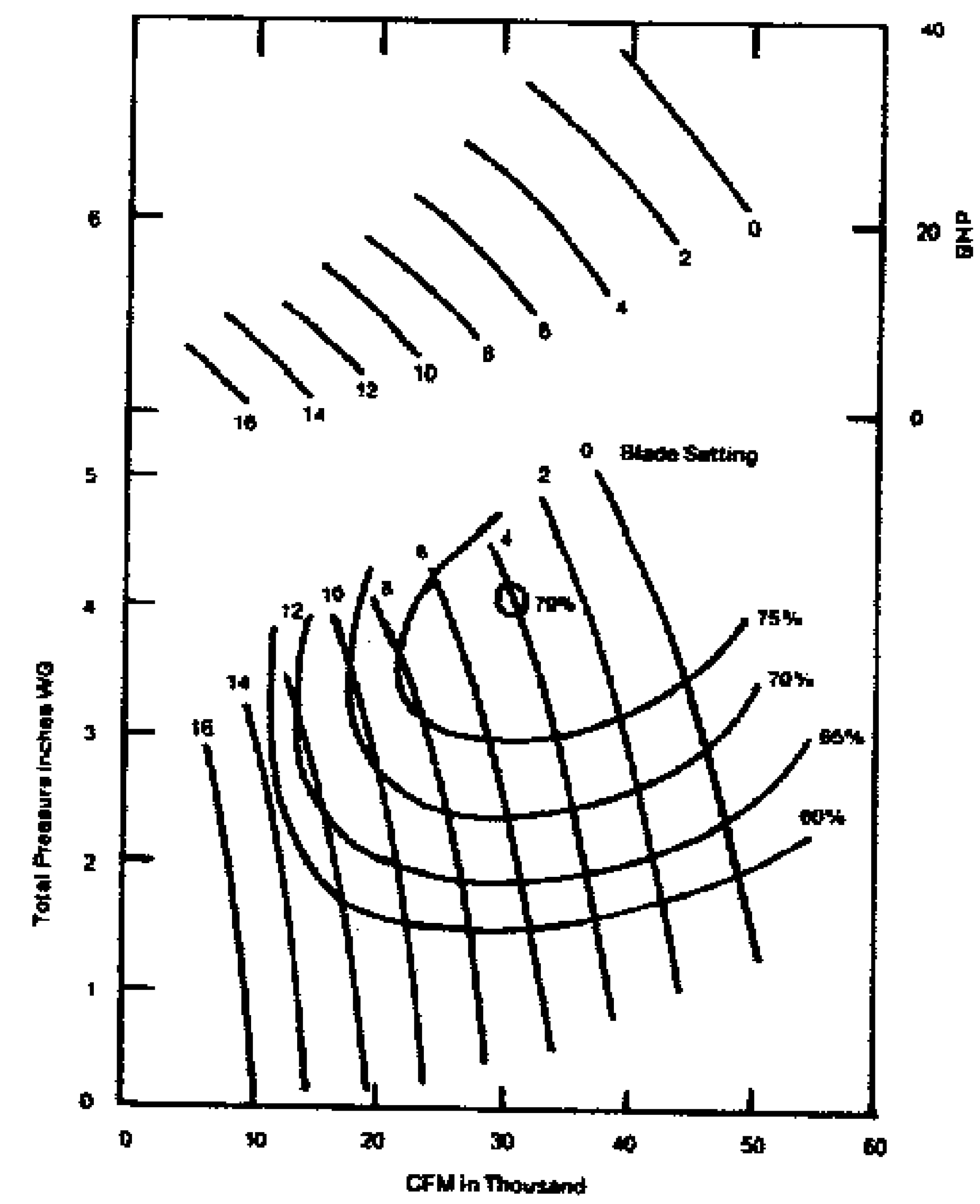


Figure 3-5. Effect of Blade Setting on the Performance of a Vaneaxial Fan.

G. Blade Angle

The blade angle β_1 and β_2 depend only on the blade shape and blade setting. The fluid entering and leaving angles depend on the blade and fluid velocities and can be determined with velocity triangles.

From Figure 3-2, it is clear that:

$$\tan\beta_1^* = \frac{V_1}{u_1} \tag{3-12}$$

The incidence angle $(\beta_1 - \beta_1^*)$ is generally kept close to zero. According to [1.2] (page 213) optimum value is usually between 2° to 7°. The best value should be determined by testing blade cascades in wind tunnels.

The deviation between fluid angle and blade angle at the trailing edge can be calculated by the following formula derived by Eck [1.1] (page 263):

$$(\beta_2 - \beta_2^*) = 0.25\,\frac{t}{l_c}\,(\beta_2 - \beta_1) \tag{3-13}$$

Ref. [1.3] (page 226) presents curves showing a relation between ψ_m and $(\beta_1 - \beta_2)_m$ based on experimental data, the subscript "m" signifying "at mean effective diameter". The curves can be approximated by the following equation for ψ_m between 0.06 and 0.40:

$$(\beta_2 - \beta_1)_m = 58\psi_m^{0.89} \tag{3-14}$$

H. Blade Profile

Radial flow of gas along the blade of an axial-flow fan lowers the efficiency, pressure, and flow rate of the fan and is one of the prime causes of fan noise. Such radial flow can be eliminated by designing the blade so that the pressure differences across the blade are the same at all sections. For details see [3.2] and [1.1].

The blade profile of the higher pressure units, such as vaneaxial and tubeaxial fans, is usually an airfoil section giving a high lift coefficient. For propeller type fans, however, the advantage of airfoil blades is small. Many good designs of blades made of thin sheets have been made, as the simplicity of construction outweighs the advantage of an airfoil shape on low pressure fans.

Many combinations of circular and parabolic arcs are used for blades. See [1.1] for detailed design of blade profiles. Also see [1.1] which gives simple methods for selecting blade profiles.

I. Tip Clearance

Some clearance between the blade tip and housing is necessary to permit fan operation. The size of this clearance has a profound effect on pressure, flow volume, and efficiency. Excessive noise can also result [1.1]. Measurements by Marcinowski [3.4] indicate that there is an approximately 2% loss in efficiency for a 1% increase in clearance. Figure 3-6 shows data for some of the fans tested by Marcinowski. A clearance of 0.1 percent of impeller outer diameter is frequently used when both impeller and housing are machined [1.3].

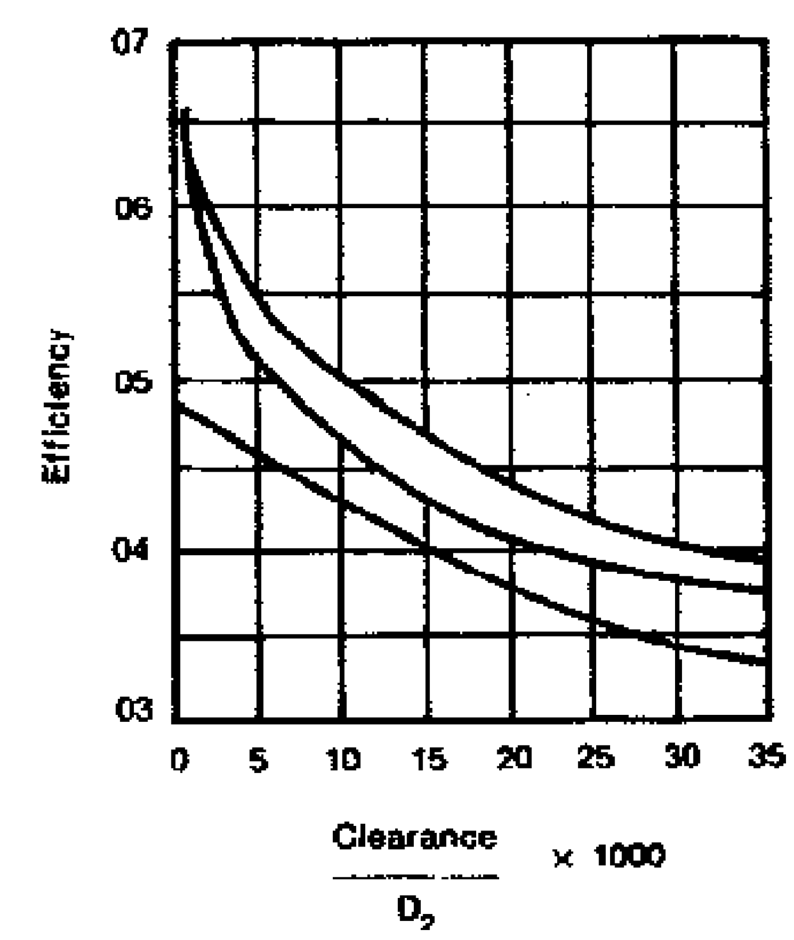


Figure 3-6. Effect of Tip Clearance on Maximum Efficiency of Three Fans Tested by Marcinowski [3.4].

The harmful effects of large clearance are most severe for high pressure fans with profiled blades. For very low pressure fans such as propeller fans, fairly large clearances can be tolerated.

J. Guide Vanes

Guide vanes may be used upstream or downstream of the impeller. Investigations [3.5] show that upstream guide vanes are useful only at very low hub to tip diameter ratios and particularly bad lift-drag ratios. This statement is valid only if there is no pre-rotation of air at entrance. If pre-rotation existed, upstream guide vanes will improve performance. In most fans manufactured, downstream guide vanes are used.

The air discharged by the impeller has a significant radial velocity component. The downstream guide vanes turn the air until this radial component of velocity is eliminated. The inlet angle of the guide vanes should be the same as the angle at blade exit, β_2^*. The guide vanes should discharge in the axial direction. Close spacing between guide vane and impeller blades is desirable when the operating point is fixed. Larger spacings are preferable when the operation will be over a wide range. The spacing should generally be about 10% of the tip diameter [1.3].

Guide vanes are economically justifiable where they bring about a significant reduction in pressure losses. An analysis by Eck [1.1] (page 282) yields the following expression:

$$\frac{\Delta p_1}{\Delta p} = \frac{\psi}{[1 - S_1^2]\, 2\eta^2} \ln\left(\frac{1}{S_1}\right) \tag{3-15}$$

where Δp_1 = pressure loss if there were no guide vanes

Δp = total pressure developed by impeller

η = impeller efficiency

ψ = pressure coefficient

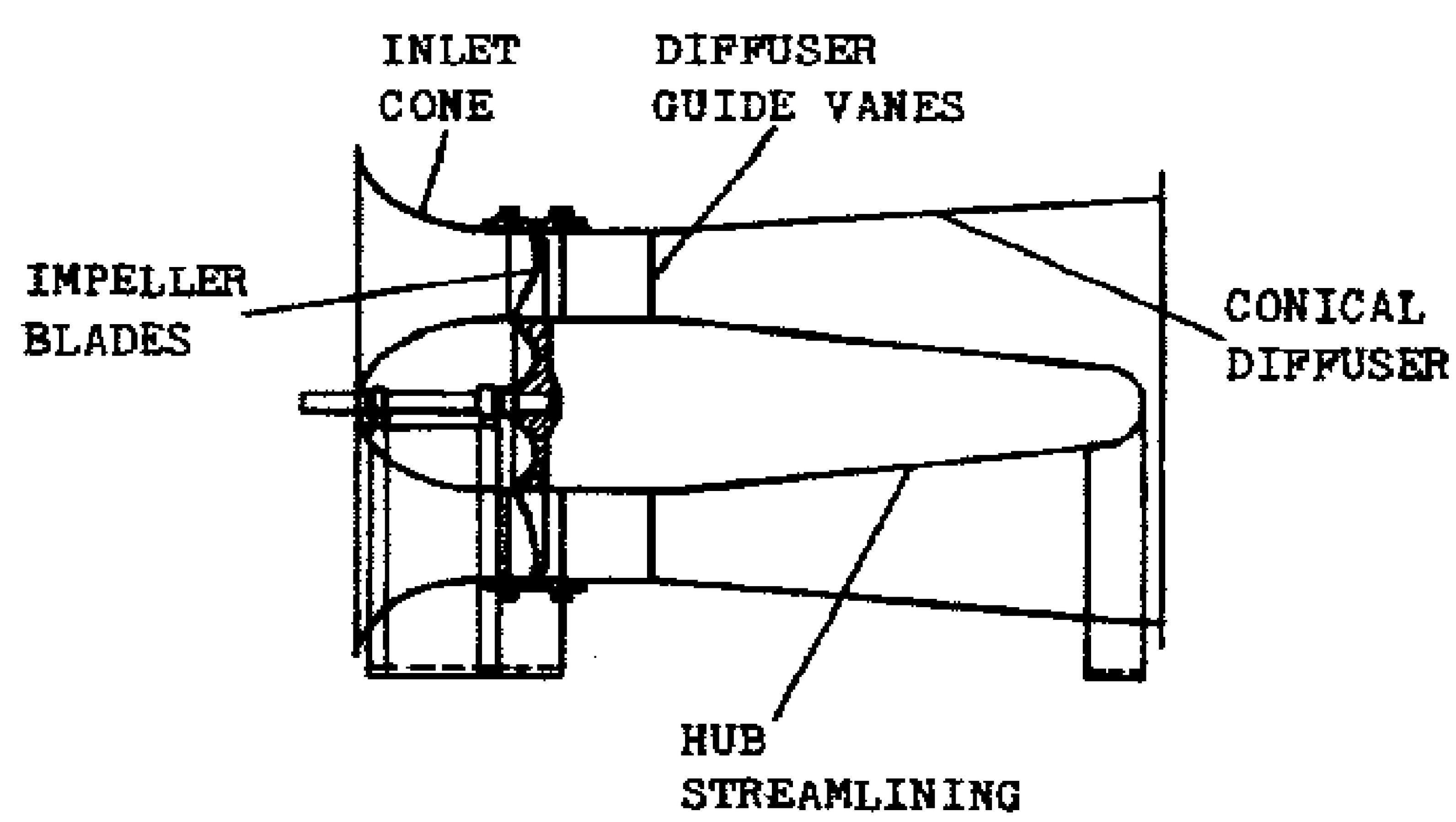
For S_t between 0.3 and 0.7, this formula gives losses of 5 to 10 percent when $\psi = 0.1$. Hence for ψ greater than 0.1, guide vanes are always desirable.

The number of guide vanes should not be the same as the number of impeller blades as it results in excessive noise. The number of guide vanes from accoustic consideration can be determined with Eq. (6-4).

K. Fan Casings and Diffusers

The casing for the usual vaneaxial or tubeaxial fans is a concentric cylinder around the rotor as shown in Figure 3-7. In meridionally accelerated fans, the casing is designed such that the flow passage area diminishes in the direction of flow.

Figure 3-7. Axial Flow Fan with Diffuser and Guide Vanes.



For fans without inlet duct connections, the use of inlet cones (also known as inlet bell) significantly improves performance. These may be an integral part of the fan casing but are more commonly bolted to the casing.

The velocity of air leaving the casing is usually high, typically around 4000 ft./minute which corresponds to 1 inch WG pressure. In many applications, static pressure rather than velocity pressure is needed. The velocity pressure can be converted to static pressure by using a conical diffuser, generally known as the outlet cone. The increase in static pressure through an ideal diffuser may be calculated as follows:

$$\Delta p_S^* = \frac{V_1^2 - V_2^2}{4000^2} \tag{3-16}$$

where V_1 and V_2 are respectively the inlet and outlet velocities in feet per minute. In actual diffusers, the static pressure is less than given by Eq. (3-13). For good diffuser efficiency, the diffuser expansion angle should be kept small. The diffuser cone is generally bolted to the fan casing. Diffuser design in general is discussed in Section 406.

To minimize losses, the hub nose and tail should be streamlined. The extended hub tail shown in Figure 3-7 is generally cut short in commercial fans to make the unit more compact. This results in a slight loss of efficiency.

L. Drive Arrangements

Both direct drive and belt drives are used. In direct drives, the fan and motor often share a common shaft. Direct drives save space but have the disadvantage that the fan speed cannot be changed. However, this is not much of a disadvantage if blade setting can be changed as the desired changes in fan performance can be obtained by changing the blade setting. If the fan is handling hot, corrosive, or erosive gases, the motor has to be kept out of the air stream and belt drive is the only practical proposition. Belts passing through the air stream generally cause some loss in performance. For belt drives, the motor is often mounted on the fan casing. The motor can be mounted on top, bottom, or any other angle.

IV. REVERSIBLE FANS

In some applications, a reversal of air flow direction is needed. In axial flow fans, the direction of air flow is reversed when the direction of impeller rotation is changed. In fans with adjustable blade setting, the flow direction can also be changed by reversing the blade setting. However, performance in reverse flow will be very inferior unless special care is taken in design. For reversible fans, the blade profiles should be symmetrical. Fans in which the performance in both directions is equal are available. However, their performance is inferior to that of fans designed to operate only in one direction.

Refs. [3.7] and [3.8] describe some recent developments in reversible fans.

I. AIR FLOW REQUIREMENTS

The air flow and pressure head of a fan is determined by the system served by it. The fan flow needed can depend on several criteria among which are:

1 Maintaining a certain temperature in a room or in a machine
2 Diluting concentration of hazardous substances
3 Providing rapid pressurization or depressurization in a space
4 Providing sufficient velocity for capturing and conveying dust and other particulates.

A. Air Flow Needed for Cooling

The windings and core of electric motors generate heat during operation. Air is blown over them to dissipate the heat and protect the winding insulation from damage at high temperatures. In an office building, cooling air is provided to offset various heat gains so that occupants can be comfortable. For detailed guidance on calculation of heat gains in buildings, see [4.1]. Rapid estimates of heat gains in engine-generators rooms may be made with [4.2].

The quantity of air needed for cooling can be determined by the following equation:

$$Q = \frac{H}{60 \rho C_p (t_2 - t_1)} \tag{4-1}$$

where Q = fan air flow, cfm
H = quantity of heat to be removed, Btu/hr
ρ = density of air at fan inlet, lb/ft^3
C_p = specific heat of air, Btu/lbm °F
t_1 = temperature of entering air, °F
t_2 = temperature of leaving air, °F

Equation (4-1) can be written in the following approximate form:

$$Q = \frac{H}{1.08\,(t_2 - t_1)} \, \frac{(460 + t)}{(460 + 70)} \tag{4-2}$$

where t is the temperature of air entering the fan, in °F. Thus if the heat gain in a room is 100,000 Btu/hr, air supply is at 70°F, and air exhaust is at 90°F, the capacity of exhaust fan needed is 4804 cfm.

B. Air Flow Needed For Dilution

In a variety of processes and plants, hazardous substances are evolved and added to the room atmosphere. For example, hydrogen is generated during battery charging and in the absence of ventilation, may form an explosive mixture with air. More often, the hazard is due to toxicity of substances. Sufficient ventilation has to be provided to keep the concentration of substance in air below dangerous levels.

The air flow needed for dilution may be calculated by the following formula:

$$Q = \frac{10^6 q}{C} \tag{4-3}$$

where q = volume of hazardous vapor generated, cfm
C = maximum permissible concentration of hazardous vapor in air, parts per million (ppm)

For further guidance on the subject, see [4.3].

C. Air Flow For Pressurization and Depressurization

In many situations fans have to be sized such that a room or building can be brought to a desired positive or negative pressure within a certain time. Such requirements are often found in buildings where the possibility of release of toxic gases exists, for example in nuclear power plants.

Shah [4.4] has given analytical solutions for estimation of pressure transients in buildings at constant temperature. For the case of a linear relation between leakage and pressure difference between inside and outside, his solution simplifies to the following relation:

$$T = \frac{V}{p_o B} \ln \left[\frac{Q - B\Delta p_i}{Q - B\Delta p} \right] \tag{4-4}$$

where T = time in minutes

Δp_i = absolute value of pressure difference between inside and outside when $T = 0$, lb/ft²

Δp = absolute value of pressure difference between inside and outside after T minutes, lb/ft²

Q = capacity of fan, cfm

V = building volume, ft³

P_o = outside pressure, lb/ft² absolute

$B = \dfrac{\text{Leakage in cfm at } \Delta p}{\Delta p}$

Formulas for some cases of non-linear relations between leakage and pressure difference are also given in [4.4].

D. Air Velocities Needed For Conveying Solid Particles

A particle in air is pulled down by gravity but its fall is opposed by viscous drag. The velocity at which the viscous drag just equals the gravitational pull is known as terminal velocity of particle. For vertical transport of the particle, air velocity should clearly exceed the terminal velocity.

Gelperin and Einstein [4.5] recommend the following dimensionless equation for calculating the terminal velocity in vertical ducts:

$$R_t = \frac{Ar}{18 + 5.22\, Ar^{0.5}} \tag{4-5}$$

where R_t = terminal Reynolds number

$= \dfrac{V_t \rho d}{\mu}$

Ar = Archimedes number

$\dfrac{g d^3 \rho_S \rho}{\mu^2}$

d = diameter of particle, ft

g = acceleration due to gravity, ft/hr²

ρ_S = absolute density of solid, lb/ft³

ρ = density of air, lb/ft³

μ = viscosity of air, lb/hr ft

V_t = terminal velocity of particle, ft/hr.

A theoretical analysis [4.6] leads to the followinng dimensionless equation for the minimum velocity required to pick up particles in horizontal ducts:

$$V = \left[\frac{4gd(\rho_s - \rho)}{3\rho}\right]^{0.5} \qquad (4\text{-}6)$$

where V is the minimum pickup velocity in ft/hr, the units for other quantities being the same as for Eq. (4-4). However this equation seems to considerably overpredict the experimental data. Zenz and Othmer [4.6] have given a graphical correlation which shows better agreement with measurements.

The air velocities required for conveying generally vary from 1500 ft/min. for very light and small particles to 5000 ft/min. for heavy and large particles.

Table 4-1 gives generally accepted conveying velocities for some common materials.

Table 4-1
Range of Velocities (in fpm) used for conveying materials in ducts

Carbon Black	3500
Castor Beans	5000
Coffee Beans	3500-4000
Cement	6000-7000
Coal, Powdered	4000
Cotton	3000-4500
Flour	3500
Foundary Dust	4500
Grain	5000-6000
Grinding Dust	5000
Iron Oxide	6500
Limestone, Pulverized	3500-5000
Lead Dust	4500-5000
Metallic Fumes	1400-2500
Sand	4000-7000
Sawdust	3000-4000
Salt	6000
Sugar	6000
Wool	4500-5000

II. ESTIMATION OF REQUIRED FAN PRESSURE

Some fans operate without duct connections, as for example ceiling fans, but more often they are used to cause air to flow through circuits consisting of a variety of flow passages. These flow passages may consist of ducts, louvers, dampers, slots, heat exchangers, etc. The resistance of the circuit component causes pressure drop. The fan is required to overcome their resistance and deliver the required amount of air.

Furthermore, certain kinds of fan inlet and outlet connections can result in deterioration of fan performance due to system effects. This deterioration is generally calculated as a pressure drop to be added to the flow circuit pressure drop.

For proper fan selection, both the flow passage pressure drop and the system effect pressure drop have to be calculated. Their calculation is now discussed.

A. Flow Passage Pressure Drop

Pressure drop in flow passages occurs due to frictional effects and due to dynamic effects. In long straight passages, the loss of pressure is primarily due to friction. In bends, expansions, contractions etc., the losses are primarily due to dynamic effects. The general practice is to calculate the pressure drops due to frictional and dynamic effects separately and then add them together to get the total pressure drop in the flow passage. Calculation of such pressure drops is dealt with in Sections 401 to 406. Further information may be found in [4.1].

The various flow passages may be connected together in series or parallel, or a combination of the two. For flow passages connected in series, the total pressure drop in the circuit is the sum of the pressure drops in each flow passage (i.e. component) of the circuit. If the flow passages are in parallel, the pressure drop in each flow passage is the same (see Manifolds, Section 404).

B. Pressure Drop Due to System Effects

The fan performance data normally published in manufacturer's catalogs are based on tests carried out with ideal fan inlet and outlet conditions. In practice, space limitations often force the designers to use inlet and outlet connections which fall short of ideal and cause deterioration in fan performance. The effect of inlet and outlet connections on fan performance is known as "system effect". It is generally calculated as an additional pressure drop to be added to the pressure drop in the flow circuit. In reality, system effects cause a reduction in pressure produced by the fan.

System effects and their calculation are discussed in [4.7] to [4.12] and [2.1]. The most comprehensive source of information for calculation of system effects is AMCA Publication 201 [2.1] which provides simple methods of calculation in graphical form. The method of calculation given here is based on the data in [2.1].

The pressure loss due to system effects is proportional to the velocity pressure and can be expressed by the following equation:

$$\Delta p^*_{fse} = K_{se}\left(\frac{V'}{4000}\right)^2 \qquad (4\text{-}7)$$

where Δp^*_{fse} = loss of total pressure due to system effects, inches of water
V' = velocity of air in ft/min.
K_{se} = system effect loss coefficient, dimensionless

The factor K_{se} generally depends on the geometry of inlet and outlet connections.

Correlations for calculation of K_{se} are given in the following. Their use will give virtually the same results as will be obtained with the graphs in [2.1].

1. Outlet Connections

a. Minimum straight duct length. For the fan to give its rated performance, the outlet connection should be a straight duct of uniform cross-section, several duct diameters long. The minimum straight duct length L_{min} needed to prevent system effect loss is given by:

$$\frac{L_{min}}{D_{eq}} = \left[\frac{V'}{1000}\right] \geq 2.5 \qquad (4\text{-}8)$$

D_{eq} is the equivalent diameter of duct. It is equal to the duct diameter for circular ducts. For rectangular ducts, it is calculated as:

$$D_{eq} = \left[\frac{4a_1a_2}{\pi}\right]^{0.5} \tag{4-9}$$

where a_1 and a_2 are the length of the two sides of the duct.

If the length of the straight duct is less than L_{min}, there will be a loss in performance. The resulting pressure loss can be calculated by Eq. (4-7), with K_{se} from Figure 4-1. See Figure 2-1 for definition of blast area.

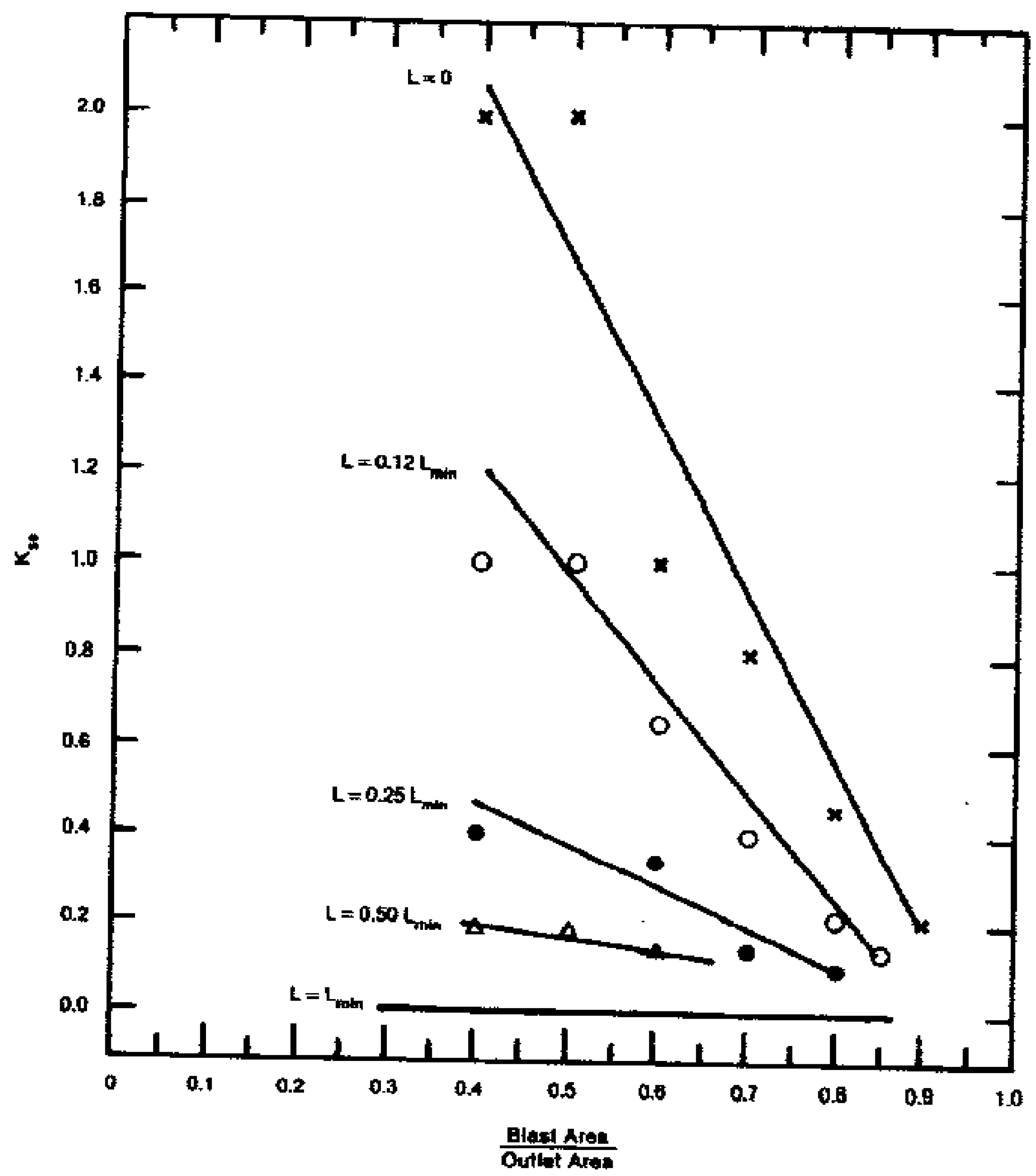


Figure 4-1. Effect of Discharge Duct Length L on K_{se}. Data from [2.1].

As far as possible, any dampers, elbows, fitting, branch ducts, splitters, should be located beyond L_{min}. Otherwise losses due to system effects will occur.

b. Elbows. If any elbows are required in the fan discharge, they should be located beyond L_{min}. Closer location will increase the losses. Elbows which rotate air in the same direction as the fan rotation have the least losses. Values of coefficient K_{se} are shown in Figure 4-2 for the worst elbow arrangement.

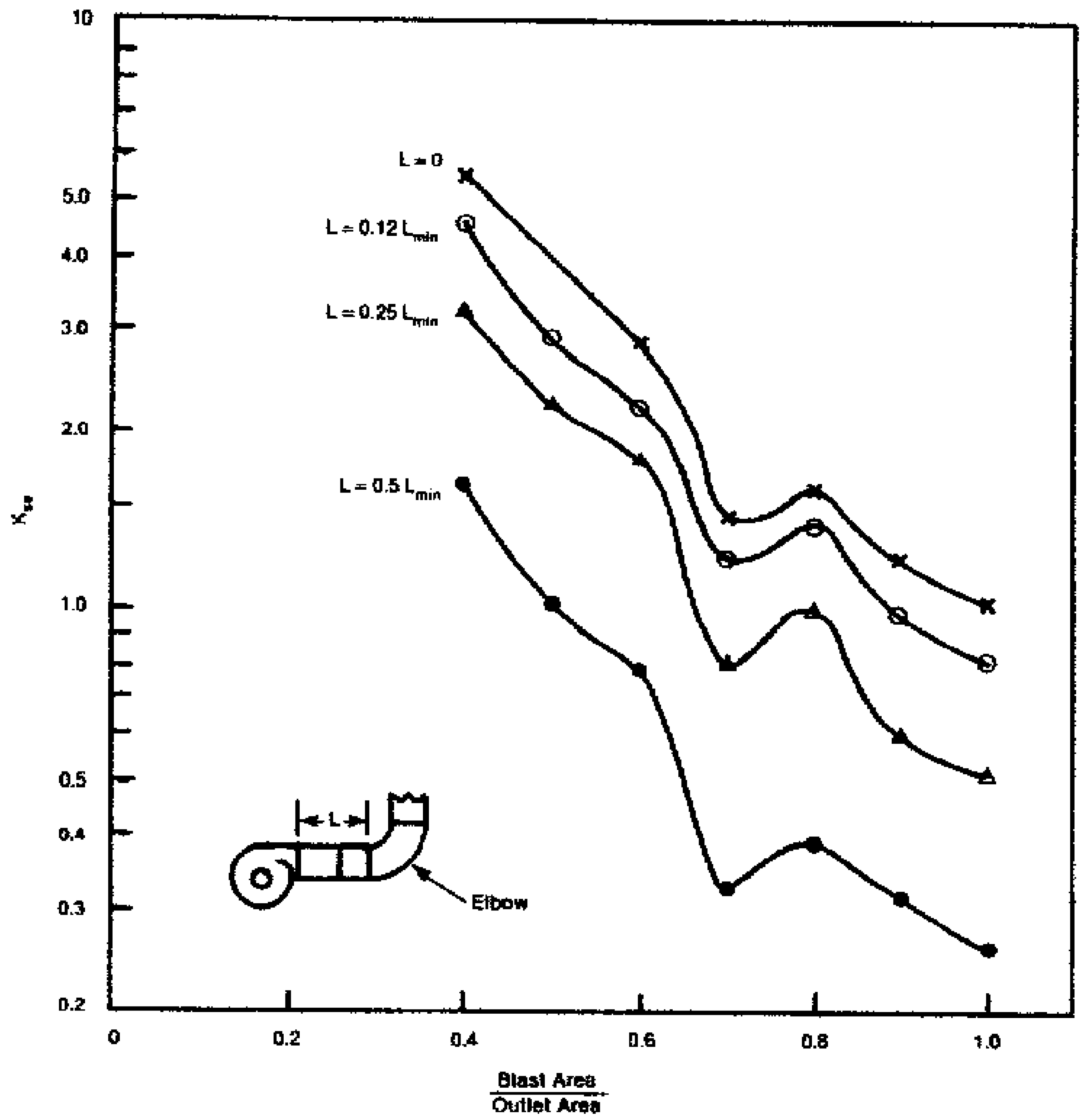


Figure 4-2. Effect of Elbow Close to Fan Discharge on System Effect Coefficient. Data from [2.0].

c. Discharge Dampers. For centrifugal fans, discharge dampers with blades perpendicular to the fan shaft generally give better results. The location of dampers close to fan discharge will increase the pressure drop of the damper above its catalog rating. The increase in pressure drop of dampers can be calculated by the following equation:

$$\frac{\text{actual } \Delta p}{\text{catalog } \Delta p} = 1.2 \left(\frac{\text{outlet area}}{\text{blast area}} \right)^2 \qquad (4\text{-}10)$$

Opposed blade dampers are preferable to parallel blade dampers as the latter result in very non-uniform velocity distribution and cause excessive pressure drop downstream.

2. Inlet Connection

Inlet conditions have a more profound effect on fan performance than outlet conditions. To obtain the rated performance, air should be uniformly distributed over the inlet area and enter in an axial direction without pre-rotation. A straight duct minimum five duct diameters long is desirable at fan inlet. The shorter the length of straight duct, the more severe the loss in fan performance. The losses increase with increasing air velocity.

a. Elbows. Elbows close to fan inlet cause non-uniform distribution of air. Some parts of impeller will be starved of air, causing loss in performance. If elbows close to fan inlet are unavoidable, they should be provided with turning vanes to minimize air rotation.

Figure 4-3 shows K_{se} for a square elbow close to fan inlet. It is seen that pressure losses are reduced to about one third if the elbow contains turning vanes. See [2.1] for data on several other types of elbows.

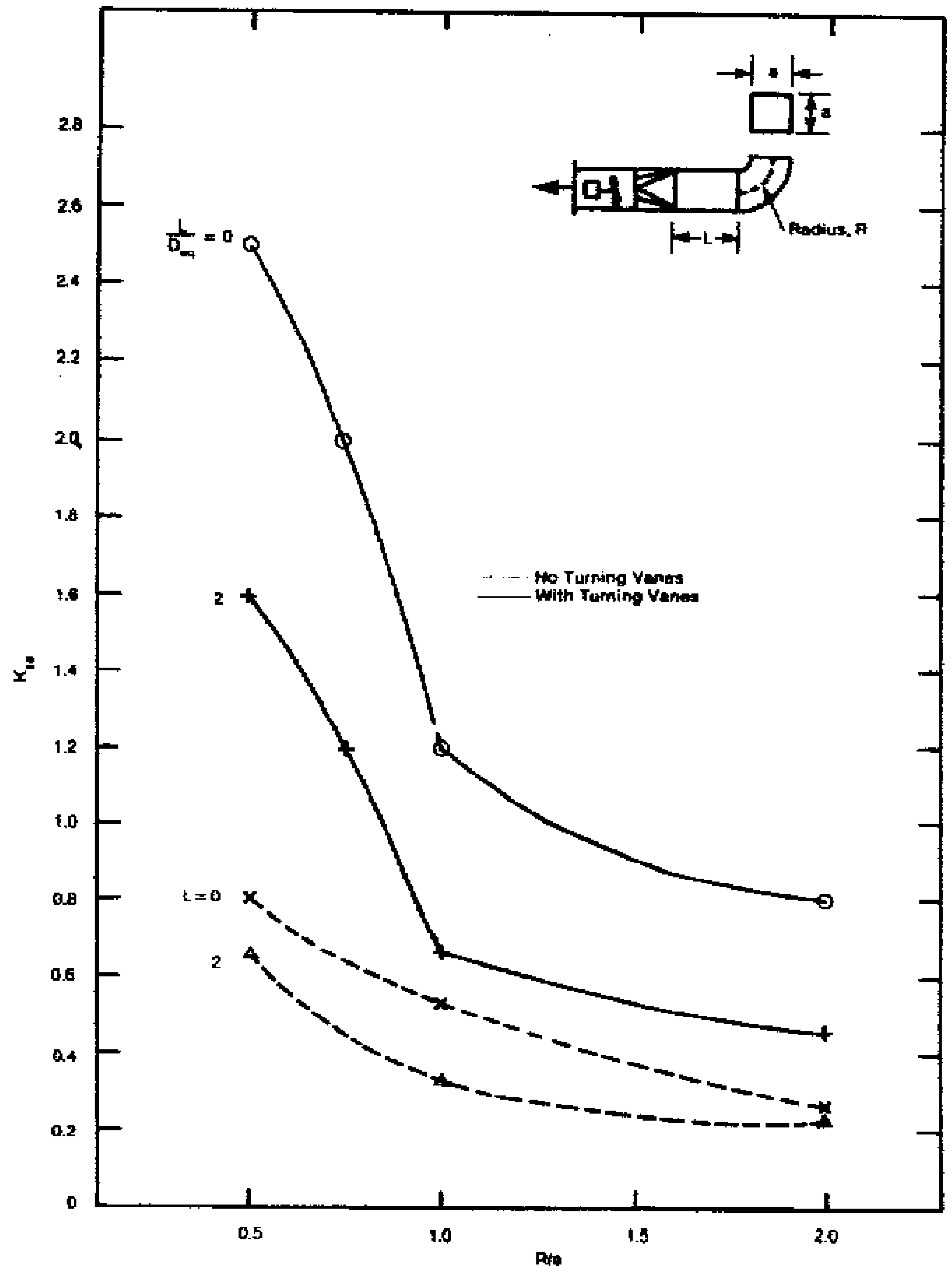
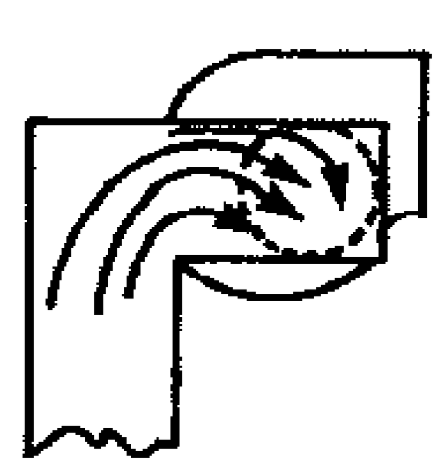


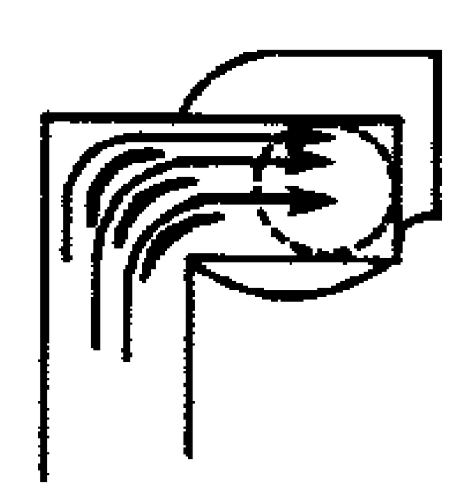
Figure 4-3. System Effect Coefficient for Square Elbow at Fan Inlet. D_{eq} by Eq (4-9). Data from [2.1].

b. Sharp-Edged Entrance. Flow into a sharp-edged orifice causes the formation of vena-contracta, resulting in loss of flow area. Fans without inlet duct connections, for example those directly mounted into a plenum, should be provided with a rounded entry. Otherwise loss in performance will occur due to reduction in inlet flow area.

c. Swirling Flow at Inlet. A poorly designed air inlet box can cause air to be swirling at fan inlet. If the rotation of air is in the same direction as the fan rotation, it causes decrease in flow and pressure but the efficiency is comparatively little affected. If the rotation of air is in the direction opposite to that of the fan rotation, fan power consumption is increased while the flow and pressure are largely unaffected. Use of turning vanes is helpful in reducing the swirl. Figure 4-4 shows an example of an inlet connection causing swirl and its mitigation with use of turning vanes.



(a) No turning vanes. Air swirling at inlet.



(b) Turning vanes in elbow. Swirl at inlet eliminated.

Figure 4-4. Prerotation of Air at Fan Inlet and Its Mitigation by Use of Turning Vanes.

d. Obstructions at Inlet. Obstructions at fan inlet may be caused by bearings and their supports, belt-guards, etc. The value of K_{se} for such obstructions is related to the reduction in flow area and is shown in Figure 4-5. The value of V' used in Eq. (4-7) should be based on the unobstructed area, i.e., the total fan inlet area minus the area obstructed.

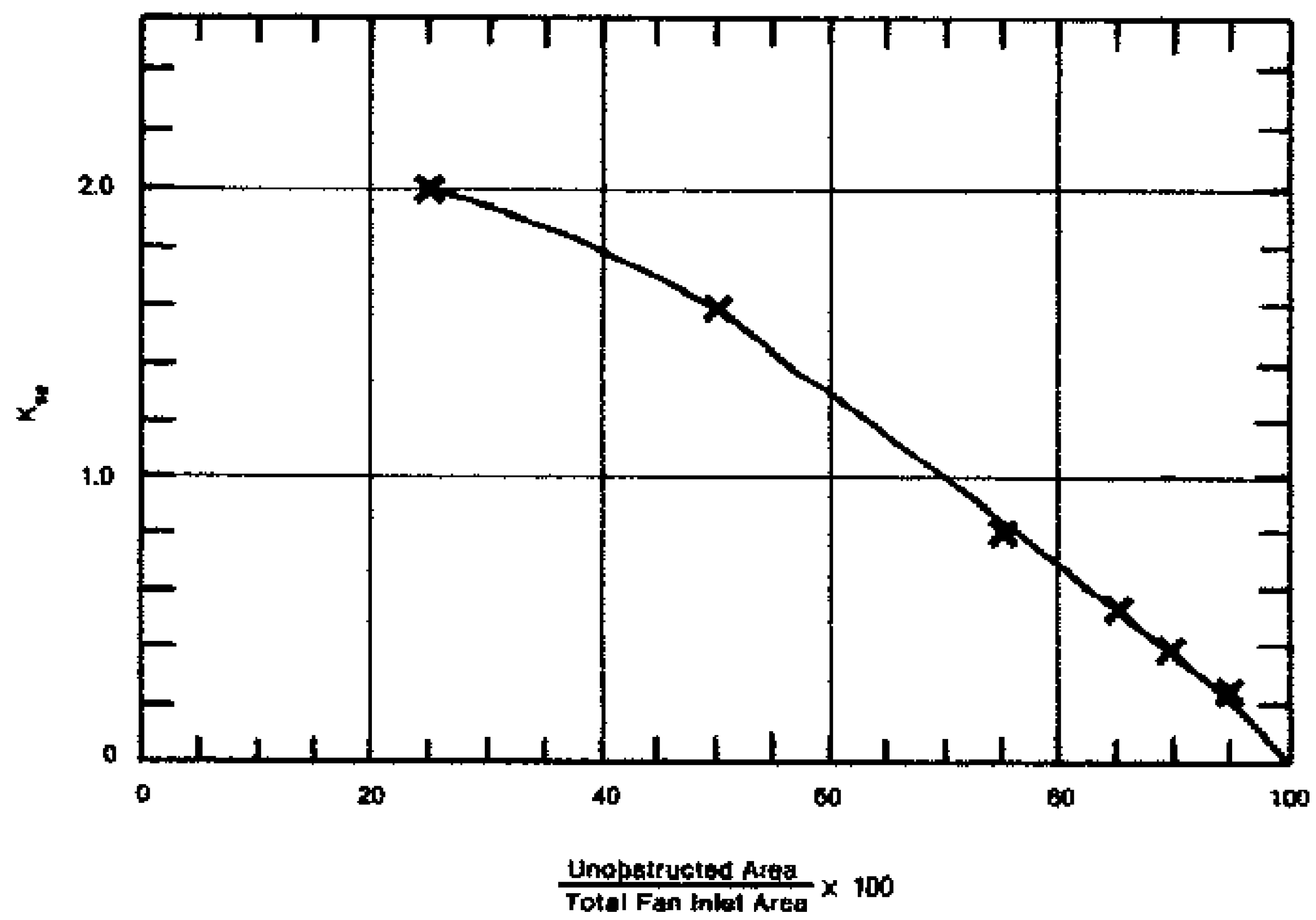


Figure 4-5. System Effect Coefficient for Fans With Obstructed Inlets. Data from [2.1].

Example

Various obstructions at the inlet of a fan cover 50 percent of the inlet area. The total area of fan inlet is 10 ft². The total flow rate is 25,000 cfm. Calculate losses due to system effects.

Solution

The free area is 50 percent of 10 ft^2 = 5 ft^2

$V' = 25{,}000/5 = 5000$ fpm

For 50 percent free area, Figure 4-5 gives $K_{se} = 1.6$ from Eq (4-7),

$$\Delta p_{fse} = 1.6\left(\frac{5000}{4000}\right)^2 = 2.5 \text{ inches of water}$$

e. Various Inlet Fittings. Table 4-2 gives the values of K_{se} which may be used in the absence of manufacturer's data:

Table 4-2

Item	K_{se}
Well designed inlet box	0.60
Inlet box damper	0.60
Inlet vane control, integral	1.40
Inlet vane control, add on	0.80

f. Fans Within Cabinets. Fans within cabinets should be located to permit unobstructed flow into the inlet area. Loss in performance will occur otherwise. The system effect coefficient for this situation can be calculated as follows:

$$K_{se} = 0.16\left[\frac{\text{fan inlet diameter}}{\text{distance of fan inlet from wall}}\right]^{1.25} \tag{4-11}$$

The distance between fan inlet and a wall should be at least 0.6 times the impeller diameter. If two double inlet centrifugal fans are in one cabinet, their inlets should be at least 1.2 times the impeller diameter apart [4.1]. The relative position of cabinet inlet and fan inlet should be such that it does not cause rotation of air at fan inlet.

C. Total Pressure Drop

The total pressure drop required to be developed by the fan is obtained by adding together the flow passage pressure drop and the pressure drop due to system effects. It must be recognized that the system effects cannot be accurately calculated until the fan has been selected. Preliminary estimates of system effects may have to be modified after fan selection has been made. Hence some trial and error may be required.

III. SYSTEM CURVE AND FAN OPERATING POINT

In most practical systems, flow is turbulent. In turbulent flows, pressure drop is approximately proportional to the square of the flow rate. Hence once the pressure drop at the design flow has been determined as described in the foregoing, that at other flow rates can be easily calculated by the following relation:

$$\frac{\Delta p_1}{\Delta p_o} = \frac{Q_1^2}{Q_o^2} \tag{4-12}$$

A plot of Δp at various values of Q gives the system curve. Its intersection with the fan curve gives the fan operating point, as shown in Figure 4-6.

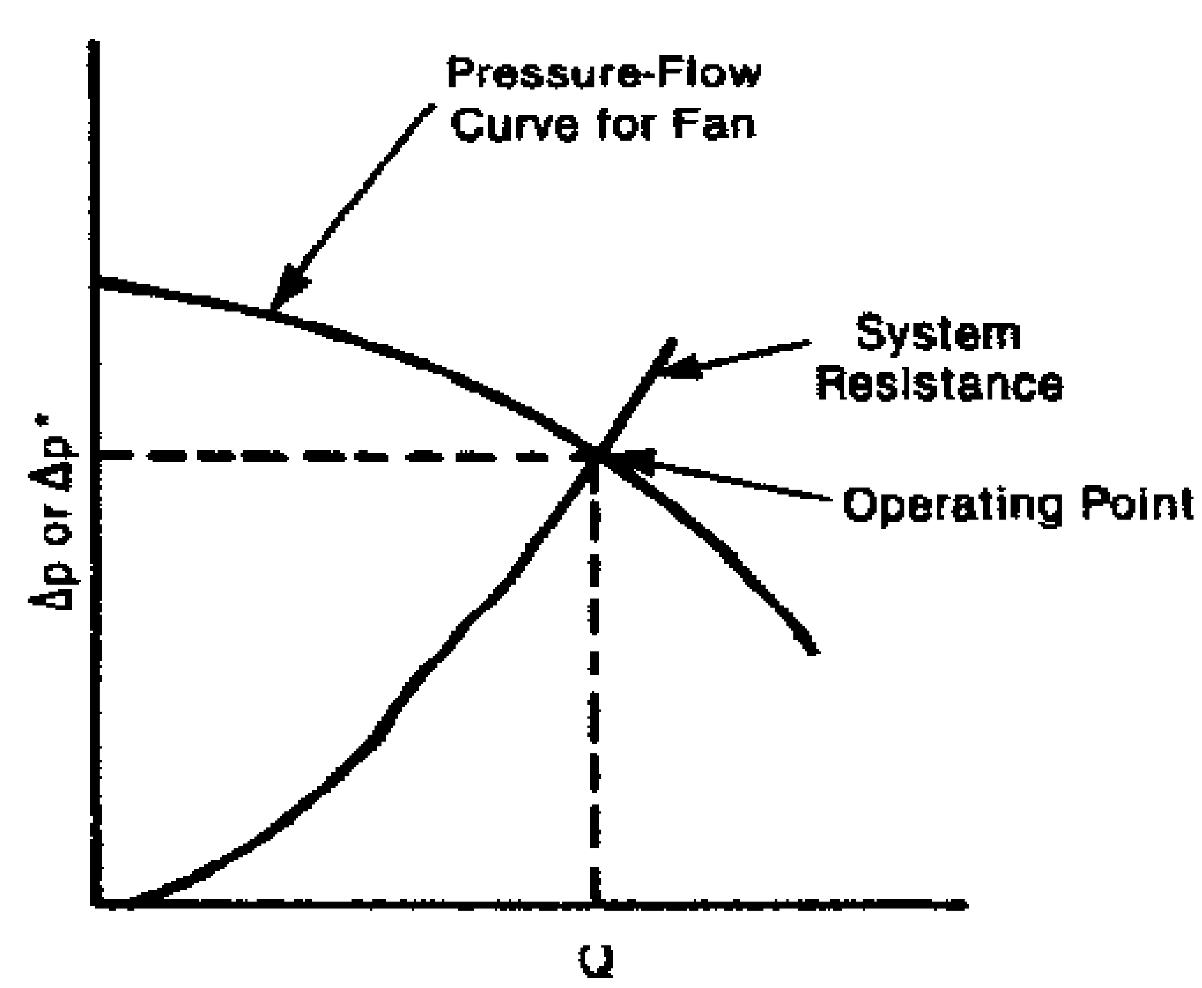


Figure 4-6. Determining Operating Point.

I. FAN LAWS

Fan laws are a set of equations which relate the performance of geometrically similar fans. Two fans are considered to be geometrically similar if all dimensions of one fan are in a fixed ratio to the corresponding dimensions of the other fan. In other words all the shape factors $S_1, S_2, \ldots S_n$, in Eq. (1-10) should be equal for the two fans. Appropriately enlarged photographs of geometrically similar fans will be indistinguishable from one another.

The fan laws for geometrically similar fans may be expressed by the following equations:

$$Q_a = Q_b \quad \frac{D_a^3}{D_b^3}\frac{n_a}{n_b} \tag{5-1}$$

$$\Delta p_a^* = \Delta p_b^* \quad \frac{D_a^2}{D_b^2}\frac{n_a^2}{n_b^2}\frac{\rho_a}{\rho_b} \tag{5-2}$$

$$P_a = P_b \quad \frac{D_a^5}{D_b^5}\frac{n_a^3}{n_b^3}\frac{\rho_a}{\rho_b} \tag{5-3}$$

where Q is the flow rate, Δp^* the total fan pressure, D the impeller diameter, and p the density. The subscripts a and b signify fan a and fan b respectively. These equations are dimensionless and any consistent units may be used. Through algebraic manipulations, these equations may be written in various other forms. See [1.3] for a number of alternative forms of fan law equations.

These laws apply to geometrically similar fans at the same rating point. Thus if data are available for fan b at 50% of the free delivery flow, the performance of fan a at 50% of the free delivery flow can be calculated by these laws. At the same point of rating, the efficiencies of geometrically similar fans are assumed equal.

II. LIMITATIONS OF FAN LAWS

As is apparent from Eq. (1-10), complete similarity requires that in addition to geometrical similarity, the Reynolds number and Mach number of the two fans be also equal. Generally, the effects of these parameters are small. However, when the geometrical scaling factor is very large, the variations in Reynolds and Mach numbers may have significant effects and the fan laws may break down.

The fan laws are based on the assumption that the fan efficiency is constant. In fact, fan efficiency generally increases slightly with increasing size. According to AMCA Standard 210, the effect of size on total efficiency may be approximately calculated by the following formula:

$$\frac{1-\eta_a}{1-\eta_b} = 0.5 + 0.5\left[\frac{\text{Reynolds number of fan b}}{\text{Reynolds number of fan a}}\right]^{0.2} \tag{5-4}$$

The fan laws are based on the assumption that effects of fluid compressibility can be neglected. This is true only if pressure variations are small. According to [2.1], these laws will be accurate only if the ratio of absolute pressures developed by the two fans is less than 1.036.

III. USE OF FAN LAWS

These laws are very useful in calculating the performance of geometrically similar fans as well as the performance of a particular fan at different speeds and air densities. For fan selection and system design, one is generally interested in knowing the effect of speed and density on a particular fan. To the fan manufacturer, the ability to predict the effect of impeller diameter is also very useful.

If the complete fan curve at one speed and air density is known, the curves at different densities and speeds can be calculated using fan laws. Fan laws are applied to various points on the known curve to find corresponding points at a different speed or density. The calculated points are then joined to obtain the performance curve at the desired condition.

The use of fan laws is now illustrated with a few examples:

Example 1

A fan delivers 10,000 cfm at a total pressure of 3 inches wg when handling air at 70°F. Calculate its flow and pressure if the air is at 200°F. Also calculate the change in power consumption.

Solution

Density of air at 70°F is 0.075. Density at 200°AF,

$$\rho_a = \frac{0.075\,(460 + 70)}{(460 + 200)} = 0.0602$$

The speed and impeller diameter are unchanged. Hence $D_a = D_b$, $n_a = n_b$. Therefore Eq. (5-1) yields:

$$Q_a = Q_b, \text{ no change in flow}$$

Eq. (5-2) yields:

$$\Delta p_a^* = 3\left(\frac{0.0602}{0.075}\right) = 2.408 \text{ inch wg}$$

Eq. (5-3) yields:

$$P_a = P_b \frac{0.0602}{0.075} = 0.802\, P_b$$

Thus the increase in air temperature from 70°F to 200°F causes no change in air flow, but pressure and power consumption are reduced by 20%.

Example 2

A fan delivers 100,000 cfm at a total pressure of 6 inch wg when operating at 1800 rpm. Calculate its flow and pressure when operating at 900 rpm. Also calculate the change in power consumption.

II. FAN NOISE LAWS

Laws for relating the noise from geometrically similar fans are available [1.3]. The most important relation is the following:

$$PWL_a = PWL_b + 70 \log_{10}\left(\frac{D_a}{D_b}\right) + 50 \log_{10}\left(\frac{n_a}{n_b}\right) + 20 \log_{10}\left(\frac{\rho_a}{\rho_b}\right) \tag{6-5}$$

The subscripts "a" and "b" indicate for fan a and fan b.

Using the fan laws given in Section 409.5 the above equation can be changed to express sound power level in terms of other parameters such as fan horsepower, cfm, and pressure. For these alternative forms, see [1.3].

Example

A fan operating at 3600 rpm in a fixed duct system has a sound power level of 60 db. What will be the sound power level if the fan speed is reduced to 1800 rpm?

Solution

$$\begin{aligned} PWL_a &= 60 + 70 \log_{10}(1) + 50 \log_{10}\left(\frac{1800}{3600}\right) + 20 \log_{10}(1) \\ &= 60 + 0 - 15.05 + 0 = 44.94 \text{ db} \end{aligned}$$

III. NOISE IN CENTRIFUGAL FANS

There are, in general, two major sources of noise in centrifugal fans. First is noise caused by periodic interruption of air flow such as that caused by the fan blades passing near a stationary projecting surface. Second is noise caused by eddies created by the fan blades as a result of sudden or nonuniform acceleration of the gas. Both classes of noise tend to increase with increase in peripheral speed.

To guard against the first cause of noise it is necessary to avoid having the fan blades pass close to projections. Also the blades should not pass close to a series of openings. In general, any condition that would cause high frequency interruptions of flow must be avoided. Noise caused by fan blades passing the cut-off point in an involute housing or the blades of a diffuser comprises a serious handicap to the most effective use of these types of housings.

Noise caused by eddies produced by the fan blades can be decreased only by careful attention to such details as blade shape and entrance and exit conditions, that is, the avoidance of sharp bends and projections into the air stream. Sharp corners should be filleted.

Careful attention should be paid to the mechanical resonances of the blades. These should not occur near rotational speed or at any frequency that is a multiple of rotational speed times number of blades.

Frequently the noise output of a fan discharging into a duct can be decreased appreciably by using a canvas connection between the fan outlet and the duct. Acoustic lining of the duct walls may also be necessary. If the system resistance has a cavity resonance or "organ pipe" frequency which matches the product of speed and number of blades, "blooping" noises are likely to occur.

Fans with larger number of blades run quieter than fans with small number of blades. Furthermore, in fans with small number of blades, there is the possibility of resonance with portions of building [1.1].

Solution

$$Q_a = 100{,}000 \frac{900}{1000} = 50{,}000 \text{ cfm}$$

$$\Delta p_a^* = 6\left(\frac{900}{1800}\right)^2 = 1.5 \text{ inch wg}$$

$$P_a = P_b\left(\frac{900}{1800}\right)^3 = 0.125\ P_b$$

Thus at half speed, the flow will be ½, pressure ¼, and power consumption ⅛, of those at full speed.

Example 3

A fan with an impeller with a 6 inch diameter delivers 1000 cfm. Calculate the cfm delivered by a geometrically similar fan which has an impeller of 12 inch diameter at the same speed and density. Also calculate the change in pressure and horsepower.

Solution

From Eq. (5-1)

$$Q_a = 1000\left(\frac{12}{6}\right)^3 = 8000 \text{ cfm}$$

From Eq. (5-2)

$$\frac{\Delta p_a^*}{\Delta p_b^*} = \left(\frac{12}{6}\right)^2 = 4$$

From Eq. (5-3)

$$\frac{P_a}{Pb} = \left(\frac{12}{6}\right)^5 = 32$$

Thus doubling the fan impeller diameter increases the flow rate 8 times, the pressure 4 times, and the power consumption 32 times.

I. SOUND POWER AND SOUND PRESSURE

Sound power level (PWL) of a fan (or any other source) is given by the following relation:

$$PWL = 10 \log_{10} \frac{\text{Sound power output in watts}}{10^{-12} \text{ watts}} \tag{6-1}$$

The PWL calculated with Eq. (6-1) is in decibels (db).

The sound pressure level (SPL) is referred to 0.0002 microbar and is given by the following relation:

$$SPL = 10 \log_{10} \left[\frac{\text{sound pressure in microbar}}{0.0002 \text{ microbar}}\right]^2 \tag{6-2}$$

SPL given by Eq. (6-1) is also in decibels. However, SPL and PWL are different quantities. PWL is independent of distance from sound source while SPL varies inversely as the square of distance. For a spherical free field, with distance X in feet, and at standard temperature and pressure, the following relation exists [1.3]:

$$PWL = SPL + 20 \log_{10} X + 0.5 \tag{6-3}$$

A. Specific Sound Power Level

Specific sound power level (PWL_S) of a fan is defined as the sound power level generated when operating at a capacity of 1 cfm and a total pressure of 1 inch of water.

Knowing the specific sound power level of a fan, the sound power level at operating pressure and flow may be determined by the following equation:

$$PWL = PWL_S + 10 \log_{10} Q + 20 \log_{10} \Delta p_t^* \tag{6-4}$$

Eq. (6-4) also permits the calculation of specific sound power level if the sound power level at a particular flow and pressure is known. Furthermore the change in sound power level of a fan due to change in flow and pressure can also be calculated by this equation.

Figure 6-1 shows typical specific sound power levels of different types of fans [6.1].

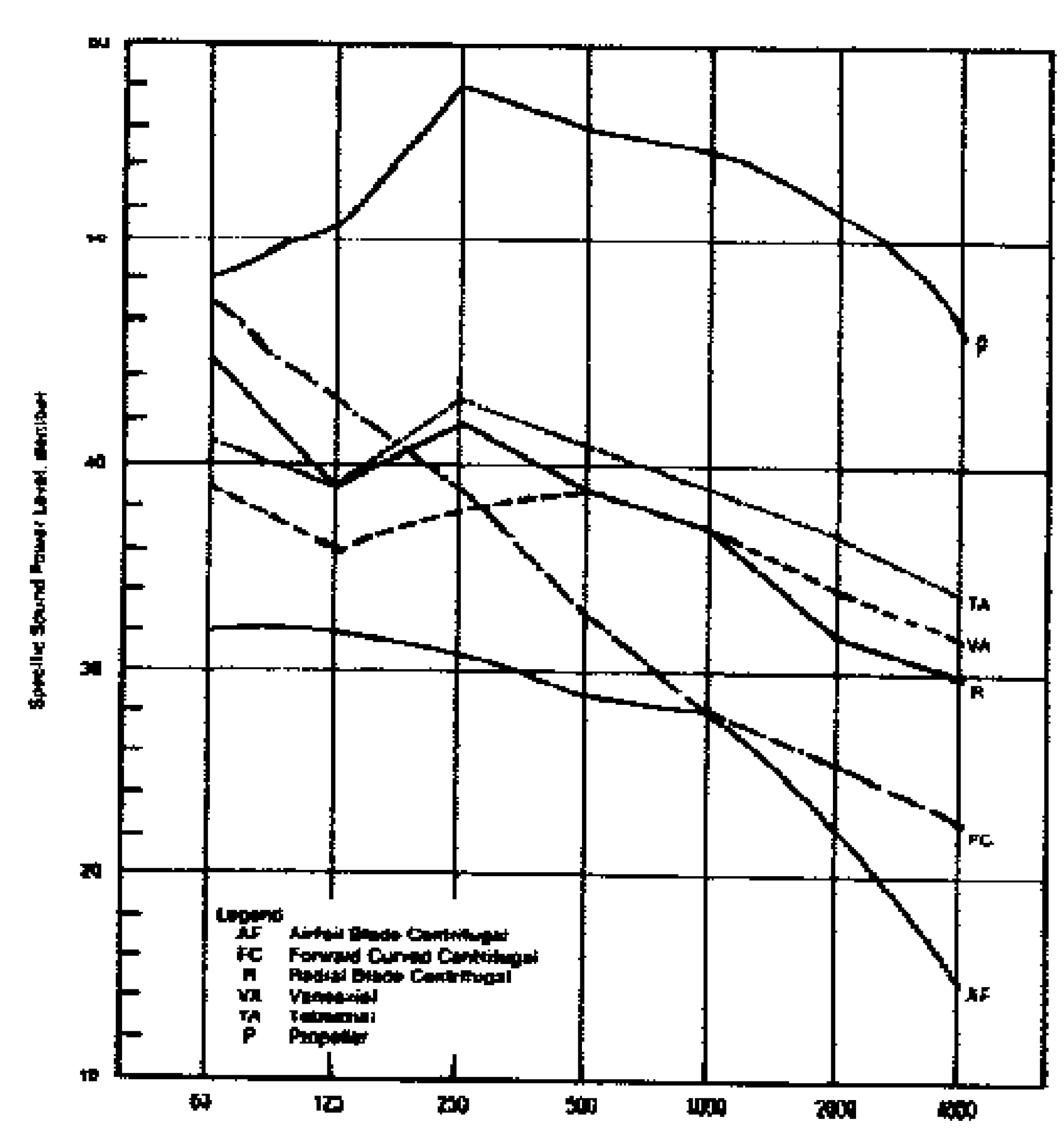


Figure 6-1. Typical Specific Sound Power Levels for Fans of Different Types. Data from [6.1]. Curves for Propeller and Forward Curved Centrifugals Apply to All Sizes. Other Curves Apply to Diameters Greater Than 36 Inch.

A fan makes the least noise when operating at highest efficiency. As can be seen in Figure 6-1, fans with airfoil blades generally make the least noise. Operation of fans under unstable conditions can generate noise. Vibrations due to poor installation can also cause noise. These points should be considered in fan selection, system design, and installation. For additional information, see[1.1], and [6.1] to [6.20].

IV. NOISE IN AXIAL FLOW FANS

Axial flow fans are generally more noisy than centrifugal fans. This often discourages their use in air conditioning systems having low noise criteria. Accoustical treatment is often required.

The number of impeller blades should differ from the number of guide vanes as equal number of the two results in higher noise level. The following formula may be used to determine the number of guide vanes [1.1] (page 488).

$$N_{guide} = \frac{N_{impeller}}{1 \pm 1/N_{impeller}} \tag{6-5}$$

Noise increases with increasing clearance between blade tip and housing [3-4]. Variations of clearance around the circumference also increase noise [1.1]. Hence tip clearance should be kept small and as uniform as possible.

For additional information, see [1.1], [6.21], [6.22], and [6.23]. Also see [6.1] to [6.7], [6.8], and [6.12] to [6.20].

Testing of fans for noise is discussed in Section 409.11.

I. STEPS IN FAN SELECTION

Fan selection involves the following major steps:

1 Determine the flow and pressure requirements

2 Determine other selection criteria such as noise, maximum power consumption, space limitations, etc.

3 Select the type of fan to be used

4 Select a particular fan using manufacturer's catalogs

5 Select fan drive

Estimation of fan pressure and flow requirements is discussed in Section 409.4.

II. SELECTION OF FAN TYPE

Selection of fan type involves consideration of flow-pressure characteristics, power consumption, noise, space limitations, etc.

A. Propeller, Tubeaxial, Vaneaxial, or Centrifugal

Propeller fans are suitable only where the required pressure is low. Their application is generally limited to pressures less than 0.75 inches of water. At these low pressures, propeller fans are generally more economical than other types of fans.

Tubeaxial fans can develop fairly high pressure but their efficiency is low. They are used where low first cost is desired. Otherwise the more expensive and more efficient vaneaxial fans are used. Tubeaxial fans are also generally more noisy than vaneaxial fans.

Over a very wide range of pressures and flow rates, both centrifugal and vaneaxial fans of comparable efficiency are available. Choice between the two is often difficult. Some factors in choosing between the two types are now discussed.

Single stage vaneaxial fans are generally significantly cheaper than centrifugal fans. The weight of vaneaxial fans is about half that of a conventional centrifugal fan and about one third of the weight of the tube-centrifugal type. The lower weight results in savings in the cost of support structures and reduces structural vibrations. Vaneaxial fans require less space than conventional centrifugal fans. Vaneaxial fans can be directly inserted into the ductwork, without the bends required by centrifugal fans. This generally reduces the duct pressure drop and system effect losses (see Section 409.4). The tubecentrifugal fans also have the same advantages but they are even more expensive and heavy than conventional centrifugal fans.

A major disadvantage of vaneaxial fans is that they are generally noisier than centrifugal fans (see Section 409.6). This often discourages their use in commercial air conditioning systems. This is generally not a significant disadvantage in large industrial applications, since all high pressure, high flow fans will make unacceptably high noise and will require accoustic attenuation.

Where modulation of flow over a wide range is required, vaneaxial fans with automatic pitch control will be usually preferrable. This method of control is available only with axial flow fans and is much more efficient and less noisy than the variable inlet vane control used with centrifugal fans. The most efficient and quietest method of capacity control is speed modulation. If speed control is technically and economically feasible, the choice of fan has to be made on other considerations.

If flow control is to be done by discharge throttling, axial flow fans should not be used, since the throttling may push the operating point into the unstable range of the fan curve. See Section 409.9.

Vaneaxial fans are best suited for handling clean gases. They can handle air with small amounts of dust but get plugged and eroded when the air contains large amounts of particulates. For handling air with high particulate contents, centrifugal fans are generally the only choice.

B. Choice of Type of Centrifugal Fan

The choice between the three basic types of centrifugal fans, namely backward-inclined, radial, and forward-inclined blade type has been discussed in Section 409.2 and summarized in Table 2-1. Fans with backward-inclined blades of airfoil design are usually the most efficient and least noisy. They are generally also the most expensive. If air contains large amounts of solid particles, radial blades usually give the best service (see Section 409.10).

Forward-curved blade fans should be avoided if flow is to be controlled by discharge throttling as it can lead to unstable operation.

Tube-centrifugal fans are used where inline installation is more convenient or essential. Fans of this type are costlier, heavier, and less efficient than centrifugal fans with conventional scrolls.

Also see [7.1], [7.2], and [7.3].

III. SELECTING A PARTICULAR FAN

Once the type of fan has been decided upon, manufacturers' catalogs are consulted to choose a particular fan. Besides the obvious requirement that the fan should be capable of providing the design flow and pressure, other important considerations are efficiency, stability, and noise. Special design features such as easy maintenance access should also be considered.

Manufacturers publish their data in the form of tables and graphs which give the performance parameters in dimensional units. The most commonly used units are cfm for flow, inches WG for pressure, and horsepower for power consumption.

A fan should normally be selected such that the point of operation is in the maximum efficiency range. This not only minimizes the power consumption but also minimizes the noise, since, for a specific line of fans, those operating in the most efficient range will also be the quietest. Multi-rating tabulations in catalogs generally give selections only in the recommended range of operation and the maximum efficiency point is identified. Ref. [7.4] describes a method for representing fan performance data.

Figure 7-1 shows the selection range for good efficiency.

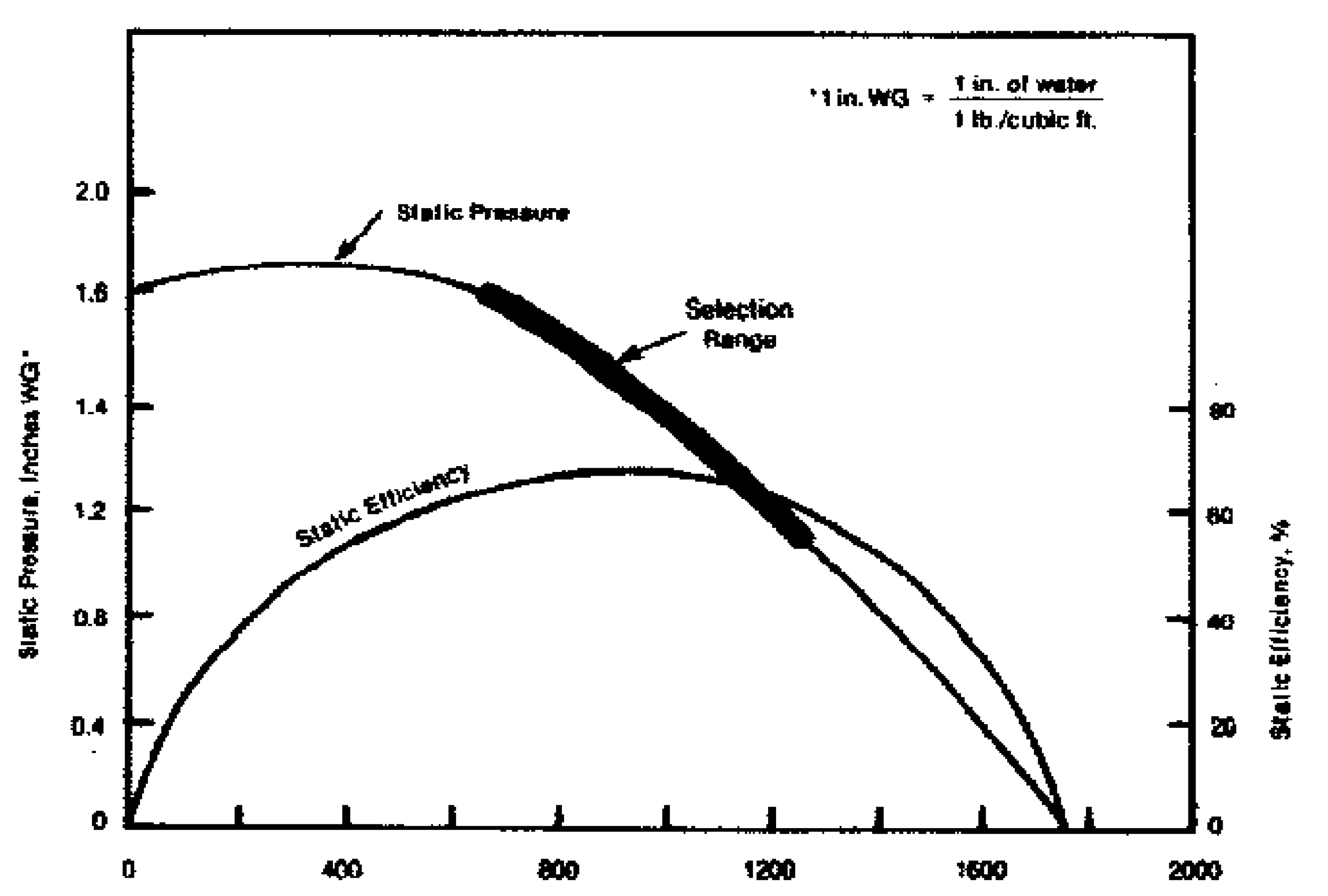


Figure 7-1. Typical Performance Curve for Backward-Inclined Centrifugal Fans Showing Selection Range for Good Efficiency.

A. Stability Consideration

Flow instability can be defined as rapid and persistent variations of flow and pressure. It results in noise and vibration and can even cause mechanical failure.

First of all, certain axial flow fans are completely unstable below a certain flow rate. Vaneaxial fans have curves similar to that in Figure 7-2(a) and are usually inoperable to the left of point A. Hence for such fans, the operating point should always be kept to the right of point A. If operation to the left of point A is intended with an axial fan, the manufacturer should be consulted to determine if stable operation is possible.

The forward-curved centrifugal fans have characteristics similar to that in Figure 7-2(a) but they can have stable operation to the left of point A provided the system curve intersects the fan curve at only one point. Thus system curves a and c intersect the fan curve only once and hence operation will be stable. On the other hand, curve b intersects the fan curve at three points and operation will be unstable. The operation will continuously shift between points e, f, and g. If capacity control by throttling is intended, one should be careful that a situation similar to system curve b does not arise.

Ideally, the system curve and fan curve should have opposite signs so that the opposing effects cancel each other, thus damping out disturbances. This is the situation with system curves a and c in Figure 7-2(a). As a minimum, the system curve should be considerably steeper than the fan curve at the point of intersection. Thus in Figure 7-2(b), which shows the curve of a typical backward-curved fan, operation can be stable with curve c even though the slope of fan curve at intersection has the same sign as the system curve. However, some backward-curved fans can have pulsations to the left of point A. Hence manufacturer should be consulted if operation in this region is being considered.

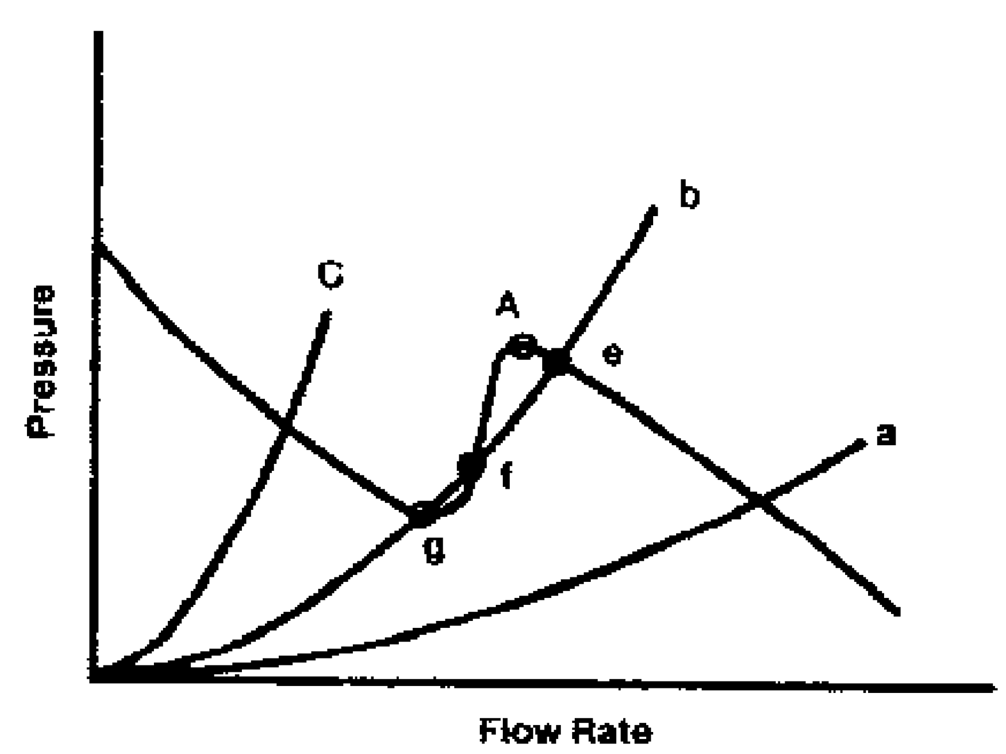


Figure 7-2(a). System Curve Intersecting Fan Curve at More Than One Point, Leading to Instability.

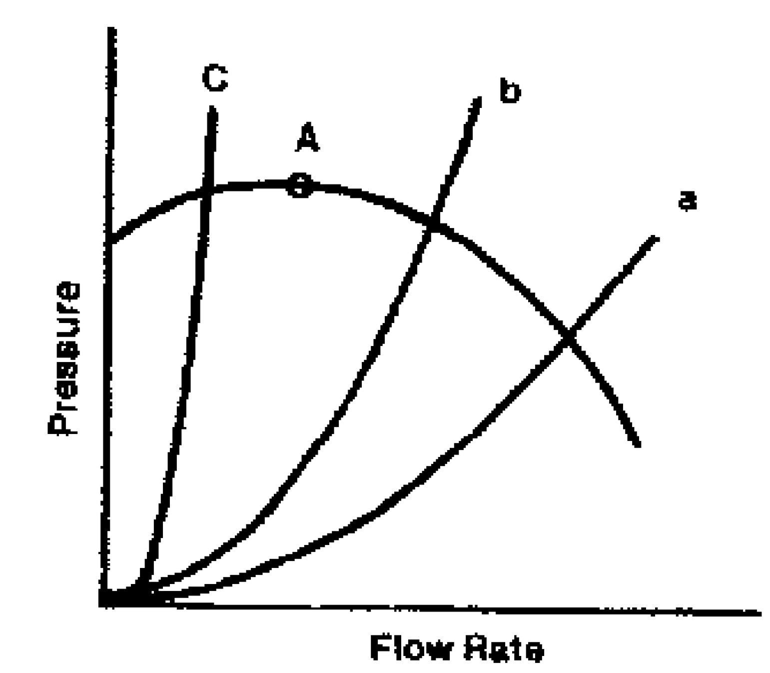


Figure 7-2(b). Fan Curve for Backward Inclined Centrifugal Fans. More than One Intersection with System Curve is Not Possible.

Parallel fan operation can lead to instabilities in all fans whose curves have a maximum pressure or points of inflection, if operated at lower flow rates. See Section 409.8 for a detailed discussion.

B. Densities Other Than Standard

Manufacturer's catalogs normally give the fan performance with air density at the standard value of 0.075 lb/ft.[2] This corresponds to dry air at 70°F at sea level. If the temperature, humidity or altitude are different, the density may be different and the fan performance will differ from that given in the catalogs. The effect of temperature and altitude on density of dry air is shown in Figure 7-3. It can also be calculated using the following formula:

$$\rho = \frac{1.328\ p}{(t + 460)} \qquad (7\text{-}1)$$

where ρ = density at desired conditions, lb/ft³

t = temperature of air in °F

p = absolute pressure inches of mercury

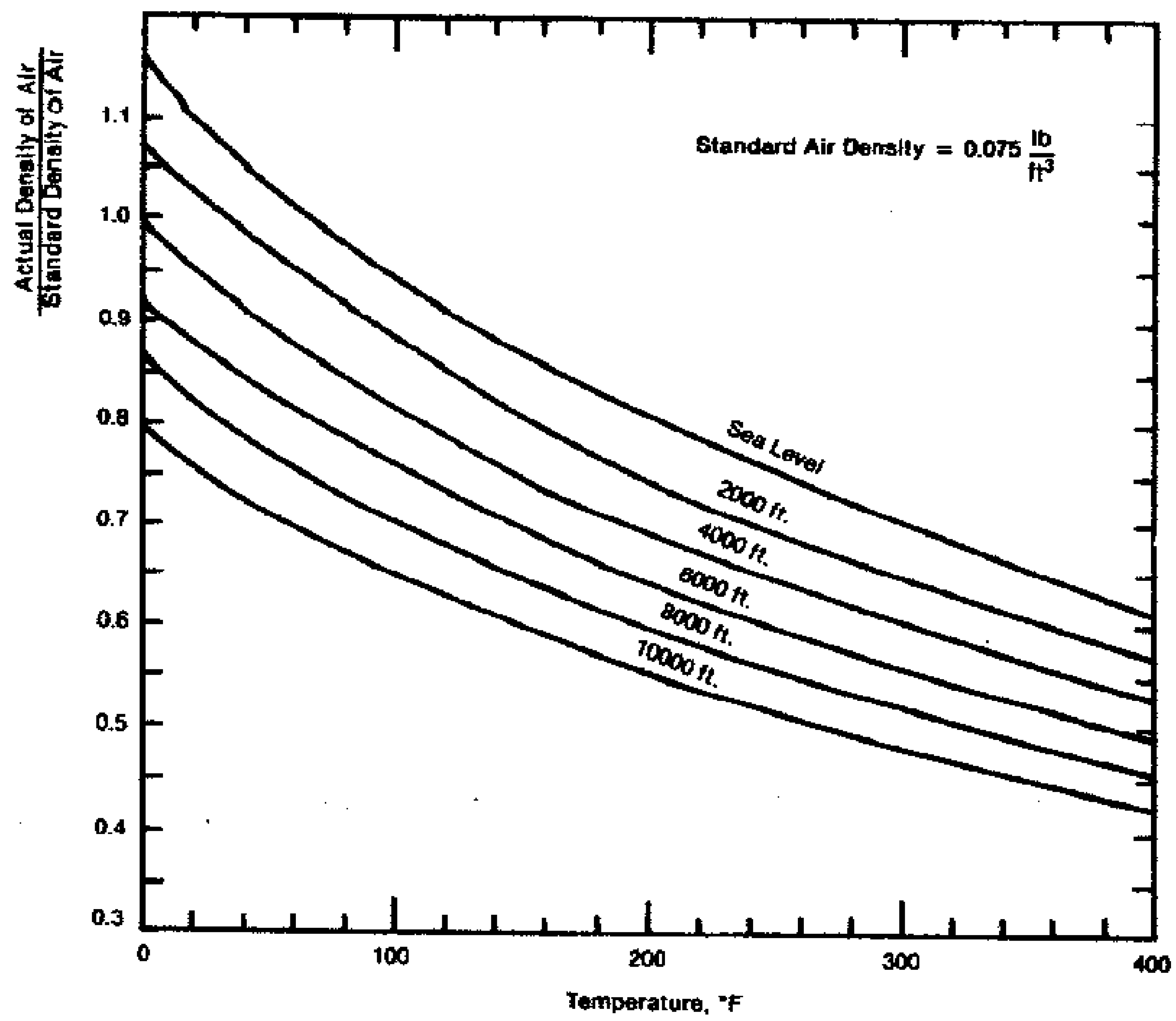


Figure 7-3. Effect of Temperature and Altitude on Density of Air.

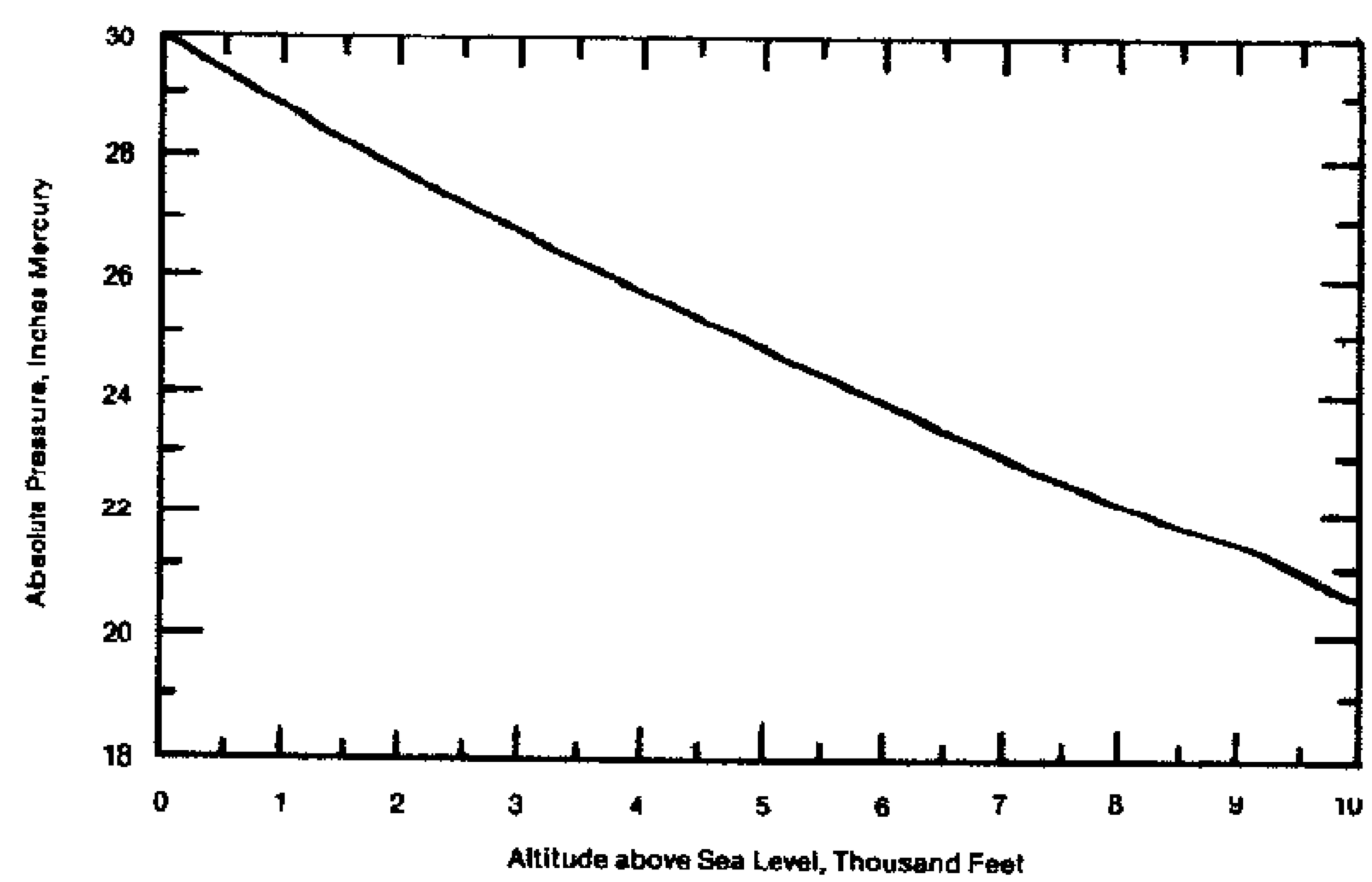


Figure 7-4. Variation of Atmospheric Pressure with Altitude.

Selection at densities other than 0.075 lb/ft² can be made in two ways. The first method is to calculate an "equivalent pressure drop" (EPD) for the system. The other method is to calculate the fan performance at the actual density using the fan laws given in Section 409.5.

The equivalent pressure drop method is more convenient when using tabulated performance data and when the fan will be used at a fixed flow rate. This method involves the following steps:

1 Calculate EPD as follows:

$$\text{EPD} = \text{Pressure drop at actual density} \times \frac{0.075}{\text{actual density}}$$

2 Enter tables with cfm and EPD to select the fan and read its rpm and bhp.

3 The bhp at actual density is calculated as follows:

$$\text{actual bhp} = (\text{bhp from table}) \times \frac{\text{acutual density}}{0.0075}$$

The rpm read from table does not need any adjustment.

When the manufacturer's data is in the form of curves and when the fan flow rate has to be varied, it is more convenient and useful to calculate out the fan performance at actual density using fan laws. Pressure and horsepower are calculated at closely spaced intervals, plotted on a graph and connected by smooth curves. The system curve is then plotted on the same graph to obtain the operating point. For variable flow rates, several system curves can be plotted to obtain the pressures and horsepowers over the entire range of interest. Figure 7-5 illustrates this method.

Note that the system curve is at the actual density. Also see [7.5].

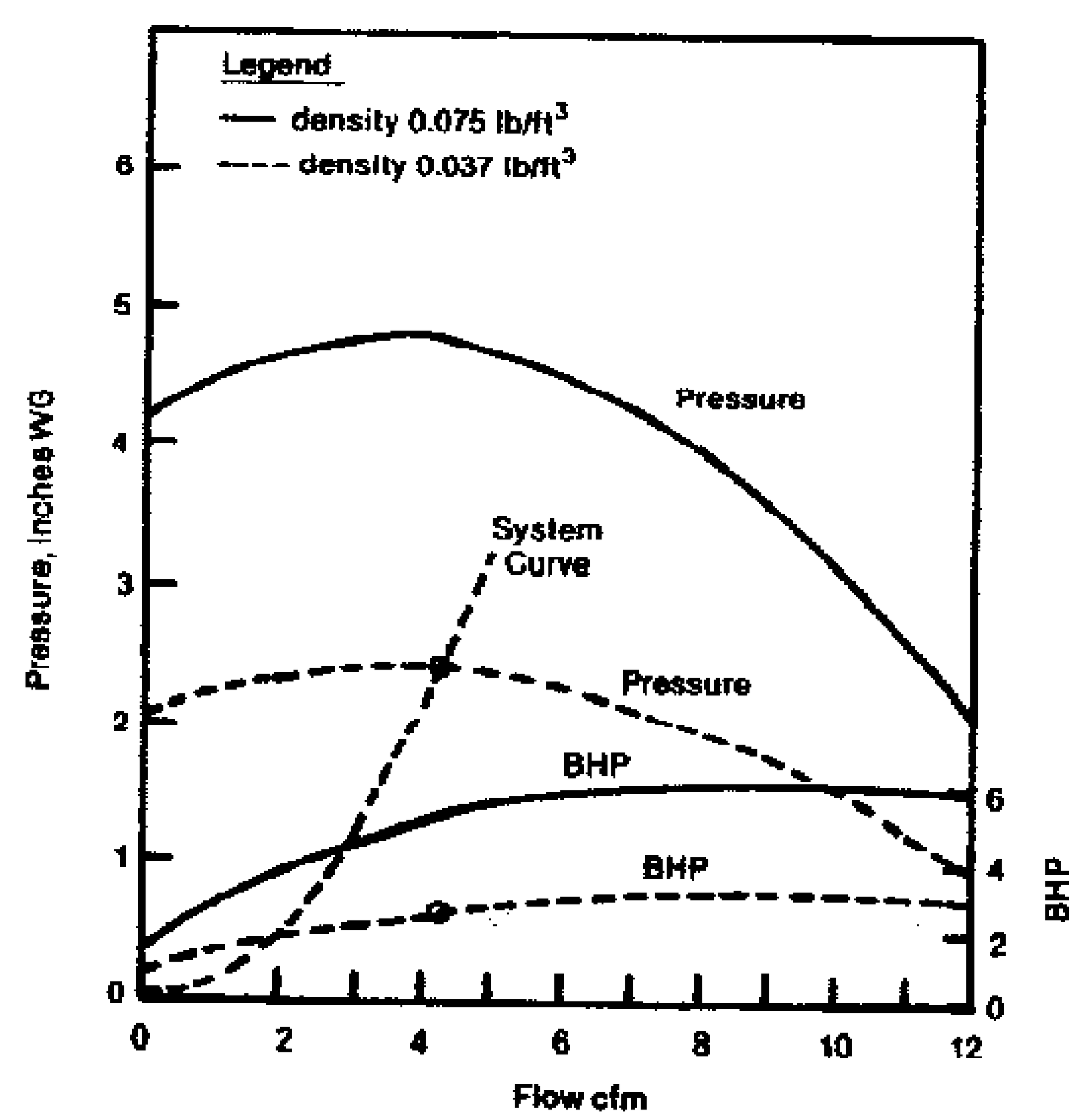


Figure 7-5. Use of Fan Laws to Determine Operating Point at Non-standard Air Densities. With 0.037 lb/ft³ Density Air, Operation is at 2.4 inch wg Pressure and BHP is 2.5.

IV. SELECTION OF FAN DRIVE AND MOTOR

A. Direct Drive or Belt Drive

A fan may be driven by an engine or turbine, Figure 7.6, but most commonly is driven by electric motors. The fan and motor may be directly coupled or be connected through a belt-drive. The advantages of direct coupling are that it is more compact, has no transmission losses, requires less maintenance, chances of failure are less, and the motor is cooled by the air stream to or from the driven fan. In direct drives, motor heat is added to the air stream. It can be a disadvantage in an air conditioning system. The belt drive has a transmission loss of a few percent. Regular maintenance is needed to replace worn belts and to adjust belt tension. The main advantage of belt drive is that the speed of the fan can be changed easily by simply replacing a couple of pulleys while the direct drive permits operation only at the motor speed. In many practical situations, system losses are found to be different from those calculated. The fan performance can then be easily modified by changing the fan speed.

B. Motor Types

The motors most commonly used are alternating current induction motors with squirrel cage rotor. These operate close to the synchronous speed. Motors are available with a variety of frames which fall into two broad categories, open and totally enclosed. See Figures 10-3 and 10-4 for some examples of motor enclosures.

The open motors permit free exchange of air between outside and inside while totally enclosed motors largely eliminate such exchanges. The most common open type enclosure is the drip proof enclosure. Open splash proof motors provide somewhat better protection. Open motors should be used only where air does not contain any harmful vapors or dust and where the possibility of large amounts of liquid falling over the motor can be excluded. Otherwise, totally enclosed motors should be used.

Totally enclosed motors are of two basic types, those which have an integral cooling fan, and those which do not have an integral fan. The totally-enclosed air-over (TEAO) motors do not have an integral fan. They require air to be blown over them for cooling. They are suitable for direct coupling to fans as the fan air has to pass over them. The totally-enclosed fancooled (TEFC) motor has an integral fan outside the enclosure but within a protective shield which circulates air over the enclosure to dissipate the motor heat. For motors located in combustible atmospheres, explosion proof enclosures are available which withstand any internal explosion and prevent sparks from reaching the surrounding combustible vapors. For motors located in atmospheres containing combustible dust, dust ignition proof enclosures are obtainable.

C. Motor Horsepower

Selection of motor horsepower requires careful consideration. The selected motor should be capable of providing the necessary power not only at the design point but also at any other point where the fan may be required to operate. Errors in pressure drop calculations may cause the fan to operate at higher or lower flow rates with the resultant effect on horsepower. Use of discharge throttling dampers can greatly affect the horsepower requirements. The fan's normal operation may be with hot gases of low density but it may have to run with low temperature gases during startup, needing much more power than at the design point. The fan bhp curve should therefore be studied over the entire range of possible operation and the selected motor should satisfy the maximum power demand. If the possibility of field adjustment of fan speed exists, the motor capacity should be selected with appropriate margin.

D. Motor Starting Torque

Another consideration in motor selection is the starting torque. The motor should provide sufficient torque to accelerate the fan to full speed within a short interval (usually 10 seconds). Otherwise the motor may be damaged or trip. It is possible that a motor may have enough horsepower for operating the fan at full speed but may have insufficient torque for starting.

Figure 7-6 shows typical motor and fan torque curves. The torque available for acceleration is the vertical distance between the two curves. The motor torque t in ft-lb. required to accelerate the fan in T seconds can be calculated as follows:

$$\tau = \frac{(WR^2)\, n_{fan}^3}{308T\, n_{motor}^2} \qquad (7\text{-}2)$$

where WR^2 = moment of inertia of fan lb/ft²

n_{fan} = fan speed, rpm

n_{motor} = motor speed, rpm

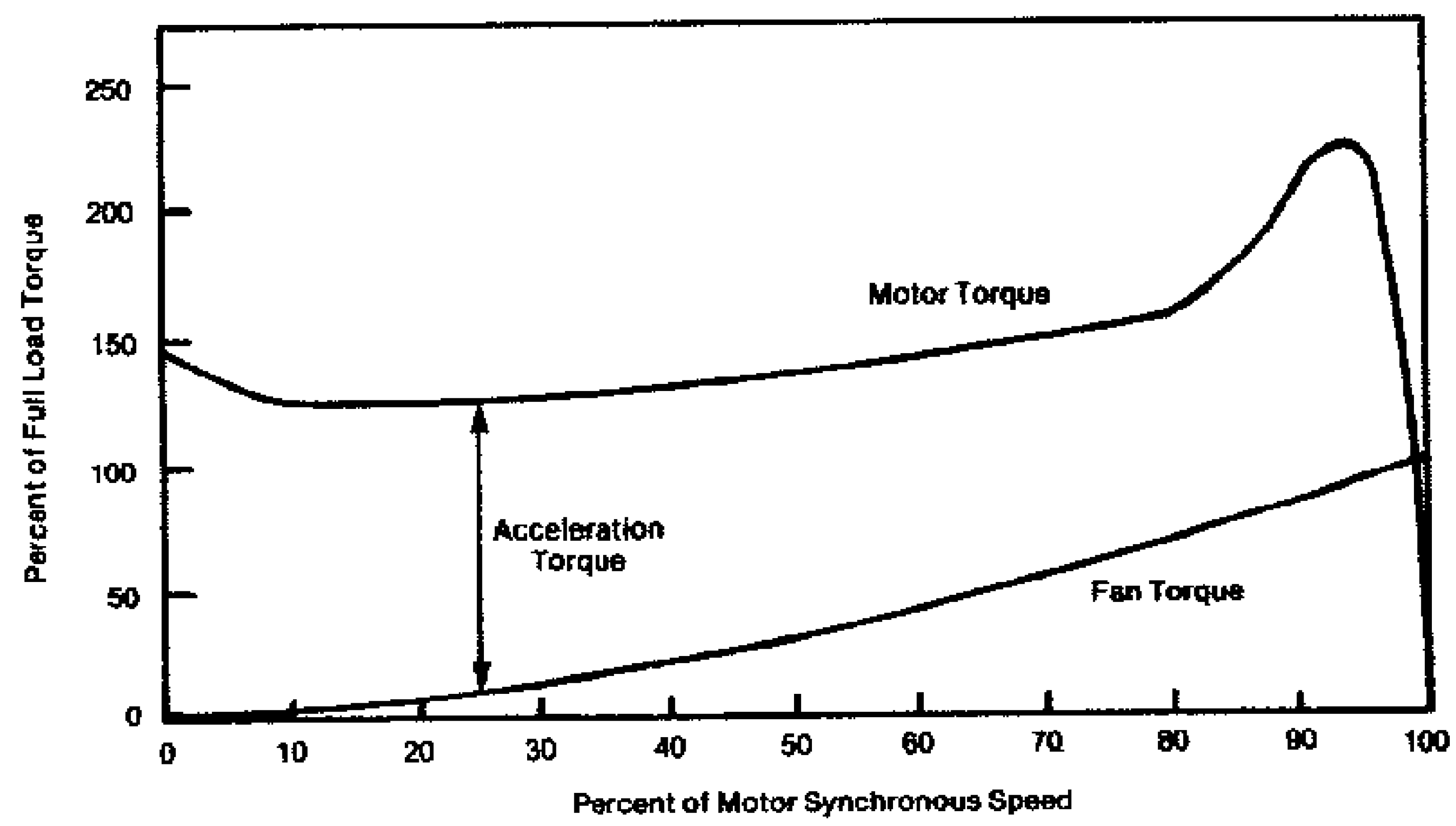


Figure 7-6. Typical Fan and Electric Motor Torque Curves.

Friction, aerodynamic drag, and drive inefficiency, increase the torque requirements. Hence the torque requirements given by the above formula should be increased by at least 10%. WR^2 data are readily obtainable from fan manufacturers and motor torque data can be obtained from fan manufacturers.

I. INTRODUCTION

Two or more fans are often used together to meet the system requirements. Fans may be arranged in series or parallel. In series arrangement, the flow through all fans is the same while the total system pressure drop is divided among them. In parallel arrangement, the total system flow is divided among the fans. The fans in either arrangement may be close together or be remote from each other. Figure 8-1 schematically shows the series and parallel arrangements.

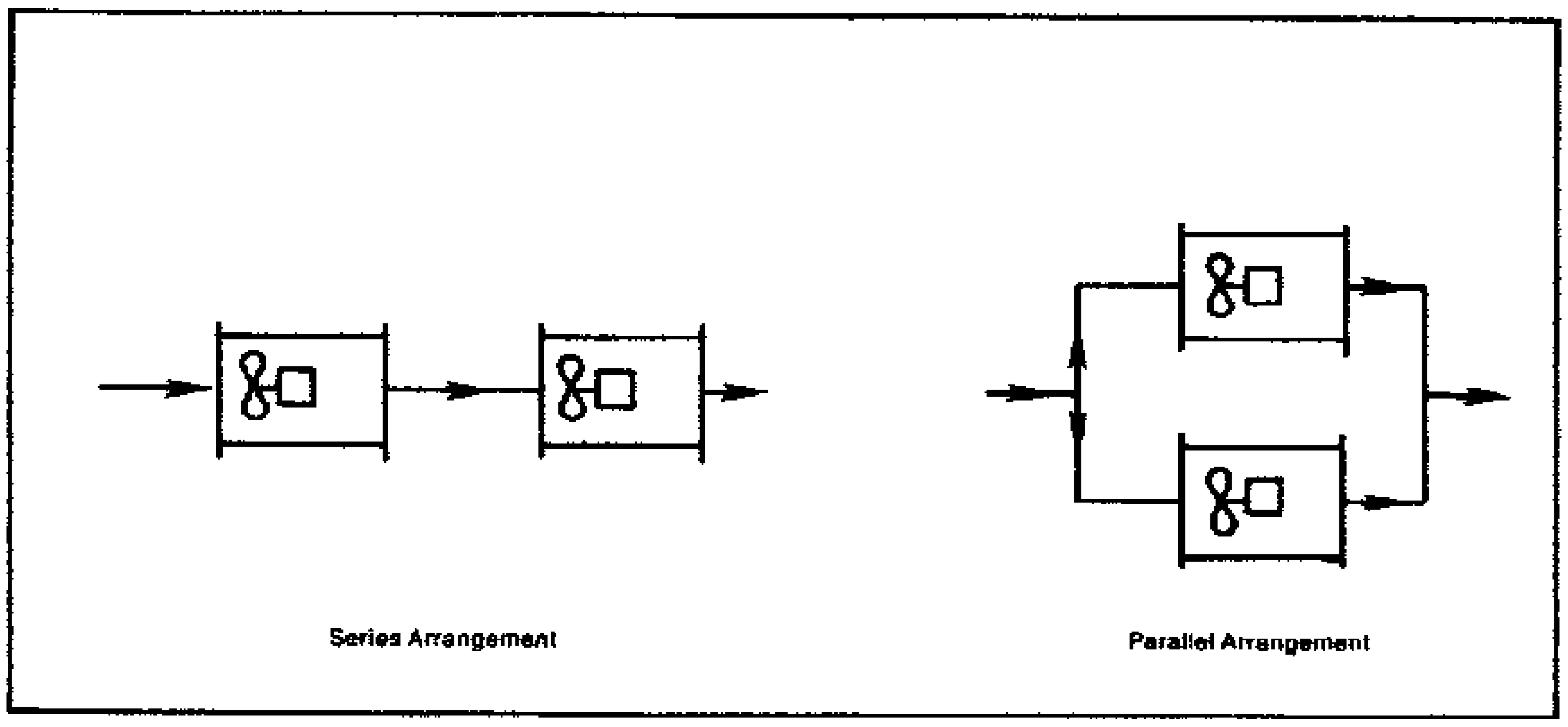


Figure 8-1. Series and Parallel Fan Arrangements.

There can be various reasons for using several fans in parallel to meet the system requirements. Use of two fans instead of one may save space. Many packaged air conditioning units use two fans for this reason. Use of several fans ensures that if one fan fails, the other fans will continue to provide enough air to meet some minimum requirements. Such arrangements may also be used for capacity control, some fans being shutdown during low flow requirements. Finally, a single fan with sufficient flow capacity may just not be available.

There can similarly be various reasons for using several fans in series. Multistage fans are simply several fans in series, arranged in a single housing. The system pressure drop could be more than could be handled by one fan alone. In air conditioning systems, separate supply and exhaust fans are often used to avoid overpressurizing the building. In boilers, separate supply and exhaust fans are often used to avoid excessive negative or positive pressures in the boiler.

II. FANS IN PARALLEL

The combined characteristic curve of several fans in parallel can be obtained by adding the flow rates at each value of pressure. If the fan curve has no maximum or point of inflection, the combined characteristic is obtained easily. Figure 8-2 shows the combined characteristic of two identical fans while Figure 8-3 shows that for two different fans in parallel.

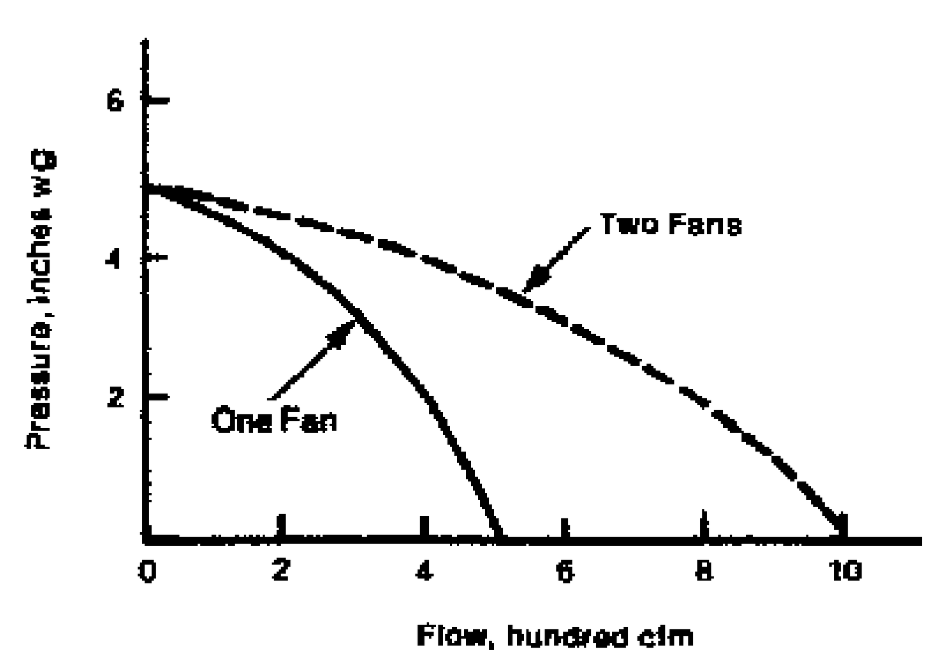


Figure 8-2. Performance Curve of Two Identical Fans in Parallel.

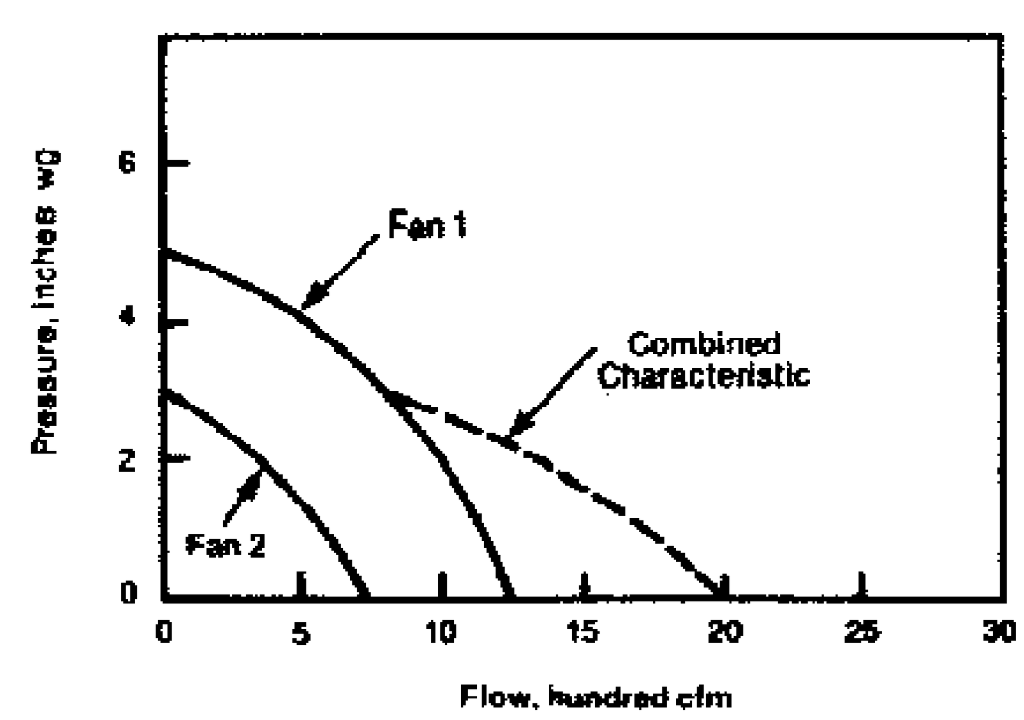


Figure 8-3. Performance Curve of Two Non-identical Fans in Parallel.

Complications arise when fan curves have a maximum or points of inflection, which is the case for most fans. In such fans, there can be more than one flow rate at a particular pressure, typically at lower flow rates. Thus the addition of flow rates at each pressure can be carried out in more than one way.

Figure 8-4 shows the characteristics of two identical fans. These characteristics are typical of axial fans and forward inclined centrifugal fans. At the lower flow rates, each fan curve has up to three flow rates at a single pressure. The combined characteristic curve is drawn by plotting all possible combinations of flow rates at each pressure. The resulting curve has a figure-eight loop. The system curve A has three points of intersection with the combined characteristic. The operation is likely to oscillate between these three points. The fans will be alternately underloaded and overloaded. Pulsations and noise will be caused. Damage to drive motors is also possible. It is seen that system curve B has only one intersection with the combined characteristic. Hence stable operation will result.

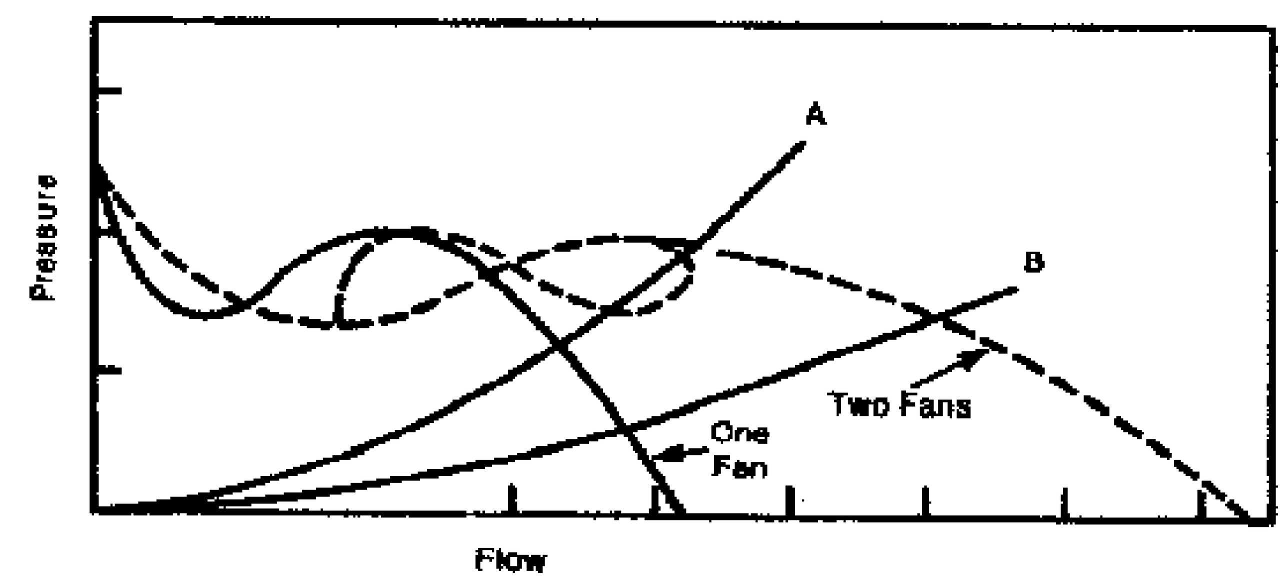


Figure 8-4. Parallel Operation of Identical Fans Whose Characteristics Have Points of Inflection.

Similar problems are faced with backward-inclined centrifugal fans at low flow rates, though to a lesser degree. Figure 8-5 shows the combined characteristics of two identical fans of this type in parallel. Two different combined characteristics result at lower flow rates. It is seen that system curve A intersects the combined characteristics at two points which will result in unstable operation.

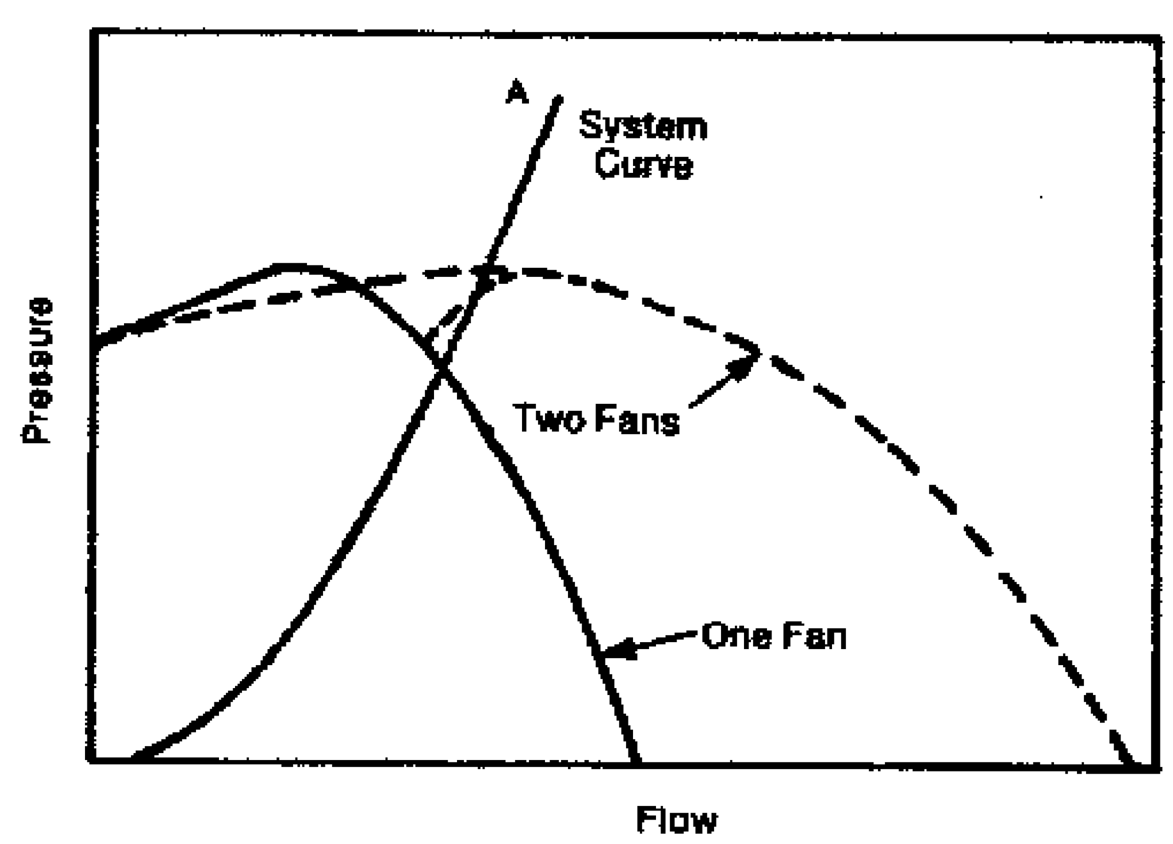


Figure 8-5. Parallel Operation of Two Identical Backward Inclined Centrifugal Fans.

It is therefore clear that if fans are operated in parallel, the operating point should be kept outside the region where there can be more than one flow at a single pressure. Thus operation is generally limited to higher flow rates. One has to be especially careful when axial flow or forward-curved centrifugal fans in parallel are used in variable volume systems.

Additional complications can arise if fans of unequal capacity having points of inflection in their curves are operated in parallel. At lower flow rates, air may be blown back through the smaller fan.

See [1.1] for extensive discussion on parallel flow systems.

III. FANS IN SERIES

Under ideal conditions, the combined performance curve of two fans in series can be obtained by adding the pressures of the fans at the same air flow rates. Figure 8-6 shows the curve obtained by such calculations.

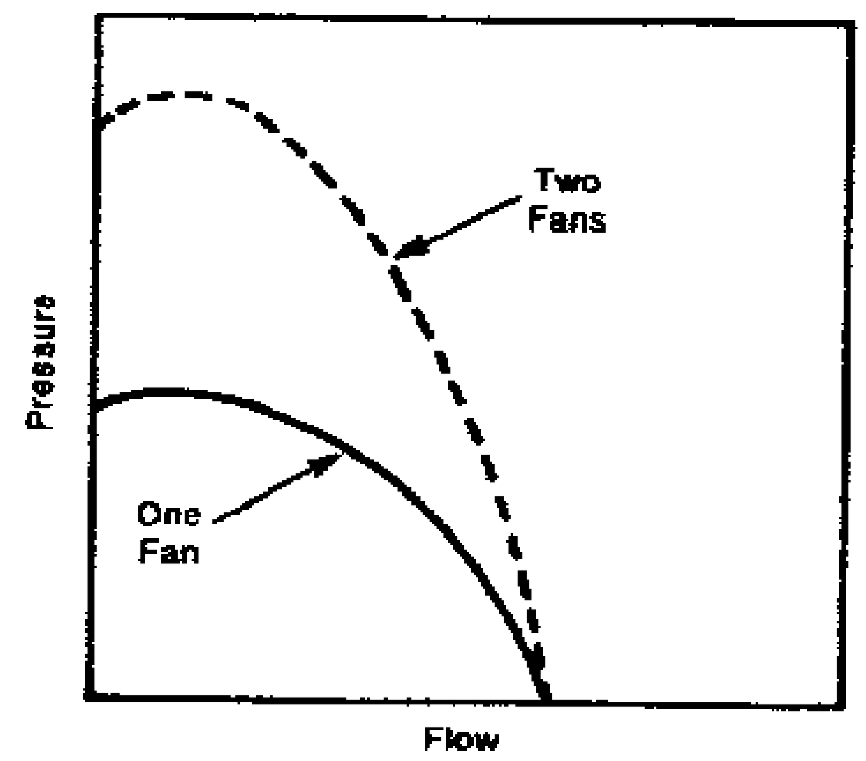


Figure 8-6. Combined Characteristic of Two Identical Fans in Series.

In practice, there will be a significant loss in performance due to non-uniform flow into the second stage. The losses may be considered to be due to system effects discussed in Section 409.4. Another point to note is that the density of air entering the second stage will be higher. However, the pressure changes in fans are so small that the effect of density variation can generally be neglected.

I. INTRODUCTION

In many systems, the amount of air required is not constant but varies over a wide range. In variable-air-volume air conditioning systems, the quantities of air handled by supply and return fans vary with the space cooling demand. The quantity of combustion air supplied to boilers varies with the steam demand. Many other examples could be cited.

To match the system requirements with fan output, two basic approaches are possible. One approach is to change the system curve and the other is to change the fan performance curve. The system curve can be changed by adding a resistance to the circuit in the form of a throttling damper. There are several ways of modifying the fan performance curve. One way is to change the fan speed through mechanical or electrical means. Another way is to use variable inlet vanes. A third way is to vary the fan blade pitch, this being available only with axial flow fans. Figure 9-1 lists the common methods of fan capacity control. See [1.1] for detailed discussions and other methods of capacity control.

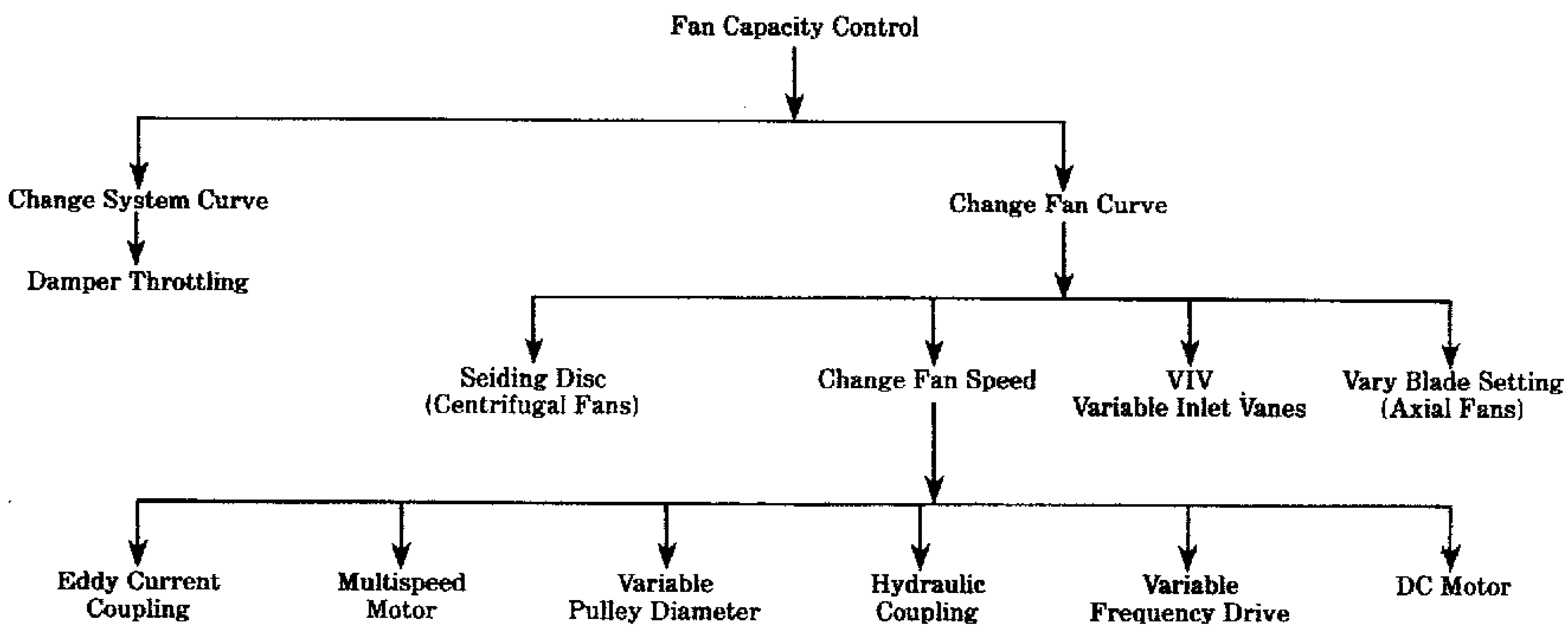


Figure 9-1. Some Common Methods of Fan Capacity Control.

The choice of capacity control method depends on several factors, such as: range and frequency of flow variations, type of fan, first cost, energy saving, etc. The various capacity control methods are now described and discussed.

See [9.1] for a discussion on the choice of capacity control method. Also see [9.2].

II. FLOW THROTTLING

A damper is placed upstream or downstream of the fan and modulated to give the desired flow rate. The change in flow is obtained due to a change in the system curve. However, if the dampers are very close to fan inlet or outlet, the change in flow may be partly due to loss in fan performance because of system effects (see Section 409.4).

This method of control has low first cost but it does not provide much reduction in energy consumption as flow is reduced. Figure 9-2 shows the performance curves of a tube-centrifugal fan. The system curve is A without throttling. With throttling, the curves B and C are obtained. It is seen that there is little change in power consumption between system curves A and B.

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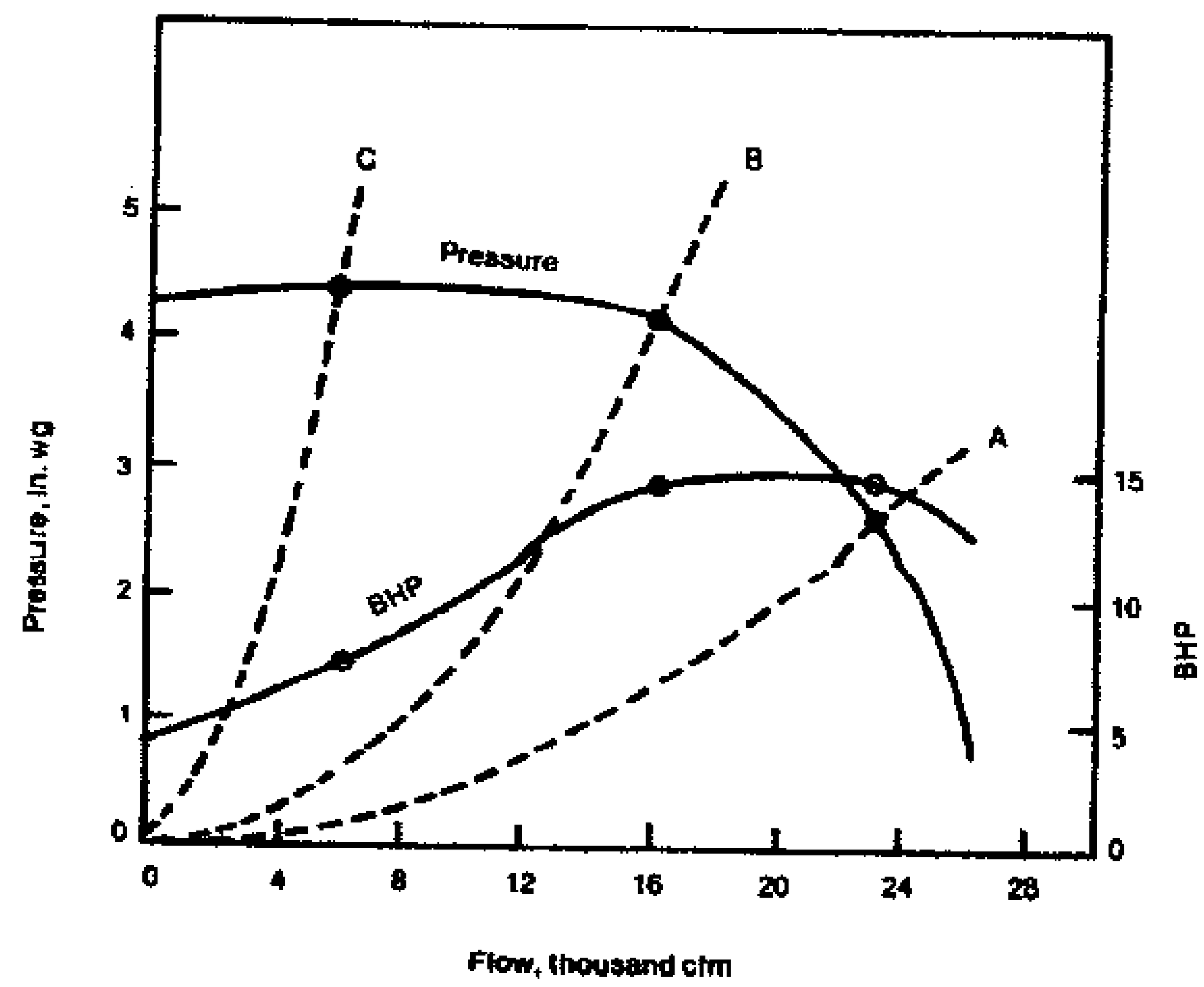


Figure 9-2. Performance of a Typcial Tube-centrifugal Fan. Throttling to Change System Curve From A to B Does Not Reduce BHP.

Many axial flow fans are inoperable at lower flow rates as was discussed in Section 409.7. Hence the throttling should not be so much that the operating point lies in the inoperable part of the fan curve. Forward-curved centrifugal fans can also exhibit flow instabilities if operated in the unstable part of their characteristic curves. See Section 409.7.

Damper throttling results in noisy operation due to the fan operating at lower efficiency and due to increased velocity and turbulence caused by the partly closed damper.

III. VARIABLE INLET VANE CONTROL

Variable inlet vane (VIV) control can be used with any kind of fan but is most commonly applied to centrifugal fans. They may be integral with the fan or added on. Their effect is mainly to change the fan performance curve by prerotating incoming air in the direction of impeller rotation. Some of the change in flow is due to the fact that they throttle the flow and thus change the system curve. However, it is customary to regard the VIV as affecting only the fan curve.

A new fan curve is obtained at every position of vanes. By use of a modulating motor to actuate the VIV, stepless variation of fan capacity is achieved. This system of control also reduces the fan power consumption. Figure 9-3 shows the effects of VIV on the characteristics of a centrifugal fan.

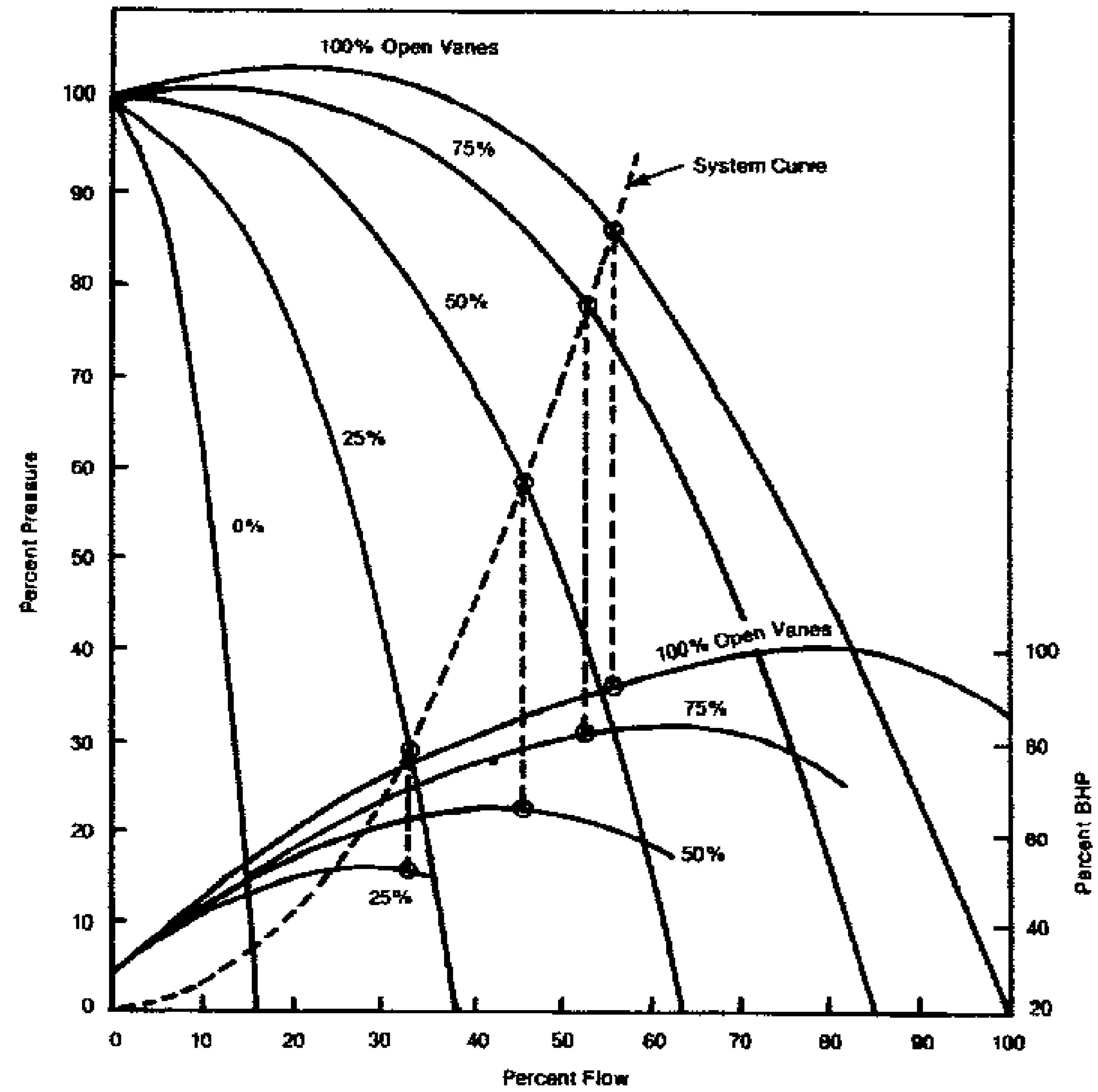


Figure 9-3. Effect of Variable Inlet Vane Control (VIV) on Performance of a Typical Backward Inclined Centrifugal Fan.

There are limits to the reduction in flow possible with this method. With the inlet vanes completely closed, the fan may still deliver up to 20 percent of its maximum flow. The reduction in power consumption at lower flow rates is also small. Operation with flow less than 50 percent of the free delivery volume may cause pulsations.

Use of variable inlet vanes increases the noise output of the fans.

IV. VARIABLE BLADE PITCH (SETTING) CONTROL

This method of control is widely used with vaneaxial and tubeaxial fans. By changing the blade pitch (stagger angle) a new fan curve is obtained. This method provides good efficiency over a wide range of blade settings and hence results in large power savngs. Figure 3-5 shows the performance of a typical vaneaxial fan at various blade settings. Noise is also reduced as the flow is reduced.

If the flow variations are needed infrequently, the blade pitch adjustments can be made manually. Fans with automatic pitch adjustment controls are also readily available. In such fans, pitch adjustment is done automatically while the fan is operating, in response to appropriate signals.

V. ADJUSTABLE DISC CONTROL

This type of control has been used with centrifugal fans. A disc passing through the blades and rotating with the impeller can be moved in the axial direction thereby cutting off part of the impeller from air. The reduction impeller can be moved in the axial direction thereby cutting off part of the impeller from air. The reduction in power consumption at part flow is somewhat more than with variable inlet vane control.

In a lower cost version of this method, the disc extends only up to the blades but not through the blades. The reduction in power consumption obtained is somewhat less than with variable inlet vane control. See [1.1] for more information.

VI. VARIABLE SPEED CONTROL

Changing the speed of the fan changes the fan characteristics according to the fan laws given in Section 409.5. As the horsepower is proportional to the cube of rpm, while the flow is proportional to the first power of rpm, there is a sharp decrease in horsepower as the flow is decreased. Figure 9-4 shows the reduction in power consumption obtained with a 100% efficient speed control device. The noise also goes down sharply according to the noise law given in Section 409.6. Hence speed variation is an ideal method for fan capacity control, provided that it can be accomplished conveniently and at reasonable cost.

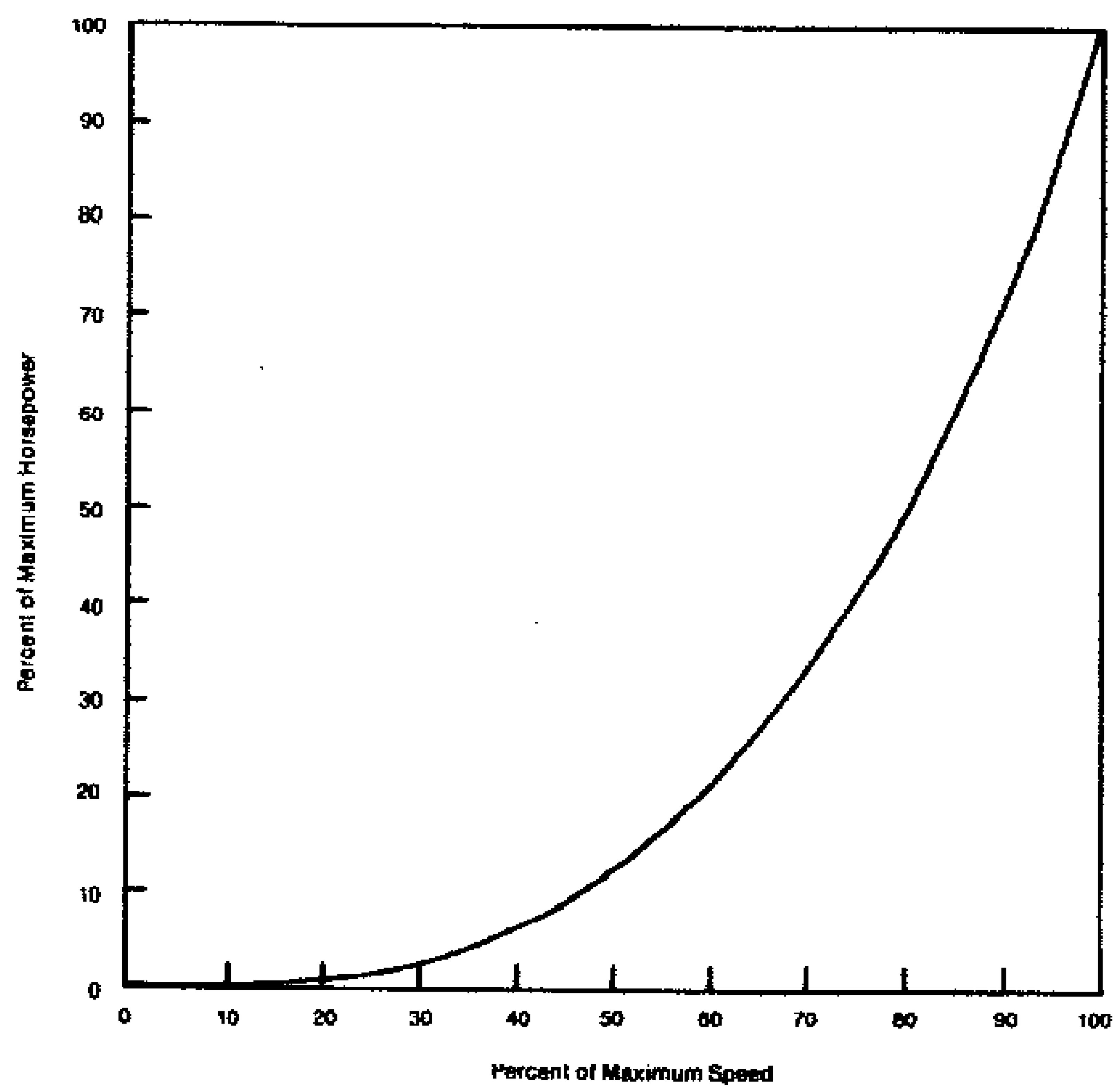


Figure 9-4. Effect of Speed Variation on Fan Power Consumption According to Fan Laws.

One method which has been employed is to use direct current motors whose speed is proportional to voltage supplied. Variable voltage direct current is supplied using a transformer and rectifier. The major difficulty with this method is that direct current motors require frequent maintenance and their part-load efficiency is poor unless elaborate and expensive controls are used.

In some systems, step variations of speed may satisfy the needs. For example, ventilating fans for a factory may run at high speed during the day and at low speed during the night. In such systems, two or three speed induction motors can be used. These operate on the principle that motor speed is inversely proportional to the number of pairs of poles. To change the speed, the number of poles is varied by suitable switchgear.

The speed of an AC induction motor is directly proportional to the supply frequency. Hence stepless speed control can be obtained by frequency modulation. A number of solid state speed controllers based on frequency modulation have been marketed in recent years. Usually voltage is also varied along with the frequency. Any standard motor can be used. No modification or attachment to fan or motor is needed. The speed controller simply replaces the motor starter. The controllers have high efficiency but tend to be expensive and large in dimensions.

Hydraulic and eddy current couplings permit infinite variation of fan speed and have good efficiency. A variety of automatic mechanical drives are readily available which permit continuous speed variations of up to 1:4. As an example, one type uses a belt drive between pulleys whose diameters are automatically varied. The motor is coupled to one pulley and the fan is connected to the other pulley. These drives have good efficiency.

VII. CHOICE OF CAPACITY CONTROL METHODS

In choosing the method of capacity control, several factors have to be considered. Among them are first cost, energy savings, reliability, space requirements, range and frequency of variations, convenience in layout, etc. All these factors should be carefully considered in making the choice. Very often, the methods which have the least first cost also give the least energy savings. Use of more expensive equipment for capacity control should be justifiable through greater energy savings which result in rapid payback of the additional investment. Thus if flow control is needed for only a few hours in a month, the less efficient discharge damper control may be preferable to the more efficient VIV control. The most important requirement is that the control method chosen should give stable performance throughout the operating range. Only those control methods can be considered which satisfy this minimum requirement.

Figure 9-5 shows typical energy savings obtainable with some commercially available systems of capacity control. It is seen that if flow variation is limited to about 80% of full flow, the power consumption with VIV is only slightly higher than with speed control or with controllable pitch fans. As VIV control is much cheaper than the other types, its use may be more economical. On the other hand if required flow is often in the range of 20% to 50%, VIV may be uneconomical because of its much higher energy consumption. Indeed VIV may not at all be operable in this range.

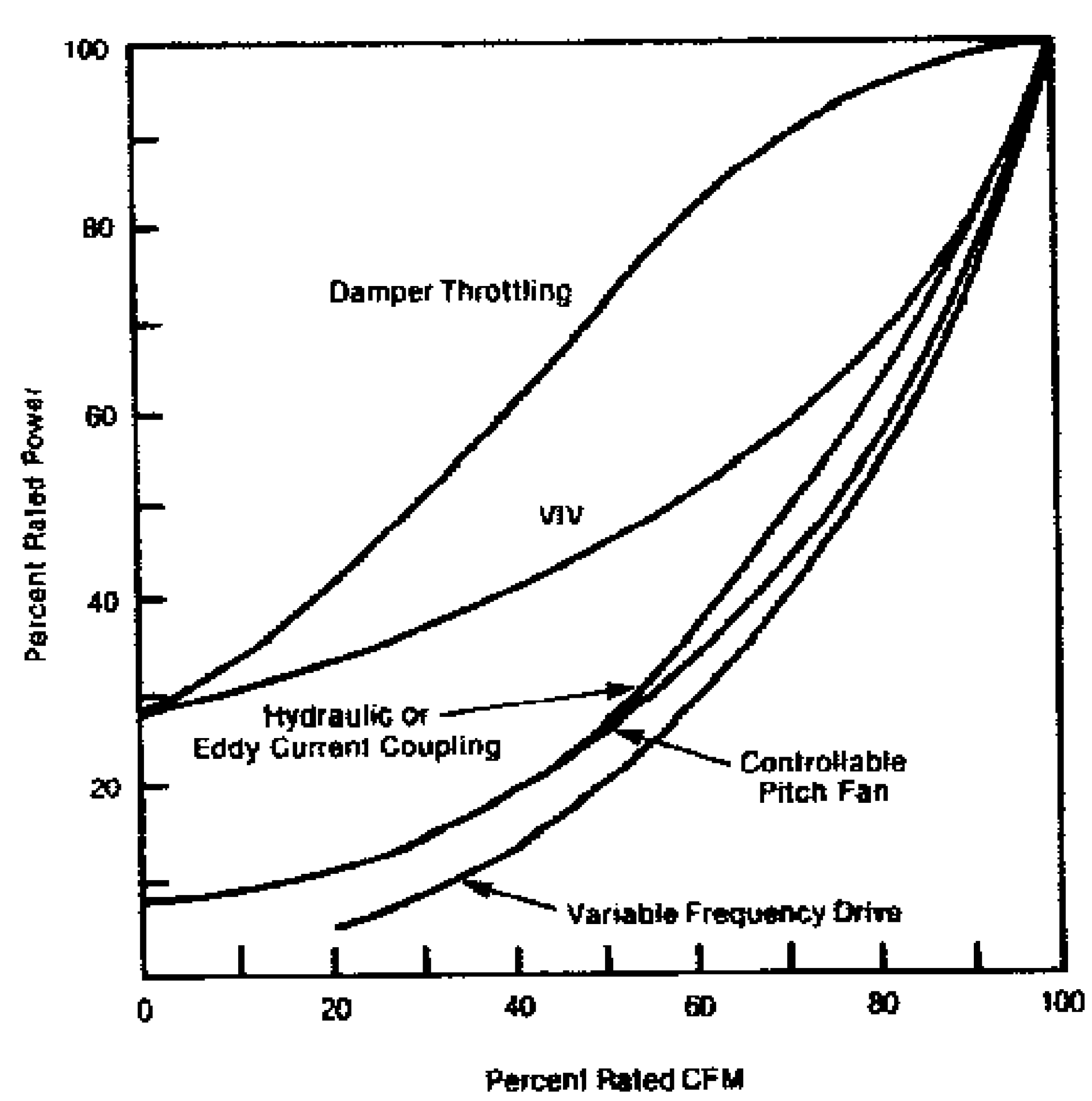


Figure 9-5. Typical Energy Savings Obtainable With Commercially Available Capacity Control Equipment.

Where noise considerations are important, speed control becomes very attractive as fan noise diminishes rapidly with decreasing flow. Variable blade pitch control also reduces the noise at lower flows but it is available only with axial flow fans. Throttling and VIV controls give more noise at part-flow than at full flow. The cost of sound attenuation should be included in the economic analysis.

I. INTRODUCTION

The requirements and features of fans for some special applications are discussed in this section. The topics covered are:

Fans handling hot gases
Fans handling corrosive gases
Fans handling gas-solid mixtures
Power plant fans
Spark-resistant fans
Fans for cooling electrical motors and generators

II. FANS HANDLING HOT GASES

In a number of processes, fans are required to handle hot gases. The strength of materials of construction goes down with increasing temperature. Hence impellers of standard design are run at lower speeds when used at higher temperatures. Design and cooling of bearings poses special problems. Bearings are kept outside the hot gas stream but they still gain heat due to radiation and conduction. At temperatures above 700°F, the bearings are usually provided with a separate base to prevent direct conduction of heat from the fan housing. Heat is still conducted through the drive shaft. At higher gas temperatures, a cooling disc or wheel is mounted on the shaft between the fan and bearing. This protects the bearing by cooling the shaft and moving air over the bearing.

Bearings may be lubricated with grease at temperatures up to 200°F but oil lubrication is necessary at higher temperatures. Forced cooling of bearings is generally required. One method is to pass the lubricating oil through a water cooled heat exchanger. In some designs air is sucked through or blown through the bearings by an auxiliary blower. Air is filtered before it enters the bearings.

Expansion of fan components due to temperature rise requires careful consideration. Unequal expansion of parts may cause interference or excessive slack. Sudden exposure to hot gases may cause the impeller to expand faster than the shaft and come loose. A fan shaft exposed to hot gases while idle may take a permanent set leading to vibrations. All such problems should be carefully considered in fan design and selection.

In some situations, it may be more economical to cool the gases before entering the fan instead of using a more expensive model suitable for high temperature duty. This can be done by diluting the gases with ambient air, by water sprays, etc. In some cases, the system requirements can be fulfilled by placing the fan on the cold side rather than the hot side of the system. For example, many boilers use a forced draft fan (handling cool air) instead of an induced draft fan (handling hot flue gases). The fan system designer should consider such possibilities.

III. FANS HANDLING CORROSIVE GASES

When handling corrosive gases, fan bearings should be outside the air stream and fan components should be fabricated from or coated with corrosion resistant materials. No single material can be considered corrosion-proof under all conditions nor can the suitability of any particular material be predicted through theoretical methods alone. The choice has largely to be based on experience.

Besides the chemical composition of gases, temperature and moisture content are also important. Susceptibility to corrosion increases with temperature. A material which may be satisfactory at low temperatures may be unsatisfactory at higher temperatures. Many materials which show no corrosion with dry gases corrode rapidly if the same gases contain moisture.

Type 304 and 316 stainless steels show good corrosion resistance to a wide variety of chemicals. Aluminum and its alloys are generally satisfactory with acids but are attacked by alkalis. Rubber and plastics have excellent resistance to a wide range of chemicals.

Fans with impellers and housings fabricated from fiber glass reinforced plastics (FRP) and polyvinyl chloride (PVC) are available but are generally limited to moderate pressures. The maximum temperature that FRP fans can withstand is about 250°F. Corrosion due to moisture alone can be resisted by a variety of paints and by galvanizing. Plastic coatings can resist a wide variety of chemicals. The success of any coating system depends strongly on the skill with which it is applied. AMCA Standard 2601 gives general guidance on this subject.

In selecting a fan for corrosive gases, guidance should be sought from fan manufacturers and corrosion specialists. As far as possible, materials used should be those which have been proven corrosion-resistant under the same conditions.

IV. FANS HANDLING GAS-SOLID MIXTURES

A variety of systems and processes require the fans to handle gas-solid mixtures. Notable examples are induced draft fans for furnaces, kilns, boilers, and sinter beds. The blower in a domestic vacuum cleaner handles air carrying dust. Fans used for pneumatic conveying systems handle air carrying a wide variety of materials such as coal, coffee, wood chips, sand, grain, sawdust, etc. Fans in a variety of industrial processes handle hot gases carrying solid particles.

The two major problems associated with solid particles are erosion of fan components and flow passage clogging. The flow passages and blades have therefore to be designed to prevent clogging and features to minimize erosion have to be incorporated in the design.

Axial flow fans are used to a limited extent. Centrifugal fans are used in most cases. Radial blades are used for the most severe applications. Forward and backward curved blades are used to a limited extent.

A. Erosion

When the solid handled consists of hard grains, erosion is the major problem. Erosion can occur on both moving and stationary parts. The severity of erosion depends on the hardness of particles and their kinetic energy. The erosion will be less in a large slow speed fan compared to a small high speed fan because of the higher velocities in the latter. Methods for reducing erosion are available but erosion cannot be completely eliminated. A very effective method is to hardface the areas subject to wear. Layers of a hard substance such as tungsten carbide 1/16 to 1/8 inch thick are deposited using electric welding machines. The life of hard-faced impellers is up to four times longer than that of ordinary impellers.

The dust particles are generally carried towards the backplate of the impeller. A common device is to provide a wear plate in this area. Weld beads at the junction of blade and backplate also help to reduce the erosion. In very severe applications, the entire blade may be covered with a wear plate. The wear plates may be welded or bolted to the blades. The bolted wear plates can be replaced in the field.

The fan scroll is also often provided with liners. These liners are plug welded to the scroll. When the liner is worn, it can be removed and replaced by a new one. In some cases, the entire scroll may be replaced. See [10.1], [10.2], and [10.10] for additional information on erosion.

B. Deposition and Clogging

Deposition of particles on impellers can lead to unbalance and vibrations. Plugging of flow passages with solids will lead to system failure. Hence the design of impellers should attempt to prevent these problems. Experience has led to a number of designs. While these designs differ in detail, the basic feature is that the impellers are open or semi-open to permit easy passage of solids. Figure 10-1 shows the semi-open design while Figure 10-2 shows the open design. The open design has neither an inlet plate nor a backplate. The semi-open design has a backplate but no rim (inlet plate). The blade heels are so shaped that material slides off easily from them. The number of blades is generally kept small. Where sticky materials are involved, internal surfaces have to be minimized.

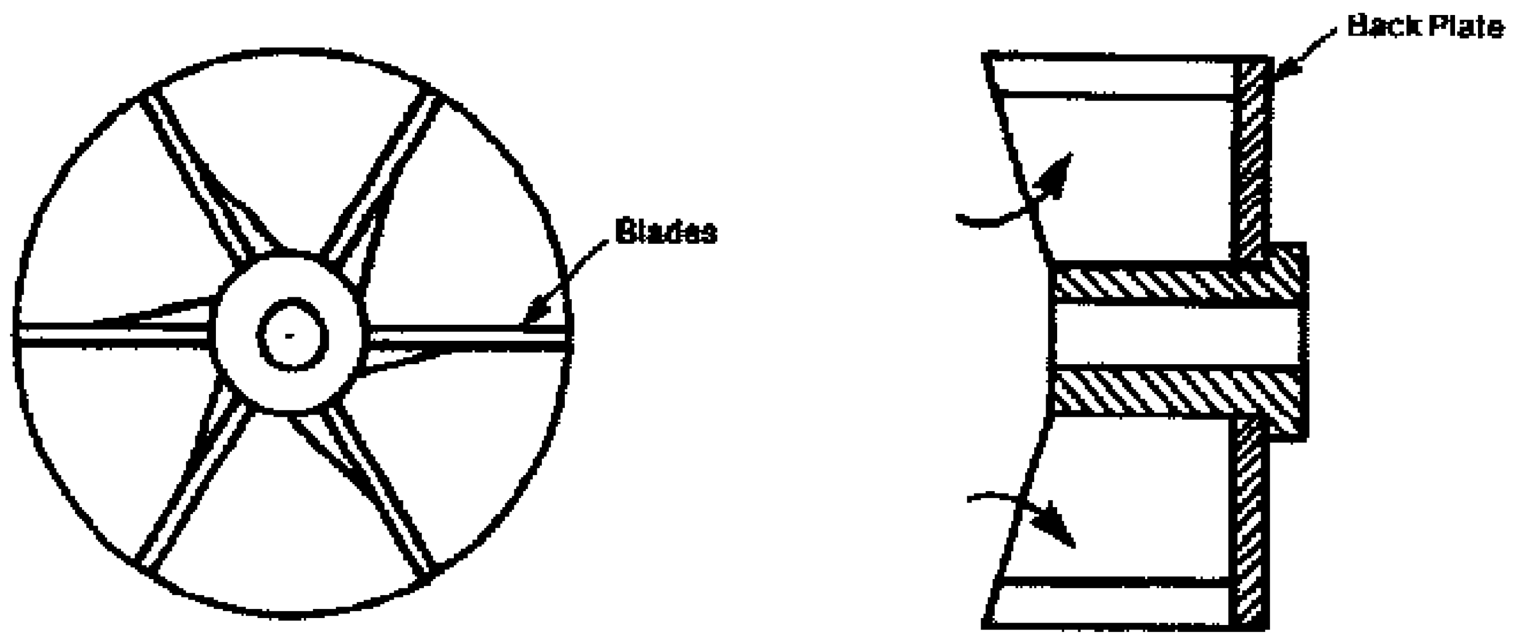


Figure 10-1. Semi-open Impeller Design.

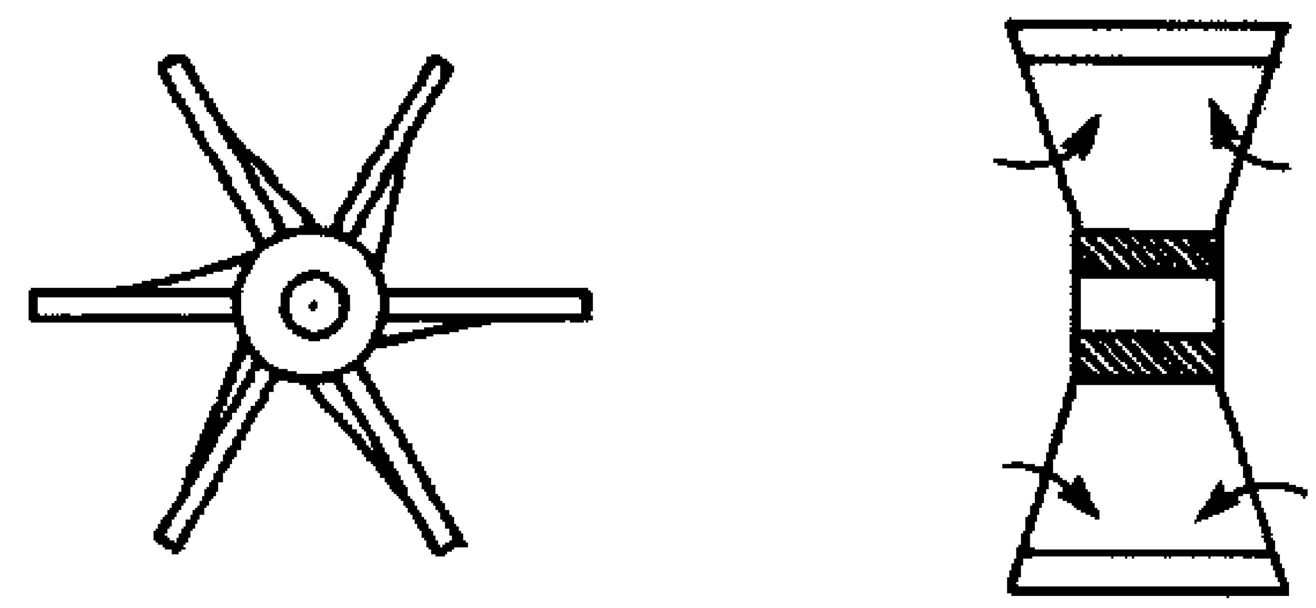


Figure 10-2. Open Impeller Design.

C. Effect of Solid Content on Fan Performance

The presence of solids can be expected to have some effect on fan performance. The effect depends on the size and concentration of solids. We first define a parameter μ as:

$$\mu = \frac{\text{Mass flow rate of solids}}{\text{Mass flow rate of gas}} \tag{10-1}$$

Experiments have shown that with small μ and fine particles, the pressure-cfm curve of the fan is the same as for pure gas. However the power consumption increases in direct proportion to the density. The mixture density ρ_{mix} is given by:

$$\rho_{mix} = \rho_{gas} (1 + \mu) \tag{10-2}$$

The power consumption is given by:

$$\frac{P_{mix}}{P_{gas}} = \frac{\rho_{mix}}{\rho_{gas}} \tag{10-3}$$

The subscripts "mix" and "gas" indicate for mixture and for 'As respectively.

If the particles are large, they do not follow the gas stream due to their large inertia and collide more frequently. Energy losses increase and the pressure head decreases. Cherkassky [1.2] gives the following semi-empirical equations for calculating pressure, power consumption, and efficiency, when larger particles are involved. These are based on extensive experimentation.

$$p_{mix} = p_{gas} (1 - k_p\mu) \tag{10-4}$$

$$P_{mix} = P_{gas} (1 + k_N\mu) \tag{10-5}$$

$$\eta_{mix} = (1 - k_p\mu)/(1 + k_N\mu) \tag{10-6}$$

The correction factors k_p and k_N are as follows; for solid particles of organic origin (peat, sawdust) 0.5 to 3 mm in size $k_p = 0.1$ to 0.45 and $k_N = 1.5$ to 1.7.

V. POWER PLANT FANS

The two major fans used in power plants are the forced draft fans and the induced draft fans. The forced draft fan handles atmospheric air and supplies it to the boiler for combustion purposes. The induced draft fan sucks the flue gases from the boiler. Fans for recirculating the hot products of combustion within the boiler are also used. These fans represent a major capital investment and power plant reliability depends on their reliability. Hence all aspects of these fans require careful consideration.

The forced draft fans handle clean air at normal atmospheric temperature. Hence they are not subject to effects of high temperature, corrosion, or erosion. Pressures developed by forced draft fans usually range from 30 to 80 inches of water. Their power consumption is in the several thousand horsepower range and flow modulation is needed because of the wide variations in boiler output. Hence good part load efficiency and stable part load operation is very important. Both centrifugal and vaneaxial fans are used for this purpose. Centrifugal fans are usually used with backward-curved air foil blades. Centrifugal fans are provided with variable inlet vane control while the vaneaxial fans may have variable blade setting control or variable inlet vane control.

The induced draft fans handle hot flue gases. In coal-fired boilers, flue gases contain erosive solids. If the fan is located after filters (flyash precipitators), only small amounts of solids pass into the fan. If located before filters, severe erosion problems have to be faced. Thus the induced draft fan may face problems due to combined effects of high temperatures, corrosion, and erosion. These problems have been discussed earlier in this section.

For induced draft fans handling relatively clean air, backward-curved airfoil centrifugals with VIV control or vaneaxial fans with automatic blade setting control are usually used. With higher amounts of solids, vaneaxial fans use the VIV control. Centrifugal fans used with high solids containing gases usually have forward-curved or radial blades.

For additional information on power plant fans, see [10.3], to [10.7] and [1.1] to [1.3].

VI. SPARK RESISTANT FANS

If a fan is handling a combustible or explosive gas, it is essential that there be no spark, otherwise a fire or explosion may result. Rubbing or striking of ferrous surfaces can cause sparks. Spark resistant fans are designed to eliminate the possibility of rubbing between ferrous surfaces.

AMCA Standard 401 gives three classifications of spark resistant construction. These are:

Type A All parts of the fan in contact with the gas being handled shall be made of non-ferrous material.

Type B The fan shall have an entirely non-ferrous wheel or impeller and non-ferrous ring about the opening through which the shaft passes.

Type C The fan shall be so constructed that a shift of the wheel or impeller or shaft will not permit two ferrous parts of the fan to rub or strike.

For all three types, it is required that bearings be outside the gas stream and the user should electrically ground all fan parts.

Fans of all three classifications are readily obtainable from manufacturers. The non-ferrous materials commonly used are aluminum and bronze.

VII. FANS FOR COOLING MOTORS AND GENERATORS

Heat is generated in electric motors and generators due to mechanical friction and electrical losses in windings and core. The heat has to be dissipated to prevent overheating of bearings and windings. In small and medium size machines, this is accomplished by passing air over the bearings and the electric windings.

The design of fans for rotating electric machinery is distinguished from the design of fans for other applications by the following features:

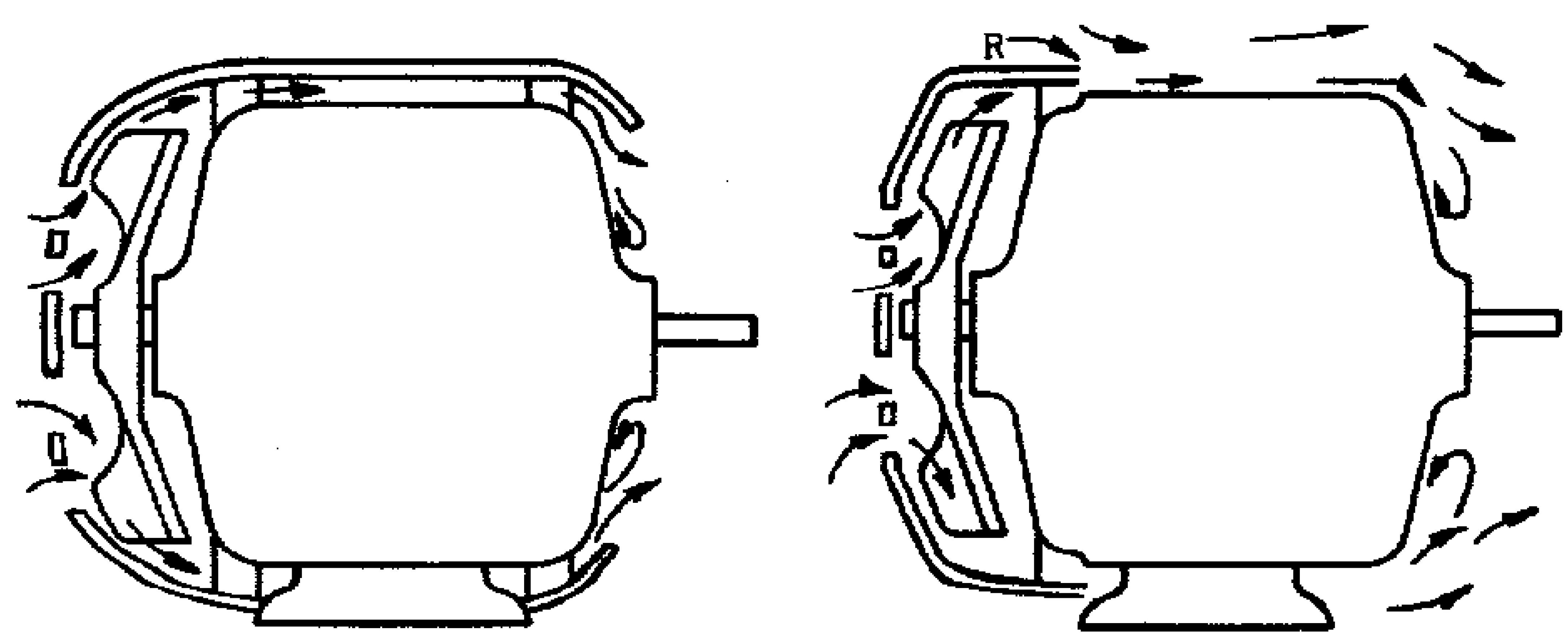
1 The speed of the fan is generally the same as the speed of the machine.
2 The maximum outer diameter of the fan is generally limited by the size of the machine.
3 The fan design is modified to suit the variety of uses to which the machine is put. Intermittent use, the requirement of reversibility, and speed range are examples of this.
4 The fan design is modified to suit the type of enclosure and whether or not the machine is self-ventilated or separately ventilated.
5 The fan design is adapted to the manufacturing facilities available and quantities to be made.

There is no best all purpose fan [10.8]. Each optimum design is a feature of the particular application. The approach to the design of a fan for a generator may be quite different from that for a motor, and considerable variation may be encountered in the design of fans for large, slow-speed machines as opposed to small, high-speed ones. Furthermore, the performance of small motor fans cannot be accurately calulated from the physical dimensions alone because of the wide variation in the many empirical coefficients used. Instead it is calculated from the performance of similar fans. This is true to a greater or lesser degree for all fans.

The various kinds of motors are described in Section 409.7.Fan impeller design and some other design features are discussed in Section 409.2.

A. Cooling of Totally Enclosed Motors

Totally enclosed motors must rely upon heat removal by conduction through the mountings. In very small sizes especially for intermittent operation, natural convection (and conduction) may be sufficient. Except for such cases however, forced convection and hence fans, must be used. Two types of enclosures for a totally enclosed fan cooled motor are indicated in Figure 10-3. Part (a) represents good cooling performance but is wasteful of material in the outer housing; 10-3(b) represents a closer balance between material utilization and effective cooling.



(a) (b)

Figure 10-3

B. Air Path Design for "Open Motors"

In "open-type" motors, the removal of heat from the windings and laminations may be accomplished by end ventilation. Air is circulated by fans at both ends and cools the end windings. For relatively large motors, however, it is usually necessary to use axial ventilation. In this case, the coolant fluid is circulated through axial ducts in the stator and in some cases the rotor. Still another type of ventilation utilizes radial ducts at intervals along the armature.

The design of fans for these applications is complicated by the fact that proper direction and baffling of the coolant flow must be achieved for effective cooling.

For open type double-end ventilated small motors the effects of various baffles and other design changes are shown in Figure 10-4 a,b,c,d & e. The systems sketched represent increasing cooling capacity (a, least; e, greatest) [10.8]. Also see [10.9].

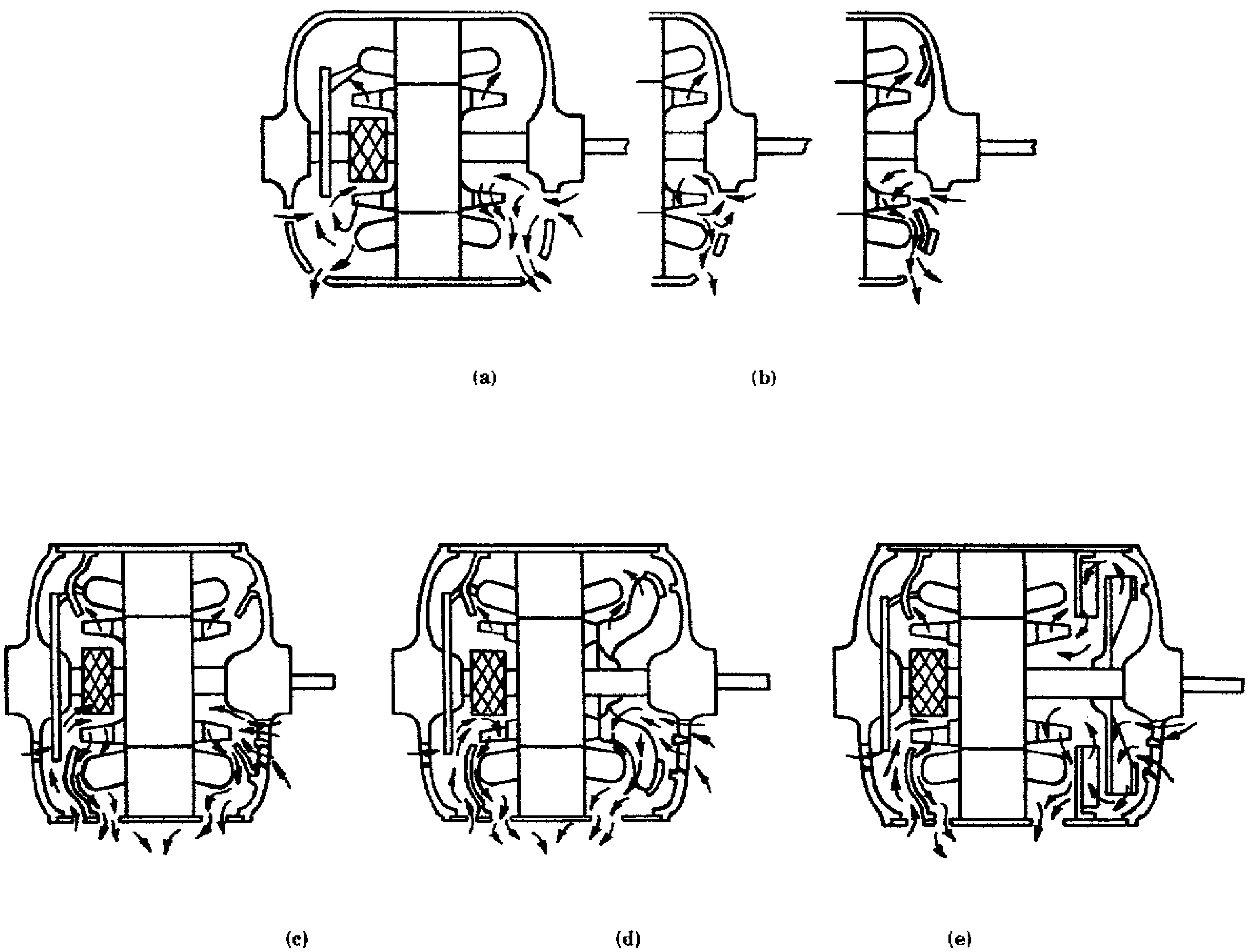


Figure 10-4. Open Type Double-end Ventilated Small Motors.

C. Operating Point

A comparison of end-ventilated and through-ventilated systems as well as the general characteristics of a good and poor fan and baffle design is shown in Figure 10-5.

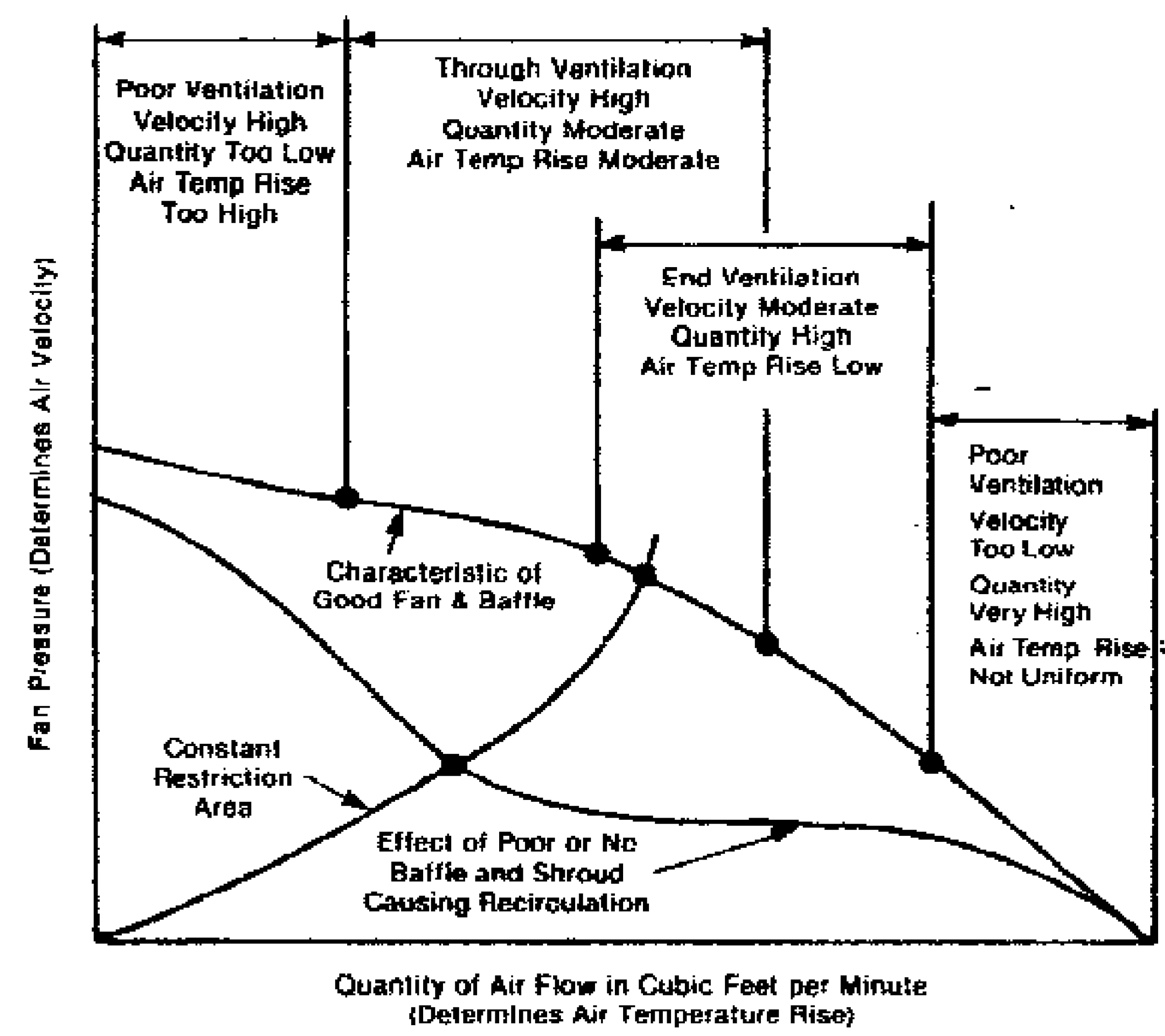


Figure 10-5.

I. GENERAL

Fans are tested in laboratories to determine their characteristics in order to provide reliable information to the user. Typically, a user needs to know the pressure developed, power consumption, and fan efficiency. Information of sound power output may also be required. Field tests of fan performance are often carried out in addition to or in lieu of laboratory tests.

II. LABORATORY TESTS FOR FAN PERFORMANCE

Laboratory tests of performance are generally carried out under conditions which eliminate the system effects discussed in Section 409.4. Inlet and outlet connections are so designed that system effects are negligible. This enables the user to carry out a meaningful comparison of fans from different manufacturers. Flow, pressure rise, and power consumption are measured with precision instruments to obtain the fan performance. For rating purposes, usually the smallest fan from a series of geometrically similar fans is tested. The performance of the other fans of the series is then calculated with the fan laws presented in Section 409.5. For a detailed discussion on laboratory test methods, see [1.1].

Most manufacturers test and rate their fans according to AMCA Standard 210 (which is the same as ASHRAE Standard 51). Most users also specify that the fans be rated according to this standard.

AMCA Standard 210 describes a number of test set-ups. The choice among these depends on considerations such as fanconnections, fan pressure and convenience. Figure 11-1 shows the general arrangement of three of the test setups described in this standard.

The arrangement of Figure 11-1(a) is suitable for fans which may be used with or without ducts, such as propeller fans. The fan sucks from a large chamber so that the air velocity near its inlet is negligible, thus simulating unducted suction such as from a room. For flow measurement, a set of calibrated nozzles is provided. Pressure drop across the nozzles is measured with manometers. To ensure uniform velocity in the chamber, devices such as screens, baffles, perforated plates etc. are provided upstream and downstream of the nozzles. As the pressure developed by such fans may be insufficient to overcome the resistance of measuring devices, an auxiliary fan may be used to supply air into the test chamber. The auxiliary fan should be capable of providing varying quantities of air. A fan with variable speed or other flow control devices has therefore to be used. Fan flow is known from nozzle pressure drop. Fan suction pressure is known from the total pressure measured near fan suction. Static pressure at fan outlet is assumed to be atmospheric. Varying the flow with the auxiliary fan, flow and pressure measurements are carried out over the range of interest.

Figure 11-1(b) shows an arrangement which is suitable for fans used with inlet ducts; there may or may not be a discharge duct. A long duct is connected to the fan inlet. Quantity of air entering the duct is varied using a symmetrical throttling device. A flow straightening device is installed in the duct to minimize velocity variations across the duct crosssection. Velocity and static pressure in the duct are measured with a pilot tube. Measurements are made at a number of points in the cross-section and their average is considered to be the correct value. A correction is made for the prpssure loss in the duct between the measurement plane and the fan inlet to obtain the true fan inlet static pressure.

Figure 11-1(c) shows an arrangement suitable for a fan used with discharge duct, with or without inlet duct. A venturimeter is used for measuring the flow. Flow through the fan is varied with a variable exhaust fan. This may be a simple throttling device or a variable flow auxiliary fan.

Whichever the test setup used, air temperature variations should be kept to a minimum. The room in which the test is carried out should be kept free of any currents which may affect the fan performance. If the fan is sold with its own bearings, it should be tested while mounted on its own bearings.

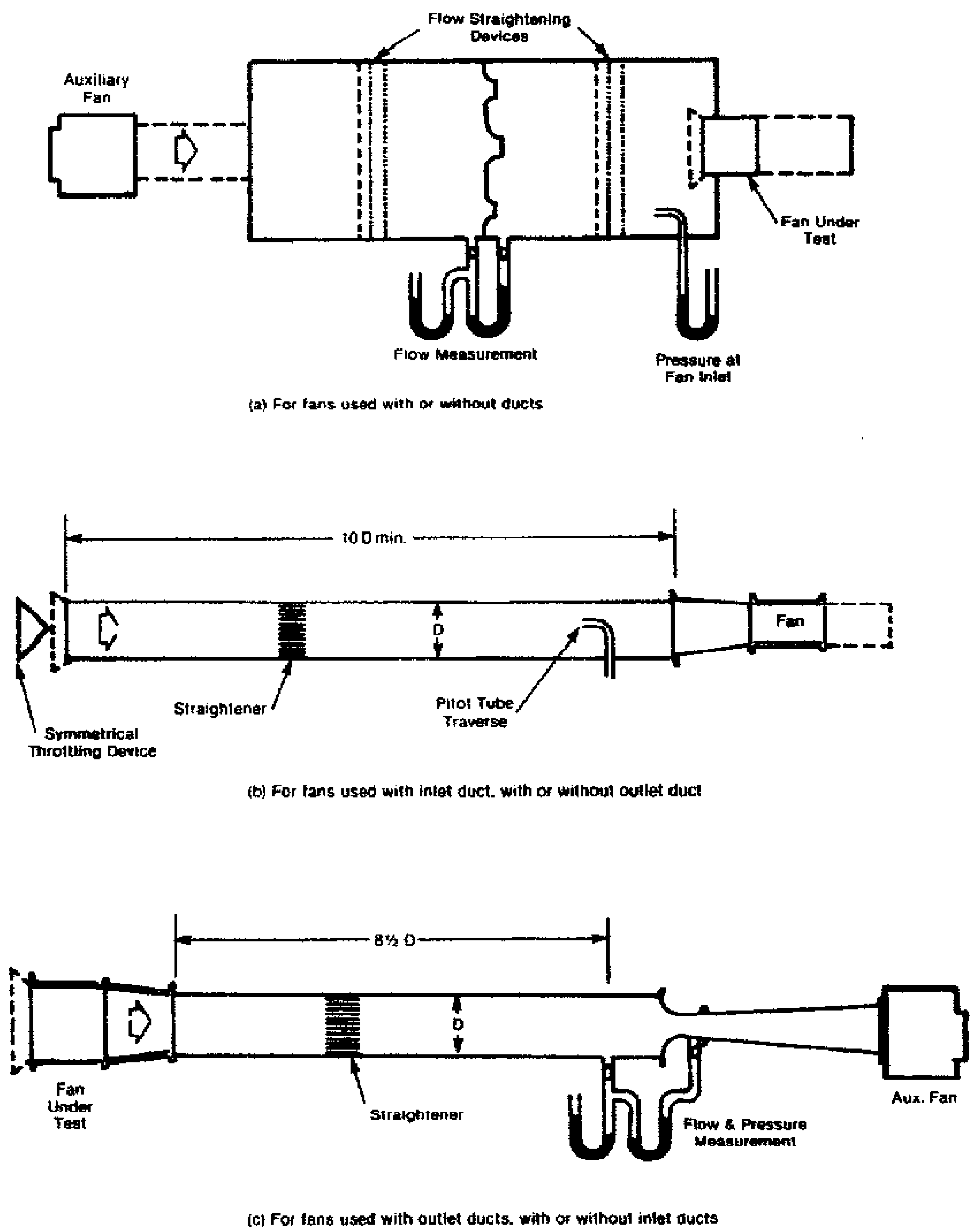


Figure 11-1. Schematic Representation of Some of the Laboratory Methods for Performance Testing of Fans for Rating Purposes According to AMCA Standard 210/ASHRAE Standard 51.

Fan power consumption may be measured using electric or hydraulic dynamometers or using calibrated electric motors. The power measured excludes losses in drive belts or couplings. Fan speed measurements are carried out using a directly coupled tachometer or a stroboscopic device. A directly coupled tachometer may slow down a small fan. Hence for small fans, stroboscopic devices have to be used.

III. FIELD TESTS FOR FAN PERFORMANCE

Field tests may be carried out in lieu of or in addition to laboratory tests. For very large fans, laboratory tests may be too difficult or expensive and field tests may be the only recourse. Another reason may be that the buyer may wish to verify the manufacturer's claims regarding performance through measurements in actual installation. Whatever the reason, accurate measurements in the field are generally difficult.

One major difficulty is caused by the fact that in most practical installations, fan inlet connections are less than ideal. As an example, space limitations often necessitate placement of dampers close to fan inlet and outlet, resulting in large losses due to system effects. Hence the fan performance falls short of the rated performance. As the prediction of system effects can only be approximate, it is generally difficult to determine from field tests whether the fan manufacturer's claims regarding performance are true or false.

Another major problem in field tests is that it is difficult to make accurate measurements. For example, accurate flow measurements require a calming length of duct, at least a few duct diameters long, upstream of the measuring instrument. Adequate calming length is often unavailable due to space limitations. Flow measurements therefore become susceptible to error. If field tests are intended, it is advisable to consider instrumentation requirements during duct layout design and provide adequate space for carrying out proper measurements.

As the variety of field installations is virtually limitless, no code or standard for field testing or performance has been developed. AMCA Publication 203 [11.1] gives general guidelines for such measurements. Ref. [11.2] to [11.8] give additional information on field performance measurements.

IV. SOUND MEASUREMENTS

Laboratory tests of sound for rating purposes are carried out according to AMCA Standard 300. AMCA Standard 301 gives a standard method for calculating fan sound ratings from laboratory test data. Sound ratings of fans published by the manufacturers are determined according to these standards. Ref. [11.9] reviews the development of sound measurement techniques.

Measurements of noise in field installations poses many difficulties. Ref. [11.10] discusses some of these problems.

I. INTRODUCTION

The codes and standards used in the U.S.A. are those by AMCA (Air Movement and Control Association), ASHRAE (American Society of Heating Refrigerating and Air Conditioning Engineers), and UL (Underwriters Laboratory). These are listed in the following.

II. AMCA STANDARDS

1-66	Product Definitions
AS-98-76	Basic Series of Preferred Numbers
AMS-100-76	Metric Units and Conversion Factors
401-66	Classifications for Spark Resistant Construction
402-66	Air Density Ratios—At Various Altitudes a Temperatures
1401-66	Operating Limits for Central Station Units
1402-66	Coil Face Areas for Central Station Units
AMS-2001-76	Metric Standard for Impeller Diameters & Outlet Areas of Centrifugal Fans
AMS-2002-76	Metric Standard Dimensions for Industrial Centrifugal Fans
AMS-2003-76	Metric Standard for Impeller Diameters & Outlet Areas of Tubular Centrifugal Fans
2401-66	Wheel Diameters & Outlet Areas for Centrifugal Fans
AS-2401-M-66	Standard Impeller Diameters & Outlet Areas for Centrifugal Fans (Metric Equivalents)
2402-66	Sizes for Industrial Centrifugal Fans
AS-2402-M-66	Standard Dimensions for Industrial Centrifugal Fans (Metric Equivalents)
2403-66	Sizes for Industrial Centrifugal Fans with Cast Housings
2404-78	Drive Arrangements for Centrifugal Fans
2405-66	Inlet Box Positions for Centrifugal Fans
2406-66	Designations for Rotation and Discharge of Centrifugal Fans
2407-66	Motor Positions for Belt or Chain Drive Centrifugal Fans
2408-69	Operating Limits for Centrifugal Fans
2409-66	Flue Gas Densities for Forced and Induced Draft Centrifugal Fans
2410-66	Drive Arrangements for Tubular Centrifugal Fans
2411-69	Wheel Diameters & Outlet Areas—Tubular Centrifugal Fans
AS-2411-M-69	Standard Impeller Diameters & Outlet Areas—Tubular Centrifugal Fans (Metric Equivalents)
2601-66	Preparation of Centrifugal Fans and Parts for Protective Coatings
AMS-3001-76	Metric Standard Dimensions for Axial Fans
AMS-3002-76	Metric Standard Dimensions for Axial Fan Transitions
AMS-3003-76	Metric Standard Orifice Dimensions for Propeller Fans

(All the standards listed above are included in AMCA Publication 99, "Standards Handbook")

210-74	Laboratory Methods of Testing Fans for Rating Purposes
300-67	Test Code for Sound Rating Air Moving Devices
301-76	Methods for Calculating Fan Sound Ratings from Laboratory Test Data

11. OTHER STANDARDS

UL507 Safety Standards for Electric Fans (1977). Sponsored by Underwriters Laboratory.

ASHRAE 51-75 Laboratory Methods of Testing Fans for Rating Purposes. It is the same as AMCA Standard 210-74

I. INTRODUCTION

A list of some U.S. fan manufacturers and the basic types of fans manufactured by them are given below. This list includes, but is not confined to, all manufacturers whose products are licensed to bear the AMCA certified ratings seal. For a more comprehensive list of manufacturers and information on their products, see the Thomas Register. The following abbreviations apply to the list of fan manufacturers:

C	Centrifugal fan
CF	Ceiling fan
M	Material handling
P	Propeller fan
PRV	Power roof ventilator
TA	Tubeaxial fan
TC	Tube-centrifugal (inline centrifugal) fan
VA	Vaneaxial fan
VAA	Vaneaxial fan with automatic blade pitch control

LIST OF FAN MANUFACTURERS

Manufacturer	Products
ACME Engineering & Manufacturing P.O. Box 978 Muskogee, Oklahoma 74401	C, P, PRV, TA
Aerovent Inc. Ash & Bauer Streets Piqua, OH 45356	C, TA, VA, P,PRV
Air Turbine Propeller Co. Feazell Road Zelienpole, PA 16063	P
Aladdin Heating Corporation P.O. Box 5208 San Leandro, CA 94577	C, PRV, TC, P, M
Airmaster Fan Co. 150 W. North Street Jackson, MI 49202	CF, P, TA, PRV
American Coolair Corporation P.O. Box 2300 3604 Maynower Street Jacksonville, FL 32203	P, PRV, TA
American Fan Co. 2930 Symmes Road Fairfield, OH 45014	CF, C, P, TA, M
American Standard Industrial Products Division P.O. Box 76 8111 Tireman Dearborn, MI 48121	C, TA, M
Ammerman Co., Inc. General Resource Corp. 201 South Third Street Hopkins, MN 55343	PRV
Bar-Brook Manufacturing Co. 6135 Linwood AVe. Shreveport, LA 71106	P, VA, TA, PRV
Barry Blower 99 N.E. 77th Way Minneapolis, MN 55432	C, TC, M
Bayley Propeller Group Lau Div. of Philips Industries Inc. 843 Indianapolis Ave. Lebanon, IN 46052	P, C, TC, M
Bonanza Fan Inc. 1622-T Browning Ave. Irvine, CA 92714	VA
G.C. Breidert Co. 13690 Vaughn Street P.O. Box 1190 San Fernando, CA 91341	C, P, PRV
Broan Manufacturing Co., Inc. 926 West State Street Hartford, WI 53027	C, P
Brod and McClung-Pace Co. 9800 S.E. McBroad Ave. Portland, OR 97222	C, PRV, TA, VA, P

Manufacturer	Product
The Brundage Co. 436 West Willard Street Kalmazoo, MI 49006	C
Buffalo Forge Co. P.O. Box 985 490 Broadway Buffalo, NY 14240	C, TA, VA, VAA, P, PRV
C-E Power Systems, Industrial Boiler Products, Combustion Engineering Inc. Windsor, CT 06095	TA, C
CEA-Carter-Day Co. Minneapolis, MN 55432	C, M
Carrier Corp. Syracuse, NY 13201	C, TC, TA, VA, P
Carnes Company 448 South Main Street Verona, WI 53593	PRV
Central Blower Company 4545 East Washington Blvd. Commerce, CA 90040	C
Champion Blower Corp. 100 W. Central Ave. Roselle, IL 60172	C, TC, M, VAA
Chicago Blower Corp. 1675 Glen Ellyn Road Glendale Heights, IL 60137	C, VA, M
Chore-Time Equipment Inc. State Road 15 North Mieford, IN 46542	P
Cincinnati Fan & Ventilator Co., Inc. 5345 Creek Road Cincinnati, OH 45242	C, TA, P, M
Clarage Fan Co. Two Clarage Place Kalamazoo, MI 49001	C, P, VA, M
Combustion Equipment Co. 3007 E. 85th Street Kansas City, Missouri 64132	C, TA
Loren Cook Co. 2015 E. Dale Street Springfield, Missouri 65803	C, P, PRV
Comfort Conditioning Div. Robins & Myers Inc. Memphis, TN	P, TA
Coppus Engineering Corp. 342 Park Ave. Worcester, MA 01610	C, TA, VA, P
The Crowley Co. P.O. Box Drawer 281 Newbury, OH 44065	P
The Cyclone Manu. Co. P.O. Box 67 Urbana, IN 46990	P
Dayton Electric Manufacturing Co. 5959 W. Howard Street Chicago, IL 60648	C, TA, VA, P, M
DeBothezat Fan Co., Inc. 4211 Campbells Run Rd. Pittsburgh, PA 15205	C, TA, P
Dresser Industries Inc. Jefferey Mining Machinery Div. Columbus, OH 43216	C, P, TA, VA
Dynamic Air Engineering, Inc. 620 E. Dyer Rd. Santa Ana, CA 92705	C, TA, VA, P
Emerson-Chromalox Div. Emerson Electric Co. 8402 Pershall Rd. Hazelwood, MO	CF
Envirofan System Inc. 730 Young Tr. P.O. Box 107 Buffalo, NY 14223	CF
Flakt Products, Div. of Flakt Inc. Ft. Lauderdale, FL 33335	VA, TA
Frigid Inc. 1250 Rockaway Avenue Brooklyn, NY 11236	TA, VA, C, P, M
Garden City Fan & Blower Co. 1701 Terminal Road Niles, MO 49120	C, P, M
General Resource Corp. Hopkins, MN	C, TA, P, M
Greenheck Fan Corp. Ross Avenue Schofield, WN 54476	C, P, PRV
Hartzell Propeller Co. P.O. Box 919 Downing Street Piqua, OH 45356	C, P, TA, VA
Heil Process Equipment Div. Xerxes Corp. 34252 Mills Road Avon, OH 44011	C, TA, VA, PRV
ILG Industries 2850 North Pulaski Road Chicago, IL 60641	C, P, TA, M
Industrial Gas Engineering Co., Inc. Westmont, IL	TA, VA, C, P, M
Industrial Air, Inc. P.O. Box 215 Amelia, OH 45102	C, TC, TA, VA, P, PRV, M
Jenn Industries Inc. 3035 Shadeland Ave Indianapolis, IN 46226	PRV, TC

Manufacturer	Product
Joy Manufacturing Co. 1200 Oliver Bldg. Pittsburgh, PA 15222	TA, VAA
Joy Manufacturing Co. New Philadelphia Div. New Philadelphia, OH 44663	TA, VAA
Lau Div. of Philips Industries, Inc. 2027 Home Avenue Dayton, OH 45401	C
Leading Edge Inc. 8814 T. S.W. 131 Street Miami, FL 33176	C
Lytron Inc. Division Fan Division Dragon Court Woburn, MA 01801	PRV
Moffit Division Steelite Inc. 1010, Ohio River Blvd. Pittsburgh, PA 15202	PRV, P
The New York Blower Co. 3155 South Shields Ave. Chicago, IL 60616	TA, VA, P, C, TC, M
Oliver & McClennan Inc. P.O. Box 478 West Street E. Hanover, NJ 07936	C, TC, P, PRV, TA, VA
Penn Ventilator Co. Gantry at Red Lions Rd. Philadelphia, PA 19115	P, PRV, C
H.K. Porter Co., Inc. 1401 West Market Street Warren, OH 44485	C, TC, P, TA, VA
Power Line Fans & Chelsea Fans & Blowers Tuttle & Bailey, Div. of Interpace Corp. P.O. Box 1313 215 Warren Street New Britain, CT 06050	C, P, TA, VA, PRV, M
Quietaire Corp 505 North Hutcheson Houston, TX 77003	C, P, PRV
Revcor Inc. 251 Edwards Street Carpentersville, IL 60110	TA, P, C, M
Rotron Inc. Dubois Road Shokan, NY 12481	C, P
Robinson Industries, Inc. P.O. Box 100 Zeleinople, PA 16063	C, TA, P, M
Sheldons Manufacturing Corp. 1400 Sheldon Drive Elgin, IL 60120	C, TC, TA, VA, P, M

Manufacturer	Product
Spendrup Fan Co. 746 Ouray Avenue Grand Junction, CO 81501	VA
Strobic Air Corp. 207 Bunting Avenue Tenton, NJ 08611	TA, VAA, P
Shipman Industries, Inc. 530 Riedlin Avenue Covington, KY 41012	PRV
Stanley Industrial Corp. 6393 Powers Avenue Jacksonville, FL 32217	P, PRV
Swartwout Industries Inc. P.O. Box 1233 Sherman, TX 75090	TA, VA, C, P, PRV
Sundance-Byco P.O. Box 1405 Greeley, CO 80632	C
Tempmaster Corp 1222 Ozark Street N. Kansas Street North Kansas City, MO 64116	C
Thermador/Waste King 5119 District Blvd. Los Angeles, CA 90040	PRV
Torin Corp. 1 Kennedy Drive Torrington, CT 06790	C, TA, VA, P, M
The Trane Co. 3600 Pammel Creek Road LaCrosse, WI 54601	C, VA
Twin City Fan & Blower Co. 550 Kasota Avenue Minneapolis, MN 55414	C, TA, VA, P, M
Universal Cooperatives, Inc. Manufacturing Division 1000 N. Union Avenue Alliance, OH 44601	P
Ventco Inc. P.O. Box 55589-TR Houston, TX 77055	TA, P
Western Engineering & Manufacturing Co. 4114 Glencoe Avenue Marina Del Rey, CA 90291	TA, VA, P, PRV
Westinghouse Electric Corp. Sturtevant Division Damon Street Boston, MA 12136	C, TA, VA, M
Woods Fan Division The English Electric Co. 500 Executive Blvd. Elmsford, NY 10523	VAA, TA, P, M
York Division Borg Warner Corp York, PA 10745	C