



ESDU 79037

Issued December 1979
Reprinted at Amendment B
March 2007

A guide to fan selection and performance

Endorsed by
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(Continued on inside back cover)

A GUIDE TO FAN SELECTION AND PERFORMANCE

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A GUIDE TO FAN SELECTION AND PERFORMANCE

1. NOTATION

		Units
A	fan casing dimension, (see Sections 4 and 7.2.2); cross-sectional area of airway according to context	m, m ²
B	fan casing dimension, see Sections 4 and 7.2.2	m
C	fan casing dimension, see Sections 4 and 7.2.2	m
D_T	impeller or rotor diameter at tip, see Sketch 3.7	m
D_H	impeller or rotor diameter at hub, see Sketch 3.7	m
d	specific diameter defined Section 3.2	–
f	frequency	Hz
f_B	blade passing frequency	Hz
J	number of sound sources	–
K	pressure loss coefficient	–
K_w	fan specific sound power level	dB
k	number of blades on impeller	–
L_p	sound pressure level	dB
L_w	sound power level	dB
L_w^*	base sound power level	dB
N	rotational speed of impeller	rad/s
n	specific speed defined in Section 3.2	rad
p	root mean square sound pressure	Pa
p_d	dynamic pressure, see Section 3.1.2 for definition	Pa
$p_{f, d}$	fan dynamic pressure defined in Section 3.1.2	Pa
p_s	static pressure associated with system	Pa
p_t	total pressure associated with system	Pa
p_0	root mean square reference sound pressure	Pa
$\Delta p_{f, s}$	fan static pressure rise defined in Section 3.1.2	Pa

$\Delta p_{f, t}$	fan total pressure rise defined in Section 3.1.2	Pa
$P_{f, s}$	fluid static power produced by fan defined in Section 3.1.2	W
$P_{f, t}$	fluid total power produced by fan defined in Section 3.1.2	W
P_R	power input to fan impeller	W
q	volume flow rate through fan	m ³ /s
r	distance	m
T	reverberation time	s
U	mean flow velocity, $U = q/A$	m/s
V	volume of reverberant test enclosure	m ³
W	sound power	W
W_H	width of impeller flow passage at hub, see Sketch 3.7	m
W_T	width of impeller flow passage at tip, see Sketch 3.7	m
W_0	reference sound power	W
η	fan efficiency expressed as a percentage, defined in Section 3.1.2	per cent
σ	relative density* re 1.2 kg/m ³ , i.e. $\sigma = \rho/\rho_0$	–
ρ	density of gas flowing through fan, nominally at inlet	kg/m ³
ρ_0	density of air at standard conditions, i.e. $\rho_0 = 1.2 \text{ kg/m}^3$	kg/m ³
ϕ	flow coefficient defined in Section 3.2	rad ⁻¹
ψ	pressure coefficient defined in Section 3.2	rad ⁻²

* This definition of relative density should not be confused with the ratio of the density of water vapour to that of dry air sometimes associated with the term "relative density".

Subscripts

ext	relates to conditions external to system
H	relates to conditions at impeller hub
i	refers to fan inlet quantity
j	quantity associated with j^{th} octave band or measuring position
m	refers to mean value of quantity
max	refers to maximum value of quantity

<i>o</i>	refers to fan outlet quantity
<i>s</i>	refers to static pressure or quantity related to fan static pressure rise
<i>t</i>	refers to total pressure or quantity related to fan total pressure rise
<i>T</i>	relates to condition at impeller tip

2. INTRODUCTION

This Item contains introductory material to assist the non-specialist with fan selection and tender appraisal. It also presents characteristic properties of the major categories of fans in a non-dimensional form that allows estimates of size, power requirements and noise emission to be made for given combinations of flow rate and pressure rise.

To a non-specialist, the particular meanings given to certain terms in the context of fans may be confusing. In addition, use is made in this Item of special non-dimensional groups for generalising the characteristics of fans. The Item therefore first leads the user, in Section 3, through comprehensive definitions and explanations of the conventions and terminology related to fans.

For certain applications, *e.g.* the handling of airborne fibres, certain types of fans are a prerequisite. Section 4 identifies the salient features of the major fan categories and presents their typical characteristics in a non-dimensional form. It will be found, however, that for many applications a choice of fans is available.

Here, apart from cost, a fan may be selected (i) according to its performance, *e.g.* stability of operation, ease of control, power consumption, *etc.*, (ii) according to its mechanical arrangement, *e.g.* convenience of installation, self-cleaning blade properties, *etc.*, or (iii), because of noise emission advantages. Users familiar with fans, their terminology and characteristics, may refer directly to Section 7 which, for several fan types operating at a given duty, enables impeller powers, speeds and physical sizes to be estimated and compared. Alternatively, where noise emission is the primary concern, Section 9 may be used to provide estimates and comparisons of sound power spectra from several fan types. This section also gives many conversion formulae and other information to enable the diverse presentations of noise data which are found in manufacturers' literature to be compared.

For users less familiar with the characteristics of fans, Sections 5 and 6 are necessary preliminary reading. Before selecting a fan, it is essential to have considered stability of operation, flexibility of system operation including growth potential, and special system requirements such as back-up/stand-by capability. Section 5 explains these system characteristics and requirements while Section 6 gives guidance on how they and the characteristics of fans may be successfully integrated.

In the majority of applications, fans are driven by electric motors on whose properties Section 8 gives general information. It covers, for example, the starting problems that can arise with certain fan and motor combinations and outlines the methods of speed control that are used for fans.

2.1 Scope

The categories of fans dealt with are: axial flow, centrifugal flow, mixed flow and crossflow. The information is limited to fans where the density of the gas varies less than ten per cent between the fan inlet and outlet; typically, this corresponds to a maximum fan total pressure rise, $\Delta p_{f,t}$, of 10^4 Pa (approximately 40 inches of water). Compressors and some high pressure ratio fans are thus outside the scope of this Item.

3. FAN TERMINOLOGY AND BASIC RELATIONSHIPS

The following paragraphs enlarge on the terms associated with fans that are not otherwise in common usage within internal flow fluid mechanics. Section 3.1 deals with the terms surrounding fan and system performance and Section 3.2 enlarges on the non-dimensional groups used to portray the properties of fans in this Item. Section 3.3 presents the scaling approximations that are known as the Fan Laws and Section 3.4 describes the mechanical conventions for fans including methods for depicting the discharge and motor positions of centrifugal fans.

3.1 Performance Terminology

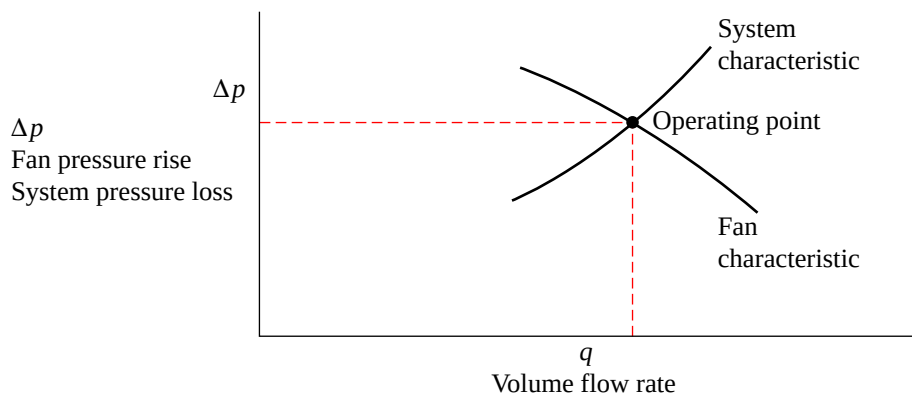
3.1.1 General terms

Characteristic

The characteristic of a device or system is the relationship that links important primary variables associated with its operation. The most commonly used fan characteristic is the relationship between pressure rise and volume flow rate for a given impeller speed. Similarly the relationship between pressure loss and volume flow rate is the most commonly used system characteristic. Fan pressure rise and flow rate characteristics are often expressed non-dimensionally in terms of ψ , the pressure coefficient, and ϕ , the flow coefficient. These coefficients, which are fully defined in Section 3.2, are independent of impeller speed and fan size.

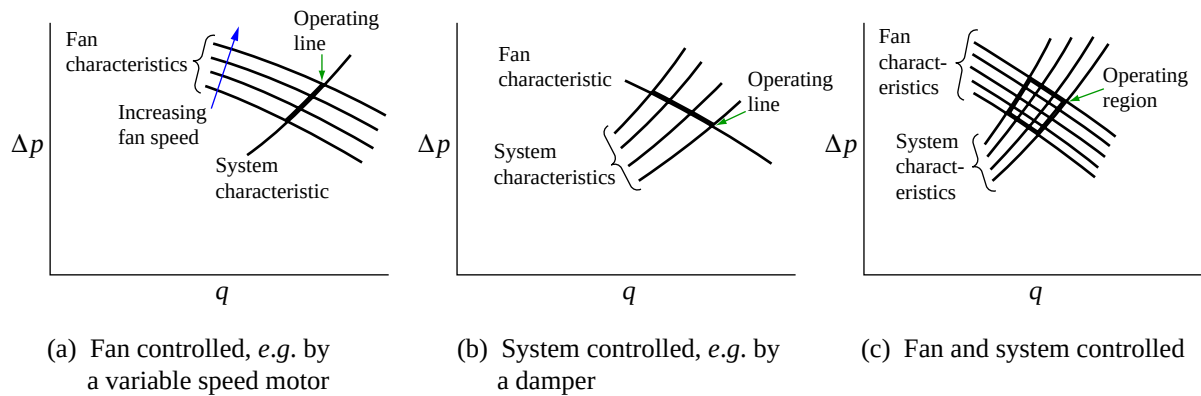
Operating Point

An operating point is defined as the fan pressure rise/volumetric flow rate condition where the fan and system are in a stable equilibrium. This corresponds to the condition at which the fan pressure rise/flow rate characteristic intersects the system pressure loss/flow rate characteristic.



Sketch 3.1 Definition of operating point

On a system or fan that has a variable characteristic, as may be obtained using either system or fan control devices, the operating point will also vary to describe an "operating line" and, where both are independently variable, an "operating region" will result, see Sketch 3.2.



Sketch 3.2 Influence of fan and system controls

3.1.2 Terms related to fan pressure rise

Fan total pressure rise, $\Delta p_{f,t}$, often loosely termed "fan total pressure", is defined as the rise in total pressure produced by the fan between its inlet and outlet. It can be defined with an open inlet, see Sketch 3.3a, or open outlet (Sketch 3.3b) or as an in-duct quantity (Sketch 3.3c). Thus for $\Delta p_{f,t}$ to be consistent for the different fan inlet and outlet arrangements, variations of intake and outlet losses arising from the different geometries are by convention neglected.

Fan dynamic pressure, $p_{f,d}$, is defined as the fluid dynamic pressure corresponding to the mean through flow velocity, U_o , at the fan outlet based on the total outlet area without any deductions for motors, fairings or other bodies. Thus, for the low Mach number flow rates to which this Item applies, the fan dynamic pressure* may be taken as

$$p_{f,d} = \frac{1}{2} \rho U_o^2. \quad (3.1)$$

Fan static pressure rise, $\Delta p_{f,s}$, often loosely termed "fan static pressure", is defined as the difference between the fan total pressure rise and the fan dynamic pressure, i.e.

$$\Delta p_{f,s} = \Delta p_{f,t} - p_{f,d}. \quad (3.2)$$

The fan static pressure rise is thus equal to the gauge static pressure at the fan outlet when the fan draws in air from the atmosphere through a well shaped intake. Note that the definition of this term is peculiar to fans and is not consistent with the normal meaning of static pressure rise. The term is derived from methods of testing the performance of fans.

Special care must be taken with calculations involving $\Delta p_{f,s}$ since it will not correspond to the actual static pressure rise between the fan inlet and outlet unless the fan draws in directly from the atmosphere or a large plenum, see Sketch 3.3a. When a fan has an inlet and outlet plane of the same through-flow area and discharges directly to the atmosphere, $\Delta p_{f,s}$ is equal to the total pressure rise across the fan for negligible density changes, see Sketch 3.3b. Note that when discharge terminals or intakes are coupled directly to a fan, they are, by convention, regarded as an integral part of the fan and not of the system. However, when they are remote from the fan such that the fan inlet and outlet stations are within the confines of the connecting duct work, the fan total pressure rise, $\Delta p_{f,t}$, and not fan static pressure rise, $\Delta p_{f,s}$, will equal the static pressure rise between the fan inlet and outlet, again for negligible density changes (see

* Strictly, dynamic pressure is the difference between total pressure and static pressure and is thus greater than $\frac{1}{2} \rho U^2$. However, for air at Mach numbers below 0.2, i.e. 68 m/s flow velocity at standard conditions, the difference is less than one per cent.

Sketch 3.3c).

The relationships in Sketch 3.3 all neglect variations in the inlet losses and the velocity profiles of the flow entering and leaving the fan. They also neglect density changes between inlet and outlet. Although the values of pressure differences calculated using the above relationships may differ slightly from those obtained by integrating the local velocities and pressures measured within the inlet and outlet cross sections, for most practical purposes they are compatible with the pressure rises required to match duct system pressure losses which are also usually calculated making similar assumptions*.

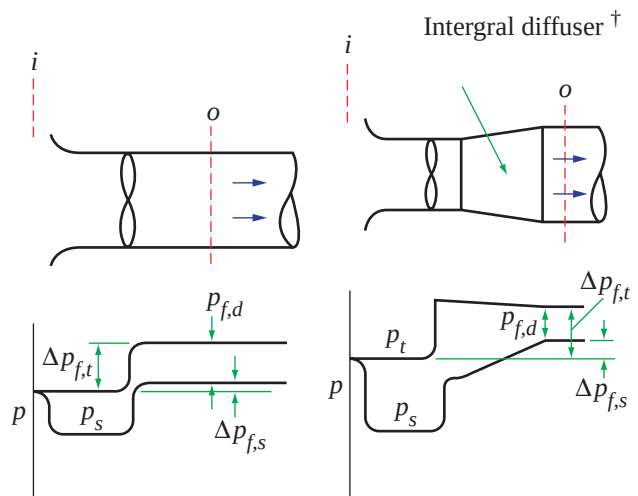
(a) **Free intake**

$$(p_t)_o - (p_t)_i = \Delta p_{f,t},$$

$$(p_s)_o - (p_s)_i = \Delta p_{f,t} - \frac{1}{2}\rho U_o^2,$$

$$(p_s)_o - (p_s)_i = \Delta p_{f,s}.$$

$$\text{Note: } (p_s)_i = (p_t)_i.$$



(b) **Free outlet**

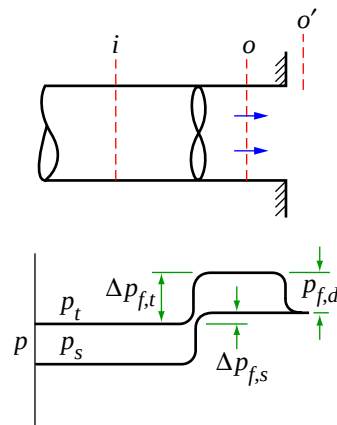
$$(p_t)_o - (p_t)_i = \Delta p_{f,t},$$

$$(p_t)_o - (p_t)_{o'} = \frac{1}{2}\rho U_o^2,$$

because fan dynamic pressure is lost at the free discharge, i.e.

$$(p_s)_{o'} = (p_t)_{o'} = (p_t)_o - \frac{1}{2}\rho U_o^2.$$

$$\text{Hence } (p_t)_{o'} - (p_t)_i = \Delta p_{f,s}.$$



* Such duct pressure loss calculations should nevertheless include an allowance for swirl which can be pronounced downstream of tube-axial fans. With the exception of diffusers, the effect of swirl is usually to increase the pressure loss of components.

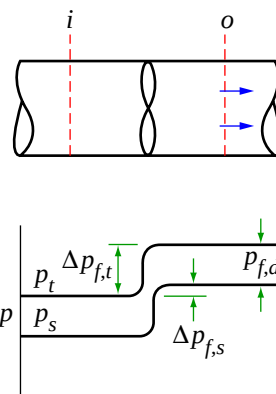
† A diffuser that is directly coupled to a fan may also be regarded as an integral part of the fan in some conventions. Here the fan dynamic pressure is that corresponding to the outlet through-flow area of the diffuser. This is illustrated in Sketch 3.3a.

(c) **Ducted inlet and outlet**

$$(p_t)_o - (p_t)_i = \Delta p_{f,t},$$

$$(p_s)_o - (p_s)_i = \Delta p_{f,t},$$

$$(p_s)_o - (p_t)_i = \Delta p_{f,s}.$$



Sketch 3.3 Fan ducting arrangements

Air power, P_o , is the power delivered by a fan to the gas and is given by

$$P_o = q \Delta p_f, \quad (3.3)$$

where the fan pressure rise, Δp_f , may be taken as either the fan static pressure rise or fan total pressure rise, *i.e.*

$$P_{o,s} = q \Delta p_{f,s} \quad (3.4)$$

or
$$P_{o,t} = q \Delta p_{f,t}. \quad (3.5)$$

Note that $P_{o,t}$ and not $P_{o,s}$ is the actual power gained by the fluid. Air static power, $P_{o,s}$, is an artifice used in the definition of fan static efficiency, η_s , described below.

Efficiency, η , is the ratio of air power, P_o , to the power input to the propeller, P_R , expressed as a percentage, *i.e.*

$$\eta = \frac{P_o}{P_R} \times 100 \text{ per cent.} \quad (3.6)$$

Air power and hence efficiency may be taken as either a static or total quantity, *i.e.*

$$\eta_s = \frac{P_{o,s}}{P_R} \times 100 \text{ per cent} \quad (3.7)$$

or
$$\eta_t = \frac{P_{o,t}}{P_R} \times 100 \text{ per cent.} \quad (3.8)$$

The static and total efficiencies are related by

$$\eta_s = \eta_t \Delta p_{f,s} / \Delta p_{f,t}. \quad (3.9)$$

Because $P_{o,t}$ is the actual power delivered to the fluid, the fan total efficiency and not static efficiency is the quantity representative of the fan energy conversion efficiency. The fan static efficiency is always less than the total efficiency but this, of course, does not mean that there is any difference in the impeller power required for a given duty; the impeller power, P_R , obtained from Equation (3.10) is identical to that obtained from Equation (3.11), *i.e.*

$$P_R = q \Delta p_{f,s} / \frac{\eta_s}{100} \quad (3.10)$$

or

$$P_R = q \Delta p_{f,t} / \frac{\eta_t}{100} \quad (3.11)$$

Nevertheless, for many systems, a fan offering a high value for η_s may be more suitable than another offering a higher η_t but lower* η_s . This is because many systems include a discharge such that the fan dynamic pressure is lost from the total pressure rise, see Sketch 3.3b. The fan static pressure rise is in such cases the total pressure difference remaining to overcome the system resistance.

On systems where the fan is ducted on both the inlet and outlet (see Sketch 3.3c), the fan total pressure rise, $\Delta p_{f,t}$, should be used to match the fan to the system losses. However, if such a system discharges from a terminal of approximately the same through-flow area as that of the fan exit plane, a fan dynamic pressure will be lost and an optimised fan static efficiency will result in a minimum impeller power, P_R .

On closed-loop systems or systems incorporating efficient diffusers, the fan total efficiency should normally be optimised.

3.2 Dimensionless Groups

Dimensionless groups are used in this Item for characterising the properties of fans of similar fundamental design but of different scale and, to a limited extent, of different proportion. Four groups are used:

specific diameter, d ,
specific speed, n ,
flow coefficient, ϕ , and
pressure coefficient, ψ .

The flow coefficient, ϕ , has a unique definition, *i.e.*

$$\phi = \frac{q}{D_T^3 N W_T / D_T} \quad (3.12)$$

Thus, ϕ is proportional to the ratio of two volume flow rates: that on the numerator is the actual flow rate through the fan while that in the denominator is equal to the product of the impeller tip speed and the impeller surface area circumscribed by the blade tips[†].

* Such a situation is possible when a fan with a large through-flow area produces a certain total pressure rise. A fan of inherently more efficient design (higher η_t) but smaller through-flow area producing the same total pressure rise at the same volume flow rate may have a lower static efficiency. The static efficiency value is penalised by the higher fan dynamic pressure associated with the smaller through-flow area.

† For axial-flow and mixed-flow fans, the impeller surface area is ascribed a nominal value proportional to the disc area such that $W_T/D_T = 1$. See comments later in this Section.

The other groups may be defined either on a basis of fan total pressure rise or fan static pressure rise. Thus

$$d_s = D_T \left[\frac{W_T/D_T}{q} \right]^{1/2} \left[\frac{\Delta p_{f,s}}{\sigma \rho_0} \right]^{1/4} \quad (3.13)$$

or

$$d_t = D_T \left[\frac{W_T/D_T}{q} \right]^{1/2} \left[\frac{\Delta p_{f,t}}{\sigma \rho_0} \right]^{1/4}; \quad (3.14)$$

d is a measure of the compactness of the fan. Generally, large diameter wide impellers are best suited to high flow rates but small narrow impellers are best suited to high resistance systems.

$$n_s = N \left(\frac{q}{W_T/D_T} \right)^{1/2} \left(\frac{\sigma \rho_0}{\Delta p_{f,s}} \right)^{3/4} \quad (3.15)$$

or

$$n_t = N \left(\frac{q}{W_T/D_T} \right)^{1/2} \left(\frac{\sigma \rho_0}{\Delta p_{f,t}} \right)^{3/4}; \quad (3.16)$$

n is a measure of the impeller speed necessary to produce a fan pressure rise at a given flow rate.

$$\psi_s = \frac{\Delta p_{f,s}}{D_T^2 N^2 \sigma \rho_0} \quad (3.17)$$

or

$$\psi_t = \frac{\Delta p_{f,t}}{D_T^2 N^2 \sigma \rho_0}; \quad (3.18)$$

ψ is proportional to the ratio of two pressures: that on the numerator is the actual fan pressure rise while that on the denominator is the dynamic pressure corresponding to the impeller tip speed.

Any of these groups may be expressed in terms of any other two, *i.e.*

$$d = \frac{1}{n \psi^{1/2}} = \frac{1}{n^{1/3} \phi^{1/3}} = \frac{\Psi^{1/4}}{\phi^{1/2}}, \quad (3.19)$$

$$n = \frac{1}{d^3 \phi} = \frac{1}{d \psi^{1/2}} = \frac{\phi^{1/2}}{\psi^{3/4}}, \quad (3.20)$$

$$\phi = \frac{1}{n d^3} = \frac{\psi^{1/2}}{d^2} = n^2 \psi^{3/2}, \quad (3.21)$$

$$\psi = \frac{1}{n^2 d^2} = d^4 \phi^2 = \frac{\phi^{2/3}}{n^{4/3}}. \quad (3.22)$$

It may be noted that, contrary to convention, the definitions of d , n and ϕ used in this Item involve the impeller width to diameter ratio, W_T/D_T . The inclusion of this term allows the performance of both double inlet centrifugal fans and cross-flow fans to be correlated in terms of specific speed without having to treat

these fans as special cases.

Sometimes, however, when it is desirable to illustrate the performance differences between separate categories of fans, conventional definitions of the groups specific diameter, specific speed and flow coefficient, that do not include the term W_T/D_T have an advantage. This is because the term W_T/D_T , which correlates the effect of different impeller geometries, tends to vary more between categories of fans than within them. By omitting the term, therefore, the difference between, for example, the flow coefficients of two different categories of fans each operating at peak efficiency are exaggerated. This approach is adopted in Sketches 4.2 and 4.3 for illustrating the inherent characteristics of different fan types. The groups specific diameter, specific speed and flow coefficient defined in this way are indicated by a prime ('), e.g.

$$\phi' = \frac{q}{D_T^3 N}.$$

The success of conventional definitions in correlating the performance of single-inlet centrifugal fans lies in the range of W_T/D_T being limited by mechanical and fluid flow constraints within each category of fan. Therefore, in the definitions of ϕ' , but especially d' and n' (where its influence is smaller), it is an acceptable approximation to absorb the effect of W_T/D_T into correlations of, say, d' versus n' for each category of centrifugal fan.

For axial-flow and mixed-flow fans, W_T/D_T is analogous to some function of the impeller hub to tip diameter ratio but, because of design constraints, it is not an independent variable in a fan optimised for a particular duty. Thus, in common with the definitions of d' , n' and ϕ' , the ratio W_T/D_T should be assigned a value of unity for evaluating d , n and ϕ for axial and mixed-flow fans, i.e. $d = d'$, $n = n'$ and $\phi = \phi'$.

3.3 Fan Laws

In Section 3.2, the non-dimensional groups ψ and ϕ are assembled from a combination of variables governing the performance of fans, i.e. D_T , N , W_T and σ , and variables that reflect their performance, i.e. q and $\Delta p_{f,s}$ (or $\Delta p_{f,t}$).

Thus, by assigning a particular value to ψ and knowing the values of D_T , N and ρ associated with a fan and its operation, the fan pressure rise is defined. Similarly, by assigning a value to ϕ and knowing D_T , N and W_T/D_T , the flow rate is defined. Thus a fan characteristic consisting of a function of Δp_f versus q can, for known values of D_T , N , W_T/D_T and σ , be alternatively represented by a function of ψ versus ϕ .

The function, $\psi(\phi)$, representing the characteristic of a given fan may be used to represent the characteristic of any geometrically similar* fan. If the system characteristic is known, the operating point may also be defined as a pair of co-ordinates, ψ and ϕ . An operating point defined in this way is known as a "point of rating". Geometrically similar fans having the same point of rating (i.e. operating at the same values of ψ and ϕ) have properties that are related by expressions known as the "Fan Laws". For example, if the subscript "1" is used to denote one fan and the subscript "2" another geometrically similar fan and if both operate at the same point of rating, the following expressions apply:

* Two fans are geometrically similar when one fan and a scale factor completely defines the other.

$$\frac{q_1}{(D_T^3)_1 N_1 \left[\frac{W_T}{D_T} \right]_1} = \phi_1 = \phi_2 = \frac{q_2}{(D_T^3)_2 N_2 \left[\frac{W_T}{D_T} \right]_2} \quad (3.23)$$

and

$$\frac{(\Delta p_f)_1}{(D_T^2)_1 N_1^2 \sigma_1 \rho_0} = \psi_1 = \psi_2 = \frac{(\Delta p_f)_2}{(D_T^2)_2 N_2^2 \sigma_2 \rho_0} \quad (3.24)$$

For geometrical similarity, $(W_T/D_T)_1 = (W_T/D_T)_2$ and the expressions may be rearranged into the form

$$(\Delta p_f)_2 = (\Delta p_f)_1 \left[\frac{N_2}{N_1} \right]^2 \left[\frac{(D_T)_2}{(D_T)_1} \right]^2 \frac{\sigma_2}{\sigma_1} \quad (3.25)$$

and

$$q_2 = q_1 \frac{N_2}{N_1} \left[\frac{(D_T)_2}{(D_T)_1} \right]^3 \quad (3.26)$$

The fan pressure coefficient, ψ , and pressure rises, $(\Delta p_f)_1$ and $(\Delta p_f)_2$, in Equations (3.24) and (3.25) may relate to either the fan static pressure rise or fan total pressure rise.

Equations (3.25) and (3.26) can be combined to give a relationship between the impeller powers, $(P_R)_1$ and $(P_R)_2$, i.e.

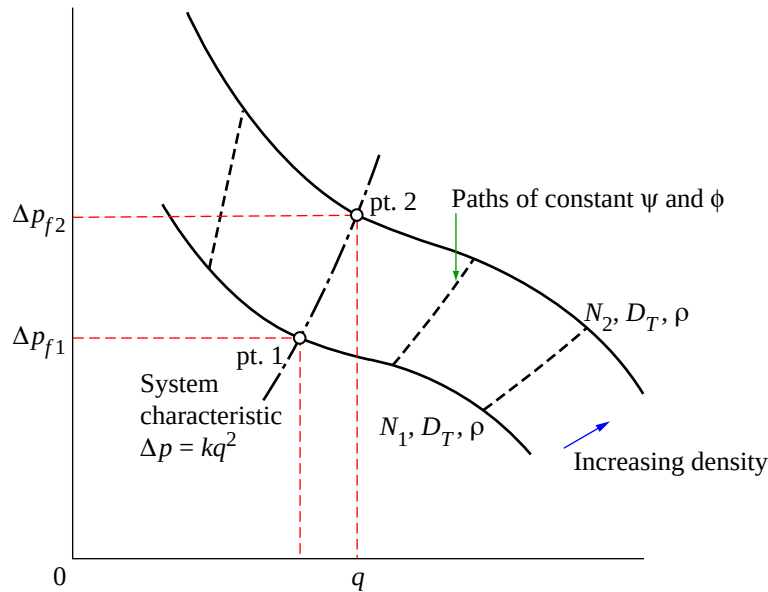
$$(P_R)_2 = (P_R)_1 \left[\frac{N_2}{N_1} \right]^3 \left[\frac{(D_T)_2}{(D_T)_1} \right]^5 \frac{\sigma_2}{\sigma_1} \quad (3.27)$$

Using a similar argument regarding the governing variables of fan noise, a relationship can be derived between the sound power levels, $(L_w)_1$ and $(L_w)_2$, i.e.

$$(L_w)_2 = (L_w)_1 + 20 \log_{10} \frac{\sigma_2}{\sigma_1} + 50 \log_{10} \frac{N_2}{N_1} + 70 \log_{10} \frac{(D_T)_2}{(D_T)_1} \quad (3.28)$$

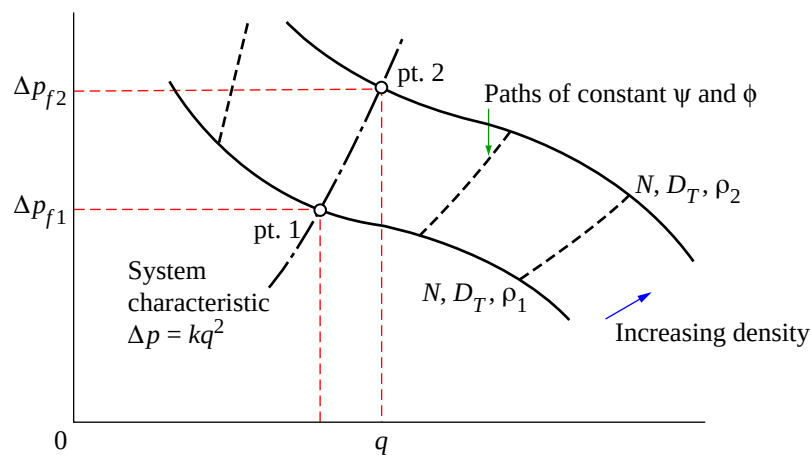
Equation (3.28) is derived from a combination of other fan laws (Equations (3.25) and (3.26)) and the function used in the noise estimation procedure that relates a change in pressure rise or flow rate to a change in sound power level. Equation (3.28) is therefore subject to the same conditions of applicability as the fan laws and the noise estimation method (given in Section 9.6). Furthermore, its use for predicting sound power level changes within individual octave bands, resulting from speed or diameter changes, is limited to those changes that are sufficiently small to maintain the blade passing frequency (see Equation (9.13)) in the same octave band.

Sketch 3.4 demonstrates one application of the fan laws. The pressure and volume flow rate characteristic for a particular fan is known at a rotational speed N_1 , and it is required to predict the characteristic for the fan at a new speed N_2 . Using Equations (3.25) and (3.26), point 1 of Sketch 3.4 is transformed, moving along a parabolic path of constant ψ and ϕ due to the increase in speed, to a new point 2. Similarly every other point on the original performance characteristic may be mapped onto the new characteristic.



Sketch 3.4 Effect of change of speed on fan characteristic

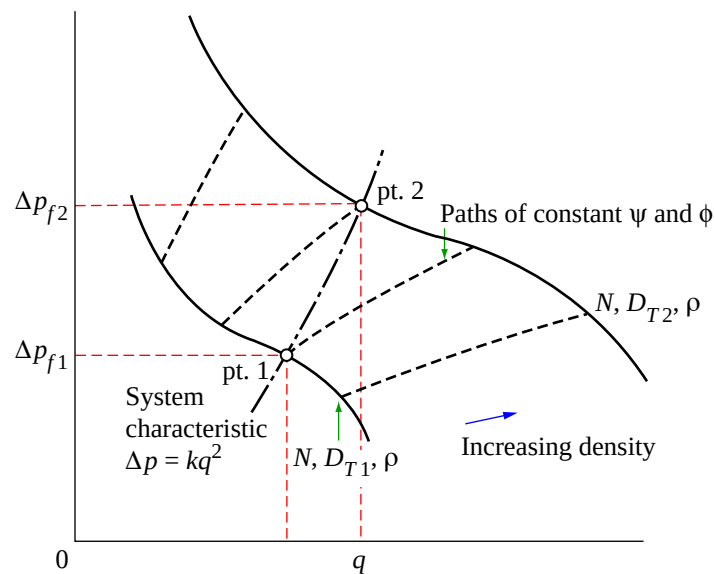
In this particular case, the parabolic path linking point 1 to point 2 coincides with the system characteristic. This is because the effect of speed implies a fan pressure rise that is proportional to q^2 , a characteristic that follows most system characteristics. The variation of density also affects the fan characteristics in a way that approximately balances the effect on most systems without changing the fan point of rating as illustrated in Sketch 3.5.



Sketch 3.5 Effect of change of density on fan characteristic

Sketch 3.6 however demonstrates another application of the fan laws for predicting, from the known characteristic of one fan, the characteristic of a similar but larger fan operating at the same impeller speed.

The paths representing constant points of rating no longer coincide with the system characteristic. It is thus not possible to predict operating point 2 from operating point 1 alone; instead a knowledge of a range of the original fan characteristic is necessary



Sketch 3.6 Effect of change of size on fan characteristic

Note that the fan laws are strictly only applicable for changes that involve exact geometrical similarity and for incompressible flows with constant Reynolds numbers. However, in the context of fans, the effect of Reynolds number variations is usually small and it will often be found that fans of different sizes available within a manufacturer's line may, for practical purposes, be regarded as geometrically similar.

3.4 Mechanical Conventions

3.4.1 Major impeller dimensions

The major impeller dimensions referred to in this Item involve diameter and width. They are illustrated in Sketch 3.7.

3.4.2 Direction of rotation

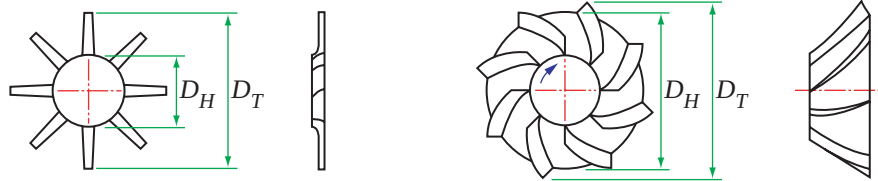
The standard²⁵ method of specifying the direction of rotation for centrifugal fans is to view the fan from the drive side and indicate whether the rotation required is to be clockwise or anti-clockwise. Note that the drive side of a centrifugal fan is considered to be the side opposite the inlet even when the actual position is on the inlet side.

The standard²⁵ method of specifying the direction of rotation for axial and mixed-flow fans is to view the fan from the discharge end and indicate whether the rotation required is to be clockwise or anti-clockwise.

3.4.3 Discharge and motor position

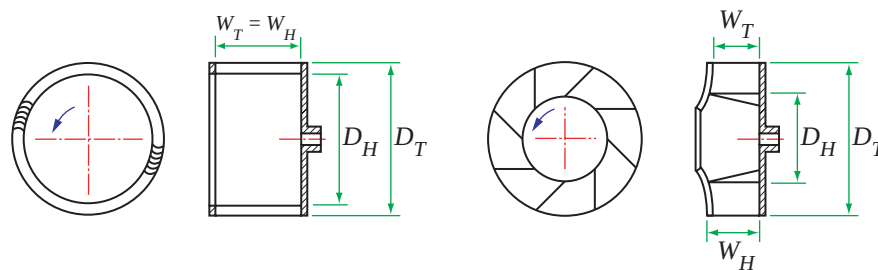
As with the direction of rotation, the specification of discharge position is normally only necessary with centrifugal fans. Sketch 3.8 shows the usual method of indicating discharge position and rotation for a centrifugal fan²⁵.

The position of the motor for centrifugal fans should also be specified, *i.e.* either directly coupled or according to Sketch 3.9.

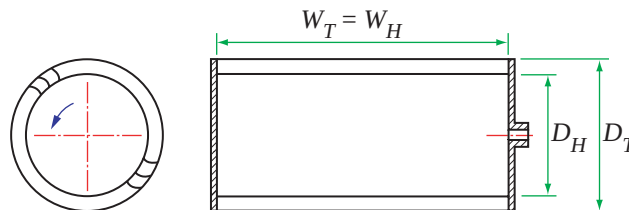


Axial-flow impeller

Mixed-flow impeller

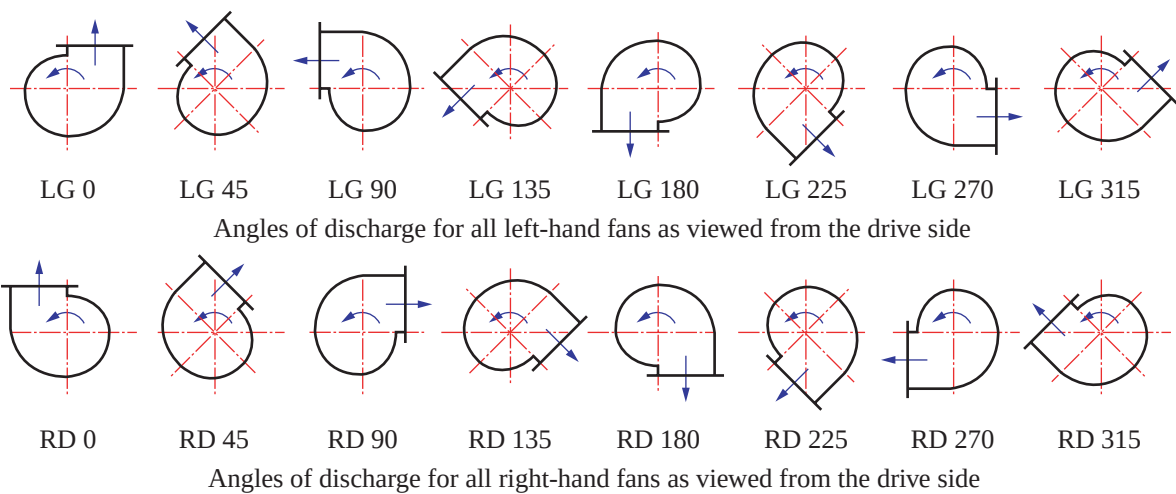


Centrifugal impellers

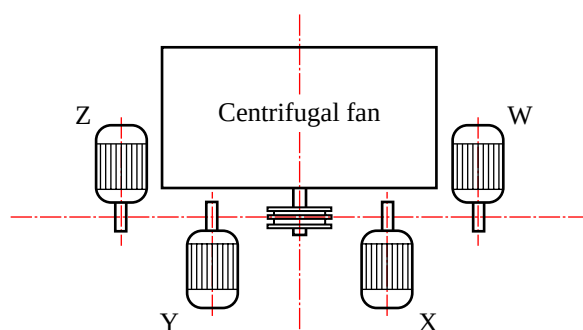


Cross-flow impeller

Sketch 3.7 Major impeller dimensions



Sketch 3.8 Rotation and discharge designs

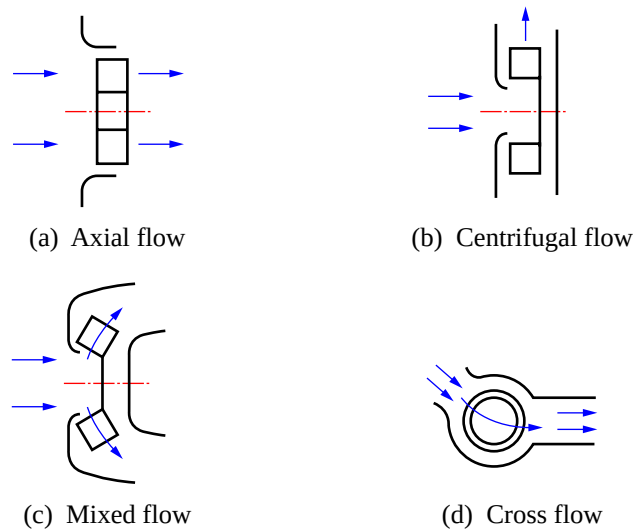


Sketch 3.9 Standard motor positions for centrifugal fans

Axial-flow fans may be driven either by a directly coupled motor or via vee belts to a motor mounted on the outside of the casing.

4. MAJOR CATEGORIES OF FANS

There are four main categories of fans: axial flow, centrifugal flow, mixed flow and cross flow. These categories are defined by the nature of the flow through the impeller blades, see Sketch 4.1.



Sketch 4.1 The major fan categories

Within each of the major fan categories, the following sub-categories are covered.

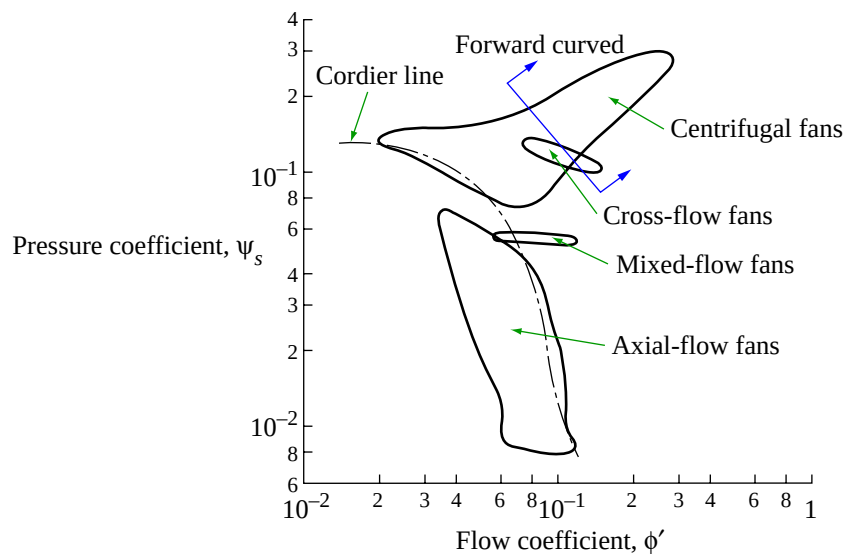
Axial Flow	Centrifugal Flow	Mixed Flow	Cross Flow
Propeller	Forward-curved		J-casing
Tube-axial	Radial-bladed	Axial-casing	S-casing
Guide-vane-axial	Backward-bladed		U-casing

Each type of fan has a different characteristic that may, or may not, be suited for the required duty. For example, a fan intended to blow cooling air slowly over electronic components will have a relatively large volume flow rate capacity but relatively low pressure gain. By contrast, a fan required to blow air through a filtration system offering a high flow resistance will have a relatively small volume flow rate capacity but relatively high pressure rise.

The characteristics of pressure rise and volume flow rate can be represented non-dimensionally in terms of ψ , the fan pressure coefficient*, and ϕ , the volume flow rate coefficient*. Fans of the same geometric proportions, though of different size and running at different speeds, when represented in terms of these coefficients can be compared directly. Another advantage of representing fan characteristics in terms of pressure and flow coefficients is for the comparison of fans from different categories although, for this purpose, it is customary to use the more conventionally defined flow coefficient†, ϕ' .

* Section 3.2 gives full definitions of these terms.

† The definitions of the non-dimensional groups flow coefficient, ϕ' , specific speed, n_s' , and specific diameter, d_s' , employed in Sketches 4.2 and 4.3 do not include the term (W_T/D_T) . Elsewhere in this Item, however, the groups are defined with a (W_T/D_T) term. See notes in Section 3.2 for a full explanation.



Sketch 4.2 Optimum efficiency fan static pressure coefficients for various types of fans

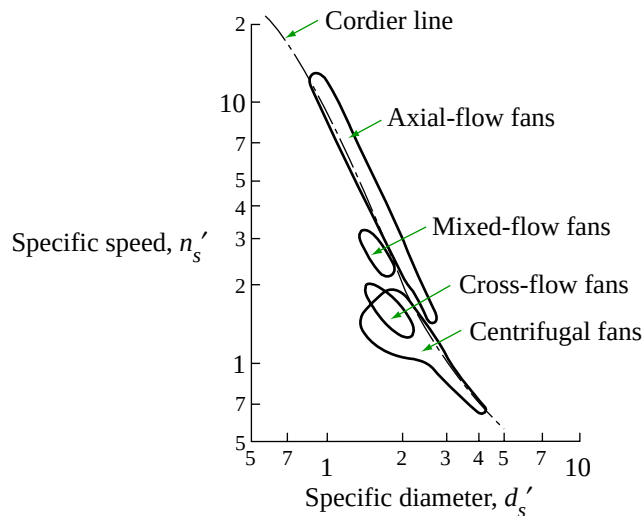
Sketch 4.2 compares the Cordier line with the pressure coefficient and flow coefficient[†], ϕ' , corresponding to optimum efficiency for the categories of fans dealt with in this Item. The Cordier line¹⁵ is intended to represent pressure coefficient and flow coefficient combinations that are produced by the most efficient designs of turbomachines themselves operating at optimum efficiency. It agrees acceptably well with most fan categories except the forward-curved and cross-flow types whose geometry is not compatible with the Cordier classes of turbomachines.

Sketch 4.2 also shows that axial-flow fans can only be designed to produce relatively low pressure coefficients. Centrifugal-flow fans, however, produce higher coefficients.

Within the envelope for axial-flow fans, the wide range of pressure coefficient is achieved by varying the solidity, *e.g.* a small number of blades gives a low pressure coefficient and vice-versa. Within the centrifugal fan envelope, the range of pressure coefficient is achieved by varying the impeller blade angles, *i.e.* backward-curved blades give the lowest pressure coefficients, radial blades give intermediate values and forward-curved, the highest. In addition, forward-curved blades produce the highest flow coefficients.

In the same way that ψ and ϕ' relationships can be used to denote the pressure rise and flow rate characteristics of a type of fan, other non-dimensional groups, namely d_s' , the specific diameter*, and n_s' , the specific speed*, may be used to typify the impeller diameter and operating speed relationship for each fan category.

* The definitions of the non-dimensional groups flow coefficient, ϕ' , specific speed, n_s' , and specific diameter, d_s' , employed in Sketches 4.2 and 4.3 do not include the term (W_T/D_T) . Elsewhere in this Item, however, the groups are defined with a (W_T/D_T) term. See notes in Section 3.2 for a full explanation..



Sketch 4.3 Diagram illustrating specific speed versus specific diameter for optimum efficiency conditions of various fans

Sketch 4.3 illustrates the relationship between specific speed and specific diameter for the different fan categories when each fan within each category runs at maximum efficiency. As with Sketch 4.2, it is noticeable how different fan categories monopolise different regions on the graphs. The sketch shows that, for a given impeller diameter, axial-flow fans run at higher impeller speeds than centrifugal fans.

It should be noted that Sketches 4.2 and 4.3 do not merely display patches of characteristics from a single fan in the vicinity of its maximum efficiency. Instead the curves in the sketches connect the ψ and ϕ' conditions of several fans in each category each operating at maximum efficiency. Detailed design variations between fans in the same category allow individual fans to be optimised for different duties within the ranges indicated in Sketches 4.2 and 4.3.

Suitability of pressure rise and volume flow rate is not the only criterion for selecting a fan. Consideration must be given to the shape of the pressure rise/flow rate characteristic, the efficiency, shape of the power characteristic, speed, size, inlet and outlet configuration and also the power and spectrum shape of the noise emitted from different fan types. In addition, the use of a certain type of fan may be determined by its mechanical suitability, *e.g.* for propelling airborne particulate matter.

The following Sections, 4.1 to 4.4, comment in some detail on general performance and mechanical features of different types of fans. Guidance on their noise characteristics is given in Section 9. Typical fan performance characteristics are illustrated in terms of ψ_s , the static pressure coefficient, and ϕ , the flow coefficient. Since these coefficients are directly proportional to the fan static pressure rise*, $\Delta p_{f,s}$, and the volume flow rate, q , the ψ_s versus ϕ charts are merely convenient methods of giving a more general presentation than their dimensional counterparts. Also included on the same charts in the following sections are typical fan static efficiencies and typical impeller power characteristics.

It should be noted that although the characteristics illustrated in Sections 4.1 to 4.4 give an indication of the behaviour of each category of fan, they are not wholly general within their category due to detailed design variations.

* Note that $\Delta p_{f,s}$ is not necessarily equal to the change in static pressure between the inlet and outlet of a fan. For a full explanation of fan static pressure rise, see Section 3.1.2

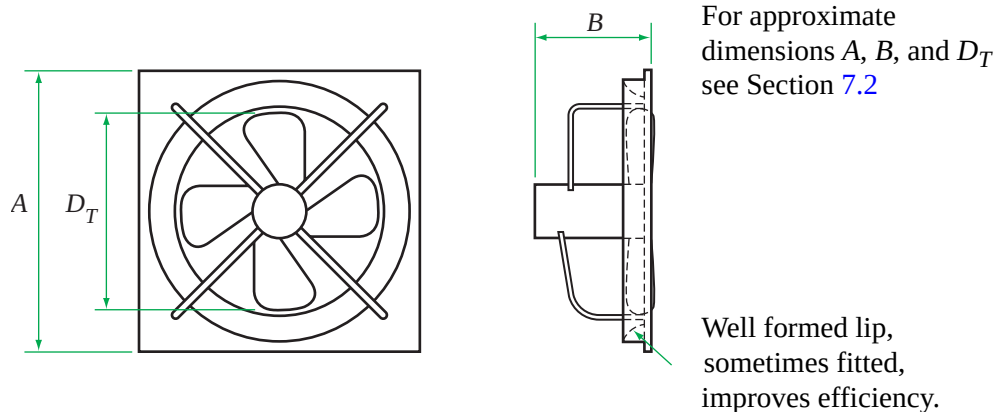
4.1 Axial-Flow Fans

Within the major category of axial-flow fans, there are four sub-categories: propeller fans, tube-axial fans, contrarotating fans and guide-vane-axial fans. Most axial-flow fans are available with many blade angle settings that in some cases may be adjusted, when stationary, by slackening a clamping mechanism in the impeller hub. Other, more sophisticated fans, have a variable pitch facility that can alter the impeller blade angles while the fan is in operation.

Changing the blade angles predominantly affects the flow coefficient, ϕ . Fans optimised to produce high flow coefficients are set with large blade angles that can give rise to stalling if the flow is over-throttled. Stalling is manifested by a rapid decrease in both the fan pressure rise and the efficiency as the flow rate is reduced, see for example, Sketch 4.11.

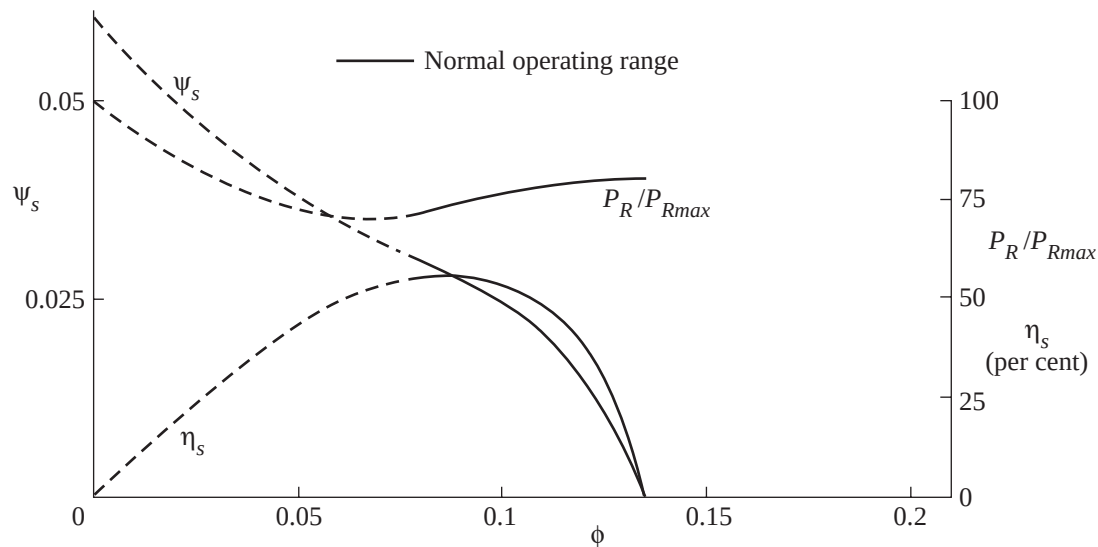
Whilst the blade angle is the chief design variable governing flow coefficient, the impeller solidity is the main variable governing the fan pressure coefficient. Generally, high solidity impellers (large number of blades and, or, high hub to tip diameter ratios) generate high pressure coefficients more efficiently than low solidity impellers which operate more efficiently at low pressure coefficients.

4.1.1 Propeller fan



Sketch 4.4 Propeller fan

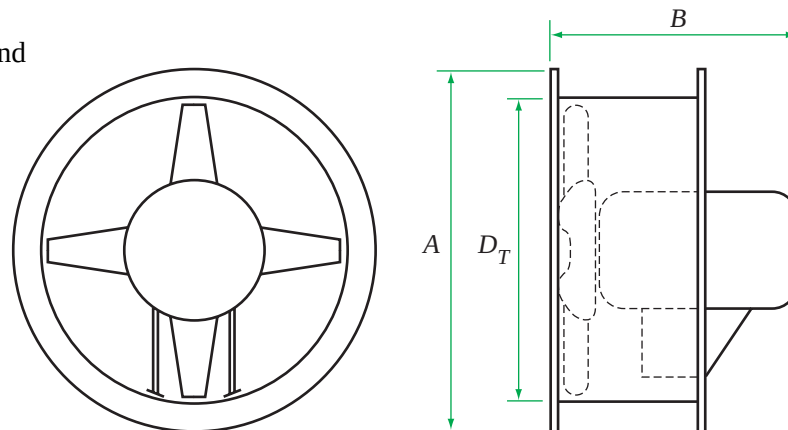
Applications for this fan mostly involve moving air through a partition from one open space to another such as the supply or removal of air through the roofs or walls of buildings or through the cabinets of electronic equipment. The shape of the lips of the mounting ring has a marked effect on the efficiency. Small sizes of these units, *e.g.* $D_T < 0.5$ m, often have impellers moulded from plastic but the larger sizes usually have impellers of pressed steel. Except for very large units, *e.g.* $D_T > 1.5$ m, most are supplied with integral electric motors and are restricted to speeds near the synchronous pole speeds (see Section 8).



Sketch 4.5 Pressure coefficient, flow coefficient, efficiency, and impeller power characteristics of a typical propeller fan

4.1.2 Tube-axial fan

For approximate dimensions A , B and D_T , see Section 7.2

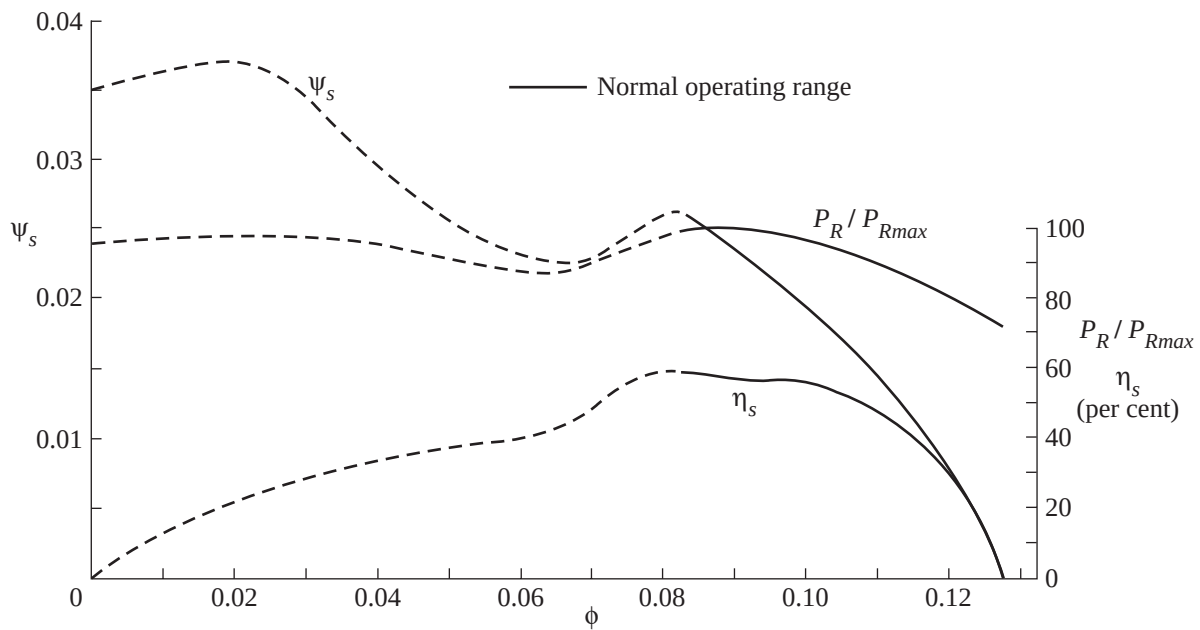


Sketch 4.6 Tube-axial fan

Applications for this fan are found in many air-conditioning systems that require a high volume flow rate but small pressure rise. In contrast to centrifugal flow fans, this type of fan can be installed directly in a duct. Design variations involving the blade geometry and number of impeller blades can significantly affect both the pressure rise and flow rate.

The swirling motion of the discharged flow dissipates energy that reduces the efficiency of the fan and can increase the pressure loss of at least the first component downstream.

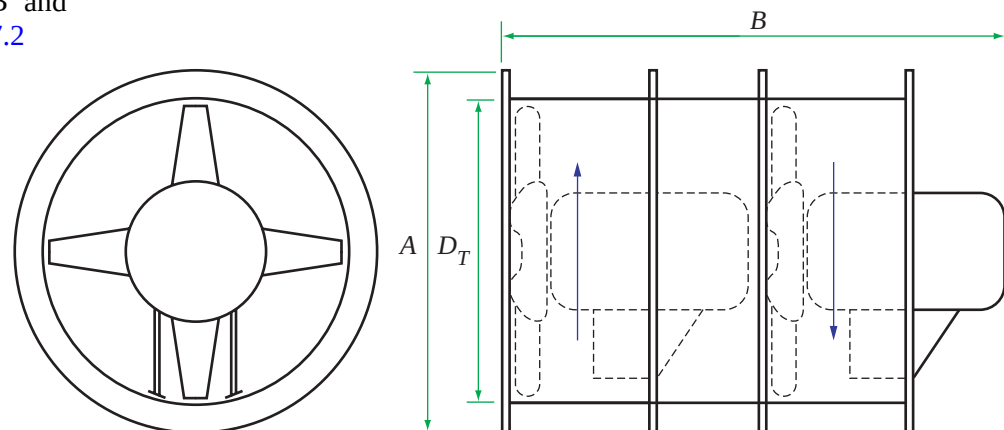
Fans of this type are frequently coupled directly to electric motors and thus have a restricted range of operating speeds but some versions are driven through vee belts by an electric motor mounted on the casing.



Sketch 4.7 Pressure coefficient, flow coefficient, efficiency, and impeller power characteristics of a typical tube-axial fan

4.1.3 Contrarotating fan

For approximate dimensions A , B and D_T , see Section 7.2



Sketch 4.8 Contrarotating fans

These fans are used for duties that require higher pressure rises than those obtainable from single axial-flow fans. The units are normally constructed from a pair of tube-axial fans so that the swirling motion imparted to the flow by the upstream fan is countered by the second fan. Thus the overall efficiency of the combination is higher than the efficiency of either component fan in isolation.

Although the second fan is often set with smaller blade angles than the first because of the swirling inlet flow, the maximum pressure coefficient obtainable from the combination can be more than twice the maximum pressure coefficient obtainable from either component fan in isolation.