WOODS PRACTICAL GUIDE

TO FAN ENGINEERING

Woods Practical Guide

to Fan Engineering

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Published by
WOODS OF COLCHESTER LIMITED
a GEC company

This Edition first published
December, 1978
Second impression July, 1979
Third impression June, 1985
Fourth impression April, 1988
Fifth impression 1990
Sixth impression 1992

Foreword

We depend increasingly on the movement of air and gases for our comfort and survival. This is not confined to ventilation and air conditioning but includes heat transfer, drying, blowing, particle transportation and filtering, the control of environments and the extraction of life threatening fumes and hot smoke.

The prime vehicle for moving air and gases is the fan. The relatively traditional appearance of fans conceals recent advances in aerodynamic, mechanical and acoustic technology. The demand for improvements in performance gathers pace with environmental awareness, safety legislation and the need to satisfy EC Directives. The knowledge needed to make these advances can be found within.

This book, published by Woods of Colchester limited, is the third edition of a work first issued in July 1952 and was originally aimed at those concerned with practical fan engineering problems. Distribution of the first edition which was translated in to French, exceeded 20,000 copies, being surpassed by that of the second edition, first published in June 1960, which reached about 27,000 copies. This included translations into French, German, Spanish and Italian. In the years since the second edition was issued the fan industry has continued to expand both in demand for its products and in the technological sophistication of customer applications and requirements. Moreover, in these intervening years the growth of sales in a world-wide sense has emphasised the need for a more systematic presentation of information relating to fan performance and the newer terminology associated with different national practices.

This revised edition of the guide, like its predecessors, is aimed at the practical as well as the professional engineer, and contains many worked examples. It provides a comprehensive compendium of information relating to fans, fan drives, fan selection and their environment of application. It commences with two chapters describing the ventilation requirements of the individual and of occupied spaces subject to various thermal environments caused by industrial, human and solar inputs. The next three chapters continue with a description of methods of heat rejection and heating, including discussion of such important current issues as pollution control and explosion hazards. The heart of the book treats extensively, in Chapters 6 and 7, the details of determination of system losses, fan performance specification and fan sizing. There the scope covers many climatic conditions whilst, for the more mathematically-minded, the basic theory of fan pressure generation is included.

An important feature of present-day society is the interest in efficient use of energy. Often fans are required to operate over their characteristic rather than merely at the flow corresponding to fan best efficiency. In such cases both the motor drive and the method of fan duty control are of primary importance. Chapters 8 and 9 provide clear illustrations of the merits of different electrical drives and fan duty controllers necessary to prevent unwanted energy dissipation. The increasing relevant subjects of noise and vibration are dealt with in Chapter 10 and include methods for calculation of fan sound power and assessing vibration acceptability. The problems of fan testing and conventions are discussed in the next chapter.

Heat transfer now represents one of the largest applications for fans and this topic is covered in Chapter 13, Heat Exchange and Drying. The growth in population, urbanisation and transport has expanded the demand for the ventilation of tunnels, mines and underground spaces and this is covered in Chapter 14. A range of tables and charts is provided whose value and usefulness are enhanced by bringing together data on many topics within a single accessible reference.

This foreword could not be considered complete without a sincere and grateful acknowledgement to the personal effort of the author, the late Mr. B.B. Daly, formerly Technical Director of Woods of Colchester Limited. This Guide represents the last major task of Mr. Daly for Woods and, as the reader will appreciate on each occasion he consults this book, Mr. Daly has brought together the practical experience of a lifetime of active involvement in the fan industry.

Those who read, comprehend and properly apply all the principles contained in this book will truly be well qualified in Fan Engineering. I hope that the benefit of this knowledge brings you success.

D. J. PRIEST
Managing Director
Woods of Colchester
Limited.

Author's Preface

This, the third, edition of Woods *Practical Guide to Fan Engineering* conforms entirely to the principles and objectives which governed the first edition in 1952. It is intended to help the user of fans-and there is hardly a field of industrial or environmental activity in which they are *not* used-and the application engineer responsible for their proper selection and installation.

The text has been completely rewritten, but this must not be taken to imply any disrespect for the work of the original editors, W. C. Osborne and C. G. Turner. The continuing demand for the book shows how well they struck and held the right note in their treatment of the subject. It is the development of the subject matter itself over the last 25 years which has necessitated extensive revision.

Increasing concern for comfort and safety and the demand for better working conditions call for environmental control in many new situations. Growing interest in energy conservation is served by increasing sophistication in process control. Empiricism has often been superseded by logical design and a more professional approach fostered by such bodies as the Institution of Heating and Ventilating Engineers (now the Chartered Institute of Building Services) and the Heating and Ventilating Research Association (now the Building Services Information and Research Association).

Great developments have taken place in the number and quantity of national standards, and more often than not these now have the advantage of international consensus. This helps the systematic presentation of data, and where standards are lacking tables of "typical" values have been freely used in this book. These give the engineer a quick, clear picture of the range of possibilities open to him-to be filled in by the detailed and up-to-date information which can only be furnished by the actual supplier of the material, equipment or service in question.

As in earlier editions mathematical analysis which would be beyond the competence of many readers has been avoided. This does not mean that figures, formulae and calculation methods are absent; these are the very essence of the engineer's daily work, and the primary object of the book is to present them in a way that is simple to use and understand

The book is unlikely to satisfy the professional specialist in one of the many fields covered, but every practising engineer knows how often he requires to understand the essentials of someone else's speciality. In a similar way it is not intended for the fan designer, but for those engineers -hundreds of times more numerous-whose combined contribution is just as essential to the success of each fan installation. It is not, however, a mere compendium of practical rules since an attempt has been made throughout to clarify principles for the benefit of the student.

The author's thanks are due to the directors of Woods of Colchester Limited for their encouragement and support, to J. P. Nelson for organising production of the book, and to Messrs. C. L. B. Noon and W. Woods Ballard for reading and checking the text. The treatment is not limited to the practice and environment of the British Isles but is backed by the experience of Woods of Colchester, a company exporting to over 120 countries, worldwide.

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Introductory notes on SI and other units

This, third, edition of Woods *Practical Guide to fan Engineering* has been written in the International Standard System of Units called "SI" (Systeme International).

The student will already be familiar with the new system. The purpose of these introductory notes is to help the engineer more familiar with the Imperial or technical metric systems.

Throughout the book the basic SI units have been used for all formulae and calculations. It is the great strength of this form of the SI system that all such formulae develop logically one from another without any arbitrary constants. A few Π's appear, but they are there because of the geometry of the application. The basic SI system has what is called "coherence": every unit is built up directly from the basic units of length, mass and time. By choosing the metre, kilogramme and second for these units, the joule and the watt are brought in for energy and power, and the established electrical system-volt, ampere, etc. follows.

The metric and general (not basic) SI systems of course use larger and smaller units as well. The SI system employs only multiples and submultiples of 10^3, 10^6, 10^9, etc. (except for 10 and 100 as in the decimetre and centimetre which we have avoided). It is of course natural to think and speak of distances across country in kilometres or the dimensions of components in millimetres. This practice has been followed where appropriate in the general text, but we strongly recommend that, before making any calculations or using any formulae, each quantity is converted into basic SI units. The result will be itself a basic SI unit which can then, and only then, be converted back into conventional form if desired.

This book will be no exception to the rule that all good rules have their exceptions. "Air changes per hour" appear in an early chapter for the ratio of the ventilation rate to the volume of the space ventilated. These form convenient numbers and are in universal currency the world over. No one talks seriously in terms of kiloseconds or megaseconds of time, although the millisecond, microsecond, and nanosecond (10^{-9} s) are well established in science and technology. Some remarks follow on the practice adopted. Basic SI units are in **heavy type.**

Length, Area and Volume. The **metre, the square metre** and the **cubic metre** are appropriate to all purposes of fan and ventilating technology. The kilometre is reserved for travel distances and the millimetre for component dimensions. The micrometre (μm) is the right size for limits and fits of machined components.

Velocity. The **metre per second** is a good unit for all air movement purposes. In talking of vehicle speeds, it is recognised that the kilometre per hour will prevail.

Volume Flow. All calculations are performed **in cubic metres per second** and this is the appropriate unit for general use in industrial fan technology. The litre per second ($10^{-3}m^3/s$) is likely to be more popular for small fan ratings but should be translated for calculations.

Rotational Speed. The revolution per second is the unit used in this book, although the more familiar revolution per minute is sometimes added, in brackets, in the text. Strictly the basic SI unit is the radian per second which, to a mathematician, is the natural unit of angular velocity. The engineer will feel that the complete revolution has the stronger grip on reality.

Mass, Mass Flow and Density. The **kilogramme,** the **kilogramme per second** and the **kilogramme per cubic metre** are suitable units for all purposes of fan technology. The gramme and milligramme are more convenient for chemical analysis, and the tonne (= 1000 kg) for bulk quantities of massive materials, but the **kilogramme** is the basic unit.

Force. It is most important that the **newton** should become established in our practice, and indeed our thinking, as the unit of force. The greatest benefit to be derived from the SI system is the elimination of the confusion caused by the gravitational units, even when these are properly named the kilogramme force (= 9.81 N) and the poundforce (= 4.45 N). While the cgs dyne (= 10^{-5} N) is as logical as the newton, it is much too small for practical use. Earlier attempts to overcome the confusion between mass (e.g. the slug) and force or weight (e.g. the poundal) can pass into deserved oblivion.

Pressure and Stress. Throughout this book the **pascal** (the name now internationally adopted for the **newton per square metre**) is used as the unit of pressure. 1 Pa (equal to about 0.1mm or 0.004in of water) is as small a unit as is ever needed for ordinary experimental work with air, while the fact that system pressures often run to a few thousand Pa does not necessitate the introduction of the kilopascal, a unit which would always need decimal places in our field.

Barometric pressures at around 10^5 Pa are admittedly inconveniently large, and deference is paid to meteorological practice by putting the millibar equivalent in brackets in the text (1 mb = 100 Pa). Nevertheless, important formulae contain the ratio of fan pressure to barometric pressure and it is important to maintain consistency. Modern barometers are calibrated in millibars, and the old mm or in of mercury should gradually disappear.

Stress in a material is the same quantity as pressure, and the megapascal (MPa) is a suitable unit. Since most stressed components will be dimensioned in millimetres rather than metres, it is convenient, though heretical, to think of unit stress as one "newton per square millimetre", which equals 1 MPa.

Energy, Power and Heat. The **joule** (= 1 newton-metre) and the **watt** (= 1 joule per second) are the basic SI units of energy and power respectively. It is an important advantage that the units are the same whether the energy is mechanical, electrical or in the form of a quantity of heat. Similarly, the rate of flow of heat is an energy flow, or power, and is measured in watts. The old mechanical equivalent of heat (4.18 joules per calorie, or 3412 BTU's per kilowatt-hour) is gone - banished by the definition of absolute temperature in thermodynamic terms.

The horse-power, cheval-vapeur, or pferdestarke are on the way out, too. Electric motors are now rated in kW, though here there is a pitfall to be avoided. Always recognise and state whether it is (mechanical) power output in kW or (electrical) power input in kW that is meant.

The convenient relationship that one kilocalorie raised the temperature of one kilogramme of water by about 1 °C is lost, but for the fan and ventilating engineer there is a bonus. The specific heat of air at constant pressure can be taken as **1000 joules per kilogramme per degree C** for practical purposes under atmospheric conditions.

The **joule** and **watt** will be used consistently in all formulae in this book. Of course a result in watts can be converted into kilowatts at the end, but it remains important to retain the basic SI calculation principle.

Expressions for heat conduction, conductance and conductivity will be found slightly less cumbersome with the use of **watts per square metre** than they were in the older units.

Viscosity. The basic SI unit of dynamic viscosity is the **pascal-second** and of kinematic viscosity the **square metre per second.** These are admittedly inconveniently large for gases, and tabulations commonly remain in centipoise (1 cP = 10^{-3} Pa/s) and centistokes (1 cSt = $10^{-6}m^2/s$). Of course, the basic SI units (to which no names have yet been allotted) must be used in the formulae for Reynolds and other dimensionless numbers.

SYMBOLS WITH SI BASIC UNITS

A a	Area of cross-section	m^2
B b	Length of rectangular section	m
C c	Width of rectangular section	m
c_p c_v	Specific heat at constant pressure; volume	J/kg °C
C_d	Drag coefficient	non-dim.
D d	Diameter	m
D_h	Hydraulic diameter = 4 (area)/(periphery)	m
E	Heat exchanger effectiveness	non-dim.
E	Modulus of elasticity	N/m^2
e	Eccentricity of mass centre (balancing)	m
F	Force	N
f	Coefficient of duct friction: $\Delta p = f\,(L/D)\,\frac{1}{2}\rho\bar{v}^2$	non-dim.
G	Heat transfer capacity of a fluid flow	W/°C
g	Acceleration of gravity	m/s^2
g_p	Pressure gradient	Pa/m
H	Heat flow	W
H h	Height; height of fluid column (head)	m
h_c h_r	Heat transfer coefficient by convection; ditto radiation	W/m^2 °C
I	Moment of inertia	kg m^2
I	Current	amps
I_d I_f I_h I_v	Radiant heat intensity—direct; diffuse; horizontal; vertical	W/m^2
J	Quantity of heat	J
K	Pressure loss factor: $\Delta p = K\,.\,\frac{1}{2}\rho\bar{v}^2$	non-dim.
L l	Length	m
L	Latent heat	J/kg
L_p L_q	Sound power correction for pressure; ditto for volume flow	dBW
M m	Mass	kg
M	Moment of a force; torque	Nm
m	Moisture content, kg moisture per kg dry air	non-dim.
n	Number. Rotational speed	rev/s
n_s n_c	Synchronous speed; critical speed	rev/s
P_t P_s P_v	Fan total pressure; fan static pressure; fan velocity pressure	Pa
p_a p_o	Absolute pressure at a point; absolute barometric pressure	Pa
p_s p_t p_v	Pressure at a point or section—static; total; velocity	Pa
p_f	Pressure drop caused by duct friction	Pa
Q q	Volume flow rate, m^3/s	m^3/s
q_m	Mass flow rate, kg/s	kg/s
R r	Radius, m	m
r_g	Radius of gyration	m
S	Stress	N/m^2
s t	Spacing; thickness	m
T	Thrust force	N
T	Absolute temperature	°K
t t_o	Temperature; ambient temperature	°C

t_a	t_w	Dry-bulb (air) temperature; wet-bulb temperature	°C
t_g	t_r	Globe temperature; mean radiant temperature	°C
U		Thermal transmittance	W/m² °C
U	u	Peripheral or other reference velocity	m/s
V	V_s	Velocity; velocity of sound	m/s
V		Electric pressure	volts
v	$\bar{v}$	Velocity at a point; average velocity at a section	m/s
W		Power	W
W_{imp}	W_{air}	Fan impeller power input; ditto output	W
W_{in}	W_{out}	Electric motor power input; ditto output	W
w		Air velocity relative to a moving part	m/s
X		Length of air jet	m
y		Deflection of a shaft	m
α	alpha	Flow angle, absolute	degrees
β	beta	Flow angle, relative	degrees
γ	gamma	Ratio of specific heats, c_p/c_v	non-dim.
ϵ	epsilon	Surface roughness	m
ϵ		Emissivity	non-dim.
η	eta	Efficiency	non-dim.
θ	theta	Included angle of expander; blade pitch angle	degrees
λ	lambda	Leakage coefficient	non-dim.
μ	mu	Dynamic viscosity	Ns/m²
ρ	rho	Density	kg/m³
ρ_s		Specific density, air = 1	non-dim.
ϕ	phi	Angle; electric phase angle	degrees
ω	omega	Angular velocity	radians/s

ABBREVIATIONS—ALSO USED AS SYMBOLS OR UNITS

DB	Dry-bulb		°C
dB	Decibel of sound pressure	Relative to 2×10^{-5}	Pa
dBW	Decibel of sound power	Relative to 10^{-12}	W
DP	Dew point		°C
ET	Effective temperature		°C
FSP	Fan static pressure		Pa
FTP	Fan total pressure		Pa
FVP	Fan velocity pressure		Pa
Ma	Mach number = V/V_s		non-dim.
ppm	Parts per million		non-dim.
Re	Reynolds number = $DV\rho/\mu$		non-dim.
RH	Relative humidity		non-dim.
SF	Safety factor		non-dim.
SPL	Sound pressure level.	dB relative to 2×10^{-5}	Pa
SW	Sound power		W
SWL	Sound power level	dBW relative to 10^{-12}	W
SWR	Sound power ratio = SW/W_{air}		non-dim.
VP	Vapour pressure		Pa
WB	Wet-bulb		°C
Δt	Temperature difference		°C
Δt_{lm}	Logarithmic mean temperature difference		°C

APPROXIMATE CONVERSION FACTORS
SI AND BRITISH UNITS

1m = 39·37in 1m = 3·28ft 1m = 1·094yd 1 km = 0·621 mile	1in = 25·4mm 1ft = 305mm 1yd = 0·914m 1 mile = 1·61 km
$1m^2$ = $1550in^2$ $1m^2$ = $10{\cdot}76ft^2$ 1 ha = 2·47 acre	$1in^2$ = $645mm^2$ $1ft^2$ = $0{\cdot}0929m^2$ 1 acre = 0·405 ha
1 litre = $61{\cdot}0in^3$ 1 litre = 1·76 pint = 0·22 gallon $1m^3$ = $35{\cdot}3ft^3$	$1in^3$ = $16{\cdot}4cm^3$ $1ft^3$ = $0{\cdot}0283m^3$ 1 gallon = 4·546 litre
1 gram = 15·43 grain 1 kg = 2·205 lb 1 tonne = 0·984 ton	1 grain = 64·8 mg 1 lb = 0·4536 kg 1 ton = 1·016 tonne
1 kg/m^3 = 0·0624 lb/ft^3 1 gm/cm^3 = 0·0361 lb/in^3	1 lb/ft^3 = 16·02 kg/m^3 1 lb/in^3 = 27·68 gm/cm^3
1m/s = 3·28ft/s 1m/s = 197ft/min 1m/s = 3·6 km/hr = 2·237 mile/hr	1ft/s = 0·305m/s 1ft/min = 0·00508m/s 1 mile/hr = 0·447m/s = 1·609 km/hr
$1m/s^2$ = $3{\cdot}28ft/s^2$	$32{\cdot}2ft/s^2$ = $9{\cdot}81m/s^2$
$1m^3/s$ = $2120ft^3/min$ (cfm) $1m^3/hr$ = $0{\cdot}588ft^3/min$	$1ft^3/min$ = 0·472 litre/s $1ft^3/min$ = $1{\cdot}70m^3/hr$
1 N = 0·225 lbf (pound force) 1 MN = 100·4 ton force	1 lbf = 4·45 N 1 ton f = 9·96 kN
1 Nm = 0·738 lbf ft	1 lbf ft = 1·356 Nm
1 k Pa = 4·01in of water 10^5 Pa (1000 mb) = 29·53in of mercury 1 Pa = 0·0209 lbf/ft^2	1in of water = 249 Pa 1in of mercury = 3386 Pa 1 lbf/ft^2 = 47·9 Pa
1 N/mm^2 (MPa) = 145 lbf/in^2 1 N/mm^2 = 0·0647 $tonf/in^2$	1 lbf/in^2 = 0·0069 N/mm^2 (MPa) 1 $tonf/in^2$ = 15·44 N/mm^2
1 Pa s (10^3cP) = 0·0209 lbf s/ft^2 $1m^2/s$ (10^6cS) = $10{\cdot}76ft^2/s$	1 lbf s/ft^2 = 47·9 Pa s $1ft^2/s$ = $0{\cdot}0929m^2/s$
See Table 14.15 for Redwood and other viscosity scales.	
1 J = 0·737ft lbf	1ft lbf = 1·356 J
1 kW = 1·34 hp	1 hp = 746 W
t °C = (1·8 t + 32) °F (Fahrenheit) T °K (t °C + 273) = [1·8 T] °R (Rankine)	t °F = [$^5/_9$ (t − 32)] °C (Celsius) T °R (t °F + 460) = [$^5/_9$ T] °K (Kelvin)
1 MJ = 0·278 kWh 1 kJ = 0·948 Btu	1 kWh = 3·6 MJ 1 Btu = 1055 J
1 W = 3·412 Btu/hr	1 Btu/hr = 0·293 W

1 kJ/kg = 0·430 Btu/lb	1 Btu/lb = 2·326 kJ/kg
1 kJ/kg °C = 0·239 Btu/lb °F	1 Btu/lb °F = 4·19 kJ/kg °C
1 W/m² = 0·317 Btu/ft² hr	1 Btu/ft² hr = 3·155 W/m²
1 W/m² °C = 0·176 Btu/ft² hr °F	1 Btu/ft² hr °F = 5·68 W/m² °C
1 W/m °C = 0·578 Btu/ft hr °F	1 Btu/ft hr °F = 1·73 W/m °C

SI, TECHNICAL METRIC AND cgs UNITS

1 ha (hectare) = 10,000m² = 0·01 km²
1 cc (cubic centimetre) = 1 ml (millilitre) = 10^{-6}m³
1 radian/second = 0·159 rev/s = 9·55 rev/min
1 km/hour = 0·278m/s = 0·540 kn (international knot)
1 q (quintal) = 100 kg = 0·1 tonne
1 N (Newton) = 10^5 dyne = 10^{-3} sthene
1 kgf (kilogram-force) = 9·81 N. 1 N = 0·102 kgf
1mm of water = 1 kgf/m² = 9·81 Pa 1 Pa = 0·102mm H_2O
1mm of mercury (torr) = 133·3 Pa 1 k Pa = 7·50mm Hg
1 atmosphere = 1013 mb (millibar) = 101·3 k Pa = 760mm Hg
1 kgf/mm² = 9·81 N/mm² (MPa) = 98·1 b (bar).
1 Pa (pascal = N/m²) = 10 dyne/cm² = 0·01 mb
1 J (joule = Nm) = 10^7 erg = 0·102m kgf 1m kgf = 9·81 J
1 kcal (kilocalorie) = 4·19 kJ 1 kJ = 0·239 kcal
1 ch (cheval vapeur) = 1 Pf (Pferdestärke) = 75m kgf/s = 735 W
1 kcal/hour = 1·163 W 1 W = 0·860 kcal/hr
1 kcal/s = 4·19 kW 1 kW = 0·239 kcal/s
1 cP (centipoise) = 10^{-3} Pa s = 10^{-3} Ns/m²
1 cS (centistokes) = 10^{-6} m²/s = 1mm²/s

Note: All these conversion factors are approximate with errors not exceeding in 1000. For precise conversions see Table 14.2.

CHAPTER 1

The air and human well-being

Man's sense of comfort and his capacity for work deteriorate quickly in poor air conditions. More seriously, his general health may be impaired in the long term by living and working in ill-ventilated buildings or an enervating climate.

The relevant factors are the temperature and relative humidity of the air, its circulation and movement over the body, and its purity. It is the function of the Heating and Ventilating Engineer to keep these within satisfactory limits.

Ventilation of buildings will give complete control over air movement and purity. Together with heating it will deal adequately with temperature and humidity and will produce tolerable conditions even in difficult cases. Ventilation means the supply of fresh outside air to a building, its circulation within the building, and its discharge together with the various polluting substances it has picked up.

Air Conditioning adds the functions of cooling and positive control of humidity. With the complete system a good measure of comfort can be assured however severe the climate and weather. Continuous control is a vital factor and since tastes differ, the ideal system will include some individual control over temperature and air movement.

1.1 The Air

1.1.1 Composition of the air

The natural atmosphere up to 11,000m altitude (at which point the troposphere in which we live ends and the stratosphere begins) consists of gases well mixed in the following proportions:

Table 1.1

Gas		Per cent of dry air by volume*	by weight
Nitrogen	N_2	78·09%	75·52%
Oxygen	O_2	20·95%	23·15%
Argon	A	0·93%	1·28%
Carbon dioxide	CO_2	0·03%	0·05%
Total dry air		100·00%	100·00%
Water vapour	H_2O	0 to 5%	0 to 3%
Total moist air		100 to 105%	100 to 103%

* And by number of molecules.

The reference of percentages to the dry air part of the total mixture is a useful convention in view of the highly variable nature of the water content. It also has advantages in calculation which will appear in later chapters. The air contains small proportions of many other substances, most of which can be classed as impurities.

1.1.2 Oxygen is of course the vital component for life support, and indeed man can function effectively when breathing pure oxygen alone. The average person at rest passes some 500 litres of air per hour through his respiratory system. This air carries 140 grams of oxygen, of which some 15% is absorbed into the bloodstream, a consumption rate of 21 gm per hour.

Oxygen deficiency, with its symptoms of weakness and lethargy, may arise in two ways. Without any change in the proportion of oxygen, the density may be reduced, as at high altitude, to the point where the respiratory system cannot handle the additional volume required. More rarely, the proportion of oxygen may be excessively reduced by consumption, dilution or displacement. Consumption or dilution occur in enclosed spaces such as submarines or sewers. The danger of displacement may arise from heavy vapours left at the bottom of empty tanks or vats. 25% reduction in available oxygen can be serious for those with respiratory or cardiac weakness. 50% is a severe test for the fittest, without acclimatisation.

1.1.3. Nitrogen. together with argon, and traces of the other inert gases, neon, helium, krypton and xenon, has no physiological effects under normal conditions. At the high pressures encountered by deep-sea divers an increased proportion of nitrogen is absorbed into the blood. Too rapid a return to normal pressure causes the painful and dangerous symptoms known as "the bends", in which the excess nitrogen reappears as bubbles in the veins and arteries.

1.1.4 Carbon dioxide also has no significant effect on man. Because we produce it, together with water vapour, in the air we exhale, the concentration rises in crowded rooms or buildings. For this reason it was at one time used as an index of the effectiveness of ventilation with limits of a half to one per cent, but this practice has died out.

1.1.5 Impurities such as methane or ozone occurring in natural fresh air at levels of less than 1 ppm (parts per million) may be ignored. Industrial processes, however, and the circumstances of town life, produce a wide variety of polluting substances which must be dealt with by ventilation.

For toxic gases and their dilution see Section 5.1.

For the extraction of dust and fumes at source see Section 5.4. For the cleansing of air from smoke and dirt see Section 5.3.

For precautions against fire and explosion hazards see Section 5.2. For fans to resist corrosion or abrasion see Section 7.14.

1.1.6 Odours may result from the presence of extremely small proportions of impurities in the air. The staleness perceptible in the air of crowded rooms is due chiefly to organic substances given off from the occupants' bodies. While these are not dangerous, and some people may even like "a good fug'", others are more sensitive. Loss of appetite is a common reaction and some may even experience nausea. The ventilation requirements to deal with body odour are dealt with in Section 2.1.2. They are not great and are usually exceeded substantially by other requirements of the system.

1.1.7 Bacteria, often associated with dust particles or water droplets, may remain in suspension in the air far long periods. In hospitals a positive programme of control is necessary. In ordinary life the spread of infection by coughing and sneezing is a direct function of the number of people congregated together in a limited space. There is positive evidence that good ventilation reduces this hazard.

1.2 Body Heat Balance

Why does anyone feel limp in hot surroundings? Why is a breeze refreshing 7 These sensations relate very much to the efforts the body has to make to maintain its temperature. The less the effort the more comfortable we feel. Man is a warm blooded animal and must maintain the temperature of his vital organs within a few degrees of 37°C throughout life.

The human body, fuelled by the food we eat, continuously produces heat, associated with chemical change and muscular activity. This metabolic rate of heat generation can exceed 1 kW with maximum exertion. Table 1.2 gives an idea of the range; individuals will vary with age, weight and other personal characteristics.

Table 1.2

Activity	Metabolic rate
Sleep and complete rest	60–100 W
Seated, without manual work	100–140 W
Light manual work	140–200 W
Moderate work; walking	200–400 W
Heavy labour	400–800 W

The body has notmuch heat storage capacity because we cannot allow the greater part of our substance to get significantly hotter or cooler without distress. Apart from minor temporary deviations we must dissipate heat just as fast as it is generated - i.e. at the metabolic rate. There are three processes by which the body loses heat: convection; radiation; evaporation.

1.3 Convection and Radiation

A layer of cool air in contact with warm skin or clothing will pick up heat, and as its temperature rises its density will fall. The lighter air now rises up away from the body, taking heat with it, and is replaced by fresh. cool air which continues the process. This is natural convection.

Even the slightest air movement around the body will increase the rate at which warm air is replaced by cool, and thus increase the heat loss. The laws governing this heat exchange by forced convection are dealt with in Chapter 12. If the air temperature is on the high side the extra movement will be felt as a pleasant breeze, and can be increased with advantage by the use of ceiling fans or circulating fans. If, on the other hand, the temperature is normal or low, movement will be felt as a draught and should be kept below the limit of perception-about 0.25m/s-for maximum comfort.

Like all matter, the body transmits heat by radiation and receives heat by the same path. If all the surfaces surrounding us were at the same temperature as our skin there would be no nett gain or loss of heat by radiation. In practice, of course, the walls and most of the surfaces are cooler and radiation carries part of the necessary heat loss. Radiant heaters have high temperatures over small areas, contributing a nett heat gain to the body. As radiant heat, like light, travels in straight lines, only the "illuminated" side of the body will be heated, but, in the absence of excessive draught, the blood stream will distribute the heat satisfactorily over the body.

Within the comfort zone of external air conditions, convection and radiation losses account for about 75% of the metabolic heat at rest, or with mild activity. When the temperature rises above or falls below the comfort zone an automatic reaction known as vasomotor regulation comes into play, with the object of preserving both internal temperature and heat loss unchanged. The tissue layers under the skin contain a network of veins which can be enlarged or contracted under the control of the nervous system. In a warm environment a copious flow of blood is allowed through these vessels, bringing the skin temperature up towards a maximum of about 35 °C, so as to maintain its temperature excess over the surroundings. In a cold environment the veins are constricted, reducing the thermal conductivity of the tissue layer so that the skin cools. The blood flow along legs and arms is also modified, allowing further cooling along their length to keep down the heat loss. Ultimately, hands and feet may be only a few degrees above the air temperature even to the extent of allowing frost-bite of fingers and toes rather than loss of temperature at a vital centre.

1.4 Evaporation

At temperatures above about *30°C* the system just described is unable to secure the necessary heat loss, even at rest. Indeed, above *35°C* convection and radiation cease to be losses and become heat gains making the body's task even harder. Reaction to this situation is known as *evaporative regulation.* It consists in the automatic activation of the sweat glands over an area of skin proportional to the corrective effort required.

The transformation of sweat into water vapour absorbs energy, just as the boiling of water does. This energy is taken from the wetted skin surface in a form known as the latent heat of evaporation (see Chapter 12). The cooling effect is powerful : the worker in a hot industrial environment may easily produce and evaporate one kilogram of sweat each hour, which will remove heat at the rate of 680 watts. The water and salt lost by the body have to be replaced, and it is natural that such workers should be copious drinkers.

Apart from the sweat mechanism, evaporation of the water vapour in the exhaled breath carries away heat at a minimum rate of around 20 watts: the proportions at rest are illustrated in Fig. 1.1. Note that the metabolic rate cannot be reduced, though we may increase it if we are cold by voluntary activity-walking briskly, stamping the feet, swinging the arms, etc. Shivering is an automatic reaction with the same purpose.

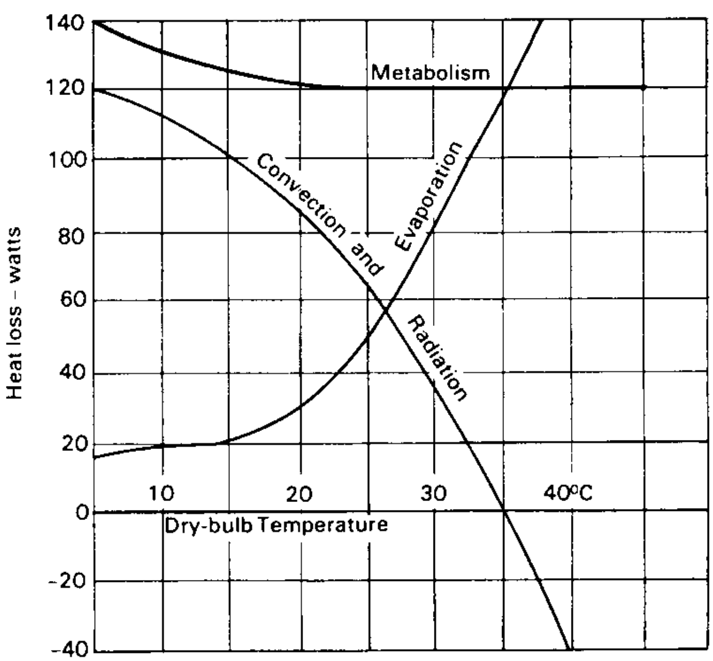


Fig. 1.1 Heat balance for average man, at rest.

In a still atmosphere the air next to the skin and trapped in the clothing becomes almost saturated, and its capacity to absorb and carry away moisture is severely limited. The sweat produced stays wet on the skin and the body's effort to give off heat is retarded. The surplus sweat which drips off or is wiped away is virtually useless for heat removal.

Currents of air flowing round the body correct this situation. Saturated air is replaced by fresh and evaporation is helped in exactly the same manner as heat removal is helped by forced convection. To understand the limitations of evaporative regulation, and the value of ventilation and air motion, we must study the subject of air humidity.

1.5 Humidity

The moisture in humid air takes the form of an invisible vapour, mixed with the other gaseous constituents. If mist or steam can be seen it means that more moisture is present than the air can hold; the excess has condensed in the form of liquid water droplets.

Air that carries the maximum possible amount of water in the vapour form is said to be saturated. The temperature at which air becomes saturated rises with its moisture content and is known as the dew point. Thus air saturated at 14°C carries about 1% of water vapour by weight. But at 25 °C it could contain 2%, so that 1 % moisture content is now only half the saturation value. It will be sufficient, for our present purpose, to say that its *relative humidity* is 50%, saturation value at the same temperature being taken as 100°.

Strictly speaking the relative humidity is the ratio of the number of molecules of water vapour present to the number in the same volume of saturated air at the same temperature and pressure. The comparison is based on equal volumes of moist air rather than equal weights of dry air, and the values are slightly higher except at 0% and 100%. Relative humidity is also equal to the ratio of the vapour pressure of the water present to the vapour pressure at saturation.

1.5 Measurement of Humidity

The true temperature of air, dry or humid, is that measured with an ordinary mercury-in-glass thermometer and is called the "dry-bulb temperature". Precautions should be taken to shield the bulb from strong radiant sources. If the bulb of a second thermometer is kept wetted, its temperature will fall by evaporative cooling. Provided the air is kept moving past it at a velocity usually specified as 2m/s minimum, it will reach, and record, a stable temperature known as the wet-bulb temperature. This is intermediate between the dry-bulb temperature and the dewpoint, and is arrived at by a process of heat balance described in Chapter 12.

The meterological type of wet- and dry-bulb instrument is designed to make use of the air currents practically always present out-of-doors, and is not suitable for use in rooms. The sling psychrometer is simple and reliable for ventilating work. As illustrated in Fig. 1.2 the two thermometers are mounted side-by-side in a frame which can be whirled round the handle, providing 3 to 4m/s air movement over the bulbs. The cloth covering the wet bulb should be clean and freshly soaked in distilled water. The other bulb must be quite clean and dry.

Air conditions for the human environment are generally described in terms of the dry-bulb temperature (°C DB) and the relative humidity (RH). There is no simple formula relating RH to °C DB and °C WB (wet-bulb temperature). Tabulated values may be used (see Table 14.8) but more information can be obtained from a Psychrometric Chart. Fig. 1.3. is a simple example suitable for the present discussion, giving, in addition

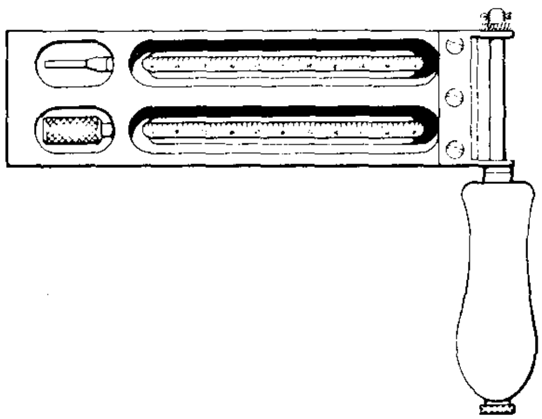


Fig. 1.2 Sling psychrometer.

to the RH, the dew-point and the moisture content. Many more complicated psychrometric charts have been drawn, with refinements of accuracy and giving additional information for the engineer concerned with air conditioning or process work. See for example Figs. 4.4 and 12.13.

1.7 Comfort Conditions

During the heating season good heating and ventilating systems will be provided with thermostatic control, both for economy and for comfort Small rooms and offices should be individually controlled to meet variations in preferred temperature, which may range from 18°C to 28°C. Spaces accommodating many people must of course have a compromise setting which is likely to leave at least 10% of the occupants slightly dissatisfied-some too warm, some too cool.

1.7.1 Air movement and humidity

Well distributed air movement in the range 0.1 to 0.25m/s is important to create a sense of freshness in the air. As well as eliminating hot spots and cold spots, it limits the temperature gradient from foot level to head

Each point on the psychrometric chart, Fig. 1.3, represents a particular air condition. Given any two of the first four quantities, the remaining three can be found. For example at the point 0:

Arrow a points to the dry-bulb temperature	41 °C
Arrow b points to the wet-bulb temperature	23 °C
Arrow c points to the relative humidity	20%
Arrow d points to the moisture content	0·010
Arrow e points to the dew point	14 °C

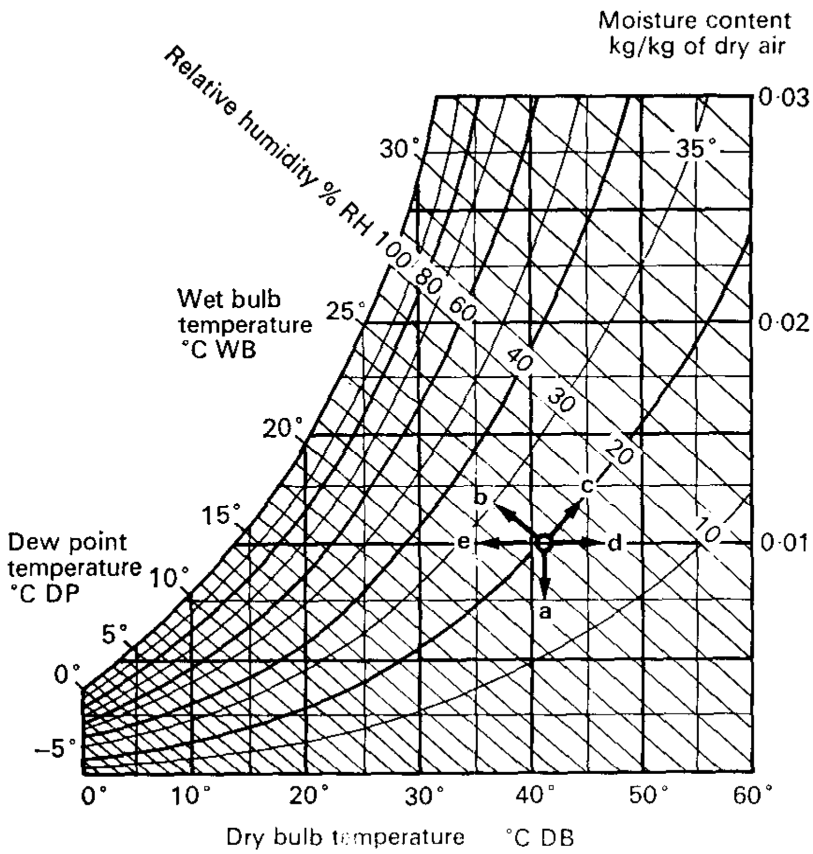


Fig. 1.3 Psychrometric chart.

level; this should not exceed 3°C. Draughts must be avoided, and research has indicated the range which is free from unpleasant sensation as a function of temperature. The suggested velocity limitation is:

Dry-bulb air temperature	20°	22°	24°	26°C
Maximum m/s on back of neck	·10	·17	·27	·42
Maximum m/s at general head level	·17	·25	·35	·50

Humidity is unimportant in the comfort zone provided it is in the range 30% to 70% RH. A rise to *90%* would call for a reduction of about 1 °C in the dry-bulb temperature to maintain comfort at rest-more if active. Humidities of 10 to 20% RH give rise to uncomfortable dryness in the nose and throat, and are likely to cause cracking in furniture and other timber seasoned at higher humidities, and electrostatic sparking.

1.7.2 Optimum temperatures

There is no international consensus on the optimum comfort temperature. Longer experience of central heating, coupled with a tendency to wear light warm-weather clothing all the year round, probably accounts for the higher temperatures preferred in North America. The following is an interpretation of national practice:

	Optimum	Range
Great Britain (IHVE)	21°C	19° to 23°C
United States (ASHRAE)	25°C	23° to 27°C
Germany (DIN)	22°C	20° to 25°C

The conditions applying to the above selection are:

0·1m/s air velocity.	Add 2°C for 0·25m/s mean.
30% to 70% RH.	Deduct 1°C for 90% RH.
Winter season.	In summer 1°C higher may be preferred.
Normal indoor clothing.	For bathing or similar add 3°C to 5°C.
Seated at rest.	Deduct up to 5°C for active occupation.
Long term occupation.	Deduct 3° to 5°C for transit areas.

Mean radiant temperature same as air temperature.

The last condition will not be met in poorly insulated buildings with convective heating. In such cases cold walls will have to be offset by higher air temperatures. Conversely, when the heating is mainly radiant a lower air temperature will give maximum comfort. This problem is dealt with in Section 3.3.

The optimum temperatures quoted will not be detrimental to any activity provided some clothing is discarded when working hard. However, with moderate and heavy work loads lower temperatures can be tolerated; for example the British Factory Acts allow 15 5°C minimum where factory work is done seated. Outdoor work can, of course, be done in freezing, or indeed Arctic, conditions given the necessary clothing, fitness, activity and acclimatisation, but rest periods in warm surroundings should be available.

1.8 Heat Stress

The scale introduced by Houghton and Yaglou in 1923 is still regarded as the best guide to the severity of exposure to high temperature and humidity, and the cooling effect of air movement. In these respects it is a measure of the effort required by the evaporative regulation system of the body. The scale is based on jury judgements of equality of warmth when passing from enclosures at various humidity levels to a standard enclosure at 100% RH. The temperature of the latter is termed the effective temperature. The diagram in Fig. 1.4 is a conversion to metric units of the upper part of the definitive chart published by ASHRAE. The chart applicable to men stripped to the waist and doing light physical work has been chosen. Instructions are given with the chart for determining the effective temperature in °C over a range of dry-bulb and wet-bulb temperatures and of air velocities.

The following table of maximum recommended exposures for industrial workers is derived from the publications of the American Conference of Governmental Industrial Hygenists. It is in terms of effective temperature.

Exposure	**Light Activity**	**Heavy Work**
Daily work, cold weather	24°C ET	24°C ET
Daily work, warm weather	30°C ET	27°C ET
3 hours work*	34°C ET	29°C ET
1 hour work*	38°C ET	32°C ET
½ hour work*	42°C ET	35°C ET

* With rest period for full recovery.

The Tower limit for cold weather is intended to guard against chill when leaving work which has caused heavy sweating. It will be seen from Fig. 1.4 that a high air velocity can lower the effective temperature by 5 °C or more, particularly if the air is humid. At the same time the velocity should be limited if the exposure is long term. Some recommended velocities from the source just referred to for use in high effective temperatures are:

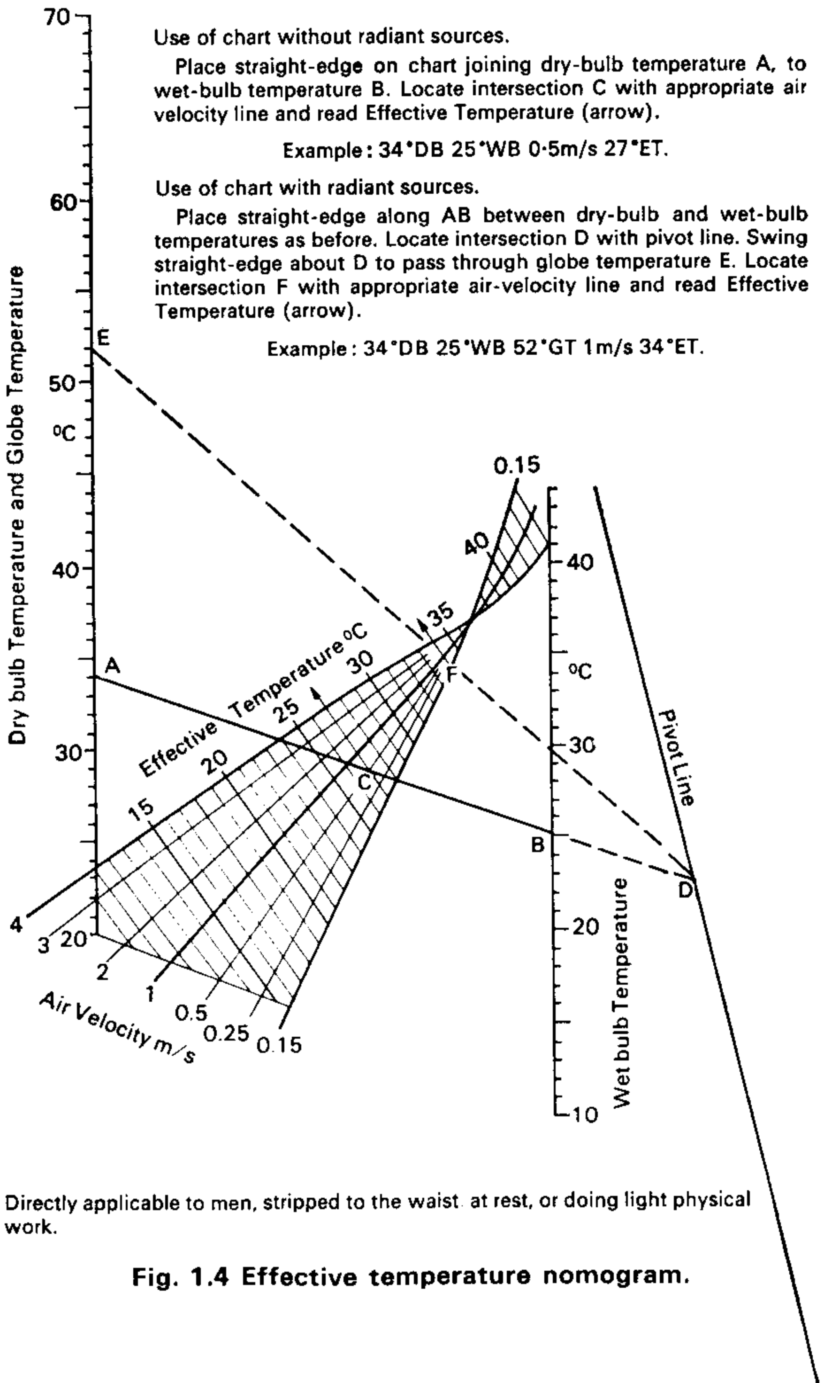
Daily work, seated	0·4– 0·6m/s
Daily work, standing	0·5– 1·0m/s
Intermittent exposure, light stress	5 –10 m/s
Intermittent exposure, moderate stress	10 –15 m/s
Intermittent exposure, heavy stress	15 –20 m/s

The stress refers to the combination of heat and work loading. Intermittent exposure may be at the site of exceptional heat stress, or in relief areas set aside for recovery after severe exposure.

1.9 Radiant Heat

1.9.1

An effective temperature determined from readings of wet- and dry-bulb thermometers and air velocity alone is appropriate only to environments where radiant heat is not a significant factor. In some industries,



Directly applicable to men, stripped to the waist at rest, or doing light physical work.

Fig. 1.4 Effective temperature nomogram.

e.g. in steel works, it is precisely the radiant sources which cause the heat stress.

The prediction and calculation of the heat loads from radiant sources is often a complex and somewhat unrewarding exercise. The ventilation engineer is more likely to be called in when certain operations in existing installations are the subject of complaint. Effective action may include shielding of heat sources, enclosure and ventilation of sensitive spots such as control desks and relief areas, or even (for very short exposure to intense heat) the use of protective clothing with an aluminised surface to minimise radiation absorption.

1.9.2 Globe temperature

If, however, improvement by general or local ventilation is contemplated, measurement becomes important. The usual instrument for assessing radiant heat is the globe thermometer. This is a copper sphere, 15cm in diameter and painted matt black inside and out, with a thermometer to measure the internal temperature. This, after 20 minutes or so to reach equilibrium, is the globe temperature (°C GT) and is such that the radiant heat gain equals the convection heat loss, including the effect of air velocity but disregarding evaporative effects.*

A number of investigators have proposed ways in which the effective temperature scale may be modified to take account of radiant heat using the globe temperature. These do not agree very closely so it has been felt permissible to introduce a very simple modification to the Effective Temperature Nomogram, Fig. 1.4. This produces results within 1 °C or so of the method suggested by Bedford, within the limited chart area drawn.

We can now assess the severity of heat stress by comparing the measured effective temperature with the recommendations in Section 1.8. Furthermore, we can estimate the effectiveness of various factors in bringing the ET down, which will form the subject matter of Chapter 2.

1.9.3 Example 1

Suppose in a particular workshop the measured dry-bulb temperature is 34°C and the measured wet-bulb temperature 25°C. Fig. 1.3 shows that the relative humidity is 45% and the moisture content 0.016. On Fig. 1.4 the location of the straight edge is already drawn as line AB, and shows that the effective temperature is 28°C with natural air movement only (0.15m/s).

This is on the high side for heavy work and could be reduced to 26 °C with general air movement at about 1m/s or to 24°C with a positive cooling stream at 4m/s. Alternatively, if the moisture content could be reduced to 1.0%, say by local extraction of escaping steam, with an accompanying reduction to 32 °C dry-bulb, Fig. 1.3 shows that we have

*The globe temperature t_g and the air temperature t_a may be used to estimate the ***mean radiant temperature,*** t_r provided the velocity, v m/s, in the neighbourhood of the globe is also measured. In Europe the Missenard globe of 10cm diameter may be used, t_g being then called the "*dry resultant temperature*".

$$t_r = t_a + (t_g - t_a)(1 + 2{\cdot}35\sqrt{v}) \quad \text{150mm globe thermometer}$$

$$t_r = t_a + (t_g - t_a)(1 + 3{\cdot}16\sqrt{v}) \quad \text{100mm Missenard globe}$$

reduced the RH to 33% and the wet-bulb to just over 20'C. This gives an effective temperature of 24°C without considering the enhanced air circulation.

Example 2

Suppose the same air conditions exist in a foundry, where the workers cannot readily be shielded from radiant heat. A globe temperature of 52°C is measured with natural air movement of about 1m/s caused by convection currents only. Fig. 1.4 shows an effective temperature of 34 °C in which only short periods of heavy work are possible.

At first sight it appears that air velocity has negligible effect. However, it must be remembered that the globe thermometer should be exposed to the same conditions as the worker. An experiment is tried with the globe in the work place exposed to a spot-cooling air stream at 4m/s and the globe temperature falls to 42°C. This gives an effective temperature of 29°C-target value for the work pattern in question.

CHAPTER 2

General ventilation

Natural ventilation, with open windows in summer, may suffice for the living rooms and bedrooms in our homes, where there is plenty of space per person and no generation of steam or cooking fumes. It is, however, unpredictable, and will fail altogether in unfavourable conditions of wind and weather. In many areas within a building, therefore, mechanical ventilation, powered by fans, is a practical (and often a legal) requirement.

The rate of ventilation, conveniently measured in litres of air per second must be sufficient to satisfy the following three requirements:

(a) Sufficient air movement throughout the room or building to prevent the formation of pockets of stale, stagnant air.

(b) Sufficient fresh air supply and foul air exhaust to limit the level of air pollution from all sources in the building, including humidity.

(c) Reduction of air temperature, within the limits set by the climate, by the removal of heat generated within the building or supplied by the sun.

2.1 Ventilation Requirements

2.1.1 Air change ratings

The simplest method of determining the ventilation rate required is to make use of the accumulated experience of the industry expressed in a

Table 2.1

Recommended air movement rates for general ventilation*

Air changes per hour	Typical situation
1–2	Residences Churches Storage areas
2–4	Libraries Banks Classrooms
4–6	Offices Assembly halls Laboratories
6–8	Hospital wards and treatment rooms Lavatories and bathrooms Bars
6–10	Theatres Cinemas Garages Workshops
8–12	Cafes Canteens Dance halls
10–15	Restaurants Domestic kitchens Laundries
15–30	Kitchens for restaurants or canteens Bakeries, dyers and cleaners Boiler houses, engine rooms Swimming baths
30–60	Paint shops Foundries and furnace rooms

*Check for the possibility of greater air changes required on a per person basis (Tables 2.2 and 2.3) or for heat, fume or moisture removal.
The larger air changes apply to the smaller and more congested examples.
The air extraction points must be so located as to deal effectively with local sources of pollution. For example, kitchen ranges, WC's, fume cupboards, grinding wheels.

table of "air change rates". The volume in cubic metres (m^3) of the space to be ventilated is estimated and multipled by the number of air changes per hour to give the ventilation rate in m^3 per hour. Division by 3.6 converts this to litres per second.

$$\text{Ventilation rate (litre/s)} = \frac{\text{Volume } (m^3) \times \text{air changes per hour}}{3{\cdot}6}$$

Table 2.1 gives a guide to the number of air changes generally recommended. Provided air inlets and outlets are properly sited, the tabulated values will be sufficient for requirement (a), air movement. They will also meet requirement (b), for pollution and moisture removal, in typical examples of the accommodation indicated, and requirement (c), heat removal, in temperate climates without heat sources.

However, particular installations should always be checked against the following questions:

Are there specific bye-laws or other legal requirements applicable in the locality?

Is the space expected to become so crowded that a higher ventilation rate might be obtained by calculating on a per person basis?

Are there abnormal sources of heat or fumes?

Is it desirable to provide a higher ventilation rate for summer cooling?

2.1.2 Ventilation rates per person

As explained in Chapter 1, each human being produces his personal quota of heat, water vapour, carbon dioxide, and body odour. The ventilation requirement for the last item will cover the needs of the other three during the heating season. It is not sufficient merely to ensure an acceptable average freshness of atmosphere throughout the room; if people are crowded together by shortage of space the average odour concentration must be reduced. This means that the fresh air supply per person must increase with the number of people in a room, and the air change rate must increase faster still.

Table 2.2

Recommended minimum fresh air supply

Air space per person	Air supply per person	Air changes per hour
3 m^3	17 litre/s	20
6	11	6·5
9	8	3·2
12	6	1·8

Increase by one-third if smoking is permitted.

These rates should be increased by a third if smoking is permitted or if there are doubts about standards of personal hygiene. If minimum legal requirements are specified, they are generally about a third less. In contrast, as little as a 1 litre/s will suffice to meet the strictly physiological needs of one person for oxygen supply and carbon dioxide limitation.

2.1.3 Recirculation

The flow rates just quoted on a per person basis must be supplied as fresh outside air. So must the air extracted to remove pollution, moisture or heat. However, the flow rate required to achieve the air change recommended in Table 2.1 will usually be considerably greater. The excess can be recirculated through ducts from that part of the building where the bulk of the air is exhausted, to the opposite side. This means that the same air may pass several times through the various rooms in the building, providing the necessary air circulation to prevent stagnation, and reducing the heating or cooling load.

Fig. 2.1 is a schematic example of a ducted system showing the winter and summer arrangements. Fig. 2.2 shows a unit system suitable for single storey factories, with winter recirculation of the warm air collected at roof level.

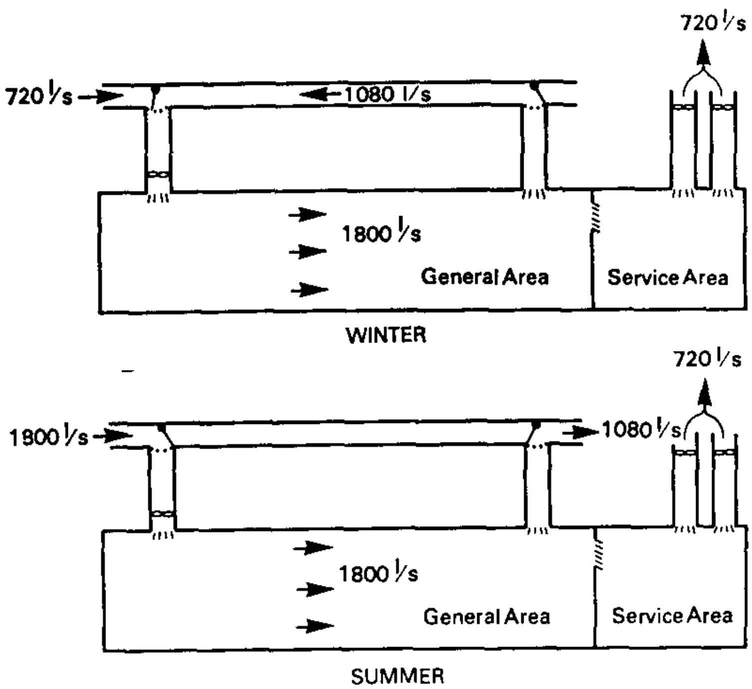


Fig. 2.1 Schematic diagram of general ventilation system with recirculation.

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There would be no value in recirculation if the outside air were clean enough and warm enough to pass, untreated, through the building. However, in winter the whole of the fresh air supplied has to be heated from the outside temperature to the internal comfort temperature. Furthermore, if the outside air is dirty and requires cleaning, the life of the filter elements will be lengthened as the volume of air they handle is reduced. Economy is, therefore, served by minimising the fresh air flow and maximising the recirculation during the heating season. On the days it is better to operate with 100% fresh air, thereby maximising the rate at which the heat generated by the occupants and other sources can be disposed of.

2.1.4 Ventilation for air conditioning

Full air conditioning systems invariably embody recirculation. The cost of cooling plant is high and should not be wasted on the cooling of more than the necessary minimum volume of hot outside air. British practice for such installations is indicated by Table 2.3 taken from the IHVE Guide, 1970. The outdoor air supply rates given take account of the likely density of occupation and the type and amount of smoking. Of the minimum rates quoted per person and per mz, the greater should be taken.

2.1.5 Example

Ventilation is required for an open office with dimensions 12m x 30m x 3m high. The number of occupants is assumed to be 90, this being the maximum permitted under U.K. regulations.

Floor area	12 × 30 =	$360m^2$
Volume	12 × 30 × 3 =	$1080m^3$

We will first select six air changes per hour from the recommendations in Table 2.1, bearing in mind that the occupancy is high. Then the ventilation rate equals:

$$\frac{1080\text{m}^3 \times 6 \text{ changes/hour}}{3{\cdot}6} = 1800 \text{ litre/s}$$

Since the air space per person is 1080/90 = 12M3, Table 2.2 recommends a fresh air supply of 6 litre/s per person. Supposing smoking to be permitted, we will increase this by a third to 8 litre/s, giving a total fresh air supply rate of 8 x 90 = 720 litre/s.

While the building is not to be air conditioned, we will note that Table 2.3 would give:

Recommended fresh air supply	8 × 90 =	720 litre/s
Minimum, per person basis	5 × 90 =	450 litre/s
Minimum, floor area basis	1·3 × 360 =	470 litre/s

We may thus adopt a recirculating system,with a wintertime budget of:

Fresh air supply	720 litre/s
Recirculating volume rate	1080 litre/s
Total ventilation rate	1800 litre/s

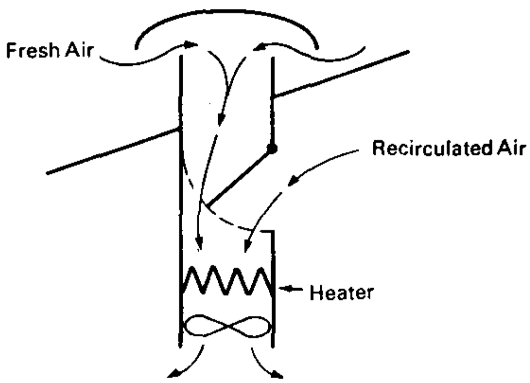


Fig. 2.2 Roof-mounted unit for the supply and heating of fresh and recirculated air.

Table 2.3
Recommended outdoor air supply for air conditioned spaces.

Type of space	Smoking expected	Recommended per person	Minimum per person	Minimum per m² floor area
		litre/s	litre/s	litre/s
Factories	None	8	5	0·8
Offices (open plan)	Some	8	5	1·3
Shops, department stores	Some	8	5	3·0
Supermarkets	Some	8	5	3·0
Theatres	Some	8	5	—
Dance halls	Some	12	8	—
Hotel bedrooms	Heavy	12	8	1·7
Laboratories	Some	12	8	—
Offices (private)	Heavy	12	8	1·3
Residences (average)	Heavy	12	8	—
Restaurants (cafeteria)	Some	12	8	—
Cocktail bars	Heavy	18	12	—
Conference rooms (average)	Some	18	12	—
Residences (luxury)	Heavy	18	12	—
Restaurants (dining rooms)	Heavy	18	12	—
Board rooms, conference rooms	Very Heavy	25	18	6·0
Corridors	Immaterial	—	—	1·3
Kitchens (domestic)	Immaterial	—	—	10·0
Kitchens (restaurants)	Immaterial	—	—	20·0
Toilets	Immaterial	—	—	10·0

The outdoor air supply may have to be increased to meet statutory requirements and local bye-laws, or for heat, fume or moisture removal.

Adjoining lavatory accommodation is planned with six urinals and nine WC's. At the recommended rate of *24* litre/s per point this will require an extraction rate of 15 x 24 = 360 litre/s.

To match the 720 litre/s supply rate, another 360 litre/s extraction will be required. This can conveniently be passed through a service area which might be used for printing, or for a small kitchen, etc. This is the system shown schematically in Fig. *2.1* and as a plan view in Fig. 3.3.

2.2 Sources of Heat

2.2.1 Ventilation rates based on heat gain

The third function of a ventilation system is to mitigate discomfort in hot weather. We are not now considering the positive cooling provided by air conditioning, which is something of a luxury in temperate climates. Here we shall discuss the removal of heat by generous rates of fresh air ventilation which may be well in excess of those recommended in Table *2.1.* The first step in this study is to determine the total heat generated within or transmitted into the enclosure to be ventilated.

2.2.2 Heat from people

The metabolic heat given off by the occupants was discussed in Chapter 1. Average values sufficient for the present purpose may be taken as:

Passive and seated, e.g. audiences	0·11 kW per person
Some movement, e.g. offices, shops	0·15 kW per person
Bench work, dancing	0·25 kW per person
General industrial activity	0·35 kW per person
Heavy industrial labour	0·45 kW per person

2.2.3 Heat from animals

Approximate values:

Sow	0·074 kW each
Piglet	0·005 kW each
Calf (12 weeks)	0·210 kW each
Layers—heavy hybrids	0·007 kW each
Layers—light hybrids	0·006 kW each
Cattle—adult	0·410 kW each
Rabbits—adult	0·007 kW each

2.2.4 Heat from lighting

Heat from lighting apparatus is simply the aggregate rating of the tubes and bulbs installed, discounting those that will not be in use when solar heat gain is a major factor. Sometimes part of the ventilation air is extracted through special recessed lighting fittings. The convected heat from the lights will then be extracted with the air, and will not form part of the room heat load.

The radiant heat load remaining will be about 80% of the rating for tungsten lamps, 60% for fluorescent.

2.2.5 Heat from electric power

With very few exceptions the whole of the energy supplied to electrical apparatus will be dissipated as heat within the building. The maximum will be the full load rating of the appliance in kW, although allowance can be made for the dispersion factor when known. This factor is the ratio of the kilowatt-hours consumed in one hour to the kilowatt rating and takes account of both part-time and part-load working. Sometimes this factor can be obtained indirectly; for example, a machine shop may have statistics of the number of kWh of electricity consumed per shift - or, better, the maximum half-hour kW demand.

2.2.6 Heat from electric motors

In the case of an electric motor the usual rating is the output in hp or kW, whereas what we require is the electrical input, including the motor losses. Typical values are:

Output in hp or kW		Efficiency	Input
$\frac{1}{4}$ hp	0·18 kW	64%	0·28 kW
$\frac{1}{2}$	0·37	68%	0·55
1	0·75	72%	1·05
2	1·5	77%	1·95
5·5	4	83%	4·8
20	15	88%	17
100	75	93%	80

2.2.7 Heat from fuel

The combustion of fuel occurs in some part of the manufacturing process in most industrial plants. In many cases it is a major factor, and general ventilation is necessary for the relief of heat stress, quite apart from local ventilation and other measures which are taken at the sources of heat.

The potential heat load is easily obtained from the calorific value of the fuel and the rate at which it is used:

$$kW = kJ/kg \times kg/s$$

However, a large part of this heat will not contribute to the environmental load. The deductions can be classified under three headings:

(a) Incomplete combustion-unburnt or partially burnt fuel passing up the flue.

(b) Flue heat loss-the heat carried up the flue by hot gases comprising the products of combustion and the excess air.

(c) Heat to stock-occurring when the material worked on (solid, liquid or gaseous) is heated and then passes, still hot, to other parts of the building.

In important cases estimates for each of these items should be available as part of the manufacturing records or plans. More usually, perhaps, the margins of error in the direct calculation of the residual heat loss remaining within the building are too great.. In such cases measurement of the actual air conditions on site, as described in Chapter 1, will be more helpful.

2.3 Solar Heat

2.3.1 Heat from the sun

Solar heat reaches the earth at a fairly constant rate of 1.36 kW/m^2. In passing through the atmosphere some of this heat will be absorbed or scattered into space. The proportion lost will be determined by the length of the path through the atmosphere and by the presence of clouds, dust, water vapour and ozone. Fig. 2.3 gives representative values for a clear sky and various heights above sea level.

The direct intensity, I_d varies with the altitude angle of the sun above the horizon. The effective intensity on any surface is further reduced by the inclination of the surface to the sun. In addition, diffuse radiation, I_f reaches earth's surface after scattering by atmospheric dust and vertical walls receive reflected radiation from the ground.

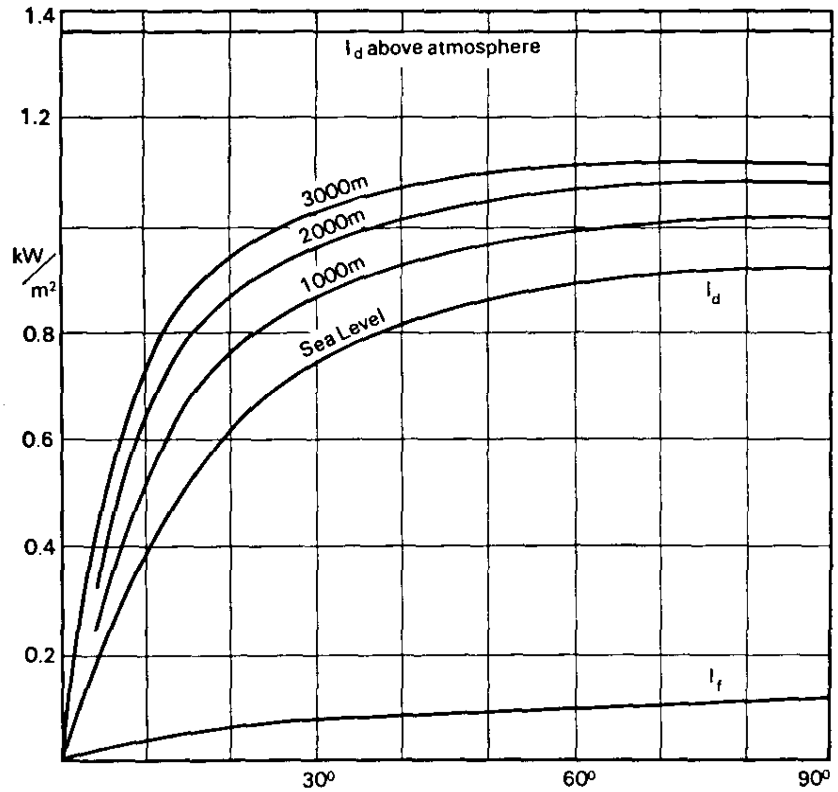
This simple statement gives complete information for calculating the solar heat reaching any exposed surface on a sunny day. However, the work involved in allowing for the sun's movements during the day, and over the year, is time consuming. Such allowances are necessary in estimating the refrigerating plant capacity and energy consumption for a full air conditioning system, and extensive charts and tables to assist in the task will be found in works of reference.

2.3.2 Maximum solar heat intensity

However, our present aim is merely to estimate the ventilation rate required to limit the internal temperature in hot weather. For this purpose we need consider only that time of day and month of the year which will give the maximum solar heat load. This has been done in drawing up Tables 2.4 and 2.5, which can be used to estimate the maximum solar intensity on walls and roofs.

Fig. 2.3 Intensity of solar heat radiation

Effect of location and orientation



α = altitude of the sun above the horizon.

I_d = Intensity of direct solar heat radiation falling on a surface directly facing the sun.

I_f = Intensity of diffuse radiation falling on a horizontal surface.

$I_h = I_d \sin\alpha + I_f$ = Total intensity of solar heat falling on a horizontal surface.

$I_v = I_d \cos\alpha \cos\beta + 0{\cdot}5\, I_f + 0{\cdot}1\, I_h$ = Total intensity of solar heat falling on vertical surface facing in a direction $\beta°$ away from the sun.

Noonday altitude of the sun at L° north or south latitude

Northern Hemisphere	June	May July	April August	March Sept.	Feb. Oct.	Jan. Nov.	Dec.
α	(113–L)°	(110–L)°	(101–L)°	(90–L)°	(79–L)°	(70–L)°	(67–L)°
Southern Hemisphere	Dec.	Jan. Nov.	Feb. Oct.	Mar. Sept.	April Aug.	May July	June

Table 2.4

Maximum solar intensity on vertical walls

Î kW/m² at sea level

N. Latitude	0°	10°	20°	30°–60°
Wall facing S	0·51	0·62	0·69	0·73
SSE or SSW	0·61	0·67	0·71	0·73
SE or SW	0·70	0·72	0·72	0·73
E or W	0·73	0·73	0·73	0·63
Wall out of sun	0·16	0·16	0·16	0·14

Table 2.5

Maximum solar intensity on roofs

Î kW/m² at sea level

N. Latitude	0°–30°	40°	50°	60°
Flat roof	1·03	0·99	0·91	0·80
Sloping roof—				
facing sun	1·0	1·0	0·95	0·85
15° away from sun	1·0	0·9	0·8	0·7
30° away from sun	0·9	0·7	0·6	0·4
45° away from sun	0·7	0·5	0·4	0·2
60° away from sun	0·5	0·3	0·2	0·1

Generally speaking the maximum intensities on various parts of the building structure occur at different times of day, and the corresponding heat loads cannot be simply added together. The following procedure is suggested as a basis for calculating the heat load to be removed by ventilation.

(a) Select the wall which may be expected to provide the greatest solar heat load, and estimate the maximum intensity on it from Table 2.4 for the direction in which it faces and the latitude.

(b) Assume that the "out of sun" intensity of Table 2.4 applies to the remaining walls at right angles and on the other side of the building.

(c) Assume that the "flat roof" intensity of Table 2.5 applies to the whole plan area of the building as if the roof were flat, even if it is actually pitched. Exclude glazed areas and any sections shaded during the heat of the day.

(d) Take the intensity applied to the actual area of skylights, north lights, etc., from Table 2.5 according to their inclination to the horizontal towards or away from the sun when it faces wall(a).

It should be noted that if the terrain is very bare and the air very dry, increased ground reflection and atmospheric clarity may increase tropical solar intensities by 20%. Conversely, Industrial atmospheres may reduce intensities by 20% to 40% even on the sunniest days. Altitude corrections can be derived from Fig. 2.3, but for moderate heights an addition of 1% per 100m above sea level will suffice. The figures in the tables are for the time of year and day giving the highest intensity for each entry. The 24-hour average midsummer intensity averaged over walls equally exposed to all points of the compass is remarkably constant at about 0.08 kW/m^2, with about 0.35 kW/m^2 on roofs. The tables may be used for the southern hemisphere by interchanging N and S.

2.3.3 Calculation of solar heat gain

Having found the intensities of solar radiation falling on the faces of a building, the next step is to determine the total solar heat gain entering through windows, walls and roofs. This is, of course, proportional to the area of each surface, but is reduced by a factor taking account of the heat excluded by reflection and external convection. Values of this factor S for typical building features are given in Table 2.6.

If A is the gross area of the window, wall or roof section under consideration in square metres, and S the appropriate transmission factor, the corresponding rate at which the sun's heat enters the building will be for any surface:

$$H = SA\hat{I} \text{ kW}$$

where $\ddot{I}$ is the effective maximum solar intensity obtained from Tables 2.4 and 2.5 in accordance with the procedure specified in Section 2.3.2 (a), (b), (c) and (d).

Account must, of course, be taken of those times when the windows are wholly shaded by neighbouring buildings and trees, or partly shaded by architectural features such as deep eaves or mullions. External sun blinds are very effective provided they are used, but internal blinds or curtains much less so. They themselves become heated, and re-radiate heat to the room.

2.4 Ventilation Rate and Temperature Rise

2.4.1 Heat capacity of air

For the detailed study of the thermal history of a building we would need to consider the daily cycle of solar and other heat gains. Then we would have to examine the thermal capacity of the various parts of the building structure and the rate at which heat is dissipated as a result of internal temperature rise. For our present purpose, however, we will ignore the contribution to temperature limitation which the building makes, supposing it to be of light construction so that ventilation must dispose of the whole heat gain.

The specific heat of air at constant pressure and normal temperature is about 1.0 kJ/kg (kilojoules per kilogram) *. It follows that a heat gain

*See also Table 14.9.

Table 2.6

Effective transmission factors for solar heat through following building surfaces	Transmission factor S
Windows	
Open area	1·00
Single glazing	0·85 to 0·80
Double glazing	0·75 to 0·65
Special heat absorbing glazings	0·60 to 0·30
Window with Venetian blind	0·55 to 0·45
Window with roller shade (light colour)	0·30 to 0·20
External sun blind or awning	0·25 to 0·15
Roofs	
Corrugated iron or asbestos (unlined)	0·25 to 0·22
Corrugated aluminium (unlined)	0·15 to 0·12
Corrugated sheeting lined with boards	0·08
Tiles laid on battens	0·25
Tiles laid on boards	0·16
Tiles or sheet on boards and felt	0·05 to 0·06
Asphalt on 15cm concrete	0·09
Asphalt on concrete with cork	0·02 to 0·03
Walls	
Corrugated iron or asbestos (unlined)	0·22 to 0·20
Corrugated aluminium (unlined)	0·13 to 0·10
Corrugated sheeting lined with boards	0·07
Wooden boarding (unlined)	0·09 to 0·07
Concrete—15cm	0·11
Concrete—25cm	0·08
Concrete or stone—40cm	0·07
Brick—12cm	0·12
Brick—24cm	0·08
Brick, cavity, plastered—28cm	0·05
Brick, cavity, plastered—40cm	0·04

Note 1 : Factors for window glazing with the sun's rays at more than 45° to the surface. As grazing incidence is approached the transmission factor falls, as well as the effective intensity.

Note 2 : The factors for walls and roofs (except aluminium) assume dark colours. For medium colours multiply S by 0·8 and for light colours (allowing for some deterioration) by 0·6.

of 1 kW or 1 kJ/s will raise the temperature of air flowing at the rate of 1 kg/s by 1 °C. Therefore:

$$\text{Temperature rise, °C} = \frac{\text{Heat gain, kW}}{\text{Air flow, kg/s}}$$

$$= \frac{\text{Heat gain, kW}}{\text{Air flow, m}^3\text{/s} \times \text{air density, kg/m}^3}$$

Air will enter the building at the outside air temperature. As it flows through the building its temperature will rise until, at the point of extraction, it has reached the value given by this formula. If there is a major heat source in the building it is clearly wise to extract air close to the source, preferably so that nobody need work in the space between. We can then discount the major source in considering the temperature rise of the general ventilation system.

Since large volumes of air can be moved quite cheaply through industrial buildings, general ventilation is a potent means of maintaining reasonable comfort during spells of hot weather.

2.4.2 Example

A single-storey machine shop has a plan area of 40m x 12m and a height of 4m to the eaves and 6m to a central ridge. The 40m wall faces 40° west of south and has five windows. A roof light on the NE side spans 32m and is 1 m wide. Rural situation in UK at 52° north latitude.

	kW
Metabolic heat:	
Sixty operators assessed at 0·35 kW each	21
Lighting load:	
Assessed at 30 W per square metre,	
40m × 12m × 0·03 kW/m² = 14·4 kW. In full sunlight say	3
Electric power:	
Motors, mainly 2 to 5 hp, total installed rating 120 kW;	
corresponding input = 120/0·80 = 150 kW.	
Taking a time and load diversity factor 0·33. 150 × 0·33	50
Solar heat gain:	
Windows in SW wall, single glazed, five at 4m² each:	
0·82 (S) × 20 (m²) × 0·73 (Î, Table 2.4)	12·0
SW wall, 28cm cavity brick, 40m × 4m –20m² = 140m²	
0·05 (S) × 140 (m²) × 0·73 (Î, Table 2.4)	5·1
Remaining walls, 40m × 4m + 2m × 5m = 280m²	
0·05 (S) × 280 (m²) × 0·14 (Table 2.4, "out of sun")	2·0
NE roof light, single glazed, 32m × 1m × = 32m²	
0·82 (S) × 32 (m²) × 0·75 (Table 2.5, 18° slope)	19·6
Roof, lined corrugated asbestos, light colour	
(S = 60% of 0·08, Table 2.6) 40m × 12m –32m² = 448m²	
0·048 (S) × 448m² × 0·89 (Table 2.5, flat roof)	19·1
Maximum solar heat gain	58
Total heat load	132

If we set a temperature rise target of 5°C we have, for the required ventilation rate:

$$\frac{132\ \text{kW}}{5\,^\circ\text{C} \times 1{\cdot}2\ \text{kg/m}^3} = 22\ \text{m}^3/\text{s}$$

This will be met by an installation of five 800mm propeller fans at 4800 litre/s each, 24m^3/s total.

$$\text{Volume of building: } 40 \times \tfrac{1}{2}\,(4 + 6) \times 12 = 2400\text{m}^3$$

$$\text{Air changes per hour} = \frac{3600 \times 24}{2400} = 36$$

$$\text{Temperature rise} = \frac{132\ \text{kW}}{24\text{m}^3/\text{s} \times 1{\cdot}2\ \text{kg/m}^3} = 4{\cdot}6\,^\circ\text{C}$$

This is the temperature rise of the air when finally extracted. The occupied zone may be expected to be about 3°C above the outside temperature.

CHAPTER 3

Air distribution and circulation

The main topic in this chapter will be the siting of air inlets and outlets and their design in relation to air movement in the ventilated spaces.

3.1 Distribution System

3.1.1 Extraction systems

These are of greatest significance for industrial buildings. The first requirement will be the local extraction of fumes or other contaminants from localised sources, using hoods and ducts as necessary; this is dealt with in Chapter 4. Usually, however, a much larger volume flow of air is needed to meet the general air change or summer time heat removal requirements.

We may take as an example the 40m x 12m machine shop of Section 2.4.2. For this we specified $24m^3/s$ of general ventilation, equalling 36 air changes per hour. Fig. 3.1 c shows how this could be arranged, fitting five 800mm propeller type exhaust fans in one long wall, with open windows providing air inlets on the opposite side.

The size of the inlets is important. Obviously, if they are too small they will restrict the air flow, and full benefit will not be obtained from the fans fitted. To match a propeller fan 1 ½ to 2 times the fan area is advisable. The fan area in the present case is $5 \times \Pi\ (0.4)^2 = 2.5m^2$. If each of the five $4m^2$ windows is allowed $1\ m^2$ opening this will be ample.

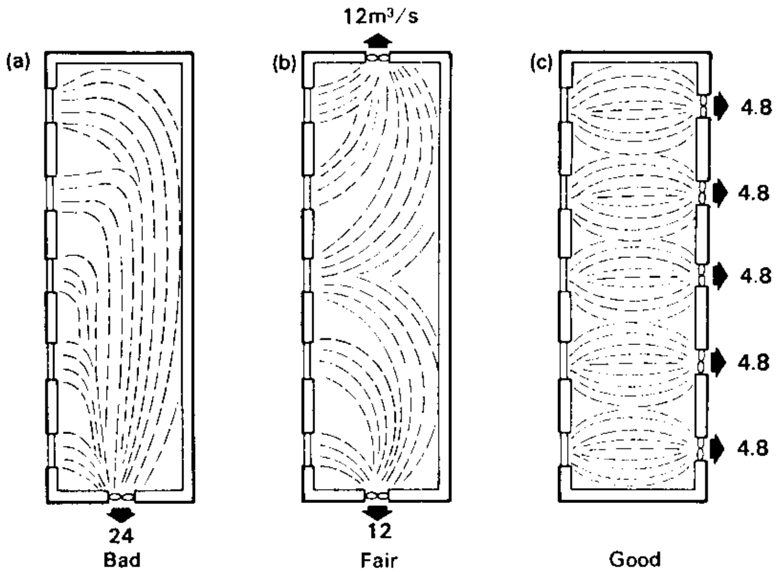


Fig. 3.1. Good and bad distribution with wall fans.

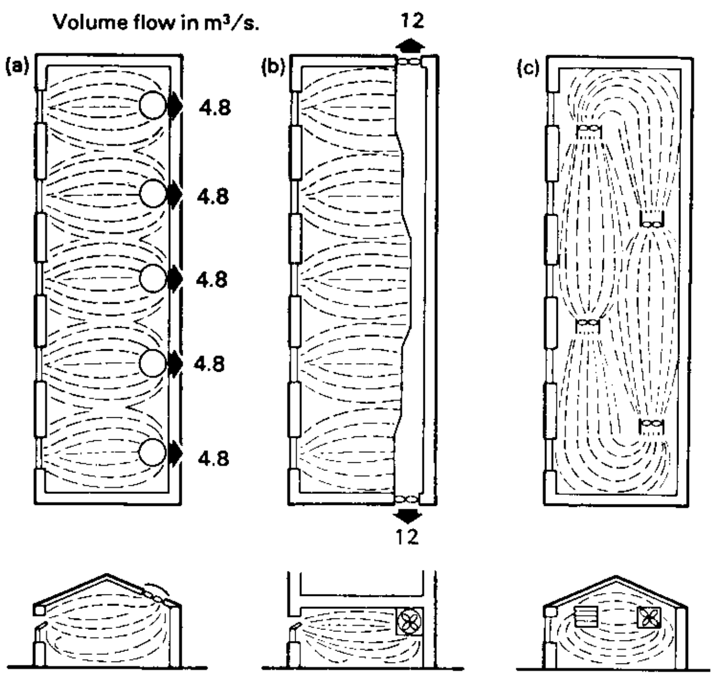


Fig. 3.2 Alternative examples of good practice.

(a) Good distribution with exhaust fans in roof units.

(b) Good distribution with ducted fan exhaust.

(c) Good winter circulation with unit heaters.

Since the inlet velocity at 24 (m^3/s)/5 (m^2) = 4.8 (m/s) is quite high it will be best to allow the opening at the top of the windows, above head level. Air will then sweep across the roof space-where the hottest air collects-and out through the fans while gentler, but still generous, air movement will be induced in the working space below.

For successful ventilation the exhaust points and natural inlets must be so arranged that fresh air is drawn across all parts of the space. This is secured in Fig. 3.1 c, but we will now suppose that the wall on which the fans were mounted is against another building and not available for extraction. A single gable-end fan as in Fig. 3.1 a would be a bad arrangement, even though it extracted the full 24M^3/s. There would be no air movement at the far end of the shop, and even the fitting of a fan at each end as Fig. 3.1 b would leave the centre inadequately served.

The difficulty might be met by ducted extraction, preferably with a fan at each end, which halves the volume to be carried by the duct and the distance the air must travel. As indicated in Fig. 3.2b adjustable registers in the duct wall secure uniformly distributed extraction along the length of the room. However, when a roof is available, the fan-powered roof units located as in Fig. 3.2a are likely to provide the most economical solution, in both first cost and running cost.

3.1.2 Supply and exhaust systems

Air inlets round the walls are insufficient for large industrial buildings. They would be too far from the extract points and the air would become excessively heated or contaminated *en route.* Roof mounted, fan powered, supply units and extract units provide a convenient arrangement for single storey buildings. They may be spaced alternately every 10 or 15 metres in such a way as to ensure cross flow over the whole area. The supply units should be ducted down to a level at which all round radial outlets will distribute the air above head level. Sometimes, however, the need to clear travelling cranes necessitates high level inlets, which must then project a jet of air downwards towards floor level. The general extraction units should take the air from as high a level as possible, where it will be hottest. Large multi-storey buildings will, of course, require supply and extraction fans and ducts with air outlets and inlets distributed along them.

So far we have considered only high-volume general ventilation systems for summer time cooling. In winter general ventilation will still be required, at a lower rate. Where there are fan-powered supply units, they will be fitted with heater batteries. These may merely "temper" the incoming air, raising its temperature to the internal comfort level only, the building fabric losses being supplied by a separate heating system. More usually, perhaps, in an industrial installation, the air supply will carry the whole of the heating load. For this purpose it must enter at a temperature considerably above the comfort level, which complicates the distribution aspect, as will be discussed later.

3.1.3 Heating with air circulation

The winter time examination of our machine shop example, which will be found in Section 4.3 shows that Jm3/s will amply supply the requirements of fresh air ventilation. This is much too little to carry the heating load, neither could it set up sufficient air movement for comfort. Fan-powered unit heaters provide an answer at low cost, and may be mounted as in Fig. 3.2c to maintain a relatively high velocity in the upper levels, with induced air circulation below.

Better control of air movement, of comfort temperature, and of heat supply can be maintained with ducted supply and extraction systems. They lend themselves to winter time recirculation with minimum fresh

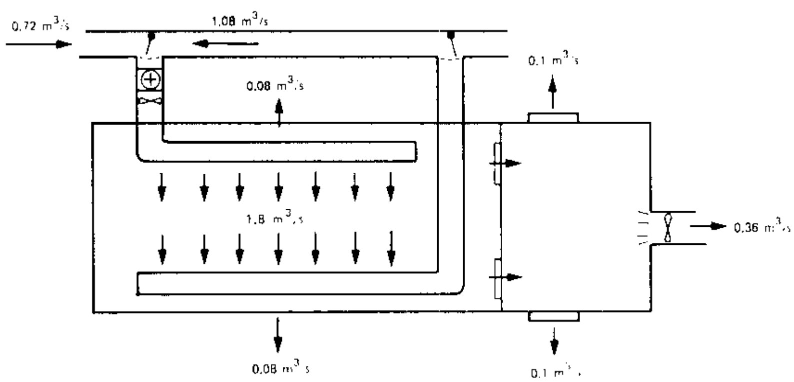


Fig 3.3 Schematic diagram of general ventilation system with recirculation—winter.

air supply and are commonly arranged with an excess of supply over extraction air, so that the surplus will escape outwards through the various leakage paths round windows, doors, etc. Fig. 3.3. takes as an example of such a scheme the 30m x 12m x 3m office of Section 2.1.5.

Parallel supply and extraction ducts establish cross-flow rather than the less desirable longitudinal flow as the main pattern of air movement while 0-36M3/s of positive extraction is established through exhaust fans in the lavatories. The remaining 0-36M3/s of fresh air is allowed to escape through grilles appropriately placed in the service area and by leakage around the office windows. The main damper is set when the system is balanced on installation, so as to allow 1.08m^3/s through the return duct and 0.72m^3/s fresh air supply.

In summer the damper is closed against recirculation and the whole supply allowed to escape, either through the outlet of the extract duct or through open windows.

3.2 Jets

3.2.1 Jet controlled circulation

While comparatively crude provisions suffice for high-volume general ventilation systems in industrial buildings, greater sophistication is needed for optimum comfort in a leisure assembly area. The points of air supply are critical, both in their location and in the manner in which the air is introduced.

The air will generally be supplied in the form of a jet in order to maintain control over a relatively large floor area. This jet will be of relatively high velocity and must be kept out of the "occupied zone"-namely the accessible parts of the room not more than 1.8 metres above floor level. As the jet advances across the room it slows down and at the same time expands, dragging into motion or "entraining" an ever-increasing volume of air from the room. When the peak velocity has fallen below 0.5m/s the jet breaks up into random motion and ceases to advance. It is mainly the entrained air, circulating back through the occupied zone to rejoin the jet nearer the supply outlet, which provides the controlled air movement, ideally in the range of 0.1 to 0.25m/s, which is required. In industrial situations velocities up to 0.4m/s would be applicable.

3.2.2 Velocity profiles

The whole pattern of the jet is determined by the jet velocity and area at the supply outlet. The following are the basic equations which apply when the entrained flow pattern is fully established:

$$V_x = \frac{K V_o \sqrt{A_o}}{X} \quad \text{m/s} \qquad 3{\cdot}1$$

$$Q_x = 2\, Q_o\, (V_o/V_x) \quad \text{m}^3\text{/s} \qquad 3{\cdot}2$$

X	=	Distance from supply outlet	m
V_x	=	maximum velocity at X (along jet centre line)	m/s
V_o	=	outlet velocity, averaged over A_o	m/s
A_o	=	minimum area of jet	m²
Q_o	=	V_oA_o = supply air flow rate	m³/s
Q_x	=	total entrained air flow rate at X	m³/s
K	=	constant, with following approximate values:	

Outlet	K
Round or square opening	6·3
Rectangular slot	5·4
Circular, annular, outlet	4·4
Grille with 40% free area	5·1
Grille with 15% free area	4·4
Grille with 4% free area	3·3

These values of K are for V_o = 4m/s and may be 10% lower at low V_o, 10% higher at high V_o.

A_o is the throat area of a nozzle outlet, the free area of a grille, the gross area of a duct outlet, or 60% of the area of a sharp edged orifice outlet.

In the absence of obstructions the velocity profile across the jet is approximately as Fig. 3.4, y being the radial distance from thecentreline. A jet discharged close to a wall or ceiling will cling to the surface, with peak velocity increased by 40% as shown. Effectively this is one half of

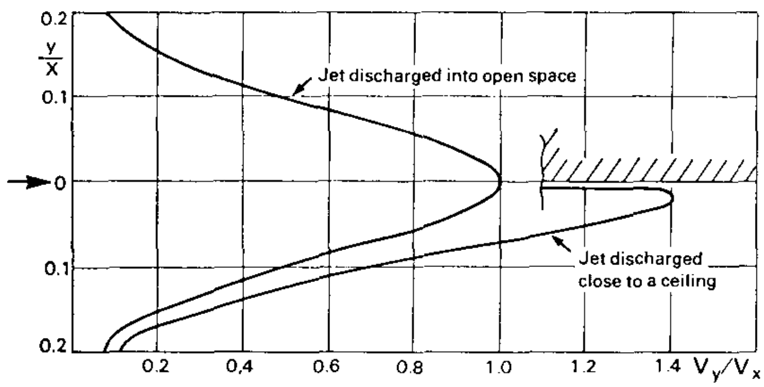
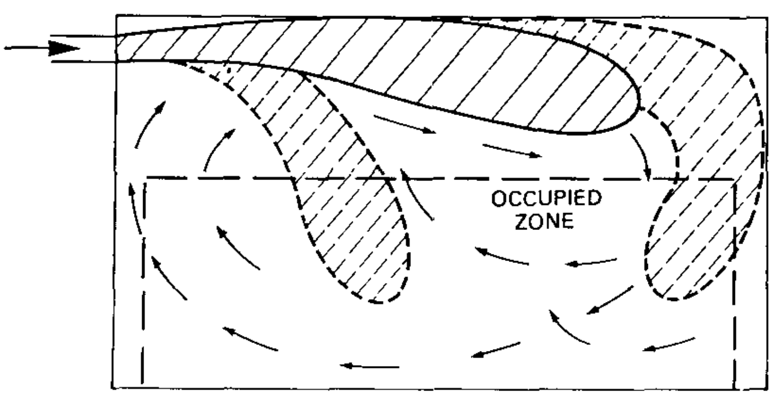


Fig. 3.4 Velocity profiles of jets distant X from the outlet .



Dotted lines show unsatisfactory results of inadequate and excessive throw.

Fig. 3.5 Good jet profile for cooling system (full line).

a jet from an outlet with double the area, the entrained volume flow being the same as it would be from the same outlet into open space.

The throw of a jet is defined as the distance X_t at which the peak velocity V_x has fallen to 0.5m/s. It is good practice to set the throw at about three-quarters of the distance to the wall opposite the outlet. Fig. 3.5 illustrates the type of air distribution aimed at. It also shows, for the case of a cooled air conditioning jet, the result of inadequate throw (draught-making velocities fall into the occupied zone) and of excessive throw (cold draughts down the far wall).

Not only the throw, X_t but also the supply air quantity, Q_o, must be right for satisfactory design, and the selection of Q_o. is dealt with in the next section. Here we may note that the basic equation can be recast to give the required outlet area in terms of Q_o, X_t and V_x. (= 0.5m/s).

$$A_o = \left(\frac{KQ_o}{X_t V_x}\right)^2 \text{ m}^2 \qquad 3{\cdot}3$$

3.2.3 Example

A room 12m x 8m x 3m high is to be air conditioned with outlet slots near ceiling level along the 12m wall. Four air changes are required.

$$Q = (12 \times 8 \times 3) \times \frac{4}{3600} = 0{\cdot}32\text{m}^3/\text{s}$$

A throw of ¾ x 8 = 6m is required, at which point the jet spread to half velocity (see Fig. 3.4) is about 0.2 x 6 = 1.2m horizontally, or rather more with the slit shape and proximity of the ceiling. It will be reasonable to divide the room into four sections, each 3m x 8m x 3m and each supplied with 0.08m^3/s through a single outlet. Inserting these values in equation 3.3

$$A_o = \left(\frac{5{\cdot}4(K) \times 0{\cdot}08(Q_o)}{6(X_t) \times 0{\cdot}5(V_x)}\right)^2 = 0{\cdot}021\text{m}^2$$

Four outlet registers, each 60mm x 600mm, with an effective area 60% of the gross area, would meet this requirement, with an outlet velocity V_o = 0.08 (m^3/s)/0.021 (m^2) = 3.7m/s.

Proprietary outlet registers are available for which performance figures are quoted directly. Preset or adjustable vanes may befitted which cause the jet to diverge and reduce the throw. Ceiling diffusers are an example, and are needed to avoid draught at head level.

3.2.4 Effects of heated and cooled jets

If we examine the results of the last section we shall find that we can provide the required throw and set in motion the required volume of entrained air, either with a small quantity of supply air at high outlet velocity, or with more air at a lower velocity. In fact, the product of the two ($\rho Q_o V_o$, the *momentum flux* of the jet) is constant for a given result.

However, in an air conditioning system, we shall probably want to use the jet to supply heat in the winter and remove heat (by supplying chilled air) in summer. A heated jet will tend to rise and a cooled jet to fall, since they have respectively lower and higher densities than the air in the room. As entrained air is mixed with the jet the average temperature difference from the room air will fall-substantially in proportion to the fall in peak velocity. However, the jet will be excessively deflected up or down if we try to load it with too much heating or cooling duty.

If a single jet is designed to provide satisfactory throw down the length, L m, of a room which is H m high by B m wide, then it should still be satisfactory when carrying a heating or cooling load of W watts, provided the supply volume flow is not less than*

$$Q_o = \frac{1{\cdot}8\,BH}{1000}\sqrt[3]{\frac{W}{B+H}}\ \mathrm{m^3/s} \qquad 3{\cdot}4$$

Strictly speaking, the limiting value is dependent on the shape of the outlet and its location relative to the room. Recent work by Jackman at the British Heating and Ventilating Research Association (now BSIRA) gives a more general treatment.

3.2.5 Example

Substituting the values $Q_o = 0.08M^3/s$, B = 3m, H = 3m from example 3.2.3. in the last equation, we find that the maximum heating or cooling load, W, is 740 W per outlet or 3 kW for the whole space. If the requirement were to be, say, 6 kW, then Q_o must be increased, and V_o decreased, in the ratio $\sqrt[3]{6/3} = 1.26$.

The revised values become:

$$Q_o = 0{\cdot}08 \times 1{\cdot}26 = 0{\cdot}10\mathrm{m^3/s} \text{ per outlet}$$
$$V_o = 3{\cdot}7/1{\cdot}26 = 3{\cdot}0\mathrm{m/s}$$
$$A_o = 0{\cdot}021 \times 1{\cdot}26^2 = 0{\cdot}033\mathrm{m^2} \text{ per outlet}$$

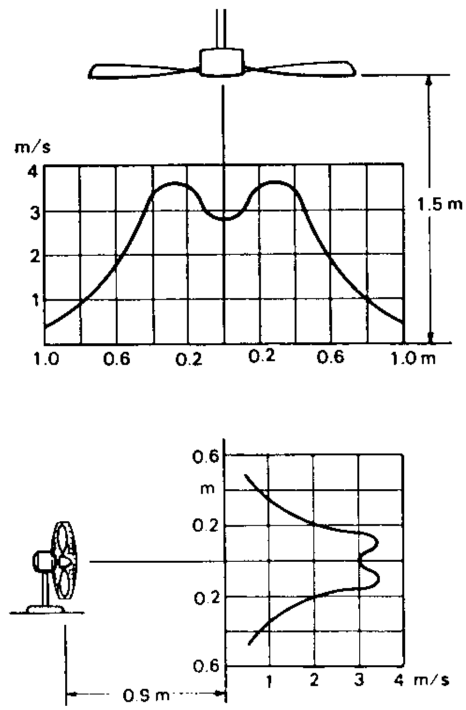
3.3 Circulating Fans

3.3.1 **Relief in hot climates**

The simplest and most commonly applied remedy for the discomfort felt in a hot climate is increased air movement. Velocities well above the 0.1 to 0.25m/s range appropriate to optimum comfort temperatures are needed to exploit the evaporative cooling capacity of the human body when the air is hot. Circulating fans setting the air already in the room into relatively high velocity, directed, jet-stream motion provide an economical and effective system.

*This expression corresponds to a maximum Archimedean Number, Ar, of 10^4. The Archimedean Number is a measure of the ratio of buoyancy to dynamic forces. If g is the acceleration of gravity (9.81 m/s^2) Dh the hydraulic diameter of the room cross section, 2 BH/(B + H), and ΔT the initial difference in absolute temperature:

$$Ar = gD_h\frac{\Delta T}{T}\left(\frac{Q_o}{BH}\right)^{-2}$$



CEILING FAN
1400mm diameter
3·5 rev/s (210 rev/min)
120W input
5·0m³/s

TABLE FAN
300mm diameter
23 rev/s (1380 rev/min)
60W input
1·0m³/s

Fig. 3.6 Velocity profiles of typical ceiling and table fans.

3.3.2 Table and ceiling fans

Table fans, for individual use, and ceiling fans, for places of public assembly, are well known and their sizes and speeds have been standardised by the I.E.C. (International Electrotechnical Commission).

Table fans

Nominal diameter	200	250	300	400	mm
Maximum speed	3400	2700	2200	1700	rev/min
	(though much lower speeds are usual)				

Ceiling fans

Nominal diameter	900	1200	1400	1500	1800	mm
Maximum speed	420	320	270	250	210	rev/min

Both types may be used, seated, with the head directly in the air stream. They are therefore tested by exploring the air velocity distribution at appropriate distances directly in front of the fans. These distances are taken as three impeller diameters in front of table fans, or 1.5 metres below a ceiling fan placed 3m above the floor. Typical performances are shown in Fig. 3.6, the air volume flows being integrated to include all the air entrained at the test plane.

Speed control is usual since the highest air velocities will be needed only on the hottest days. Oscillation mechanisms to swing the jet stream through an arc of 60° to 120° are a refinement. The advantage here is that air motion rising periodically to a peak is found more pleasant than a steady velocity of the same average value.

3.3.3 Pedestal fans

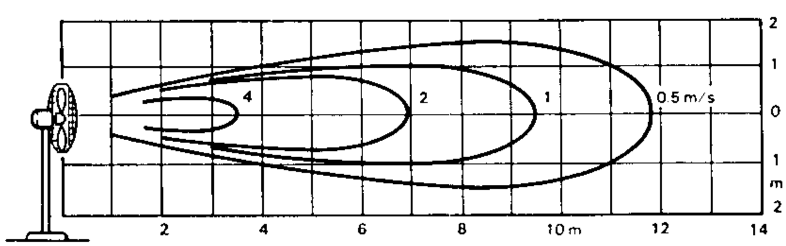
Pedestal fans are similar to table fans but of larger size and longer throw. The oscillating jet-stream is generally directed above the heads of the occupants to promote vigorous entrained air movement over a considerable floor area. A typical plot of constant-velocity contours in the jet stream is shown in Fig. 3.7 and I.E.C. standard sizes are:

Pedestal fans

Nominal diameter	300	400	500	600	mm
Maximum speed	2200	1700	1400	1100	rev/min

It will be seen that ceiling and table fans are capable of producing air movement in the range 1 to 4m/s over the upper part of the body. Velocities at the higher end of this range are welcome when the temperature is very high. At more moderate temperatures the fan speed may be lowered or table fans may be moved to moderate the air movement.

Pedestal fans are perhaps most useful for occasional hot spells in temperate climates when air movement is otherwise inadequate. Being generally designed with a view to attractive appearance and quietness they are most often found in restaurants and similar surroundings.

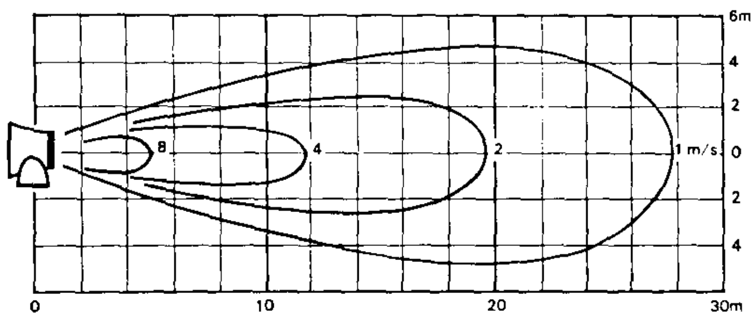


PEDESTAL FAN
600mm diameter 15 rev/s (900 rev/min)
150W input

Fig. 3.7 Velocity contours of a typical pedestal fan.

3.3.4 Mancoolers

The portable cooling fan or mancooler is a much more powerful jetblast device. It is used for the relief of industrial heat stress (see Section 1.8) rather than comfort cooling. A constant-velocity contour plot of the jet from a typical mancooler is shown in Fig. *3.8.* It will be seen that the worker can be provided with a local air stream at velocities from 4 to 20m/s according to the distance the fan is from him. With this assistance he can perform short tasks in a spot at which the heat would otherwise be insupportable.



MANCOOLER FAN
500mm diameter 48 rev/s (2880 rev/min)
2250W input

Fig. 3.8 Velocity contours of typical circulator fans.

CHAPTER 4

Heating and cooling of buildings

We have seen that a part, or indeed the whole, of the heating and cooling requirements of a building may be carried by the ventilating air. Brief attention will, therefore, be given to the principles governing the estimation of these loads.

A significant part of the load is often provided by the unintended infiltration of outside air caused by the wind. A discussion of wind effects on buildings is therefore included in this chapter.

4.1 The Heating Load

4.1.1. Design temperatures

Heat flows through the wall of a building in proportion to the temperature difference between the space inside, t_c, and the air outside, ta. The flow, Hf, is further proportional to the wall area, A, and to a constant, U, called the thermal transmittance and dependent on the wall structure.

$$H_f \text{ (watts)} = A \text{ (m}^2\text{)} \times U \text{ (W/m}^2 \text{ °C)} \times (t_c - t_o) \text{ °C}$$

to will be the same as the comfort index temperature discussed in Chapter 2. t_o is usually taken as the "external design temperature" established by convention in different parts of the world. This is not the extreme minimum temperature on record, but is a value that can be expected on several days in an average year. Here are some typical figures-though the local convention should always be ascertained. See also Table *14.16.*

Great Britain	t_o = − 1°C
North Mediterranean coast	t_o = 0° to − 3°C
USA (New England area)	t_o = − 10° to − 18°C
Denmark	t_o = − 12°
Central Europe	t_o = − 15° to − 18°C
Canada (Great Lakes area)	t_o = − 20° to − 27°C
Inhabited Arctic areas	t_o = − 35° to − 45°C

4.1.2 Thermal transmittance

Some typical values for thermal transmittance, U, are given in Table *4.1.* These are for illustration only and the heating engineer will need to consult much more comprehensive data for the particular materials and construction at his disposal. Good examples are those in the IHVE and ASHRAE Guides. ,

For each value of U the corresponding gross area of *wall,* window, roof, etc., will be calculated. Partitions, floors and ceilings should not be forgotten, as they may not always be exposed to the same temperature, t_c, on the far side. Floors on the ground are listed in Table *4.1* with the inclusion of ground transmittance through to t_o. Heat will flow to the nearest exposed wall and a way of dividing the floor into lengthwise strips and corner sections with different U values is indicated in the key diagram.

4.1.3. Total heat load

The contributions of the sections are now added together to obtain the total heat load, H_f, to maintain the comfort temperature, t_c, with outside temperature t_o, considering losses through the building fabric only.

$$H_f = A_1 U_1 \Delta t_1 + A_2 U_2 \Delta t_2 + \dots \text{ watts}$$

There will bean additional heat load, H_v required to raise the temperature, of the fresh air brought in, Q_v, from t_o to t_a (not necessarily the same as t_c - see Section *4.2).*

$$H_v \text{ (Watts)} = 1200 \times Q_v \text{ (m}^3\text{/s)} \times (t_a - t_o) \text{ C}^\circ$$

Solar and other heat gains discussed in Chapter *2* will form the major part of the heat load on a cooling system. If the building is air conditioned there will be an additional load due to the heat flowing *into* the building from outside temperature t_o to the lower internal comfort temperature, t_c. This will be $H_f + H_v$ calculated as before.

4.2 Relative Influence of Radiation and Convection

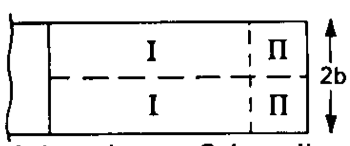
Comfort temperatures are assessed on the assumption that the mean radiant temperature and the air temperature are the same. Heating the room by warming the air only (100% convection) will mean that the

Table 4.1

Typical thermal transmittances for buildings

U values	W/m² °C
Walls	
75mm precast concrete panels	4·0–4·8
105mm brick or 150–200mm cast concrete	3·0–3·6
220mm brick	2·2–2·4
220mm brick, plastered inside	1·8–2·2
260mm cavity brick, plastered inside	1·3–1·6
Various lightweight concrete/brick combinations	0·8–1·0
Concrete sandwich structures with insulating layers	0·6–1·1
Cavity wall with expanded polystyrene in cavity	0·7
Curtain walling—glass and expanded polystyrene	0·8–1·5
Curtain walling—steel and asbestos board	1·1–1·8
Curtain wall with metal mullions projecting both sides	2·0–3·6
Windows	
Single glazed, metal framed	5·0–7·0
Single glazed, wood frames	4·0–5·0
Double glazed	2·5–3·5
Single glazed skylight	6·0–8·0
Skylight or lantern light with laylight under	3·0–4·0
Roofs	
Corrugated iron or asbestos cement	6·0–8·0
Corrugated aluminium (bright)	4·0
Corrugated sheeting, insulated and plasterboard lined	1·0–2·0
Tiles on felt or board with plasterboard ceiling	1·3–1·5
Ditto with 50mm glass-fibre insulation between joints	0·5
Asphalt or felt-bitumen on 150mm concrete	3·1–3·7
Ditto with screed and plaster ceiling	2·1–2·3
Asphalt or felt-bitumen on 150mm hollow tiles	2·1–2·3
Ditto with screed and plaster ceiling	1·5–1·7
Light flat roof on joists with plasterboard ceiling	0·9–1·4
Ditto with 25mm glass-fibre insulation between joists	0·6

Floors directly on ground



Column I
Single-edged
areas, b wide

Column II
Double-edged
corners, b × b

b	I	II
1m	1·2	2·0
2m	0·90	1·4
4m	0·52	0·80
8m	0·26	0·42
15m	0·16	0·26
30m	0·10	0·15

walls will be relatively cold. This will increase the heat which the body disposes of by the radiation route. To maintain constant heat loss, and therefore comfort, the body must secure an equivalent reduction in the heat disposed of by the convection route. This can be done if the air temperature is raised, though the rise required is less if the air movement is substantial. At an air movement rate around 01m/s the rise in air temperature needed is about 2°C for every 1 °C fall in the mean radiant temperature. At 0.2 to 0.4m/s it will be nearer to 1 °C for every 1 °C.

Precisely the opposite effect occurs if the heat is supplied mainly by radiation. The body's radiant heat loss will fall, its convective loss must rise, and the air temperature for comfort will be below the comfort index temperature by 2°C per 1 °C radiant excess at 0.1m/s or 1 °C per 1 °C at the higher air velocity.

Since the whole approximation process is fairly crude it is sufficient to class heating systems as mainly convective or mainly radiant. The following is a general classification by type, with typical proportions:

	Convection	Radiation
Mainly convective:		
Fan convectors. Unit heaters	100%	0%
Steam or hot water "radiators"	70%	30%
Mainly radiant:		
Heated floors	50%	50%
Heated ceilings. Overhead panels	30%	70%
Luminous panel heaters and "fires"	20%	80%

Fortunately the influence of the thermal insulation of the room or space under consideration can be estimated quite simply. The heating design will have determined the total fabric loss, H_f, to maintain the desired comfort temperature t_c with design external temperature t_o. Calculate the total area A_t of walls, floor and ceiling. Then, for optimum comfort in the case of a mainly convective heating system:

Mean radiant temperature $t_r = t_c - 0{\cdot}1\ H_f/A_t$

Air temperature $t_a = t_c + 0{\cdot}2\ H_f/A_t$ (0·1m/s)

Air temperature $t_a = t_c + 0{\cdot}1\ H_f/A_t$ (0·2 to 0·4m/s)

Imagine the case of a 100% radiant system. The radiant heat will travel directly to the walls without heating the intervening air. If there is no ventilation the air will ultimately reach the same temperature t_r as the walls. However, there must actually be fresh air introduced at a rate Q_v m^3/s requiring H_v watts to reach t_c °C. This air can, however, be preheated to a lower temperature t_a which will enable a new balance to be found maintaining comfort temperature t_c. In the case of a mainly radiative heating system:

Mean radiant temperature $t_r = t_c + 0{\cdot}1\ H_v/A_t$

Air temperature $t_a = t_c - 0{\cdot}2\ H_v/A_t$ (0·1m/s)

Air temperature $t_a = t_c - 0{\cdot}1\ H_v/A_t$ (0·2 to 0·4m/s)

An example will show that a mainly radiative system will consume rather less heat than a mainly convective one:

4.3 Example of Heating System

The machine shop of example 2.4.2 is to be heated to a comfort temperature of 18°C with an external design temperature of -1 °C.

Dimensions: 40m × 12m × 4m to eaves, 6m to ridge

Volume: 40 × 12 × 5 = 2400m³

Surface Area: 40 (4 + 12 + 4 + 6·3 + 6·3) + 12 (5 + 5)	= 1420m²
Temperature difference for fabric heat loss = 18 − (−1)	= 19°C
Single glazed windows. 5 at 4m² each	
5 (U) × 20 (m²) × 19 (°C)	= 1,900 W
Long walls. 260mm cavity brick. 2 × 40m × 4m − 20m²	
1·5 (U) × 300 (m²) × 19 (°C)	= 8,600 W
Short walls. 260mm cavity brick. 2 × 12m × 5m	
1·5 (U) × 120 (m²) × 19 (°C)	= 3,400 W
Single glazed roof light. 32m × 1m	
6·5 (U) × 32 (m²) × 19 (°C)	= 4,000 W
Roof. Lined corrugated asbestos. 40m × 12·6m − 32m²	
2 (U) × 472 (m²) × 19 (°C)	= 18,000 W
Floor. Concrete laid on ground. 40m × 12m	
0·35 (U–I) × 2 × 6m × 28m × 19 (°C)	= 2,200 W
0·55 (U—II) × 4 × 6m × 6m × 19 (°C)	= 1,500 W
Total fabric loss, H_f	= 39,600 W

Minimum winter ventilation for 60 persons at 8 litre/s each = 480 litre/s. Installation of five window fans at 200 litre/s each is planned, extracting from small supervisors' and shop clerical offices, gauge room, lavatories, etc., along NE wall. Total extraction 1000 litre/s, equalling 1 .5 air changes per hour, and balanced by natural infiltration without planned intakes.

Ventilation loss, H_v:

$$1{\cdot}2\ (\mathrm{kg/m^3}) \times 1{\cdot}0\ (\mathrm{m^3/s}) \times 1000\ (\mathrm{J/kg\ ^\circ C}) \times 19\ (^\circ\mathrm{C}) = 22{,}800\ \mathrm{W}$$

$$\text{Total loss} = H_f + H_v = 62{\cdot}4\ \mathrm{kW}$$

Clearly this winter heat load will be supplied in substantial part by the casual heat gains in the building (see example 2.4.2). However, 60 kW of heating plant will be installed for early morning and other slack period heating, assuming ventilation loss at the natural infiltration rate of about 10 kW only for these periods.

Consider first convection heating with four unit heaters of 15 kW rating each and 500 litre/s air throughput. The distribution has been illustrated in Fig. 3.2c and it is clear that, with entrained air circulation, there will be substantial air movement. We shall, therefore, take the factor in the air temperature formula as 0.15.

$$\text{Mean radiant temperature} = 18^\circ - 0{\cdot}1 \times \frac{39{\cdot}600\ (H_f)}{1420\ (A_t)} = 15{\cdot}2^\circ\text{C}$$

$$\text{Air temperature} = 18^\circ + 0{\cdot}15 \times \frac{39{\cdot}600\ (H_f)}{1420\ (A_t)} = 22{\cdot}2^\circ\text{C}$$

The revised ventilation loss will be based on $22{\cdot}2 - (-1) = 23{\cdot}2^\circ$C:
$H_v = 1{\cdot}2\ (\text{kg/m}^3) \times 1{\cdot}0\ (\text{m}^3/\text{s}) \times 1000\ (\text{J/kg}\ ^\circ\text{C}) \times 23{\cdot}2\ (^\circ\text{C})$
$= 27{,}900$ W

As a radiant heating alternative, consider twelve 5 kW overhead gas heated panels. Air movement will now be low, and we will take the air temperature factor as 0.2.

$$\text{Mean radiant temperature} = 18^\circ + 0{\cdot}1 \times \frac{22{\cdot}800\ (H_v)}{1420\ (A_t)} = 19{\cdot}6^\circ\text{C}$$

$$\text{Air temperature} = 18^\circ - 0{\cdot}2 \times \frac{22{\cdot}800\ (H_v)}{1420\ (A_t)} = 14{\cdot}8^\circ\text{C}$$

The ventilation loss is now based on $14{\cdot}8 - (-1) = 15{\cdot}8^\circ$C:
$H_v = 1{\cdot}2\ (\text{kg/m}^3) \times 1{\cdot}0\ (\text{m}^3/\text{s}) \times 1000\ (\text{J/kg}\ ^\circ\text{C}) \times 15{\cdot}8\ (^\circ\text{C})$
$= 19{,}000$ W

This is a saving of 9 kW, but the lack of air movement could be unsatisfactory and there may be difficulty in siting the heaters to avoid excessive local intensity.

4.4 Wind Effects

The wind provides the principal motive power both for the natural ventilation of buildings and for the unwanted infiltration of cold or hot air. Wind speed is the customary measure of this power and the meteorological wind speed quoted in weather reports and statistics has a specific meaning: in the UK it is the average air velocity 10 metres above the ground in open, level country. This is substantially below the velocity reached at high altitude owing to the drag forces exerted by the ground. In towns the buildings cause a further reduction in average velocity, though eddies between and around the buildings account for local peaking.

4.4.1 Wind speed statistics

Fig. 4.1 shows the standard variation of wind speed with altitudes for a meteorological wind speed of 9m/s. Typical lower values experienced in suburban areas and in a modern city centre with many tall buildings are also plotted for the same meteorological conditions,. 9m/s is recommended by the IHVE as representative of severe winter weather in the UK for the design of heating systems. Higher values may be considered for exposed locations, reaching a maximum of perhaps 13 to 14m/s on a hill top or cliff edge exposed to strong prevailing winds. For meteorological wind speeds other than 9m/s all three curves of Fig. 4.1. should be adjusted in proportion.

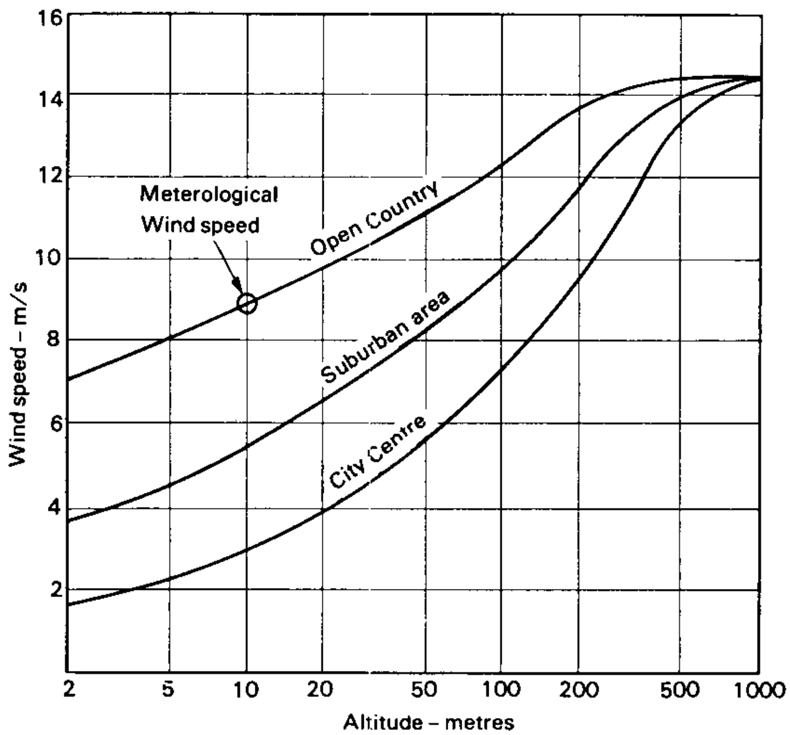


Fig. 4.1 Wind velocity profiles for winter design.

To be of significance for heat loading, both directly at the building surface, and indirectly by cold air infiltration, the wind speed must be sustained. Observations at Heathrow Airport, London, show that temperatures below 2.2°C are combined with wind speeds as follows in an average year:

2·5m/s or more for at least 6 hours on 46 days.
5 m/s or more for at least 6 hours on 13 days.
7·5m/s or more for at least 6 hours on 3 days.

Such cold winds tend to come from a predominant direction-in this case north-east. In fact 50% of the above cases are within t 45° of NE and 30°.6 within ± 22.5°.

The maximum wind speed likely to be reached during the lifetime of a building is the important factor for structural design-including the mechanical design of cowls and other exposed components of air conditioning systems. A map of the UK published by the Building Research Station gives contours of maximum gust speeds (at the 10m, open ground, level) which are likely to be exceeded once only in 50 years. They range from 38m/s in the London area to 55m/s in the Outer Hebrides.

4.4.2 **Natural ventilation** involves open windows, doors and ventilators of substantial area deliberately provided to promote air change by wind and stack effects. The volume flow passing through an opening in a wall exposed to the full impact of the wind would theoretically equal the free area of the opening multiplied by 65% of the unobstructed wind speed. Half this would be a more likely value in an actual building, mainly because of the obstructions to flow through the building and out on the leeward side.

A fairly conservative estimate of the volume flow through a building with no internal partitions is:

$$Q = 0{\cdot}5\, A_e V_w \ \ m^3/s$$

Here V_w (m/s) is the free wind velocity at roof level and A_e is derived from the total free area of inlet openings on the wall facing the wind, A_1m^2, and the total free area of outlet openings on the remaining faces, A_2M^2:

$$A_e = A_1\sqrt{\frac{A_2{}^2}{A_1{}^2 + A_2{}^2}}$$

4.4.3 **Stack effect** arises from the buoyancy of the warm air inside a building which will be at a lower density than the cold air outside. It is most effective in high, single-storey, factory buildings with ridge mounted or other ventilators in the roof. An approximate formula for the volume flow is:

$$Q = 0{\cdot}17\, A_e \sqrt{H\ (t_i - t_o)}\ m^3/s$$

A_e is calculated from the formula in the preceding section, with A_2M^2 the free area of the roof outlets and A_1m^2 the free area of the inlets which should be not far from ground level. H_1m is the height of the outlets above the inlets, to°C the outside temperature, and t_o°C the average inside temperature over the height H.

Example

A factory bay 40m x 14m x 15m high to the ridge, volume 7500m^3, has ridge-mounted natural ventilators of 16m^2 free area, and bucket type window ventilators 2m above the ground along each wall of 20m^2 total free area. If the average air temperature is 25 °C inside and 20°C outside what is the ventilation rate?

$$A_e = 20\sqrt{\frac{256}{400 + 256}} = 12{\cdot}5m^2$$

$$Q = 0{\cdot}17 \times 12{\cdot}5 \times \sqrt{(15-2)\ (25-20)} = 17m^3/s$$

$$\text{Air changes} = 17 \times 3600/7500 = 8 \text{ per hour}$$

If the temperature reached at the ridge outlet is 30°C, the ventilation will remove:

$$17\ (m^3/s) \times 10\ (^\circ C\ \text{rise}) \times 1{\cdot}2\ (kg/m^3) \times 1{\cdot}0\ (kJ/kg\ ^\circ C)$$

equals 200 kW of process heat, 1.2, and 1.0 being the density and specific heat of the air, respectively.

4.4.4 Infiltration

When windows are closed, inward and outward leakages still take place round the edges, and have a significant effect on the heating requirements. The leakage depends on the pressure across the window and its sealing qualities. As a guide to estimation three standard leakage coefficients, Aw have been established for the infiltration rate per metre of joint which are representative of the upper limits to be expected from modern, metal-framed windows:

	λ_w
Weather-stripped and pivotted	0.*05* litre/s per metre at 1 Pa
Weather-stripped and sliding	*0.125* litre/s per metre at 1 Pa
Not weather-stripped	*0.25* litre/s per metre at 1 Pa

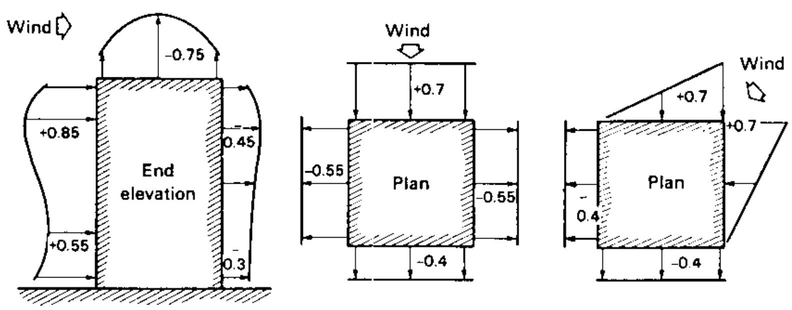
When the length of window joint is L_e metres and the pressure difference Δ_p Pa, the leakage rate of standard air expected is:

$$Q = \lambda_w \cdot L_e \cdot (\Delta_p)^{0.63} \text{ litre/s}$$

The distribution of pressure over a building of the rectangular office block type has been studied with wind-tunnel models, and typical results reported by Jackman (HVRA report *53)* are summarised in Fig. *4.2.* This expresses the positive and negative pressures over the windward and leeward faces of the building as multiples of ½ ρ v^2_w where v_w m/s is the unobstructed wind speed at the height of the roof of the building.

As a general conclusion the gross pressure difference providing the motive force for the inward and the necessarily equal outward leakage flows is taken to be 1.2 x ½ *ρ* v^2_w. Half of this is assumed to act across the inflow side.

To evaluate the effective length of window joint on the inflow faces, L_e, consider a rectangular building facing north, south, east and west with total lengths of window joint LN, Ls, LE and Lw on the respective faces. La would be LN for a north wind, but is increased to $\sqrt{L_N^2 + L_E^2}$ to take account of the worst wind direction in the north-east



Positive and negative pressures are in terms of the velocity pressure of the undisturbed wind at roof level.

Fig. 4.2 Typical pressure distribution over a tall rectangular building.

quadrant. Fig. 4.3 is plotted to give the leakage rate directly from the wind speed at roof level derived from Fig. 4.1 in accordance with:

$$Q = \lambda_w \cdot L_e \cdot (0{\cdot}3\,\rho\, v^2_w)^{0{\cdot}63} \quad \text{litre/s}$$

Example

An open plan office block in a suburban area is 30m x 15m x 50m high and has 200 windows each on the north and south faces, 100 windows each on the east and west. The windows are not weatherstripped and each has a periphery of 5m. What winter infiltration rate should be allowed?

Take $\lambda_w = 0{\cdot}25$

At a meteorological wind speed of 9m/s and an air density of 1·2 kg/m³ $v_w = 8{\cdot}3$m/s (Fig. 4.1).

$$L_e = \sqrt{1000^2 + 500^2} = 1120\text{m}$$

$$Q = 0{\cdot}25 \times 1120\,(0{\cdot}3 \times 1{\cdot}2 \times 8{\cdot}3^2)^{0{\cdot}63}$$

$$= 2100 \text{ litre/s} = 2{\cdot}1\text{m}^3\text{/s}$$

$$\text{or } Q = 1{\cdot}9 \times 1120 = 2100 \text{ litre/s from Fig. 4.3 at } 8{\cdot}3\text{m/s}$$

$$\text{Air changes} = \frac{2{\cdot}10 \times 3600}{30 \times 15 \times 50} = 0{\cdot}34 \text{ per hour}$$

4.4.5 Infiltration-air change basis

Buildings with irregular fenestration and internal subdivision into rooms and corridors do not lend themselves to leakage calculations. Empirical values have been published by the IHVE for use in rough estimates of winter heat loading. The rates in Table 4.2 have been extracted from their tabulation.

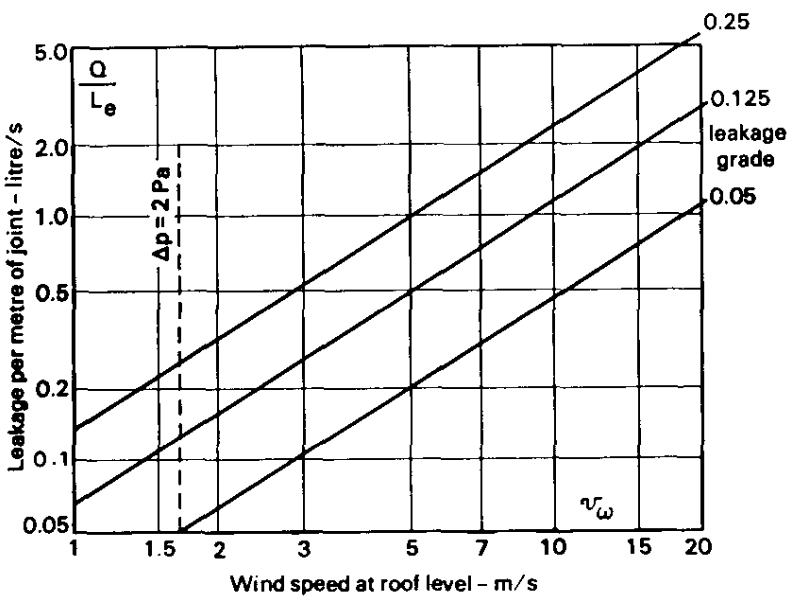


Fig. 4.3 Infiltration through closed windows.

Table 4.2

Typical infiltration allowances

	Air changes per hour
Departmental stores; warehouses; large churches and exhibition halls	$\frac{1}{4}$
Large shops; swimming baths; small churches and assembly rooms	$\frac{1}{2}$
Small shops; restaurants and bars; living rooms; offices	1
Entrance halls; corridors and foyers	$1\frac{1}{2}$
Bathrooms; hospital wards; classrooms	2

Volume, m^3	<300	300–3,000	3,000–10,000	>10,000
Brick or concrete	$1\frac{1}{2}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$
Lined curtain wall	$1\frac{3}{4}$	1	$\frac{3}{4}$	$\frac{1}{2}$
Unlined sheet	$2\frac{1}{4}$	$1\frac{1}{2}$	1	$\frac{3}{4}$

4.5 Cooling Systems

A full air conditioning system with refrigerating and dehumidifying plant is of course capable of securing complete comfort in the hottest climate. The component parts of such a system can be assembled in many different ways, and the subject is outside the scope of this book.

4.5.1 Evaporative cooling

One simple cooling system, known as evaporative cooling, which requires a fan as its only moving part, may be described here.

It is relatively inexpensive but is practicable only in a hot dry climate (with relative humidity usually less than, say, 50%) where nevertheless a supply of running water is available.

It was pointed out in Chapter 1 that a considerable amount of heat is necessary to convert liquid water into water vapour. This is known as the latent heat of evaporation and amounts to more than 2400 kJ/kg (kilojoules per kilogramme) at atmospheric temperatures.

In an evaporative cooling installation hot air from outside the building is drawn through and over layers of fibrous material kept continually wetted by gravity flow or capilliary action, or through sprays. This promotes evaporation, and the only sources from which the corresponding latent heat can be supplied are the air and water flows. Almost all the latent heat must come from the air, heating or cooling the water to wetbulb temperature only accounting for 1 or 2% of the total.

The additional moisture content of the air leaving the evaporator carries with it a corresponding quota of latent heat. This equals the heat extracted from the dry air part of the air water vapour mixture so that the total heat of the mixture (called the enthalpy) remains the same before and after the evaporation. The heat content of the dry air part is called the sensible heat. When this is reduced the dry-bulb temperature falls at the rate of about 1 °C for each 1000 joules of sensible heat lost per kilogramme of dry air. Two very simple approximate relationships follow:

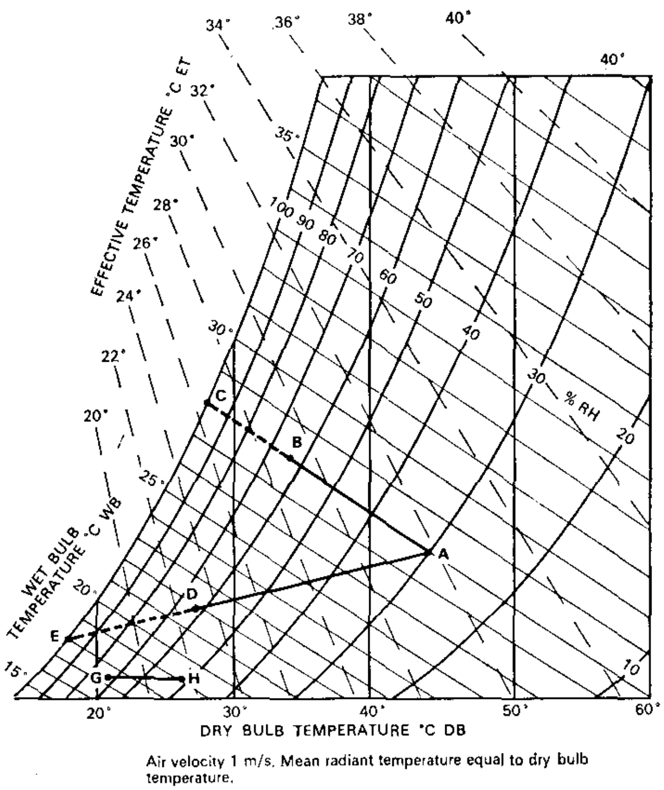
A cooling rate of 1 kW (= 1 kJ/s) applied to an air flow of 1 kg/s will reduce the dry-bulb temperature by 1 °C.

Latent heat at the rate of 1 kW is extracted by the evaporation of water at the rate of:

$$\frac{1{\cdot}0\ (\text{kJ/s} \times 3600}{2400\ (\text{kJ/kg})} = 1{\cdot}5 \text{ kg per hour}$$

Combining these, 1.5 M_t litre per hour of water must be evaporated to cool M kg/s of air by t°C dry-bulb.

It must be realised that the dry-bulb temperature of the air has been reduced at the expense of increasing its relative (and absolute) humidity. This reduces the cooling effect available, which however still remains



Air velocity 1 m/s. Mean radiant temperature equal to dry bulb temperature.

Fig. 4.4 Psychrometric chart for hot climates.

substantial provided the air was not too humid to start with. The process can be followed on a portion of the psychrometric chart selected in Fig. 4.4 to suit hot climate applications. On this chart lines of constant effective temperature have been drawn-the broken lines sloping steeply down. These have been plotted from a chart similar to Fig. 1.3 but drawn for people with customary indoor clothing, sitting, with light activity. Also for an air velocity of 1 m/s as the lowest value appropriate where cooling air streams from table and ceiling fans are available.

4.5.2 Example of evaporative cooling

Suppose the air outside is in the condition represented by A on the diagram (44°C, 30% RH). As it is drawn across the wetted surfaces its dry-bulb temperature will fall, along a constant wet-bulb temperature line, towards saturation at C where wet-bulb and dry-bulb would be equal at 28°C. In practice it will only cool a certain fraction of the way towards saturation, emerging from the cooler in condition B shown as 34°C dry-bulb-62% of the way towards saturation.. This saturation efficiency of 62% is typical of simple wetted fabric coolers. Spray type coolers may reach 90%, but will use water many times in excess of the minimum necessary amount, unless the water is recirculated.

It will be seen that we have reduced the effective temperature from nearly 32°C ET at A to 29°C ET at B, a useful amelioration of an unpleasantly hot environment. This is achieved with a simple construction in which panels of chemically treated fabric, glass fibre or wood-straw are fed with a controlled quantity of water calculated to match the required cooling rate. Given a volume flow of 4M^3/s, mass flow 4.5 kg/s at 1 .12 kg/m^3 density, the evaporation rate is:

$$1{\cdot}5 \times 4{\cdot}5 \text{ (kg/s)} \times 10 \text{ (°C)} = 68 \text{ litres per hour of water.}$$

4.5.3 Comparison with refrigeration methods

If the same outside air were passed through a cooler battery of pipes fed with chilled water, the air condition would move along a line such as ADE. E is the dew-point maintained on the outside surface of the pipes at 18°C. Supposing the saturation efficiency to be 65%, the average condition of the air leaving the battery would be D. 17° drybulb and more than 8° effective temperature reduction have been achieved, and the moisture content has been reduced by condensation on the pipes, though the relative humidity is up to 63%.

With full air conditioning more heat and moisture would be extracted at the cooling stage, by the use of refrigerant in the cooler battery in the place of chilled water. This cools the air still further, but at the expense of a rather high relative humidity, say at point G. A reheat stage follows at constant moisture content (i.e. horizontally on the chart) to H at the target dry-bulb temperature of 26°C with a satisfactory relative humidity. It is the overcooling and reheating stages which give the complete control over the final air condition characteristic of full air conditioning.

CHAPTER 5

Pollution control

The ventilating system has the major role in the control of pollution within buildings. The techniques required to deal with toxic and explosive gases, vapours, fumes and dust form the subject of this chapter.

5.1 Toxic Gases and Vapours

Complete exclusion of poisonous material from the human body is impracticable. Traces of virtually any substance may from time to time be found in the food we eat or the air we wreathe. However, the body can successfully eject or absorb such foreign invasions, provided the quantities involved are not too great.

Medical science has identified and studied the action of a great many harmful substances. Many of these appear as gases or vapours in industrial atmospheres and experience has shown what concentrations are tolerable. Table 5.1 lists some common gases with the limitations usually recommended but it is intended only to illustrate the application of ventilation techniques. Threshold limits are officially specified in many countries and must be observed.

National legislation administered by the Factory Inspectorate or Public Health Authority must always be studied and respected. The regulations deal in detail with many more chemical formulations than could be listed here and are kept abreast of current medical opinion. The maximum allowable concentration, known as the "threshold limit value" is such that lifetime exposure during the working day should not damage the health. Nevertheless, some individuals may be particularly susceptible to certain toxic agents; medical tests can often identify such people, and prevent their employment in areas harmful to them.

Some poisons are cumulative in their effect, and the allowable concentrations are then far below levels that would be quite safe for an isolated exposure. Others cause immediate harm, and these can be identified in Table 5.1 by noting that the short period maximum is the same as the daily average.

5.1.1 Dilution ventilation for toxic hazards

General ventilation will deal with many of the toxic hazards arising in industry. The manufacturing process must be studied to determine the rate at which the pollutant is generated. If this is expressed in milligrams per second* of gas or vapour, with density ρkg/m^3 at atmospheric temperature and pressure, then the ventilation rate required for dilution is:

$$Q\ (m^3/s) = \frac{SF \times M(mg/s)}{\rho(kg/m^3) \times ppm} \qquad (51)$$

In this formula both M and ppm (parts per million by volume) may be either daily average or short-period values, whichever gives the higher value of Q. SF is a safety factor, commonly chosen between 3 and 6, according to the uncertainty of the data and to possible defects in the ventilation distribution. Good distribution was discussed in Chapter 3; the danger is of local gas concentration in places where workers may spend a significant part of their time.

5.1.2 Example

Acetone is used as a solvent in a workroom of 50M^3 volume. The maximum recorded consumption is 5 litres in an eight hour shift.

The density of liquid acetone is 790 kg/m^3, that is 0.79 kg/litre, so that the vapour generation rate is about 4 kg in eight hours, ½ kg/hr or 140 mg/s.

The maximum daily average concentration of acetone vapour from Table 5.1. is 1000 ppm and the vapour density is 2.44 kg/m^3. Taking the factor of safety as 5, the required general ventilation rate is:

$$Q = \frac{5 \times 140}{2{\cdot}44 \times 1000} = 0{\cdot}29 m^3/s$$

$$\text{That is: } \frac{0{\cdot}29\ (m^3/s) \times 3600\ (s/hour)}{50\ (m^3 \text{ room volume})} = 21 \text{ air changes per hour.}$$

*1 kilogramme per hour = 278 milligrams per second.

Table 5.1

Some examples of Recommended maximum gas concentration

Gas or vapour	Maximum ppm* Daily average	Short period
Phosgene	0·05	0·05
Bromine	0·1	0·3
Iodine	0·1	0·1
Ozone	0·1	0·3
Chlorine	1	3
Fluorine	1	3
Nitrobenzene	1	3
Formaldehyde	2	2
Aniline	5	10
Hydrogen chloride	5	5
Nitrogen dioxide	5	5
Phenol	5	10
Sulphur dioxide	5	10
Acetic acid	10	20
Carbon tetrachloride	10	20
Hydrogen cyanide	10	20
Hydrogen sulphide	10	20
Naphthalene	10	20
Ammonia	25	37
Benzene	25	25
Chloroform	25	25
Nitric oxide	25	37
Carbon monoxide	50	75
Amyl acetate	100	150
Toluene	100	150
Trichloroethylene	100	150
Xylene	100	150
Hexone	100	150
Methyl alcohol	200	250
Pentane	500	750
Acetone	1000	1250
Ethyl alcohol	1000	1250
Propane	1000	1250
Carbon dioxide	5000	6250

*ppm: Parts of gas or vapour per million parts of air by volume.

5.1.3 Fume cupboards

General ventilation is not recommended for the more highly toxic gases (such as those listed at less than *50* ppm). Nor will it be economic if an excessive volume flow of air is called for, requiring to be heated in winter. Local extraction is the answer, the sources being preferably enclosed within individual hoods on the lines discussed in Section *5.4* on fume removal.

For general purpose laboratory use the fume cupboard is usual, and Fig. *5.1.* illustrates the principles of construction. With the sash type door fully open for access, room air should enter at an average velocity of 0.25 to 0.75m/s according to the toxicity of the fumes handled. The sash should not close fully, 25mm opening being left at the bottom to maintain airflow, through the cupboard.

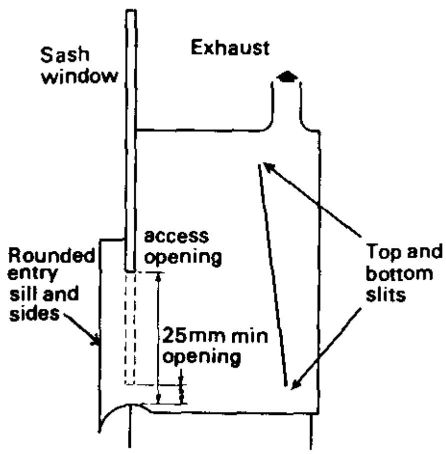


Fig. 5.1 Fume cupboard

Extraction is through top and bottom slits in a baffle leading to the exhaust duct. This ensures the extraction of heavy gases collecting towards the floor of the cupboard, as well as light gases at the top.

When dangerous micro-organisms or highly radio-active materials have to be handled, protection from contact must be as complete as possible. Glove-boxes with air-locked entry arrangements for specimens and high-efficiency filters monitored for pressure drop in the exhaust ducts are used, see Fig. 5.2.

5.1.4 Dust hazards

Certain dusts encountered in mining and quarrying cause pneumoconiosis, a serious group of diseases in which the lungs are progressively damaged, over years of exposure, by the formation of scar tissue. Some of these are listed m Table 5.2, with recommended maximum concentrations either on a weight, or on a number of particles, basis. Many chemical and metallurgical dusts and fumes are highly toxic, and will be subject to precautions well known in the industries concerned. Ordinary dust derived from the soil and from combustion is, however, relatively harmless, but should be kept within the industrial nuisance limits quoted in the interests of comfort and safety (e.g. good visibility).

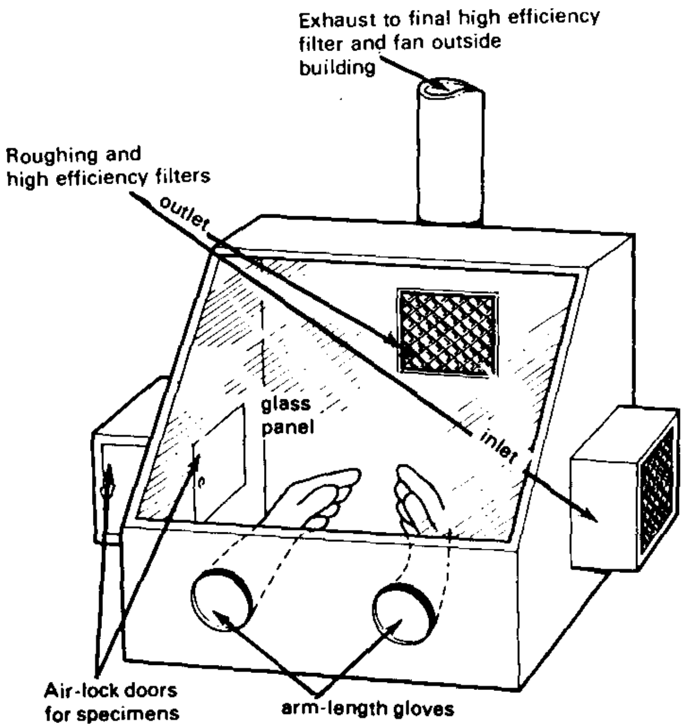


Fig. 5.2 Glove box.

Dust clouds can also explode, but only in concentrations exceeding the 10mg/m^3 quoted as the maximum for industrial atmospheres. Aluminium and magnesium powders cause particularly violent explosions, while organic dusts such as starch are easily ignited. Good housekeeping is important, with accumulations of dust in odd corners rigorously excluded. Otherwise a small local explosion may raise a dense dust cloud resulting in a massively destructive blast.

General ventilation is not effective for the control of industrial dust. Local extraction at source will be required, followed by dust collectors to prevent dust nuisance outside the plant. These are discussed later in this chapter.

Table 5.2

Recommended limits for mineral dusts

Substance	Million particles/m^3	mg/m^3
Asbestos or talc (fibres 5 μm long)	5	—
Cristobalite silica	50	0·05
Quartz and fused silica (100%)	100	0·1
Graphite (natural)	500	—
Mica, soapstone, amorphous silica	700	—
Coal dust (bituminous)	—	2
General limit for industrial nuisance	1000	10

5.2 Explosion Hazards

5.2.1 Dilution ventilation for explosive gases

Most explosive gases and vapours are toxic. To form an explosive mixture with air they must be present in a proportion rarely less than 1 % by volume, that is 10,000 ppm. It follows that workshop atmospheres which are satisfactory for health are unlikely to present an explosion risk.

However, local extraction systems will produce much higher concentrations of pollutant in the exhaust ducts. Sufficient air must always be extracted along with the gas or vapour removed to dilute the latter to a safe concentration. Table *5.3* lists some dangerous substances giving the lower limit of the range of flammable (and therefore explosive) concentrations.

There is also an upper limit, but it is never safe to operate an extraction system on the assumption that high non-explosive concentrations can be maintained.

The ventilation rate for dilution can be calculated by formula *51,* the same as for toxic gases. It is good practice to ensure that the peak local concentration of pollutant does not exceed *25%* of the lower flammable limit. This means that the factor of safety, SF, based on total airflow and gross gas or vapour production rates, must substantially exceed 4. For example, SF = 10 is the lowest value recommended for batch ovens, even when precautions are taken to ensure air circulation round the heated stock.

5.2.2. Example

A batch oven dries painted sheet steel products at a temperature of *180°C. Xylol* solvent is driven off at an estimated rate of 1.*5* kg per hour (= 420 mg/s). The lower explosive limit for xylol is 10,000 ppm (Table *5.3)* and the density of its vapour is *3.8* kg/m^3 at room temperature.

$$Q = \frac{10\ (\text{SF}) \times 420\ (\text{mg/s})}{3{\cdot}8\ (\text{kg/m}^3) \times 10{,}000\ (\text{ppm})} = 0{\cdot}11\,\text{m}^3/\text{s}$$

As both the air and the vapour will leave the oven at 180°C, the flow rate to be produced in the extract duct (proportional to absolute temperature, i.e. 273 + t °C) will be:

$$\frac{273 + 180}{273 + 20} \times 0{\cdot}11 = 0{\cdot}17\text{m}^3/\text{s}$$

5.2.3 Properties of flammable gases and vapours

When supplied at a limited rate, as through a burner, and ignited, gases and vapours will continue to burn at the point where they are diluted with sufficient air to enter the flammable concentration range. The danger of fire can be judged by the characteristic properties given in Table 5.3, thus:

(a) The upper and lower explosive limits, or limits of flammability with air. A wide range, e.g. the 4% to 74% of hydrogen, indicates particular danger of ignition and explosive propagation. At the other extreme, methyl bromide with 13.5% to 14.5% is practically noninflammable, and is, in fact, successfully used as a fire-extinguishing medium.

(b) The *ignition temperature*. This is the minimum temperature at which a flammable mixture of air and gas or vapour will ignite. It is chiefly important when it is abnormally low, as in the case of carbon disulphide. This ignites at about 120°C, and may be fired by a steam pipe, or even by the hot surface of an electric motor.

(c) The flash point applies to liquids. It is the lowest temperature at which sufficient vapour is given off to cause explosive burning when exposed to a means of ignition. It is a quantity which is dependent to a considerable extent on the method of test; a widely-used rating based on the closed cup method is listed in Table 5.3. Liquids with closed-cup flash points below 21 °C should be regarded as highly inflammable.

5.2.4 Control of explosive atmospheres

Many industrial processes, including such common activities as cellulose-spraying and petrol-dispensing, are carried on in spite of the increased fire hazard associated with the materials handled. Precautions must then be taken to limit the chance of fire or explosion to a very low level.

This is best achieved when the probability of disaster is the product of two probabilities, each in itself small. Thus we may ensure:

(a) That sources of ignition are excluded from the hazardous area, and can arise only by accident or human error.

(b) That flammable concentrations of the hazardous gas are excluded by design, and can arise only in abnormal circumstances, e.g. failure of the ventilation system.

Thus if the chance of a flammable concentration is 1 in 1,000 and the accidental ignition hazards five per annum, then a fire will occur once in about 200 years - or longer since the two abnormal happenings must coincide in space as well as time.

Table 5.3
Classification of industrial gases for explosion hazard

Group (IEC 79)	Gas vapour or volatile liquid	Flammable limits ppm by volume Lower	Flammable limits ppm by volume Upper	Flash point closed cup
I	Coal-mining industry			
	Methane	50,000	150,000	Gas
IIA	Petroleum industry			
	Propane	21,000	94,000	Gas
	Butane	18,000	84,000	Gas
	Pentane	14,000	78,000	− 40°C
	Hexane	12,000	74,000	− 22°C
	Heptane	11,000	67,000	− 4°C
	iso-Octane	9,000	32,000	13°C
	Decane	7,000	26,000	46°C
	Coal-Tar products			
	Benzene (benzol)	14,000	21,000	− 11°C
	Xylene (xylol)	10,000	60,000	17°C
	Cyclohexane	12,000	78,000	− 17°C
	Industrial solvents			
	Acetone	25,000	128,000	− 18°C
	Ethyl methyl ketone	18,000	95,000	− 1°C
	Methyl acetate	31,000	155,000	− 10°C
	Ethyl acetate	22,000	114,000	− 4°C
	n-Propyl acetate	18,000	80,000	6°C
	n-Butyl acetate	14,000	76,000	22°C
	Amyl acetate	11,000	—	25°C
	Chloroethylene	62,000	159,000	13°C
	Methanol (methyl alcohol)	67,000	265,000	12°C
	Ethanol (ethyl alcohol)	33,000	190,000	13°C
	iso-Butanol (butyl alcohol)	—	—	—
	n-Butanol	14 000	113,000	29°C
	Amyl alcohol	12,000	—	43°C
	Ethyl nitrite F	30,000	500,000	− 35°C
	Other industrial gases			
	Ammonia	155,000	270,000	Gas
	Blast furnace gas	—	—	Gas
	Methane (with > 10% hydrogen)	—	—	Gas
	Carbon monoxide	125,000	742,000	Gas

F: Ignition temperature less than 200°C.

Table 5.3—*continued*

Classification of industrial gases for explosion hazard

Group (IEC 79)	Gas vapour or volatile liquid		Flammable limits ppm by volume Lower	Upper	Flash point closed cup
IIB	Buta1, 3diene		20,000	115,000	Gas
	Ethylene		27,000	286,000	Gas
	Diethyl ether	F	18,000	365,000	− 45°C
	Ethylene oxide		30,000	800,000	Gas
	Town Gas (coal gas)		—	—	Gas
IIC	Hydrogen		40,000	742,000	Gas
Excluded	Acetylene		25,000	800,000	Gas
	Acetaldehyde	F	40,000	570,000	− 27°C
	Carbon disulphide	F	12,000	500,000	− 30°C
	Ethers (except diethyl ether	F	—	—	—
	Water gas		60,000	700,000	Gas
	Natural gas		40,000	140,000	Gas
	Petrol (gasolene)		13,000	60,000	− 46°C
	Kerosene		—	—	10°C

F: Ignition temperature less than 200°C.

In keeping with this philosophy hazardous atmospheres have been classified into three types or Zones. In Zone 0 the probability of a flammable mixture being present approaches 100%, in Zone 2 it is very small; and in Zone 1 intermediate between these extremes.

5.2.5 Electrical apparatus for explosive atmospheres

The International Electrotechnical Commission has published as IEC 79 guidance for the safe use of electric motors, control gear, lighting equipment, etc. in flammable atmospheres. This is based on the experience and research of various national bodies, and in most countries corresponding specifications exist - in the UK BS 4683. These specifications are generally followed by the responsible authorities in approving particular industrial installations and apparatus, though judgement and discretion remain necessary. Both the apparatus and the environment require study, and an outline of the classifications adopted appears on the next page.

In Zone 0 areas it is preferable to exclude all electrical apparatus except light current intrinsically safe equipment classed Ex ia. This cannot draw enough power from the electrical supply to overheat or produce an incendive spark (i.e. one with more than 0001 joule energy). When motors or other electrical equipment must be used in a continuously explosive atmosphere, Ex p, pressurised, machines can be considered, though, with all the necessary precautions, they tend to be excessively costly except in the largest sizes.

Outline of IEC 79—1971—

Hazard classification for electrical apparatus

Classification by frequency of hazard

Zone 0: Where flammable gas, vapour or volatile liquid is continuously present in concentrations between the lower and upper limits of flammability.

Zone 1: Where a flammable concentration may occur at times during normal operation.

Zone 2: Where a flammable concentration is possible only under abnormal or fault conditions of short duration.

Classification according to industry and hazardous gas

Group I: Methane gas in the coal-mining industry.

Group IIA,
Group IIB: Other industrial gases and vapours, grouped in order of increasing severity of explosion hazard.

Group IIC: See Table 5.3.

Classification of apparatus according to construction and design

Ex N: Normal apparatus with special attention to certain critical features. Zone 2 only.

Ex e: Non-sparking, "increased safety" apparatus with design features to specified limits, and with protection against overheating. Zones 1* and 2.

Ex d: Apparatus with robust "Flameproof" enclosure capable of withstanding an internal explosion without igniting flammable gas outside. Zones 1 and 2.

Ex p: Apparatus fed with air or inert gas to exclude hazardous gas from the enclosure. "Pressurised" at 50 Pa minimum with automatic disconnection on failure. Zones 0*, 1 and 2.

Ex ia: Instruments, control or communications equipment certified for "intrinsic safety", that is, as incapable of overheating or incendive sparking. Zones 0, 1 and 2.

Classification of apparatus according to maximum surface temperature

Class:	T1	T2	T3	T4	T5	T6
Maximum °C:	450°C	300°C	200°C	135°C	100°C	85°C

Example of apparatus marking: Groups IIA, IIB Ex d T5.

*Use in this Zone subject to specific approval of installation.

Ventilation may provide a solution by reducing a Zone 0 to a Zone 1 or 2 danger level. A plantroom or cubicle may be so treated, reducing the level of machine construction required to Ex d or Ex e. The approach for Ex e certification is to make sure that the machine is so designed and controlled that failure causing sparking or overheating is very unlikely. With Ex d protection on the other hand it is assumed that the most explosive mixture of the gases in question has penetrated the enclosure and been ignited. It is then ensured (by type test) that the enclosure will withstand the resulting pressure, which may reach ten atmospheres; also that the hot gases escaping through the specified spigot joints, gaps around shafts, etc. will be sufficiently cooled not to ignite the explosive mixture outside.

5.2.6 Fans for hazardous atmospheres

Fans may be driven by Ex d, Ex a or Ex n electric motors and controllers. If the motor is outside the fan casing, as with centrifugal or bifurcated axial types, Zone 1 conditions will certainly be met. To secure the more relaxed Zone 2 rating it is necessary to ensure that the pressure within the fan casing is below atmospheric pressure; consideration must also be given to the possibility of flammable gases coming into contact with the motor by leakage or otherwise when the fan is stopped and restarted.

Axial fans with Ex d flameproof motors in the air stream have been widely used in the past, but the authorities now look at such installations more critically. There is no doubt that Zone 1 conditions will be met with proper dilution of the hazardous fumes. In such applications as cellulose spray extraction, however, there is concern that overspray deposits on the motor carcase might cause overheating and consequent ignition. The fourth hazard classification-maximum surface temperature - is relevant here, although at the time of writing it is not backed by much data on the ignition temperatures of different gas or vapour mixtures. Motor carcase temperatures will generally be within 100°C by design - Class T5. Therefore, if adequate precautions are taken against overheating, only gases marked F of those in Table 5.3 should be suspect.

Another factor to be considered is the danger of incendive sparking if metals or stones are struck together in an explosive atmosphere. A a number of explosions in coal mines have been traced to this cause and equipment in steel or cast iron is now specified. Magnesium, and to a lesser extent some aluminium, alloys have been found to produce more energetic sparks, more likely to ignite the methane in the atmosphere of fiery mines.

5.3 Air Cleaning

The atmosphere out-of-doors carries in suspension a wide range of solid particles classified as dusts, fumes or smokes, as well as liquid droplets in the form of mist or rain. Typical quantities of solid matter lie within the following ranges:

Rural and suburban areas	0.05 to 0-5mg/m^3
Metropolitan districts	0.1 to 1 mg/m^3
Industrial districts	0.2 to 5 mg/m^3

Temporary local concentrations of 100 or 1000mg/m^3 are common in mining, quarrying or dusty factory operations, but they should not be allowed to persist. "Smokeless zone" legislation has had considerable success in cleansing the air of cities.

Even in clean atmospheres the air supply to public buildings is generally filtered. This makes only a minor contribution to health and comfort, since our nasal passages provide excellent filtration for the air we breathe. The value is economic, since the cost saved in less frequent cleaning and redecorating far outweighs the cost of servicing the filters. Much higher standards of filtration can be provided for the clean rooms and sterile rooms needed for sensitive industrial and laboratory operations.

Attention must also be paid to the possibility of nuisance when contaminated air is discharged from a factory. A high chimney may suffice, relying on the wind to spread the contaminant over many square kilometres of countryside. Nevertheless, anti-pollution legislation is always tightening standards, and when planning new plant it is wise at least to allow space for the insertion of air cleaning equipment at a later date, if the slightest doubt exists.

5.3.1 Classification of dust

The upper part of Fig. 5.3 gives a general impression of the size range of the terms commonly used to describe particulate matter. Fumes are solid particles formed when the vapours of substances normally solid condense in the atmosphere. Size is the main factor determining the behaviour of the particles in air, and the three linear charts of Fig. 5.4 result from exact calculations for spherical particles. Of course, actual particles are not so conveniently shaped, nor are they graded into species of constant size. Nevertheless the extremely rapid change of each quantity with size gives significance to even the roughest estimation.

Dust concentrations can be estimated by weighing filter papers through which measured volumes of the air have been passed. Numbers and sizes of particles are estimated by examining samples captured on microscope slides. Statistical analysis can then give the research worker separate distributions of number, weight or surface area by size. Each may then be expressed in terms of equivalent spherical particles.

The terminal velocity reached by particles in free-fall through the atmosphere determines the method to be used for their removal. In steady fall the atmospheric drag force holding the particle back will equal the downward force of gravity on it. Several distinct flow regimes can be distinguished.

Turbulent flow applies to particles larger than 600μm or so, when the drag force increases with the square of the velocity.

Streamline flow is predominant in the 0.1 to 100 μm range. The drag force is viscous and proportional to velocity.

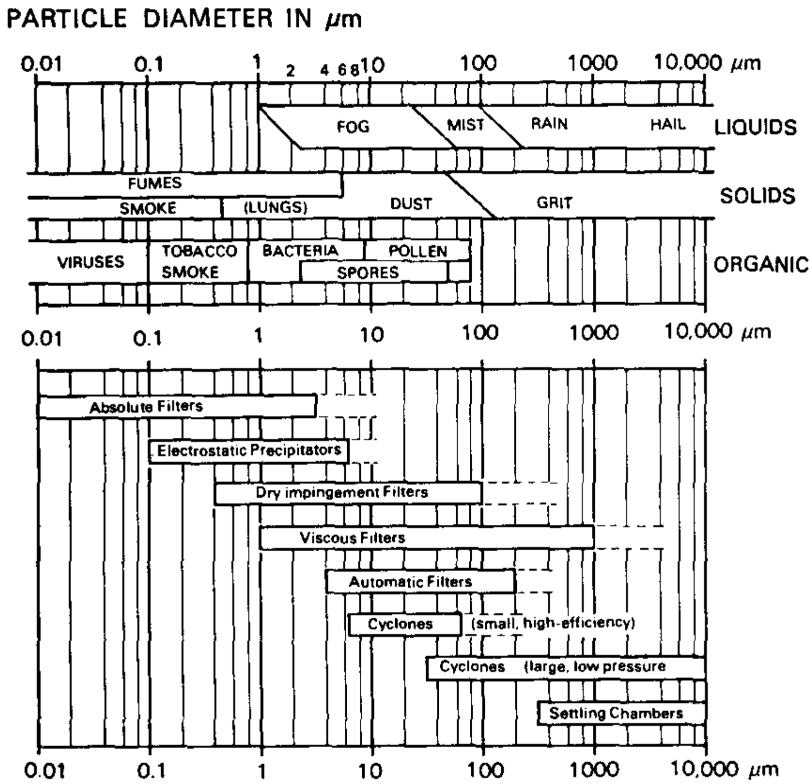
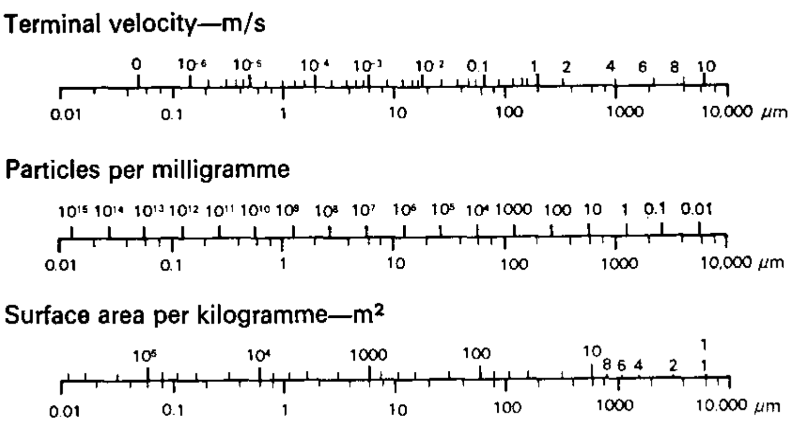


Fig. 5.3 Classification of particles and filters.



Particles assumed to be of unit specific gravity (ρ = 1000 kg/m^3).
For other values of ρ multiply particles per mg and surface area per kg by $(1000/\rho)$
Multiply terminal velocity by $(\rho/1000)$ for particles smaller than 50μm and by $\sqrt{\rho/1000}$ when larger than 1 mm.

Fig. 5.4 Data for spherical particles in standard air.

G

Transitional flow occurs between the turbulent and streamline forms, the drag force being higher than in either.

Brownian movement exists below 0.1μm. Such small particles partake of the random motion of the air molecules around them, with no net tendency to move downwards.

The air itself is always in motion, and when its velocity is well above the terminal velocity of a particle, that particle will be carried with the air. Thus particles in the 1 to 100μm range are classed as temporary impurities - quite mild air movement will keep them in suspension for a substantial period. Smaller particles move so slowly under gravity that they are likely to remain as permanent impurities. Larger particles will settle more or less rapidly, but can be transported, and thereby removed, by deliberate air currents of sufficient magnitude.

5.3.2 Performance of air cleaners

There is little standardisation in the design of air filters and dust collectors so that manufacturers' data should be studied and their advice sought in case of doubt. The *efficiency* of an air cleaner is that percentage (by weight) of the dust in the incoming air which is captured by the cleaner, the remainder passing through to the clean air outlet.

As the efficiency is very much dependent on the size of the particles, rating tests are carried out with certain standardised dusts; the size distributions of some of these are illustrated in Fig. 5.5. Efficiencies from 70% or 80% to 99.99% are available, but an unnecessarily high performance attracts disproportionately high maintenance and first costs.

Air cleaners capture particles by a variety of mechanisms which may be summarised as follows:

Gravity and centrifugal force effective in transporting large heavy particles to a collecting point.

Screening of particles which are too large to pass through the air passages in a filter medium.

Adhesion to sticky or wetted surfaces off which the larger particles would otherwise rebound.

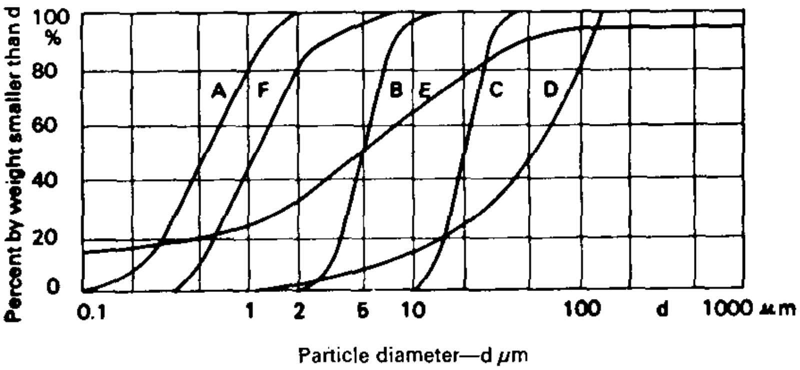
Impingement on dry surfaces by particles small enough to be retained by the surface forces associated with intimate contact.

Adsorption of gas or vapour molecules which occurs, selectively, by condensation on the extended internal surface of certain microporous substances.

5.3.3 Classification of air cleaners

Fig. 5.3 shows the range of particle size for which each of the main types is best adapted.

Settling chambers. While settlement is relied on to dispose of dust clouds in the natural surroundings of mines and quarries, settling chambers are rarely used. The size of chamber is generally too great in relation to the volume flow to be handled.



A: UK BS 2831 No. 1, methylene blue aerosol (BS 3928 NaCl similar).
B: UK BS 2831 No. 2, graded aluminium oxide.
C: UK BS 2831 No. 3, graded aluminium oxide.
D: UK BS 1701, crushed silicon (simulating road dust).
E: USA ASHRAE, carbon black, silicon dust and cotton lint.
F: Germany, Staubferschungs lust, silicon dust.

Fig. 5.5 Standard test dusts for filter rating.

The cyclone is a dynamic settling chamber in which a stable vortex motion is established by a high velocity jet of dusty air entering tangentially. Centrifugal force drives the heavier particles to the wall. There they travel round and downwards into a hopper, as illustrated in Fig. 5.6, while the cleaned air emerges upwards at the centre.

Fans should not be connected to a cyclone outlet without consulting the fan makers. The rapidly swirling air stream from the cyclone will greatly increase the power loading of a fan rotating in the opposite direction. Conversely the performance of a fan rotating in the same direction will be greatly reduced.

Cyclones are used in the conveying and handling of grain and flour, in collecting the dust from grinding, buffing, sanding or crushing processes, and for recovering re-usable industrial dust. Pressure drop ranges from 200 to 500 Pa.

High-efficiency cyclones. The centrifugal force on a particle of mass m moving in a circle of radius r is $m\,v^2/r$. Air cleaners comprising a multiplicity of small cyclones operating in parallel exploit this fact. By reducing r the force is increased, and much smaller particles can be extracted. Finer grades of chemical, pharmaceutical and food products are handled, and pulverised fuel fly-ash can be extracted from high temperature flue gases. Pressure drop ranges from 500 to 1500 Pa.

Wet collectors may take the form of cyclones in which the collecting surface is wetted with a continuous water spray. These will be more efficient than the corresponding dry cyclones, but power must be supplied to pump the water, and the water itself will usually need to be filtered before discharge.

Spray chambers or towers containing wetted surfaces designed to deflect the air repeatedly will extract dust. They are widely used for the same industrial purposes as cyclones - particularly in the ceramics industry. They may also reduce fire and explosion risk.

The washers and scrubbers used in air conditioning plants are primarily for humidity control, and should not be relied on to remove fine atmospheric dust.

Viscous filters contain panels of fabric or fibrous mat, often in plastic, glass or metal strands, and coated with an appropriate oily liquid. The oil must have good wetting power and be odourless, non-volatile and noninflammable. The tortuous path of the air through the panel causes dust particles to impinge on the strands where they adhere to the oily surface.

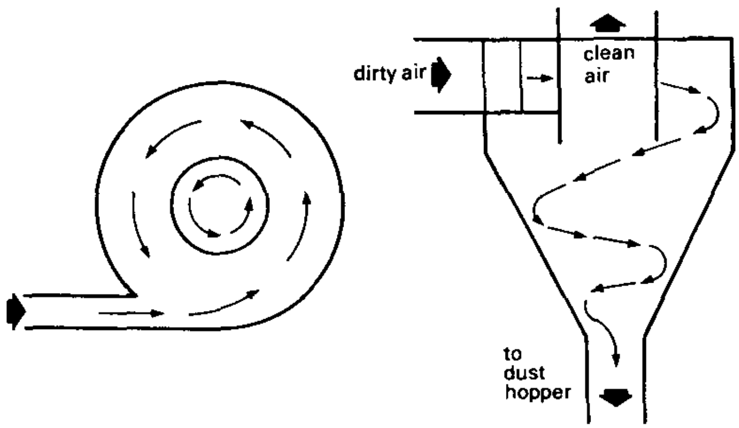


Fig. 5.6 Cyclone.

The strands of viscous filters are generally relatively coarse and open and most are at their best with particles in the4to 10Nmrange. The panels are usually washable and can be re-oiled and re-used repeatedly. Automatic filters can, therefore, be constructed from endless bands of filter material continuously circulating through a bath which serves for both washing and re-oiling.

Viscous filters find their major application in the cleansing of the air supply to buildings. They are particularly useful in cities with badly polluted atmospheres, as they can deal economically with heavy dust loads. Pressure drop is low - 100 to 150 Pa.

Dry filters are also made in panels or rolls of fabric or fibrous material, natural or synthetic.

For air conditioning supply systems they have finer, more closely packed strands and will remove most of the carbonaceous dusts in the 0.5 to 5m range responsible for dirtying fabrics and decorations. The material is generally thrown away when dirty, but can be cleaned in special solutions.

To maintain a moderate pressure drop-25 to 60 Pa when new, 100 to 250 Pa when soiled - the material is generally folded into a V-formation, see Fig. 5.7. In this way the customary face velocity for all air conditioning applications – 1.25 to 2.5m/s - can be maintained with a much lower air velocity through the dense material.

Fabric arresters are dry filters made in bag or stocking form for the capture of industrial dust. The purpose may be recovery of valuable material or prevention of dust nuisance. The fabric is chosen to suit the dust and, unlike cyclones, can be effective on particles of less than 1 μm.
The material is removed periodically by reverse blowing or shaking most of it builds up as a dust layer on the upstream side of the fabric. The method is used for textile fibres, chemicals, ceramics, grain, cement and woodworking dust. To limit the space occupied pressure drops are high - 500 to 1500 Pa at 0.01 to 0.05m/s through the fabric.

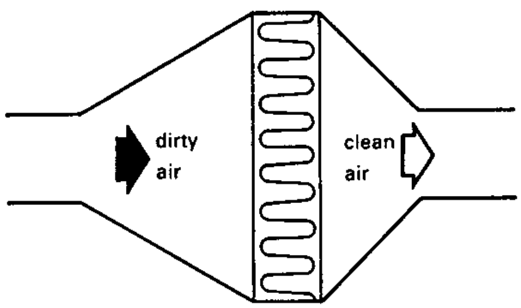


Fig. 5.7 Multiple vee fabric filter.

Electrostatic-precipitators as used in air conditioning are two-stage devices. The incoming air is first passed through an ioniser. The assymetric electric field between earthed rods and fine wires held at a positive potential of up to 13,000 volts gives to each dirt particle a positive charge. In the second stage the air with its load of charged particles passes between plates alternately earthed and held at + 6000 volts.

Each particle is repelled by the positive and attracted to the (earthed) negative plate, to which it becomes attached. The plates are periodically washed to remove the dirt. This principle ensures that the smallest particles will be removed, but large particles can cause electrical breakdown and those above 5μm should have been first removed by a viscous filter or otherwise.

The first cost of electrostatic precipitators is high although they are cheap to maintain. They come into their own in very large sizes to control industrial pollution. Such applications include the cleansing of flue gas from power stations and blast furnaces, or of contaminated air discharged from large chemical plants. For these installations ionisation and collection are combined in one stage with wires at 30,000 to 75,000 volts (negative) between earthed plates. The pressure drop of all electrostatic precipitators is very low, and the expense is associated with the power pack to produce the high DC voltages and the safety precautions these voltages necessitate.

Absolute filters are made from a dry filter medium akin to paper or felt. They are made possible by the very small diameters in which glass and asbestos fibre can now be produced. The pores are so fine that the diffusion or Brownian movements of particles as small as 0.01 μm will cause impingement and retention. The medium can be quite thin so that, with deep corrugations, maximum face velocities of 1.25 to 2.5m/s can be reached. Pressure drop is likely to reach 250 to 500 Pa, and soiled filter medium must be thrown away. If the filter is used to capture bacteria only it can be sterilised and re-used several times.

Such filters are used in the air supply to clean rooms or sterile rooms of the highest standard. Operating theatres, pharmaceutical, photographic and biological laboratories and the trapping of radio-active dust may be mentioned. If necessary an efficiency of 99.995% can be reached with methylene blue test dust. The cost is high and pre-filters are necessary to remove coarse dust and lengthen the life of the absolute medium.

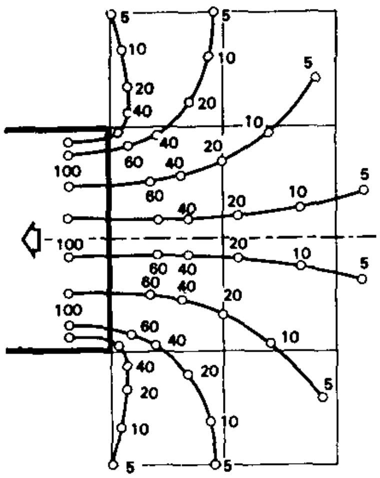
Activated carbon filters can be used for the adsorption of gases and vapours. The carbon is produced in granulated form, packed into panels through which the room air is circulated and recirculated. The granules need not be tightly packed, and many of the large molecules characteristic of odours are captured in an active way reminiscent of the capillarity of liquids. Each carbon granule is a structure with a large surface area formed on a multitude of internal pores. The medium has a long life since it can absorb up to half its own weight of gas or vapour, which can then be driven off by heating and the carbon used again. 95% efficiency is obtainable and the pressure drop can be constant in clean air at about 60 Pa. These filters are perhaps most commonly found in cooker hoods for domestic kitchens.

5.4 Dust and Fume Collection

The importance of local extraction at source has been mentioned several times already in this chapter. The requirement is to see that the great majority of grit, dust or fume particles are transported from the point where they are generated into a duct along which they are conveyed to a discharge or collection point.

The critical phase is the capture of the particle by an air stream sufficiently strong to direct and transport the particle towards the duct. The air direction must be proof against excessive deflection by draughts; 0.5m/s is usually enough to secure this. 0.5m/s is also sufficiently above the terminal velocity given in Fig. 5.4 for the capture of most dusts. Industrial grit, however, is usually too heavy for air capture; hoods must be placed in line with the expected flight from grinding wheels and similar sources. Similarly the natural flow of air and vapour upwards from boiling pans or other heated sources must be respected in the siting and sizing of a hood above.

The velocity of the air towards a suction point falls rapidly with the distance away-in fact with the square of the distance when this is more than about twice the diameter of the suction opening. The distribution of velocity close to circular suction openings is shown in Fig. 5.8, the figures being air velocities in per cent of average duct velocity. Wide flanges will improve the inward velocities in their neighbourhood, but will make little difference to the approach velocities along the centre line.



Circular Suction Entry

Fig. 5.8 Flow lines into ducts.

5.4.1 Hoods and canopies

The first rule for design is that the hood should enclose the source of fumes or grit just as completely and closely as is possible without impeding production operations. Maintenance operations should be allowed for only by making the hood, or part of it, removable. Not only will such a hood intercept and capture the inevitable stray particles: it will also need less volume flow to maintain the required approach and capture velocities.

A number of examples are shown in Figs. 5.9 to 5.13. In each case a nominal inlet area is identified together with the average air velocity that should be maintained over this area. Multiplying the two together gives the m^3/s to be extracted Where a range of velocities is indicated, the higher values are for higher densities of grit, high cutting velocities, and greater rates of particle production. Over boiling pans, kettles or tanks the air volume flow must exceed the highest possible rate of steam or vapour production.

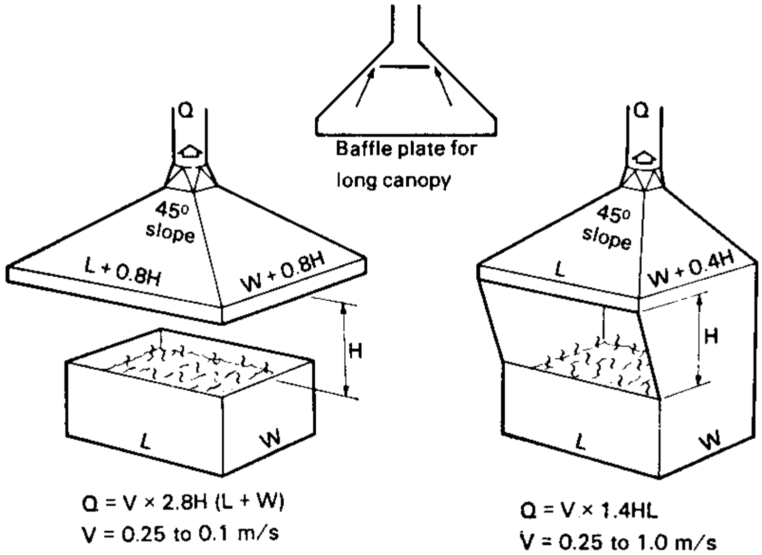


Fig. 5.9 Canopies or hoods for open tanks with hot or light fumes.

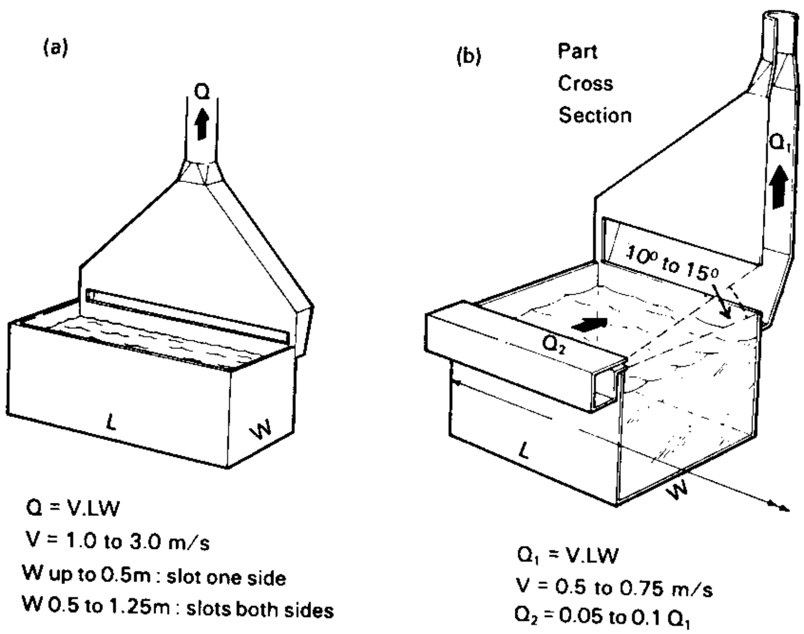
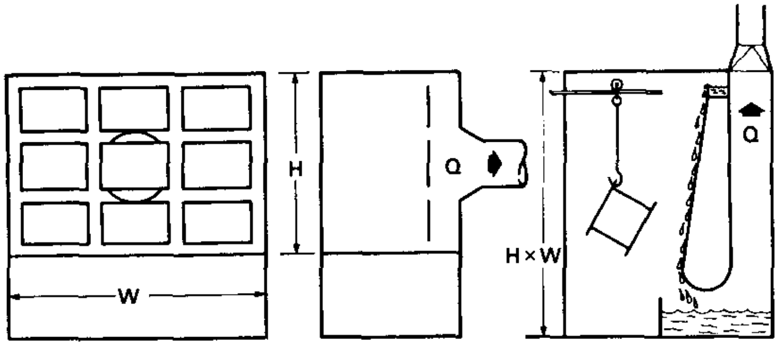


Fig. 5.10 Lateral exhaust systems for open tanks with cool or heavy fumes.



(a) Dry baffle plate type.
Baffle plate area: 0.6 WH.
Q = V.WH

(b) Walk-in, water wash type.
Q = V.WH

For both types V ranges from 1.0 m/s for small booths to 0.4 m/s for large, deep booths.

Fig. 5.11 Paint-spraying booths.

Very large canopies must be subdivided with separate exhaust outlets to make sure that one end is not starved. Alternatively, a baffle plate as Fig. 5.9 may be used to ensure even intake distribution with a single exhaust duct. Fumes from cold tanks can be removed by a full length extract slot as Fig. 5.10a. Volume flow ventilation and heat loss may be reduced by a supply slot providing an air curtain across the tank surface as Fig. 5.10b, while heat loss from the tank is often limited by a layer of plastic balls floating on the surface.

Hood width = wheel width + 60mm
Q should be $0{\cdot}5m^3/s$ per metre of wheel diameter, minimum

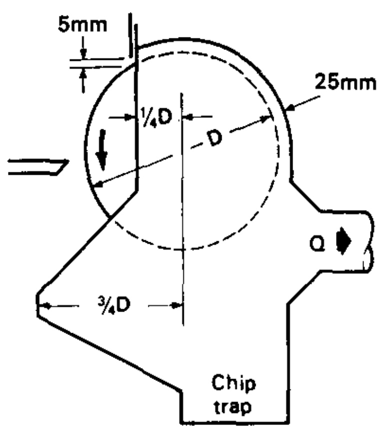


Fig. 5.12 Hood for grinding wheel.

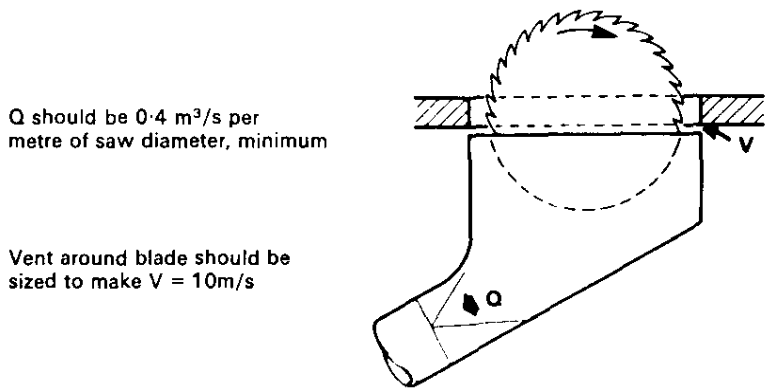


Fig. 5.13 Hood for circular saw.

5.4.2 Dust transport

When the dust-laden air has been collected transport must be assured by an adequate air velocity in the exhaust duct. Table 5.4 gives some recommended conveying velocities.

Table 5.4
Air velocities for dust transport

Type of dust or vapour	Branch duct velocity, m/s
Paint and varnish fumes	8–10
Textile lint and sawdust (dry)	10
Asbestos and limestone dust	13
Metal fumes and dusts	15
Grain and flour	18–20
Grinding, shot- and sand-blasting	18–22
Pulverised coal, stonecutting	20
Wood chips and shavings	20–25
Lead dust	28

CHAPTER 6

Air flow and pressure in ducted systems

The air flow rates and velocities required for adequate ventilation have been discussed in the first five chapters. This chapter is concerned with the fan pressure needed to circulate the flow.

6.1 Definitions of Pressure

The opening discussion of the meaning of pressure may seem too detailed and too elementary to many. However, the exact meaning of the terms used is vital, and experience has shown that misunderstanding can easily arise and persist. A summary of the facts will be found in Table 6.1.

6.1.1 Pressure, absolute and atmospheric

It will be accepted that atmospheric air experiences a pressure, caused by the weight of the air above. This is the barometric pressure p_o and is quite substantial, amounting to about 100,000 pascals. In the units used by meterologists 100,000 Pa is 1,000 millibars, since 1 mb = 100 Pa exactly; older barometers will still read the gravitational equivalent - 750mm of mercury.

If air is blown into a child's balloon it will be commonly described as "under pressure" - a pressure applied by the tension in the rubber skin of

p_o p_a

the balloon. The air inside the balloon does experience a greater pressure than the atmospheric air outside, though the difference will not be very great-perhaps 105,000 Pa compared with 100,000 Pa in the atmosphere.

The term "pressure" without qualification describes both p_o = 100,000 Pa outside the balloon and p_a = 105,000 Pa inside - the air being supposed to be still in both cases. If there is any danger of misunderstanding, this form of pressure can be particularised as *absolute pressure*.

$$p_s = p_a - p_o$$

For the purposes of fan engineering it is the difference between the absolute pressure at the point under consideration and the atmospheric pressure which is of importance. This is called the *static pressure*, and in the above example the static pressure of the air in the balloon would be 105,000-100,000 = 5,000 Pa. Static pressure is reckoned positive when the absolute pressure is above atmospheric pressure, negative when below.

The main advantage of working with static pressures rather than absolute pressures is that the natural variations in atmospheric pressure are compensated. Thus if the barometric pressure in our example had been p_o = 960 mb instead of 1,000 mb we should have found that the same effort would result in blowing up the balloon to p_a = 101,000 Pa, when the tension in the skin would have also been the same. This corresponds to an unchanged static pressure of 101,000-96,000 = 5,000 Pa.

Calculations in terms of static pressure will be independent of atmospheric pressure only so long as the static pressure is relatively small.

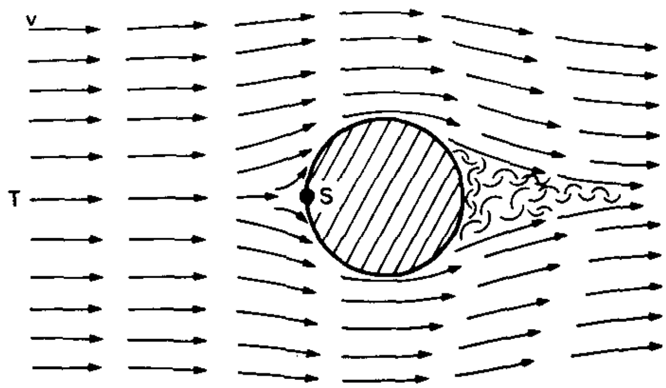
The air can then be treated as an incompressible fluid (except for density determination) in which case the laws of motion are relatively simple. Errors due to compressibility are just perceptible at 2,000 Pa static pressure and become significant above 5,000 Pa. In this book we shall deal with the influence of compressibility on fan testing where we want precise results. Otherwise we shall assume that the air is incompressible.

6.1.3 Velocity pressure, p_v

$$p_v = \tfrac{1}{2}\rho v^2$$

It is common experience that the wind exerts a force on any object standing in its path. This is mainly because the absolute or static pressures on the windward side are greater, on average, than those on the leeward side. The wind is not, of course, stopped in any final sense - it flows round the body. There is, however, always one point on the surface at which the air is actually brought to rest, without flowing either to right or left, as illustrated in Fig. 6.1. At this point, called the *stagnation point*, the pressure at S (both absolute and static) will have risen by an amount p_v above the pressure in the undisturbed air stream. p_v in Pa is exactly related to the air density, ρ kg/m^3, and velocity, v m/s, in the undisturbed stream by the relation $p_v = \frac{1}{2}\rho v^2$.

p_v is called the *velocity pressure* and is an important quantity to which all the pressure and drag effects of a moving airstream can be related. It is the velocity of the body relative to the undisturbed air which counts. It does not matter whether moving air is slowed down from v to zero at the stagnation point on a stationary body, or whether still air is accelerated from zero to v at the stagnation point on a moving body. In both cases the change in velocity is v (strictly -v or +v) and in both cases the rise in pressure is the velocity pressure, $\frac{1}{2}\rho v^2$, which is always positive.



at T: $p_s = 0 \quad p_v = \frac{1}{2}\rho v^2 \quad p_t = \frac{1}{2}\rho v^2$

at S: $p_s = \frac{1}{2}\rho v^2 \quad p_v = 0 \quad p_t = \frac{1}{2}\rho v^2$

Fig. 6.1 Air flow past an obstacle.

6.1.4`Total pressure, p,

$$p_t = p_s + p_v$$
$$p_t = p_s + \tfrac{1}{2}\rho v^2$$

At any point in the air the sum of the static pressure and the velocity pressure is called the *total pressure*.

We must now observe that in order to raise the pressure (absolute or static) of a sample of air we must do work on it; in other words the air must absorb energy. In our example we must work with the lungs to raise the pressure in the balloon. Conversely, if the air pressure falls it must give out energy. Limiting ourselves to moderate static pressure changes for which the air can be considered incompressible we can neglect changes in density and temperature and forget about the corresponding complications of thermodynamics. It is then found that the change in static pressure is equal to the energy absorbed or given out by unit volume of air. Dimensionally we can easily see, thanks to SI, that this is right.

$$1\ \text{Pa} = 1\ \frac{\text{N}}{\text{m}^2} = 1\ \frac{\text{Nm}}{\text{m}^3} = 1\ \frac{\text{J}}{\text{m}^3}$$

Next we remember that the ordinary formula for the kinetic energy of a mass m kg moving with a velocity v m/s is $\frac{1}{2}mv^2$ joules. Therefore the kinetic energy of air which has a mass of ρ kilogrammes per cubic metre is $\frac{1}{2}\rho v^2$ joules per cubic metre when it is moving with velocity v. This is also its velocity pressure in Pa and, dimensionally:

$$\text{Pa} = \text{N/m}^2 = \left(\frac{\text{kg.m}}{\text{s}^2}\right) \Big/ \text{m}^2 = \left(\frac{\text{kg}}{\text{m}^3}\right)\left(\frac{\text{m}}{\text{s}}\right)^2$$

We can now see that the formula at the end of this section in terms of pressures has a corresponding expression in terms of energies:

Total pressure = static pressure + velocity pressure in pascals.
Total energy = static energy + kinetic energy in joules per m³.

Here total pressure and total energy both have the restricted, relative, meaning of their definitions. They are not "total" in an absolute sense since the basic pressure and energy of the atmosphere are excluded.

6.1.5 Pressures at a duct cross-section

p_s = static pressure averaged over the cross sectional area.
$V = Q/A$ = average axial component of air velocity.
$p_v = \frac{1}{2}\rho V^2$ = conventional evaluation of velocity pressure.
$p_t = p_s + p_v$ = conventional evaluation of total pressure.

The pressures considered in earlier sections have been defined as applying to a point in an air stream. In practice average values are required which can be applied to the whole volume flow passing through successive cross-sections of a duct or system of airways.

These are not easy to measure, or even define, in the immediate neighbourhood of an airway element such as a fan or a bend, liable to create a chaotic pattern of air velocity, both in magnitude and direction. However, after traveling a short distance along a straight duct, the departures from the axial direction will have been largely removed and the static pressure will have become uniform over the whole cross section (unless the flow is rotating as a whole). A good average value of static pressure pr can then be measured at a simple hole in the duct wall.

The equations at the head of this section show how values of V p_v and p_t may be defined in terms of the cross-sectional area A and volume flow Q. If the velocity were uniform and axial over the whole area these would be true average values, and the corresponding kinetic energy of the flow would be $Q\frac{1}{2}\rho V^2$. In fact the air velocity, even if axial, is far from uniform, rising from zero at the duct wall to a maximum substantially in excess of V. It follows that the true kinetic energy of flow must exceed $Q\frac{1}{2}\rho V^2$ since the local energy flux is proportional to qv^2, or v^3, and the average of a number of cubes must exceed the cube of their average.

However this excess kinetic energy, and additional excess associated with any rotation of the flow, are by convention, ignored. This simplification is justified because the excess velocity pressure components are unlikely to contribute to the useful work of circulating air round the system.

6.1.6 Total pressure drop and fan total pressure

In Section 6.1.3 it was stated that air experienced a static pressure rise of $\frac{1}{2}\rho v^2$ when slowing down to rest at a stagnation point. The velocity will have fallen from v to zero and the velocity pressure from $\frac{1}{2}\rho v^2$ to zero, so it follows that the total pressure, which is the sum of the static and velocity pressures remains the same. The total energy per unit volume is therefore unchanged, which means that the process was loss-free, the whole of the kinetic energy being transformed into static energy. More generally, velocity changes will not be loss-free. Some energy will be lost (or rather turned into heat which is not significant when the fluid is incompressible) and the total pressure will fall by *the total pressure drop*:

$$p_{t1} - p_{t2} = K.\tfrac{1}{2}\rho V_1^2$$

K depends on the geometry of the system and $\frac{1}{2}\rho V_1^2$ is the conventional (see 6.1.5) velocity pressure at the beginning of the section considered. $\frac{1}{2}\rho V_2^2$may be chosen if it is the larger, but which must be made clear when defining K. Total pressure falls in the direction of flow from 1 to 2, corresponding to the loss of energy which always occurs whether v falls, rises or remains the same. If for example, a duct is expanded to double its original area through a cone of 30° included angle, 20% of the initial kinetic energy will be lost (see Fig. 14.7 b). For this process, therefore, K is 0.2 and the total pressure drop from inlet (1) to outlet (2) of the conical expander will be:

$$p_{t1} - p_{t2} = 0{\cdot}2 \times \tfrac{1}{2}\rho V_1^2$$

With simple incompressible flow the only exception to the rule that total pressure falls in the direction of flow will be a fan. A fan does work on the air and the total pressure will rise from fan inlet to fan outlet by an amount called the fan total pressure, which will be considered in more detail later.

6.1.7 Static pressure difference

The pressure difference between two cross-sections of a duct system, 1 followed by 2 in the direction of air flow is p_{s2} - p_{s1} positive if the pressure is rising, negative if it is falling. Note particularly that the pressure drop p_{s1} - p_{s2} is always opposite in sign to the pressure difference. From the relation of Section 6.1.4 we may write the static pressure difference as:

$$\begin{aligned} p_{s2} - p_{s1} &= (p_{t2} - \tfrac{1}{2}\rho V_2^2) - (p_{t1} - \tfrac{1}{2}\rho V_1^2) \\ &= (\tfrac{1}{2}\rho V_1^2 - \tfrac{1}{2}\rho V_2^2) - (p_{t1} - p_{t2}) \end{aligned}$$

The second part of this expression is the total pressure drop which will always make a negative contribution to the static pressure difference. The first part is the change in conventional velocity pressure, with a negative contribution to static pressure difference if the air speeds up between 1 and 2, a positive contribution if it slows down.

Thus the static pressure can, unlike the total pressure, rise in the direction of airflow. In the example of the conical expander (Section 6.1.6) $V_2 = ½V_1$, and $K = 0.2$, giving a net rise in static pressure - a process known as *diffusion*.

$$\begin{aligned} p_{s2} - p_{s1} &= \tfrac{1}{2}\rho V_1^2 - \tfrac{1}{2}\rho\,(0{\cdot}5V_1)^2 - (p_{t1} - p_{t2}) \\ &= 0{\cdot}75 \times \tfrac{1}{2}\rho V_1^2 - 0{\cdot}2 \times \tfrac{1}{2}\rho V_1^2 \\ &= 0{\cdot}55 \times \tfrac{1}{2}\rho V_1^2 \end{aligned}$$

Table 6.1

Summary of pressure relationships

p	=		absolute pressure at a point
p_o	=		atmospheric pressure at a point
p_s	=	$p - p_o$	static pressure at a point
p_v	=	$\frac{1}{2}\rho v^2$	velocity pressure at a point
p_t	=	$p_s + p_v$	total pressure at a point
p_n	=	p averaged across airway section n	
p_{sn}	=	p_s averaged across airway section n	
p_{vn}	=	$\frac{1}{2}\rho\left[\frac{\text{volume flow through section n}}{\text{area of section n}}\right]^2 = \frac{1}{2}\rho V^2$	
p_{tn}	=	$p_{sn} + p_{vn}$	conventional total pressure at n
Δp_{12}	=	$p_2 - p_1$	absolute pressure difference, which
	=	$p_{s2} - p_{s1}$	static pressure difference, also
Δp_{t12}	=	$p_{t2} - p_{t1}$	total pressure rise (note sign)
P_{tF}	=	$p_{t2} - p_{t1}$	fan total pressure, 1 being the fan inlet and 2 the fan outlet
P_{vF}	=	$\frac{1}{2}\rho V_2^2$	fan velocity pressure
P_{sF}	=	$P_{tF} - P_{vF}$	fan static pressure
P_{f12}	=	$p_{t1} - p_{t2}$	total pressure drop (note sign)
g_{12}	=	P_{f12}/L	total pressure gradient over length L from section 1 to section 2

6.1.6 Relative nature of p_s p_v and p_t

The absolute pressure p_a and the density ρ are the only true physical properties possessed by the air. All the rest are abstractions convenient for calculation but variable according to the frame of reference adopted for the problem in question.

In the great majority of cases the frame of reference will be the still atmosphere. Then velocities will have their ordinary meaning relative to stationary objects, and static and total pressures will be measured above or below atmospheric pressure.

Sometimes, however, a different choice is to be preferred. Suppose, for instance, that the ventilation system for a railway carriage is to be studied. The atmosphere traveling inside with the carriage will be the frame of reference for the small velocities and pressures of internal air movement. The outside atmosphere then forms a relative wind at train speed. The resulting external pressure distribution can be studied on a model in a wind tunnel to help in locating inlet and outlet openings and considering leakage - which will jointly set the carriage pressure in relation to the outside atmosphere.

In this way the system pressures of the order of 100 Pa which generate internal air circulation can be isolated for calculation from external pressure differences which may exceed 1500 Pa.

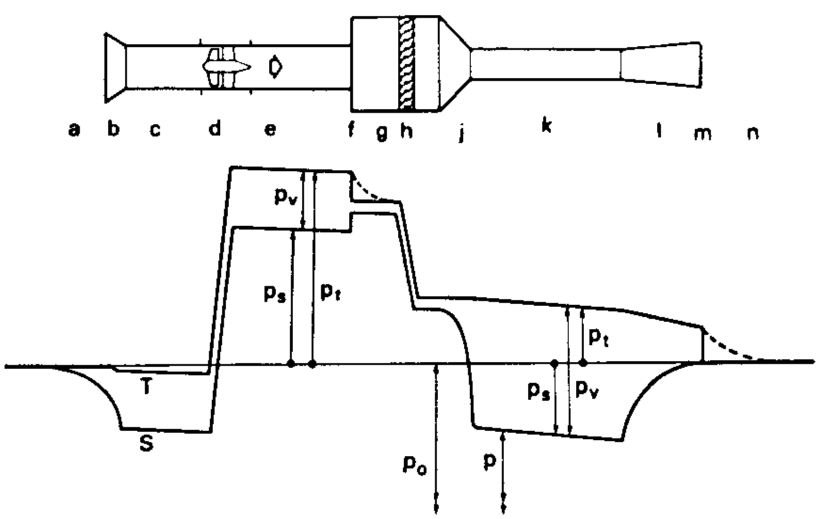


Fig. 6.2 Pressures in a ducted fan system.

6.2 Pressures in a Ducted Fan System

Fig. 6.2 illustrates most of the definitions of pressure which have just been discussed. The heavy lines S and T follow the changes of static pressure and total pressure respectively from inlet to outlet. p_o is the atmospheric pressure and p the absolute pressure in the duct, both measured from an absolute zero far below the bottom of the page. The other quantities can be identified by the arrows on the diagram as:

$$p_s = p - p_o$$
$$p_v = \tfrac{1}{2}\rho V^2$$
$$p_t = p_s + p_v$$

Following the lettered stages through the system with reference to the fig. numbers where numerical data will be found

a. Air velocity increases from zero in the free atmosphere towards the duct entry. There is no loss, so p, is constant at its free atmosphere zero value. As V increases p_s falls with $\frac{1}{2}\rho V^2$.

b. Small drop in p_t corresponding to entry loss.

c, e. p_t falls gradually at friction gradient.
p_v is constant and $p_s = p_t - p_v$.

d. p_t rises by the fan total pressure. In the diagram p_s rises by the same amount, but this is only because inlet and outlet areas are equal in the case illustrated. Note that the change in p_s is never equal to the "fan static pressure", for the definition of which see Chapter 7.

f. When the duct area is suddenly enlarged, p_t will fall corresponding to the loss of energy. The nominal fall is sharp, as shown by the full line. The broken line illustrates the lengthwise distribution of energy loss which in fact occurs. p_s will rise sharply, both nominally and actually as measured by the wall pressure. This, "static pressure regain" arises because the whole of the drop in kinetic energy from e to g is not lost.

h. The drop in p_t here represents "useful work" done in overcoming the flow resistance of a necessary element in the system-for example a heater battery. p_s drops by the same amount because V_g has the same, low, value both approaching and leaving the battery.

j. The air is accelerated with little loss of p_t but with a large fall in p_s, corresponding to the large rise in p_v to $\frac{1}{2}\rho V_k^2$.

k. p_t and p_s fall with pressure gradient corresponding to the velocity in duct k - which is greater than that in duct c.

I. The gradual increase of duct area is accompanied by a moderate energy loss and drop in p_t together with a substantial "static pressure regain" in p_s associated with the large drop in V. This is an example of a diffuser deliberately inserted to improve the efficiency of the system.

m. At any system outlet to the free atmosphere, the whole of the kinetic energy of the flow will be lost. Thus, in contradistinction to the inlet a, where p_s fell and p_t remained zero, at an outlet p_t will fall from its value just before the outlet to zero while p_s will reach zero before discharge at m, and remain zero.

n. The broken line again distinguishes the actual loss of energy along the outlet jet from the sharp fall in nominal p_t at m.

6.3 Design of a Ducted System

An orderly procedure is necessary and is best discussed in relation to an example. A building ventilation system will be chosen for this purpose.

(i) The volume flows to be supplied to and extracted from each room in the building will first be determined, as discussed in the preceding chapters.

(ii) The building site will next be considered to settle the best areas for intake from and discharge to the atmosphere, avoiding the dangers of recirculation and excessive wind and weather exposure.

(iii) The choice between unitary or central systems will be made, and whether one or more plant rooms are needed according to the requirements for accommodation, maintenance and noise control of the equipment. The plant will be sited as centrally as possible to minimise the distance air has to be moved.

(iv) The building plans will be studied, and the building itself if it exists, to find the best location for room inlets and outlets and the duct runs joining them to the fans and plant. These must meet the requirements for good room air distribution, while keeping the duct layout as simple as possible with room for gradual changes of duct area, shape and direction.

(v) Having settled the route and flow rate for each section of main and branch duct, the size and shape of each grille, duct, bend and section change must be determined, which is the subject matter of this chapter. One objective is to find the right balance between running costs-represented by the total pressure drop, assuming that maintenance snags have been avoided-and first cost, which tends to be higher the lower the loss of each system element.

(vi) There are several ways of approaching duct system design. The simplest is the velocity method, which involves selecting main and branch air velocities on the basis of experience. This is probably worth doing in any case as a first step, before moving to more refined methods. Trial and error is an almost inevitable feature of design, and while this can be systematised, and handled by computer programme, it remains important to understand the successive steps.

Two factors influence velocity selection. Firstly velocities must fall as the size of duct is reduced to avoid increasing pressure gradients. Secondly, noise generation increases rapidly with velocity at grilles, bends, and other fittings where the flow separates from the walls, leaving turbulent eddies in its wake. Table 6.2 gives some typical recommendations. High velocity systems require noise control by means of sound absorbent units between the duct system and room outlets or inlets. They are mainly to save space in city-centre buildings, where every available cubic metre commands a high rental, offsetting the additional running cost.

The higher values apply to the larger ducts, to areas less sensitive to noise, and to locations farther from the occupants. Inlet and outlet grille velocities may be influenced by a noise control survey.

Table 6.2
Typical air velocities in ducted systems

(Velocities in m/s)	Main ducts	Branch ducts	Supply grilles	Exhaust grilles
Residences	4 – 5	3 – 4	2 – 3	1·5 – 2
Quiet public buildings	5 – 8	4 – 6	3 – 5	2 – 3
Busy public buildings	8 –11	6 – 8	5 – 8	3 – 4
Factories	8 –15	6 –10	5 –10	4 –10
High velocity systems	12 –25	10 –20	3 – 8	2 – 4

6.4 Estimation of Total Pressure Drop

While the resistance of a ducted system can be estimated in terms of static pressure changes, the total pressure method is adopted here. This avoids the use of the somewhat confusing "static regain" concept, and is logical in that total pressure drop is a true measure of energy loss. When using total pressure it is important to remember that the loss of total pressure at system outlet must always be included. This applies whether the air leaves through an outlet grille into a room, leaves through weather louvres to the atmosphere or is directly discharged by the fan itself. In each case there is an item of total pressure drop equal to the velocity pressure corresponding to the average velocity at outlet.

The loss in each element of the system (bend, duct length, etc.) is dependent on the average velocity through it, which is taken as:

$$\text{Average velocity } V(\text{m/s}) = \frac{\text{Volume flow, Q (m}^3\text{/s)}}{\text{Gross cross section, A (m}^2\text{)}}$$

From this velocity, and the air density, ρ(kg/m^3) the conventional velocity pressure is determined:

$$\text{Velocity pressure} = \tfrac{1}{2}\rho V^2 \text{ Pa.}$$

This is then multiplied by a factor, K, and the result is the total pressure drop from the inlet (1) to the outlet (2) of the element. Since we are calling it a pressure drop, we can ignore the negative sign which it would have as a pressure change (from high to low in the direction of flow).

$$\text{Total pressure drop } P_{f12} = K.\tfrac{1}{2}\rho V_1^2$$
$$\text{or } P_{f12} = K.\tfrac{1}{2}\rho V_2^2$$

Typical values of K applying to V_1 or V_2 as specified have been established for the commoner system elements, and a number are given in Sections 6.6.1 to 6.6.5. Different authorities do not always agree on the values, but exactitude is not to be expected. The fact is that the velocity distribution across the inlet cross-section affects the loss in each element, which in turn affects the element next following. In a practical duct system an overall accuracy of ± 10% in the total pressure drop estimation would be very good indeed, and would only cause some

± 2 or 3% variation in the volume flow handled by the fan. The elements of total pressure drop are added together round the system, and may be classified as follows

Losses at entry to the system from atmosphere. Losses due to friction in duct lengths.
Losses at changes of duct area or shape.
Losses at bends and changes of direction.
Losses at division of flow into branches.
Losses caused by obstructions, grilles and louvres. Losses in filters, heaters, and other "useful" elements. Losses at discharge from the system to atmosphere. Change in atmospheric pressure from inlet to outlet.

The last item brings out the fact that a ducted system should be regarded as a complete circuit. Of course the air entering the system is not (or should not be) the same air that leaves the system outlets. But it is flowing at exactly the same volume (strictly mass) rate and can be looked upon as ultimately circulating round at almost zero velocity through the almost infinite area offered by the free atmosphere.

The data in section 6.6 are derived from a number of sources, the most important being the research results and survey published by BHRA (Internal Flow - a guide to losses in pipe and duct systems. D. S. Miller, British Hydromechanics Research Ass. Bedford). Slight liberties have been taken in smoothing and extrapolating the data in the interests of simplicity and clarity, but this is a field where differences of 20 to 25% between one authority and another are by no means unusual.

The K values given apply to medium to large volume flows at medium to high velocities-these being, of course, the conditions under which the resulting energy losses will be most significant. In smaller systems, as judged by the *Reynolds number* criterion (see 13.5.2), K will be greater. The factor by which it should be increased is roughly as in the following table for low-loss bends and similar components; the increase is less for sharp bends, and in other cases where the flow separates from the walls.

Reynolds Number $\times 10^{-4}$	1·0	1·5	3	5	10	50 and larger
D_e at 5 m/s	30	45	90	150	300	1500mm
D_e at 15 m/s	10	15	30	50	100	500mm
Multiply K by	2·0	1·8	1·6	1·4	1·2	1·0

The table shows the effective diameter, $D_e$mm, corresponding to each Reynolds number with atmospheric air at 5 and 15 m/s velocity.

6.5 Discussion of System Resistance Elements

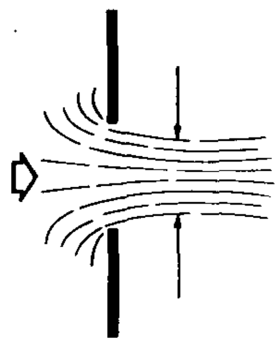
6.5.1 inlets. A plain intake should always be avoided unless the velocity is very low - as for a filter. Flanging will nearly halve the loss, but a 60° cone D/6 long is best for the usual installation. Curved bell-mouth intakes are the optimum, but require tooling and are mostly limited to proprietary equipment, such as fans.

6.5.2 **Outlets**. Always remember to include one velocity pressure for the final discharge to atmosphere - whether from a duct or a fan. This velocity pressure is already included in Fig. 6.8 and the tables for outlet louvres and cowls to show the overall effect of outlet area selection in these cases.

6.5.3 **Sudden expansion** cannot be improved upon for area ratios up to 1.3. At higher ratios diffusers should be considered.

6.5.4 **Sudden contraction** does not produce much loss at area ratios down to 0.8 but does generate separated flow and some noise. A 60° or less cone contraction eliminates these defects, and is particularly necessary at the inlet to a fan.

6.5.5 **Orifices** produce substantial loss, and will only be used for a specific purpose such as flow measurement or flow control. As sketched here the air stream contracts after passing through the hole, raising the velocity up to 1 -66 times, the increased velocity pressure being substantially lost. Rounding the inlet side would reduce the loss, but in fact the pressure drop is usually the whole purpose of inserting the diaphragm, damper or valve. Much the same drop will be produced with single holes of any shape having the same area ratio.



6.5.6 **Guards** must have mesh sizes meeting the safety requirements and the wire or bar of which they are made must be sufficiently robust. Once these needs are met any further reduction in free area will cause unnecessary loss. Locate the guard in a low velocity section if possible. Wire mesh or perforated plate designed to make K = 2 is used in test airways to produce a velocity distribution as uniform as possible over the cross section.

6.5.7 **Obstructions**. Every effort should be made to avoid passing pipes or struts through air ducts. If it cannot be avoided streamlining may be considered. A simple rounded nose and 20° wedge-shaped tail may be sufficient.

If a conflict cannot be avoided (a failure of overall planning) it may be better to bypass the obstruction provided gradual expansions or limited changes of direction can be achieved.

6.5.6 **Filters**, **heat exchangers** and the like should have pressure loss data provided by the supplier. Filters in particular show a K value decreasing as the air velocity rises, since the pressure drop does not increase as fast as the velocity squared - an effect of the low Reynolds number in the material.

6.5.9 Diffusers joining two ducts are usually required to expand through a specified area ratio while the length is at the designer's choice. The loss falls with increasing length to a minimum (K = 0.06) with an included angle of 7° or less. A 7° diffuser is very often too long in practice, bearing in mind that 2 to 4 diameters of uniform duct are needed after the diffuser for full recovery. If the limited length (L/De) available (or economical) places the design in the region of Fig. 6.7 above the broken line, it is better to truncate the diffuser as illustrated in the example. Note that diffusers with an included angle more than 25° are rarely worth while - sudden expansion is nearly as good.

While the total pressure falls, the static pressure rises from inlet to outlet of a diffuser. This rise is called the static pressure regain.

6.5.10 Outlet diffusers save energy by reducing the velocity at which the air is finally discharged. If they expand at too steep an angle the extra gain in velocity pressure will be more than offset by extra diffusion losses. Generally speaking the designer will need to limit the length of the diffuser for cost reasons or because space is limited - a diffuser outlet should be at least 1½ times its small diameter from a wall. He can, however, choose the outlet area, and Fig. 6.8 shows the area ratio for minimum loss as a function of length. The figure also gives the corresponding loss and included angle.

6.5.11 Bends are a prolific source of unnecessary loss - and unwanted noise. As shown by the high K values a sharp throat is the worst feature - it is good practice never to make the throat radius less than half the width of the duct. If there is no room for a decent throat radius the loss may be reduced by fitting various types of vane to organise the direction change of air. The high loss of a sharp bend is caused by separation of flow at the throat, and unless a length of duct follows the bend, none of the high velocity pressure thus generated will be recovered - resulting in a spectacularly large outlet loss.

6.5.12 Interaction between bends which are close to one another affects the loss in the second bend. See 6.6.6. Provided the bends have a throat radius of at least half the width the results are favourable - a reduction in total loss. Sharp bends on the other hand must be spaced at least two diameters apart, or else the losses will be greatly increased for the reasons discussed in the last paragraph. To insure against under estimating the losses, diffusers and expanders should be spaced three diameters from bends.

6.5.13 Branches. Where flows combine or divide at branch junctions losses occur affecting both main and branch flows. A study of Figs. 6.11 to 6.14 shows the advantage of entering or leaving the main flow with a 45° direction change only - or, if a right angle branch is necessary, by a well radiused or chamfered junction. Under these circumstances combining flow causes little overall loss, there being rather a transfer of energy from branch to main or main to branch according to which has the higher velocity. In a dividing branch the loss accompanies velocity reduction, and it is best to design for roughly equal velocities. The same applies to Y-type flow divisions.

For minimum dividing branch loss it is best to use a rectangular duct in which branches are taken off with half-throat-radius bends of constant height, while the main duct area is proportionately reduced at each branch. Velocity reduction by more than 10% should be avoided both in branch and main, though acceleration by 30% or more is quite safe.

6.5.14 Wind pressure must be taken account in system design in exposed locations-which include high buildings. Air taken in on the leeward side and forced out on the windward side will encounter an adverse pressure difference which should be included as an item of total pressure drop. Locations can be found on the roofs of some buildings where the air pressure is always much the same as it is on the leeward side. These are clearly favourable for system outlets since the wind pressure will make a zero or negative contribution to the system pressure drop, and may therefore be neglected. See Section 4.4 for wind pressure estimates.

6.5.15 Straight lengths. A short length, L metres, of straight duct may be roughly estimated for total pressure drop P_f by the use of the following approximate values of the loss factor K in the basic formula.

$$P_f = K.\tfrac{1}{2}\rho V^2 \qquad \text{Pa}$$

$$K = 0{\cdot}02\frac{L}{D} \qquad \text{Cylindrical duct, diameter D metres}$$

$$K = 0{\cdot}01\frac{L\,(A+B)}{AB} \qquad \text{Rectangular duct, A} \times \text{B metres}$$

However, when the duct is long enough to make a significant contribution to the total pressure drop of the system, account must be taken of the variability of the friction factor, f, in the formulae which should really be written.

$$K = f\frac{L}{D} \qquad \text{for a cylindrical duct}$$

$$K = f\frac{L}{D_e} \qquad \text{for a rectangular or other duct of effective diameter } D_e.$$

The friction factor f varies in quite a complex manner with the gas velocity, the properties of the gas, the diameter of the duct, and the effective roughness of its walls. The general case, applicable to long airways in mining, tunnelling and industrial process work, is discussed in Chapter 13. For ventilating, and most air handling work it is adequate, and much simpler, to use a chart which takes into account the variations of f with duct diameter and velocity when atmospheric air is the gas flowing.

Fig. 6.17 is such a chart, drawn up to give the pressure gradient directly with air of standard density, = 1.2 kg/m^3, and with wall roughness, ε μm (micrometres) representative of typical ventilating duct construction. Full details of the basis and use of this chart are given, with an example, on pages 98 and 99. Corrections are given for density - exact when the gas is predominantly air - and wall roughness. The roughness correction is approximate only, but then so, usually, is the evaluation of e. The chart is plotted from the formula:

$$g = \frac{P_f}{L} = f.\frac{\rho V^2}{2D} = f.\frac{8\,\rho Q^2}{\pi^2 D^5}.\ \text{Pa/m} \qquad (61)$$

6.5.16 Equivalent diameters

In order to use Chart 6.17 for ducts of other than cylindrical crosssection an appropriate value must be found for D. This is not simply the value giving the same area. It can be shown that a dimension Dh called the hydraulic diameter, should be used in the basic equation, and will cover cross-sections of any shape.

$$P_f = f.\frac{L}{D_h}.\tfrac{1}{2}\rho V^2 \qquad (62)$$

$$\text{where } D_h = \frac{4 \times (\text{cross-sectional area of duct})}{(\text{Length round periphery of cross-section})}$$

This is still not the value to use in the chart. What is wanted is a genuinely equivalent cylindrical duct which, if substituted for the rectangular or other duct, would pass the same volume flow, Q, under the same pressure gradient. The area and velocity for such a duct would be different, and so therefore would the friction factor f. Allowance for this can be made (see Chapter 13) by the somewhat awkward formula below under the same conditions of standard air and wall roughness. Use of the formula can be avoided by referring to Chart 6.18 which gives, for a rectangular duct a x b mm, an effective diameter D_q mm transferrable directly across to chart 6.17.

$$D_q = 1{\cdot}265\,\frac{a^{0\cdot6}\,b^{0\cdot6}}{(a+b)^{0\cdot2}} \qquad (63)$$

6.6 Losses in Duct System Elements

For each system element the drop in total pressure P_f equals

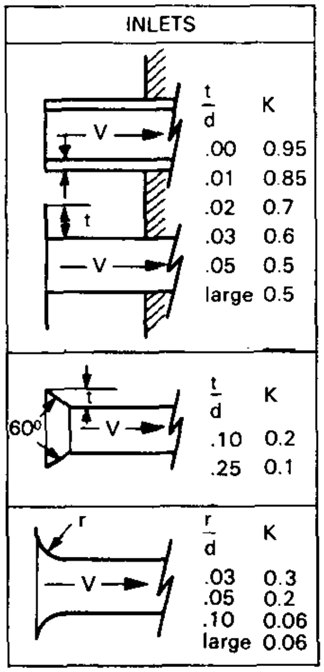
$$p_{t1} - p_{t2} = K.\tfrac{1}{2}\rho V^2 \quad \text{Pa}$$
$$= 0{\cdot}6\ KV^2 \quad \text{in air of standard density.}$$

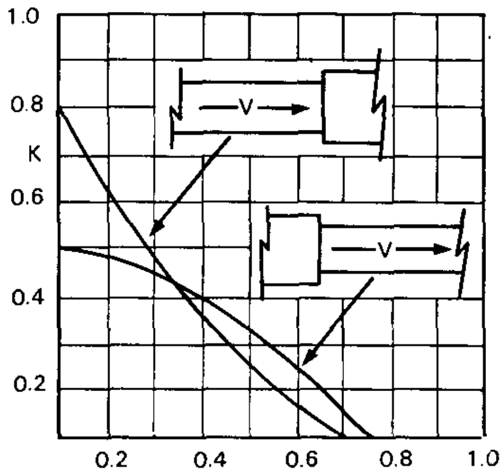
V (m/s) is the average velocity in the section indicated in each diagram.

K includes allowances for the influence of the component on duct losses downstream, and for the duct length occupied by the component.

The system loss is the sum of all the $K\frac{1}{2}\rho V^2$ items.

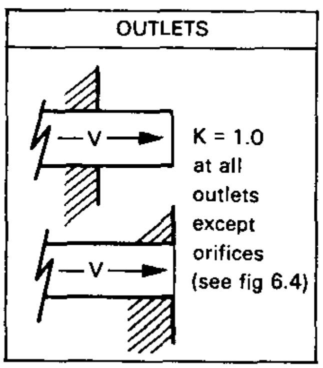
6.6.1 Losses at inlet, outlet and sudden section change

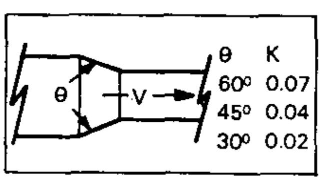


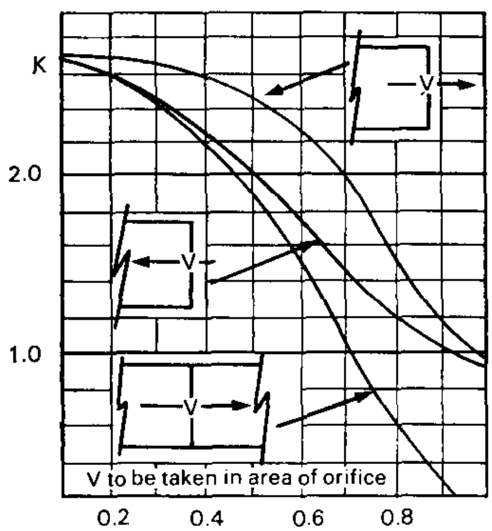


Smaller duct area/larger duct area.

Fig. 6.3
Sudden expansion and contraction.



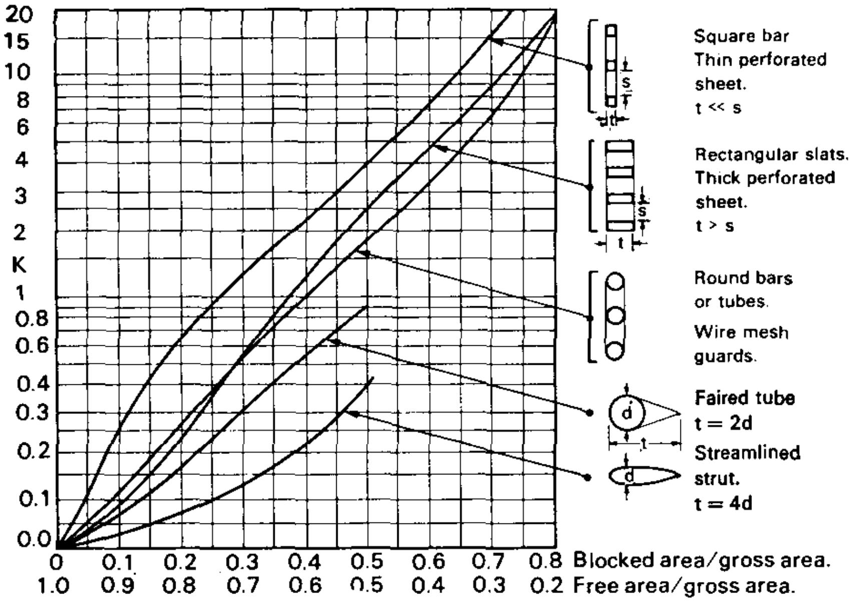




Orifice diameter/duct diameter.

Fig. 6.4
Losses at sharp-edged orifices.

6.6.2 Losses at obstructions to air flow



K is the loss factor within a square or round duct.

At duct outlet $K_{out} = K + 1$

At duct inlet $K_{in} = K +$ inlet loss factor (approx.)

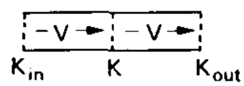
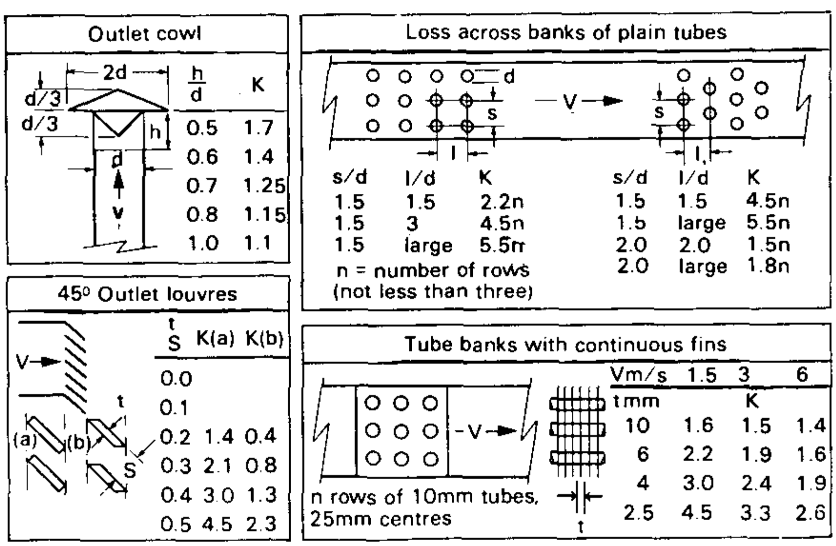


Fig 6.5 Losses at guards, screens and obstructions.



Outlet cowl

h/d	K
0.5	1.7
0.6	1.4
0.7	1.25
0.8	1.15
1.0	1.1

45° Outlet louvres

t/S	K(a)	K(b)
0.0		
0.1		
0.2	1.4	0.4
0.3	2.1	0.8
0.4	3.0	1.3
0.5	4.5	2.3

Loss across banks of plain tubes

s/d	l/d	K
1.5	1.5	2.2n
1.5	3	4.5n
1.5	large	5.5n

n = number of rows (not less than three)

s/d	l/d	K
1.5	1.5	4.5n
1.5	large	5.5n
2.0	2.0	1.5n
2.0	large	1.8n

Tube banks with continuous fins

Vm/s	1.5	3	6
t mm		K	
10	1.6	1.5	1.4
6	2.2	1.9	1.6
4	3.0	2.4	1.9
2.5	4.5	3.3	2.6

Fig 6.6 Losses across tube batteries.

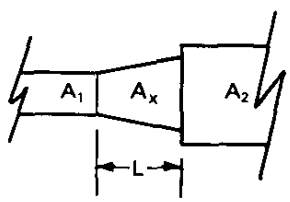
6.6.3 Losses in diffusers

Fig 6.7 gives the loss factors for expanding diffusers joining two ducts of areas A_1 and A_2. De is the effective diameter of the inlet duct, defined as follows for various expander shapes:

Circular (D_1 diameter) to circular (D_2)	$D_e = D_1$
Circular (D_1) to rectangular ($B_2 \times C_2$)	$D_e = D_1$
Rectangular ($B_1 \times C_1$) to rectangular ($B_1 \times C_2$) (Expanding on the shorter side C only)	$D_e = C_1$
Rectangular ($B_1 \times C_1$) to rectangular ($B_2 \times C_2$)	$D_e = \dfrac{2\,B_1C_1}{B_1 + C_1}$

The downstream duct should be at least $4D_2$ long, for full pressure recovery.

Truncated Diffusers. *If* the length available for a diffuser is limited, it is better to use the whole length L to expand to the area A_x given by the area ratio at the intersection of the broken line Kx with the appropriate L/D_e ratio, and then to expand abruptly to the downstream area A_2 assumed greater than A_x. The overall loss factor can then be taken as:



$$K = K_x + \left(\frac{A_1}{A_x}\right)^2 - \left(\frac{A_1}{A_2}\right)^2 \qquad (64)$$

Example. A length L = 2.4m is available to expand from 1.6m diameter (A_1 = 2.02M^2) to 2.7m (A_2 = 5.71m^2)

$$D_e = D_1 = 1{\cdot}6\text{m} \qquad L/D_e = 2{\cdot}4/1{\cdot}6 = 1{\cdot}5 \qquad A_2/A_1 = 2{\cdot}85$$

From Fig 6.7 (point A) K = 0.25 if the diffuser expands to 2.7m.

If a truncated cone is adopted the intersection X shows that K_x = 0.12 and the area ratio is 2.2. Therefore A_x = 2.2 x 2.02 = 4.44M^2 and D_x = 2.4m.

$$K = 0{\cdot}12 + (2{\cdot}02/4{\cdot}44)^2 - (2{\cdot}02/5{\cdot}71)^2 = 0{\cdot}20$$

Open outlet diffusers. *For* a given length L there is an optimum expansion ratio which will minimise loss and maximise static pressure regain. This area ratio and the corresponding loss factor K are given by the full line curves in Fig 6.8 for normal duct approach flow.

An exceptionally uniform approach velocity (as in a venturi-nozzle flow meter) allows the more rapid expansion and lower loss given by the broken lines.

For an optimum diffuser at fan outlet the manufacturer should be consulted.

Total pressure drop = $K.\frac{1}{2}\rho V^2$

Static pressure regain = $\left[1 - K - \left(\frac{A_1}{A_2}\right)^2\right].\frac{1}{2}\rho V_1^2$ (in duct diffuser)

Static pressure regain = $[1 - K].\frac{1}{2}\rho V_1^2$ (open outlet diffuser)

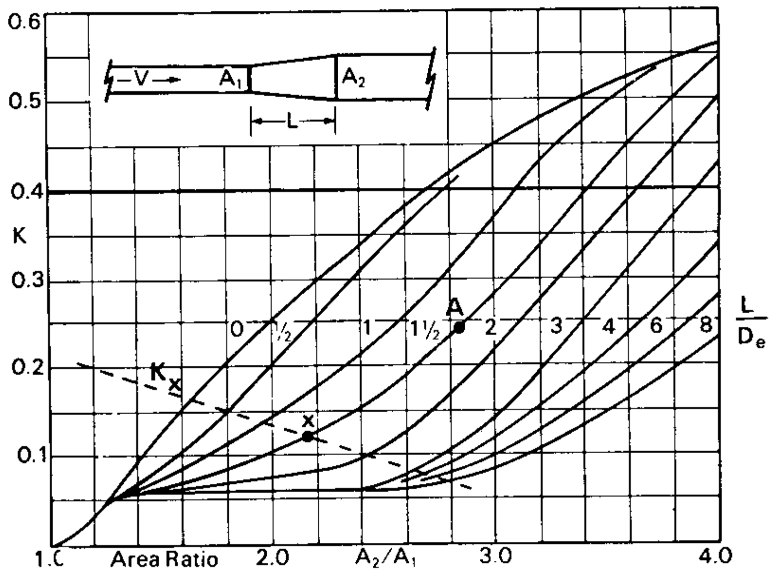


Fig 6.7 Losses of in-duct diffusers

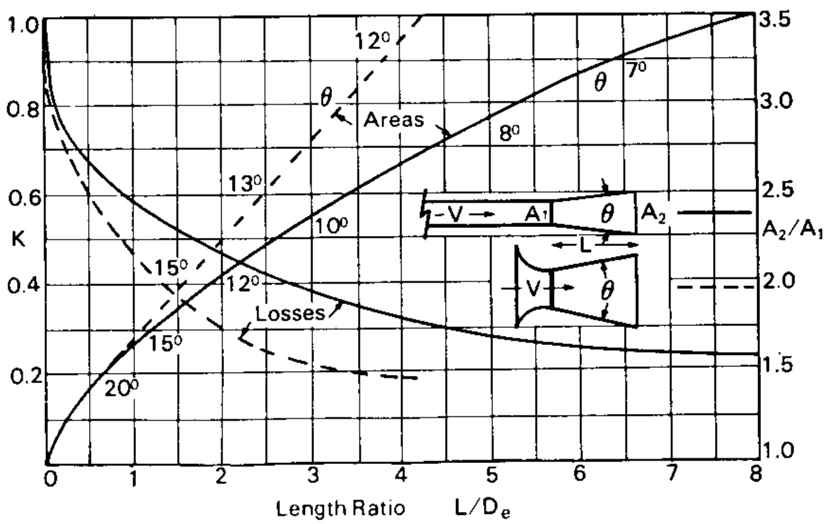
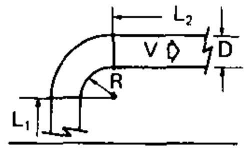


Fig 6.8 Losses of open-outlet diffusers of optimum proportions.

Applies to circular/circular or rectangular/rectangular expansions.

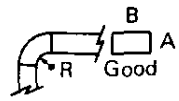
6.6.4 Losses in bends

The bend loss $P_f = K.½\rho V^2$ is to be added to the duct friction loss calculated up to bend entry L_1 and on from bend exit L_2.

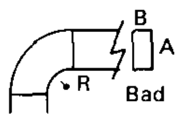


The ratios R/D and R/A plotted relate to the Throat radius R shown here. Some authorities measure radius to the centre line of the bend, quoting a radius ratio equal to:

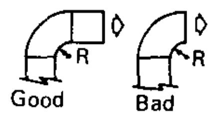
$$(R/D + \tfrac{1}{2}) \text{ or } (R/A + \tfrac{1}{2}).$$



For a rectangular duct A x B, A is the dimension in the same plane as R. Losses depend on the *aspect ratio* B/A and are higher when A is the larger dimension.

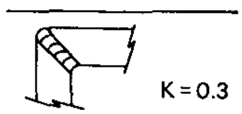


An outlet duct length at least equal to D or A is required for pressure recovery after the bend. If this is omitted, i.e. if the bend is *open outlet*, the value of K may be multiplied up to two and a half times.

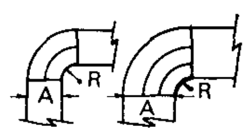


When space does not permit the use of a generous throat radius, the following vane constructions may be considered.

Turning vanes of thin sheet circular arc form, spanning a mitre bend in a round or rectangular duct.

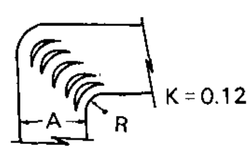


Splitter vanes in a rectangular bend of small throat radius R. One splitter has a radius = $\sqrt{R(R + A)}$. Two splitters have radii = $\sqrt[3]{R^2 (R + A)}$ and $\sqrt[3]{R(R + A)^2}$ respectively.



For K see Fig 6.10

Aerofoil vanes of solid or hollow construction in a rectangular bend. (n-1) vanes will form n passages each with a throat to width ratio of 1.0. The throat radius of the main bend must be A/n. Each vane has an inside radius of 2A/n and an outside radius of A/n plus straight extensions.



K for bends turning through angles other than 90° can be expressed approximately as a percentage of the 90° value.

Angle of bend	15°	30°	45°	60°	75°	90°	120°	180°
Easy bend (R/D > ½)	17	33	50	67	83	100	115	125%
Sharp bend (e.g. mitre)	5	15	30	50	75	100	140	— %

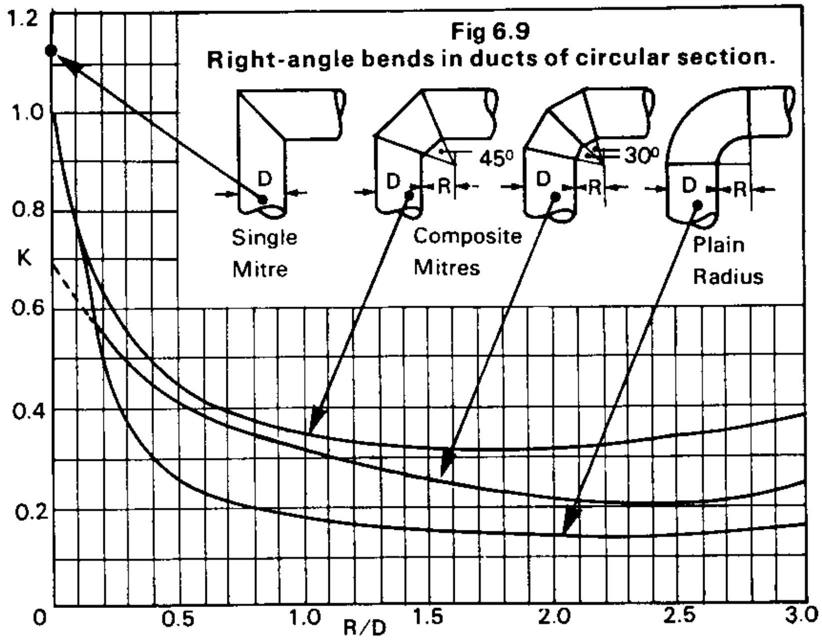


Fig 6.9
Right-angle bends in ducts of circular section.

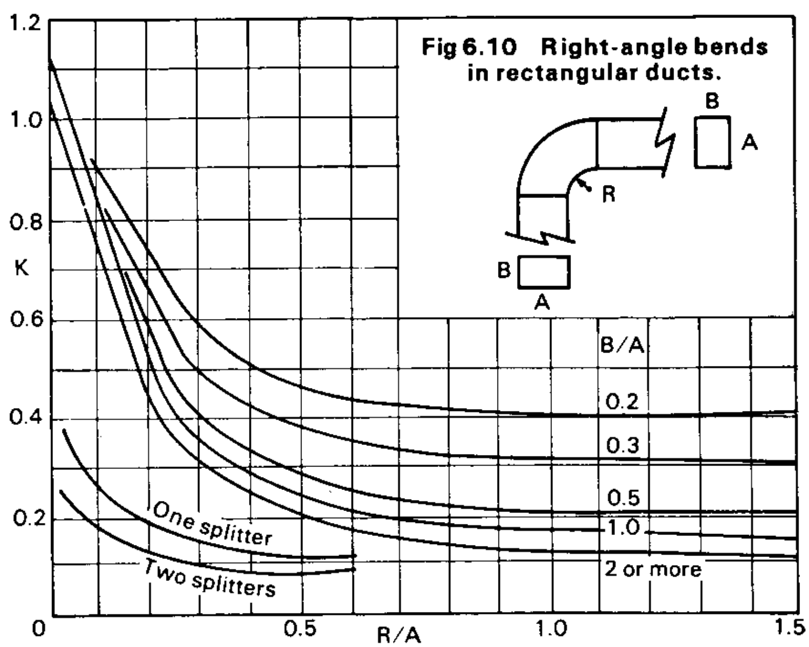
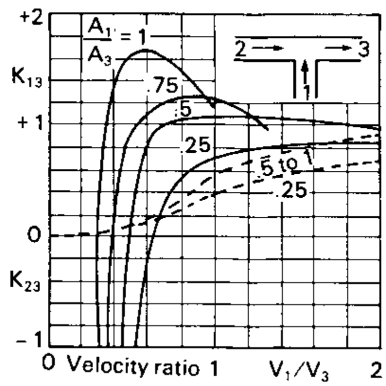
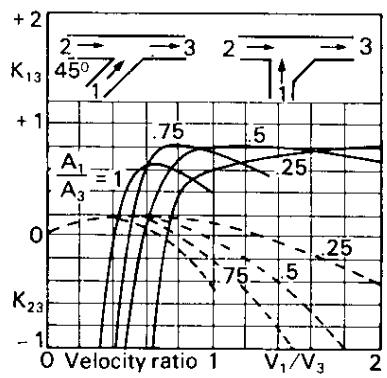


Fig 6.10 Right-angle bends in rectangular ducts.

6.6.5 Losses at branches

These loss factors apply to round or rectangular ducts with constant area main. They depend on the branch area ratio A_1/A_3 and the branch velocity ratio V_1/V_3 - therefore on the volume flow ratio $Q_1/Q_3 = A_1V_1/A_3V_3$. Figs. 6.12 and 6.14 apply to the 45° branch case, but right angle branches with 0.5 or more throat radius or chamfer will be nearly as efficient. Y- and T- flow division is dealt with in Fig. 6.15.

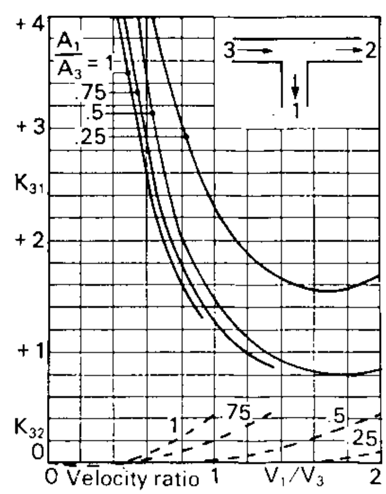


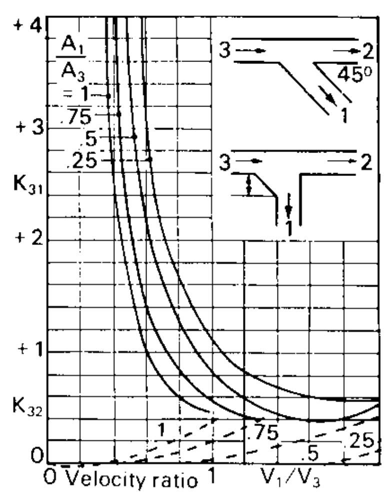


Δp_{13} from branch to main $= p_{t1} - p_{t3} = K_{13}.\frac{1}{2}\rho V_1^2$ ———

Δp_{23} across branch $= p_{t2} - p_{t3} = K_{23}.\frac{1}{2}\rho V_3^2$ — — — —

Fig. 6.11 Flow entering main. **Fig. 6.12 Flow entering main.**





Δp_{31} from main to branch $= p_{t3} - p_{t1} = K_{31}.\frac{1}{2}\rho V_1^2$ ———

Δp_{32} across branch $= p_{t3} - p_{t2} = K_{32}.\frac{1}{2}\rho V_3^2$ — — — —

Fig. 6.13 Flow leaving main. **Fig. 6.14 Flow leaving main.**

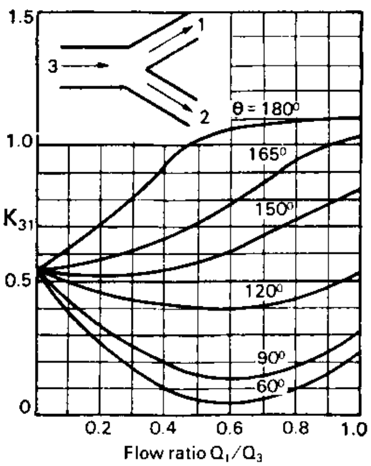


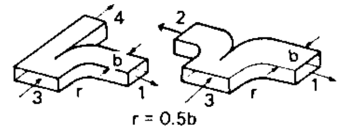
Fig 6.15 Y-type flow division.

$\Delta p_{31} = p_{t3} - p_{t1} = K_{31} \cdot \frac{1}{2} \rho V_3^2$

K_{32} = value of K_{31} at $(1 - Q_1/Q_3)$

$A_1 = A_2 = \frac{1}{2} A_3$ for $\theta = 60°$ and $90°$

$A_1 = A_2 = A_3$ for $\theta = 120°$ to $180°$



Low-loss right angle branches and tees

Loss from main 3 to branch 1 = $K_{31} \cdot \frac{1}{2} \rho V_1^2$
Loss from main 3 to branch 2 = $K_{32} \cdot \frac{1}{2} \rho V_2^2$
Loss from main 3 to branch 4 = $K_{34} \cdot \frac{1}{2} \rho V_4^2$

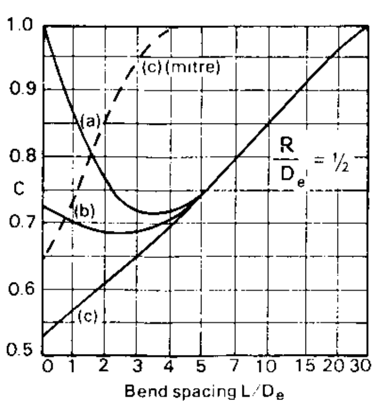
These loss factors apply only so long as V_1/V_3, V_2/V_3 and V_4/V_3 all lie between 0·9 and 1·3

Q_1/Q_3	0.1	0.15	0.25	0.35	0.5	0.7	0.9
K_{31}	0.5	0.4	0.3	0.25	0.2	0.2	0.25
K_{34}	0.04	0.04	0.04	0.05	0.1	0.15	0.3
K_{31}	0.5	0.4	0.3	0.25	0.2	0.2	0.25
K_{32}	0.25	0.25	0.2	0.2	0.2	0.25	0.5

6.6.6 Influence of one bend on another

The overall loss of two closely spaced, well radiused, bends is less than the sum of their separate losses, unlike most inter-action effects which are unfavourable as are those for sharp bends in arrangements (a) and (b).

Multiply the sum of the losses in the two bends and the duct between them by C from Fig. 6.76. Correction for angles other than 90° can be used for type (a), but not for type (b) or (c) configurations.



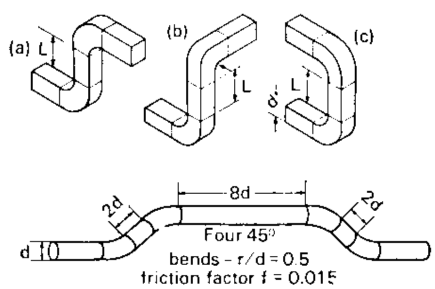


Fig 6.16 Interaction of bends.

Example: Offset to clear an obstruction.
One 90°, 0.5 r/d bend from Fig 6.9: K = 0.26
Correcting to 45° (6.12.4): 0.26 × 50% = 0.13
Sum of loss factors for bends and spacers:
4 × 0.13 + (2 + 8 + 2) × 0.015 = 0.70
Correction factor for 2d long spacers is 0.76 from Fig 6.16, curve (a). Ignore effect of 8d.
Overall loss is 0.76 × 0.70 = 0.53 × $\frac{1}{2} \rho V^2$.

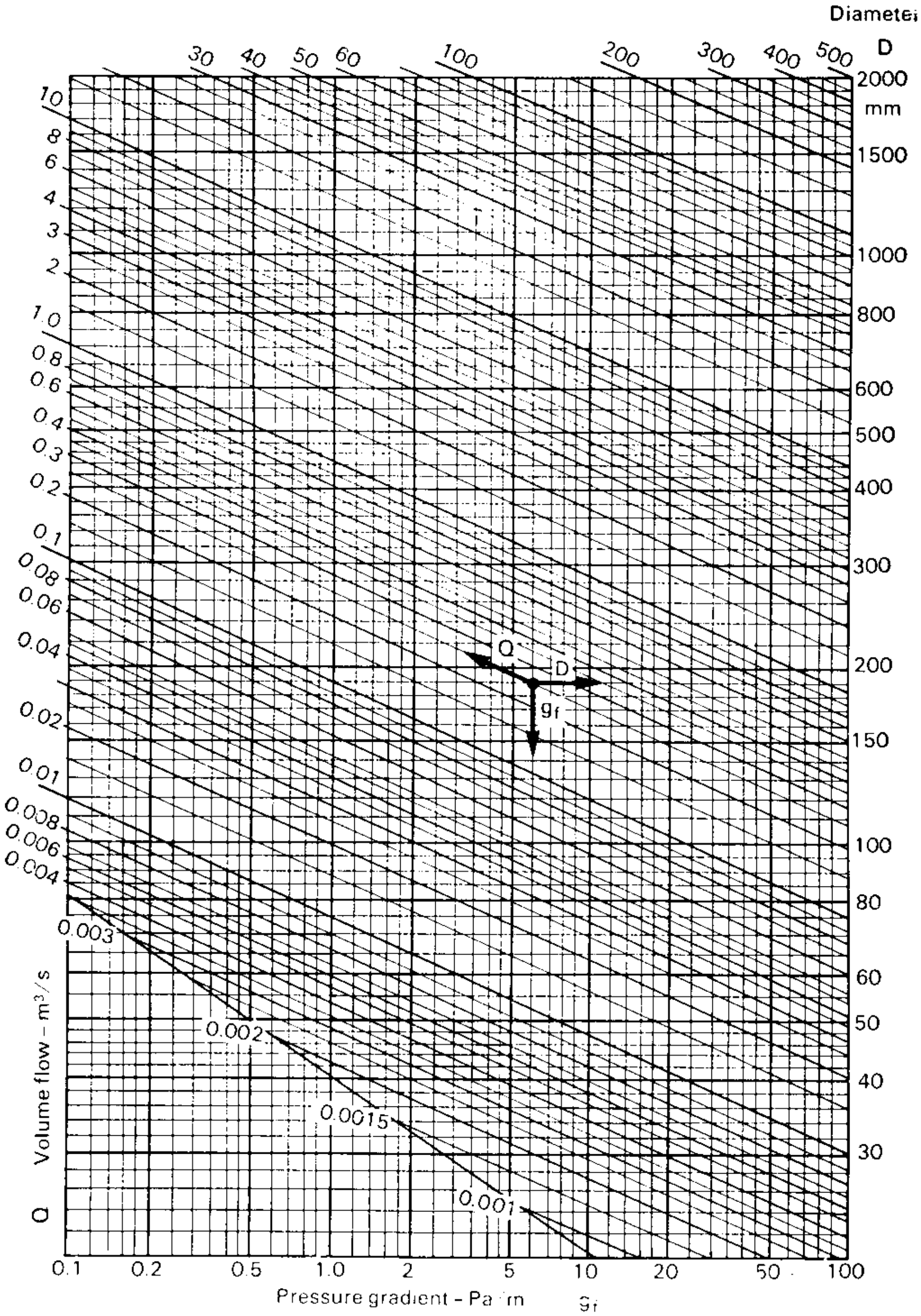
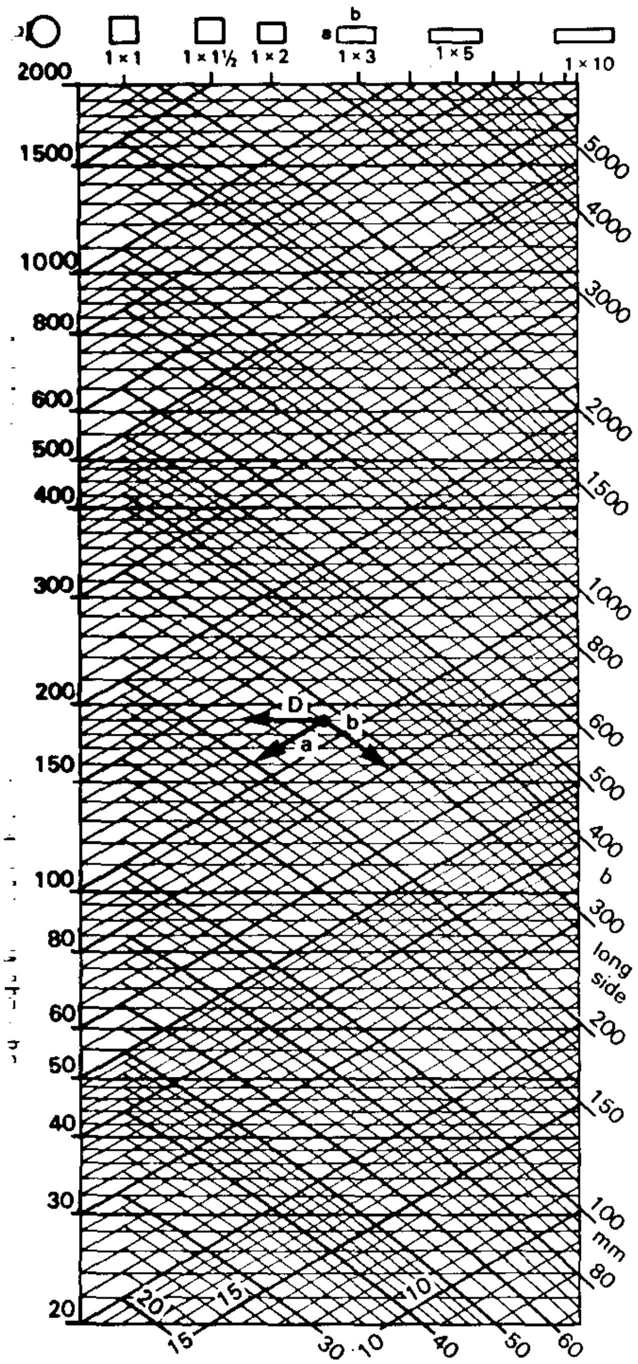


Fig. 6.17 Pressure gradient in circular ducts.



6.6.7. Losses due to duct friction

See sections 6.5.15 and 6.5.16.

Density 1·2 kg/m³. At other densities multiply by $\rho/1{\cdot}2$.

Roughness of duct wall:

0 to 50mm diameter:
ϵ less than 15μm.
Smooth bore tubing.

50 to 200mm diameter:
ϵ of intermediate value.
Aluminium and similar ducts.

200mm upwards:
ϵ = 150μm.
Galvanised metal ducts with flanged joints about every metre.

Factors (approximate) for ducts in other materials over 200mm dia.

Spirally wound R.I.	× 0·9
Cement or plaster faced	× 1·1
Welded and painted	× 1·25
Fair faced brick	× 1·5
Rough brick or concrete	× 2·2

The equivalent circular duct has the same wall roughness as the rectangular duct and gives the same pressure gradient at the same volume flow.

Example:

A welded and painted steel rectangular duct 40m long with cross-section 110 × 280mm carries 0·25m³/s of hot flue gas with a density of 0·6 kg/m³.

From Fig. 6.18, 110 × 280mm is equivalent to 190mm diameter.

From Fig. 6.17 0·25m³/s in a 190mm duct gives 6·0 Pa/m gradient.

For 40m length	× 40
For 500μm roughness	× 1·25
For 0·6 kg/m³, 0·6/1·2	× 0·50

Therefore the pressure drop from inlet to outlet is 6·0 × 40 × 1·25 × 0·50 = 150 Pa.

Fig. 6.18 Equivalent diameter of rectangular ducts.

6.7 Duct Sizing in Branched Systems

The equal friction method is often used for ventilating systems in which the ducting is extensively branched. The basic idea is that branches with only a fraction of the main duct flow should be operated at lower velocity so that the pressure gradient would be everywhere the same. A pressure gradient of 1 Pa/m makes a good first trial for an ordinary low velocity system. The method will be readily followed from the following example, making direct use of Fig. 6.17.

	m^3/s per duct	Pa/m gradient	mm diam	V, m/s (calculated)
Fan duct	12·0	1·0	1140	11·8
4 main ducts	3·0	1·0	680	8·3
40 final branches	0·3	1·0	286	4·7

The calculated diameters would, of course, be rounded off or converted into reasonably equivalent rectangles. For example, to keep main and branch ducts of common height we might choose, using Fig 6.18:

Final branch ducts	400mm high × 175mm wide
Main ducts	400mm high × 1,000mm wide

with the main ducts tapering in width as the m^3/s fall. The ratio of branch velocity to main velocity is, however, low for efficient junction and it would be better to bring them into closer agreement. See Fig.6.13.

In a branched system the total pressure drop is usually calculated for the run to the farthest branch outlet or inlet. The total pressure drop to a nearer branch would be less, and as equal volume flow at all outlets or inlets is probably the design objective a damper would be needed in each branch to absorb the excess pressure. The system would be "balanced" after installation, the damper in the farthest branch being set fully open and the remainder closed by trial until equal volume flow is achieved at each.

A static regain method is sometimes recommended for long supply ducts with many take-off branches. The idea is to maintain constant static pressure at all take-off points in the main duct thereby minimising the damper losses. The main duct is resized for a lower velocity after each take-off and it is assumed that about 25% of the fall in velocity pressure will be lost at each junction, the remaining 75% being "regained" as a rise in static pressure. The velocity reduction is chosen so that the regain just balances the loss of total pressure at the junction plus the duct friction of the section.

Fuller data on branch and junction losses is now available, see Figs. 6.11 to 6.15, and we may judge by how much (if at all) the main velocity may exceed the branch velocity without incurring excessive junction loss. See the last paragraph of Section 6.9 for an example of satisfactory balance.

6.8 Influence of Density on Total Pressure Drop

It is a good plan always to calculate the total pressure drop in an airway system, Pr first on the assumption that it is handling air at 1.20 kg/m^3 standard density. This will give the correct volume flow rate irrespective of the actual density (or gas) when used in conjunction with the standard fan characteristic as described in the next chapter. The reason is that both fan total pressure and total pressure drop are equally proportional to density at constant volume flow, and therefore remain in balance as density changes.

At any density, ρ kg/m^3, the total pressure drop is:

$$P_f = \frac{\rho}{1 \cdot 2} \times (P_f \text{ at } 1 \cdot 2 \text{ kg/m}^3)$$

Bearing in mind that Pi estimation to plus or minus 10% is good, the influence of density can be ignored in the majority of systems at low and medium fan pressure which are open to the atmosphere. Correction is required with high pressure fans, at high altitude where the barometric pressure is low, and in closed circuits containing air or other gas, probably at non-standard temperature and pressure. In these cases the density can be calculated from one of the following formulae:

$$\text{For air } \rho = \frac{347}{273 + t} \times \frac{p_a}{100{,}000} \text{ kg/m}^3 \qquad (65)$$

$$\text{For other gases } \rho = \rho_s \times \frac{347}{273 + t} \times \frac{p_a}{100{,}000} \text{ kg/m}^3 \qquad (66)$$

In these formulae it is sufficient to take the temperature t°C and the absolute pressure p_a pascals at the average of their values round the system. Remember that $p_a = p_o + p_s$ for circuits open to the atmosphere, p_o being the barometric pressure in pascals (= 100 x millibars). Also that the pressure in a closed circuit may be quoted as a "gauge pressure", in which case the barometric pressure must be added to obtain p_a.

For gases other than air p_s is the value at standard temperature and pressure relative to air as given in Table 14.11. The standards are conventionally p_o = 101,325 Pa (760mm of mercury) and 20°C. For ease of calculation they are here taken as p_o = 100,000 Pa and t = 16°C, at which all densities remain standard. For precise density calculation, taking into account humidity, see Chapter 10.

Example 1: An air conditioning plant in Johannesburg (altitude 1700m) circulates air at 25°C through a high velocity supply system with a total pressure drop of 2,000 Pa, calculated at ρ = 1.2 kg/m^3.

The normal atmospheric pressure at 1700m is given in Table 14.5 as 82,500 Pa. The average static pressure in the system can be taken as:

$$p_s = (2{,}000 + 0)/2 = +1{,}000 \text{ Pa}$$

$$\text{Therefore } p_a = 82{,}500 + 1{,}000 = 83{,}500 \text{ Pa}$$

$$\text{Therefore } \rho = \frac{347}{273 + 25} \times \frac{83{,}500}{100{,}000} = 0{\cdot}97 \text{ kg/m}^3$$

$$\text{and} \quad P_f = 2000 \times \frac{0{\cdot}97}{1{\cdot}20} = 1620 \text{ Pa}$$

Further correction for average p_s = 810 Pa and p_a = 83,310 Pa is not worth making.

Example 2: A process plant circulates carbon dioxide at a gauge pressure quoted as 1.5 kgf/cm^2 at 120°C. The total pressure drop as calculated at 1.2 kg/m^3 is 3,000 Pa. What is the pressure drop under working conditions?

1.5 kgf/cm^2 = 15,000 kgf/m^2 = 147,000 Pa gauge pressure. Taking p_o = 100,000 Pa barometric pressure gives p_a = 247,000 Pa absolute pressure, which will be assumed to be the average round the system.

Table 1411 gives the density, ρ_s of carbon dioxide as 1.53 relative to air at standard temperature and pressure.

$$\rho = 1{\cdot}53 \times \frac{347}{273 + 120} \times \frac{247{,}000}{100{,}000}$$

$$= 3{\cdot}34 \text{ kg/m}^3$$

$$P_f = 3000 \times \frac{3{\cdot}34}{1{\cdot}20} = 8350 \text{ Pa}$$

6.9 Example of System Pressure Drop Calculation

Fig. 6.19 illustrates a ventilation system to supply fresh warm air to a factory building. A total volume flow of 16m^3/s is to be supplied at 2m^3/s each through eight overhead outlets in line of which four are shown. The fan proposed is a 1250mm 720 rev/min adjustable pitch model. What fan total pressure is required?

The results of the successive stages of calculation are tabulated below the diagram. At each section the following are determined: the area A - the intended volume flow Q - the mean air velocity V - the corresponding velocity pressure p_v - the length L of the section, where applicable. The loss factor K or gradient g is then estimated from the information in Section 6.6 and multiplied by p_v or L to give the contribution of the section to the overall pressure drop. The steps in this process are commented on, line by line, below.

(a) Proprietary weatherproof louvres are proposed with a rated loss of 50% of p_v in the duct.

(b) Fig. 6.7 applies. A_2/A_1 = 3.60/1.96 = 1.84.

D_e = 2 x 1.4 x 1.4/(1.4 + 1.4) = 1.40m. Therefore L/D_e = 0.54 and K = 0.15.

(c) From the rules for a truncated diffuser in Section 6.6.3 the additional loss factor will be $(1.96/3.60)^2 - (1.96/6.25)^2 = 0.20$.

(d) The pressure drop across the battery is quoted by the makers as 57 Pa for a face velocity of 2.56m/s.

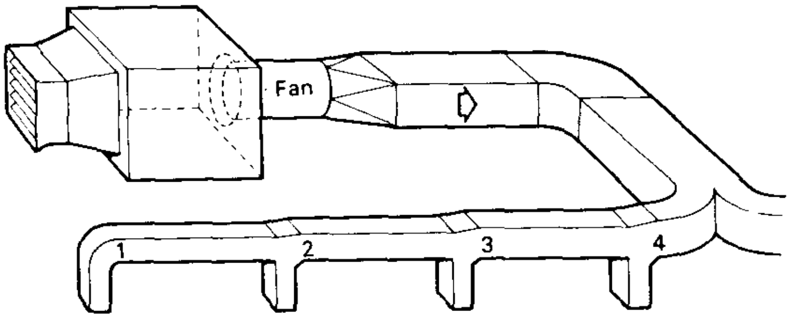


Fig. 6.19 Supply ventilation system.

	System element	Outlet	A	Q	V	P_v	L	$K \times p_v$ or Pa/m	Pressure Drop
		m×m	m²	m³/s	m/s	Pa	m		Pa
a	Intake louvres	1·4×1·4	1·96	16	8·16	40	—	0·5×40	20
b	Taper expansion to	1·9×1·9	3·60	16	4·45	12	0·75	0·16×40	6
c	Sudden expansion to	2·5×2·5	6·25	16	2·56	4	—	0·20×40	8
d	4 banks finned tube	2·5×2·5	6·25	16	2·56	4	—	—	57
e	Fan intake fairing	1·25 diam	1·23	16	13·0	102	0·25	—	—
f	Outlet transition to	1·0×1·2	1·20	16	13·3	106	1·25	·05×106	5
g	Straight duct length	1·0×1·2	1·20	16	13·3	106	20	1·4 Pa/m	28
h	90°, 0·6m R, bend	1·0×1·2	1·20	16	13·3	106	—	0·26×106	28
i	Straight duct length	1·0×·12	1·20	16	13·3	106	5	1·4 Pa/m	7
j	90°, 0·3m R, tee to	1·0×0·65	0·65	8	12·3	90	—	0·20×90	18
k	Straight duct to 4	1·0×0·65	0·65	8	12·3	90	4	1·8 Pa/m	7
l	Straight duct 4 to 3	0·75×0·65	0·49	6	12·3	90	10	2·0 Pa/m	20
m	Straight duct 3 to 2	0·55×0·65	0·36	4	11·1	74	10	2·1 Pa/m	21
n	Straight duct 2 to 1	0·3×0·65	0·19	2	10·2	63	10	3·0 Pa/m	30
o	90° chamfered bend	0·3×0·65	0·19	2	10·2	63	—	0·2×63	13
p	Straight down duct	0·3×0·65	0·19	2	10·2	63	1	3·0 Pa/m	3
q	Outlet diffuser	—	—	2	10·2	—	—	adjusted	45
r	Total pressure drop								316

(e) The fairing is that provided for an open inlet fan and any loss is allowed for in the fan performance.

(f) K would only be 0.01 for a 2½% sudden contraction without change of shape (Fig. 6.3). Allow a nominal K = 0.05 for the transition from round to rectangular.

(g) Fig. 6.18 gives D_q = 1200mm as the equivalent diameter of a 1000 x 1200mm section. Fig. 6.17 gives g = 1.4 Pa/m at Q = 16 and D = 1200. 1.4 x 20m = 28 Pa.

(h) Fig 6.10 applies. R/A = 0.6/1.2 = 0.5. B/A = 1.0/1.2 = 0.83. Therefore K = 0.26.

(i) As item (g) g = 1.4 Pa/m and 1.4 *x* 5m = 7 Pa.

(j) The data in Section 6.6.5 for low loss right angle tees applies. V_1/V_3 = 12.3/13.3 exceeding the minimum specified as 0.9. r/b = 0.3/0.65 is slightly better than the 0.5 specified. At Q_1/Q_3 = 8/16 = 0.5, K_{31} = 0.2.

(k) From Fig. 6.18, 650 x 1000 gives 880mm D_g. From Fig. 6.17, Q = 8 and D = 880 gives g = 1.8 Pa/m and multiplication by 4m gives 7Pa. Similarly:

(l) D_q = 770. Q = 6. g = 2·0 Pa/m. P_f = 20 Pa.

(m) D_q = 650. Q = 4. g = 2·1 Pa/m. P_f = 21 Pa.

(n) D_q = 470. Q = 2. g = 3·0 Pa/m. P_f = 30 Pa.

(p) D_q = 470. Q = 2. g = 3·0 Pa/m. P_f = 3 Pa.

(o) From Fig. 6.10 at *A = 0.*3m. *B = 0.*65m, taking the 0.15m chamfered bend as equivalent to 0.5 *R/A* with *B/A* = 2.2 gives K =0.20.

(q) *A* proprietary adjustable pressure loss outlet diffuser is to be used with a minimum drop equal to 50% of the approach velocity pressure. We can check that this is suitable for equalisation of branch volume flow with the aid of Section 6.6.5. Fig. 6.14 applies to the present case with the following results for successive junctions, the Pv to which K_{31} applies being 63 Pa in all cases.

	A_1	A'_3	A_1/A_3	Q_1	Q_3	V_1/V_3	K_{31}	Δp_{31}
At junction 4	0·195	0·650	0·30	2	8	0·83	1·4	88 Pa
At junction 3	0·195	0·487	0·40	2	6	0·83	1·2	75
At junction 2	0·195	0·358	0·55	2	4	0·92	0·9	56
At bend 1	0·195	0·195	1·00	2	2	1·00	0·2	13

We may note in passing that we were correct to ignore junction losses in the main duct, K_{32} being negligible for all cases in Fig. 6.14. This conclusion is reinforced by the successive reductions in main duct area, which reduce the decelerations which would be the cause of any loss.

A table can now be constructed to determine the pressure drop to the outlet of each branch. The volume flow must remain the same as assumed. Pressure differences are taken up by setting the outlet diffuser for increased drop, the minimum of 0.50 pv_1 = 32 Pa occurring at junction 2.

At the branch from junction	4	3	2	1
P_f for common items a to k	184	184	184	184 Pa
P_f for straight ducts l, m and n	0	20	41	71
Δp_{31} from main to branch	88	75	56	13
P_f in the down duct	3	3	3	3
Outlet diffuser as adjusted	41	34	32	45
Total pressure drop	316	316	316	316

CHAPTER 7

Fans and fan performance

Fans are built in widely differing types and designs, with performances just as various. Yet, in theory at any rate, one and the same duty could be met by fans to any of these designs, given the right size and speed. This chapter deals with those features of performance and construction which direct the choice to a limited number of alternatives.

While it is not the purpose of this book to enter the province of the fan designer, an elementary treatment of the mechanism of pressure generation is given at the end of the chapter. It can be passed over by those with a distaste for even a little mathematics, and it is to be hoped that they will be satisfied by the purely descriptive explanation in Clause 7.5.

7.1 What is a Fan?

A fan is defined as a rotary, bladed, machine maintaining a continuous flow of air. Continuous because the air flows steadily into, through, and out of the fan, a feature distinguishing it from positive displacement (piston, vane or lobe) machines, which generally produce a pulsating flow. Afan has a rotating impeller invariably carrying blades of some kind. These blades exert force on the air, thereby maintaining the flow and raising the (total) pressure.

The pressure rise is low or moderate in a fan, distinguishing it from a compressor, where the pressure rise is comparatively great. There is no hard and fast dividing line, though fans do not normally raise the absolute pressure more than 30% (30,000 Pa fan total pressure with standard air at inlet). Compressors and high-pressure fans are sometimes called *blowers* if they are used to take in atmospheric air and raise its pressure, exhausters if they suck in at low pressure, discharging at atmospheric pressure, but this distinction does not correspond to any difference in the fans themselves.

A distinct subclass is described by the term circulating fan. This includes ceiling fans, table fans and pedestal fans, and is distinguished by the absence of a fan casing or of any barrier between the inlet and outlet sides. They, therefore, generate no usable pressure, their function being to deliver a jet of air along the discharge axis. This jet entrains a much larger volume of surrounding air which circulates around the room in which the fan is placed. See Chapter 3 for a discussion of jet action.

7.2 The Fan Characteristic

Fan performance cannot be adequately described by single values of pressure rise or volume flow. Both quantities are flexible, but they have a fixed relationship with one another, and this relationship is best defined graphically in the form of a *fan characteristic*. Fig. 7.1 is an example; the volume flow rate is always plotted along the base, with the fan total pressure (and other performance quantities) as ordinates.

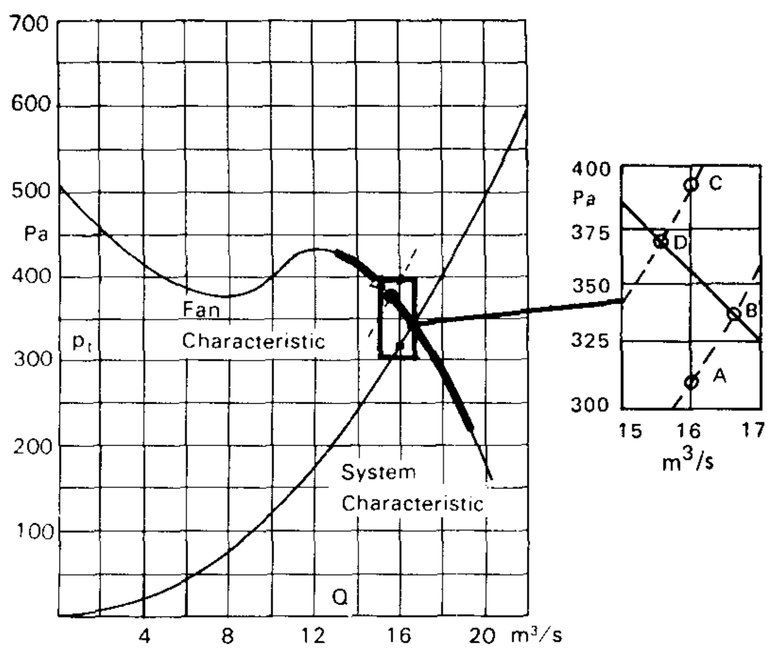


Fig. 7.1 Characteristic curves.

One point on the characteristic can always be found at which the efficiency of the fan is a maximum. This is called the best efficiency point-d on the diagram. As well as providing the lowest power consumption for a given duty, operation at this point usually secures the lowest noise level for that particular fan.

However, the fan can be successfully operated at other points on the characteristic. The region of satisfactory operation should be defined by the manufacturer as the guaranteed performance range, and such a range is indicated on Fig. 7.1 by the heavy line part of the characteristic. Outside this range the performance may be uncertain and is likely to be commercially unacceptable for efficiency, noise or cost. Part of the characteristic may also be positively forbidden because operation there would cause overloading or undercooling of the electric motor fitted, or vibration associated with unsteady flow.

7.3 The System Characteristic and Operating Point

Chapter 6 showed how to find the total pressure drop caused by the resistance of a system of airways to the required volume flow rate. These two quantities can be plotted on the same graph as the fan characteristic to show the desired duty point A. In general the fan characteristic will not pass through A-though it can be made to come very close when selecting from an adjustable pitch axial or adjustable speed centrifugal series. What will happen if this fan is installed in this system?

To solve this problem we assume that the loss factors K for the various items in the system remain constant when the flow rate is varied. The losses of total pressure are then proportional to the various velocity pressures, and each of these, and therefore their sum when they are added together, Pr, is proportional to the square of the volume flow rate. This relationship can be shown on the graph by plotting the line marked "system characteristic" which passes through the duty required at A and follows the "square law"

$$\frac{P_f}{P_{fA}} = \left[\frac{Q}{Q_A}\right]^2$$

At the point B where the system characteristic intersects the fan characteristic the operating conditions for both the fan and the system are met. B is therefore the operating point.

7.3.1 Example

Consider the airway system of section 6.15 in the last chapter. The desired duty point is at 16.Om3/s and 316 Pa, as plotted at A in Fig. 7.1. The square law system characteristic is then drawn. The fan characteristic in this figure is for a 1250mm diameter adjustable pitch axial fan, running at 12 revolutions per second and set at a pitch angle allowing 10% margin over the estimated total pressure drop at 16M3/s. If the system resistance is exactly as estimated, the operating point will be at B where:

The volume flow is	$16{\cdot}6m^3/s$
The total pressure drop is	340 Pa
The fan total pressure is	340 Pa

It should be noticed that the mutual accommodation of the fan characteristic and the system characteristic to one another has a strong stabilising effect on the performance. Thus suppose that the total pressure drop in the example had exceeded the estimate by 25%, becoming 395 Pa at $16.0m^3/s$ - point C in place of A. The dotted system characteristic through C cuts the fan characteristic at D where the volume flow becomes $15.5m^3/s$ at 370 Pa. This is a loss of only 7% from $16.6m^3/s$, resulting from a system resistance error of 25%.

7.4 Fan Static Pressure and Fan Velocity Pressure

In Fig. 7.2 four more curves have been added to give a set of complete fan characteristics. They are called complete when they run from free flow (zero P_s) to no flow (zero Q). The parts outside the guaranteed performance range of Q are given for information only, plotted from test results of progressively increasing uncertainty.

The fan total pressure is divided into two parts: the *fan velocity pressure*, which is a conventional measure (see 6.6) of the kinetic energy at fan outlet, and the *fan static pressure*, which represents the work done in compression. The fan static pressure is fully available to the user, but the fan velocity pressure suffers from the handicap that some of it will inevitably be lost, although this loss appears as part of the system pressure drop when calculated by the method in the last chapter. At an open fan outlet, or at the end of a duct system kept at fan outlet area all the way, the loss is 100% of the fan velocity pressure. If the duct area is gradually expanded, or a diffuser (conical expander) fitted to the fan outlet, the loss can be limited by careful design to 25% of the fan velocity pressure. Most practical cases lie between these extremes.

In order that the user may know what he is getting, it is usual in the English-speaking world to quote or plot both the fan total pressure and the fan static pressure; the fan velocity pressure can then be found by subtraction. Alternatively, one of the pressures may be given, together with the outlet velocity or area from which the others can be calculated. In some European countries, however, only one "fan pressure" is recognised. This is usually equal to our "fan static pressure" for an open outlet fan or to our "fan total pressure" for a ducted outlet fan. It seems to the writer illogical that 100% of the fan velocity pressure should be credited to a fan connected to a duct, where some is bound to be lost, and none to an open outlet fan when some can be recovered, and no more need be lost.

1250mm Axial Fan. Free Inlet, ducted outlet. 12 rev/s 1·20 kg/m³

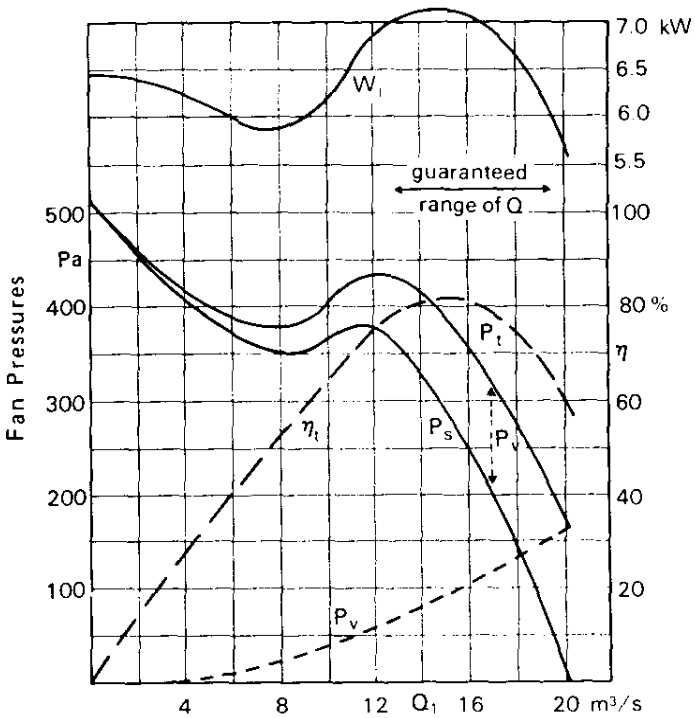


Fig. 7.2 Complete fan characteristics.

Q_1 is the volume flow rate at inlet density.
P_t is the fan total pressure, which is the rise in total pressure from inlet to outlet, as measured in a standardised test circuit.
P_v is the fan velocity pressure which is calculated from the average forward velocity over the gross fan outlet area.

$$P_v = \tfrac{1}{2}\rho \left[\frac{Q_2}{A_2}\right]^2 \qquad (71)$$

P_s is the fan static pressure which is equal to the fan total pressure less the fan velocity pressure.

$$P_s = P_t - P_v \qquad (72)$$

W; is the power supplied to the fan impeller by the driving motor.
η_t is the fan total efficiency, which is the ratio of work done to work supplied, calculated as follows:

$$\eta_t = \frac{Q_1\ (m^3/s) \times P_t\ (Pa)}{W_i\ (watts)} \qquad (73)$$

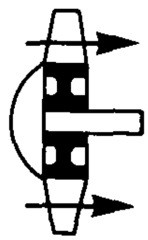
multiplied by 100 if required as a percentage. At high pressures the expression needs modifying for the effects of compression - see Chapter 11.

7.5 The Main Types of Fan

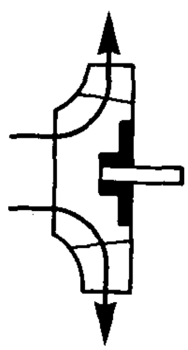
Fans can be broadly classified according to the path taken by the air through the impeller and the resulting mechanism of pressure generation. All fans work by aerodynamic action in so far as this implies the action of forces exerted by the impeller blades on the air.

The total pressure rise takes place wholly within the volume swept by the rotating blades. Neglecting losses the total pressure is constant up to the blade entry, and constant again downstream of the blade exit.

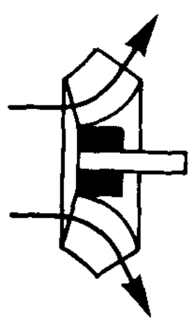
7.5.1 Axial flow. In a pure axial fan the effective progress of the air is straight through the impeller at a constant distance from the axis. The primary component of blade force on the air is directed axially from inlet to outlet, and thus provides the pressure rise by a process that may be called direct blade action. The blade force necessarily has an additional component in the tangential direction, providing the reaction to the driving torque: this sets the air spinning about the axis independently of its forward motion.



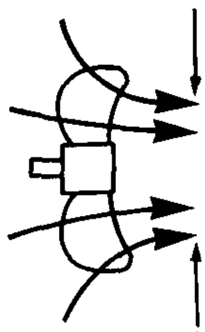
7.5.2 Radial flow. In a pure centrifugal fan the air enters the impeller axially, turns through 90°, and progresses radially outwards through the blading. The blade force is mainly tangential, causing the air to spin with the blades. The centrifugal force resulting from the spin is now in line with the outward progress of the air and is the main cause of the rise of pressure. There may or may not be an outward blade force component adding pressure by blade action.



7.5.3 Mixed flow. In a mixed flow fan the air enters axially and turns outwards through an angle which may range from 30° to 90°. An essential feature is that the impeller blading extends over the curved part of the flow path. In this region the blade force has a component in the direction of the arrow on the diagram as well as a tangential component and the pressure rise is developed partly by blade action and partly by centrifugal action.

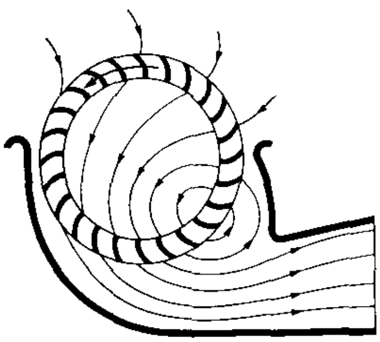


7.5.4 Orifice flow. This is characteristic of a propeller fan operating at zero or low fan static pressure. The air enters the impeller inwardly from all inlet-side directions, and curves round into a parallel stream at the *vena contracta* on the outlet side. The fan pressure is wholly developed by blade action. The total pressure rise required is the velocity pressure in the vena contracts plus any static pressure difference between the inlet and outlet sides.

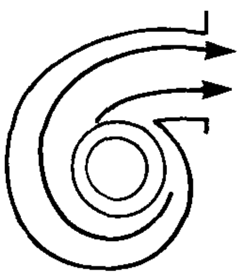


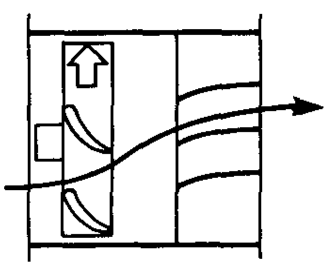
Vena contracta

7.5.5. Cross flow. The action of cross flow fans is radically different. A vortex is formed and maintained by the blade forces and has its axis parallel to the shaft and near to a point on the impeller circumference. The outer part of this vortex is peeled off and discharged through an outlet passage. An equal volume of entering air joins the inward flowing side of the vortex.



7.5.6 Outlet flow. The air leaving the impeller carries with it kinetic energy proportional to its velocity pressure. The component of this in the spin direction is unlikely to be effective when the fan is installed. It can be recovered for use by redirecting the velocity towards the fan outlet. A volute casing does this for centrifugal fans, and guide vanes for axial fans. The casing may also be shaped to allow a gradual enlargement of area as the outlet is approached. This diffusion does not increase fan total pressure, but it does convert some fan velocity pressure into fan static pressure.





7.6 Axial Flow Fans-General-Purpose

See Fig. 7.22 for performance comparisons with centrifugal fans.

7.6.1 The general-purpose range

Axial flow fans are well established for air conditioning and industrial process applications of all kinds. Manufacturers offer standard ranges of general-purpose fans commonly running from 300 or 400mm to 1800 or 2000mm diameter and displacing 0.5 to 50m^3/s at *pressures up* to 800 Pa or so. Standard units can be operated in parallel or series to extend the range of volume flow or pressure almost indefinitely.

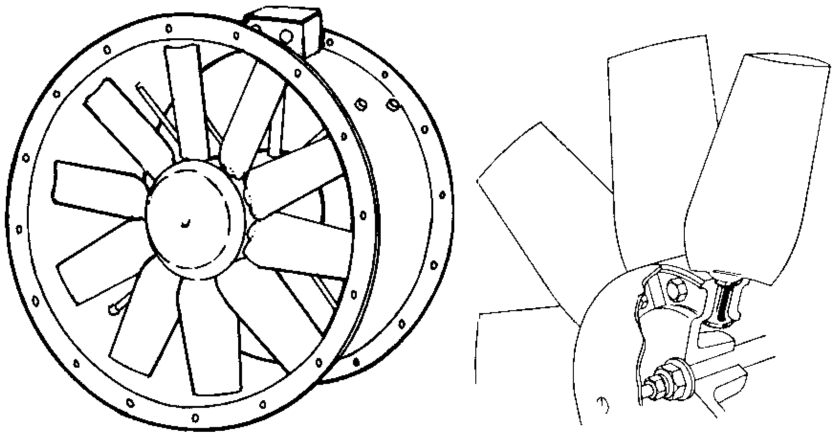
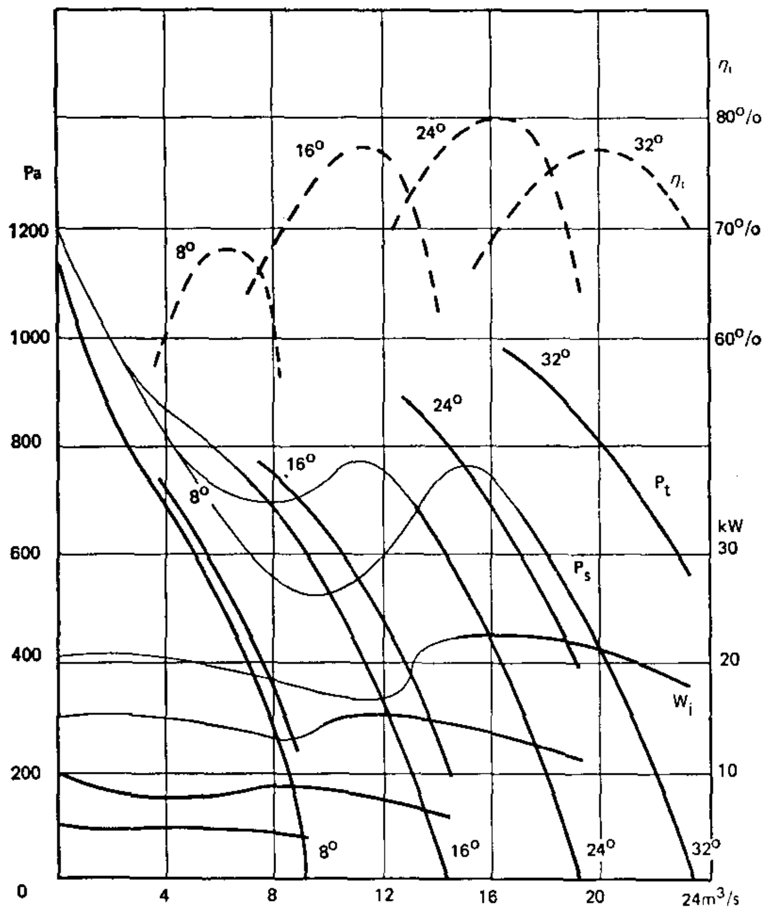


Fig. 7.3 General-purpose axial fan with adjustable pitch blades.

Impellers usually have blades cast in aluminium alloy with crosssections of aerofoil shape. As compared with curved sheet blades aerofoils can apply greater force to the air, thereby increasing maximum pressure, and can maintain better efficiency over a wider range of volume flow. Also by increasing the thickness and curvature of the inner sections the blades can be made stiffer; this limits flutter and allows the impellers to be run at higher speeds.

7.6.2 Pitch adjustment

The blades of general-purpose fans are often attached to the hubs by a system which allows the *pitch angle (see* Fig. 7.3) to be set at any desired value on assembly. This *adjustable pitch* feature enables the fan to be supplied pre-set for exactly the volume flow required, eliminating the waste of power inseparable from a range of fixed steps from which the next larger must be chosen. *Variable pitch* or controllable pitch fans are special models in which the blade angle can be continuously varied while the fan is running. They are naturally a great deal more costly, and their use is discussed in Chapter 9.4. Fig. 7.4 shows the range of pressure and volume flow covered by a typical adjustable pitch fan.



Performance at 8° 16° 24° and 32° pitch angle settings.

Fig. 7.4 1000mm 1475 rev/min adjustable-pitch axial fan.

7.6.3 Tip clearance

Axial flow fan impellers are generally mounted on the shaft extension of an electric driving motor. This is supported in a cylindrical casing surrounding the impeller with the minimum practical running clearance, a typical value for this (radial) clearance being 0-25% of the impeller diameter. Departures from the designed tip clearance will affect mainly the peak pressure development, and, therefore, the flow rate when the fan is operating close to peak pressure. The following table is an illustration for a particular fan, and should not be regarded as general.

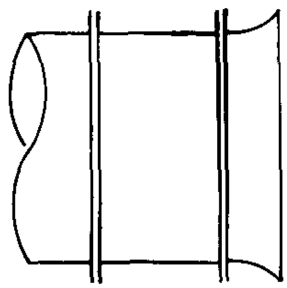
Radial clearance	0·1	0·2	0·25	0·3	0·4	0·6	1·0	% diameter
Peak fan pressure	114	104	100	97	92	85	75	% standard

At a normal operating point-say 85% of peak pressure - the loss of volume flow would only be about 2% at double the standard tip clearance. It is unwise to design to operate steep pitch axial fans much closer to peak pressure, as little margin would be left for under-estimation of the system resistance.

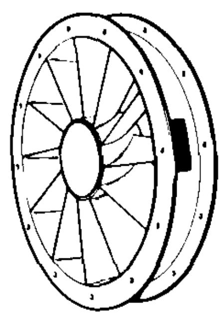
7.6.4 Variants on the standard range

General-purpose axial fans are particularly compact units with straight through flow and no wasted space. Accessories can be made available to modify the performance or adapt to particular installations. This practice can be illustrated from the Woods Aerofoil range.

(a) The standard casing normally has a flange for inlet duct connection. When operating without an inlet duct an inlet flare should be fitted to guide the air smoothly in without breaking away from the casing (see Fig. 6.31). To be effective the inlet needs a length of at least 15% and a radial depth at least 10% of the fan diameter.



(b) *Guide vanes* downstream of the impeller are sometimes incorporated in the fan as standard, sometimes as an optional extra. They will certainly improve the pressure development and efficiency (see 7.5.6) though no general rule can be given - it depends on the fan and on what is connected to the outlet. Reduction of outlet swirl is particularly important when discharging into along high velocity duct. Swirl persists for great distances in such a case, and can increase the pressure drop several times.

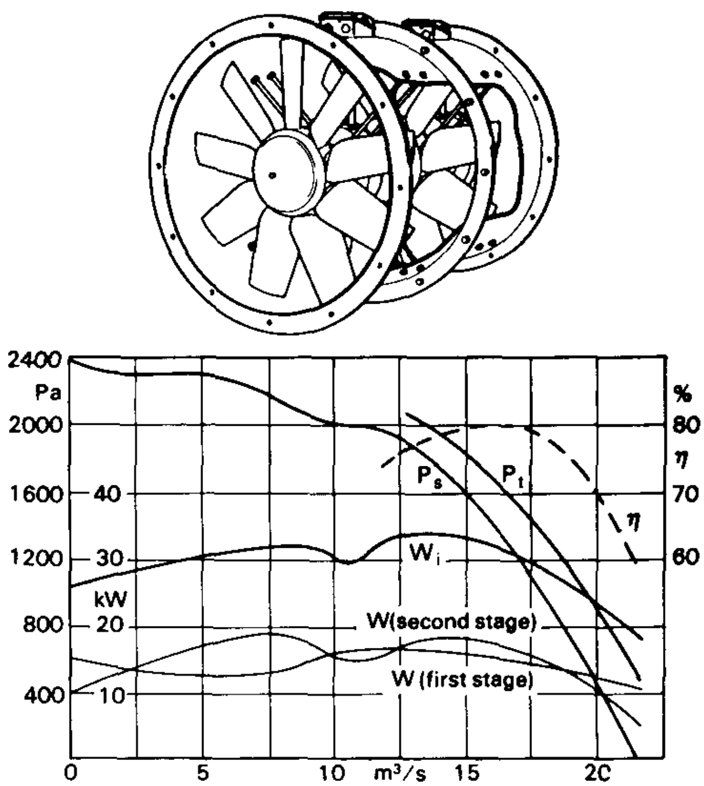


(c) *Upstream guide vanes* cause the incoming air to swirl in the opposite direction to the impeller rotation, serving to cancel out the swirl introduced by the impeller. They are not often used since they improve efficiency less than downstream guide vanes, and have an unfavourable effect on noise level. However, they do cause a substantial increase in peak pressure and will sometimes overcome a pressure limit set by a fixed motor speed.

(d) *Diffusers* will convert some of the fan velocity pressure into fan static pressure, which is the only useful form at an open outlet. They will reduce the system loss when used to connect the fan outlet to a larger duct or chamber. Their performance in these roles is very much dependent on the flow pattern at the fan outlet, and for this reason manufacturers will generally quote the combined performance of the fan and a diffuser of recommended dimensions.

(d) If two identical guide vane fans are run in *series* the fan total pressure would be approximately doubled at a given volume flow, apart from losses due to mutual interference, the fan velocity pressure remaining the same. Two non-guide-vane fans will only produce about 50% more pressure than one, since the second stage will be unloaded by the swirl produced by the first. The effect may be looked upon as a reduction in the rotational speed of the second impeller relative to the air entering it, which is spinning in the same direction.

(f) Contra-rotating fans provide a better system of series connection. Each impeller is driven by its own motor and they rotate alternately in opposite directions. Thus each impeller cancels out the swirl of the one before and guide vanes are unnecessary. Furthermore, the second, and all even-numbered, stage impellers receive air which is swirling in the opposite direction. This is equivalent to an increase in relative speed of rotation and the fan total pressure of a contra-



Set at 24° first stage pitch angle, 21° second stage

Fig. 7.5 1000mm 1475 rev/min contra-rotating axial fan.

rotating pair will be about two-and-a-half times that of a single stage.

The blades of the even-numbered and odd-numbered stages differ fundamentally from each other since one must be designed as a right-handed and the other a left-handed screw. Pitch angles are generally adjusted so that impeller powers are equal around the best efficiency point. This automatically secures an output flow free from swirl.

(g) *Reversal* of the direction of rotation of an axial fan reverses the direction in which the air flows. The performance of guide vane fans in reverse is extremely poor, but non-guide-vane and contrarotating fans will deliver 60% to 70% of the forward volume flow when reversed on a given system. The reduction is due to the fact that the aerofoil sections are operating tail-first and have their camber (curvature) in the wrong direction. A *truly reversible* impeller can be built from standard parts by rotating every other. blade through 180°. Half will then be running nose-first and half tail-first, the volume flow being about 85% of normal in each direction.

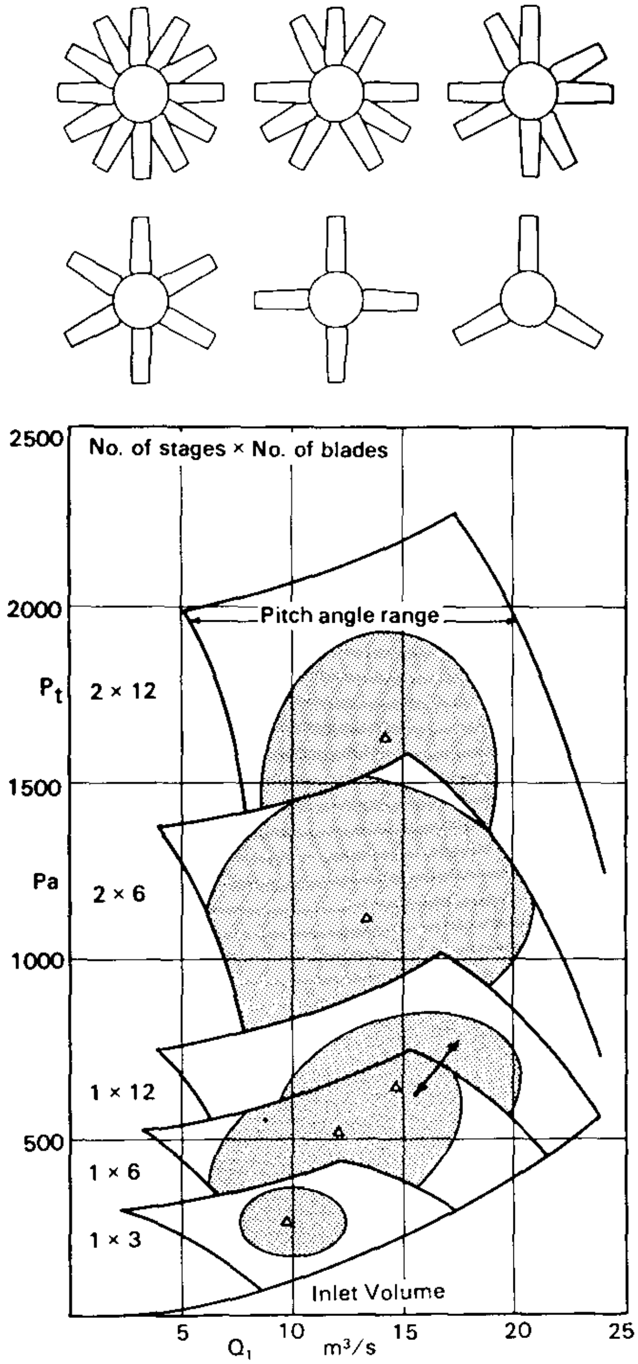
(h) *Fractional solidity* impellers can be assembled on a standard hub by omitting some of the blades. Mechanical balance must, of course, be preserved, but there is no need for the blades to be evenly spaced. Peak pressure is reduced and the best efficiency point moves to a lower pressure and volume so that the speed must be increased for a given duty. This can be an advantage when the impellers are directly driven by electric induction motors. Such motors have better efficiency and lower cost at higher speeds-a point which can be particularly significant with large low speed fans. Fig. 7.6 shows the performance range of fans with 12 left- or right-handed adjustable pitch blades, which could be assembled with 10, 9, 8, 6, 4, 3 or 2 blades, and multi-staged.

7.7 Axial Flow Fans-More specialised designs

7.7.1 High pressure axial fans such as that in Fig. 7.7 are designed with hub diameters between 50% and 70% of the impeller diameter, compared with 30% to 40% for a general-purpose range of competitive cost. Aerodynamically this reduces the pressure limitation set by the slowmoving roots of the blades. Mechanically the short blades can be made far stiffer so that the impeller can be run at higher tip speeds without danger of flutter. The ratio of the annular flow area to the total blade area decreases, making guide vanes or contra-rotation essential to recover the increased swirl energy.

7.7.2 High efficiency justifies axial fan designs of relatively high first cost when the power absorbed is measured in hundreds of kilowatts. Among features distinguishing such designs from the general-purpose type are:

(a) Hub diameters of 50% or more to improve the aerodynamic balance of the design from blade root to tip.



1000mm impeller. 24 rev/s. 12 blades (full solidity).
Efficiency exceeds 75% within the shaded area. Δ = Peak efficiency.

Fig. 7.6 Family of axial fans with fractional solidity.

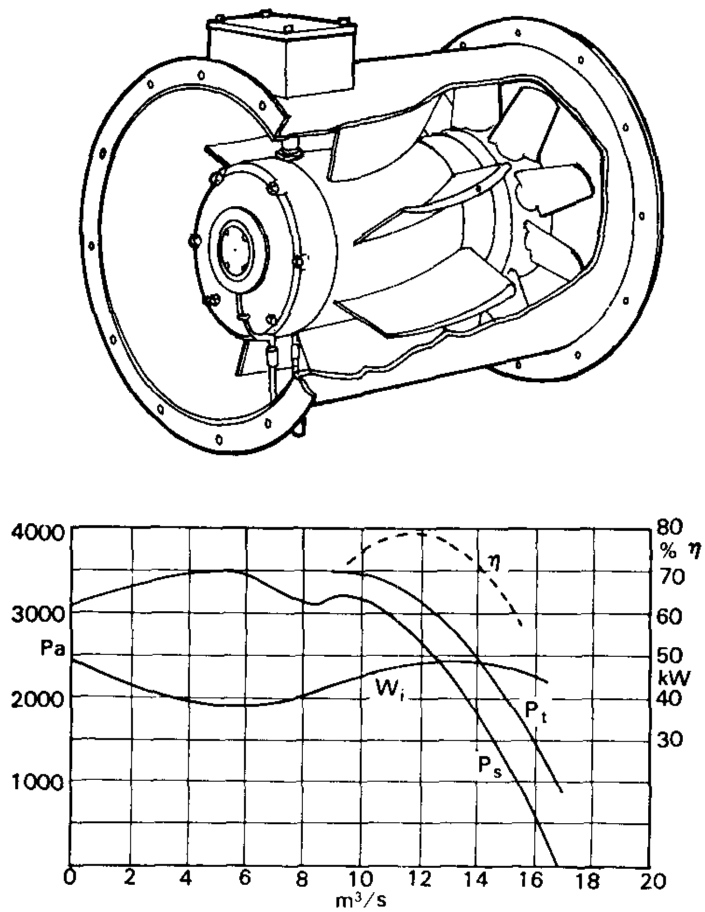


Fig. 7.7 800mm 2950 rev/min axial fan for ducted air supply during tunnel construction.

(b) Blade form designed specifically for the required duty. When dieforming is not justified, this entails increased labour to provide a good surface finish.

(c) Aerofoil-section guide vanes, again designed specifically for the required duty.

(e) Careful streamlining of the annulus passage, and fairing of bearing supports or other obstructions. This may entail moving the driving motor right out of the casing, introducing the necessary transmission elements to the impeller.

(e) Space for a long tail-fairing following the impeller hub and guide vanes to maximise fan total pressure by conversion of annulus velocity pressure.

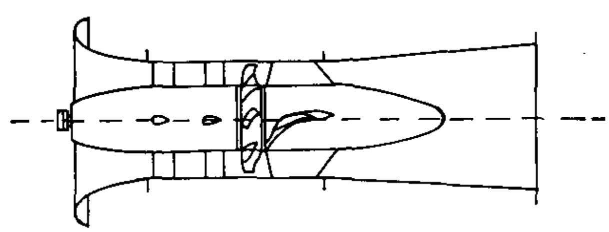
(f) Space for a long, gradually expanding diffuser to minimise outlet velocity pressure, and maximise fan static pressure.

These measures may raise the peak fan total efficiency to 90% compared with 80% for a good general-purpose model at optimum duty.

7.7.3 Small axial fans continue the general-purpose range downwards, but are more often manufactured with plastic impellers and sometimes casings also. In the 100 to 200mm diameter range the high pressure conformation is often used to counter the 50 or 60 rev/s speed limitation set by induction driving motors. In the aircraft field small axial fans of exceptionally high power-weight ratio are made, using the high induction motor speeds possible on a 400 Hz supply.

7.7.4 Low pressure axial fans are met with particularly in large sizes with volume flows from 50M3/s upwards at fan static pressure from 100 to 200 Pa. As an example they may be applied singly, discharging from the top of evaporative cooling towers, or in multiple, circulating air across extensive banks of heat exchange tubes.

Hubs are small and the blades long, slender, and few in number three, four or six. Blades were at one time of timber, but now hollow glass-reinforced polyester, or similar, mouldings, or hollow aerofoil sections from steel or aluminium sheet are more usual. Guide vanes are unnecessary.



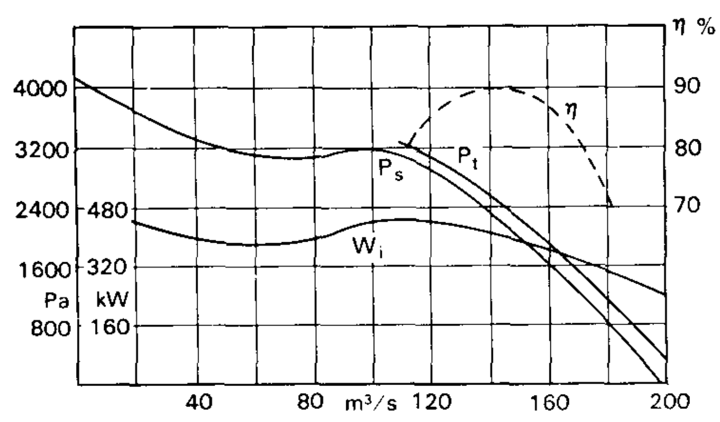
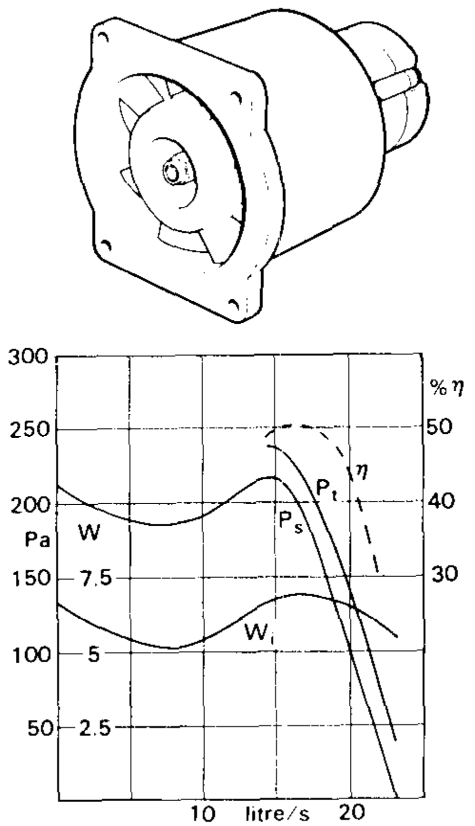


Fig. 7.8 2500mm 980 rev/min mine fan—3200mm outlet.



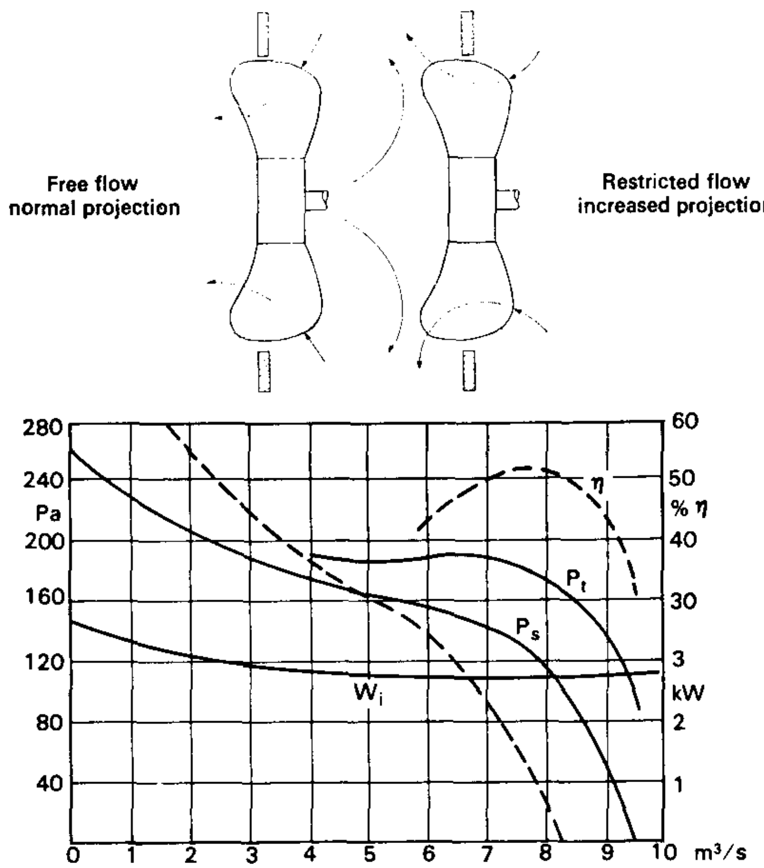
For 115 V 400 Hz 3 phase aircraft supply. The impeller is a polyester moulding and the casing, guide vanes, and motor carcase a one-piece aluminium casting.

Fig. 7.9 60mm 11000 rev/min axial fan.

7.8 Propeller Fans

These may be regarded as a special type of axial fan designed to operate without a casing, the impeller being situated in a hole in a wall or partition. The fans are simple low cost units with broad-bladed impellers usually formed from sheet metal. The blades are shaped to co-operate with an orifice flow pattern, deflecting the air with the minimum flow separation or vortex formation. Design may perhaps be considered as much an art as a science, though flow visualisation with stroboscopically viewed smoke trails can be helpful.

The blade form is usually optimised for pressure differences across the partition from zero to about *100 Pa.* Above the designed pressure the flow pattern changes drastically. The outlet jet assumes an expanding conical form with reverse circulation at its core as sketched. Towards zero volume flow, discharge is radially outwards, and the centrifugal mechanism is now responsible for pressure development.

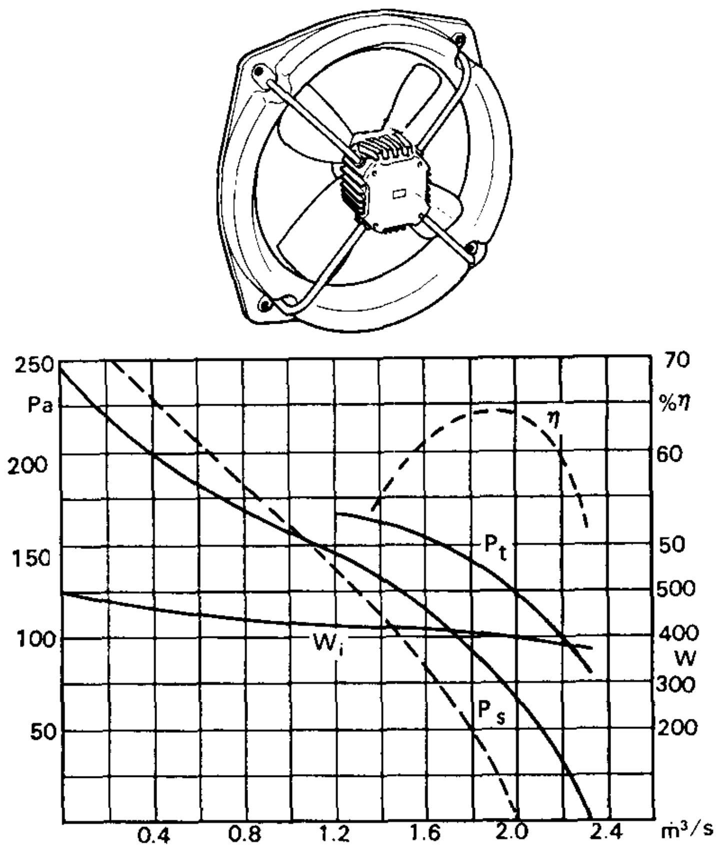


Effects on Ps of increasing downstream projection of impeller shown thus: – – –

Fig. 7.10 1000mm 710 rev/min propeller fan, diaphragm mounted.

Propeller fans are quiet and effective for ventilation purposes, both supply and exhaust. They are also used for unit heaters and similar applications where some resistance is encountered. For these an experimental matching of the fan and the unit is important since the pressure development and the flow pattern over the heat exchanger are very dependent on the blade and orifice plate positions.

As more of the impeller projects on the outlet side of the orifice, the free flow volume falls, because the inlet orifice flow no longer covers all the blade. At the same time the pressure at low flow rises, because more blade is exposed on the outlet side for centrifugal action. The free flow can be substantially increased by rounding the orifice edge or fitting a rounded inlet ring. This is because the vena contracta is expanded (see section 7.5.4) and less velocity pressure is required for a given volume flow. Moderate pressure performance is also helped, but high pressure development is impaired. If the rounding is enlarged into a true



Effect on Ps of omitting inlet ring shown thus: – – –

Fig. 7.11 500mm 1360 rev/min propeller fan with bell mouth inlet ring.

bell mouth and a short tunnel formed around the impeller, the fan becomes in effect an axial fan, and is better served by an aerofoil section impeller.

The impellers of propeller fans are almost invariably mounted on the shaft of the driving motor. The air flow cools the motor, which can be totally enclosed to keep out dust. The impeller power rises rather sharply if the volume flow is drastically restricted, and the motor could be overheated, particularly if on the downstream side, where centrifugal flow starves it of cooling air. However, propeller fans are not often used in systems where such excessive resistance could arise.

Table and pedestal fans may be fitted with very similar impellers to propeller fans. The orifice flow pattern is the same, though the zero fan static pressure condition only has to be considered and no plate orifice or ring is needed.

7.9 Centrifugal Fans-General-Purpose

The general-purpose centrifugal fan has a volute or scroll-shaped casing as illustrated in Fig. 7.12. The air enters through a circular inlet adapted for use with either open inlet from the atmosphere or duct connection. The outlet is rectangular, and intended for duct or diffuser connection only (without such a connection the performance would suffer). Three or four alternative impellers are often available, such as those described in 7.9.1, 7.9.2 and 7.9.3 with differing performance characteristics. The impeller may be direct driven by an external motor, but is more commonly belt driven for flexibility in speed selection.

See Fig. 7.22 for performance comparisons with axial fans.

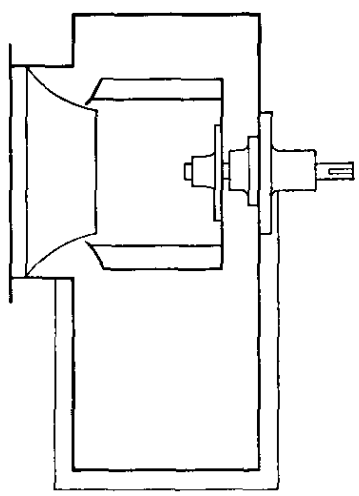


Fig. 7.12 Cross-section of general-purpose centrifugal fan.

7.9.1 Forward-curved multi-vane type

For upwards of three-quarters of a century the multi-vane impeller has been used for the established general-purpose type of centrifugal fan in small and medium sizes. It provides the most compact and probably the quietest form, and the most competitive in first cost. The chief disadvantage is a limited efficiency - 60% to 70% at most - and a steeply rising power characteristic towards free flow, necessitating a relatively high-powered driving motor.

Small impellers in the 50 to 250mm diameter range are commonly made from punched strip rolled into cylindrical form. Better performance is obtained in larger sizes with individual blades which can be fully formed to conform to the aerodynamic requirements. The blade crosssection is commonly a circular arc of 90° or so, the outside, trailing, edge making an angle with the radius from 45° upwards, forward in the direction of rotation. The air velocity relative to the impeller changes from backwards at blade inlet to forwards at blade outlet with little change in magnitude. However, the air leaves the impeller with a very high tangential absolute velocity - higher than the blade velocity. This means that a good deal of the pressure rise must come from velocity conversion in the volute casing.

With the usual proportions - impeller width about 50% and blade depth about 8% of the diameter - mechanical considerations limit the tip speed so that the maximum pressure is about 900 Pa. This can be

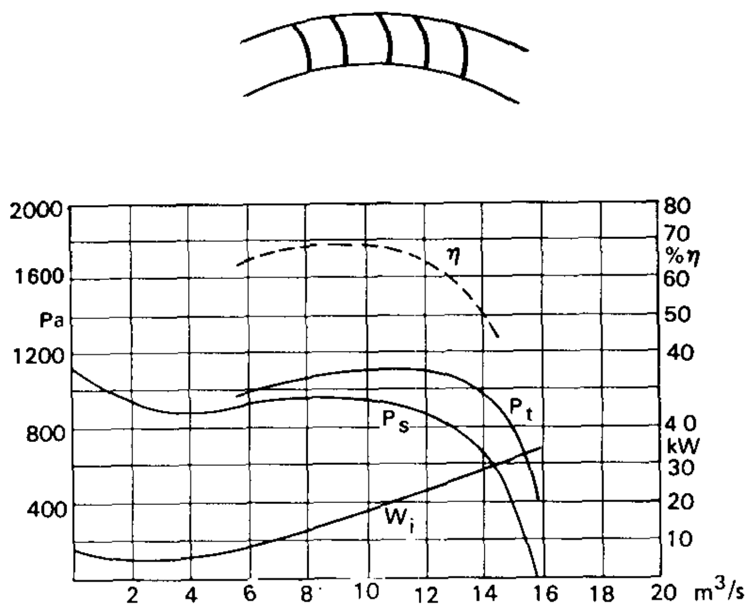
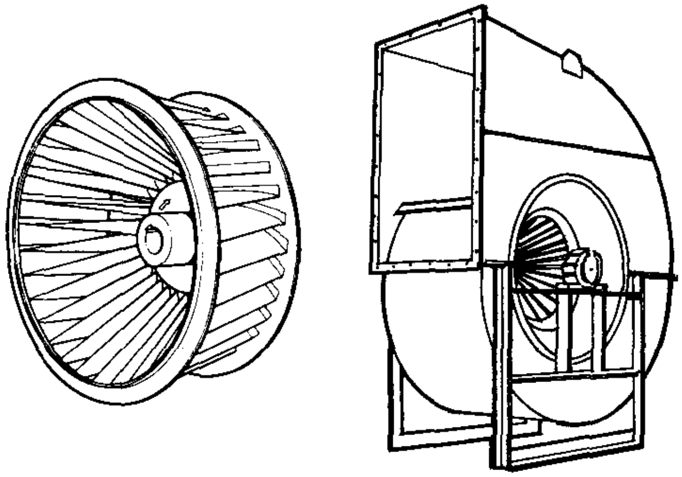


Fig. 7.13 850mm 600 rev/min forward-curved centrifugal fan with multi-vane impeller.

raised to 1500 Pa by strengthening the blades with a mid-span stiffening ring and spoke-type ties to the inlet ring. Alternatively, specially shaped forward-curved blades may be designed, typically of increased depth at the back plate end.

The double inlet fan comprises two impellers back to back in a double width casing and will generally deliver rather less than double the volume flow at the same pressure. This is best suited to free inlet, ducted outlet installations. In small direct motor-driven sizes, narrow width impellers serve to fill the performance gaps between diameter steps. Larger sizes are often belt-driven so that duty adjustment is by pulley and speed change.



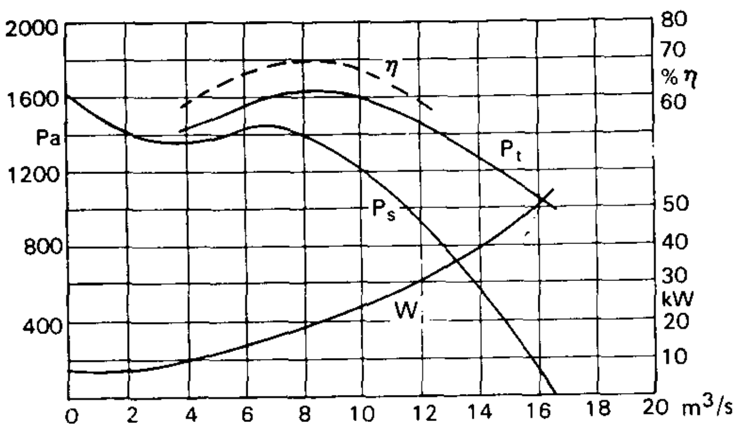


Fig. 7.14 700mm 1100 rev/min forward-curved centrifugal fan with blading of stiffened section.

7.9.2 Backward-curved impellers

In these the outer, trailing, edges of the blades are inclined backwards rather than forwards, and the curvature is reversed with respect to the direction of rotation. Again there is little change in the relative air velocity within the impeller, but there is also no reversal of direction, so that the air leaves the impeller with comparatively low velocity. For a given duty the casing is bulkier and the tip speed higher than the forward curved requires. However, the low velocities and, therefore, friction loss, together with avoidance of flow separation, make backward-curved fans much more efficient, 80% to 85% being attainable in general-purpose models.

The geometry of the blading permits fewer blades of much greater radial depth to be used, so that pressure development is largely by centrifugal action. Mechanically much higher tip speeds are possible and full-width general-purpose models will commonly develop up to 2500 Pa.

Another advantage is their non-overloading power characteristic, by which is meant a power input which does not peak at either free flow or no flow. These fans may be regarded as the standard industrial centrifugals in the larger sizes, where their aerodynamic performance justifies increased first cost.

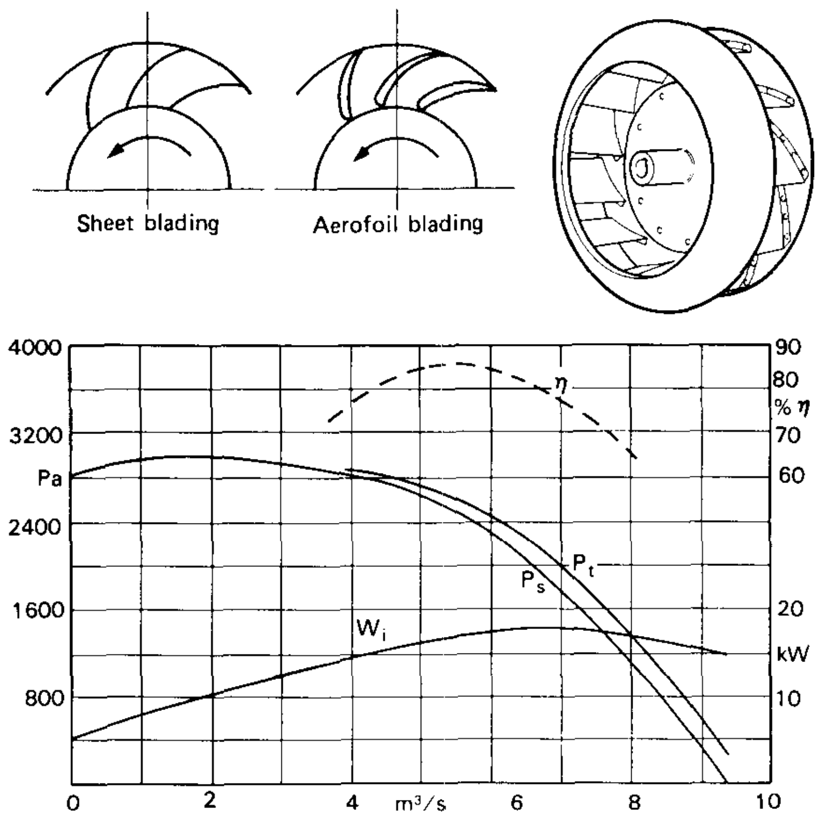
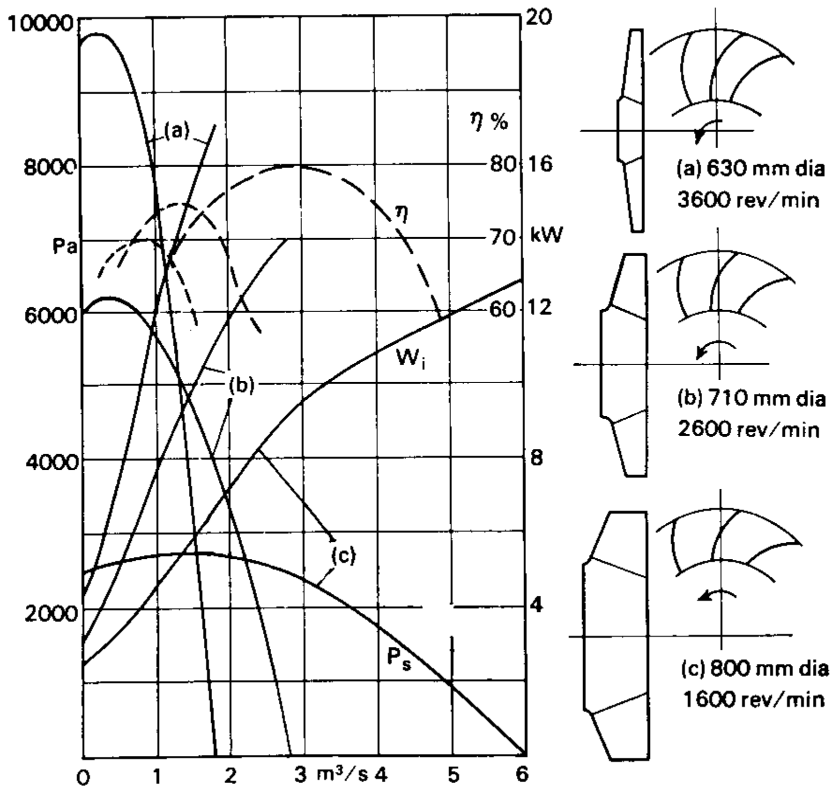


Fig. 7.15 700mm 2200 rev/min backward-curved centrifugal fan performance with impeller blades of aerofoil section.

7.9.3 Aerofoil-bladed impellers

The larger sizes of full-width backward-curved impellers may be manufactured with blading of aerofoil cross-section, formed from hollow sheet. Specially refined designs of this type may reach 90% efficiency and there is probably some advantage in efficiency spread over the characteristic and in noise generation. It is fair to say, however, that comparable results have been claimed for constant thickness blading of complex curvature to co-operate with the optimum streamlines as these change shape from back plate to shroud.



The sizes have been selected for equal output power at the maximum tip speed of each impeller.

Fig. 7.16 Three members of a family of backward-curved fans.

7.9.4 Increased pressure range

If the width of a backward-curved impeller is reduced the tip speed can be increased without overstressing the blades in bending. As the optimum volume flow is reduced the inlet eye diameter can be reduced also, increasing the radial depth of the blades and the pressure development by centrifugal action. A whole family of impeller designs with progressively increasing pressure rise and decreasing volume flow can thus be provided. The efficiency will fall away as the width is reduced, and the backward curvature becomes less necessary until, in the narrowest widths, radial blades become usual with the advantage of no bending stresses to limit the tip speed. Fig. 7.16 is an example of such a performance family.

7.10 Centrifugal Fans-more specialised designs

7.10.1 Radial and paddle-bladed fans

In the textile industry the air handled may contain lint, or waste strands of various kinds, which would catch in the blading and soon clog a backward- or forward-curved impeller. Widely spaced radial *blading* is much more satisfactory in such cases; the air performance is intermediate between the backward and forward types, but efficiency generally inferior to either. In the *paddle-bladed* impeller backplate and shroud are omitted also, the flat radial blades being supported on a central spider. These are still more effectively self-cleaning, and the blades are easily replaced if worn by abrasive particles in the air stream.

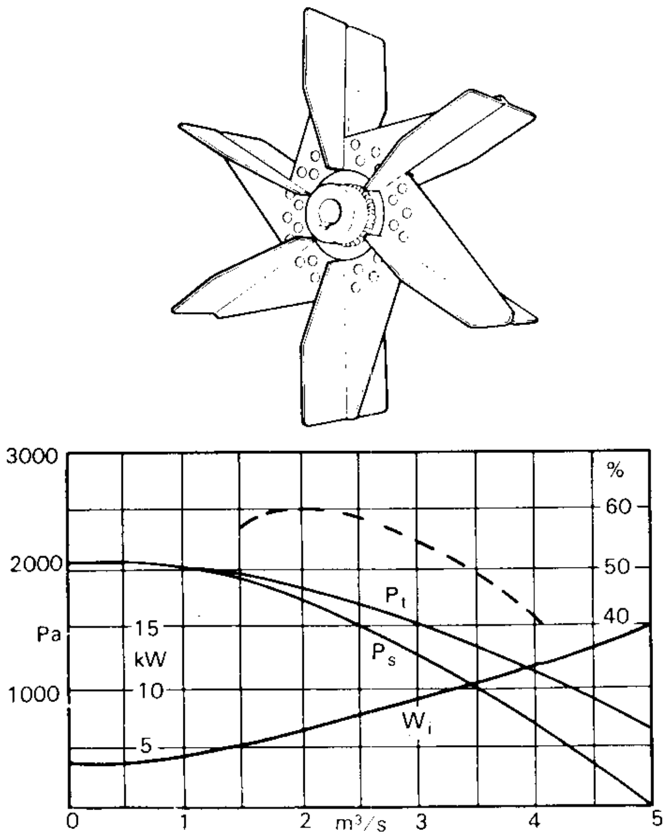


Fig. 7.17 500mm Paddle-bladed fan at 1460 rev/min for handling air carrying fibrous waste.

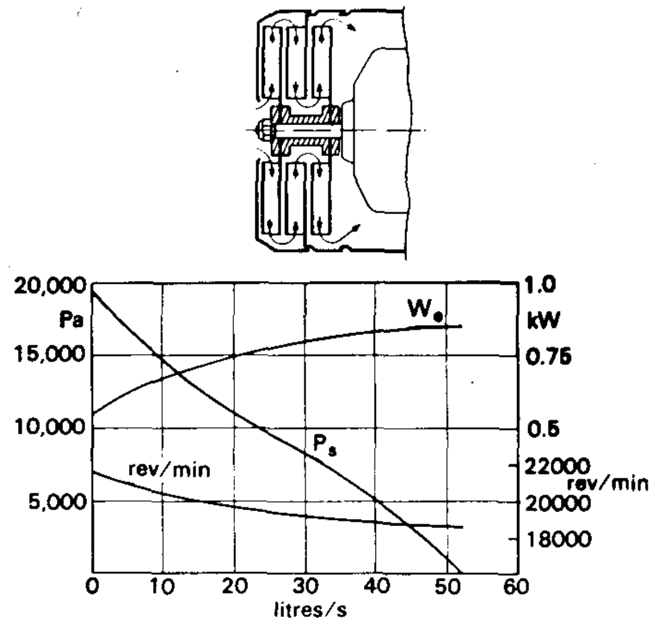
7.10.2 In-line centrifugal fans

Instead of using a volute casing to collect the swirling air from a centrifugal impeller, it may be allowed to spin forwards into a concentric annular casing. Guide vanes will then convert the swirl velocity pressure into fan static pressure, and an outlet duct can be fitted in line with the inlet duct as for axial fans. Performance tends to be somewhat inferior to the corresponding volute model, and the chief advantage is avoidance of the transverse bulk and right-angle direction change associated with a standard centrifugal fan.

7.10.3 Multi-stage centrifugal fans

Since the maximum pressure required of any fan can be provided in a single centrifugal stage, some special reason is needed for multistaging.

One such reason is economic, as illustrated by the example of the domestic vacuum cleaner. The need for low cost motors and impellers sets a limit to the tip speed, and hence the pressure development of one stage. The in-line centrifugal arrangement is adopted but, instead of putting guide vanes in the annulus, they are placed in a radial passage returning the air to the eye of the next impeller in the series.



Motor input watts and speed given for AC commutator driving motor.
Exhaust to atmospheric pressure.

Fig. 7.18 100mm two-stage centrifugal exhauster for domestic cylinder cleaner.

7.11 Mixed Flow Fans

7.11.1 Mixed flow with axial discharge

This type can be regarded as a modification of the axial fan in which the blades operate in a conically expanding passage, which is then curved back, through guide vanes, to an axial outlet. A substantial part of the pressure development is by centrifugal action and the fan static pressure is high compared with an axial fan at the same speed. With advanced design, efficiency and noise level comparable to those of a backward-curved centrifugal fan is obtainable, with a more compact, in-line, casing. They have not become widely popular however, perhaps because quantity production manufacturing techniques, necessary for competitive cost, are difficult to develop, while flexibility in pressure and volume duty is somewhat limited.

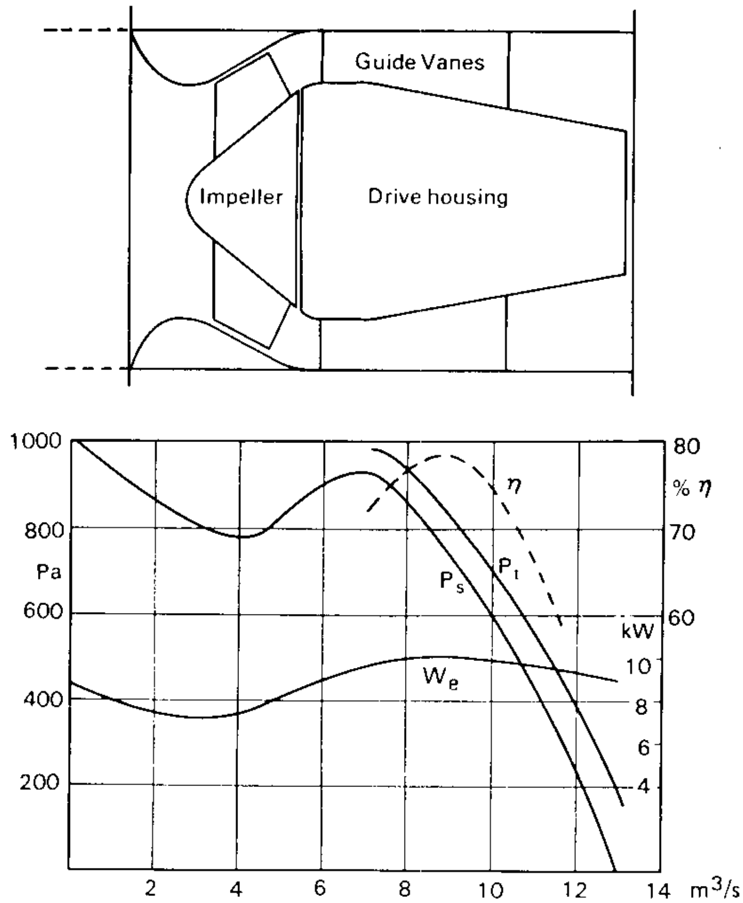


Fig. 7.19 1000mm 1100 rev/min in-line mixed flow fan.

7.11.2 Mixed flow with volute casing

Axial entry mixed flow blades can be extended into a radial flow centrifugal formation leading to a volute outlet casing. High pressure development can be combined with high efficiency given careful design. Performance is superior to that of the normal narrow-bladed centrifugal blower, giving the fan a place in the high power applications of heavy industry. In fact, the type is an extension into the fan field of the high strength, high tip-speed designs used for turbo-compressors and superchargers.

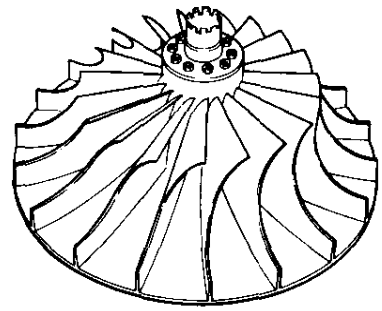


Fig. 7.20 Turbo-compressor rotor for a gas turbine.

7.11.3 Mixed flow with radial discharge

The roof ventilator is an example of a free outlet fan in which it is advantageous to discharge the air radially in all directions under a cowl excluding rain and snow. Propeller, axial, and centrifugal impellers are all used in roof units, but the mixed flow design is in many ways the ideal. No compromises for general use are necessary, the impeller, cowl and roof-mounting curb being designed as a single-purpose aerodynamic and mechanical unit. Pressures of 250 Pa or so to deal with long ducts or maintain duty in adverse winds can be developed more quietly than with axial fans. At the equally important zero pressure condition efficiency is better than with centrifugal fans, and the power characteristic can be completely non-overloading.

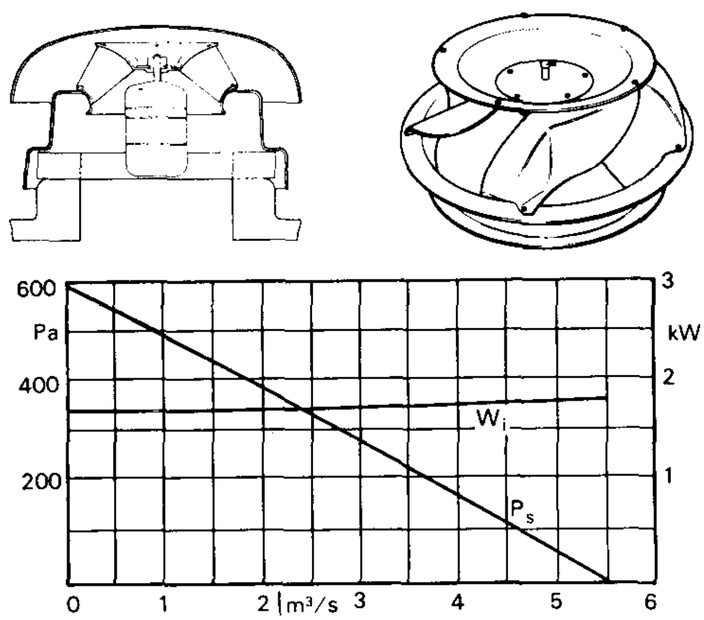


Fig. 7.21 630mm 940 rev/min mixed-flow roof extract unit.

7.12 Comparative Performances of General-Purpose Fans

Dimensionless coefficients and charts, with such concepts as "specific speed" help the fan designer in type selection, and are touched on in Chapter 11. The user, however, is concerned only with those fans which are economically available. One limitation this places on his choice is a maximum tip speed for each class above which construction costs tend to soar.

Fig. 7.23 illustrates this with typical examples from each of the main general-purpose fan types. To provide a basis of comparison each is chosen for an output, that is product of volume flow and fan total pressure, of 10,000 watts. It is clearly seen that they cater for very different pressure duties, though axial and centrifugal ranges overlap at moderate pressures. The range could be extended almost indefinitely with higher pressure centrifugals and higher volume axials.

The maximum power absorbed at any point on the constant-speed characteristic is listed to illustrate the difference between overloading and non-overloading types. Kilowatts absorbed at the 10 kW output duty are based on typical efficiencies for each class.

The fan diagrams indicate graphically, to the same scale, the relative space occupied, and all these comparisons hold of course for other sizes, speeds, and duties within the range of each type.

Selection of a fan below its top rated speed will increase space and cost per kilowatt of output, but has the advantage of quieter operations.

A Backward-curved
Half-width
630mm
42 rev/s
13·5 kW at Δ
17 kW at Ø

B Backward-curved
Full-width
630mm
36 rev/s
12 kW at Δ
14 kW at Ø

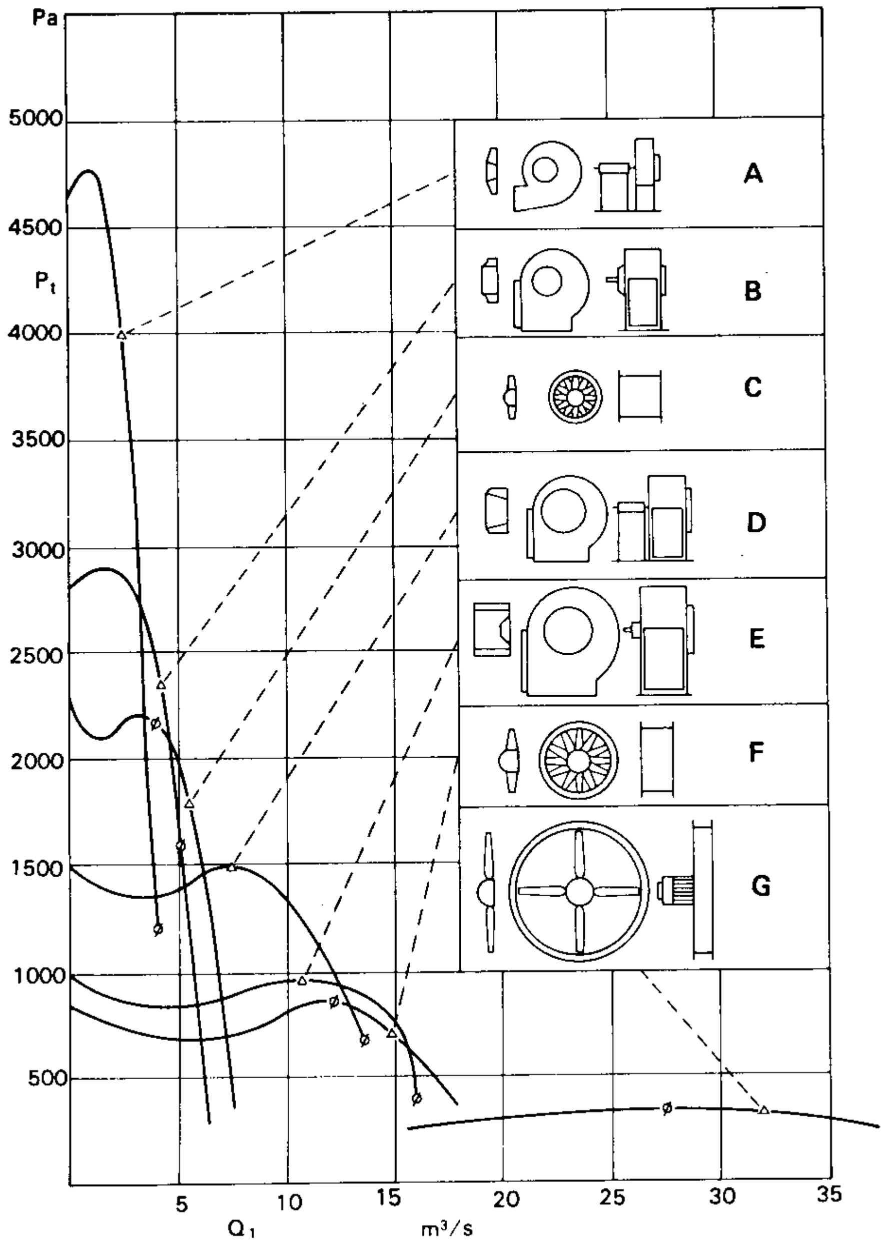
C Axial
50% hub
630mm
48 rev/s
13·5 kW at Δ
15 kW at Ø

D Forward-curved centrifugal
700mm
18 rev/s
15 kW at Δ
30 kW at Ø

E Multi-vane centrifugal
850mm
9 rev/s
15 kW at Δ
30 kW at Ø

F Axial
35% hub
1000mm
24 rev/s
13 kW at Δ
15 kW at Ø

G Axial
25% hub
2000mm
12 rev/s
12·5 kW at Δ
14 kW at Ø



Each fan, operating at top speed and best efficiency point Δ is chosen for an output $Q \times P_t = 10\,kW$.

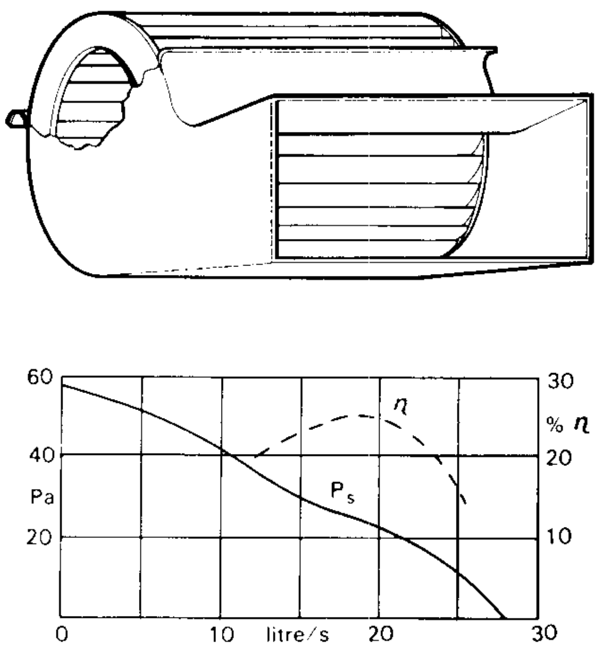
Peak input power is taken at Ø

Drawings are to a uniform scale of 1 : 120

Fig. 7.23 Comparative fan designs at equal output power.

7.13 Cross-Flow Fans

Improvements in casing design have brought this type into prominence for use in certain small domestic appliances. Impellers similar to those of multi-vane centrifugal fans are used, with the difference that they can be extended indefinitely in width, provided bearings are fitted at intervals to preserve mechanical stability. It therefore becomes possible to use inlets and outlets of long, narrow, rectangular shape, which opens up new possibilities of functional and appearance design for such



Driven at 1200–1400 rev/min by shaded-pole motor for domestic application.

Fig. 7.22 67mm diameter by 180mm long cross-flow fan.

appliances as table fans and electric fan heaters. Efficiency is low, but the fans are fairly quiet for their duty.

To obtain reasonable performance an adequate outlet diffuser is necessary since pressure mainly comes from conversion of the high velocity pressure leaving the impeller. It is also necessary to design the air circuit through the appliance and the impeller as one unit, to ensure that the build-up of the vortex will not be disturbed. Larger models are occasionally made where the rectangular flow passage is helpful. An example is the provision of air curtains over the open doorways to exclude the weather.

7.14 Special-Purpose Fans

A fan maybe aerodynamically suitable for a given duty, but disqualified for other reasons. Special hazards are raised by polluting substances in the air stream. A classification from this point of view follows.

7.14.1 **General-purpose fans** maybe defined as fans suitable for operating continuously in normal indoor atmospheres at temperatures up to 40°C. Particular general-purpose fans may meet the milder cases of the hazards below. To ensure long life, however, special-purpose fans should be considered.

7.14.2 **Weather resistance**. Higher grades of exposed plastic components and paint finishes are required for repeated exposure to rain, fog, and the ultra-violet radiation of the sun. Ingress of rain and snow must be limited to protect moving parts and delicate components. Severe climates pose special hazards. For example, termite and fungus attack in tropical environments on insulation and other organic materials; lubricant stiffness and material embrittlement at Arctic temperatures; marine strength standards for deck-mounted fan casings.

7.14.3 **Moisture resistance**. Dripping water and repeated condensation from a saturated atmosphere are to be regarded as abnormal hazards. Higher grade materials or finishes may be required to resist corrosion. Water must be kept out of bearing housings by seals or flingers. Motor frame joints must be sealed or gasketted to keep out liquid water. Humid air will nevertheless enter and condense when the motor cools; drain holes on the underside are advisable to prevent accumulation of condensate.

Special fans to handle air containing condensed liquid are called *wet gas* fans. The impellers are constructed from material selected to resist erosion, that is pitting at the points of impact with the liquid drops.

7.14.4 **Corrosion resistance**. Heavily polluted industrial atmospheres will be sufficiently corrosive to shorten the life of most general-purpose fans. Provided the concentration remains within what might be described as the "breathable" range, additional coats of protective paints will usually suffice. Chlorinated rubber paints are commonly used for this purpose.

Heavier concentrations in the air handled by the fan demand special surface treatments going right back to the basic steel or aluminium. Baked enamels, sprayed metal powders, or thick fused plastics coats may be cited, each requiring a carefully planned process schedule.

Fans to handle highly corrosive gases, particularly when wet or hot, may require to be constructed from special corrosion-resistant metals or plastics. Selectivity is necessary since no material is uniformly resistant to all reagents, though rigid PVC has a wide range of application in the chemical industry.

7.14.5 **Heat resistance**. General-purpose fans with motors out of the air stream will handle air with temperatures well above 40°C, the first limitation being generally reached with the bearing lubricant. Standard lithium-based ball bearing greases will give satisfactory life at *60°C* to 80°C housing temperature according to speed. Pedestal housings can be isolated to maintain such temperatures with much hotter air in the fan. Oil-lubricated bearing systems can be designed for any practical condition, with circulation through an oil cooler, or water cooling for the worst cases.

Electric motors operating in the air stream of a fan are normally rated for a maximum ambient temperature of 40°C. Specially wound motors insulated with Class F or Class H materials may be rated to operate in temperatures up to 70°C and 100°C respectively. At higher temperatures the motor should be moved outside the air stream as with most centrifugal fans and bifurcated axials.

For *hot gas fans* operating at really high temperatures, the materials of construction must be changed. Maximum impeller speeds and stresses may require progressive limitation at temperatures above 200°C for aluminium alloy and 400°C for mild steel construction.

At temperatures from *600°C* to 1000°C special steels are available to resist the dimensional changes known as creep, and the increased corrosion rates which would otherwise occur.

7.14.6 Abrasion resistance. The impellers of general-purpose fans may be worn away by the passage of too much dust of an abrasive character. If the particles cannot be sufficiently removed, as by a cyclone, an abrasion resistant fan must be specified. This may be constructed of steel alloy, harder than the material to be handled, or alternatively easily replaceable plates may be fitted at the areas subject to wear. Rubber blades moulded to aerofoil shape on a metal core are sometimes used for axial fans, and will deal with light particles of any hardness.

7.14.7 Explosion hazards. Any possibility of an inflammable or explosive concentration in the gases handled by a fan must be taken seriously. The requirements for electrical equipment were discussed in Chapter 5. A *flameproof fan* may be necessary with the appropriate class of electric motor, terminal arrangements, and cabling.

Sparkproof construction may also be specified for the mechanical parts of the fan. This involves extra precautions against rubbing contact between moving parts and the avoidance of materials prone to incendive sparking on impact with foreign bodies. These include magnesium and (to a lesser extent) aluminium alloys. Steel and cast iron are relatively safe.

7.14.8 Leakage avoidance. For the conveyance of toxic or radioactive gases a *gastight fan is* likely to be specified. The same requirement may apply to an inflammable or valuable gas, e.g. hydrogen or helium in a closed cooling circuit. The permissible leakage rate at a given pressure should be laid down, and will determine the jointing and gasketting system for the casing together with the type of rubbing seal if the drive is external.

7.15 Elementary Theory-Axial Fans

For those who wish to obtain a clearer understanding of the mechanism of fan pressure generation, a little mathematics is necessary. Fig. 7.24a is an isometric view of the path of the air through an axial fan; a small portion of the total flow is considered, confined between two imaginary cylinders of slightly differing diameter. Radial forces are assumed to be so balanced that the air which enters flows wholly between the two cylinders with no tendency to escape, outwardly or inwardly.

As the air passes through the rotating impeller, forces exerted by the blades give it a spinning motion so that it leaves the impeller in helical paths as if guided by a very long-pitch screw thread. The condition of radial balance enables the imaginary cylinder to be unrolled flat so that the helical paths become straight lines, easier to handle mathematically.

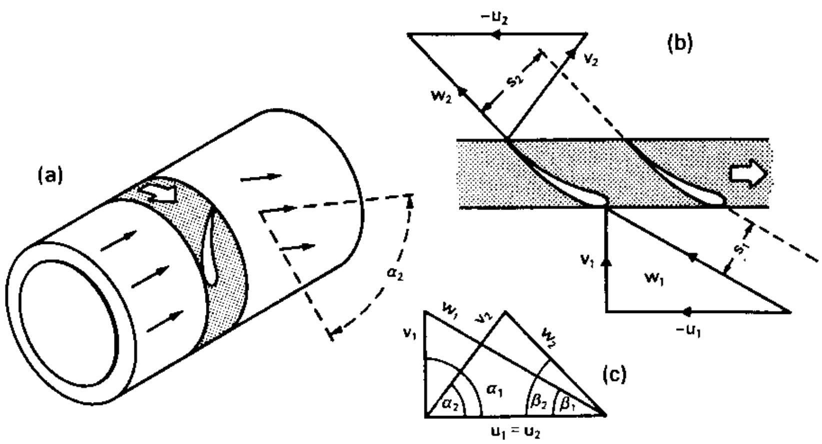


Fig. 7.24 Axial flow.

Fig. 7.24 shows this development. Velocities are represented by vectors giving the magnitude and direction of the air flow. These are distinguished as follows:

v_1 v_2 — actual or absolute air velocities.
u_1 u_2 — peripheral velocity of blade at radius considered.
w_1 w_2 — air velocities relative to the rotating impeller.
$_{1,\,2}$ — suffixes indicating inlet and outlet sides of impeller.

In the absence of external forces air will enter the fan in the axial direction v_1. To find its velocity w_1 relative to the leading edge of the moving blade, we must subtract the blade velocity vector u_1 (that is add $-u_1$) to the air velocity vector v_1. This operation is performed by drawing the vector triangle as in Fig. 7.24b. The impeller will be so designed that, at the "design point" volume flow and corresponding inlet velocity w, will meet the blade at the most favourable angle of approach.

The blade is given an aerofoil shape designed to deflect the relative velocity in the direction of rotation from w_1 to w_2. Note that the components of w_1 and w_2 in the axial direction must be the same since there is the same volume flow in the same annulus area at inlet and outlet. It follows from the geometry of the flow (note also the increase in blade passage width from s_1 to s_2) that $w_2$must be less than w_1. The corresponding fall in the relative velocity pressure, ($\frac{1}{2}\rho w_1^2$ - $\frac{1}{2}\rho w_2^2$) results in an equal rise in static pressure ($p_{s2} - p_{s1}$) within the impeller from blade inlet to blade outlet.

Now static pressure, as explained in Section 6.1.2, is an absolute property of the air which does not change with the frame of reference. Therefore the rise of static pressure within the impeller is also the rise of static pressure external to the impeller from the inlet to outlet side. A fall in relative kinetic energy has been transformed into a rise in absolute static energy. The vector diagram on the outlet side, u_2 being the same as u_1 shows that the absolute velocity v_2 is increased as well as deflected from v_1. With a little rearrangement the two diagrams can be combined into the composite vector diagram 7.24c, which shows the velocity changes quite clearly. We can now determine the total pressure rise across the impeller (p_{t2} - p_{t1}).

Since

$$p_{s2} - p_{s1} = \tfrac{1}{2}\rho\,(w_1^2 - w_2^2)$$

$$P_t = p_{t2} - p_{t1} = (p_{s2} + \tfrac{1}{2}\rho v_2^2) - (p_{s1} + \tfrac{1}{2}\rho v_1^2)$$

$$= \tfrac{1}{2}\rho\,(w_1^2 - w_2^2 + v_2^2 - v_1^2) \qquad (74)$$

An alternative expression can be found with the aid of the absolute flow angles a shown, together with the relative flow angles β in Fig. 7.24c. For our particular case a_1 = 90° and $u_2 = u_1$ while there is no change in the axial component of v, ie. $V_2 \sin \alpha_2 = v_1$.

$$w_1^2 = v_1^2 + u_1^2$$

$$w_2^2 = v_1^2 + (u_2 - v_2 \cos \alpha_2)^2$$

Therefore

$$w_1^2 - w_2^2 = 2u_2\, v_2 \cos \alpha_2 - v_2^2 \cos^2 \alpha_2$$

$$v_2^2 - v_1^2 = v_2^2 - v_2^2 \sin^2 \alpha_2$$

Substituting in equation 74 and remembering that $\sin^2 \alpha_2 + \cos^2 \alpha_2 = 1$

$$P_t = \rho u_2 v_2 \cos \alpha_2$$

7.16 Elementary Theory-Centrifugal Fans

A portion of the flow through a centrifugal fan is shown in Fig. 7.25a. It is again confined between two imaginary surfaces, initially cylindrical but then curving smoothly outwards into two parallel radial planes. The area between the planes is no longer constant, but they follow the natural streamlines of flow as determined by the casing and impeller. This includes a loss-free turn to the radial direction when approaching the blades.

The velocity vectors at blading inlet and outlet have the same meaning as for axial fans, although u_2 is no longer equal to u_1. Once again the relative velocity vector w_1 determined by the inlet side vector triangle, meets the leading edge of the blade at a favourable angle at design volume flow. The radial component at outlet v_2 is fixed by the volume flow and outlet area, while the assumption that the relative velocity w_2 is in the direction of the trailing edge of the blade completes the outlet velocity triangle. Transformation of relative kinetic energy within the blade passages will produce the static pressure rise $\frac{1}{2}\rho w_1^2 - \frac{1}{2}\rho w_2^2$ in the axial case, and the total pressure rise will again include the change in velocity pressure, $\frac{1}{2}\rho v_2^2 - \frac{1}{2}\rho v_1^2$. There is, however, an additional pressure rise in the centrifugal fan case, derived as follows.

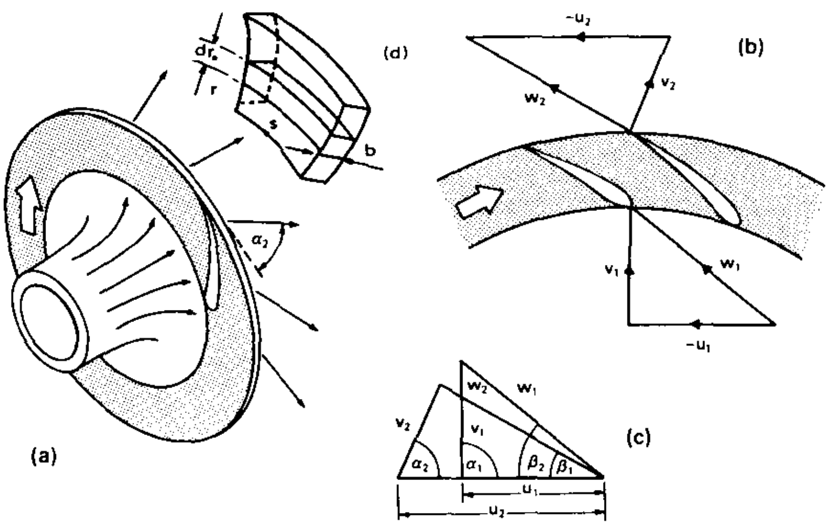


Fig. 7.25 Radial flow.

Consider the element of air shown in Fig. 7.25d. Between radii r and (r + dr) and between successive blades spaced s circumferentially and of width b, the mass of this air will be:

$$m = \rho \,.\, sb \,.\, dr$$

This air must have a circumferential velocity component a equal to that of the impeller, that is 2π rn where n is the revolutions per second. Irrespective of any other relative motion it will experience a centrifugal force:

$$\frac{mu^2}{r} = \rho \,.\, sb \,.\, 4\pi^2 \, n^2 \, r \,.\, dr$$

This force, exerted over an area of sb will produce an increment of pressure from r to (r + dr)

$$dp = \rho \,.\, 4\pi^2 n^2 r \,.\, dr$$

Integrating this from r_1 to r_2 the contribution of centrifugal force to the static pressure rise from impeller inlet to impeller outlet is obtained as:

$$\rho . 4\pi^2 n^2 \int_{r_1}^{r_2} r . dr = \rho . 4\pi^2 n^2 . \tfrac{1}{2} (r_2^2 - r_1^2)$$
$$= \tfrac{1}{2}\rho (u_2^2 - u_1^2)$$

There are thus three component parts to the total pressure rise across the impeller:

Fall in relative velocity pressure	$\tfrac{1}{2}\rho (w_1^2 - w_2^2)$
Rise in absolute velocity pressure	$\tfrac{1}{2}\rho (v_2^2 - v_1^2)$
Pressure caused by centrifugal force	$\tfrac{1}{2}\rho (u_2^2 - u_1^2)$

and the sum of these is the fan total pressure:

$$P_t = \tfrac{1}{2}\rho (w_1^2 - w_2^2 + v_2^2 - v_1^2 + u_2^2 - u_1^2) \qquad (76)$$

This expression is in fact quite general for impellers of any kind, including axial and mixed flow types. It is also unnecessary for the inlet velocity to be axial, deflection with and against the impeller rotation being respectively defined by positive and negative values of α_1. Formula 76 still applies, the alternative formula in terms of flow angles being:

$$P_t = \rho (u_2 v_2 \cos \alpha_2 - u_1 v_1 \cos \alpha_1) \qquad (77)$$

7.17 Elementary Theory-Impeller Power

If the cross-sectional areas of the portions of the total flow illustrated in Figs. 7.23 and 7.24 are a_1 at impeller inlet and a_2 at impeller outlet, we can write down the mass flow per second through the portion considered as:

$$q_m = \rho_1 a_1 v_1 \sin \alpha_1 = \rho_2 a_2 v_2 \sin \alpha_2$$

At outlet this flow is swirling with a circumferential velocity component $v_2 \cos a_2$ at a radius r_2. It therefore carries with it each second angular momentum to the value of:

$$q_m r_2 v_2 \cos \alpha_2$$

Similarly the flow entering the impeller brings with it angular momentum at the rate $q_m r_1 v_1 \cos \alpha_1$. The difference between these quantities is the rate at which angular momentum is being added to the air stream, which must equal the torque applied by the impeller:

$$q_m (r_2 v_2 \cos \alpha_2 - r_1 v_1 \cos \alpha_1)$$

At a speed of n revolutions per second this torque requires a mechanical power input of:

$$2\pi n . q_m (r_2 v_2 \cos \alpha_2 - r_1 v_1 \cos \alpha_1)$$
$$= q_m (u_2 v_2 \cos \alpha_2 - u_1 v_1 \cos \alpha_1) \qquad (78)$$

But, the *power output* of the fan is Q_1 P_1 where Q_1 is the whole volume flow. Similarly, the power output of the portions of flow we have been considering will be q_1 P_t where $q_m = \rho q_l$. Substituting for P, from equation 77 the element of power output becomes:

$$q_m (u_2 v_2 \cos \alpha_2 - u_1 v_1 \cos \alpha_1)$$

It is not surprising that the aerodynamic power output should equal the mechanical power input of equation 78 since all losses have been ignored. These losses have the effect of reducing the pressure rise and effective output together with other factors discussed in the next section. The mechanical input power obtained by summing the equation 78 over the whole flow through the fan is, however, correct.

7.18 Practical Impeller Design

This idealised treatment is only a starting point. Further factors the designer must consider include the following

(a) The air will not leave the impeller at the angle set by the trailing edge of the blade. There will be a *deviation* angle tending to reduce input and output power. This is a function of aerofoil shape and blade loading.

(b) The air will meet the leading edge of the blade at the optimum angle only at the design volume flow. At other flows the incidence angle will change. The effect of this change on performance over the whole fan characteristic is again a function of aerofoil shape and loading.

(c) To keep the portion of flow considered between the imaginary surfaces defining its intended path requires a balancing transverse pressure gradient. In the axial case, for example, only the so-called free vortex pressure and velocity distribution will secure flow at constant radius along concentric cylinders. This distribution requires a constant value of rv cos α, i.e. a spin component which is smaller the bigger the radius. This limits the work done towards the blade tips, giving a fan of poor power-size ratio; to overcome this weakness forced vortex designs are usual, increasing tip work. To maintain balance the streamlines tilt outwards through the impeller, so that r_2 and u_2 are greater than r_1 and u_1 making it necessary to use equation 76 rather than 74.

(d) Viscous drag forces at the blade surface and wake effects behind the blade convert some of the work input into heat instead of useful pressure rise.

(e) Tip clearance effects and boundary layer retarded flow along casing, hub, backplate and shroud spoil the flow pattern at the ends of the blade and limit the work done (input and output).

(f) The velocity leaving the blades is usually far from uniform in magnitude or direction. Since the energy is proportional to the square of the velocity, more is required by the peak velocities than is saved in the troughs. Thus excess kinetic energy is supplied which will not all be available when the air has reached the downstream test plane.

CHAPTER8

Fan drives and speed control

The vast majority of general-purpose fans are driven by electric motors. Motors suitable for fan drive are discussed in this chapter, together with their installation and speed control. Fan duty may be controlled by speed adjustment or by aerodynamic means and comparison of the methods available is reserved for Chapter 9.

Of course, large numbers of specialised fans are not electrically driven. Consider, for example, the cooling of internal combustion engines, by impellers mounted on a take-off from the main shaft. Even in the automobile field, however, the use of electric motor drive is spreading, since it overcomes the incompatibility of power economy at high engine speed with adequate cooling at idling speed.

8.1 AC Motors

8.1.1 polyphase induction motors are the normal standard drive for industrial fans, where the necessary three phase AC *(alternating current)* supply is almost always available. The usual *squirrel-cage rotor* has the merits of minimum maintenance, robustness and low cost; the electrical conductors are of aluminium cast in place through holes in the iron rotating member *(rotor).* In large sizes copper bars brazed to copper end rings may be preferred.

The stator (stationary iron member) is wound with coils of insulated wire inserted into slots and connected to the supply. The winding pattern determines the number of poles or peaks of magnetic flux round the circumference.

These magnetic poles rotate at a synchronous speed which is also the speed of the motor on no-load.

$$\text{Synchronous speed (rev/s)} = 2 \times \frac{\text{supply frequency (Hz)}}{\text{number of poles}}$$

Table 8.1
Synchronous speed and pole number

Supply Frequency	50 Hz		60 Hz	
Number of poles	rev/sec	rev/min	rev/sec	rev/min
2	50	3000	60	3600
4	25	1500	30	1800
6	16·7	1000	20	1200
8	12·5	750	15	900
10	10·0	600	12	720
12	8·3	500	10	600
14 (rare)	7·1	429	8·6	515
16	6·25	375	7·5	450

When an induction motor is loaded the speed falls, the drop from synchronous speed being known as the *slip*. The slip at full load ranges from 1% of synchronous speed for large motors to 10% for fractionals, typical actual full load speeds being given in Table 14.24. It is the slip of the rotor conductors past the synchronously rotating magnetic field which generates the rotor currents; these in turn react with the field to produce torque.

The slip is approximately proportional to load up to full load, but thereafter it increases more rapidly until at *pull-out* the maximum torque is reached. The relationship between slip and torque is dependent on rotor design, and two examples are illustrated in Fig. 8.1. Since the initial load at starting is that due to friction only, high torque motors are not necessary for fan drives. Standard rotors are normal practice, though there is a tendency for these to incorporate special cage features to flatten the torque curve and improve starting. In small sizes where specialised quantity production is justified, fan motors may have limited pull-out torque and a suitably tailored torque-speed curve to improve voltage-speed control and limit starting current.

While standard industrial motors may be used for direct fan drive with the impeller mounted on the shaft extension, modifications are often necessary and specialised fan motor designs are produced. Before specifying a standard motor to replace one which may or may not be standard, the following points should be checked:

Will the bearing system take the weight load and aerodynamic thrust? See 8.6.3, 8.6.4 and 8.6.5.

Is the shaft extension suitable to secure the impeller? A shaft shoulder and shaft-end clamp washer may be necessary.

Will the fan air stream and the motor cooling flow conflict? Removal of the motor cooling fan and cowl, with re-rating in the fan air stream, may be necessary for motors in the air stream.

Is the motor insulation and lubrication system and its rating correct for the air conditions, running hours, etc.? See 8.6.1 and 8.6.6.

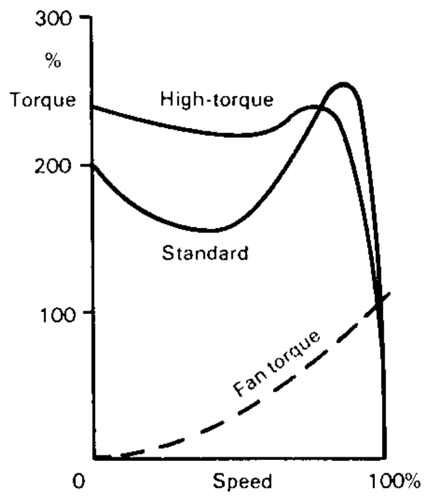


Fig. 8.1 Torque curves of three phase induction motors.

Will the motor size and shape, the necessary mounting arms or platform and such features as projecting terminal boxes obstruct the air flow to or from the impeller, or set up noise generating air flow patterns?

Will the motor start a high inertia impeller quickly enough to avoid overheating or unnecessary tripping of the overload protection?

8.1.2 Split phase induction motors

A stator winding connected to single phase AC-the usual domestic supply-cannot produce any starting torque. Special auxiliary windings are necessary which are either of high resistance (split-phase type) or incorporate a condenser (capacitor-start, induction-run type). In the standard fractional horse-power motor the auxiliary circuit has a very short time rating-a minute or so. It is disconnected by a centrifugal or current-controlled switch at about two-thirds of full speed, when a somewhat inefficient form of rotating magnetic field has developed. Such motors have sufficient torque to overcome excessive starting friction generated by a badly-adjusted belt drive or sealing gland.

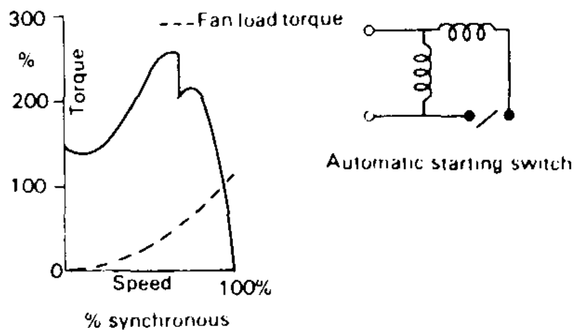


Fig. 8.2 Typical speed—starting torque curve of split-phase motor.

8.1.3 Permanent split capacitor motors

When the fan impeller is directly mounted on the motor shaft extension the *permanent-split capacitor* design is to be preferred. Provided the impeller will never absorb less than, say, half full load, it is possible to design main and auxiliary windings to share the load equally. A given motor frame will then develop the same power on single phase as on three phase supply, with equal efficiency and noise level, and at near unity power factor. This favourable result is peculiar to fan drive, with its roughly constant load; if such a motor were allowed to run light the phase split would become very unbalanced, applying excessive voltage to the capacitor and overheating the auxiliary winding.

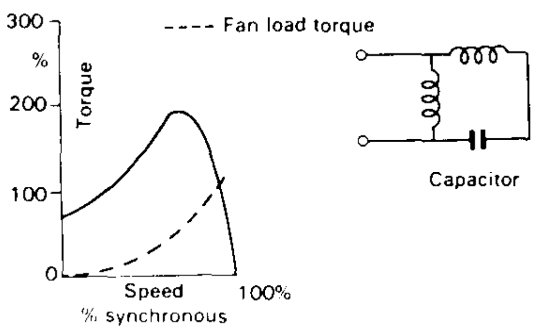


Fig. 8.3 Typical speed—starting torque curve of permanent-split capacitor motor.

8.1.4 Capacitor-start, capacitor-run motors

At full load outputs exceeding a few kilowatts the permanent split capacitor motor will not develop sufficient starting torque or will take too long to accelerate a high inertia impeller. If a three phase supply is not available, as may happen in isolated rural installations, a *capacitor-start, capacitor-run* motor is suitable. In this type an additional, short-time-rated, capacitor is switched in at low speeds to provide extra starting torque.

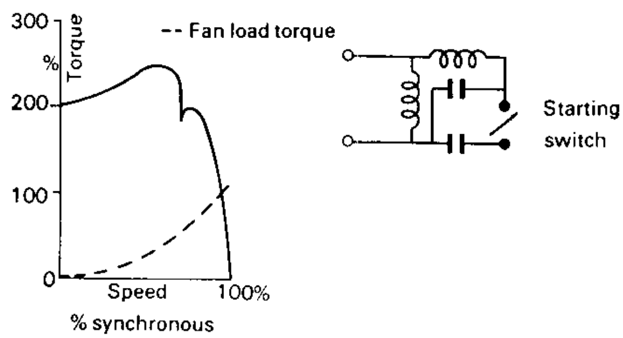


Fig. 8.4 Typical speed—starting torque curve of capacitor-start capacitor-run motor.

8.1.5 Shaded-pole motors

For outputs of 20 watts or less the *shaded-pole* induction motor is the common type for fan drive. No switch or capacitor is needed, the auxiliary winding taking the form of a closed loop of solid copper, offset to one side of the centre line of each main pole. Currents induced in this loop by transformer action develop a rotating magnetic field. This is poorly distributed, resulting in a low efficiency, seldom exceeding 25%, but the motor is quiet, reliable and cost-effective in the smallest sizes. The squirrel cage rotor always rotates from the main pole towards the nearest copper loop, unlike all other induction motors, which can be reversed by reversing the connections of one stator winding.

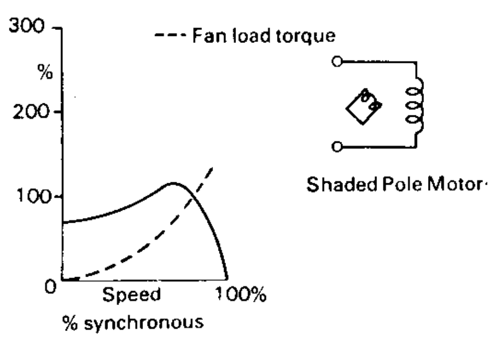


Fig. 8.5 Typical speed—starting torque curve of shaded-pole motor.

8.1.8 External rotor induction motors

A specialised mechanical type of induction motor has an external rotor carrying a squirrel cage rotating round an internal wound stator. One arrangement has blades mounted on the rotor which thus forms the hub of an axial impeller. The rotor may be overhung on a stub axle running in bearings buried within the stator body. Alternatively, the rotor may have bearings either side of the stator on a stationary shaft which is hollow to carry the leads to the stator. The latter arrangement is usual for ceiling fans and is also occasionally used for small double inlet multi-vane centrifugal fans.

8.1.7 Synchronous motors

These are rarely used for fan drive, unless the fan load is continuous and large, forming a substantial fraction of the total load in the plant. It is then possible to use the unique ability of a synchronous motor to draw leading power-factor current from the supply. This will compensate for the lagging power-factor current drawn by induction motors in the plant, correcting the overall power factor and so qualifying for better rates under the supply company's tariff.

The stator is similar to that of a three phase induction motor, but the rotor is quite different. It has an insulated winding fed with direct current, either from a DC commutator-type generator on the same shaft or through slip rings and a rectifier from the AC mains. This holds it in exact synchronism with the rotating magnetic field of the stator. Starting and synchronising on load is difficult and designs sometimes involve running the motor up light and subsequently accelerating the fan impeller through a fluid clutch or other slip-type coupling.

8.2 Speed Control of Induction Motors

8.2.1 Voltage control

Small squirrel cage induction motors driving fans can have their speed reduced or regulated by reduction of the applied voltage, provided they are of the shaded-pole, permanent-split capacitor, or polyphase types. Split phase and other single phase motors incorporating switches cannot be regulated in this way because the short-time rated auxiliary windings might be brought into circuit.

Reasonably stable running is possible because of the proportionality of fan torque to the square of the speed. Fig. 8.6 shows the way in which

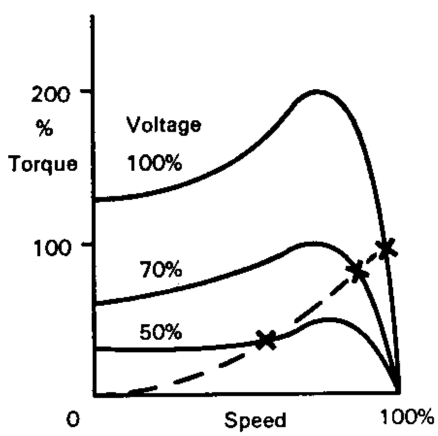
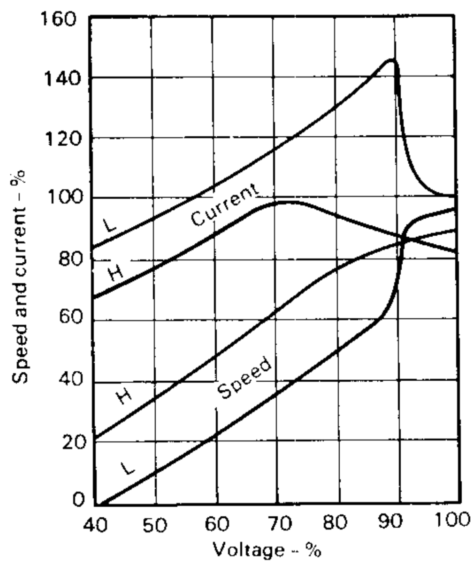


Fig. 8.6 Speed regulation by voltage control.

the motor torque-speed relationship is reduced with successive reductions in supply voltage. At full speed and first regulated speed there is a well defined intersection X with the fan torque-speed curve identifying stable running speeds. In the last case the intersection is not so well defined and the running speed will fluctuate somewhat with motor temperature, bearing friction, and other factors.

Substantial speed reduction is inevitably accompanied by energy loss in the squirrel cage conductors. The loss reaches a maximum at two-thirds of synchronous speed, and is then 15% of the power the impeller would absorb at synchronous speed. This loss can be accommodated



L = low, H = high, rotor resistance.

Fig. 8.7 Thyristor control of single phase capacitor motor.

in motors with full load ratings up to about 350 watts output. At higher outputs the motor frame must be derated, and a practical maximum for voltage regulation is about 1 kW output.

Small fan motors are often designed with relatively high resistance cage rotors because, as explained under wound-rotor motors, this increases stability. Voltage control by autotransformer is best for stability also, since the output volts are independent of load, but series chokes or resistors are often used.

The permanent-split capacitor motor is best regulated by control of the main winding voltage, the auxiliary winding and capacitor remaining at full voltage. This arrangement maintains a better balance between the field strength and phase angles of the two windings. increasing stability and reducing regulator load.

Solid state *transistor* or *thyristor* controllers can give continuous rather than stepped speed control. The losses in the controller are negligible, although motor losses can be substantially increased if the motor is not specifically designed for this form of control. Stability is also dependent on appropriate design, Fig. 8.7 showing as an example the unfavourable effect of too low a rotor resistance. On three phase it is best if the supply has a neutral connection available, connected through the controller to the star point of the motor stator, thus providing three independently controlled phases.

The principle of thyristor operation is the disconnection of the supply at each current zero, reconnecting at a controlled time interval later so that the motor receives power for part of each cycle only. The controller must incorporate protective devices for the thyristors of proved adequacy in relation to the motor control duty. Radio interference must be suppressed.

8.2.2 Wound rotor induction motors

In these machines the squirrel cage is replaced by an insulated winding connected through slip rings and brushes to an external variable resistor. When the resistance is set at zero the motor runs at full speed and load exactly like a squirrel-cage motor. As the resistance is increased the motor torque-speed characteristic is progressively changed as shown in Fig. 8.8, enabling the fan to run stably at any speed below full speed.

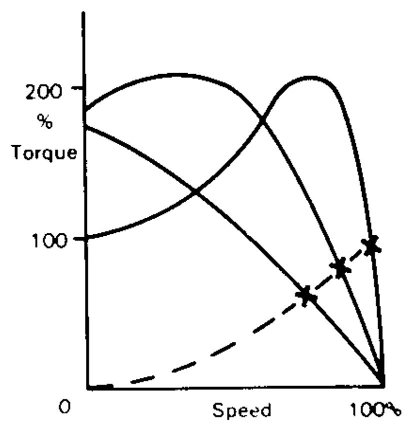


Fig. 8.8 Speed regulation of wound rotor motor.

Since the energy loss has been removed from the rotor winding there is no longer any limit to the power output for which the motors may be built. Unfortunately, the loss remains inevitable. Apart from certain rather costly systems of regenerative recovery it must be dissipated in the resistor elements. If the full load fan power is W_i watts at n_1

revolutions per second, the synchronous speed being n_s and the regulated speed n, the loss in the resistor and (a small item) the rotor winding will be:

$$\left(\frac{n_s n^2 - n^3}{n_1{}^3}\right) W_i$$

Such motors may be considered for short-time speed control above the range covered by de-rated squirrel-cage motors, but some de-rating is still necessary and a more efficient method should in any case be considered if the running hours are long or the load heavy.

8.2.3 Frequency control

A three phase squirrel-cage motor will operate efficiently at speeds proportional to the supply frequency provided the supply voltage is varied in the same proportion. With its use of standard fan motors and freedom from energy loss this would be the ideal method if it were not for one drawback-high first cost. It may, nevertheless, be economic for large fans, or if a large number of fans are to be simultaneously controlled from a single frequency converter, or if specially accurate speed control is wanted.

Solid state devices have taken the place of the older rotating machine sets for the conversion of the 50 or 60 Hz supply to variable output frequency. Thyristors are used for the repeated switching of the motor current from phase to phase of the supply at controlled instants in the cycle. The cost arises largely from the complex protective circuits required to guard the thyristors from the danger of shortcircuit or overvoltage surge; as well as to control the frequency. High frequency harmonic currents decrease motor efficiency by a few per cent., and may require limitation to satisfy the supply authority. There are two main types. The rectifier-inverter system first converts the alternating supply currents into a unidirectional (DC) current and then reconstitutes AC at a different frequency controlled by an oscillatory circuit. The cycloconverter system dispenses with the DC stage, splitting the supply into pulses at several thousand Hz, which are then modulated at output frequency. Cyclo-converter outputs are preferably below 30 Hz, which may be favourable for large low speed fan motors. since they will be wound with fewer poles.

8.2.4 Multi-speed windings

Three phase squirrel-cage induction motors can be wound with more than one pole number. By switching from one to another different synchronous speeds can be selected. For each pole number the motor speed will be the same as that of the corresponding single speed machine, though the efficiency and output will be somewhat reduced owing to the necessary reduction of stator conductor cross-section, or winding factor.

One method involves putting two, or occasionally three, separate windings into the same stator slots. Since, on fan load, the power required falls with the cube of the speed, the low speed can be wound with a much smaller wire cross-section. The bulk of the slot space remains available for the high speed winding, which will not usually need to be derated more than 30% to 35%.

The other method is known as pole amplitude modulation (PAM). In this a single stator winding is tapped at selected places so that the coils may be rearranged by a two-position switch to change the number of poles. The system is flexible, it having been shown that quite unsymmetrical coil arrangements-designed in accordance with the Rawcliffe patents-will perform without much loss of output or efficiency.

Common pole number combinations are 4 and 6, 4 and 8, 6 and 8, but the choice is virtually unlimited.

It is possible to provide:

2 speeds—by two normal or one PAM winding

3 speeds—by three normal or one PAM, plus one normal winding

4 speeds—by two PAM windings.

8.3 Commutator Motors

8.3.1 Direct current motors

Constant speed DC motors will not be used for fan drive if an AC supply is available. This limits them in practice to transportation applications, often at 6, 12 or 24 battery voltage, or to isolated locations. In both cases the power is likely to be low.

When the impeller is mounted directly on the motor shaft a series wound motor is usual. The single field winding is connected in series with the armature through the brushes and commutator. A series wound motor should not be used for indirect fan drive because, if the load be disconnected, for example through belt failure, the speed will rise to a dangerous level. The latter weakness is avoided in the compound wound motor. This has, in addition to the series field winding, a shunt field winding which is connected across the supply and will maintain sufficient magnetic field at light load to prevent excessive speed. Pure shunt motors without a series winding are unsuited to fan drive.

8.3.2 AC series motors

Small series wound motors with ratings up to about 500 W can be designed to operate on AC as well as DC. At speeds of 100 to 200 revolutions per second the AC and DC performances are quite similar, and such machines are called universal motors. However, with the disappearance of DC public supplies this feature has lost its importance. The ability to operate far above the 50 or 60 rev/s maximum of the induction motor is nevertheless valuable for such applications as domestic suction cleaners. As these are also for intermittent use, the rather short brush and commutator life characteristic of AC commutator motors is not a serious disadvantage. Radio interference suppression is always necessary.

8.3.3. Speed control of DC motors

DC (and AC) series fan motors may be reduced in speed by the series connection of a variable resistor. At higher powers the resistor may also act as a starter, but the energy losses are substantial, and for fan powers in excess of 1 kW or so more efficient methods should be chosen.

One such method is shunt field control. A compound wound motor with a relatively weak series winding, and additional series connected commutating poles to suppress sparking, will be used. The shunt field can be weakened using a variable resistor or potentiometer whose losses are quite small. As the field drops the speed rises so that maximum speed and load occur at the weakest magnetic field. This reduces the output available from a given motor frame and limits the speed range available to 1 ½ or 2 to 1 at the outside.

8.3.4 DC and other commutator motors for AC supply

The successive developments of mercury arc, semi-conductor, and thyristor types of *rectifier* have enabled large DC motors, with shunt field speed control, to be operated off an AC supply. Of greater significance is the phase control or "chopped-wave" system of output voltage control available in addition to rectification when thyristors are used. In this system the shunt field of the compound wound DC motor is maintained at full strength through a small auxiliary rectifier. The main thyristors supply the armature and series field. In addition to full wave rectification they have a variably delayed closure in every cycle which controls the effective voltage applied to the armature. This reduces the motor speed to any desired value, without energy loss, apart from a slight increase in motor losses caused by high frequency pulsations. Although relatively costly this system provides an answer to any problem requiring wide range, accurate speed control with minimum loss.

Several types of AC commutator motor are made for three phase AC supply and are capable of operating over a range of speed. Their use for fan drive is mainly confined to heavy industry where skilled maintenance is available to look after the brush gear and commutators. Also many designs involve physical movement of brushes, induction regulator cores, etc., and may be more suited to manual than to automatic control.

8.4 Induction Motor Performance

8.4.1 Efficiency and power factor

In the smaller sizes single phase permanent-split capacitor or capacitor-start capacitor-run motors can be made with the same efficiency as standard three phase induction motors and a power factor generally in excess of 0.95.

On DC the product of the supply voltage and supply current equals the input power in watts. On AC, however, there is an additional interchange of energy each cycle between the magnetic field and the electric supply, measured in "reactive volt-amps", which does not, on average, supply any power. The useful energy which is supplied is measured in watts. These two quantities can be treated as *phasors*, at right angles to one another, and their phasor sum is known as the "volt-amps" input.

$$\text{Volts-amps} = \sqrt{(\text{Watts})^2 + (\text{reactive volt-amps})^2}$$

$$VA = \sqrt{W^2 + VAR^2}$$

$$kVA = \sqrt{kW^2 + kVAR^2}$$

The ratio of the watts to the volt-amps is known as the "power factor" and it may be expressed as "cos Ø" where 0 is the phase angle between the supply voltage and the supply current. The relations between the supply volts, V; the supply current, I amps; the supply power, W_e watts; the output power, W_m watts and the efficiency, η, are then:

$$\text{DC} \quad W_m = \eta W_e = \eta VI$$

$$\text{Single phase AC} \quad W_m = \eta W_e = \eta VI \cos Ø$$

$$\text{Three phase AC} \quad W_m = \eta W_e = \eta \cdot \sqrt{3} \cdot VI \cos Ø$$

In the last case V and I are the line voltage and line current respectively.

8.4.2 Power factor correction

The capital cost of transmission and power station electrical plant is largely dependent on the kVA rather than the kW. Hence the power supply tariff is often based partly on the kilowatt-hours energy supplied and partly on the kVA maximum (30 minute) demand. Fans are likely to be continuously running, and therefore to enter fully into the maximum demand of a plant. If the aggregate kVA of all fans is considerable, and particularly if the fan motors are low speed, and therefore of rather low power factor, power factor correction is worth consideration. Permanent-split capacitor motors are inherently corrected, so the issue will arise only with three phase motors.

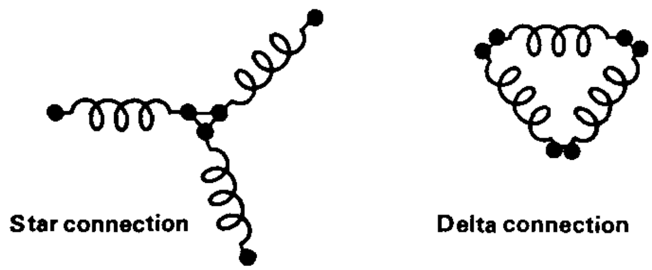
Banks of static power-factor correction capacitors can be purchased for connection across the incoming supply terminals. They will have a rating kVAC but the power factor is "leading" rather than "lagging" which implies the opposite sign to the WAR of the motors. We thus have:

$$kVA = \sqrt{(kW)^2 + (kVAR - kVAC)^2} \qquad (81)$$

The maximum demand will therefore be a minimum, and equal to kW, when kVAC = WAR = kW tan Ø. The formula shows that kVAC can fall substantially short of WAR before the kVA rises appreciably above the minimum.

8.4.3 Supply voltages and windings

The windings of three phase induction motors up to 2 kW input are normally connected in star for supplies of 380 to 415 volts. With each end of each phase brought out to 6 terminals it is possible to reconnect in delta for 220 or 240 volt three phase supplies as used in some countries for low-power installations. This makes stock motors more widely available.



The practice may be extended to 5 kW unless there is a requirement for star-delta starting (see Section 8.5.1) on the higher voltage. Such a requirement necessitates motors with delta-wound stators.

Similar considerations sometimes lead to single phase motors being wound in two parts for parallel connection on 110 volts and series connection on 220-240 volts, and even for a compromise design capable of operating on either 50 Hz or 60 Hz in the shaded-pole range.

8.4.4 High inertial loads

Particularly heavy impellers can in some cases influence the selection of motors. It can be shown that the heat dissipated in the rotor of an induction motor while starting equals the kinetic energy stored in the rotating parts when full speed is reached. Additional heat is generated if external work is done during the starting period. The danger is that the rotor will be overheated, particularly if starts are frequently repeated, as they may be if the fan operates on a time-cycle or temperature-cycle. The stored kinetic energy of the rotating parts is:

$$KE = 2\pi^2 (I_1 n^2{}_1 + I_m n_m{}^2) \text{ joules} \qquad (82)$$

where the impeller rotates at n, rev/s and the motor at n_1 rev/s - which may of course be different from n_1 for an indirect drive. I_1 and I_m are the moments *of* inertia of the impeller and motor rotor respectively, found from the formula:

$$I = m \cdot r_g{}^2 \quad \text{kgm}^2$$

in this formula m is mass in kg of the rotating part considered and r_g its *radius of gyration* in metres. R_g is likely to be between 25 and 30% of the diameter for an axial impeller, 30 and 36% for a backward-curved centrifugal, and 36 and 42% for a forward-curved. For a cylindrical rotor r_g is 35% of the diameter.

The torque of an induction motor directly coupled to an impeller is plotted in Fig. 8.9, but for the present purpose it is sufficient to estimate an average value, M_m newton-metres, over the whole starting speed range. A torque M_1 is needed to drive the impeller at full speed, n_1 rev/s, and varies with the square of the speed as plotted, leaving a variable margin for acceleration. Fig. 8.10 gives the dependence of two factors, A and B, on the ratio of full-load torque to average starting torque. Then, for the general case when n_m is not equal to n_1:

$$\text{Acceleration time} = A.\frac{2\pi n_m}{M_m}\left(I_m + I_1\frac{n_1{}^2}{n_m{}^2}\right) \quad \text{secs.} \qquad (83)$$

$$\text{Rotor heat generation} = B\,.\,2\pi^2\,(I_1\,n_1{}^2 + I_m\,n^2{}_m) \quad \text{joules} \qquad (84)$$

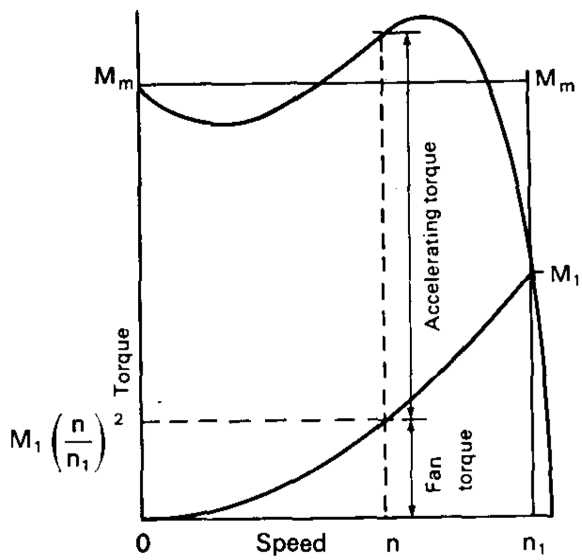


Fig. 8.9 Fan motor starting.

Motor makers usually quote the inertia of the motor rotor, together with the maximum allowed starting time or kinetic energy of rotating parts. If these are exceeded an increase of motor frame size is necessary, though this is only likely to occur when a heavy centrifugal impeller operates at well below its maximum speed, using a correspondingly low power driving motor.

8.4.5 Example

A 1250mm heavy-duty, backward-curved impeller, weight 350 kg and radius of gyration 430mm, absorbs 15 kW when belt driven at 12 rev/s by a 24 rev/s motor.

Can a 15 kW D160 motor be used with 0.12 kgm^2 rotor inertia and an average torque 1 .7 times full load?

$$\text{Motor torque } M_m = \frac{15{,}000\ W \times 1{\cdot}7}{2\pi \times 24\ rev/s} = 170\ Nm$$

At $1 \div 1{\cdot}7 = 0{\cdot}59$ full load to starting torque ratio A = 1·32
Impeller inertia $I_1 = 350\ kg \times (0{\cdot}43m)^2 = 65\ kgm^2$

Therefore, from equation 83, the acceleration time is:

$$1{\cdot}32\ (A) \times \frac{2\pi \times 24\ (rev/s)}{170\ (Nm)} \left[0{\cdot}12 + 65 \times \left(\frac{12}{24}\right)^2\right] = 19\ \text{seconds}$$

This is too close to the recommended maximum starting time of 20 seconds. Considering a D180 frame with average starting torque 1.7 times full load of 22 kW the torque ratio for Fig. 8.10 becomes 15/1.7 x 22 = 0.40 giving A = 1.18, M_m = 250 N_m and acceleration time = 12 seconds, which is satisfactory.

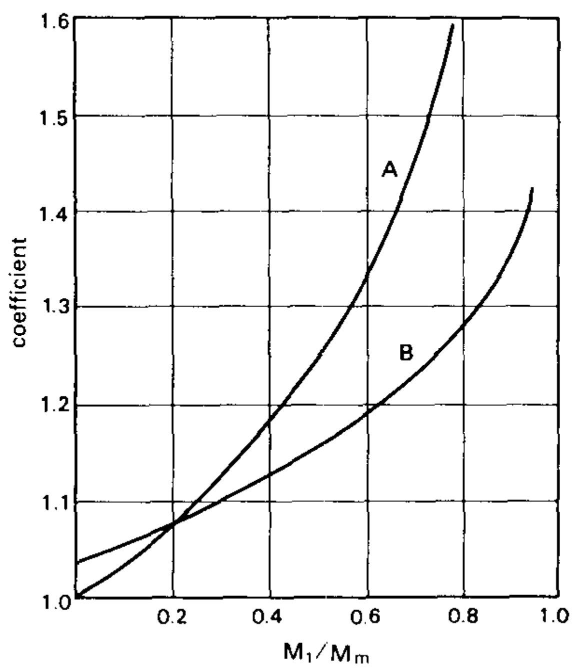


Fig. 8.10 Starting time and loss coefficients.

8.5 Motor Control Gear

8.5.1 Direct-on-line and star-delta starters

Any fan may be started *direct-on-line* (d.o.l.) provided it is not outside the range allowed by the supply authority. When some limitation of starting current is called for the first choice on a three phase AC supply is usually the *star-delta* starter. The motor windings are connected to six terminals as in 8.4.3 and are designed for delta connection when running. At starting a star connection is made, and has the effect of reducing both the standstill current and the standstill torque to approximately one-third of the values they would have for d.o.l. starting in delta.

One-third of normal starting torque is always sufficient to get the impeller started, but it may sometimes happen that there is insufficient torque in star to accelerate beyond about two-thirds of full speed. Switching to delta will, of course, finish the job, but at the expense (in some cases) of a current rush only a little less than it would have been with d.o.l. starting. Modern squirrel-cage motors have "double-cage" or other specialised rotor designs which enable a good torque to be maintained over the whole speed range. Nevertheless, a requirement for star-delta starting on fan load should be specified to the motor manufacturer, who may recommend derating a particular motor frame, though rarely more than 10% or 20%.

8.5.2 Other starting methods

Large squirrel cage motors are more likely to require special starting arrangements if the demands on the supply need to be limited. The autotranslormer starter connects the motor terminals to reduced supply voltage during the run-up period. The star-delta starter is equivalent to an autotransfoimer set at $1 / \sqrt{3}$ = 58% of supply voltage. This may well be the first autotransformer step with one or more intermediate steps inserted during the run up to full voltage. *Part-winding* motors are popular in America, for the smaller three phase motors. The stator winding is in two parallel sections, one of which is disconnected at starting, increasing the impedance to current flow.

Resistance starters insert successively reduced steps of metal strip resistors between the motor terminals and the supply. They may be relatively cheap though requiring selection to match the motor. An interesting variant is the liquid electrolyte starter. As the electrolyte is heated by the passage of the starting current, its resistance falls, thereby steadily increasing the effective motor voltage. This is particularly suitable for fan load with torque proportional to $(\text{speed})^2$ and is inherently automatic, the final step being closure of a switch short-circuiting the electrolyte.

Wound rotor motors with step-by-step metal, or continuously varied liquid, resistors in the rotor circuit provide the ideal start with minimum kVA demand at all speeds from start to full load.

8.5.3 Overload protection

The first stage is usually the HRC (high rupturing capacity) fuse to protect the motor, starter and installation wiring from damage in the event of a short-circuit. This is done by breaking the circuit within the first half cycle (1 /100th of a second) before the very high *prospective current*, which would flow if the circuit were not broken, is reached.

Next the standard starter will contain a bi-metal or fusible metal device which will open the contacts after an interval dependent on the current in the manner of Fig. 8.11. This should disconnect a faulty motor and should also protect the windings from overheating if the high current starting period is unduly prolonged for any reason. Slow starting on high inertia loads may cause unnecessary tripping if the starter is not well-matched with the motor (i.e. is too sensitive).

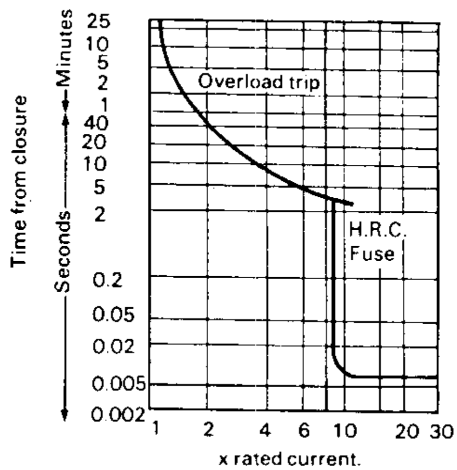


Fig. 8.11 Typical starter tripping schedule.

Thermal protection devices inserted in the motor windings themselves can replace current-operated thermal elements in the starter with advantage. Unnecessary tripping is avoided since they automatically match the heating rate of the windings. Protection is also provided against overheating not associated with excess current. Thus fan motors in the air stream may be overheated either by excessive temperature of the air handled or through accumulation of conveyed material, dirt, etc., impairing the motor cooling, or through excessive reduction of the air flow.

These devices take two main forms, each small enough to be inserted between the coils of the stator end windings during manufacture. The winding *thermostat* is a temperature operated switch, commonly taking the form of a bi-metallic disc, snapping over from concave to convex formation at a pre-set temperature. These can be used in series with the supply connections so that they will directly open the motor circuit without the need of a separate contactor on single phase up to 1 kW or so motor rating. At higher ratings thermostats in each phase can be used in series to trip the starter.

The *thermistor uses* a PTC *(positive* temperature coefficient) resistor in the form of a small bead of semi-conductor material. This has a low heat capacity, and will therefore closely follow a rapidly heated winding. One is embedded in each phase of the motor winding and they are connected in series with one another and with an electronic relay which will in turn operate the overload release of the starter. The resistance of the PTC bead rises from a few hundred to several thousand ohms over a range of a few degrees, the trip level of 1000 ohms being passed at a predetermined temperature. The recommended temperatures to limit premature thermal ageing of the insulation are:

Table 8.2
Thermal protection

Class of Insulation	E	B	F
Thermistor to trip at	130°C	140°C	160°C
Steady winding peak	155°C	165°C	190°C
Stalled winding peak	215°C	225°C	250°C

8.6 Other Motor Features

8.6.1 Class of insulation

Insulating materials for electric motors are classified according to the maximum temperature at which they will provide a satisfactory life. The life is not exactly specified, acceptability being judged rather by experience with traditional materials in practical use over many years. Accelerated life test methods are adopted by the motor manufacturer to ensure that new materials at their limiting temperatures will be at least as satisfactory as the traditional materials are at theirs. To ensure mutual compatibility these tests are often done on model wire coils inserted into slots by practical manufacturing methods.

The insulation class is therefore dependent on the whole insulation system, not just the materials used. Nevertheless it is of interest to note the following common selections for classes E, B, F and H.

Wire enamels	Polyvinylformal or polyurethane for classes E and B. Polyester for class F.
Sheet materials	Polyester film and polyester paper composites for classes E and B. Suitably impregnated glass cloth, mica splittings or polyamide-paper composites for class F.
Impregnants	Oil modified synthetic resins for classes E and B. Polyester or epoxy resins for class F.

Class H systems Silicone resins and bonding compounds.
Silicone elastomers (rubber-like materials).
Thermally superior grades of polyester-based enamel and polyamide film.

The motors themselves are accepted on the basis of temperature tests which determine the average temperature of the winding by reference to its electrical resistance. With copper wire this changes in the following ratio:

$$\frac{R_2\ (\text{ohms})}{R_1\ (\text{ohms})} = \frac{t_2\ (^\circ C) + 235}{t_1\ (^\circ C) + 235}$$

Since a failure anywhere will incapacitate the insulation system it is obviously the temperature of the hottest spot which is really important. However, this cannot be measured, and a conventional allowance is made above the measured average. Standard motor ratings are for an ambient temperature of 40°C, which covers the effective outdoor tem-pe,ature over most of the world but makes no allowance for additional air heating or for exposure to the sun's radiation.

Class E insulation is standard in most of Europe, Class B in the U.S.A. Class F, which is a little more costly, may be used either to increase the output rating of a given motor frame (by some 10% to 20%) in standard ambient conditions or to increase the permissible ambient temperature to 140°C less the actual temperature rise. Class H is substantially more costly, and is used primarily for operation in ambient temperatures above those permissible with Class F. Clearly Classes F and H are particularly useful for motors operating in the air stream of fans handling heated gases.

Table 8.3

Temperature limits of motor windings (as specified in the recommendations of IEC)

Class of Insulation	E	B	F	H
Maximum temperature rise by resistance	75°C	80°C	100°C	125°C
Average temperature at 40°C ambient	115°C	120°C	140°C	165°C
Limiting "hot-spot" temperature	120°C	130°C	155°C	180°C

All organic insulating materials deteriorate with time, and the end of their life is reached when degeneration has sapped most of their mechanical strength and rendered them permeable to moisture, with increasing ionisation and current leakage.

Roughly speaking their life is halved for every 8° to 10°C rise in temperature although no guarantees can be given above the rated maximum temperatures. The rule implies that a full rated standard motor would lose a quarter of its working life-and therefore be at some slight statistical risk of failure-by exposure to the following ambient temperatures and times (including an allowance for increased motor losses at higher temperatures).

6 months at 60°C

3 weeks at 80°C

3 days at 100°C

12 hours at 120°C

High altitude will impair motor cooling, since a given volume flow of air will carry away less heat at a lower density. Standard motor ratings apply to any altitude up to 1000m. Above this 1% should be deducted from the allowable temperature rise for each 100m above 1000m. The motor manufacturer should always be consulted about derated or specially wound motors since temperature rise is not simply proportional to output.

8.8.2 Motor cooling and enclosure

A rather complex classification of methods of cooling (designated IC with a two digit number) and protecting against damage to the machine and injury to the user (IP) will be found in the publications of IEC (International Electrotechnical Commission) and the corresponding British Standard BS 4999 in its many parts.

The great majority of fans in the 1 to 100 kW range driven by external electric motors, use *totally enclosed fan cooled* (TEFC) machines. This form of enclosure protects the internal parts from build-up of dust and damage, while the cooling fan and ribbed carcase dissipate heat in most cases as efficiently as the through-air-flow of a ventilated machine. In larger sizes internal and external air circuits coupled by built-in heat exchangers come into consideration, together with open ventilated systems if the surroundings are clean.

Axial fans with the impeller mounted on the shaft extension utilise totally enclosed, air stream cooled motors. These are often TEFC motors with the cooling fan and cowl omitted, and the TEFC rating will be achieved provided the air velocity over the motor is not less than the values indicated by Table 8.4. In fact the cooling is generally so good that higher ratings are justified. These should be left to the fan manufacturer to determine, since a number of factors are involved, including particularly air distribution.

Table 8.4
Air velocities required over totally enclosed air stream cooled motors

IEC Frame Number	Synchronous Speed (rev/min)			
	3000	1500	1000	750
	Minimum velocity—m/s			
80	10	8	7	5
90	12	9	7	6
100	15	10	8	7
112	17	11	9	8
132	18	12	10	8
160	19	13	10	9
180	20	13	11	9
200	21	14	12	10
225	22	15	12	10
250	23	15	13	10

8.8.3 Bearing loading

A fan impeller may be mounted on the shaft of the driving motor or on a separate shaft which is coupled to or belt driven by the motor. In either case the shaft is usually carried in rolling bearings which must carry both journal lie. radial) and thrust (i.e. axial) loads.

The thrust on an axial fan impeller is in the opposite direction to the air flow through it. It may be taken as the product of the maximum fan total pressure and the gross area of the impeller disc:

$$\mathbf{T\ (newtons)} = P_t\ (Pa) \times \frac{\pi}{4} D_r^2\ (m^2)$$

Strictly speaking there should be deductions for the static pressure regain in guide vanes and over a downstream hub fairing if stationary (not if attached to the impeller). Also an addition for total pressure losses outside the impeller. Such refinements are rarely justified however, and the simple formula should err on the safe side.

In the case of a single inlet centrifugal impeller the thrust will be in the opposite direction to the air flow entering the inlet. It may be taken as the product of the maximum fan static pressure and the area of the impeller eye, i.e. at the inside diameter of the shroud, D_e.

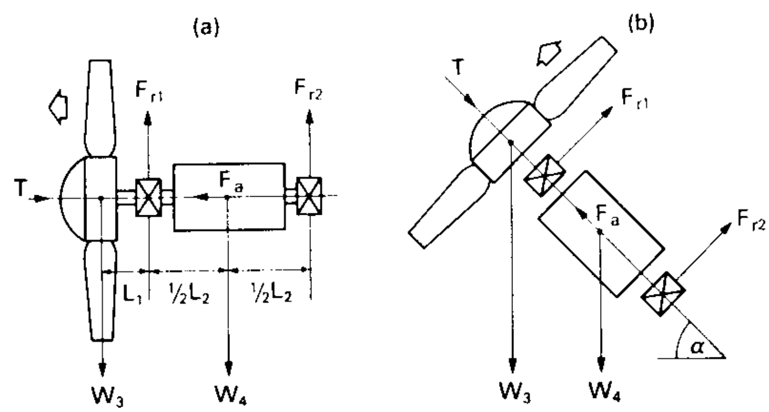
$$\mathbf{T\ (newtons)} = P_s\ (Pa) \times \frac{\pi}{4} D_e^2\ (m^2)$$

Once again this is a safe maximum value, since deductions should be made for reduced pressure in the air behind the backplate, most of which rotates at half the impeller speed. A double inlet impeller develops no thrust.

Fig. 8.12a illustrates the forces acting on a horizontal rotor (4) driving an impeller (3). The weight force W acting on a mass of m kilograms is mg newtons where g is the acceleration of gravity (9.81 m/s^2).

The radial reaction forces, or journal loads, F_{r1}, and F_{r2} are found by taking moments; the resulting formulae being given below Fig. 8.12. Additional allowances are needed for unbalanced magnetic pull on the rotor and mechanical out-of-balance.

The axial reaction force, or thrust load, of the bearing system F,, must equal the aerodynamic thrust, T. It cannot, however, be shared between two bearings spaced apart. Excessive manufacturing precision would be required, and could not in any case be maintained against thermal expansion. One bearing must have inner and outer races axially locked to take the whole thrust.



Horizontal spindle ($\alpha = 0$)

$$F_{r1} = \tfrac{1}{2} W_4 + W_3 (1 + L_1/L_2)$$
$$F_{r2} = \tfrac{1}{2} W_4 - W_3 (L_1/L_2)$$
$$F_{a1} \text{ or } F_{a2} = T \text{ (fan blowing left or right)}$$

Vertical spindle ($\alpha = 90°$)

$$F_{r1} = F_{r2} = 0$$
$$F_{a1} \text{ or } F_{a2} = W_3 + W_4 + T \text{ (fan blowing upwards)}$$
$$= W_3 + W_4 - T \text{ (fan blowing downwards)}$$

General case

$$F_{r1} = \tfrac{1}{2} W_4 \cos\alpha + W_3 (1 + L_1/L_2) \cos\alpha \qquad (85)$$
$$F_{r2} = \tfrac{1}{2} W_4 \cos\alpha - W_3 (L_1/L_2) \cos\alpha \qquad (86)$$
$$F_{a1} \text{ or } F_{a2} = (W_3 + W_4) \sin\alpha + T \text{ (blowing up)} \qquad (87)$$
$$= (W_3 + W_4) \sin\alpha - T \text{ (blowing down)}$$

Fig. 8.12 Forces acting on the bearing system.

For moderate thrust, bearing 2 is often the one locked, bearing 1, which must carry the heavierjournal load, being freed of thrust by giving a sliding fit to the outer race, or by making it a roller bearing. For heavy thrust, however, it may be better to make bearing 1 capable of carrying the combined journal and thrust load. Bearing 2 is then a light duty type, and a ball bearing is to be preferred to a roller bearing; axial spring loading in the housing can slide the outer race to take up the inevitable clearance between balls and races.

Fig. 8.12b illustrates the case of a shaft inclined at an angle to the horizontal. Forces W_3 and W_4 are resolved along and at right angles to the shaft with the results summarised below the diagram. The thrust loading is increased for fans blowing upwards (Form AU, BU) reduced for fans blowing downwards (Form AD, BD).

When the impeller shaft runs in its own bearings its weight, W_4 can be neglected. The bearings must not be too close together or the increase in L_1/L_2 will make F_{r1} excessive. A belt drive will apply a substantial radial force to a pulley outside bearing 2. Being of somewhat uncertain value it is generally sufficient to suppose that the belt pull is wholly counterbalanced by an equal journal load F_{r2} on bearing 2. Bearing 1 will then carry only the impeller journal load while the thrust load will be unaffected.

The belt pull T_b depends on the power transmitted, W_m, the effective diameter of the smaller pulley, D_b, and its rotational speed n, in accordance with this formula:

$$T_b \text{ (newtons)} = \frac{K_b \times W_m \text{ (watts)}}{\pi \times D_b \text{ (m)} \times n \text{ (rev/s)}}$$

K_b is a factor accounting for the excess of the actual belt tension over the minimum required to transmit the power. An estimate for an accurately adjusted belt can be derived from the maker's data, but the following range will cover the majority of cases:

V-Belts	$K_b = 2 \text{ to } 2{\cdot}5$
Flat belts	$K_b = 3 \text{ to } 4$

8.8.4 Example of bearing capacity

The capacity of a ball bearing under combined radial and axial loading is best explained by means of an example. Fig. 8.13 relates to a 1600mm axial fan operating at 16.2 rev/s (975 rev/min) on a D250 electric motor. The mass of the rotor is 153 kg and of the impeller 92 kg and the fan is to be capable of working with the spindle at any angle to the horizontal. The drive end bearing is a 70mm medium ball journal to carry the thrust load, and a life of 20,000 hours is required. Applying formulae 85 and 87:

$$\begin{aligned} F_{r1} &= \tfrac{1}{2} \times 153 \times 9{\cdot}81 \cos\alpha + 92 \times 9{\cdot}81 \left(1 + \frac{150}{600}\right) \cos\alpha \\ &= 1880 \cos\alpha \text{ newtons} \end{aligned}$$

$$\begin{aligned} F_a &= (153 \times 9{\cdot}81 + 92 \times 9{\cdot}81) \sin\alpha + T \\ &= 2400 \sin\alpha + T \text{ newtons} \end{aligned}$$

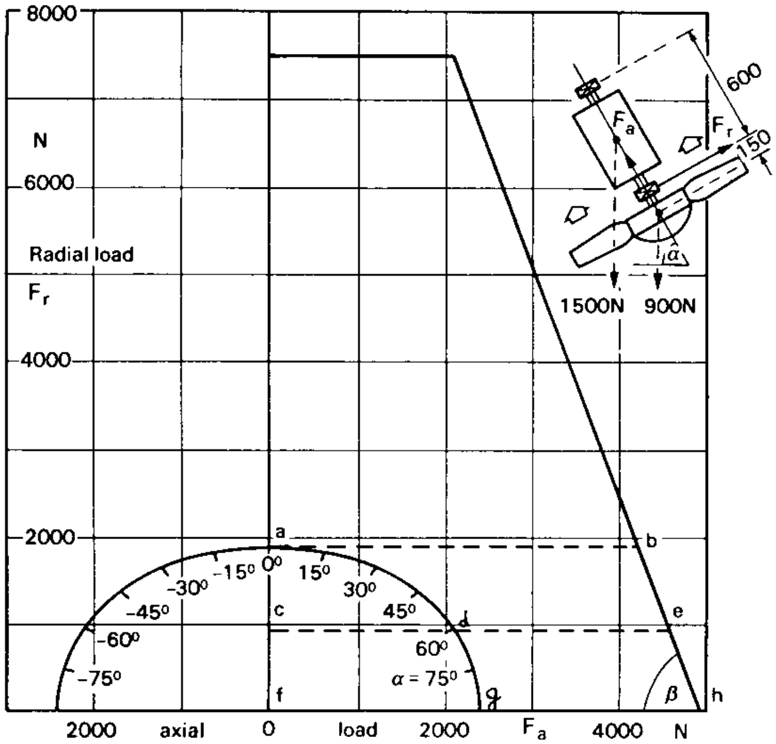


Fig. 8.13 Example of drive end bearing loading.

The elliptical plot at the bottom of Fig. 8.13 shows the relationship between F_r and F_a with T = 0 for all angles between + 90° (blowing upwards) and - 90° (blowing downwards). Unbalance forces are neglected.

The straight lines on Fig. 8.13 show the combined journal and thrust capacity of the 70mm bearing at 975 rev/min for a life of 20,000 hours. A radial, journal, load of 7500 N can be carried with no influence from thrust until the latter reaches 2100 N. From that point on the radial capacity is steadily reduced until it has vanished altogether when the thrust has reached 4900 N.

Reverting to the example, with horizontal spindle, point a, the radial load is 1880 N and the thrust capacity ab = 4200 N. With vertical spindle, blowing upwards, fg becomes the weight load of 2400 N and the remaining capacity for thrust, gh, is 4900 - 2400 = 2500 N. At an intermediate inclination the weights of the components apply a radial load fc and an axial load cd to the drive end bearing. This leaves a thrust capacity de for aerodynamic loading, and de passes through a minimum of 2420 N when α = 65°, obtained by applying the condition tan α = (1880-2400) tan β

The gross area of a 1600mm diameter impeller is 2.01 m^2. Therefore the maximum fan total pressure which can be maintained at any angle to the horizontal is 2420/2.01 = 1200 N/m^2 or Pa for a bearing life of 20,000 hours.

8.8.5 Bearing ratings

The internationally accepted method of rating ball bearings for combined journal and thrust loads is given in ISO 8281, and is used in bearing makers' catalogues. It is a little complicated and Table 14.22 and Figs. 14.3 and 14.4 in Chapter 14 have been devised to simplify estimation.

The table gives the pure radial, or journal, load that can be carried for 20,000 hours at the speed quoted. Actually the life is not determined in terms of hours, but of the total number of revolutions made. Thus the life in hours will be shorter the higher the speed as indicated by the nomogram of Fig. 14.3. This figure gives the factor K_B for ball or K_R for roller bearings by which the basic capacity, P_B, P_A or P_R should be multiplied to take account of speed and life requirements. Further multiplication by K_J and K_T give the journal and thrust capacities for any combination of the two. It will be seen that K_T for a ball bearing is dependent on the speed-life factor K_B, but this is not the case for an angular contact bearing.

The base speed n_o rev/min is so chosen that the static capacity C_o in the makers' tables is equalled by the dynamic capacity for 2000 n_o revolutions. 2000 is an arbitrary multiple chosen to give reasonable values of n_o, but by making it a constant one uniform diagram, Fig. 14.4, is obtained for all sizes and speeds of bearing.

It should be mentioned that, while identical externally, different makes of bearing have differing internal ball and track designs. Basic capacities therefore differ and the highest or lowest values are by no means always peculiar to one maker. For Table 14.22 the lower ratings have been selected from two major manufacturers' catalogues. For precision, preliminary selections thus made should be checked against the catalogue of the maker used.

A life of 20,000 hours is reasonable for ordinary general-purpose use. It equals 10 years each of 50 weeks of five days of eight hours. Other lives may be selected on the following lines:

10,000 hours	**Where the running time is a few hours a week only, or first cost is more important than life.**
20,000 hours	**General industrial use on an 8 hours a day basis.**
50,000 hours	**For important plant running 24 hours a day.**
100,000 hours	**Large machines requiring an exceptionally high level of reliability.**

Bearing life is a statistical concept only, and loads are rarely maintained sufficiently accurately to warrant much precision of estimation. The nominal life is the figure attained or exceeded by *90%* of a large batch on test. About *96%* should survive for half the life and 50% may be expected to last five times the nominal value. Of course, badly fitted, poorly lubricated, or dirty bearings can fail very quickly.

The comparative journal capacities of roller bearings are included in Table 14.22. These, of course, have no thrust capacity. The speed dependence of roller bearings is different from that of ball bearings, the load for equal life being proportional to:

Ball bearings	$(n_o/n)^{0.333}$
Roller bearings	$(n_o/n)^{0.3}$

Angular contact ball bearings are very useful for increasing thrust capacity. They are made with differing angles of contact between balls and races, and the angle with the highest capacity usually available 40° (from the radial direction) is selected for listing in Table 14.22. They should be used in pairs or as a single double-row assembly and will then take thrust in either direction as well as radial loads. The shape of the combined load diagrams is no longer dependent on life, which leads to simpler formulae.

8.6.8 Bearing lubrication

Ball and roller bearings require lubrication for three main purposes:

To prevent corrosion. While the balls and races are made from corrosion resistant chromium steels, their highly finished rolling surfaces would soon lose their perfection if subject to moisture attack.

To lubricate the rubbing surfaces of the cage against the balls or rollers.

To preserve an oil film, even if only of molecular dimensions, to separate the metal surfaces in spite of the high pressures to which they are subjected. If this film is lost, local welding will occur, minute particles will be torn from the surfaces, and failure will result.

Greases are commonly used, and should be of a premium grade approved as a result of rig tests by the bearing manufacturers. Lithium base greases with a rust-inhibiting additive have the most general application. General-purpose grades may be used down to - 10°C and up to 100°C grease temperature, which may correspond to 70° to 80° ambient temperature with averagely well-cooled bearing housings. Special greases containing diester or silicone oils may extend the range down to - 40°C or up to 150'C. Standard bearings should not, however, be subjected to more than about 125°C temperature, although specially heat treated grades are available up to 350°C.

Bearing makers recommend that not more than 30% to 50% of the available space in a bearing housing should be filled with grease. This is particularly important at high speeds where the churning action of the balls forcing their way through would raise the temperature and cause rapid loss of oil. Where lubricators are fitted this means that an escape path for excess grease must be provided-either through a clearance around the shaft or by means of a relief valve. Dirt is fatal to all bearings, and great care must be taken to prevent its entrance along with fresh grease.

Bearings and bearing housings cannot be "sealed for life" except for very. intermittent use or low speed. The grease will deteriorate and must be periodically replaced, either through lubricators or by dismantling, washing and repacking the bearings with fresh grease. The latter procedure is to be preferred if performed with due care to avoid damage, but re-lubrication through nipples or "Stauffers" is serviceable enough provided it is done with care and discretion. There should be a regular maintenance schedule, and the lubricators must be operated neither too often nor at too long intervals.

The re-lubrication intervals recommended by one bearing maker are summarised in Fig. 8.14. The chart applies to ball bearings at grease temperatures not exceeding 70°C, which may be taken as corresponding to 50°C ambient at low or 40°C ambient at high speeds. The following factors may be applied to the re-lubrication interval at other temperatures, and roller or angular contact bearings.

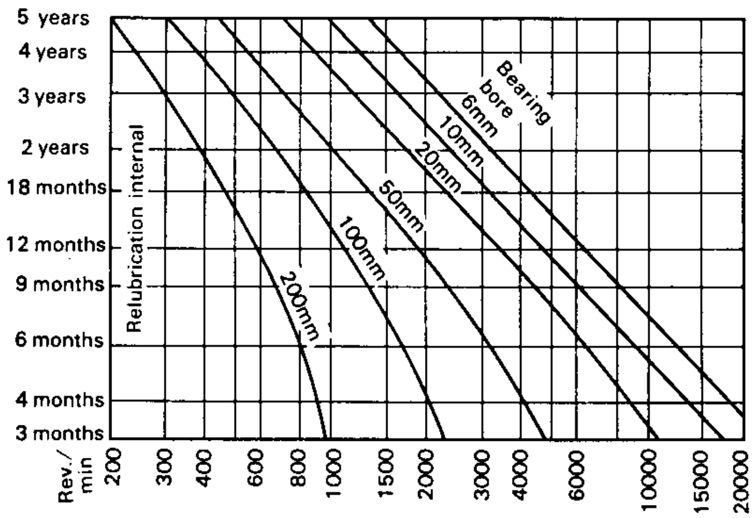
Table 8.5
Factors for re-lubrication interval

Ambient temperature	Grease temperature	Standard ball bearings Multiply by	Roller or angular contact Multiply by
20° to 30°C	50°C	2	1
40° to 50°C	70°C	1	$\frac{1}{2}$
55° to 65°C	85°C	$\frac{1}{2}$	$\frac{1}{4}$
70° to 80°C	100°C	$\frac{1}{4}$	$\frac{1}{8}$

A grease is a honeycomb structure of tiny cells containing oil, the walls being thin membranes formed by emulsification with sodium, calcium or lithium soaps. The function of the cellular structure is to hold the oil in place in bearing housings from which free oil would escape. In contact with the moving elements of the bearing the cells are ruptured and oil is released. A grease that is too *"soft" will* lose all its oil too soon. A "hard" grease will not release it quickly enough for the application.

It is dangerous to re-lubricate with a different grease from the one already in the housing. If they are incompatible the resulting mixture will be unduly *soft.* On the other hand, grease may harden simply through standing idle in the housing. The grease should be examined if a new machine has stood for a year or more before entering service.

The lubricating properties of a grease are entirely due to the oil content, and the oil can be used perfectly well by itself for rolling bearings. The difficulties are solely those of a practical nature. Efficient seals are necessary to prevent oil from escaping, and this makes horizontal spindle operation simpler than vertical. An oil bath should reach the centre of the lowest ball or roller and is suitable for low speeds only. At higher speeds the oil should be circulated through a filter, and changed as oxidation develops. The highest speeds necessitate an oil mist directed at the bearing from an atomiser fed with dry compressed air. High temperatures can be dealt with by passing the circulating oil through an oil cooler, or by water cooling the housing.



Grease temperature 70°C. Age-resistant grease.
For other bearings and temperatures see Table 8.5.

Fig. 8.14

Re-lubrication intervals for continuous-running ball bearings.

Good quality mineral oils are used for the oil lubrication of rolling bearings, and it is important that the oil viscosity should have the right value at the working temperature. This value is dependent on the linear speed of the balls or rollers, and on the load. Too low a viscosity would violate the hydrodynamic conditions for establishing an oil film at the rolling surfaces. Too high a viscosity would cause excessive friction. Fig. 8.15 gives one maker's recommendation of optimum viscosity at working temperature taking the loading corresponding to 20,000 hours' life. For well-cooled bearings the working temperature and corresponding viscosity might be taken as 40°C. Oil makers will quote the viscosity at various temperatures, but SI units may not be used. A comparison with Redwood and other scales is given in Table 14.15, Chapter 14.

Plain oil-lubricated sleeve bearings are still used for some large low speed fans, or for very small domestic fans in the interests of silence. Their engineering aspects are outside the scope of this book.

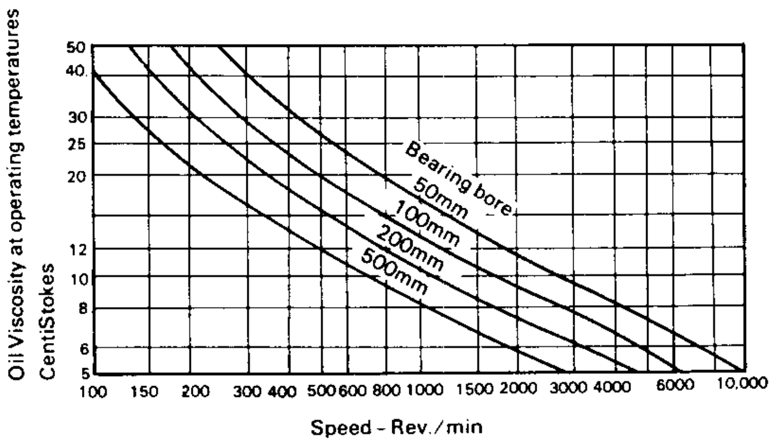


Fig. 8.15

Oil viscosity at operating temperature for ball or roller bearings.

CHAPTER 9

Fan duty control and energy saving

Speed control of the fan driving motor was discussed in the last chapter. Other methods of controlling the performance may well be more favourable in particular circumstances, and a comparative study is the subject of this chapter.

Often the full volume and pressure for which the fan is selected are not required continuously and to leave the fan operating as if they were is very wasteful of energy. Half or more of the kilowatt-hours supplied to the motor can commonly be saved by adjusting the fan performance automatically to match requirements and fuel economies resulting from control of the air duty may multiply the savings. A significant UK government exercise was reported in 1974. 306 office and public buildings were fitted with advanced heating control systems at an average cost of £1,500: the resulting annual saving in fuel costs per building was £860 (that is 30% to 50% of the original annual cost). This is good business under any accounting system.

Variable volume flow is a feature of several of the best air conditioning control systems currently available and its inclusion in new plant will enable advantage to be taken of new ideas in a rapidly developing field.

In the chemical, food processing, and many other industries control is necessary to maintain consistent quality. Variations in throughput, raw material analysis, ambient temperature, etc. must be compensated, and if this can be done by adjusting the flow of air or gas there will be accompanying savings in circulating energy.

9.1 Step-by-Step Systems

9.1.1 Multi-speed motors, described under *8.2.4,* page 150 are probably the lowest in first cost of all control means. They are also, at the speeds available and in applications with constant system resistance, the most efficient. This is because the fan operates at a constant point on the non-dimensional system characteristic so that both static and total efficiencies remain constant, while the motor efficiency is very little below the optimum for each duty. They are not suitable for constant pressure applications since, if the pressure can be developed at low speed, the fan will be operating well below optimum pressure at high-speed.

The chief drawback of multi-speed motors for fans is the size of the step to the first regulated speed. This may prejudice their consideration except for simple applications such as ventilation where the criteria for successful operation are somewhat vague. The maximum ratio of the first regulated speed to full speed depends on the pole number of the motor at full speed (see Table *8.1)* in the following manner:

Motor poles at top speed	2	4	6	8	10	12
Regulated speed/full speed	0·50	0·67	0·75	0·80	0·83	0·86

In the case of belt- or gear-driven fans it may be noted that some choice is possible by selecting the top motor speed.

9.1.2 Fans in parallel can obviously be used for flow control by switching off one or more of the units. This system is mainly used with axial fans for relatively large volume flows. A line of parallel fans may also best suit the building or plant layout.

It is essential to block the passage through the fan casing against reverse flow when the fan is not running. This should be done by fitting air- or motor-operated automatic shutters at inlet or outlet to close while the fan is slowing down.

It is best also to pass the combined air flow at relatively low velocity through a plenum chamber between the fans and the system and to choose fans having a fairly low ratio of outlet velocity pressure to fan static pressure-secured if necessary by fitting outlet diffusers. Otherwise the mismatch, inevitable at some duties, between the fan outlet velocity and the system velocity will waste energy.

The characteristics of fans in parallel are illustrated in Fig. 9.1. for the case of three identical units. Volume flows at constant pressure are of course proportional to the number of fans running. The dotted system characteristics show that constant pressure is ideal for fans in parallel, fan efficiency being maintained at peak value. In such a system fans would be switched in to maintain the plenum pressure applied to a series of resistive outlets as fresh outlets were opened up. On the other hand constant resistance systems can only be operated efficiently over a fairly small control range.

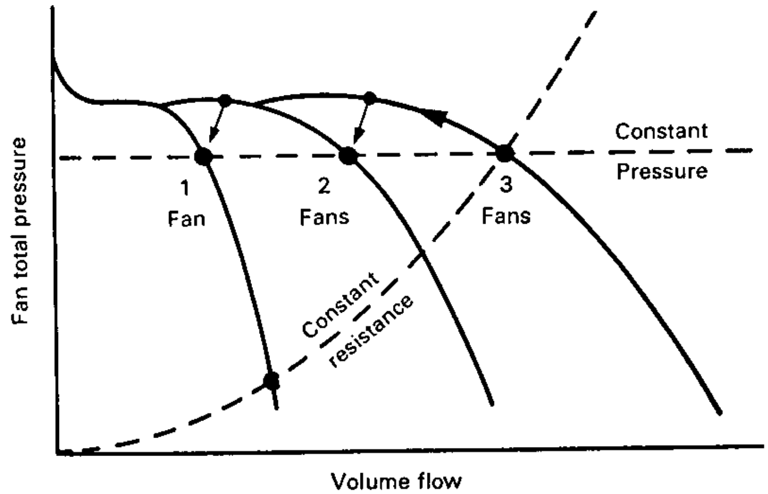


Fig. 9.1 Duty control by fans in parallel.

A stable characteristic is important for fans in parallel. Fig. 9.2 illustrates the danger of using a fan with a marked drop in pressure at restricted volume flow. The dotted system resistance line shows the volume flow per fan *on the assumption that* two fans share the volume flow equally between them, operating at A. If one fan is shut down the other will operate at point B, carrying the whole volume flow in place of the half share plotted at E. If the shut down fan is now re-started it will be unable to move out of the stalled region finishing up at C, while the other operates at D. The average volume flow per fan is now at F well short of the intended volume and pressure point A.

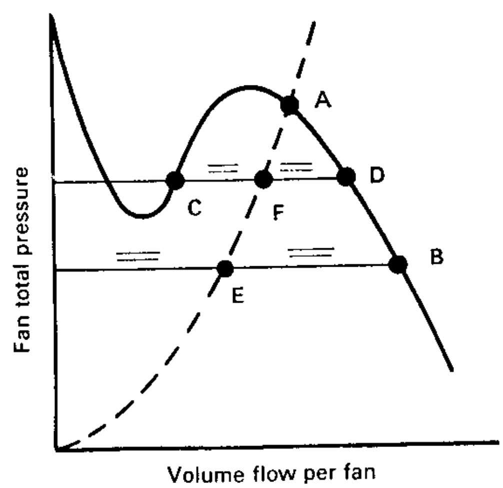


Fig. 9.2 Effect of unsuitable fan characteristic.

9.1.3 Fens in series can provide for duty control if of the contra-rotating axial type (see 7.6.4 (F) *).* When the motor driving one of the impellers is disconnected from the supply, it will idle at about two-thirds of full speed, absorbing around 50% of fan velocity pressure in aerodynamic loss. Centrifugal fans and axial fans with guide vanes have much greater idling losses and are not used for this form of duty control.

As indicated by Fig. 9.3 fans in series are best suited to pressure control at fairly constant flow rate (constant volume system) ; not a great deal of fan efficiency is lost on a constant resistance system, however. Since the pressure rise across each driven impeller depends on the swirl

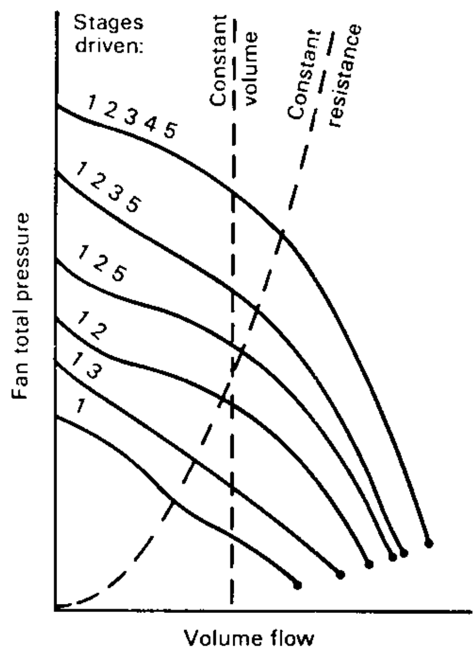


Fig. 9.3 Duty control by five contra-rotating fans in series.

preceding it, the overall pressure is not simply proportional to the number of impellers driven. The switching order is important to avoid excessive counter-swirl which would overload one of the motors, and the makers' advice should be sought on this point.

Contra-rotating fans in series are best suited to applications requiring pressures of 1000 Pa upwards. Simple standard impellers can be used, not the more specialised mechanical and aerodynamic designs needed to develop such pressures in a single stage. Motor costs are not much affected over 10 kW rating (two 50 kW motors may well be slightly cheaper than one for 100 kW) and rather less noise is generated.

The added facility of step-by-step duty control is, therefore, obtained very reasonably.

9.1.4 energy saving. Throughout this chapter a uniform system will be adopted for assessing the efficiency of a control system. The reference standard will be the energy consumed in the absence of control - that is to say the power input to the driving motor at full fan duty multiplied by the time per day, week or year during which any air flow is required.

As a rule the volume or mass flow is the quantity actually required by the user. This requirement is supposed to vary over a *control range* expressed as a percentage of the maximum flow, at full fan duty. On the rarer constant volume systems the fan total pressure is the variable. For comparison purposes it will then be assumed that any volume flow within the control range is equally likely - if the range is split into a

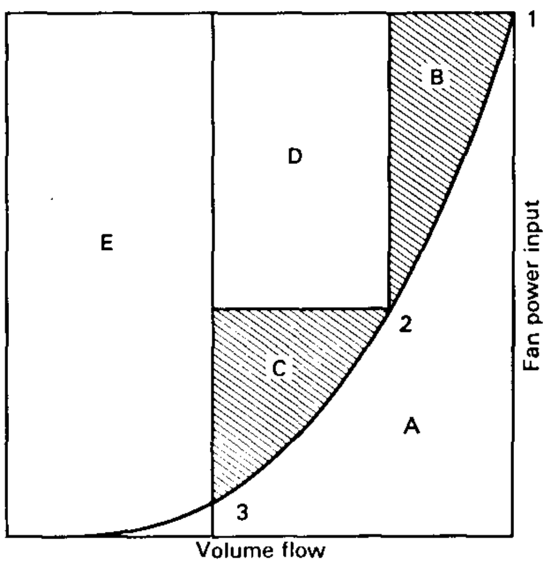


Fig. 9.4 Power and energy consumption with two-step control.

number of equal intervals of volume flow, an equal time is spent in each. The case where the distribution of volume flow with time is known is touched on in 9.6.

A constant system resistance case is sketched in Fig. 9.4. The pressure varies with the square, and the power input with the cube of the volume flow, and the latter is plotted along 1 2 3. A two-step control is considered, without significant variations in fan or motor efficiency. The fan operates at full power until the requirement has fallen to point 2, from which point the reduced power suffices until the end of the control range at 3. The energy saved by the two-step control is represented by area D (power reduction x proportionate time at reduced power). The reference power is the whole area A + B + C + D and the percentage ratio giving the *control saving* is 100D/(A + B + C + D).

For any control range 1 to 3 there is an optimum volume flow 2 at which to make the change from full to reduced power for maximum control saving. This can be calculated and its location is shown by the broken line on Fig. 9.5, as a function of the control range. As well as the two-step case with optimum location of change point *2,* the optimum three-step case has been calculated, and also the most efficient control possible, continuous control, in which the only energy used is that for the pressure and volume flow actually required from moment to moment.

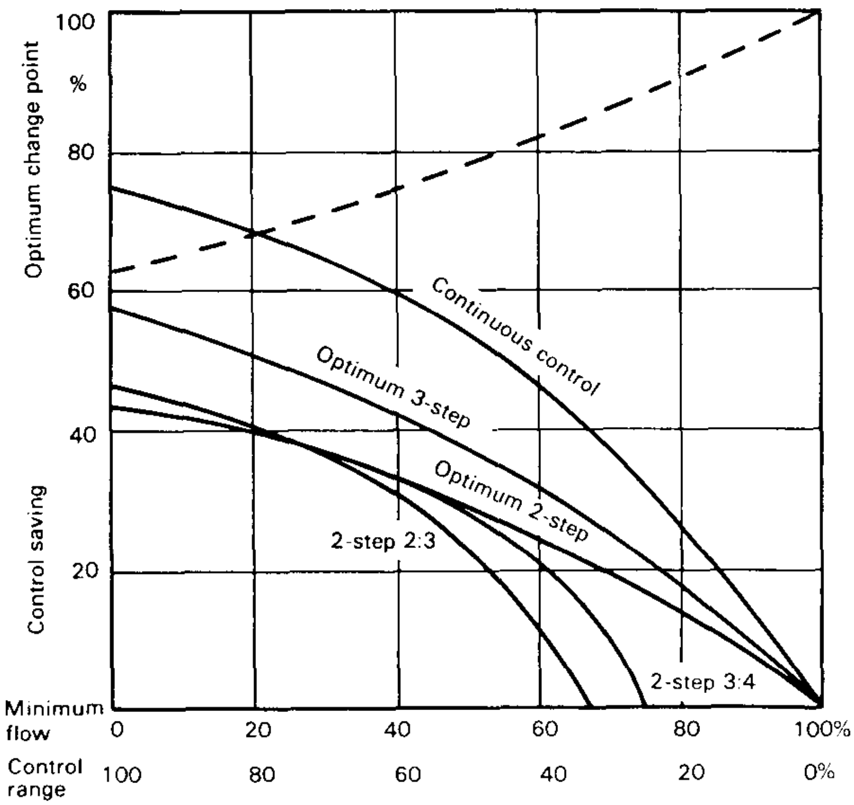


Fig. 9.5 Control savings with uniformly spread volume flow requirement.

Finally, the control savings for two-speed *4-6* pole *(2:3)* and *6-8* pole *(3:4)* motors are shown. While ineffective for a limited volume reduction it will be seen that these are practically as good as the optimum two-stage change point for control ranges exceeding *60%* and *40%* respectively. This would not be the case, of course, if more than proportionate time were spent at the higher volume flows.

9.1.5 Example of stage control. The three methods of duty control described in *9.1.1, 9.1.2* and *9.1.3* are combined in this example. The fans are used to ventilate a 1.*5* km vehicle tunnel with a capacity of

2,000 cars per hour joining Kowloon and Shatin at Hong Kong. There are supply and exhaust units at each end of the tunnel and each of the four plants consists of 8 stages of 1500mm axial fans in four parallel sets of contra-rotating pairs. Each impeller is driven by a 40 kW 970 - 485 rev/min two-speed induction motor.

The fans are connected to supply and exhaust ducting on the transverse principle (see Chapter 13), and the carbon monoxide content of the air is monitored at several sampling points in the tunnel. By switching the fans in different combinations the volume flow per unit can be varied from 15 to 140M^3/s in eight steps according to the pollution level, the top rate being for emergency use only in case of a vehicle fire. Since all the stages are not normally required, spares for maintenance or breakdown are minimised.

Fig. 9.6 shows the fan characteristics for the combinations usually employed, and their working intersections A to H with the system

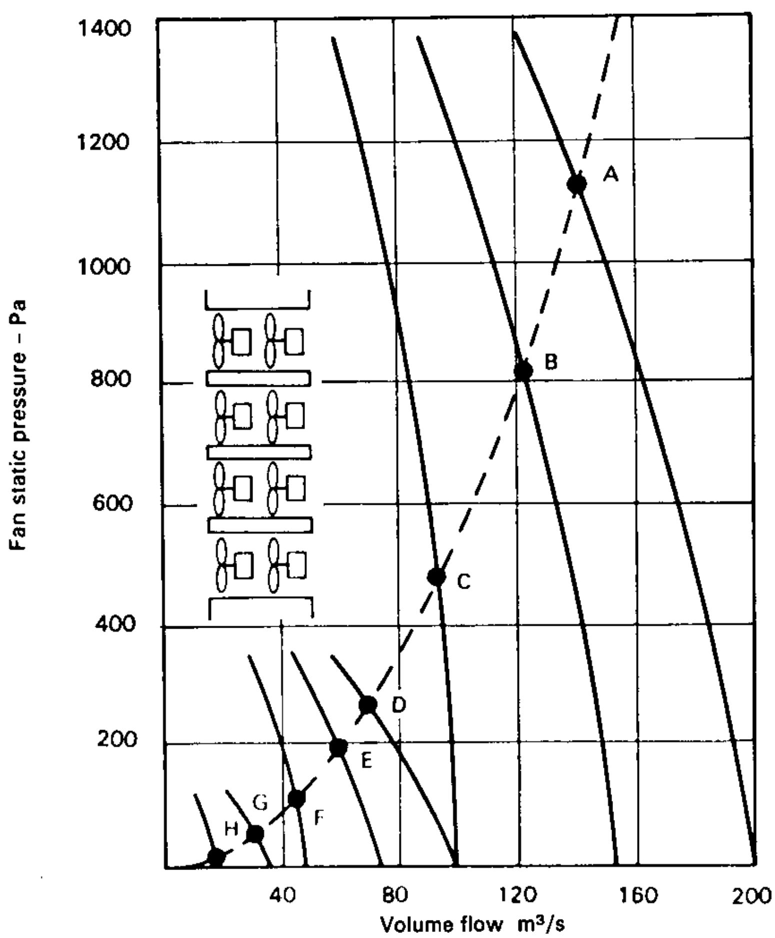


Fig. 9.6 Example of stage control.

characteristic of the installation. The corresponding data for each operating point are listed in Table 9.1. It will be seen that the average daily energy consumption of 3200 kWh for the four plants compares with 103 x 24 x 4 = 9900 kWh to maintain the full normal duty of 90m3/s, a control saving of (9900 - 3200)/9900 = 68%.

Table 9.1

Ventilation of vehicle tunnel

Point Fig. 9.6	No. in Parallel	No. in Series	Speed rev/min	Fan Eff. %	Motor Eff. %	Vol. Flow m^3/s	Input kW	Average daily hours	kWh
A	4	2	975	82	91	142	254	E	
B	3	2	975	81	91	120	178	E	
C	2	2	975	76	91	91	103	4	412
D	4	2	485	82	82	75	34	10	340
E	3	2	485	81	82	60	24	—	
F	2	2	485	76	82	45	15	—	
G	2	1	485	55	80	28	6	4	24
H	1	1	485	30	80	14	4	6	24
									800

E: For emergency use only.

9.2 Continuous Speed Control

9.2.1 A variety of fan drives by electric motors capable of stepless speed variation were discussed in Chapter 8. It is also possible for a constant speed motor to drive a variable speed fan through some form of steplessly variable transmission. Many different principles have been exploited and examples are to be found driving fans in test and research laboratories and in heavy process equipment where specialised mechanical design is justified.

However, they are not often found in general-purpose installations. The standard fan and its electric motor drive are commonly capable of 20,000 to 50,000 hours' life with the bare minimum of maintenanceeven sometimes with none at all. Variable speed transmissions are seldom designed with that kind of reliability in mind and this may be the reason for their comparative rarity. It is also the case that nearly all variable duty fan systems are designed for automatic, or at any rate remote, control. A variable ratio fan transmission should therefore be capable of operation by pneumatic or electric servo motors or thrusters.

9.2.2 Examples of variable speed transmission

Some of the commonest systems are briefly listed here:

V-belt drives are made with pulleys of variable effective diameter secured by varying the width between the inclined flanges. Ratings up to about 10 kW are available, and this is the design most commonly found in test laboratories with manual control-although units with motorised control are also available.

Fluid couplings can be operated with continuous slip if fitted with scoops or other means of controlling the internal fluid circulation.

Losses are typical of slipping devices and they are most useful to ease the motor starting load with a capacity for occasional short time regulation.

Torque converters are similar in effect to fluid couplings with other than 1 : 1 speed ratio. Away from the speed reduction for which they are designed they entail losses of the same order as those of slipping devices.

Eddy-current couplings have a solid iron driven element carrying a magnetic field generated by the driving element. Currents in the iron similar to those in a squirrel cage rotor arise as the slip increases and produce torque which is capable of close control by varying the relatively small magnetising current. The fan speed changes to balance the applied torque.

A fluid-power system has two elements which can be located a little distance from one another. One is a positive displacement pump driven by a constant speed motor but producing nevertheless a variable flow of oil through a variable stroke piston or other means. The oil is passed through high pressure piping to a constant displacement hydraulic motor driving the fan, whose speed will be proportional to the rate of oil flow.

Variable ratio gearboxes are made, usually, with a rolling element under pressure to provide a friction drive at an adjustable radius between driving and driven surfaces.

To these must be added, for heavy industrial use, steam- or gasturbines, geared down to fan speed, and controlled by throttle and governor systems.

9.2.3 Power consumption with continuous speed control

Of course, every control system has its own particular sources of energy loss, dependent also on size, speed, and speed ratio. Nevertheless, two broad categories can be recognised each of which has a basic minimum power consumption over and above which the losses incidental to the particular detailed design must be added.

Variable ratio systems:

Variable V-belt pulleys; variable ratio gear; fluid power drives; variable frequency motor supply; DC and AC commutator motors.

Variable slip systems:

Fluid couplings; torque converters; eddy-current couplings; wound rotor induction motors.

For a constant resistance system the fan efficiency will remain constant over the speed range. The basic power input will fall in proportion to the cube of the speed ratio for variable ratio systems as plotted in Fig. 9.7. In the case of variable slip systems the full fan torque is borne by the driving motor without any "gearing" ratio" reduction; this means that the basic input power is proportional to the square of the speed ratio.

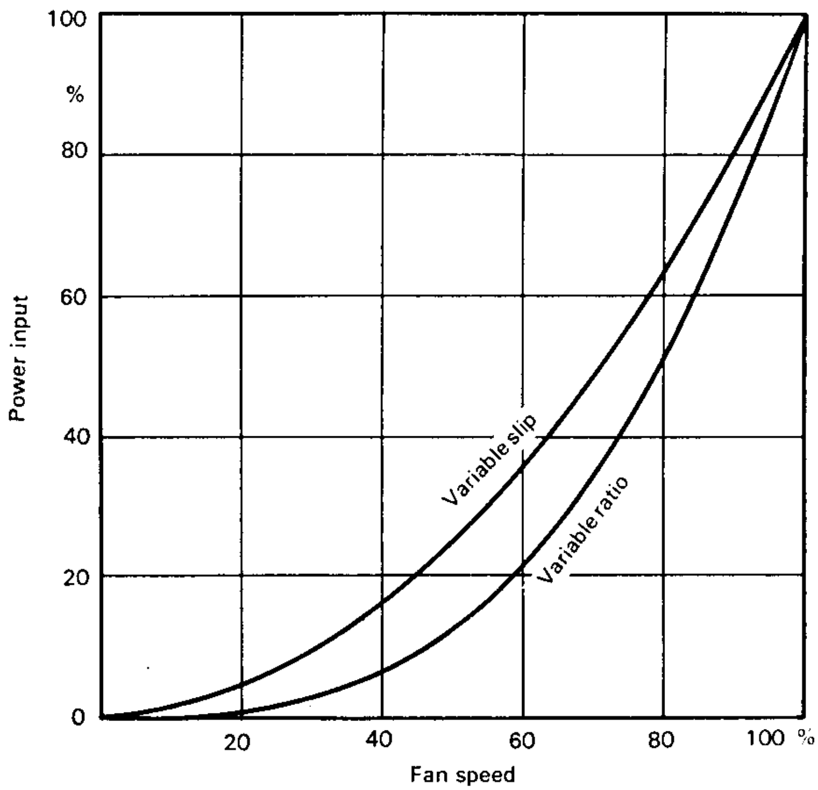


Fig. 9.7 Basic power input with continuous speed control.

The losses additional to the basic power input will be those in the driving motor and in the transmission system which can be determined from the overall efficiency of the assembly at full fan speed. In the absence of detailed knowledge it is a safe assumption that these additional losses will remain constant over the speed range. In fact most of them will fall with speed and load, but not by any means in proportion.

9.3 Damper Control of Fan Duty

9.3.1 Most air conditioning systems and many industrial installations employ dampers to control the air flow, and in consequence to vary the fan duty. While these are effective for their purpose and indeed necessary when division between branches is to be adjusted, they quite fail to realise the potential savings of energy unless accompanied by control of the fan itself.

All dampers serve simply to throttle the flow in the airway in which they are placed. Back pressure is built up by accelerating the air to high velocity and then allowing its kinetic energy to be dissipated in turbulence downstream. They do not have an "efficiency" in energy terms but their characteristics in respect of noise generation, stability, and smoothness of control are important.

While most dampers are merely placed across the airway in which the airflow is to be reduced, an alternative exists which may or may not be preferable according to the fan characteristic and other circumstances. This is the by-pass duct containing a damper which, when opened, permits some air to flow back from the fan outlet to the inlet, thereby reducing the flow in the main system. The same effect can be secured by simply opening a damper in the airway wall downstream of the fan to "bleed off" by-pass air, or alternatively to open an entry port for by-pass air in a suction airway upstream of the fan. In both the latter cases possible disturbance to conditions outside the duct must be considered.

9.3.2 Influence of the fan power characteristic

In the absence of any other form of fan duty control, variation of volume flow by the closure of a damper will move the operating point to a new position on the fan characteristic. This will involve a change in fan efficiency (generally for the worse) and a change in the fan power input. Classification into fans with rising, falling, or non-overloading power characteristics is important in this respect.

Forward-curved fans have rising characteristics, particularly so in the multi-vane type which may have an impeller power at full open flow six or eight times that at no flow. High pressure blowers, paddle bladed fans and others with radial outlet blading have a rising characteristic which is less marked, four or five being more typical ratios of maximum to minimum power.

Falling characteristics are peculiar to the propeller fan which, if allowed to develop centrifugal outlet side flow and corresponding pressure in a diaphragm mounting, may take at no flow 1½ to 2½ times the free flow power.

Non-overloading characteristics are usual in backward-curved centrifugal and axial designs, particularly those with high efficiency. The power taken at the best efficiency point on the characteristic is quite close to the maximum, and then falls away both towards free flow and no flow. Typical power characteristics for these classes are compared in Fig. *9.8,* taking the volume flow and power input at the best efficiency point as 100% in each case.

Assuming that fans have been chosen to operate near the best efficiency point at full duty, it will be clear that series dampers should be used for fans with rising power characteristics, so that fan flow and system flow fall together and the power input is reduced. On the other hand series dampers, or indeed any undue flow restrictions, are bad for a propeller fan; if duty control is necessary the by-pass or bleed-off method is always better for this type-if it can be arranged.

The backward-curved fan is more flexible. There is little prospect of power saving however, what there is being usually better with series than with by-pass damper control. Axial fans depend very much on the

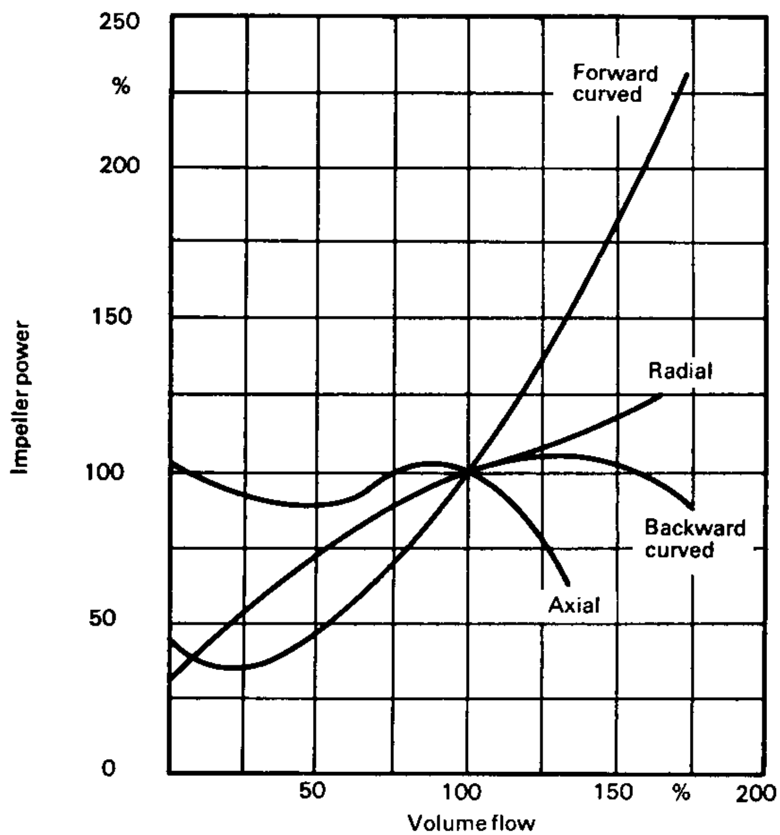
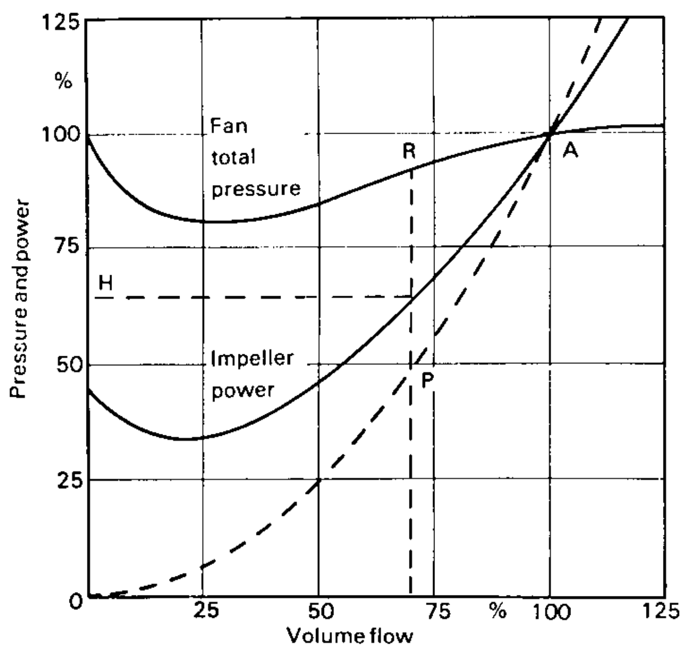


Fig. 9.8 Typical power characteristics.

blade pitch angle; low volume, shallow pitched blading places them in the same class as backward-curved fans. High volume, steep pitched impellers on the other hand should not be subjected to much volume reduction by series dampers or the unstable, stalled flow region may be entered. By-pass control is better, and may permit power saving of *20%* to 40%.

9.3.3 Performance with damper and by-pass control

Figs. 9.9 and 9.10 illustrate the application of damper and by-pass control respectively to a constant resistance, "square-law" system characteristic. The most suitable fan designs for power saving are selected for the illustrations, and it will be seen that a reduction of impeller power slightly greater than the reduction in volume flow is achieved in both cases. The electrical losses in the driving motor are not likely to fall quite so fast, and it must also be noted that a forward-curved fan for series damper control is unlikely to be as efficient at full duty as an axial fan for by-pass control.



A:	Full duty	100% volume flow; pressure and power.
P:	Regulated duty	70% volume flow; 49% system pressure.
R:	Regulated fan duty	70% volume flow; 90% fan pressure.
H:	Regulated fan power	65%, 35% power saving.
PR:		41% pressure drop in damper.

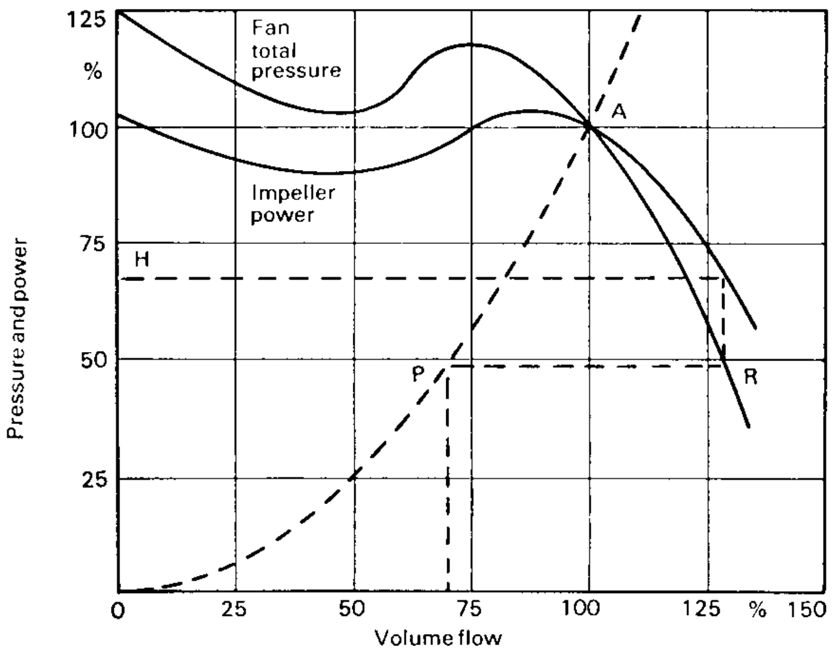
Fig. 9.9 Forward-curved fan with series damper control.

It is interesting to draw a comparison between the best constant speed fan-damper or fan by-pass combinations and variable ratio or variable slip speed control. On *constant resistance* systems the ratio of input power on regulation to full duty input power will be approximately equal to:

(Volume flow ratio)^3 with variable ratio speed control.
(Volume flow ratio)^2 with variable slip speed control.
(Volume flow ratio) with optimum damper or by-pass control.

If the system requires *constant volume* flow with a fluctuating system pressure, a series control damper may be used with the fan remaining at full duty and power input throughout. However, power can be saved by adopting by-pass control and an axial or other fan with a falling power characteristic. The by-pass is adjusted to dump the surplus air when the pressure required is low.

To maintain *constant minimum pressure* in an enclosure (e.g. a dustfree workshop) with a varying air change requirement the control damper should be placed in the outlet from the room. By-pass control of the supply is then functionally possible, but it would be simpler and more economical to choose a backward-curved or other fan with a



A:	Full duty	100% volume flow, pressure and power.
P:	Regulated duty	70% volume flow, 49% system pressure.
R:	Regulated fan duty	130% volume flow, 49% fan pressure.
H:	Regulated fan power	67%, 33% power saving.
PR:		60% volume flow in by-pass.

Fig. 9.10 Axial fan with by-pass control.

fairly flat fan static pressure characteristic and a rising power characteristic over the volume flow range required. The outlet damper would then suffice to control volume flow and efficiency could remain quite high.

Many designs of damper, when partially closed, deflect the air strongly over to one side of the duct in which they are placed (or to both sides in the butterfly case). Placed less than, say, five duct diameters upstream of a fan inlet these will badly affect its performance; if closer spacing is necessary the damper should be of the opposed vane or similar symmetrical, non-swirl-generating type. Similar considerations apply to the flow from a by-pass duct entering the suction duct close to a fan inlet.

Where a by-pass is installed to accept part of the flow velocity inequalities approaching the by-pass will introduce substantial branching losses (see Section 6.7).

9.4 Variable Pitch Fans 9.4.1 General

A principal factor determining the performance of all fans-axial, mixed flow, or centrifugal-is the angular setting of the blades, particularly the angle made by the outlet edge with the tangent to the direction of rotation (α_2 in Fig. 7.24). As this angle is increased the forward component of air velocity on which the volume flow depends is increased while the swirl component on which the pressure depends remains substantially constant. It follows that the volume flow could be controlled if the angular setting of the blades could be altered while the impeller was rotating.

The variable pitch aircraft propeller and the Kaplan runner for hydraulic turbines show that pitch adjustment in the axial configuration can be developed to the highest levels of reliability. Fans with variable pitch axial impellers have been in successful service for many years, though in an unnecessarily limited range of application. With increasing emphasis on flexibility of control and energy economy, both in air conditioning and process work, their use is spreading. Centrifugal impellers have been produced with variable pitch trailing edges to the blades, rather after the pattern of aircraft wing-flaps. However, the mechanical design problems of this configuration are severe and it is seldom encountered.

Variable pitch axial fans can be used with constant resistance, constant volume flow, or constant pressure systems-in fact with any form of system characteristic. The overall energy savings are among the highest available with any form of duty control. A unique feature is the ability to control volume flow down to zero even at constant pressure, and to produce reverse flow if required. Standard pneumatic or electric continuous control system can be employed, and the speed of response tailored to requirements. Noise level falls with reduction of volume flow, whereas it tends to rise with damper or vane control; the fall is not nearly so rapid as it is with speed control, but the use of a multi-speed motor with a variable pitch impeller largely bridges this gap.

9.4.2 Performance of typical variable pitch fan

The aerodynamic design of a variable pitch axial fan can be exactly the same as that of the general purpose adjustable pitch range described in Section 7.6. Fig. 9.11 is an example: a 1600mm fan with downstream guide vanes and a diffuser expanding to a 2000mm diameter outlet duct in which the fan performance is measured. The fan total pressure and inlet volume flow are plotted for a pitch angle range from 32° (at the blade tips) to - 8°; this range could be extended if negative flow was required. Contours of constant fan total efficiency are also plotted from the peak of about 86% down to 50%.

The broken lines A, B, C and D are examples of the range of system characteristic which can be dealt with by this fan.

A is a constant resistance characteristic for which the required total pressure drop equals:

$$0{\cdot}40\,Q^2$$

This is the commonest system, and applies when the required volume flow is delivered (or exhausted) through a system of airways, filters, heater batteries, etc. with the specified total pressure drop varying with the square of the volume flow. In the example the volume flow can be

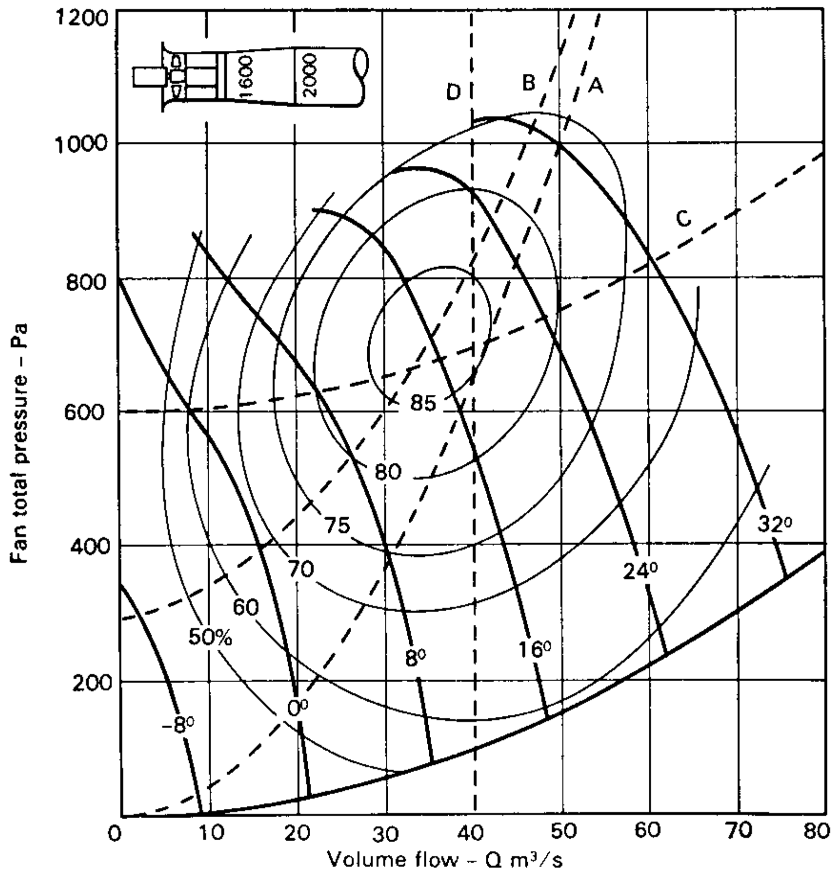


Fig. 9.11

Variable pitch axial fan performance: 1600mm, 975 rev/min

continuously and stably adjusted from 50m^3/s to zero - or to reverse flow if required. The fan efficiency is between 75% and 85% from 50 down to 31 m^3/s, that is from 66 down to 16 kW impeller power input.

B is a part constant pressure, part constant resistance characteristic with a total pressure drop equal to:

$$300 + 0{\cdot}33\,Q^2$$

Characteristics of this type are met with in air conditioning systems requiring a fixed pressure (here 300 Pa) to operate the final room outlet distribution and control units. The rest of the pressure drop arises in the airways and equipment supplying the outlet units. The example shows a volume flow adjustable from 46M3/s down to zero with 75% to 85% efficiency from 46 to 21 m^3/s - 60 to 13 kW input.

C is a constant pressure system maintaining a constant static pressure of 600 Pa in the 2000mm duct. It is assumed that the velocity pressure at the outlet of this duct (which equals 0.06 Q^2) is all lost (though some could well be recovered) which leads to the following total pressure requirement:

$$600 + 0{\cdot}06\ Q^2$$

Characteristics of this type arise when the fan supplies a plenum chamber feeding a number of outlets, only some of which are in service at any one time. These may be process supply points, each requiring a fixed volume flow and pressure, or the plenum may be on the suction side of the fan, drawing from a varying number of fume cupboards, for example. In air conditioning the supply might be to a number of floors in a building, each with its own conditioning unit, and each operating for different periods according to occupancy. The fan pitch would be automatically adjusted to maintain constant pressure or suction in the plenum. In the example the volume flow may be adjusted from $60m^3/S$ to zero and 75% to 85% efficiency is maintained over the wide range of $58m^3/s$ to $18m^3/s$ - 62 to 15 kW input.

D is a constant volume system maintaining a volume flow of $40m^3/S$ over a range of total pressure drop from 1000 Pa down to zero (or negative pressure if required). 75% to 85% efficiency is maintained over a pressure range from 1000 to 400 Pa with corresponding power input from 53 to 22 kW. This type of characteristic is required when a specified volume or mass flow is to be maintained against a variable flow resistance, as might arise for example in the fuel bed of a combustion process, in a flotation chamber or pneumatic transport system for powered material, or across a filter as it becomes fouled. The fan pitch would be automatically adjusted to maintain constant mass flow irrespective of pressure, and the set flow rate could be altered from time to time as required.

9.4.3 Applications of variable pitch fans

The versatility of the variable pitch axial fan will be evident from the above example. It will be seen that over 75% efficiency is maintained over a power input range of 100% to 25% in each variable volume case; efficiency is still 50% at less than 10% of full power. It is not necessary for the whole duty to be dealt with by one fan. Large volumes or pressures can be dealt with by using one or more variable pitch fans in parallel or series with a number of similar fixed pitch units, which are switched in and out to provide coarse control steps, the fine control between steps being provided by the variable pitch unit.

Among successful applications of variable pitch axial fans may be cited:

Air conditioning systems in a variety of designs for commercial and public buildings.

Vehicle tunnels and garages where control of carbon monoxide and smoke is required.

Underground railways for station ventilation and heat removal.

Power stations for forced and induced draught and gas circulation.

Heavy industry, chemical and metallurgical, for process control, drying and cooling.

Textile industry for humidity control.

Oil industry for large scale "fin-fan" heat exchange.

Laboratories for fume cupboards, wind tunnels, etc.

Mines for main ventilation.

Factories for ventilation and pollution control.

9.4.4 Mechanical features of variable pitch fans

Centrifugal forces are predominant in the mechanical design. The centrifugal force on a mass of m kg revolving at n rev/sec at a radius of r metres is:

$$mr(2\pi n)^2$$

For the 1600mm 975 rev/min fan of 9.4.2 $2\pi n$ = 102 radians/sec. Taking M, the mass of one blade, as 3 kg, and its radius of gyration, r_g, as 550mm, the centrifugal force per blade will be:

$$3 \times 0{\cdot}55 \times 102 \times 102 = 17{,}200 \quad \text{N}$$

This is nearly 600 times the weight of the blade, which is 3 x 9.8 = 29.4 newtons and the situation is sometimes spoken of as equivalent to operating in a gravitational field of 600g. g = 9.8m^2/s is the acceleration due to gravity at the earth's surface.

Such a force, and indeed forces several times as great, are within the static capacity of ordinary ball-thrust or needle roller thrust bearings, and these are commonly used to carry the centrifugal force at the blade root while permitting blade rotation with minimum friction. The taper-roller thrust bearings used for aircraft propellers are normally unnecessary for fans.

Levers at the base of each blade translate the common pitch angle adjustment into axial movement of a sliding member within the hub. This may be actuated in four ways:

(a) Automatically, by the expansion of a pneumatic bellows of reinforced rubber within the hub against a spring. The bellows is fed with compressed air through a rotary air seal on the shaft extension.

(b) Automatically, by an external pneumatic or electric thruster through a lever system which applies pressure to the stationary race of a ball thrust bearing the revolving race of which is coupled to the sliding actuator within the hub.

(c) Automatically, by an external pneumatic or electric thruster which applies actuating force as in (b) while the reaction force is transmitted from the hub to the body of the thruster through a second thrust bearing. This arrangement relieves the main fan bearing of control thrust load.

(d) Manually, by means of a screw jack when the fan is at rest.

Fig. 9.12 shows the cross-section of the hub of a variable pitch axial fan with inbuilt pneumatic control of type (a). The air pressure in the bellows is adjustable by the control system to any value between 20 and 100 kPa above atmospheric pressure. For each value there is a corresponding compression of the spring, position of the sliding actuator, and pitch angle of each blade. When the fan is running forces must be applied to each fan blade to keep it at the required pitch angle. Left to itself it would rotate to a position near zero pitch angle where the centrifugal forces on it were in balance. Weights W are attached to the blade root in such a position (at right-angles to the blade) that they apply a counterbalancing turning moment and minimise the actuating force required.

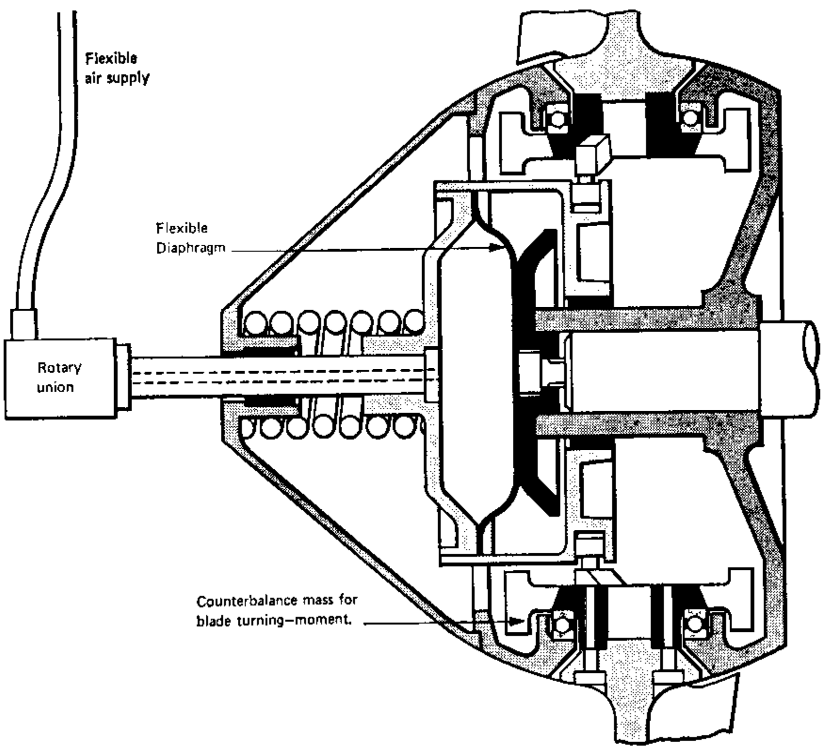


Fig. 9.12

9.5 Vane Control

9.5.1 Principles of vane control

A set of movable vanes is used, placed in the airway at or near the inlet to the fan. They are usually pivotted on radial axes arranged like the spokes of a wheel and are linked together so that they can be moved in unison. Unlike a damper they do not merely throttle the airflow but are designed to produce a controllable degree of swirl in the air entering the impeller. This has the effect of altering the fan performance - moving the fan characteristic across the pressure-volume flow chart in fact. They are thus more efficient than plain dampers, and with some types of fan they are capable of raising as well as reducing the normal fan duty.

In Section 7.15 it was shown that the total pressure rise across any fan impeller-axial, mixed flow or centrifugal - was given (ignoring losses) by:

$$u_2 v_2 \cos \alpha_2 - u_1 v_1 \cos \alpha_1$$

Vane control operates on the inlet swirl angle α_1 With the vanes in the neutral position the inlet flow is axial, cos α_1, and the second term are zero. Swirl in the same direction as the impeller rotation (a, less than 90° and cos α_1 positive) will reduce the pressure rise. Swirl in the opposite direction (a, more than 90° and cos α_1 negative) may raise the fan pressure, unless the impeller blading is incapable of carrying the increased deflection so that α_2 is not maintained.

The impeller power will be decreased or increased in proportion to the pressure. In the first case some of the torque needed to produce the outlet swirl α_2 will have been transferred from the impeller to the vanes; in the second case the impeller has to apply additional torque to counteract that applied by the vanes. Since the guide vanes are not rotating they can apply torque without consuming power, unlike the impeller.

9.5.2 Vane control for centrifugal fans

Vane control is the standard aerodynamic method of varying the performance of a fixed speed centrifugal fan. The radial vane system just described is the most usual, although other arrangements are possible. With flat plate vanes the resistance when set in the neutral position, 90°, is small so that the normal fan characteristic is hardly affected. Vane control is most effective with high efficiency, backward-curved, centrifugal fans, and a typical set of characteristics for such a combination is shown in Fig. 9.14.

Hardly any increase on normal performance is obtainable and angles greater than 90° are not used in this case. The duty may be controlled down to zero with constant resistance, constant pressure, or constant volume flow system provided the vanes are capable of being set for complete closure as dampers; reverse flow is not obtainable. The method is best suited to rather small ranges of duty control. On a constant resistance system, for example, the efficiency has dropped from 85% to 75% with only 10% reduction in volume flow and 30% in impeller power. The equivalent variable pitch axial fan of Fig. *9.11* covered 38% volume reduction and 75% power reduction for the same change in efficiency.

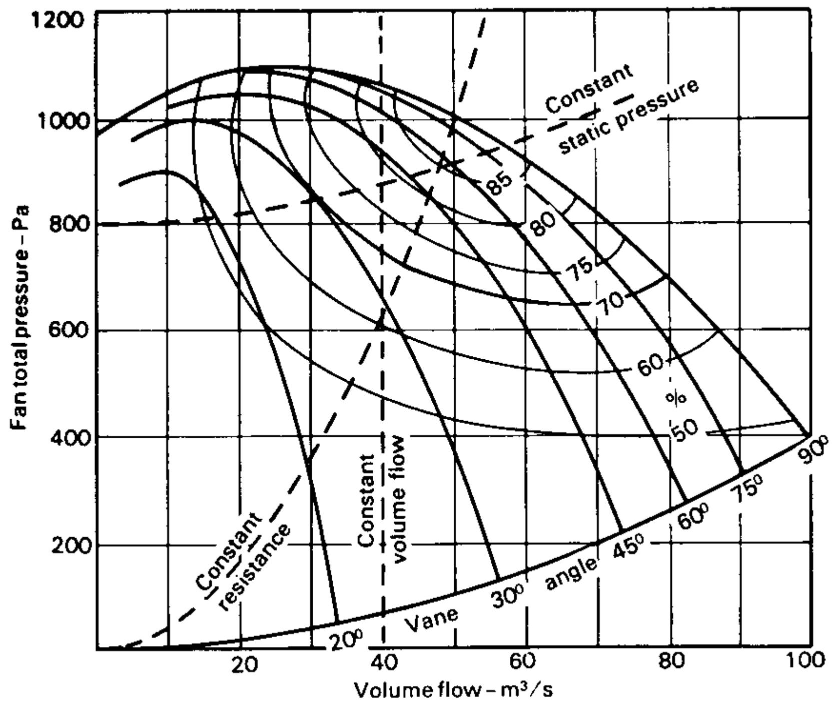


Fig. 9.14 Vane control of a backward-curved centrifugal fan: 2500mm diameter, 400 rev/min.

9.5.3 Vane control for axial fans

Vane control is seldom used for axial fans, being inferior in efficiency and range to variable-pitch control. However, it is lower in first cost and could provide a useful method of regulation for the smaller, simpler units. Some 10% to 20% increase above the normal fan pressure or volume flow is usually available at negative settings, though at reducing efficiencies and increasing noise level.

The method is best suited to constant volume flow operation with reasonably efficient control of pressure down to zero. On constant system resistance a typical unit could be regulated to 60% of normal volume flow at 36% pressure with 30% to 40% reduction in power consumption. At lower flows the power taken rises towards equality with the value for damper control when the vanes are completely closed. At constant pressure vane control is likely to encounter instability problems unless the impeller pitch angle is very small.

9.6 Comparative Energy Savings

9.6.1 Example of control at constant resistance

Fig 9.15 gives a direct comparison of most of the methods previously discussed. The maximum duty is supposed to be $50m^3/s$ at 1000 Pa fan total pressure as in Figs. 9.11 and 9.14. The maximum available control of volume flow is required on a system in which the pressure varies with the square of the volume flow.

The "ideal control" curve represents an unattainable ideal solution in which a fan with an efficiency of 85% maintained at all control volumes is driven without either motor or control losses. At 100% volume flow the power input of 59 kW is taken as 100% and reduces with the cube of the volume flow.

The band marked SP covers the probable range of electrical input power with the more efficient forms of *speed control* discussed in Section 8.2, these include AC commutator (Schrage) motors, induction motors with thyristor control of frequency, or DC motors with thyristor control of voltage. The fan will retain its full efficiency of 85%, but the electrical losses will be greater than those of a constant speed 60 kW induction motor, which would have an efficiency of about 92%.

The band marked SL covers the probable range of electrical input power of variable slip systems. These include wound-rotor induction motors, and standard induction motors driving fluid couplings, torque converters and eddy-current couplings, and the power input is seen to be considerably greater. Neither SL nor SP have been taken below 30% volume flow because of the probability of instability with most methods at lower speeds.

The damper control curves for backward- and forward-curved fans follow the respective power characteristic examples of Fig. 9.8, with motor losses added assuming motor efficiencies to be:

	Full load	$\frac{3}{4}$ load	$\frac{1}{2}$ load	$\frac{1}{4}$ load
Efficiency	92%	92%	90%	80%

In the forward-curved case a probable maximum fan efficiency of 70% at full duty has been taken, compared with 85% for the backward-curved and axial cases. By-pass damper control would be used in place of series damper control for the axial fan with its falling power characteristic, and this limits the volume flow reduction to 40% in the example of Fig. 9.10. Vane control for a backward-curved centrifugal fan (as Fig. 9.14) will introduce a little extra fully open loss, but is clearly better than any type of damper control. At zero flow vane and damper control coincide, merely serving to close off the system.

The variable pitch axial fan shows to great advantage over almost the whole control range; the example of Fig. 9.11 has been taken. The power input at maximum duty is up a little, corresponding to an efficiency of 76% at full duty. This is because the ability to increase performance by raising the pitch angle from the optimum 20° to 32° has been used, thereby maintaining efficiency over a 100% control range. For a narrower range, e.g. 50% volume flow, the 20° pitch angle could be used for maximum duty, giving 85% efficiency there, the fan diameter being increased from 1600mm to 1800mm at the same speed.

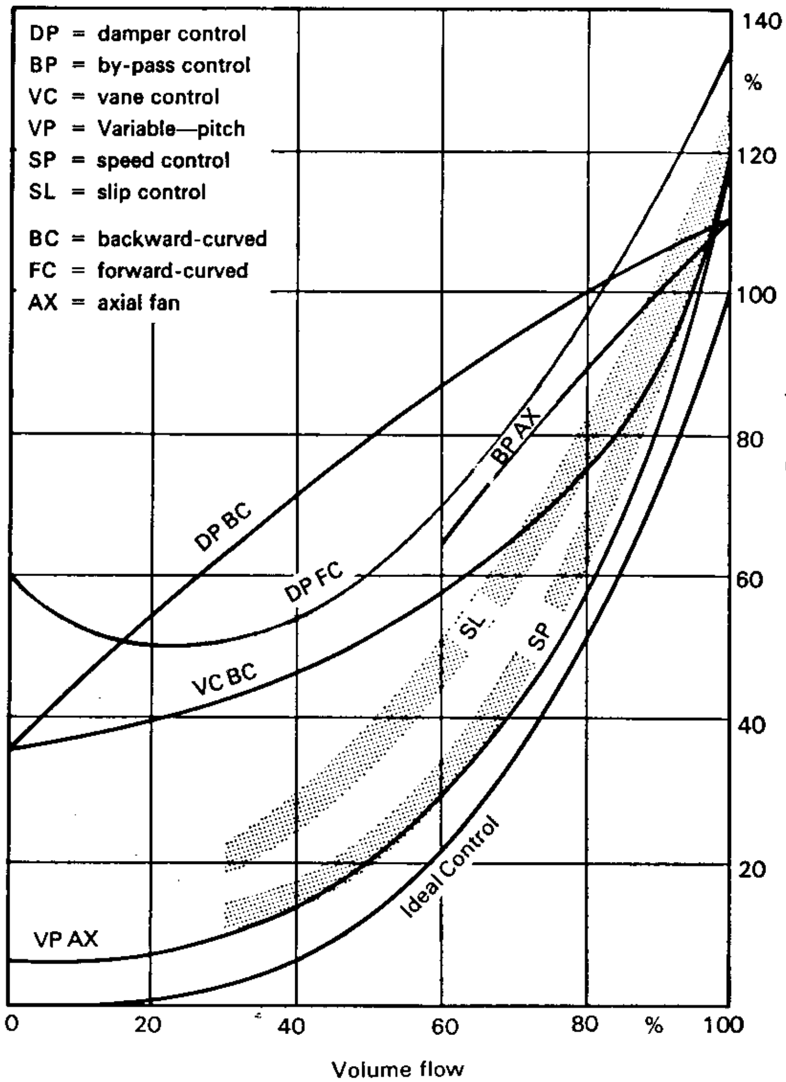


Fig. 9.15 Control methods at constant system resistance.

In Fig. 9.15 "ideal control" represents the mechanical input to a fan which maintains a constant efficiency of 85% over the whole control range. All other curves include the electrical and other losses in the driving motor and control means.

8.6.2 Example of control at constant pressure

In examples summarised in Fig. 9.16 a constant total pressure requirement has been assumed to avoid the complication of allowing for the recovery of any particular proportion of the fan velocity pressure. The

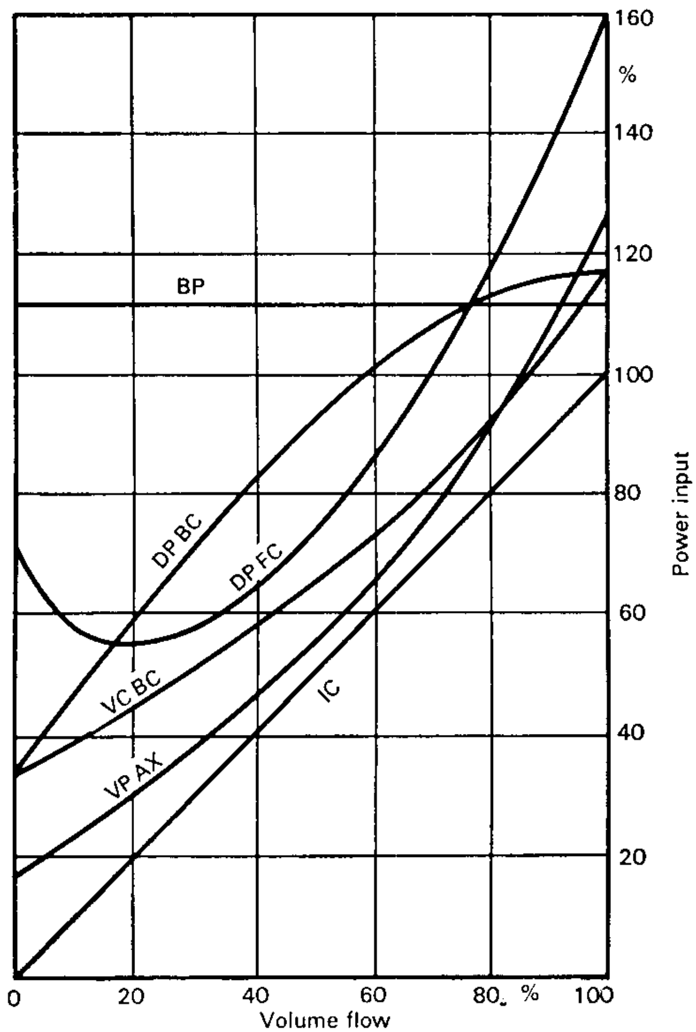


Fig. 9.16 Control at constant total pressure.

symbols have the same meaning as in Fig. 9.15, IC being the power absorbed by a fan of constant efficiency (85%) with no allowance for motor or other driving losses, conditions which are of course unattainable in practice. At constant total pressure, power is then clearly proportional to volume flow.

Of damper control, systems the by-pass (curve BP) is effective for control of volume flow over a range up to 25% reduction; it is applicable to any fan, which continues to operate at full duty and efficiency, the unwanted volume flow being simply dumped. There is no control saving, however, and for a large control range series dampers will be more efficient for centrifugal fans in which there is a substantial fall of power with volume flow. Indeed, if a forward-curved centrifugal fan is used, series damper control is best for all ranges (curve DP FC noting that 70% rather than 85% fan efficiency is assumed).

For a backward-curved centrifugal fan vane control (curve VC, BC) shows to best advantage on a constant pressure system, the example of Fig. 9.14 at 900 Pa coming within 5% of the optimum energy saving for control ranges up to 50% volume flow reduction. The variable pitch axial fan is even better, however, maintaining a good efficiency over a 100% control range. It should be noted that curve VP AX in Fig. 9.16 (corresponding to the example of Fig. 9.11 at 800 Pa) is chosen to suit a 100% control range. For a narrower range the increased pitch angle of 32°, which accounts for the increased power consumption at 100% volume flow, need not be used. A slightly larger fan could be operated over the 80% to 40% range of Fig. 9.16, giving within 3% of the optimum saving over a 50% volume flow range.

9.6.3 Examples of control at constant volume flow

For an application in which a constant volume flow is required, the ideal, unattainable power consumption will clearly be proportional to the total pressure required to maintain the flow (curve IC in Fig. 9.17 which is again the impeller power at 85% constant fan efficiency).

As the pressure required to maintain flow falls the excess developed by the fan can be absorbed in a series damper (curve DP) but there will then be no energy saving. An axial fan with a falling power characteristic will gain somewhat from by-pass damper control (curve BP AX from Fig. 9.10) and a backward-curved centrifugal fan from vane control (curve VC BC from Fig. 9.14 at 50m^3/s). In neither case is more than about half the available energy saving secured.

Speed control is suitable for constant volume flow (though not for constant pressure). Curve SP assumes 83% full load efficiency for the variable speed motor and control circuitry combined, a figure which will in practice depend on the system adopted. At lower speeds and pressures the electrical losses will fall, but the fan will move to points of lower efficiency on the fan characteristic as the free flow point is approached (10% total pressure in Fig. 9.17). Curve SL allows for the additional losses involved in speed control with electrical or mechanical slip.

Once again the example of variable pitch axial fan control gives the best energy saving over a wide range of volume control. (Curve VP AX from Fig. 9.11 at 40m^3/s.) Again selection of optimum pitch angles will bring energy saving within 3% of the ideal value for a more limited control range - 40% in this case.

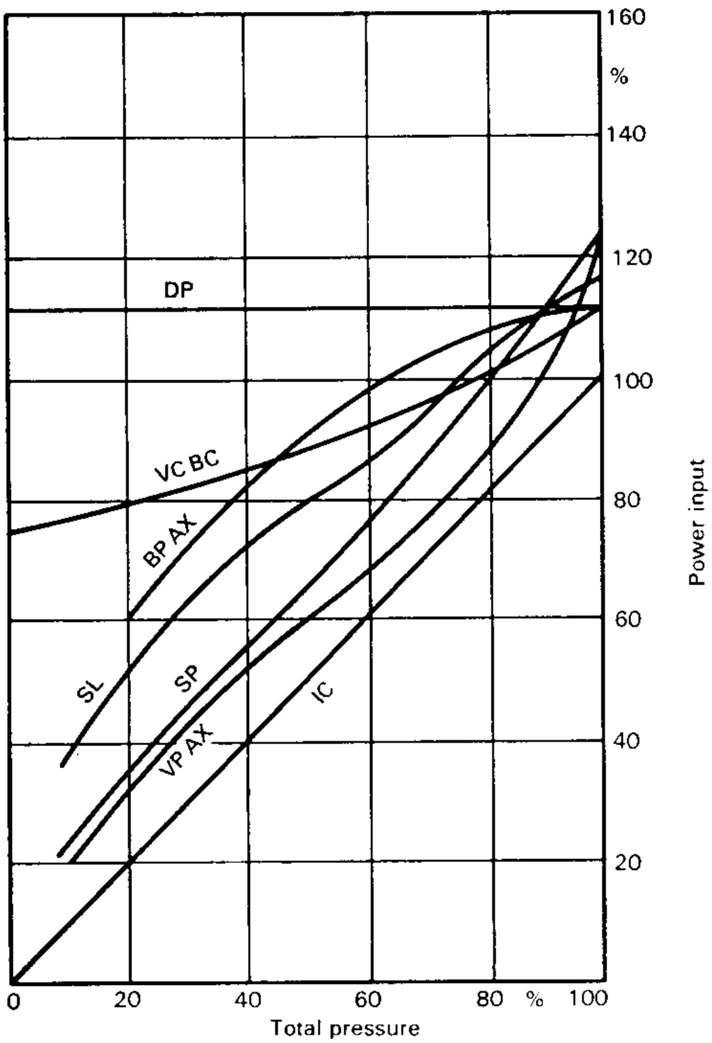


Fig. 9.17 Control at constant volume flow.

9.6.4 Energy savings

Figs. 9.15, 9.16 and 9.17 showed typical values for the (normally electrical) input power at all points in a control cycle. The significant factor is the total energy consumption in kilowatt-hours* over a representative period of time. To find this we must know the proportion of running time occupied at each fraction of full duty, and sum up the total kWh. This will now be done for the cases summarised in the last three sections.

'Gigajoules is the approoriate SI Unit. 1 GJ = 278 kWh.

For distribution in time the simplest assumption is that all volume flows in a constant resistance or pressure system and all pressures in a constant volume system are equally likely. With such a control regime the energy consumption over a representative time period will be simply proportional to the area under the appropriate curve in the figures cited, over the control range. This assumption has been made in Table *9.2.*

Table 9.2
Examples of energy consumption with duty control

Minimum volume flow or pressure %		80	60	40	0
Volume control at constant resistance					
Theoretical minimum average power %		74	54	40	25
Actual average power % for:					
VP AX	Variable pitch. Axial fan	76	58	46	33
SP	Variable motor speed	84	64	50	35
SL	Variable slip control	89	74	60	—
5:4	Two-speed, 8–10 pole	—	78	68	—
3:2	Two-speed, 4–6 pole	—	—	78	53
VC BC	Vane control. Backward-curved	86	72	65	55
BP AX	By-pass control. Axiai fan	89	79	—	—
DP BC	Damper. Backward-curved	94	89	85	70
DP FC	Damper. Forward-curved	104	89	80	65
Volume control at constant pressure					
Theoretical minimum average power %		90	80	70	50
Actual average power % for:					
VP AX	Variable pitch. Axial fan	92	83	74	56
VC BC	Vane control. Backward-curved	92	84	77	64
DP BC	Damper. Backward-curved	108	105	100	83
DP FC	Damper. Forward-curved	123	110	95	80
Pressure control at constant volume					
Theoretical minimum average power %		90	80	70	50
Actual average power % for:					
VP AX	Variable pitch. Axial fan	92	83	74	56
SP	Variable motor speed	98	90	80	60
SL	Variable slip control	104	96	90	70
VC BC	Vane control. Backward-curved	94	92	90	85
BP AX	By-pass control. Axial fan	102	100	95	—

As a standard of comparison the electrical energy which would be consumed over the same period if there were no reduction in fan duty will be taken as 100%. This will be calculated for a fan efficiency of 85% and a motor efficiency of 92%, both appropriate to the duty of $50m^3/s$ at 1000 Pa total pressure for which the examples have been selected.

The "theoretical minimum average powers" in Table 9.2 assume that 85% aerodynamic efficiency and 92% motor and drive efficiency could be maintained over the whole control range. The more realistic values of fan, motor and drive losses indicated by Figs. 9.11, 9.14, 9.15, etc. have been used to compute the "actual average powers" in the Table.

9.6.5 Example with constant system resistance

An air conditioning system is expected to operate for 1,250 hours per annum at a duty to be controlled over the following range, all volume flows being considered equally likely.

Maximum: $50m^3/s$ at 1000 Pa total pressure.
Minimum: $20m^3/s$ at 160 Pa total pressure.

This is a constant resistance system and Table 9.2 may be applied to compare the annual energy consumption of the fan with various control means. The reference power for the Table will be the product of maximum volume flow and pressure increased by the losses of an 85% efficient fan and 92% motor.

$$100\% \text{ power for Table 9.2} = \frac{50\ (m^3/s) \times 1000\ (Pa)}{0{\cdot}85 \times 0{\cdot}92} = 64000\ W$$

Annual energy consumption is 64 kW x 1250 hours = 80,000 kWh at 100% reference level. Table 9.2 for a minimum volume flow 40% of maximum gives:

Variable pitch axial fan:	46% × 80,000	=	37,000 kWh
Variable speed motor:	50% × 80,000	=	40,000 kWh
Vane controlled, backward-curved:	65% × 80,000	=	52,000 kWh
Two-speed 8–10 pole motor:	68% × 80,000	=	55,000 kWh
Damper controlled, forward-curved:	80% × 80,000	=	64,000 kWh

9.6.6 Example of constant volume flow control

An industrial process requires a flow of $50m^3/S$ to be maintained against total pressure having a most probable value of 850 Pa varying between extreme limits of 700 Pa and 1000 Pa. On the assumption of "normal statistical distribution" the hours spent in each 50 Pa band of pressure in a total of 1,000 hours will be as in the table below. Fig. 9.17 applies and the reference power for this figure ignores motor losses.

100% power for Figs. 9.15, 9.16 and 9.17

$$= \frac{50\ (m^3/s) \times 1000\ (Pa)}{0{\cdot}85} = 58000\ W$$

Multiplying this reference power by the percentage in Fig. 9.17 for each pressure band and by the number of hours spent in that band, and then adding the results will lead to the total energy consumption per 1,000 hours.

Pressure Pa	Time hours	Variable pitch axial % kW	kWh	Vane-controlled backward-curved % kW	kWh
950–1000	70	116	4800	110	4500
900– 950	170	108	10800	108	10800
850– 900	260	100	15300	105	16000
800– 850	260	93	14200	103	15700
750– 800	170	87	8700	100	10000
700– 750	70	80	3300	97	4000
	1000		57100		61000

CHAPTER 10

Noise,vibration and fatigue

The ideal fan would no doubt perform its function silently, smoothly and without attention for as long as it was needed. In the practical world these qualities are unattainable, and it is the joint responsibility of the fan designer and the engineer responsible for selection and installation to see that no nuisance is caused, and that life expectancy is satisfactory.

If the fan duty is substantial and particularly if the pressure is high, both noise and vibration are likely to need control measures to ensure that no annoyance is caused to people in the vicinity. That subject is thoroughly treated in our companion volume "Woods Practical Guide to Noise Control", and this chapter deals only with the generation mechanism and the practical assessment of noise, vibration and balance quality.

10.1 Rating System for Fan Noise

10.1.1 Sound power

The noise control engineer needs to know "how much" noise the fan will generate when installed. This data should be provided by the manufacturer, and there is now general agreement on the form it should take, expressed in such standards as BS 848: Part II in Great Britain, or AMCA Standard 300 in North America. Sound pressure and sound level are deprecated as units of fan rating because their value is entirely dependent on the location of the test microphone. Instead the rating should be the Sound Power Level (SWL) of the fan as defined, expressed in decibels (dBW) above a reference level of 10^{-12} watts. This provides a fixed starting point for noise control calculations dealing with the effects of the environment.

$$\text{SWL} = 10 \log_{10} \frac{(\text{Sound power, SW—watts})}{10^{-12}\ \text{watts}} \quad \text{dBW} \qquad (101)$$

The word "decibel" is here used in its basic sense-the logarithmic expression of the ratio of two powers. It must not be confused with the decibel (dB) of sound pressure or the dBA of sound level on the A weighting scale. See Woods Practical Guide to Noise Control.

The fundamental *sound power* is the rate at which sound energy leaves the fan and travels along a duct attached to the inlet or outlet. Strictly speaking there is one value for the inlet and another for the outlet, but with many fans the two are sufficiently close for a single common value to be quoted.

If the duct is removed the quantity radiated into the surrounding atmosphere is called the "open inlet (or outlet) sound power". At low frequencies this is less than the ducted sound power, some of the sound energy generated being reflected back into the fan casing. Alternatively, by analogy with an electric generator, we may say that the higher "acoustic impedance" presented by the open atmosphere to the fan outlet "terminal" reduces the output power.

10.1.2 Frequency spectrum

The fan sound power level is incomplete as a performance rating without additional information as to the frequency distribution of the sound energy. This is not only because of the differing sensitivity of the human ear to different frequencies. It is also because the sound transmission and absorption of air distribution and noise control systems are also very frequency dependent. Generally speaking the higher frequencies are more readily absorbed than the lower frequencies.

A weighting system such as the A scale of sound level is quite unsuitable for sound power rating. The spectrum shape at the point where the sound is heard, and therefore the appropriate weighting, will change with every change in sound absorption and listening point. An octave band frequency spectrum of sound power level gives suitable evidence of fan performance for noise control purposes. The standard centre frequencies of the bands are:

63 125 250 500 1000 2000 4000 8000 Hz

Of these the 63 and 8000 Hz bands are subject to greater uncertainty of measurement (particularly 63 Hz) and may generally be omitted without detriment to the achievement of noise control. The 125 and 250 Hz bands (for centrifugal fans) or the 250 and 500 Hz bands (for axial fans) are often the critical ones for noise control purposes, but it is unwise to ignore the remaining frequencies.

Fan noise tests are often made with analysing equipment which automatically plots the results in third octave bands. These may serve to locate and roughly assess the magnitude of prominent pitched tones in the noise. The 24 bands are too numerous for practical noise control work, however, and they should be reduced to octave form for quotation.

10.1.3 The sound power scale

A sense of the magnitude of sound power levels (SWL) will grow with familiarity, but there is an unfortunate initial tendency towards confusion with sound pressure levels (SPL). The two would be numerically the same in the somewhat imaginary circumstances of a very small sound source, radiating equally in all directions, and at a distance of about 30cm from the ear. At 3m distance the sound power level would be unchanged but the sound pressure level would be 20 d8 less (SWL = SPL_3 + 20).

Noise control, even ordinary transmission through ducts, will make the difference greater and the inexperienced user must not be intimidated by the high-sounding values of fan sound power levels. Table 10.1 gives an idea of the whole gamut of sound power, together with air duties at which the fans are likely to feed similar sound powers into a ducted system.

Table 10.1

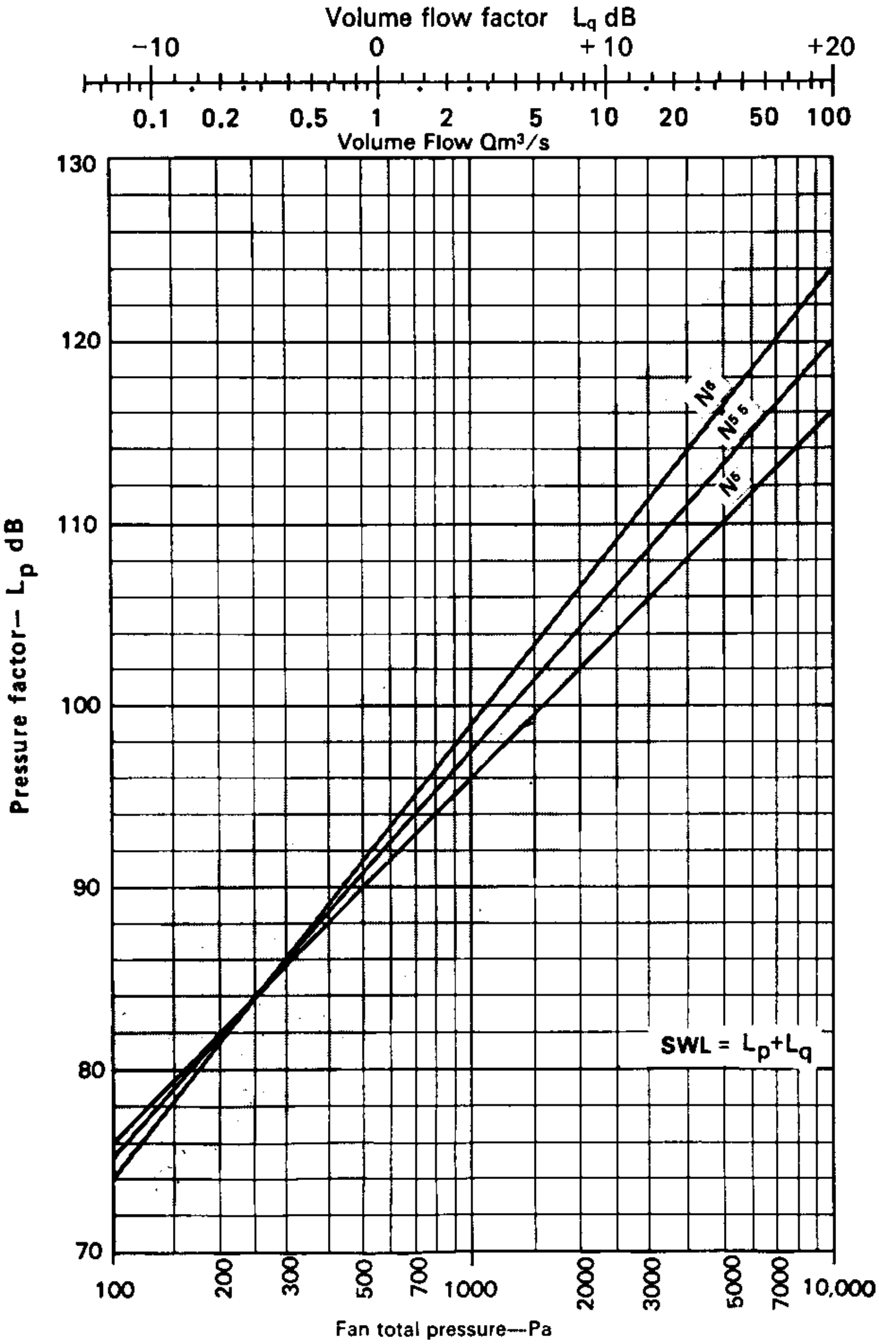
Order of magnitude of sound powers

Sound Power Level SWL	Sound Power SW	Source	Fan duty for equal aerodynamic sound power	
			Volume flow m^3/s	Fan total pressure Pa
195	30 MW	Saturn rocket at blast-off		
160	10 kW	Airliner with turbo-jet engines		
130	10 W	Orchestra at fortissimo peak	30	10,000
110	0·1 W	Blaring radio	10	2,500
90	1 mW	Voice—sustained shouting	3	250
70	10 μW	Voice—average conversation	0·3	100
50	0·1 μW	Electric clock	0·03	25

10.2 Typical Fan Sound Ratings

10.2.1 Sound power levels

Fig. 10.1 is a chart drawn up to illustrate the range of sound power levels normally produced by fans of standard type. The fan is supposed to be operating near the best efficiency point on its characteristic with



Range of levels for properly installed fans of differing types and designs.

Fig. 10.1 Sound power level and fan duty.

air of standard density, and SWL is the level of sound power which would enter and travel along a duct connected to either the fan inlet or the fan outlet. Lp is derived from the fan total pressure and Lq from the volume flow.

$$SWL = L_q + L_p \quad \text{dBW } 10^{-12} \text{ watts}$$

The air duty has been used rather than the size, speed, or mechanical power input sometimes employed in similar surveys; this makes for a fairer comparison between fans of differing type and mechanical efficiency. Any range of sizes and speeds all to one particular fan design will have L_p values close to a straight line sloping upwards at an angle between 45° and 50°.

A manufacturers noise ratings for a standard fan series will be based on a standardised test procedure (see Chapter 11) leading to such a design line or its equivalent. These ratings should always be used they are as much a function of the particular product as is the fan efficiency. Nevertheless, some general observations can be made on the values to be expected in typical cases.

The shaded area in Fig. 10.1 covers the range within which Lp may be expected to lie. Standard axial and forward-curved multi-vane centrifugal ranges will be located about the middle of the band. Backward-curved centrifugal and mixed flow designs should be found in the lower half of the area, while the lowest L_p values may be approached by high efficiency aerofoil bladed backward-curved centrifugal fans, but are unlikely to be improved upon by any design. The bottom limit of the band has been sloped steeply upwards at the higher pressure because designs for these pressures are likely to approach the radial tipped form, which is not so quiet.

In general, fans of standard types should only be found with ratings near the top of the band if they are poorly selected-in particular for pressures well below the best efficiency point. On the other hand such levels may well be reached or exceeded in practice if the installation has poor arrangements for controlling the inlet flow. Fig. *10.2* has been reproduced from *Woods Practical Guide to Noise Control* to illustrate this important point. Axial fans are illustrated, but similar considerations apply to all types. Any sharp change of direction or area, and any obstruction to the flow should be kept well away from the fan inlet.

10.2.2 The fan laws for sound power

The use of L_q and L_p with any straight line on Fig. 10.1 implies the following proportionality for sound power:

$$\text{SW is proportional to } Q.(P_{Ft})^a$$

Now the volume flow Q is proportional to the fan speed n rev/sec and the fan total pressure P_{Ft} to n^2. Therefore, for a given fan

$$\text{SW is proportional to } n^{(2a+1)}$$

SW is often taken as proportional to ns (i.e. a = 2.0) a value which applies reasonably closely to some centrifugal fans, but for accurate prediction an experimental value should be used. In the case of the Aerofoil axial range SW $\propto n^{5.5}$ (i.e. a = 2.25) and the index for a pure dipole source (see 10.3.1) would be 6 (a = 2.5).

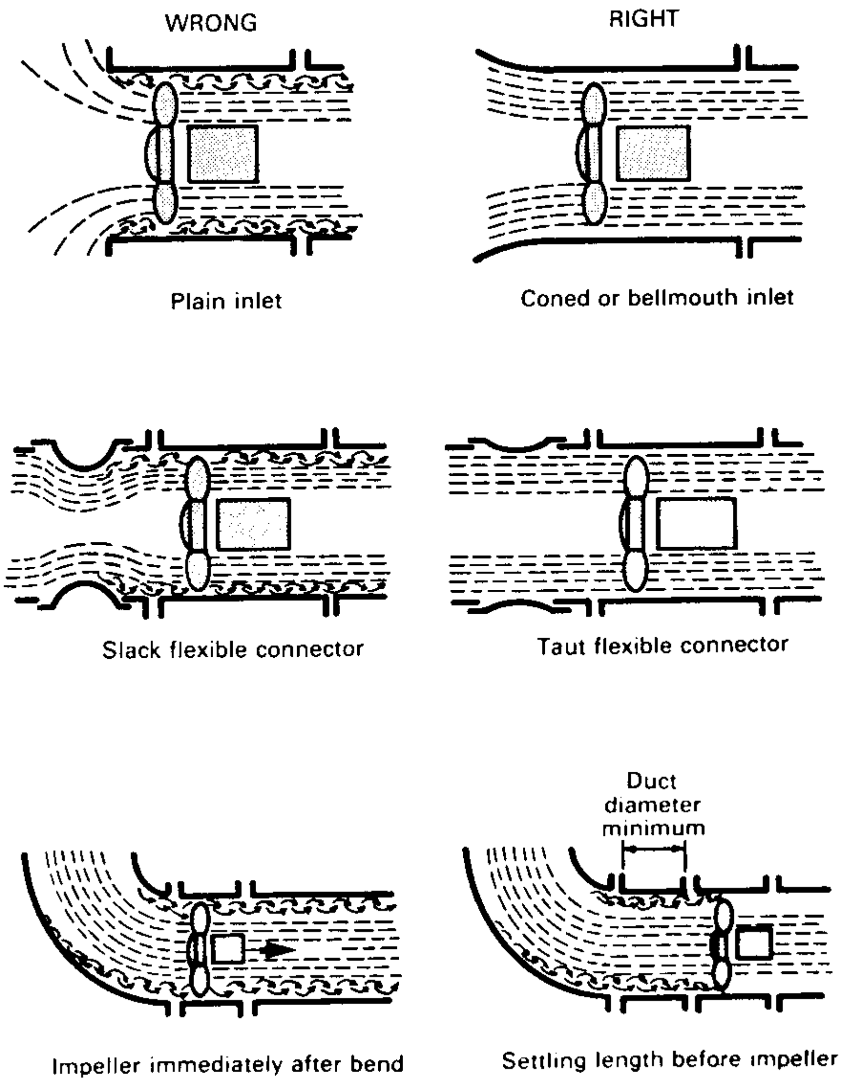


Fig. 10.2 Examples of noise generating installations.

Sound power (SW, watts) has the same units as air output power (Q x P_{Ft} m^3/s x Pa = watts) and it is instructive to express the first as a fraction of the second. This fraction is known as the *sound power ratio* (SWR) on a duty basis.

$$SWR = \frac{SW}{Q \cdot P_{Ft}} \qquad (102)$$

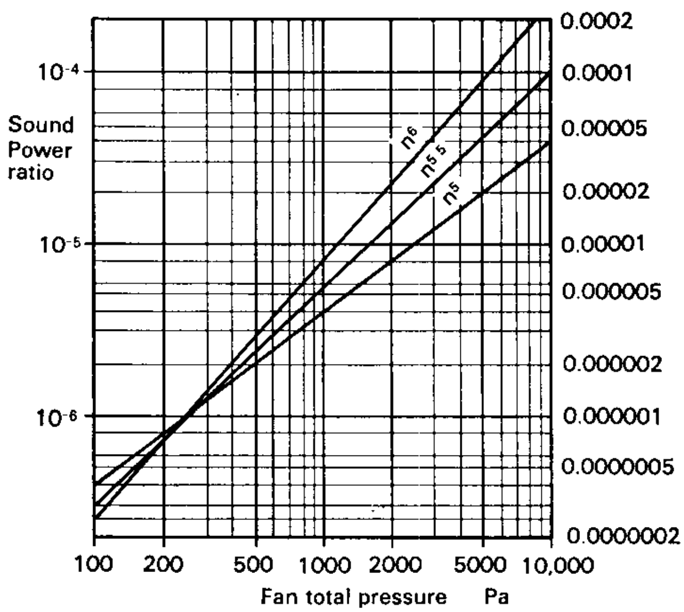
A representative value of SWR is 10^{-6} at 250 Pa fan total pressure. Taking Q as 1 m^3/s the air output at this pressure is 1 x 250 = 250 watts and the sound power 250 x 10^{-6} = 250 microwatts. The corresponding sound power level is:

$$SWL = 10 \log_{10} \frac{250 \times 10^{-6}\ (W)}{10^{-12}\ (W)} = 84\ dBW$$

Since SW is proportional to $Q.(P_{Ft})^a$ it follows that SWR is proportional to $P_{Ft}^{(a-1)}$. On Fig. 10.1 three straight lines have been drawn through 84 dBW, 250 Pa at slopes a = 2.0, 2.25 and 2.5 corresponding to SW $\propto$ n^5 $n^{5.5}$ and n^6. The equations of these lines are:

$$\begin{aligned} SWL &= 36 + 10 \log_{10} Q + 20 \log_{10} P_{ft} \\ SWL &= 30 + 10 \log_{10} Q + 22{\cdot}5 \log_{10} P_{ft} \qquad (103) \\ SWL &= 24 + 10 \log_{10} Q + 25 \log_{10} P_{ft} \end{aligned}$$

The corresponding relationships between the sound power ratio, SWR and the fan total pressure are plotted in Fig. 10.3.



For sources with SWR = 10^{-6} at 250 Pa.

Fig. 10.3 Variation of sound power ratio.

10.2.3 Change of sound power with duty point

So far we have considered only the fan sound power level at the best efficiency point on the fan characteristic. This is generally the lowest value, and the typical examples in Fig. 10.4 show the way in which the sound power increases away from this point. Variations of *5* to 10 dB are common and may be quite sharp if the characteristic contains a marked stall point. The shapes shown are representative of the fan types quoted.

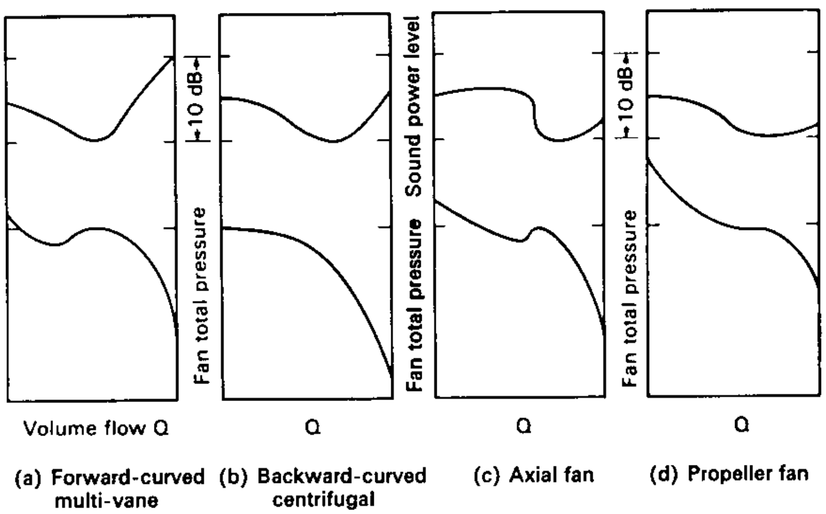


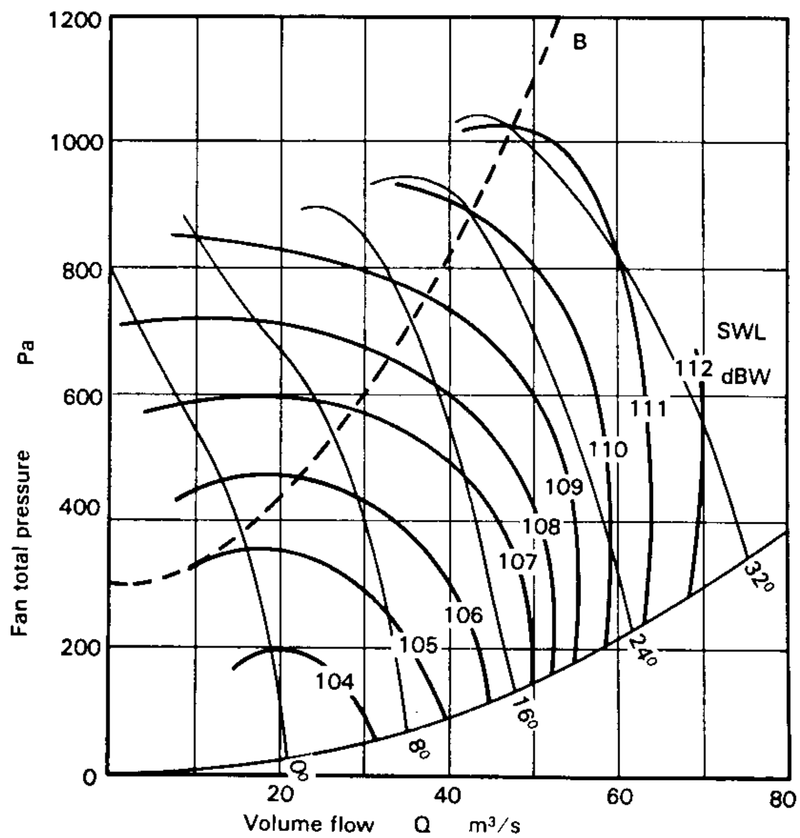
Fig. 10.4 Typical shapes of sound power level characteristics.

When the characteristic of the fan itself is capable of modification as by pitch control of an axial or vane control of a centrifugal fan other methods of representation are needed. Fig. *10.5* illustrates the use of contours of constant sound power level laid over the volume flow-fan total pressure chart. The fan is the same variable pitch axial model whose performance with constant efficiency contours was plotted in Fig. *9.11.* An example will show the use of this chart:

Example: The 1600mm *975* rev/min variable pitch axial fan of Figs. *9.11* and *10.5* operates in an air conditioning installation along system characteristic B. The following table can be drawn up, reading SWL values to 1 dBW from the nearest contour on Fig. *10.5.*

Pitch Angle	32°	24°	16°	8°	0°	
Volume Flow Q	47	42	35	26	16	m^3/s
Fan Total Pressure P_{Ft}	1020	880	710	530	390	Pa
Sound Power Level	111	110	109	107	105	dBW
L_q from Fig. 10.1	+ 17	+ 16	+ 15	+ 14	+ 12	dB
L_p = SWL − L_q	94	94	94	93	93	dB

The last line in this table is the experimental value of L_P corresponding to the observed value of SWL. Comparing these results with the L_p band on Fig. 10.1, the result for 16° lies on the mid line. The 24° and 32° results are better than average, and steep pitched axial fans working near peak pressure generally do give the most favourable noise levels for a given duty. The L_P values at 0° and 8° are not so good, partly because



1600mm 975 rev/min variable pitch axial fan.

Fig. 10.5 Contours of equal sound power level.

of the relatively low usage of available pressure, and partly because the blade is aerodynamically "off-design" for these angles, being designed for optimum performance around 24°.

It is interesting to note the strong correlation between the sound power and the aerodynamic loss of a fan. Reasons for this correlation will appear in Section 10.4, and to demonstrate it Fig. 10.6 has been drawn to the same scale as Fig. 10.5. The aerodynamic losses in watts can be expressed on the same decibel scale as the sound power and over the whole of the volume-pressure chart the difference is between 48 and 52 dB. At 50 dB the sound power would be one hundred-thousandth of the aerodynamic loss. For fans of the same design, but different speeds and diameters this ratio would vary with (tip speed)$^{2.5}$.

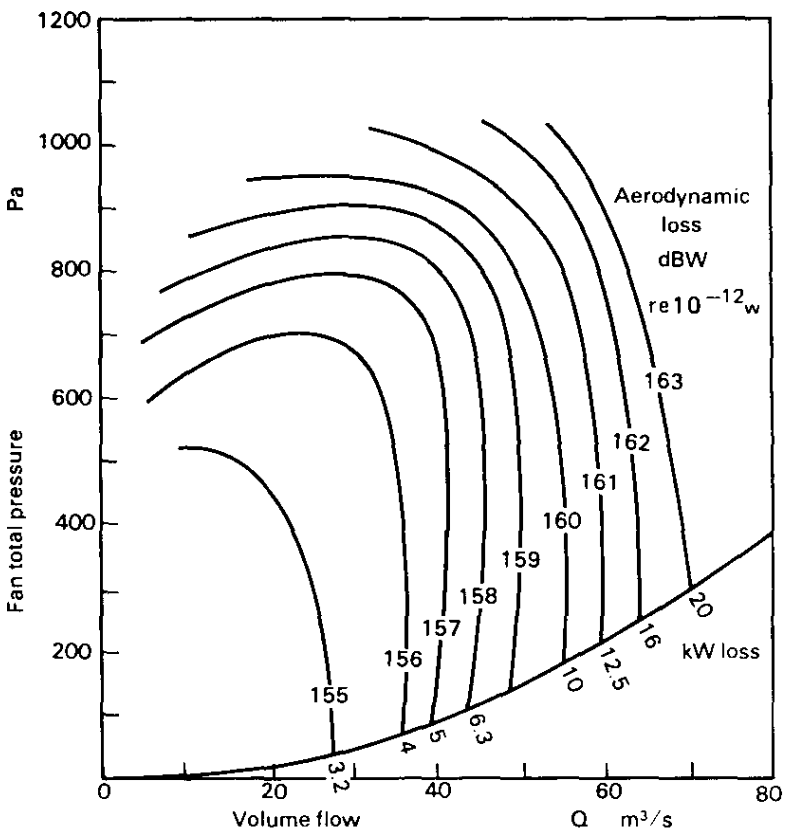


Fig. 10.6 Aerodynamic losses for comparison with Fig. 10.5.

10.2.4 Typical fan sound spectra

The total sound power level quoted for a fan should exclude any measurements in bands below 63 Hz. Their inclusion might rdise the total by several dBW, the measurement would be very unreliable, and the sensitivity of the ear to sound in this region is less than onethousandth of that at the middle frequencies. The remaining power should be split between the eight standard octave bands giving the *octave band sound power level* in each.

A convenient method adopted in fan catalogues is to quote from a number of sound spectra one which is applicable to the fan in question. Each *sound spectrum* will give for each octave band the dB deduction to be made from the total sound power level in dBW to obtain the band level. Some typical examples are quoted in Table 10.2, but it is important to use the manufacturers data since variations are substantial, e.g. from - *2* dB to - 12 d8 in the 63 Hz band, and - 15 dB to - 35 d8 in the 8000 Hz band.

Table 10.2
Typical sound power spectra

Octave Band:	63	125	250	500	1000	2000	4000	8000 Hz
Fan type				dB correction				
Backward-curved	− 4	− 6	− 8	− 10	− 15	− 20	− 26	− 32
Forward-curved	− 3	− 6	− 10	− 14	− 18	− 22	− 27	− 32
Axial fan	− 10	− 7	− 5	− 7	− 8	− 12	− 18	− 24

The generally higher frequencies of the axial fan sound output make it more audible when heard directly, but on the other hand it is more readily attenuated by noise control. The spectra quoted apply to fans for around 10m^3/s at *250* Pa. The sound powers will move towards lower frequencies at lower pressures or larger volumes, towards higher frequencies at higher pressures and smaller volumes.

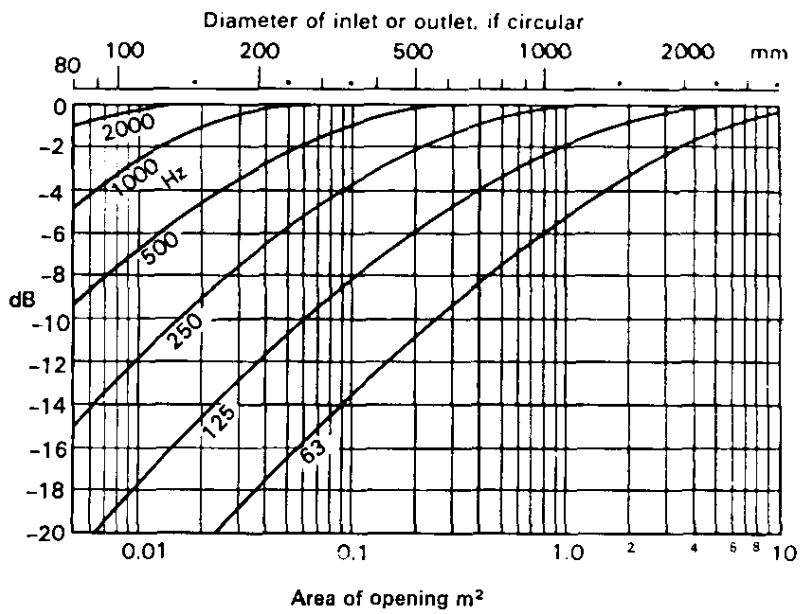


Fig. 10.7 dB reduction due to reflection at open inlet or outlet.

These values apply to the sound power flow along a duct. If there is no duct, less power will be radiated into surrounding space from the open outlet or inlet. A rough idea of the difference may be obtained by assuming plane wave reflection at the opening. The d8 deduction to be made to determine the octave band sound power level radiated depends on the area of the inlet or outlet opening and the band frequency as shown in Fig. 10.7.

The measured sound power of a fan will be predominantly broad band or random frequency noise. However, it is liable to include some discrete frequency sound particularly at a frequency equal to the rate at which the fan blades pass a fixed point. This becomes identifiable to the ear when it raises the appropriate third-octave band 5 dB above its neighbours (making only *2* dB contribution to the octave band) and can be objectionable if it is much greater.

Example: The 1600mm 975 rev/min variable pitch axial fan of the last example has an overall sound power level of 110 dBW at *24°* pitch angle. What is the sound power spectrum entering the plant room from an open fan inlet? The manufacturers spectrum correction, the reflection correction from Fig. 10.7 and the resultant octave band sound power levels are as follows:

Octave band	63	125	250	500	1000	2000	4000	8000	Hz
Overall SWL	110	110	110	110	110	110	110	110	dBW
Spectrum correction	− 12	− 6	− 5	− 7	− 10	− 15	− 21	− 27	dB
In-duct octave band SWL	98	104	105	103	100	95	89	83	dBW
Reflection correction	− 3	− 1	0	0	0	0	0	0	dB
Open inlet octave band SWL	95	103	105	103	100	95	89	83	dBW

10.3 The Mechanism of Aerodynamic Sound Generation

Aerodynamic noise is caused by the fluctuating forces generated by the flow of air or gas. Regular variations of force produce pitched tones of recognisable frequency. Randomly fluctuating forces produce broad band noise. A steady force in a fixed location does not generate sound, thus streamline or laminar flow is silent. In practical installations, however, the flow is always turbulent, in part at least, and noise is produced by one of the mechanisms described below.

10.3.1 Lift and drag forces

Examples are noise from: fan blades, turning vanes, grilles, dampers, bends, sudden contractions and enlargements in ducts and obstacles to flow.

In all these cases there are forces between the air and a solid body. The forces are unsteady owing to turbulence in the approaching air,
turbulent boundary layers, flow separation, and wake shedding. These are the principal sources of noise in fans and air distribution systems. They are best limited by good aerodynamic design and by keeping down

the flow velocity. The forces are random and produce broad band noise, being known acoustically as dipole sources. If V is the relative velocity of the air and the surface, the sound power will vary with V^6.

10.3.2 Interaction noise

This arises at a fan and is caused by guide vanes, bearing supports, or poorly designed approach airways.

The essential feature is that some regular pattern, repeating each revolution of the fan, is imposed on the otherwise random force fluctuations between the fan blades and the air.

Upstream guide vanes are a potent source, but downstream guide vanes should be satisfactory if more than 1½ blade chords from the impeller to smooth out the fluctuations in the wake of the fan blades. Inevitable upstream obstructions should be shaped to minimise the turbulent wake. Fan inlets are carefully shaped to provide smooth, uniform flow into the impeller. The installation engineer must see that these precautions are not ruined by defects such as those illustrated in Fig. 10.2 presenting the blade tips with a boundary layer which thickens locally at particular places in each revolution. This will cause interaction noise.

Interaction noise is an obtrusive pitched tone at blade passage frequency and its harmonics. It is a dipole source with sound power varying as V^6.

10.3.3 Jet noise

Examples are noise from : air blast; steam exhaust; aircraft jet engines.

This noise is generated within the air in the mixing region at the edge of the jet. As shown by Lighthill in 1952 it is caused by the interaction of turbulence with the high shear stress caused by the steep velocity gradient. It diminishes rapidly with air speed, and is negligible in duct systems except perhaps at nearly closed dampers. In quiet surroundings however, it limits the velocity of jet-throw distribution systems. The remedy is to introduce more air at lower velocity for the same momentum.

The source elements are known acoustically as "lateral quadrupoles". The noise is broad band and the sound power proportional to V^8.

10.3.4 Propeller noise

This is characteristic only of aircraft (or hovercraft) airscrews.

It is caused by the main propulsive force at each airscrew blade, which is steady in time, but is moving to and fro at very high speed, relative to an observer away from the axis, as the airscrew revolves. With only 3 or 4 blades, moving close to the speed of sound, powerful propeller noise is heard. The lower speeds and more numerous blades make this source entirely negligible in fan impellers.

The source is dipole, but the mechanism of propagation makes the sound power in the transonic velocity range vary as V^9 upwards.

10.3.5 Pulsating flow

Examples are positive displacement compressors, reciprocating internal combustion engines and sirens.

The noise source is a regular pulsation in the mass-flow-rate of the air or gas stream, entering or leaving the atmosphere. To limit the noise the pulsation must be reduced, as by a reservoir, a cavity resonator if the speed is fixed, or a diesel engine type silencer.

A pitched tone is produced at pulsation frequency with many harmonics. A powerful source, particularly when the flow change has a steep wave front. Acoustically a *monopole* source with sound power varying as V^4.

10.3.6 Induced vibration

Examples of noise from this source include the drumming of duct walls, the flapping of canvas or fabric, and the rattle of loose joints or fixtures.

The fluctuating forces between a solid body and the air flowing over it tend to produce vibration. The structure must be stiff enough, or damped sufficiently, to limit the vibration. Curved sheets have inherent stiffness, but flat sheets require stiffening ribs, particularly if they are of large area, and therefore, efficient generators of airborne sound.

With random frequency aerodynamic forces the structure will vibrate at its own resonant frequencies.

10.3.7 Other aerodynamic sources

Whistles, rumbles and rattles are sometimes heard in air systems which, when traced to their source, prove to have elements in common with musical instruments. Organ pipe resonance, cavity resonance, reed vibration, string vibration, surface drumming and edge tones may all be encountered. Experiments with stiffening or deadening the parts, closing leaks or diverting the local flow will usually provide the remedy.

10.4 Theory of the Fan Laws for Sound

Dimensional analysis indicates that the sound power ratios SWR = SW/W of a geometrically similar series of fans should be dependent only on Mach number, Ma = V/V_a, (the ratio of a typical fan velocity to the velocity of sound) and Reynolds number, Re = $DV\rho/\mu$ which is the ratio of the dynamic to viscous forces involved,

$$SWR = \text{constant} \times (Ma)^x \times (Re)^y \qquad (104)$$

Taking the rotational speed N, diameter D, and output power W of the impeller as the typical quantities:

$$W \propto N^3D^5 \qquad Ma \propto ND \qquad Re \propto ND^2$$

$$\text{Therefore } SW = \text{constant} \times N^{3+x+y} D^{5+x+2y}$$

Acoustical theory, considering only the Mach number effect, yields the following relations for the source types mentioned in 10.3

Monopole source $x = 1$ so that SW varies as V^4

Dipole source $x = 3$ so that SW varies as V^6

Quadrupole source $x = 5$ so that SW varies as V^8

Fan noise is principally dipole in origin and, for the simplest cases of boundary layer separation, there is reason to believe that the Reynolds number exponent y should be about – 0.4, leading to:

$$SW = \text{constant} \times N^{5 \cdot 6} D^{7 \cdot 2}$$

Axial fans seem to follow this relationship quite closely (e.g. $N^{5.5} D^{7.5}$), but other types may well introduce more complex generation mechanism. For example, centrifugal fans are usually considered to follow the law:

$$SW = \text{constant} \times N^5 D^7$$

Geometrical similarity as it affects Reynolds number is not well maintained in practice from size to size (thickness, clearances, even number of blades). It is wise to keep the index of D 2 greater than the experimental index of N and to treat small changes with size as a scale effect on the constant. The influence of atmospheric conditions is small -generally within 1 dB.

10.5 Vibration

A fan, like any other rotating machine, inevitably vibrates to some extent, and is a source of vibration in the structure on which it is mounted. Unless the fan is small and light, or the foundation heavy and solid it is advisable to mount the fan on vibration isolators, and this subject is treated in *Woods Practical Guide to Noise Control*. The fan manufacturer will normally advise on the choice of isolators, and sometimes provides them built-in between the driving motor and the mounting points.

This section deals with standards that have been established for assessing the severity of vibration, and with the limitation of the commonest source in fans - out of balance.

10.5.1 Measuring units for vibration

There are three things about a vibrating surface which can be measured, and each has its function.

Displacement is the natural factor to consider if one is concerned about reducing running clearances or causing rubs. It is also the most obvious to the eye.

Velocity is generally agreed to be the best general measure of vibration severity, particularly in relation to its disturbing effects on people and other equipment.

Acceleration is the factor which gives rise to forces and stresses within a machine, and between a machine and its foundation.

These three quantities have a simple and important relationship in the particular case of sine-wave vibration. This is a vibration in which the displacement is proportional to sin ωt where ωt is an angle which advances from 0° to 360° in one complete cycle of the vibration. ω is the angular velocity and equals $2\pi n$ where n is the cycles per second, or Hertz.

In such a vibration the velocity and acceleration are sine waves also, advanced in time by one quarter (90°) and one half (180°) of a cycle respectively.

The velocity is best represented by its rms (root-mean-square) value which is $1/\sqrt{2}$ times the peak value for a sine wave. All these quantities are illustrated and tabulated in terms of v with reference to the sine wave in Fig. 10.8. Vibration which has its origin principally in out-of-balance is sine-wave in character, but vibrations arising from such sources as rough bearings or air turbulence are not. The relations of Fig. 10.8 then break down, and acceleration in particular may be much greater than is indicated in relation to displacement.

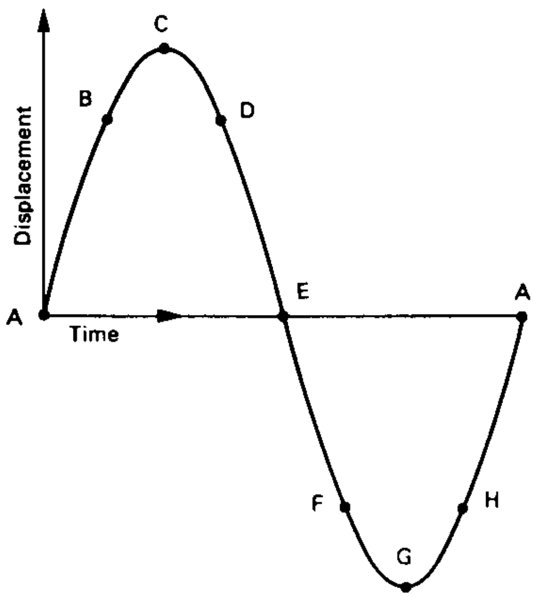


Fig. 10.8 Sine-wave vibration.

Point in cycle	Displacement	Velocity	Acceleration
A 0°, 360°	o	+ 1·4 v	o
B 45°	+ v/ω	+ v	− vω
C 90°	+ 1·4 v/ω	o	− 1·4 vω
D 135°	+ v/ω	− v	− vω
E 180°	o	− 1·4 v	o
F 225°	− v/ω	− v	+ vω
G 270°	− 1·4 v/ω	o	+ 1·4 vω
H 315°	− v/ω	+ v	+ vω

ω = 2π × frequency   v = rms velocity
v/ω = rms displacement   vω = rms acceleration

10.5.2 Vibration severity

For practical assessment the smallest significant step in vibration severity is considered to cover the ratio 1 to 1.6. A scale of quality judgement is proposed in BS 4675: 1971 with four grades A, B, C, D each covering two such steps, i.e. a ratio of 1 to 2.5. Table 10.3 summarises part of this scheme. The vibration to be considered is that on the external surface of the complete machine, with particular reference to feet or other fixing points and to bearing housings. So far no specific recommendations have been established for particular products, but it will be seen that the grading increases in severity towards Class I which is for small units likely to be fitted in more delicate situations. It is intended that, as experience is gained, the grades should become quoted in specifications.

Table 10.3

Quality judgement of vibration severity

Quality judgement BS 4675	Class I Small machines. Individual parts. Motors to 15 kW	Class II Medium size complete machines. Motors to 75 kW	Class III Large machines on heavy rigid foundations
	v—mm/s	v—mm/s	v—mm/s
A	up to 0·71	up to 1·12	up to 1·8
B	up to 1·8	up to 2·8	up to 4·5
C	up to 4·5	up to 7·1	up to 11·2
D	over 4·5	over 7·1	over 11·2

The grading applies only to frequencies from 10 Hz to 100 Hz. Vibration severity is given as v, rms.

10.5.3 Static out of balance

Two forms of balance defect can be recognised: *Static unbalance* and *couple unbalance*. Complications arise with flexible rotors, but fan impellers can always be taken as rigid from the balancing point of view.

Static unbalance simply implies that the rotor has a heavy side which can be counterbalanced by a single mass m added at a radius r exactly opposite. If the shaft were free from all constraint at its bearings it would vibrate keeping always parallel to itself - more precisely the shaft axis would rotate in a circular path (end view) with a radius e. The centre of this circle would be the centre of gravity (c.g.) of the whole rotating assembly which would thus become stationary - a state of natural running balance, like a top.

In practice, of course, the bearings must be supported, and if they are rigidly held they will experience the full centrifugal force due to the whole mass M at a radius e - since the shaft axis is now stationary instead of the c.g. The product Me is called the *unbalance* and the rotor will be statically balanced by counterweight m if:

$$mr = Me$$

suitable units are gm.mm for mr and kg.μm for Me.

The counterweight m will not be perfect, and there will be a *residual unbalance* Δm.r = Δe.M. With a completely soft bearing mounting, speed n rev/s = $2\Pi\omega$, and residual c.g. eccentricity Δe = Δm.r/M the vibration will have:

Peak displacement	= Δe	μm (micrometres)
Peak velocity	= $\Delta e.\omega$	μm/s
Peak acceleration	= $\Delta e.\omega^2$	$\mu m/s^2$

10.5.4 Balance quality grading

Fig. 10.10 shows the relationship of permissible eccentricity a with speed n for a number of balance quality grades, G1 , G2.5, etc. specified in ISO 1940. The international committee responsible for this specification has tentatively (1973) proposed quality classifications for particular products, extracts from which are given in Table 10.4. It will be seen that G 6.3 is the grade considered appropriate for fan impellers.

Note that the product eω defining the grade is a peak velocity. Since out-of-balance vibrations are generally sine wave it follows that the rms velocity = 0.71 eω. This is the vibration severity measure of Table 10.3 and it will be seen that quality grades A B C and D correspond to balance grades G1, G2.5, G6.3 and G16 + respectively for Class I small machines and components.

This equivalence of vibration and out-of-balance grades applies only to the motion of soft mounted rotor bearings. The vibration will be reduced in proportion to any added mass of bearing housing, and to the further constraints of semi-rigid or rigid mounting. The significant vibration severity with isolation mounts is that on the far side of the mount-vibration of the machine itself is then relatively harmless.

10.5.5 Couple (or dynamic) unbalance

A rotor may be in perfect static balance and yet produce vibration at its bearings by the rocking motion indicated in Fig. 10.9. Equal and opposite counterweights m at radius r in correction planes spaced a apart are said to correct a couple unbalance equal to m.r.a.

Any residual couple unbalance Δmra will act, unchanged, at the bearings, and if these are I apart the unbalance at *each* bearing will be Δmra/I. This is the same unbalance as would be produced by a static unbalance half-way between the bearings of:

$$2\Delta mr\frac{a}{l}\ \text{gm mm}$$

Clearly the bearings must be well spaced (I not too small) if a small couple unbalance is not to produce large effects. With a rotating mass M the effective value of Δe for balance quality grading is:

$$\Delta e\ (\mu m) = \frac{\Delta m\ (gm) \times 2a\ (mm) \times r\ (mm)}{M\ (kg) \times l\ (mm)}$$

This eccentricity need not be added to the residual static unbalance eccentricity Δe to give the overall balance quality grading of the whole machine, though each must be within limits. They may be additive for internal bearing loading, however, and for loads on the isolation mounts.

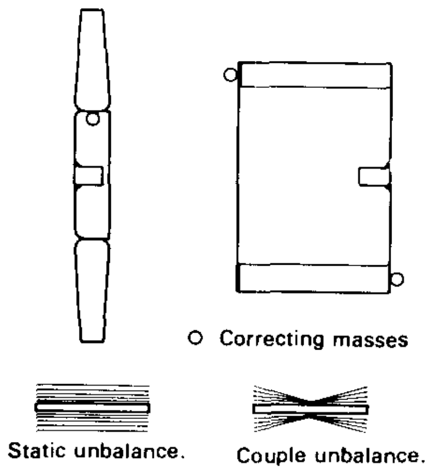


Fig. 10.9 The two kinds of rotor unbalance.

10.5.6 Balancing

Narrow impellers (Fig. 10.9) can be corrected by static balancing, particularly if produced by a dimensionally consistent method such as diecasting. Close tolerances on the shaft-hub fit are essential, and must not be spoiled by careless fitting. Dynamic unbalance will be produced by a small run-out such as might result from forcing the impeller on against a bruised shaft or shoulder.

Impellers with a breadth exceeding some 20% of the diameter are likely to require dynamic balancing. This involves driving the rotor in the bearings of a balancing machine which are soft mounted in a horizontal direction. The resulting bearing vibrations are picked-up, analysed and displayed to show the positions and values of counterweights to be attached in two selected correcting planes.

Built up impellers may have their component parts separately balanced subject to a final check on the complete assembly. For example, an axial impeller with a broad heavy hub might have this component dynamically balanced, followed by a static balance when a set of comparatively light and narrow blades had been attached. This

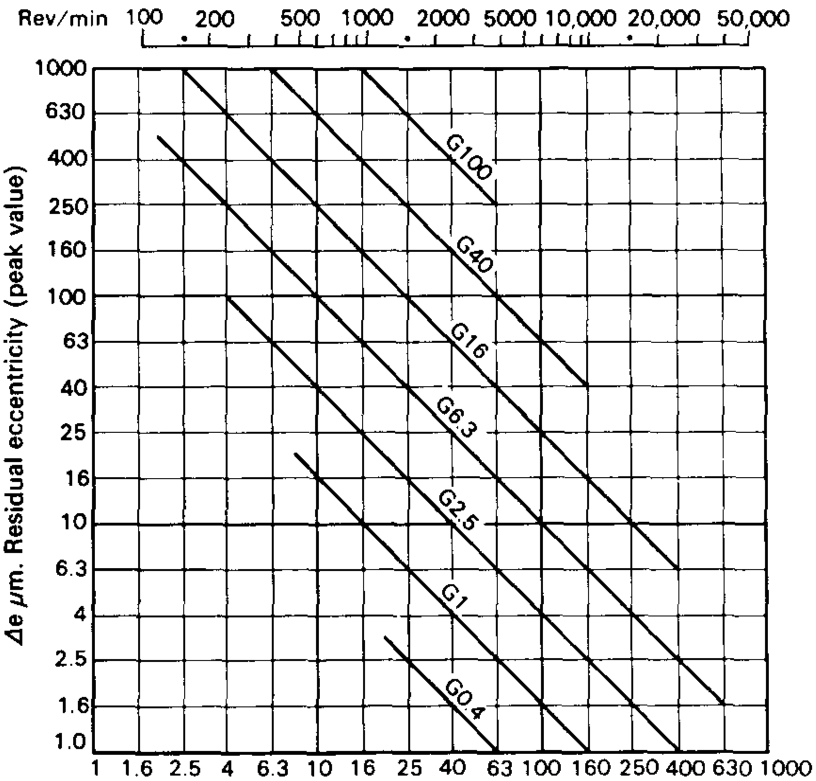


Fig. 10.10
Permissible eccentricity for ISO balance quality grades.

Table 10.4
Tentative classification of balance quality (ISO 1940)

Balance quality grade	Peak velocity $e\omega$ mm/s	rms velocity v mm/s	Typical rotating elements likely to be acceptable within grade
G 0·4	0·4	0·28	Gyroscope rotors.
G 1	1·0	0·71	Grinding machine drives. Gramophone drives.
G 2·5	2·5	1·8	Small electric armatures. Rotors.
G 6·3	6·3	4·5	Fan and pump impellers. General machinery.
G 16	16	11·2	Agricultural machinery.
G 40	40	28	Car wheels and crankshaft drives.
G 100	100	71	Complete internal combustion engines.

final stage could be more accurate than a dynamic operation since the sensitivity of the latter may be impaired by the masking effect of the fluctuating aerodynamic forces on the moving blades.

10.5.7 Dynamic balancing examples

The way in which the various unbalances may be combined and corrected is best shown by examples

Example 1. A 1000mm, 1800 maximum rev/min axial fan impeller weighing 25 kg is to be statically balanced by adding weight at 180mm radius. Within what limits should the weights be adjusted ?

The G6.3 balance quality grade is appropriate. At 30 rev/s, Fig. 10.4 gives $\Delta e = 32\mu m$ as maximum c.g. eccentricity. The corresponding maximum out-of-balance weight is:

$$\Delta m = \frac{\Delta e.M}{r} = \frac{32\ (\mu m) \times 25\ (kg)}{180\ (mm)} = 4{\cdot}5\ gm$$

Example 2. Fig. 10.11 (a) illustrates the operation of dynamically balancing a 250mm diameter, 3600 maximum rev/min (60 rev/s) multivane impeller weighing 2.5 kg. Two counterbalance weights are to be attached at 110mm radius in correcting planes 150mm apart. Initial static unbalance: 240gm.mm, 50mm from one plane. Initial couple unbalance: 120gm.mm x 150mm at right-angles to the static unbalance.

The 240gm.mm static counterbalance components will be divided between the correcting planes so as to give equal and opposite couples

$$160\ gm.mm \times 50mm = 80\ gm.mm \times 100mm$$

The couple counterbalance components of 120 gm mm in each correcting plane will directly oppose the couple unbalance moments.

Combining the counterbalance vectors in each plane gives **200 and** 144 gm mm respectively in the directions shown. These determine the following counterbalance weights:

$$\frac{200\ (gm.mm)}{110\ mm} = 1{\cdot}82\ gm \qquad \frac{144\ (gm.mm)}{110mm} = 1{\cdot}31\ gm$$

Example 3. Fig. 10.11 (b) illustrates the impeller of the last example overhung from a housing weighing 1.5 kg with bearings 80mm apart. Suppose the impeller to have been balanced just to the G6.3 limits, viz Δe = 16 Nm at 60 rev/s both statically and dynamically.

The static unbalance:

$$\Delta e.M = 2{\cdot}5\ (kg) \times 16\ (\mu m) = 40\ gm.mm$$

The couple unbalance:

$$\Delta mra = \frac{\Delta e\ Ma}{2} = \frac{2{\cdot}5\ (kg) \times 16\ (\mu m) \times 150\ (mm)}{2}$$
$$= 3000\ gm.mm^2$$

Note that these are in each case one-sixth of the initial unbalance.

The static unbalance will produce a reaction unbalance force of 40 gm.mm which will be located half-way between the bearings. These two opposite forces, 140mm apart, will produce a couple unbalance of 40 x 140 = 5600 gm, mm^2 to be resisted by the bearings.

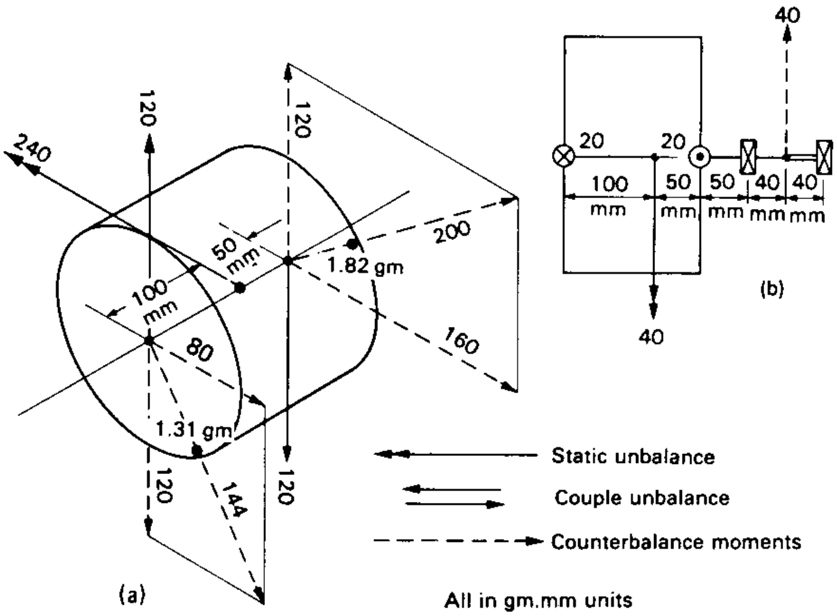


Fig. 10.11 Example of dynamic balance correction.

The couple unbalance of the impeller will be transferred unchanged, giving 3000 gm.mm^2 at right-angles to the *5600* gm.mm^2. Combining these the total couple at the bearings is:

$$\sqrt{5600^2 + 3000^2} = 6400 \text{ gm.mm}^2$$

On the total mass of *2.5 + 1.5 = 4*.0 kg this will produce with soft mounted bearings:

$$\Delta e = \frac{2\,\Delta mra}{MI} = \frac{2 \times 6400\ (\text{gm.mm}^2)}{4\ (\text{kg}) \times 80\ (\text{mm})} = 40\ \mu\text{m}$$

$$\text{or v (rms)} = 0{\cdot}71\ \Delta e\omega = 0{\cdot}71 \times 0{\cdot}040\ (\text{mm}) \times 2\pi \times 60\ (\text{rev/s})$$
$$= 10{\cdot}7\text{mm/s}$$

This is much too large a vibration for a Class I application as Table 10.3 will show. The solution, however, is not to attempt to improve the balance quality of the impeller but rather to stiffen the bearing housing against rocking motion. This may be done either by attaching the housing rigidly to a base of substantial mass (increasing M) or by fitting the housing with rigid arms of substantial span attached for example, to the fan casing (increasing the effective I).

10.6 Critical Speeds

10.6.1 Resonant vibration

There exists for any rotating shaft, rotor and bearing combination, a certain speed at which resonant vibration develops. The slightest unbalance will cause the shaft to bend, increasing the unbalance and bending without limit except for damping effects and loss of exact resonance. This is a dangerous condition and causes intense vibration so that the critical speed must never be held for any length of time. Apart from the first, and lowest, critical speed there are others corresponding to more complex modes of vibration.

While high-speed turbines are designed to pass through the first, and often other, critical speeds during run up, this is never the case for fans. Fan shafts are designed with sufficient stiffness to keep below the first critical speed at all times, although a resiliently mounted fan will pass through low speed resonance as a complete assembly. In this case there is no bending of the shaft.

10.6.2 Extended fan shafts with overhung impellers

It is sometimes desirable to mount an impeller at a substantial distance from the nearest bearing. The fan may be handling gases which are so hot that oil circulation or water cooling would be needed if the bearing housing were immersed in the gas. The impeller may be running inside an oven or a sealed and insulated stove with the running gear outside for servicing.

Fig. 10.12 (a) illustrates such a case. The impeller, of mass M, will not be perfectly balanced. Suppose its centre of gravity is distant a from the centre line of the shaft which is straight when unloaded. When bent through a deflection y, there will be a proportionate reaction force F = ky. This must equal the centrifugal force produced by M kg rotating in a circle of radius (y + e) with angular velocity $\omega = 2\pi n$. That is:

$$ky = M\omega^2 (y + e) \quad \text{N} \qquad (106)$$

$$\text{Therefore } y = \frac{M\omega^2 e}{k - M\omega^2} \quad \text{m}$$

The deflection thus becomes indefinitely large when $M\omega^2$ equals k, which determines the critical speed:

$$n_c = \frac{\omega_c}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{k}{M}}$$

While fans should never be allowed to reach the first critical speed, it is of theoretical interest to note that, if they do, y changes over to a negative value. This means that the impeller centre of gravity CG now rotates inside, rather than outside, the shaft centre as shown in Fig. 10.12 (c). As the speed rises the c.g. will move nearer and nearer the centre line through the bearings, as a smaller and smaller eccentricity suffices to supply the force required to bend the shaft. The deflection y ultimately approaches e.

At all speeds these shaft forces will be balanced by reactions at the bearings. The bearing housing must be rigidly mounted so as not to yield; if it does y will increase while k and the critical speed will fall. The alternative of soft mounting is not usually satisfactory because of excessive impeller off-centre when passing through low speed resonance.

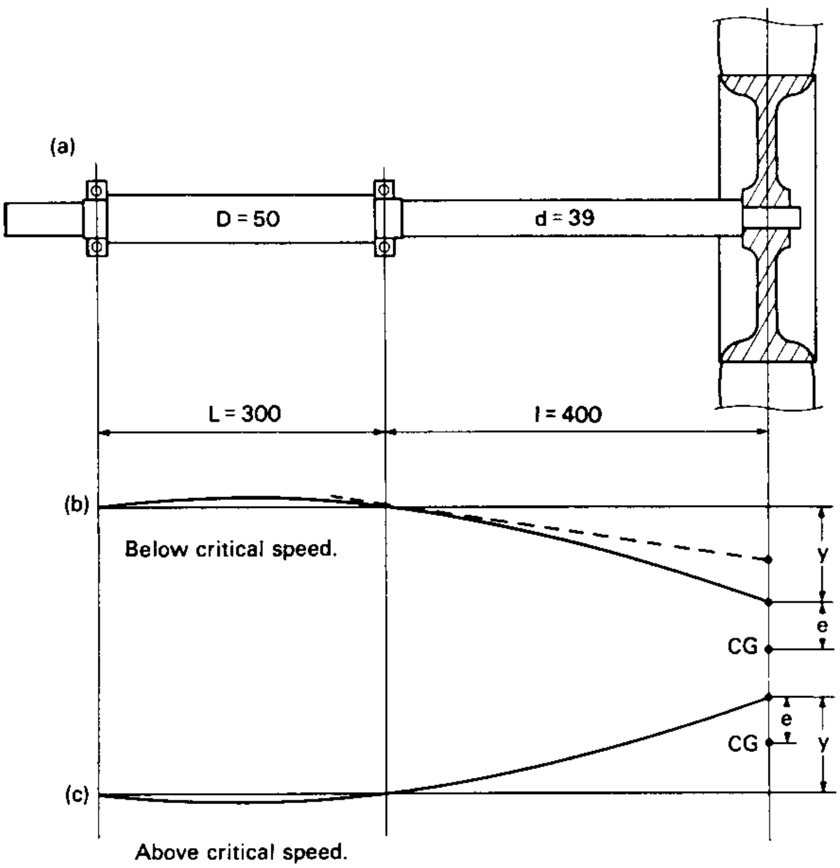


Fig. 10.12 Critical speed for an overhanging impeller.

10.8.3 Calculation of critical speed

The formulae for bending of shafts yield the following expression for the end deflection y m under an end load F newtons. Y_1 is due to the bending in length L and y_2 to that in I.

$$y = y_1 + y_2 = \frac{FLl^2}{3EI_D} + \frac{Fl^3}{3EI_d}$$

$$I_D = \pi D^4/64 \qquad I_d = \pi d^4/64$$

$$\text{Therefore } k = \frac{F}{y} = \frac{3\pi E}{64\left(\frac{Ll^2}{D^4} + \frac{l^3}{d^4}\right)}$$

Taking E = 2 x 10^{11} N/m^2 for steel, and converting dimensions L, I, D and d from m to mm for convenience, we find the critical speed n_c with an effective mass M kg at the end of the shaft to be:

$$n_c = \frac{1}{2\pi}\sqrt{\frac{k}{M}} = \frac{865}{\sqrt{M\left(\frac{Ll^2}{D^4} + \frac{l^3}{d^4}\right)}} \quad \text{rev/s} \qquad (107)$$

Example. Consider the extended shaft assembly shown with mm dimensions in Fig. 10.12 (a). Can a 630mm impeller weighing 10 kg be operated safely at the end of such a shaft with a speed of *24* rev/s *(1440* rev/min) ?

To allow for the distributed mass of the shaft extension I about *40%* of its mass of 3.8 kg should be added to the impeller mass making M = 11.5 kg.

$$\text{Therefore } n_c = \frac{865}{\sqrt{11{\cdot}5\left(\frac{300 \times 400^2}{50^2 \times 50^2} + \frac{400 \times 400^2}{39^2 \times 39^2}\right)}}$$

$$= 42{\cdot}9 \text{ rev/s} = 2570 \text{ rev/min}$$

The operating speed is thus 56% of the first critical speed, which is satisfactory - 60% to 65% is a reasonable maximum if the rigidity of the bearing housing can be relied on.

10.7 Structural Fatigue

10.7.1 Limits of alternating stress

In common with most machinery, fans operate with mechanical stresses far below the ultimate capacity of their materials. In part the reason is uncertainty of loading, though the working loads are more predictable than usual, the worst hazards being often the shocks and blows of transit and installation. The fan designer has a particular problem, however-the vibrations produced by fluctuating aerodynamic forces.

It is well known that a load many times applied and removed will cause a part to break when it could indefinitely withstand two or three times that load if steadily applied. The relationship between the maximum alternating stress S_e and the number of cycles (complete reversals) to failure is indicated in Fig. 10.13. These are typical values only, the actual fatigue performance depending on the alloy, and the heat treatment and other processes used in its manufacture, even for standard test specimens.

When a steady stress is applied in addition the alternating part must be reduced for the same life although the permissible peak stress rises. The straight line relationship of Fig. 10.14 is a fairly safe assumption for most structural materials. For any desired life applying to both S_a and S_e:

$$\frac{S_a}{S_e} = 1 - \frac{S_m}{S_u}$$

where

S_a = alternating component in the case of mixed stress
S_e = endurance limit with pure alternating stress
S_m = mean or steady component of mixed stress
S_u = ultimate limit with pure steady stress

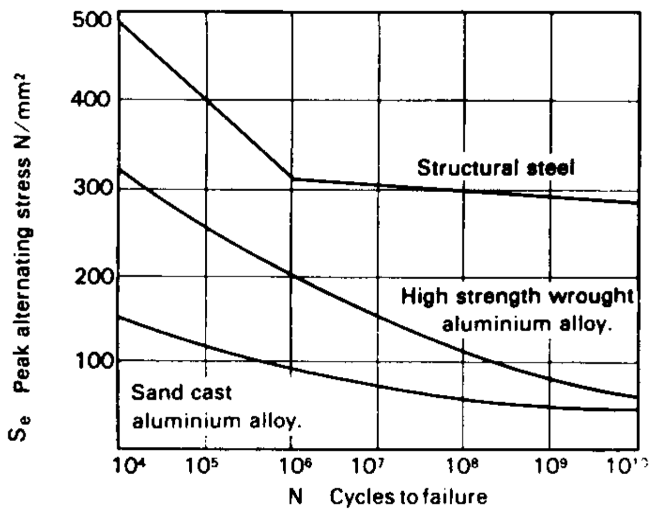


Fig. 10.13 Typical S-N diagrams.

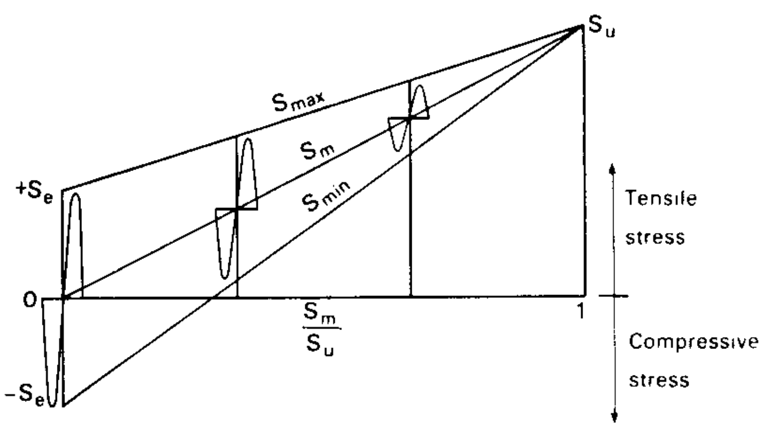


Fig. 10.14
Proportions of steady and alternating stress for equal life.

10.7.2 Alternating stress in axial fan blading

The fan designer must take into account the alternating stresses experienced by an impeller. As an example consider one blade of an axial fan. A steady mean lift force is applied to the blade corresponding to the pressure rise, but this has a randomly varying component in addition as discussed in *10.3.1.* Both forces produce a bending moment and corresponding stresses which reach a maximum near the blade root, and add to the centrifugal stresses. This is a mixed stress condition similar to that of Fig. 10.14.

These stresses can be examined experimentally. The first step is to coat the blade with a test layer of a special brittle lacquer. When the blade surface is stretched by tensile stress, cracks appear in the lacquer. These cracks run at right-angles to the direction of the stress, and are closer together the greater the value of the stress. The points on the surface where the tensile stress is a maximum can thus be identified, and strain gauges are attached at these points, one on each side of the blade. Completing an electrical bridge network and bringing out the four

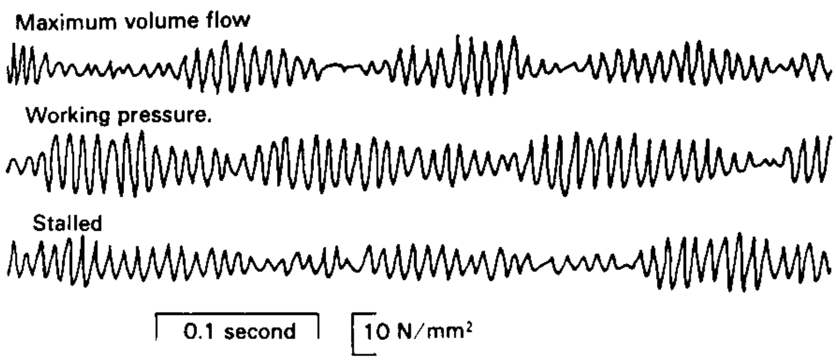


Fig. 10.15 Vibration patterns of an axial impeller blade.

corners of the bridge through small precious metal slip rings it is possible to display the alternating component of the bending stress.

Fig. 10.15 is a record of such a display. A consistent vibration frequency is immediately obvious, and this proves to be equal to the natural frequency of vibration of a stationary blade, if struck. On the other hand it is clear that the vibrations do not build up indefinitely as would a resonant vibration. The random nature of the air forces has a damping as well as an exciting action. Of course, resonance with any fixed pattern imposed on the random forces must be avoided. Such a pattern would be caused by the interaction force of *10.3.2,* and it is wise to design for a natural blade frequencyfive orsix times maximum rotation frequency to be well above any likely excitation.

As the examples show the maximum vibration amplitude occurs when the fan pressure is a maximum. The more irregular forces of separated flow in the stalled region are not so effective. This facilitates a development programme for determining the maximum permissible tip speed of a specific design. Maximum operating stress amplitudes can be related to static life tests on actual production wings.

CHAPTER 11

Fan testing and performance prediction

Air is a difficult subject for the tester. It cannot be seen and measured: nor can it be caught in a container and weighed. Furthermore, its motion is three-dimensional and turbulent. Thus a great many measurements of velocity and direction would be needed to integrate a total volume flow, unless the air were first collected, steadied, and organised in a suitable test airway.

11.1 Types of Test

Such *standardised airways* are used to provide a reliable test of fan performance, and they are discussed later in this chapter. Often, however, we shall want to measure the airflow in an actual, working installation. If the installation is a ducted one we shall look for the place, often quite remote from the fan, where the air velocity is most nearly steady and parallel, for our measurements. This is a site test for volume flow and will check whether the system resistance has been correctly estimated and the fan is giving its standard performance.

If the volume flow does not match requirements we shall want to find out why. A measurement of the pressure difference across the fan will help to decide whether the fan or the system is at fault. However, site conditions are unlikely to be such that we shall get a good measurement of fan total pressure. If fan performance is in dispute it can only be properly checked by a test in a standardised airway. If a very large fan is to be tested, standardised airways may be considered too costly, or the power of available motors may be insufficient. In such cases an acceptance test may be conducted on site, by agreement between the parties, but such an agreement presupposes an airway layout which is not too different from one of the standardised arrangements as regards pressure measurement.

A *model test* may well give more reliable performance data for a large fan than a site test with an unfavourable airway layout. The model is made geometrically similar to the large fan, with an impeller diameter usually between 500mm and 1000mm, and is tested with air in a standardised airway, preferably at about the same tip speed (adjusted if a different gas is to be used on site). The efficiency of the large fan is likely to be slightly greater than that measured on the model, but such a *scale effect* improvement may only be claimed by agreement between the parties, based on evidence for the fan type in question. For the corresponding reason a model test is not allowed for the acceptance of a fan smaller than the model.

The fan laws are rules for calculating the performance of a fan at a speed or gas density different from that at which it was tested, and also for converting the results of a model test to full scale. They hold with precision for relatively small changes of speed or density, but, if the pressure rise is large, the effects of change in compression ratio must be considered; see 11.4.3.

Methods of test for fan sound power are dealt with in our companion volume - *Woods Practical Guide to Noise Control*, Chapter 3. Reference should also be made to Sections 10.2.4 and 10.4 in this book, where the fan laws for noise are discussed.

11.2 Methods of Measurement

11.2.1 Pressure

The only two quantities that can always be measured at a point in an air stream are the temperature and the pressure. The pressure will be measured separately, in two parts:

Firstly, absolute or barometric reference pressure of the still atmosphere close to the test airways.

Secondly, the gauge pressure, or pressure difference between the reference atmosphere and the still air within a small hole exposed at the required point in the air stream.

If the air is flowing parallel to the surface in which the hole is made, the gauge pressure will be the static pressure of the air stream. If, however, the hole is the open end of a tube facing the airflow, the air stream will be brought to rest locally, and the gauge pressure measured will be the total pressure of the air stream; see Section 6.1.3.

The instruments used to measure these pressures are nearly always of the liquid column type, and may be roughly classified according to the pressure to be measured, as follows:

For less than 100 Pa: a micromanometer;
For 100 to 1000 Pa: an inclined-tube manometer;
For 1000 to *15,000* Pa: a vertical, or U-tube manometer;
For *15,000* to 110,000 Pa: a mercury manometer or barometer.

The liquid level in a micromanometer is determined with the aid of a micrometric, magnifying, eyepiece. In the Betz type the scale is etched on a glass stem carried on a float in the suction column which rises centrally above a large liquid reservoir. The small movement of the reservoir surface is allowed for in the scale. This instrument can be read to *0-2* Pa, and can maintain ± 0.25% accuracy up to several thousand Pa, so that it may also be used as a reference standard for calibrating the inclined tube instruments.

Inclined tube manometers are the general-purpose instruments in most common use. The multiplying factor is the sine of the inclination to the horizontal and may be up to twenty to one while, in some models, the tube may be reset vertically for direct reading. The scale may be drawn to allow for the angular setting, the movement of the reservoir surface, and the density of the fluid used and it may be calibrated in mm of water or Pa or kPa. The former is, of course, the gravitational unit of pressure actually observed and, if the vertical height of the liquid column is h,

$$p\ (\mathrm{Pa}) = \rho\ (\mathrm{kg/m^3})\ g\ (\mathrm{m^2/s})\ h\ (\mathrm{m})$$

Strictly speaking g should be the local acceleration of gravity and p the actual density of the fluid, less the density of the air or gas over the reservoir. However, except in precision research, g may be taken as *9.8* and ρ as the average room temperature density of the liquid alone. For a water column, therefore, with ρ close to 1000 kg/m^3

$$1\,\mathrm{mm\ of\ water} = 9{\cdot}8\ \mathrm{Pascals;\ and}$$
$$1\,\mathrm{mm\ of\ mercury} = 133\ \mathrm{Pascals}$$

Inclined tube manometers must be carefully levelled and, for acceptance tests on fans, calibrated *in situ.* The gauge fluid must consistently wet the glass to preserve a meniscus of constant shape-water needs a wetting agent. Pressures fluctuate continuously in turbulent flow, and may cause the meniscus to dance up and down in unreadable fashion. Damping must then be applied by restricting either the liquid connection from tube to reservoir, or the air tubing, until the average level can be read to, say, ± 1 %. The damping must be symmetrical to air or liquid flow in both directions, otherwise a false level will be obtained, high in the direction of easy flow.

11.2.2 Velocity by Pitot-static tube

The standard instrument for measuring the air velocity at a point within a duct is the Pitot-static tube. A variety of designs has been developed, all with a strong family resemblance, and with individual advantages which tend to appear at the research level only. Those illustrated in

Fig. 11.1 may be used without correction in ordinary work provided the following conditions are met:

(a) The tube must be aligned with the direction of airflow within 5° (0.5% error) or 10° (1% error).

(b) The tube diameter must not exceed one twenty-fifth times the duct diameter or one twentieth times the smaller side of a rectangular duct. It must also not exceed 15mm.

(c) The velocity should be between 3m/s and 70m/s for standard air.

(d) The pressure difference $\Delta\rho = p_t - p_s$ is measured, together with the air density ρ, and the velocity calculated from:

$$v\ (\text{m/s}) = \sqrt{\frac{2\,\Delta p\ (\text{Pa})}{\rho\ (\text{kg/m}^3)}}$$

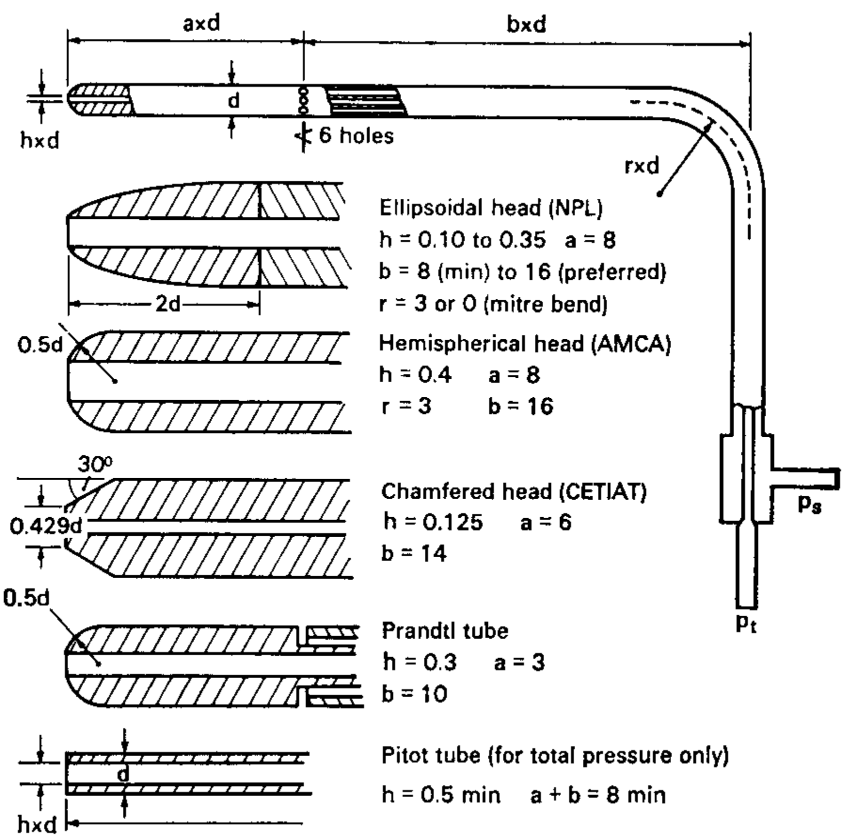
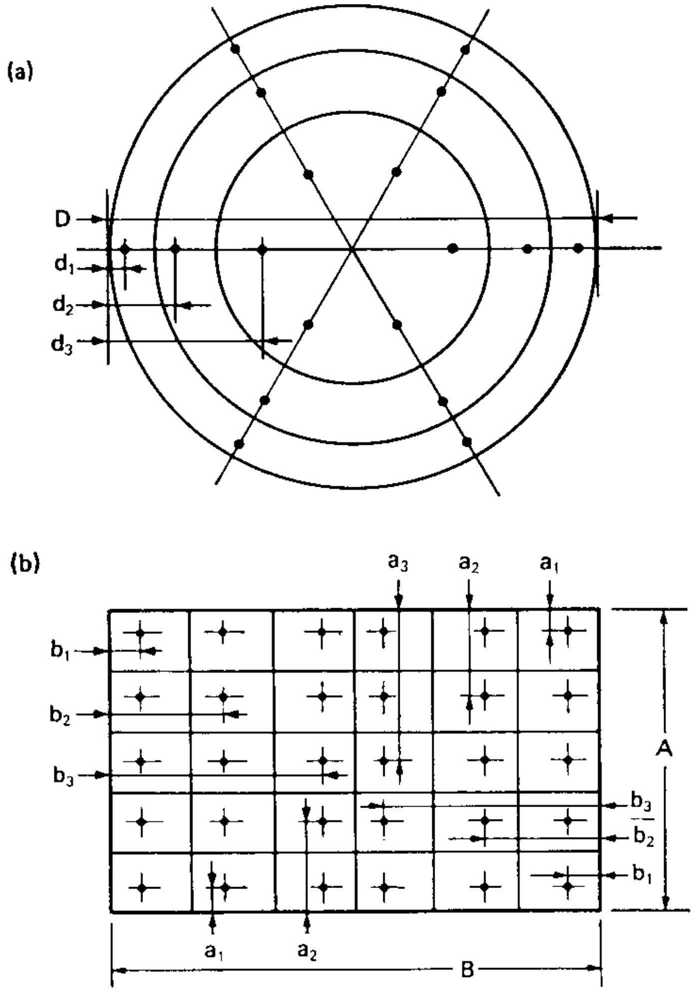


Fig. 11.1

Pitot-static and other tubes for velocity measurement.



(a) Circular ducts.

3 Zone, 18 point, traverse

$\frac{d_1}{D}$	$\frac{d_2}{D}$	$\frac{d_3}{D}$
0·032	0·135	0·321

4 Zone, 24 point, traverse

$\frac{d_1}{D}$	$\frac{d_2}{D}$	$\frac{d_3}{D}$	$\frac{d_4}{D}$
0·021	0·117	0·184	0·345

(b) Rectangular ducts.

5 Zones per side, 25 points.

$\frac{a_1}{A}$	$\frac{a_2}{A}$	$\frac{a_3}{A}$
0·074	0·288	0·500

6 × 5 Zones, 30 points

$\frac{b_1}{B}$	$\frac{b_2}{B}$	$\frac{b_3}{B}$
0·061	0·235	0·437

7 × 5 Zones, 35 points

$\frac{b_1}{B}$	$\frac{b_2}{B}$	$\frac{b_3}{B}$	$\frac{b_4}{B}$
0·053	0·203	0·366	0·500

Fig. 11.2 Location of velocity measurements in ducts.

Condition (a) is usually met by aligning the tube parallel with the duct axis and checking that there is nothing tending to produce swirl in the air stream upstream of the tube. If there is, straightening vanes must be inserted to ensure reasonably parallel flow. Contractions, bends and other disturbing features should preferably be at least 10 duct diameters upstream of the measurement.

Condition (b) usually rules out velocity measurement by Pitot-static tube in ducts smaller than about 200mm diameter. Measurements of rather less accuracy can still be made by means of a small "facing tube" or Pitot tube within the maximum diameter limits to determine $p_t.p_s$ is measured at holes drilled in the duct wall level with the open end of the tube. It is vital that all static and total pressure openings should be clear, square and free from burrs and deformities of all kinds.

11.2.3 Determination of volume flow

The velocity will vary from point to point over the cross-section of a duct, usually, but by no means always, peaking near the middle. We must, therefore, measure the velocity at a number of points, so distributed as to give an unbiased average. A suitable method for circular ducts is to divide the area into three or four concentric parts of equal area, and to find the velocity in each by averaging six velocities taken at 60° intervals round a circle - 18 or 24 points in all. The radii at which measurements are taken have been selected by the "log-linear rule", being off-centre by an amount which takes account of the most probable velocity distribution across the element area. Fig. 11.2a shows the location of these measuring points, and all the calculated velocities can be averaged together. Do *not* average the velocity pressures, which would overweight the higher velocities.

A rectangular duct should be divided into at least 25 equal rectangular areas (5 divisions each side). For long thin rectangles better accuracy is secured by dividing the long side into 6 (30 points) or 7 (35 points) parts. The points for velocity measurement are determined in this case by the "log-Tchebycheff rule" in the locations shown in Fig. 11.2b. Again a simple average of all velocities, multiplied by the total duct area, gives the volume flow rate.

Other distributions of measuring points appear in various national and international flow-measurement and fan-test codes. Differences will not be great, though the accuracy is likely to be slightly better when a large number of points are taken.

11.2.4 Volume flow by anemometer traverse

The vane anemometer is a convenient instrument for volume flow measurement in the "walk-in" airways of mines and tunnels. The velocity may be low for the Pitot-static tube and, while the anemometer is not so precise an instrument, it is quite good for averaging a velocity which is fluctuating both in magnitude and direction. It will measure velocities as low as 0.5m/s.

The usual form of instrument has a windmill-type rotor, driven round at a speed which is proportional to the air velocity except at low speeds where friction forces are significant. The number of revolutions made is indicated on a dial or may be sensed electronically and compared with a time base to give rotational speed directly. A calibration curve is necessary, and should be frequently checked.

The anemometer axis should not be closer than 1 ½ rotor diameters to the airway wall, which means that it is most suitable for rectangular airways. It should be held successively for not less than one minute in each of the measuring positions of Fig. 11.2. The operator should be at least 1.5m on the downstream side to avoid disturbing the flow past the anemometer, which he will usually hold and control remotely by rods. It is better to time a suitable number of revolutions with a stop-watch rather than repeatedly to start and stop the rotor.

An alternative method is to start both rotor and stop-watch together at the first measurement point, and allow each to run continuously to the last point, the anemometer being held for at least 20 seconds at each point before moving to the next. This process is repeated until three velocity measurements are obtained which do not differ by more than 2%, the average of these three being multiplied by the airway area to obtain the volume flow.

11.2.5 Location of site test measurements

The test plane for flow measurement must be substantially free from swirl, a condition which can be checked by observing the direction taken by small "flags" of thread or wool moved over the section on the end of a stick. A place should also be sought where the flow is as steady and symmetrical as possible. Both these conditions are best met, as a general rule, on the inlet side of the fan, and in a place where the duct is straight and uniform for a length equal to several duct diameters. The flow test plane should not be closer than 1 ½ duct diameters upstream of the fan inlet, or 4 diameters from the fan outlet if it has to be on that side.

Measurements of the pressure difference across the fan cannot, of course, be made too far away from the inlet and outlet, and may therefore have to be in less favourable flow conditions. A Pitot-static tube should be traversed over the measuring points indicated in Fig. 11.2 and the readings of static pressure taken and averaged. The less uniform these measurements are across the section, the less reliable the average will be. If in particular the static pressure at the duct centre is markedly less than that at the walls it shows that the airflow is swirling; individual static pressure measurements are then likely to be in error, and their average does not represent the true effective pressure.

Suppose the average static pressure at the test plane upstream of the fan is p_{s1}, and that on the downstream side p_{s2}. The volume flow rate q may have been measured at one of these test planes, or it may have been measured elsewhere. In either case we can calculate the average velocities v_1 and v_2 at the test planes from their areas together with the volume flows q,, q, correcting if necessary for density changes. Then the conventional total pressure rise between the test planes is:

$$p_{t2} - p_{t1} = p_{s2} - p_{s1} + \tfrac{1}{2}\rho_2 v_2^2 - \tfrac{1}{2}\rho_1 v_1^2$$

This quantity should equal the calculated total pressure drop for a volume flow q round the system of airways outside the test planes. Any differences must be accounted for either by errors of estimation or by errors of test, or by system faults such as leaks or badly-matched flanges. P_{t2} - P_{t1} plus the calculated pressure drop in any duct sections between the test planes and the fan, should equal the rated fan total pressure P_{Ft}. Any differences may be due to a faulty fan or fan installation, or to site test errors, or to departures from the standardised airway layout and provisions for pressure measurement.

11.3 Fan Tests with Standardised Airways

11.3.1 Installation types

A recent survey of the national fan performance test codes of eleven industrialised countries showed 65 distinct test methods in use. No doubt the majority of tests carried out in accordance with these codes yield comparable results, but serious discrepancies can arise. A committee of the International Organisation for Standardisation (ISO) has been charged with the preparation of a uniform system and, although agreement is as yet by no means complete, considerable progress has been made.

The performance of a fan is affected by the connections made to its inlet and outlet. From the infinite variety of possible connections, two have been selected as representative of the range of performance:

In one the fan inlet or outlet is open directly to the unobstructed free atmosphere.

In the other the fan inlet or outlet is connected to a long straight duct of the same area as the inlet or outlet.

These two alternative connections may be combined in four possible installation types as follows:

Type A: Free inlet, free outlet;
Type B: Free inlet, ducted outlet;
Type C: Ducted inlet, free outlet; and
Type D: Ducted inlet, ducted outlet.

Each of these installation types has, in principle, a different performance characteristic. In practice the Type B and Type D characteristics will coincide closely provided the fan is supplied, in its free inlet form, with a properly-shaped entry cone or bell-mouth. With the same proviso Type A performance will coincide with Type C. The essential difference remaining is that between free outlet and ducted outlet performance, which is significant for fans of all kinds though it diminishes as the ratio of fan velocity pressure to fan total pressure falls.

11.3.2 Examples of standardised airways

Fig. 11.3 gives, for each type of installation, one example out of many standardised airways capable of realising the principles which have international acceptance.

The space marked "fan" represents a cased fan under test, which may be of any type-centrifugal, axial or mixed flow-having an inlet and outlet, one or both of which may be designed for duct connection. Airflow is in the direction of the arrow.

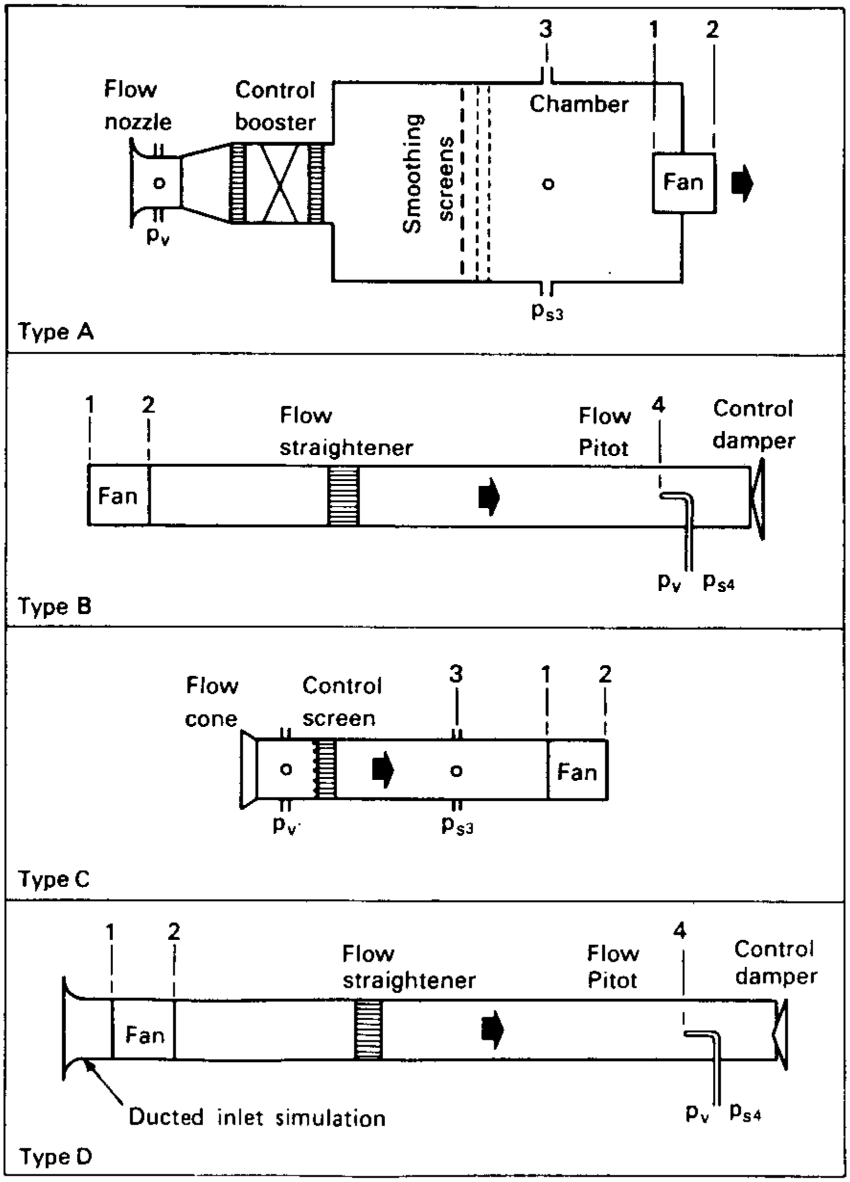


Fig. 11.3 Examples of standardised airways.

No definitive standards have as yet been agreed, but in the meantime many national standard methods, including those listed below from the British and American codes, will provide a fair basis of comparison The differences between the individual tests relate mainly to methods o1 flow measurement and flow control. Some are best for a permanent general-purpose installation, others for an airway purpose-built for a large fan, others for small fans, and so on.

Methods broadly conforming to international principles are:

Type A
BS 848 : Part 1 : 1963 : Test methods 3, 6, 7 and 8.
AMCA : Std. 210 : 1974—Figs. 13, 14 and 15.

Type B
BS 848 : Part 1 : 1963 : Test method 3 (centrifugal fans only).
AMCA : Std. 210 : 1974 : Figs. 7, 8, 9 and 10.

Type C
BS 848 : Part 1 : 1963 : Test methods 1 and 2.
AMCA : Std. 210 : 1974 : Fig. 16.

Type D
BS 848 : Part 1 : 1963 : Test method 5 (centrifugal fans only).
AMCA : Std. 210 : 1974 : Fig. 7, 8, 9 and 10 with inlet bell.

11.3.3 Fan pressure conventions

Satisfactory measurements of pressure cannot be made at the fan inlet or outlet. It is necessary to establish pressure test stations some distance upstream and downstream where the flow can be normalised. The quantity measured at these stations is the static pressure, to which is added a conventional velocity pressure to obtain the effective total pressure.

When the fan draws air directly from the test enclosure as Type B, Fig. *11.3,* or effectively directly as Type D, and also when air is discharged directly to the test enclosure as Types A and C, the test station is any point in the surrounding atmosphere where the air is still and unaffected by the flow from the fan. At this test station:

$$p_s = 0 \qquad p_t = 0 \qquad v = 0$$

When the test station is in a duct or chamber the static pressure may be measured by side tappings (Types A and C Fig. *11.3)* or by averaging the static pressures of a Pitot-static traverse (Types B and D of Fig. *11.3).* In these cases P_s is as measured relative to the ambient atmosphere.

The fan total pressure is equai to the effective total pressure at the outlet side test station minus the effective total pressure at the inlet side test station plus an allowance for pressure loss between the fan and the test stations. In a ducted outlet Type B or D test the added allowance represents the friction of the ducts between the fan and the test stations. In an open outlet Type A or C test, however, the allowance also includes the conventional fan outlet velocity pressure. This is because the whole of the outlet kinetic energy is in fact dissipated within the outlet jet as the air slows to near zero velocity at the test station where the test enclosure ambient pressure is $p_{s4} = p_{t4} = 0$

The fan static pressure is by definition the fan total pressure minus the conventional fan outlet velocity pressure. In the case of open outlet Type A or C tests this deduction exactly cancels out the added velocity pressure mentioned in the last paragraph, so that the fan static pressure equals zero minus the effective total pressure on the inlet side, e.g. for a Type A test:

$$\begin{aligned} FSP &= FTP - \tfrac{1}{2}\rho_2 v_2^2 = (p_{t4} - p_{t3} + \tfrac{1}{2}\rho_2 v_2^2) - \tfrac{1}{2}\rho_2 v_2^2 \\ &= 0 - p_{t3} \end{aligned}$$

Pressure formulæ for Fig. 11.3

Diagram	Outlet side	Inlet side	Fan total pressure (FTP) Fan static pressure (FSP)
Type A	p_s 0 $v = 0$ $p_t = 0$	$p_s = p_{s3}$ $v_3 = q_3/A_3$ $p_t = p_{s3} + \tfrac{1}{2}\rho_3 v_3^2$	$FSP = -p_{s3} - \tfrac{1}{2}\rho_3 v_3^2$ $FTP = FSP + \tfrac{1}{2}\rho_2 v_2^2$ $P_f = 0$
Type B	$p_s = p_{s4}$ $v_4 = q_4/A_4$ $p_t = p_{s4} + \tfrac{1}{2}\rho_4 v_4^2$	$p_s = 0$ $v = 0$ $p_t = 0$	$FTP = p_{s4} + \tfrac{1}{2}\rho_4 v_4^2 + P_{f24}$ $FSP = FTP - \tfrac{1}{2}\rho_2 v_2^2$
Type C	$p_s = 0$ $v = 0$ $p_t = 0$	$p_s = p_{s3}$ $v_3 = q_3/A_3$ $p_t = p_{s3} + \tfrac{1}{2}\rho_3 v_3^2$	$FSP = -p_{s3} - \tfrac{1}{2}\rho_3 v_3^2 + P_{f31}$ $FTP = FSP + \tfrac{1}{2}\rho_2 v_2^2$
Type D	$p_s = p_{s4}$ $v_4 = q_4/A_4$ $p_t = p_{s4} + \tfrac{1}{2}\rho_4 v_4^2$	$p_s = 0$ $v = 0$ $p_t = 0$	$FTP = ps_4 + P_{f24} + \tfrac{1}{2}\rho_4 v_4^2$ $FSP = FTP - \tfrac{1}{2}\rho_2 v_2^2$

The above are algebraic quantities, and sign must be carefully considered. All velocity pressures are positive, but static pressures are positive if above the surrounding atmosphere, negative if below. On the inlet side static pressures are generally negative. Thus if in a Type C test

$$p_{s3} = -500 \text{ Pa} \qquad \tfrac{1}{2}\rho_3 v_3^2 = 200 \text{ Pa} \qquad P_{f31} = 6 \text{ Pa}$$
$$FSP = -(-500) - 200 + 6 = 306 \text{ Pa}$$

The friction losses P_f in the test airways are taken as those that would arise if the flow were axial, with the distribution finally reached in a long straight duct. This is the "handbook value" which a user would employ in estimating the resistance of his system. The actual friction losses are likely to be greater owing to the disturbed flow close to the fan, but this excess is, by convention, ignored - that is it is not claimed as part of the fan output although the fan does have to produce it. The loss due to an average velocity v in a duct diameter D and length L is:

$$p_f = f\,.\,(L/D)\,.\,\tfrac{1}{2}\rho v^2$$

In most current codes f is taken as 0.02, and L is increased to include an allowance for the flow straightener. The code convention must, of course, be followed in reporting fan performance under any particular code. In practice f depends to a substantial extent on Reynold's number, a proposed relationship appropriate to the normal range of surface roughness found in test airways being:

$$f = 0{\cdot}02\left(\frac{10^5}{Re_D}\right)^{0\cdot17}$$

Where $Re_D = \dfrac{Dv\rho}{\mu} = \tfrac{2}{3}\,Dv\,.\,10^5$ approximately for atmospheric air.

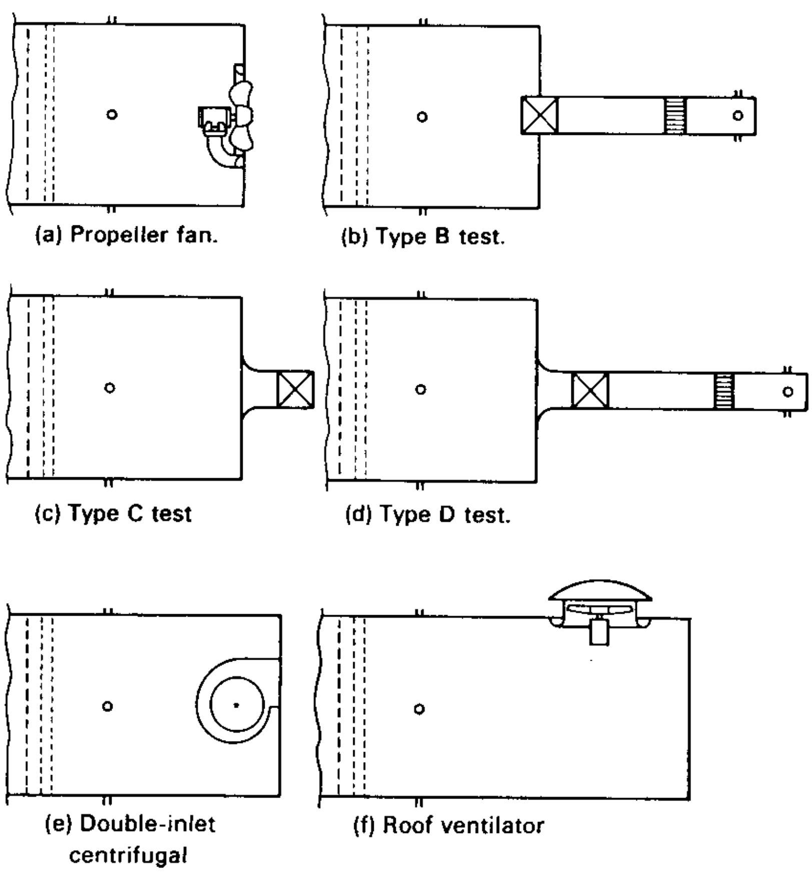


Fig. 11.4 Standardised airway tests with an inlet side chamber.

11.3.4 Type A-free inlet, free outlet-standardised airways

In the Type A layout of Fig. 11.3 the inlet side test chamber has only one essential requirement: that the flow pattern at the fan inlet should not differ significantly from what it would be in a completely unobstructed free atmosphere. An area ratio chamber: fan inlet of at least 6: 1 secures this, together with smoothing screens to provide reasonably uniform, axial, approach flow. An auxiliary booster fan is shown to overcome the resistance of the test airways, enabling the fan under test to be operated through to zero or even negative fan static pressures. This might be a variable pitch axial booster fan, variable to negative pitch angles to provide control resistance, and mounted between anti-swirl units to protect the test fan and the flow measurement nozzle. Nozzles are discussed in 11.3.7.

An inlet side test chamber is a versatile device, which may be permanently constructed and instrumented to produce, rapidly and accurately, performance data for fans in a very wide range of sizes and types. The test fan may be fitted with short, auxiliary airways to provide Type B, C or D characteristics as well as Type A, as indicated in Fig. 11.4, b, c and d. Actual inlet side installation conditions may be reproduced as for the propeller type wall fan of Fig. 11.4a. An awkwardly-shaped unit such as a double inlet, free outlet, centrifugal fan (Fig. 11.4e) or a crossflow fan may have casing and inlets wholly immersed in the chamber. Fig. 11.4f shows an adaptation for fan-powered roof ventilators which need to be tested with a vertical axis because they have gravity-operated shutters.

11.3.5 Type B-free inlet, ducted outlet-standardised airways

The Type B diagram in Fig. 11.3 shows a Pitot-static tube traverse for volume flow measurement, the differential pressure, P_{s4} between the static holes and atmosphere being also measured and averaged. Flow control is by damper, although an auxiliary exhaust fan might also be used for control. Pitot-static traverses are time-consuming, and are mainly used for one-off acceptance tests on large fans to save the cost of building more complicated airway systems.

Three other methods of test for volume flow, each requiring only a single pressure-difference measurement to integrate the whole flow rate, are shown in Fig. 11.5. By fitting a diffuser of 15 ° included angle to the venturi nozzle, expanding to rather more than the original duct diameter, almost all the fan characteristic can be covered, though an auxiliary exhauster could be used if preferred. As the diagram demonstrates, the immersed orifice plate is best suited to small fan tests because of the length involved; an auxiliary exhaust fan would be needed to cover the whole characteristic. The orifice at system outlet, used both for flow measurement and flow control (by changing the orifice plate) is the flow measurement principle of the French standard NFX 10 (AFNOR, formerly CETIAT).

It will be noticed that all the airway layouts in Fig. 11.5 show a common duct design from the fan outlet to plane AA, some five duct diameters downstream, containing a standardised flow straightener in a standard location. International standardisation of this "common part" would ensure that different standardised airways would lead to the same values of fan total pressure for Type B and Type D installation tests.

The differing values of fan total pressure that arise without such standardisation stem from the character of the airflow leaving the fan. Even when free from swirl the velocity distribution at outlet can, as illustrated in Fig. 11.6, be far from uniform. This results in an excess of kinetic energy over the conventional allowance of ½ $\rho_2 v_2^2$, caused by the

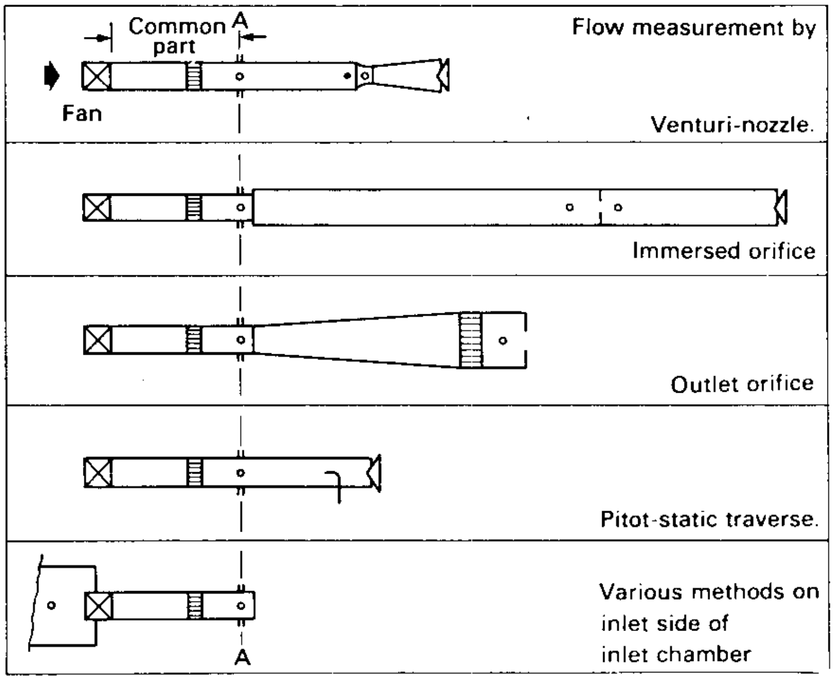


Fig. 11.5
Principle of the common part applied to Type B test airways.

proportionality of kinetic energy to the local value of ρv^3 (mass flow x velocity pressure) so that the excess where v is high exceeds the deficit where v is low.

Now the non-uniformity of the *axial* velocity components diminishes as the flow proceeds down the duct and the excess energy reaches a minimum of a few per cent. of ½ ρv^2 within a length equal to two to three duct diameters. Part of the original excess is lost as indicated by the broken line in Fig. 11.6 but part is converted into additional static pressure, as indicated by the full lines, the conventional velocity pressure remaining constant. This addition to the fan static pressure and the conventional fan total pressure is available for overcoming external resistance, and in order to credit it to the fan, as it should be for Type B and Type D installations, the test station for outlet side pressure measurement should be more than three duct diameters from the outlet.

On the other hand the *swirl* energy at fan outlet is never recovered in a straight uniform duct, and only decays over long distances-more than 100 diameters. In the presence of swirl simple measurements of effective pressure or volume flow are impossible, and it must therefore

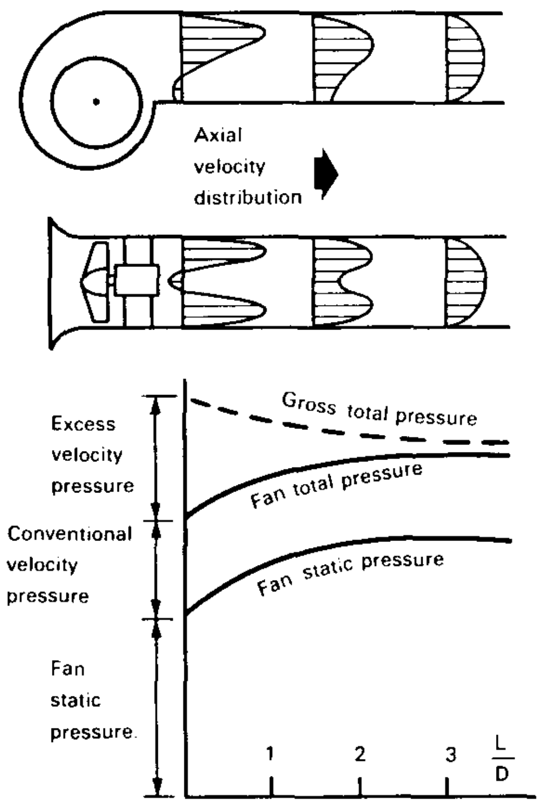


Fig. 11.6 Velocity diffusion downstream of a fan.

be removed when tests are to be taken in a duct on the outlet side for Type B or D performance. An effective flow straightener will do this, and if it removed just the swirl energy and no more the minimum energy convention would be satisfied. Unfortunately, the energy actually removed is very dependent on the combination of swirl pattern and straightener design, hence the need to standardise the latter.

In practice a fan with pronounced outlet swirl ought not to be selected for use with a *long* straight outlet side duct, because the friction loss in the latter will be substantially increased. Guide vanes should be fitted which will remove and recover (instead of removing and destroying) the swirl energy. The flow straightener will then merely ensure that the test conditions are satisfactory in the downstream duct; the relatively small outlet swirl components from centrifugal, guide-vane axial or contra-rotating fans will be removed without significant disturbance to the performance.

11.3.6 Type C-ducted inlet, free outlet-standardised airways

These difficulties of definition and test do not arise with a Type C installation. Satisfactory conditions for measuring volume flow and (negative) effective total pressure are easily achieved in an inlet side duct. Subtracting this total pressure from the static pressure at fan outlet, which is simply taken as zero, gives the fan static pressure. The fan velocity pressure $\frac{1}{2}\rho_2 v_2^2$ is added to give the fan total pressure for a meaningful statement of fan efficiency. This $\frac{1}{2}\rho_2 v_2^2$ will be taken away again as the outlet loss part of the system resistance (see Chapter 6) leaving the fan static pressure as the effective pressure for overcoming the upstream part of the system resistance when the fan has free outlet.

A Type C test installation can be extended to provide Type D performance by fitting a "common part" outlet duct as illustrated in item 2 Fig. 11.7, here shown with an open outlet in the position occupied by the static tappings of Fig. 11.5, giving virtually the same result. This is valid for a fan with limited outlet swirl, but unreal differences can arise between Type C and Type D performances as Fig. 11.7 illustrates for an extreme case - an axial fan without guide vanes having a high proportion of fan velocity pressure. When swirling air is discharged into the free atmosphere it forms an expanding vortex which does convert part of the swirl energy into useful pressure, so that curve 1 is the true Type C performance characteristic. When swirl is removed in a straightener, curve 2, this conversion does not take place, leading to increasing loss of nominal Type D fan pressure as the resistance rises and the fan outlet swirl correspondingly increases.

As explained in the preceding section airway *2* does not give a practical characteristic for a long duct application when there is swirl, since it includes a non-existent flow straightener. Only a test with the actual long duct could determine the true performance, and it is in any case bad practice to use a fan with pronounced swirl in such a case. With a short outlet duct, however, the swirl energy conversion will still take place at the duct outlet, and will more than compensate for the increased friction in the duct. The UK and USA national test codes have for many years included methods with an outlet duct two or three diameters long to represent ducted outlet. For fans without outlet swirl this practice will provide a true Type D characteristic. When the fan has pronounced swirl the characteristic, of which curve 3 is an example, will include both axial excess energy and swirl energy recovery. This will be a fair characteristic for *short* ducts, say up to five diameters long, but cannot be termed a Type D performance.

Even better recovery of swirl energy is obtainable with an outlet diffuser to control the vortex expansion. Steeper diffuser angles are possible without flow separation, but it is essential to determine the characteristic for the fan and diffuser in combination as one unit. The fan velocity pressure is now the lower value at diffuser outlet, and the broken line 4 shows that there is a loss of *total* pressure in the combination. The fan static pressure is then a better standard of comparison, giving the pressure available to overcome the upstream system resistance, without need to specify or allow for the area selected for the fan or diffuser outlet.

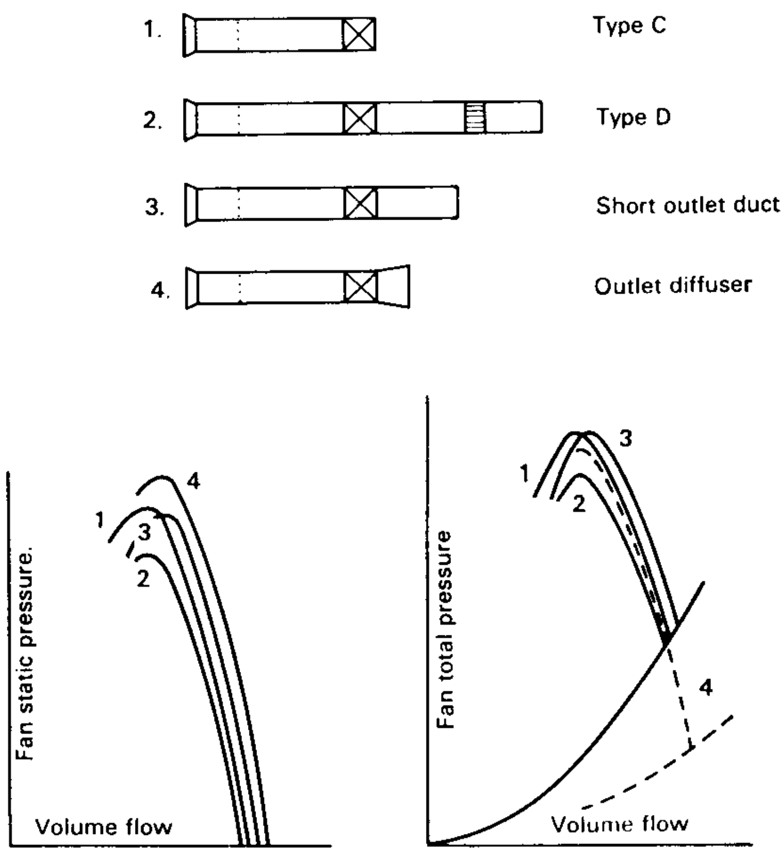


Fig. 11.7 Fan characteristic with outlet swirl.

The performance of a fan with outlet swirl can be presented by plotting fan static pressure, fan velocity pressure and fan (total) efficiency against volume flow for the Type C, open outlet, configuration. A factor based on test results can then be quoted by which the fan velocity pressure should be multiplied to give the additional static pressure available when a short outlet duct, a diffuser of specified dimensions, or any other outlet side component is fitted.

11.3.7 Flow measurement by nozzle and orifice

Volume flow in standardised airways may be determined by Fitot-static tube traverse exactly as described in paragraphs 11.2.2 and 3. With careful observance of the requirements of a national test code it is possible to limit the error to ± 2%.

Nozzles and orifices depend on the measurement of the pressure drop across a restriction through which the whole flow is passed and several varieties are illustrated in Fig. 11.8. The *orifice* is in one sense a simple hole in a plate, but it is also an instrument, and must be made and maintained with careful attention to code requirements - particularly the sharpness of the upstream edge of the hole. A *nozzle* is a smoothly-formed contraction to a parallel throat. Dimensional details are fairly

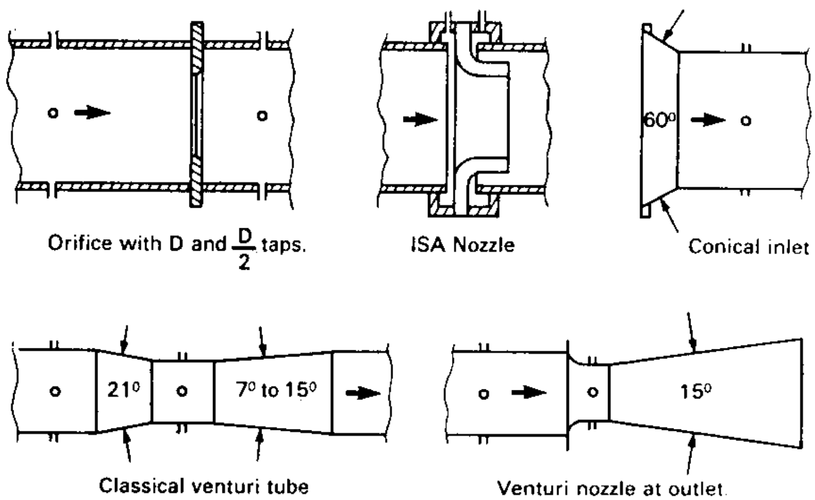


Fig. 11.8 Flow measurement devices.

complicated, and should be taken from a standardised test code. A nozzle becomes a venturi-nozzle when the throat is followed by a gradual expansion designed to recover most of the pressure drop, which is now measured by a static tapping in the throat. A classical venturi-tube is the forerunner of the venturi-nozzle, formed with a 21 ° cone in place of the curved contraction.

General flow measurement codes such as BS 1042, DIN 1952, ISO 8541 and ISO 8781 are limited to the use of orifices and nozzles inserted in a length of duct or pipe. However, several fan test codes give coefficients which have been established when the same devices are fitted at the inlet to a duct from free space or at the outlet from a duct into space.

11.3.8 Calculation of flow rate

The flow rate can be calculated either in terms of volume flow:

$$Q_v = \alpha \, \epsilon \frac{\pi}{4} d^2 \sqrt{\frac{2\Delta p}{\rho_1}} \ m^3/s$$

or of mass flow:

$$Q_m = \alpha \, \epsilon \frac{\pi}{4} d^2 \sqrt{2 \, . \, \Delta p \, . \, \rho_1} \ kg/s$$

d is the throat diameter of nozzle or orifice in m.

D is the duct diameter at the upstream pressure tapping in m.

Δp is the upstream static pressure minus the downstream or throat static pressure in Pa.

$\boldsymbol{\rho_1}$ is the density at the upstream tapping in kg/m^3.

$\mathbf{Q_v}$ is measured in m^3/s at the upstream density, ρ_1

$\mathbf{Q_m}$ is constant right through the airway system in kg/m^3.

α is the flow coefficient, as given in the appropriate code.

ε is the expansibility factor, as given in the appropriate code. For testing with atmospheric air ε may be taken as 1.00 provided Δp does not exceed 1000 Pa.

α is sometimes, but less conveniently, replaced by CE where

C is the coefficient of discharge and

E is the velocity of approach factor $\doteq \dfrac{D^2}{\sqrt{D^4 - d^4}}$

Both a and C are dependent on

β= d/D the diameter ratio and

Re_D or Re_d the duct or throat Reynold's number which is either based on the average velocity and air conditions in the approach duct D, or those in the nozzle throat or orifice opening d.

The Reynold's number need only be known quite roughly since the changes it produces in α or C are small. When testing with atmospheric air it is sufficient to use the approximate formulae below, using either Q_v the expected volume flow, or Δp the observed pressure difference.

Preliminary values of α and a may be taken from Fig. 11.9. Having found Re a precise value of α can be looked up in the charts or tables in the code, which will also give precise data for ε, appropriate to the particular device.

Quantity	Re_D	Re_d
Exact value	$\frac{Dv_D\rho_D}{\mu_D}$	$\frac{dv_d\rho_d}{\mu_d}$
Approximately, given Q_v	$85{,}000\,\frac{Q_v}{D}$	$85{,}000\,\frac{Q_v}{d}$
Approximately, given Δp	$86{,}000\,\alpha\,\epsilon\,\frac{d^2}{D}\sqrt{\Delta p}$	$86{,}000\,\alpha\,\epsilon\,d\sqrt{\Delta p}$

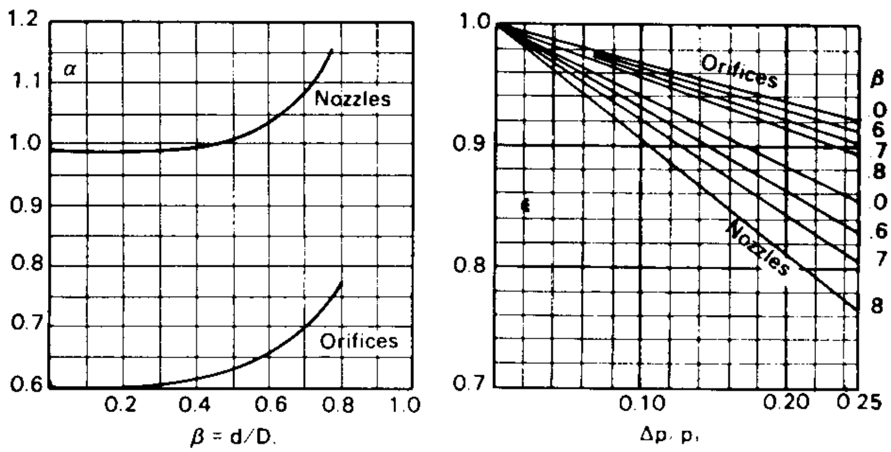


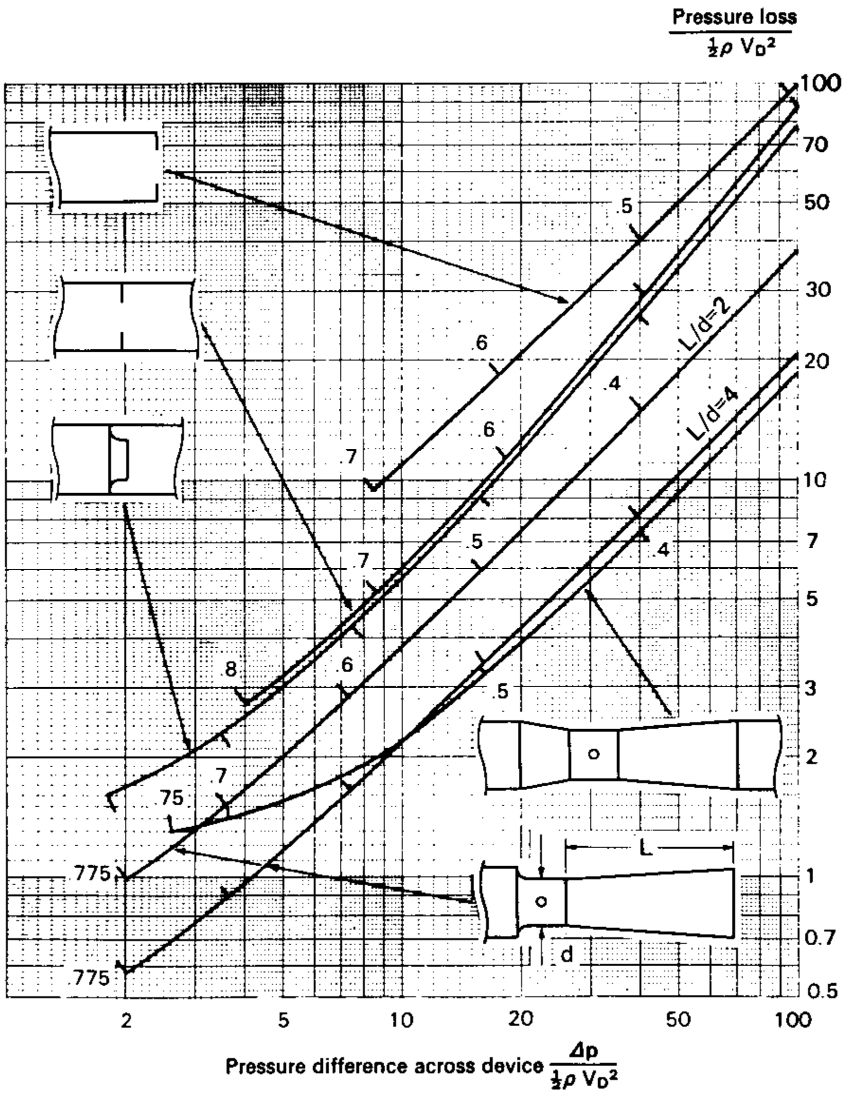
Fig. 11.9 Approximate nozzle and orifice coefficients for air.

Any flow measurement device uses up some of the fan total pressure, and representative pressure losses are given in Fig. 11.10. It is assumed that they are placed in an outlet side test duct, and the velocity pressure lost at final outlet from the duct or device is included. The loss is plotted against the useful pressure, Δp, both in terms of the velocity pressure upstream of the device. The lower the loss the closer we can carry the fan characteristic towards completely free flow without needing an auxiliary booster fan.

11.3.9 Calculation of air density

The first step is to calculate the density of the ambient air in the test room. For ordinary purposes measurement of the barometric pressure, p_o Pa, and the dry-bulb temperature, t_o °C, is sufficient, the density being:

$$\rho_o = 1{\cdot}20\left(\frac{289}{273 + t_o}\right)\left(\frac{p_o}{100{,}000}\right)\ \text{kg/m}^3$$



D = diameter of approach duct

d = diameter of orifice or nozzle throat

V_D = air velocity in approach duct

Figures on curves are values of d/D.

Loss includes velocity pressure loss at final airway outlet.

Fig. 11.10

Pressure difference and pressure loss in nozzles and orifices.

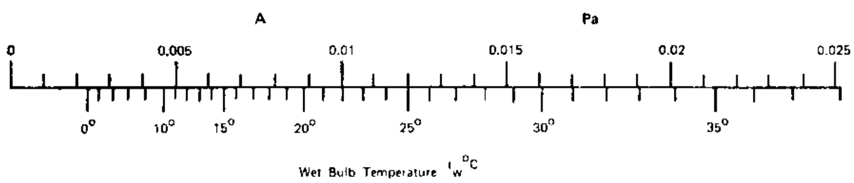
$$p_o \text{ (in pascals)} = 100 \times \text{(pressure in millibars)}$$
$$= 133 \times \text{(pressure in mm of mercury)}$$

The above formula is correct for air with 1% moisture content by weight and holds within ± 0.5% for any condition between dry air at any temperature and saturated air up to 25°C.

For precision tests, and in air of tropical heat and humidity, it is necessary to measure the wet-bulb temperature, t_w °C, preferably with a sling or Assmann hygrometer. The density may then be found from the following formula, which is correct within ± 0.1%, or from tabulated data or psychrometric charts. See also Table 14.10 and Fig. 14.1 in Chapter 14.

$$\rho_o = 1{\cdot}205 \left(\frac{289}{273 + t_o}\right) \left[\left(\frac{p_o}{10^5}\right) \left(1 + \frac{t_o - t_w}{4000}\right) - A \right] \text{kg/m}^3$$

The factor A depends on t_w only and may be obtained from the following linear chart:



$$A = 0{\cdot}378 \times 10^{-5}\, p_{sat}$$

Fig. 11.11 Saturated vapour pressure factor –A

In a standardised airway test the density, ρ_x, is required at the section of pressure measurement upstream of the fan and also upstream of the flow measurement device; in each case the static pressure, p_{sx} Pa, and the dry-bulb temperature, t_x °C, will be observed. Then noting that p_{sx} may be positive or negative:

$$\rho_x = \rho_o \left(\frac{273 + t_o}{273 + t_x}\right) \left(\frac{p_o + p_{sx}}{p_o}\right) \text{kg/m}^3$$

Example. Upstream of the fan in a Type C standardised airway the pressure p_s = -760 Pa and the temperature 28°C. The reference barometric pressure is 101,300 Pa, dry-bulb 24°C and wet-bulb 12°C. What is the fan inlet density ρ_x?

Ignoring humidity:

$$\rho_o = 1{\cdot}20 \left(\frac{289}{273 + 24}\right) \left(\frac{101{,}300}{100{,}000}\right) = 1{\cdot}18 \text{ kg/m}^3$$

$$\rho_x = 1{\cdot}18 \left(\frac{273 + 24}{273 + 28}\right) \left(\frac{101{,}300 - 760}{101{,}300}\right) = 1{\cdot}16 \text{ kg/m}^3$$

Using the more precise formula, note that at t_w = 12°C, A = 0.0054.

$$\rho_o = 1{\cdot}205 \left(\frac{289}{297}\right) \left[1{\cdot}013 \left(1 + \frac{24-12}{4000}\right) - 0{\cdot}0054\right]$$

$$= 1{\cdot}173\,[1{\cdot}013 \times 1{\cdot}003 - 0{\cdot}0054]$$
$$= 1{\cdot}173 \times 1{\cdot}011 = 1{\cdot}185\ \mathrm{kg/m^3}$$

$$\rho_x = 1{\cdot}185 \left(\frac{297}{301}\right) \left(\frac{100{,}540}{101{,}300}\right) = 1{\cdot}161\ \mathrm{kg/m^3}$$

11.3.10 Calculation of efficiency

The *impeller* power measures the work done to drive the impeller round against the aerodynamic forces. The shaft power is often the quantity actually measured. It includes the bearing losses, which make a negligible difference except in very small fans, provided the bearings are of ball or roller type. Losses in transmission elements such as belt drives must be estimated separately, and deducted before calculating the *fan total efficiency*. An *overall efficiency* is sometimes quoted to include the losses in an electric driving motor, or other source of power.

The power input is best measured by a swinging frame dynamometer if a suitable one is available. Usually, however, the quantity measured will be the electrical power input at the driving motor. This may have an efficiency calibration or the efficiency may be calculated by the summation of losses method. In both cases it is important to make sure that the working temperature of the motor windings is about the same as that on which the efficiency estimate is based. If:

Inlet volume flow	= Q_{v1}	$\mathrm{m^3/s}$
Fan total pressure	= P_{Ft}	Pa
Impeller power	= W_i	W

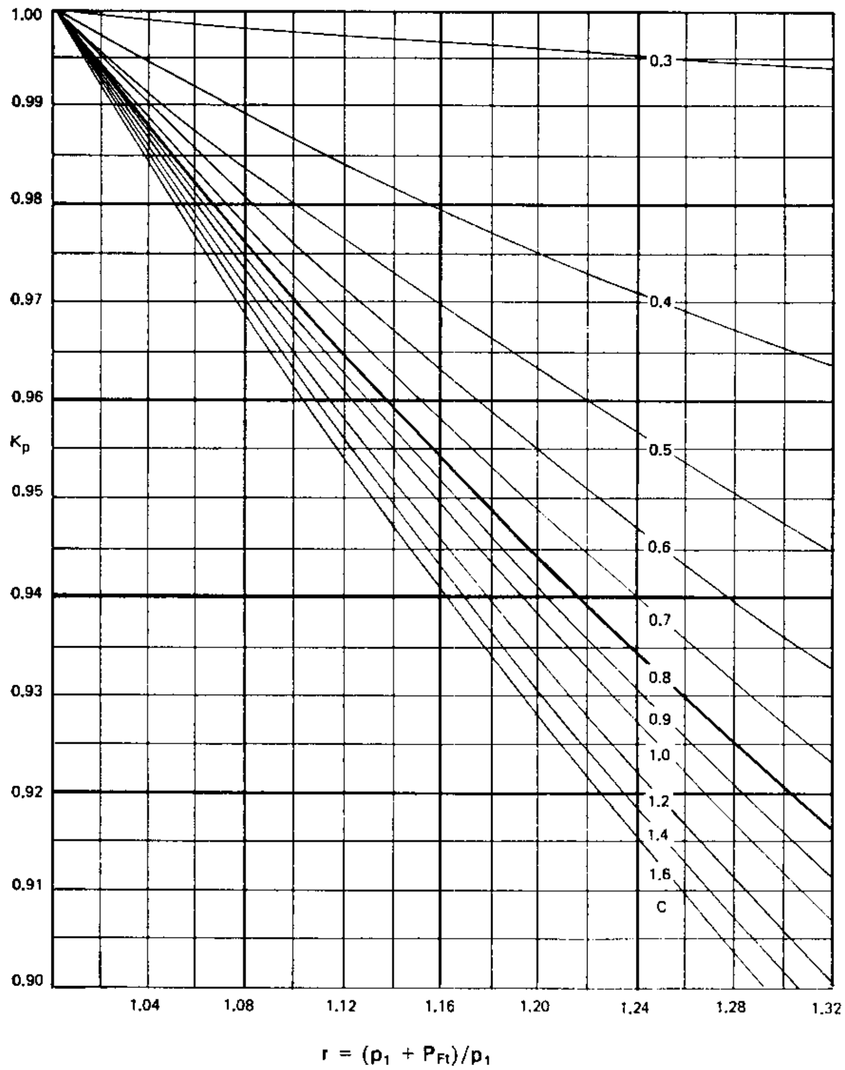
Fan total efficiency = η_t per unit (or $100\eta_t$) %

$$\eta_t = \frac{Q_{v1}\,P_{Ft}}{W_i}$$

The above simple formula holds for moderate fan pressures-say up to 2500 Pa-when the air can be considered incompressible. At higher pressures it is necessary to take account of the rise in air density through the impeller: the air discharged will be compressed to a volume flow rate Q_{v2} smaller than Q_{v1} Both the work done and the power input are reduced by these changes, and, to avoid overestimating the efficiency, 0.1 must be reduced to a mean value K_p $Q_{v1}$where K_p is called the *compressibility factor*.

$$\eta_t = \frac{K_p\,Q_{v1}\,P_{Ft}}{W_i}$$

The change in density is not a simple function of the pressure rise because the air is also heated, both by the compression and by the losses in the fan. At first sight there is a difficulty here because we do not know the losses until we know the efficiency, and we do not know the



$$C = \frac{q_{v1} \cdot P_{Ft}}{W_i} \text{ for air or other gas with } \gamma = 1{\cdot}4$$

$$C = \frac{\gamma \cdot q_{v1} \cdot P_{Ft}}{3{\cdot}5\,(\gamma - 1)\,W_i} \text{ for any gas with isentropic exponent } \gamma$$

Fig. 11.12 Compressibility factor K_p.

efficiency until we know the losses! Fortunately, trial and error methods can be avoided by applying the thermodynamic laws of *polytropic compression* which yield:

$$K_p = \frac{E \log_{10} [1 + (r - 1)]}{\log_{10} [1 + E (r - 1)]}$$

$$\text{where } E = \frac{(\gamma - 1) W_i}{\gamma . Q_{v1} . P_{Ft}}$$

$$r = \frac{p_1 + P_{Ft}}{p_1} \text{ the absolute pressure ratio}$$

γ = 1·4 for air, the "isentropic exponent"

Calculation of K_p can be avoided by the use of Fig. 11.12, which is drawn for direct use with air, i.e. with $\gamma = 1.4$.

Example

Q_{v1} = 12·0m³/s	P_{Ft} = 4000 Pa
W_i = 60 kW	$p_1 = p_o$ = 102,000 Pa

From the incompressible flow formula:

$$\eta_t = \frac{12{\cdot}0 \times 4000}{60{,}000} = 0{\cdot}800 \text{ or } 80{\cdot}0\%$$

but, at 4000 Pa, the compressible flow formula should be used:

$$r = \frac{102{,}000 + 4000}{102{,}000} = 1{\cdot}0392$$

$$\frac{Q_{v1}\, P_{Ft}}{W_i} = 0{\cdot}800 \text{ as before, so } E = \frac{1{\cdot}4 - 1}{1{\cdot}4 \times 0{\cdot}800} = 0{\cdot}357$$

$$\text{Therefore } K_p = \frac{0{\cdot}357 \log_{10} 1{\cdot}0392}{\log_{10} 1{\cdot}0140} = \frac{0{\cdot}357 \times 0{\cdot}01670}{0{\cdot}00604} = 0{\cdot}987$$

$$\text{Therefore } \eta_t = 0{\cdot}80 \times 0{\cdot}987 = 0{\cdot}790 = 79{\cdot}0\%$$

There is not a big difference, but, for a machine approaching the limiting pressure rise for a fan, r = 1.3 or 30,000 Pa with atmospheric pressure at inlet, the drop would be from 86% apparent to 79% true efficiency.

11.4 The Fan Laws

11.4.1 Change of speed and density

A standardised airway test will be carried out at a speed, N rev/s, and inlet density, ρ, kg/m³, which are unlikely to be exactly those specified. To find the performance at the required speed, N^1, and density, ρ^1, (which will normally be 1.20 kg/m³) the observed values of inlet volume flow, Q_{v1}, fan total pressure, P_{Ft} etc. should be converted to $Q^1{}_{v1}P_{Ft}$ etc. by the use of the following relationships:

Volume flow $$\frac{Q'_{v1}}{Q_{v1}} = \left(\frac{N'}{N}\right)$$

Fan total pressure $$\frac{P'_{Ft}}{P_{Ft}} = \left(\frac{N'}{N}\right)^2 \left(\frac{\rho'_1}{\rho_1}\right)$$

Impeller power $$\frac{W'_i}{W_i} = \left(\frac{N'}{N}\right)^3 \left(\frac{\rho'_1}{\rho_1}\right)$$

Fan total efficiency $$\eta'_t = \eta_t$$

Sound power level $$SWL_1 = SWL + 10a \log_{10}\left(\frac{N'}{N}\right) + 22 \log_{10}\left(\frac{\rho'_1}{\rho_1}\right) - 12 \log_{10}\left(\frac{p'_1}{p_1}\right)$$

where 10a ranges from 50 for some centrifugal fans to 55 for some axial fans and p_1 is the absolute inlet pressure (see Section 10.4). The coefficients 22 and 12 are approximations to the small corrections required.

Note that the efficiency does not change. The fan velocity pressure and fan static pressure change in the same ratio as the fan total pressure. a, the speed exponent for sound power, will be between 5.0 and 5.6 according to the type and design of fan. All the formulae are sufficiently accurate for use at fan total pressures up to 2500 Pa in atmospheric air. Above that pressure account should be taken of compressibility - see 11.4.3.

11.4.2 Change of size-model tests

When the performance of a model fan of diameter D m is to be used to predict the performance of a larger, geometrically similar, fan of diameter D' the rules for speed and density change are extended as follows:

Volume flow $$\frac{Q'_{v1}}{Q_{v1}} = \left(\frac{N'}{N}\right)\left(\frac{D'}{D}\right)^3\left(\frac{K_p}{K'_p}\right)$$

Fan total pressure $$\frac{P'_{Ft}}{P_{Ft}} = \left(\frac{N'}{N}\right)^2\left(\frac{D'}{D}\right)^2\left(\frac{\rho'_1}{\rho_1}\right)$$

Impeller power $$\frac{W'_i}{W_i} = \left(\frac{N'}{N}\right)^3\left(\frac{D'}{D}\right)^5\left(\frac{\rho'_1}{\rho_1}\right)$$

Fan total efficiency $$\eta'_t = \eta_t$$

Sound power level $$= SWL' = SWL + 10a \log_{10}\frac{N'D'}{ND} + 20 \log_{10}\frac{D'}{D} + 22 \log_{10}\frac{\rho'_1}{\rho_1} - 12 \log_{10}\left(\frac{\gamma' p'_1}{\gamma p_1}\right)$$

In the formula for sound power level a has the same significance as in Section 11.4.1, but account has been taken of the possibility of change in inlet absolute pressure p_1 and in the gas, y. These do not affect the other formulae.

The provisos of the previous section still hold. The fan tested must always be the smaller of the two, and if the size difference is great, the larger fan may be expected to develop slightly more than the fan total pressure given by the above formula, with corresponding slight improvement in efficiency. This is the so-called scale effect, and adjustments to the simple formula can only be made if both parties to the test agree that there is satisfactory experimental evidence for a scaling formula applying to the type and design of fan in question.

If the performance of a whole range of fans is to be predicted it is best to take thorough performance tests at diameter intervals of not more than 2: 1. Not only is there a Reynold's number effect, but there are inevitably some significant departures from true geometric similarity, notably in surface roughness (which should increase with size, but does not), blade thickness and impeller to casing clearances.

11.4.3 Influence of compression on the fan laws

Suppose that a high-pressure fan is tested at half speed, and that it raises the air density by 2½% in passing from inlet to outlet of the impeller. If the speed is now doubled to full speed the compression will become 10% which means that, while the peripheral speed at impeller outlet has increased by 100%, the axial air velocity has only increased by 85%. Clearly this alters the flow pattern through the impeller, and we cannot expect the full speed performance to be accurately predicted by applying the simple speed change law to the half speed test. While geometrical similarity of impeller and casing are preserved geometrical similarity of air flow has been lost.

However, the prediction is found to be much better if, instead of applying the simple laws to the inlet volume flow and density Q_{v1} and ρ_1, we apply them to a mean air condition, Q_{vm} and ρ_m, which can be visualised as existing half-way through the impeller. Equal and opposite distortions of the flow patterns at inlet and outlet will minimise the errors. A suitable mean density is defined by the use of the compressibility factor K_p (see 11.3.10).

$$\rho_m = \rho_1/K_p \qquad \rho_2 = \rho_m/K_p \text{ approx.}$$
$$Q_{vm} = K_p\, Q_{v1} \qquad Q_{v2} = K_p^2\, Q_{v1} \text{ approx.}$$

The mean volume flow is not directly applicable to performance ratings or practical application, so the mean flow form of the laws is best converted by reference to the inlet condition thus:

Volume flow $$\frac{Q'_{v1}}{Q_{v1}} = \left(\frac{N'}{N}\right)\left(\frac{D'}{D}\right)^3\left(\frac{K_p}{K'_p}\right)$$

Fan total pressure $$\frac{P'_{Ft}}{P_{Ft}} = \left(\frac{N'}{N}\right)^2\left(\frac{D'}{D}\right)^2\left(\frac{\rho'_1}{\rho_1}\right)\left(\frac{K_p}{K'_p}\right)$$

Fan velocity pressure $$\frac{P'_{Fv}}{P_{Fv}} = \left(\frac{N'}{N}\right)^2\left(\frac{D'}{D}\right)^2\left(\frac{\rho'_1}{\rho_1}\right)$$

Impeller power $$\frac{W'_i}{W_i} = \left(\frac{N'}{N}\right)^3\left(\frac{D'}{D}\right)^5\left(\frac{\rho'_1}{\rho_1}\right)\left(\frac{K_p}{K'_p}\right)$$

Fan total efficiency $\eta'_t = \eta_t$

Fan sound power:

Further correction of the SWL formula is unnecessary.

The compressibility coefficient K'_P applying to the predicted performance is not known in advance. It can, however, be calculated from the following formula, in terms of quantities which are all known from the test.

$$\frac{K_p}{K'_p} = 1 + (m-1)\ (1 - K_p) \text{ approx.}$$

$$\text{where } m = \left(\frac{N'}{N}\right)^2 \left(\frac{D'}{D}\right)^2 \left(\frac{\rho'_1}{\rho_1}\right) \left(\frac{p_1}{p_1'}\right)$$

$$= \left(\frac{N'}{N}\right)^2 \left(\frac{D'}{D}\right)^2 \left(\frac{273 + t}{273 + t'}\right)$$

The fan laws in this form need not be applied below 2500 Pa, nor for K_p changes of less than 0.01. They should hold for values of K'_P within the range ($K_p \pm 0.03$). For larger changes in compression ratio they will give the best available prediction, when supplemented by any necessary scale effect adjustments for size change. If there is a change of gas with isentropic exponent γ' in place of γ and a computer is available the exact value of K_p/K'_P could be calculated from:

$$\frac{K_p}{K'_p} = \frac{[1 + m\ E'\ (r-1)]^{K_p/E'} - 1}{m\ (r-1)}$$

$$\text{where } E' = \frac{(\gamma' - 1)\ W_i}{\gamma'\ .\ Q_{v1}\ .\ P_{Ft}}$$

11.4.4 Examples of the application of the fan laws

Conversion from test to standard conditions

A 1000mm fan is tested at 1380 rev/min (23.0 rev/s) in a Type B standardised airway with the following results:

$q_{v1} = 20 m^3/s$. $P_{Ft} = 1520$ Pa. $W_i = 40$ kW.

SWL = 112 dBW. $p_1 = p_0 = 98{,}000$ Pa. $t_1 = 20°C$.

Therefore $\rho_1 = 1{\cdot}16\ kg/m^3$

What is the performance at 1470 rev/min (24.5 rev/s) and 1.20 kg/m^3 inlet density?

$$q'_{v1} = 20{\cdot}0 \times \frac{24{\cdot}5}{23{\cdot}0} = 21{\cdot}3 m^3/s$$

$$P'_{Ft} = 1520 \times \left(\frac{24{\cdot}5}{23{\cdot}0}\right)^2 \left(\frac{1{\cdot}20}{1{\cdot}16}\right) = 1790 \text{ Pa}$$

$$\eta'_t = \eta_t = \frac{20{\cdot}0 \times 1520}{40{,}000} = 76\%$$

$$W'_i = \frac{21{\cdot}3 \times 1790}{0{\cdot}760} = 50{\cdot}1 \text{ kW}$$

$$\text{SWL} = 112 + 55 \log_{10} \frac{24{\cdot}5}{23{\cdot}0} + 22 \log_{10} \frac{1{\cdot}20}{1{\cdot}16}$$

$$= 112 + 55 \times 0{\cdot}0275 + 22 \times 0{\cdot}0147$$

$$= 112 + 1{\cdot}5 + 0{\cdot}3 = 114 \text{ dBW}$$

Performance prediction with change of size

What is the expected performance of a 1250mm fan at 1770 rev/min (29.5 rev/s), geometrically similar to that above?

$$q'_{v1} = 20{\cdot}0 \times \left(\frac{29{\cdot}5}{23{\cdot}0}\right) \times \left(\frac{1{\cdot}25}{1{\cdot}00}\right)^3 = 50{\cdot}0\text{m}^3/\text{s}$$

$$P'_{Ft} = 1520 \times \left(\frac{29{\cdot}5}{23{\cdot}0}\right)^2 \times \left(\frac{1{\cdot}25}{1{\cdot}00}\right)^2 \times \left(\frac{1{\cdot}20}{1{\cdot}16}\right) = 4020 \text{ Pa}$$

$$\eta'_t = 76\% \quad \text{Therefore } W'_i = \frac{50{\cdot}0 \times 4020}{0{\cdot}76} = 264 \text{ kW}$$

$$\text{SWL} = 112 + 55 \log_{10} \frac{29{\cdot}5 \times 1{\cdot}25}{23{\cdot}0 \times 1{\cdot}00} + 20 \log_{10} \frac{1{\cdot}25}{1{\cdot}00}$$

$$+ 10 \log_{10} \frac{1{\cdot}20}{1{\cdot}16} = 112 + 10{\cdot}3 + 2{\cdot}0 + 0{\cdot}1$$

$$= 124 \text{ dBW}$$

Correction for compressibility

At the 4020 Pa of the last example we are in the range where compressibility should be considered. In the original, 1000mm, 1380 rev/ min, test:

$$r = \frac{98{,}000 + 1520}{98{,}000} = 1{\cdot}01551$$

$$E = \frac{0{\cdot}4 \times 40{,}000}{1{\cdot}4 \times 20 \times 1520} = 0{\cdot}376$$

$$K_p = \frac{0{\cdot}376 \log_{10} 1{\cdot}01551}{\log_{10} 1{\cdot}00583} = 0{\cdot}995$$

$$m = \left(\frac{29{\cdot}5}{23{\cdot}0}\right)^2 \left(\frac{1{\cdot}25}{1{\cdot}00}\right)^2 \left(\frac{273 + 20}{273 + 16}\right) = 2{\cdot}60$$

$$\frac{K_p}{K'_p} = 1 + (2{\cdot}60 - 1)\ (1 - 0{\cdot}995) = 1{\cdot}008$$

Exact calculation using the computer formula gives 1.0077.

$$q'_{v1} = 50{\cdot}4\text{m}^3/\text{s} \qquad P_{Ft} = 4050 \text{ Pa} \qquad W'_i = 266 \text{ kW}$$

The changes in this case are insignificant, justifying ignoring compressibility below 3000 Pa.

11.4.5 Non-dimensional coefficients

Some readers will be surprised that no reference has been made to the non-dimensional volume flow, fan pressure, and power coefficients *Φ, Ψ* and *λ*. Confusion can be caused by the lack of international agreement on the reference quantities for these, leading to differing numerical values, and practical requirements are met by the above treatment. They are, in fact, of more use to fan designers than fan users, and each designer will have his own personal definitions.

11.5 Accuracy

11.5.1 Uncertainty *of* measurement

Standard fan test codes do not always state the accuracy levels appropriate to the measurements to be made, and Table 11.1 contains some suggested values for use in default of specific code instructions. The first column is a normal standard, suitable for most tests in standardised airways. A need may be felt in some cases for greater precision, particularly when testing large, high efficiency, fans which contribute significantly to the running costs and performance of a plant. For this reason a second column has been included suggesting the accuracy likely to be attainable with first-class equipment and trained staff, together with the additional calculation steps which then become justified.

There are basic uncertainties in the values of the flow coefficient a for the standardised nozzles and orifices. These range from ± 0.5% to 1.5% and combine with the errors of measurement to produce a probable uncertainty of ± 2.0% to 2.5% for flow measurement in a standard test. In a precision test readings should be repeated until the desired level of statistical consistency is reached, which should secure a flow measurement uncertainty of ± 1.5% if the fan produces reasonably smooth and steady flow.

The uncertainty in determining fan pressures depends on the proportion of fan velocity pressure to fan total pressure and may be obtained from the following expression in which a = 1.5% to 2.0% and b = 5% to 10% :

$$\text{uncertainty of fan pressure} = \sqrt{a^2 + \left(\frac{P_{Fv}}{P_{Ft}}\right)^2 b^2}$$

To summarise the lowest uncertainties that can be expected from standard test procedures will occur around the best efficiency point and are likely to be:

In volume flow ± 2·0%

In fan total pressure ± 1·5% when P_{Fv}/P_{Ft} is low

In fan total pressure ± 2·0% when P_{Fv}/P_{Ft} = 25%

In fan total pressure ± 3·0% when P_{Fv}/P_{Ft} = 50%

In fan total pressure ± 5·0% when P_{Fv}/P_{Ft} = 100%

The percentages quoted here are all 95% *confidence limits.*

This is a statistical term and means that:

the quoted error is likely to be exceeded in 5% of cases;

$\frac{1}{2}$ × the quoted error is likely to be exceeded in 32% of cases;

$1\frac{1}{4}$ × the quoted error is likely to be exceeded in 1% of cases.

Table 11.1
Suggested accuracy of individual measurements

Quantity measured	Standard tests	Precision tests
Barometric pressure:		
Read scale to	± 1mm Hg or 2 mb	± 1 mb (100 Pa)
Gravity corrections	Not required	Apply
Temperature corrections	Not required	Apply
Probable overall accuracy	± 0·5%	± 0·2%
Temperature:		
Calibrated and read to	± 1 °C	± 0·5 °C
Correction to stagnation temp.	Not required	Apply over 30m/s
Wet-buld temperature	Not required	± 2 °C
Density calculation:		
Use formula correct to (11.3.9)	± 0·5%	± 0·1%
Pressure:		
Manometer calibrated to	± 1% or 1 Pa	± 0·5% or 0·5 Pa
Damp to read to	± 1%	± 0·5%
Average of repeated readings	Not required	to ± 0·5%
Dimensions:		
Nozzle and orifice diameters to	± 0·1%	± 0·1%
Airway diameters to	± 0·2%	± 0·1%
Airway areas to	± 0·5%	± 0·25%
Lengths as specified within	− 1% + 10%	Measure to ± 1%
Rotational speed:		
Absolute or calibrated accuracy	± 0·5%	± 0·25%
Average of repeated readings	Not required	to ± 0·25%
Input power:		
Overall accuracy of method	± 2%	± 1% if possible
Electric meters:		
Class index of accuracy	0·5	0·2
Compressibility:		
Use K_p at pressure above	3000 Pa	600 Pa

11.5.2 Tolerance on fan performance

It would be futile to guarantee fan performance to closer limits than the uncertainties of measurement, and to those above must be added errors in measuring speed and air density. In addition there will be the manufacturing tolerances - all fans to a given design cannot be expected to be identical. Fortunately, the combination of the fan supplied with the user's system greatly reduces the deviation from the specified volume flow due to departures from the specified fan performance, and normally this is the significant issue.

Suppose that a fan has been tested, and a smooth fan characteristic, the full line in Fig. 11.13, drawn through the test points. At any point on the fan characteristic we can plot the tolerance range of volume flow horizontally and fan pressure vertically. Now the extreme positive value of q occurs once in 40 tests only and so does the extreme positive value of p. So they are extremely unlikely (once in 40 x 40 = 1,600 tests) to occur simultaneously. In fact the mutual dependence of the two errors produces an elliptical area within which the *true* test point has a 95% probability of lying. Strictly speaking the way in which the fan pressure is calculated gives the pressure error a slight dependence on

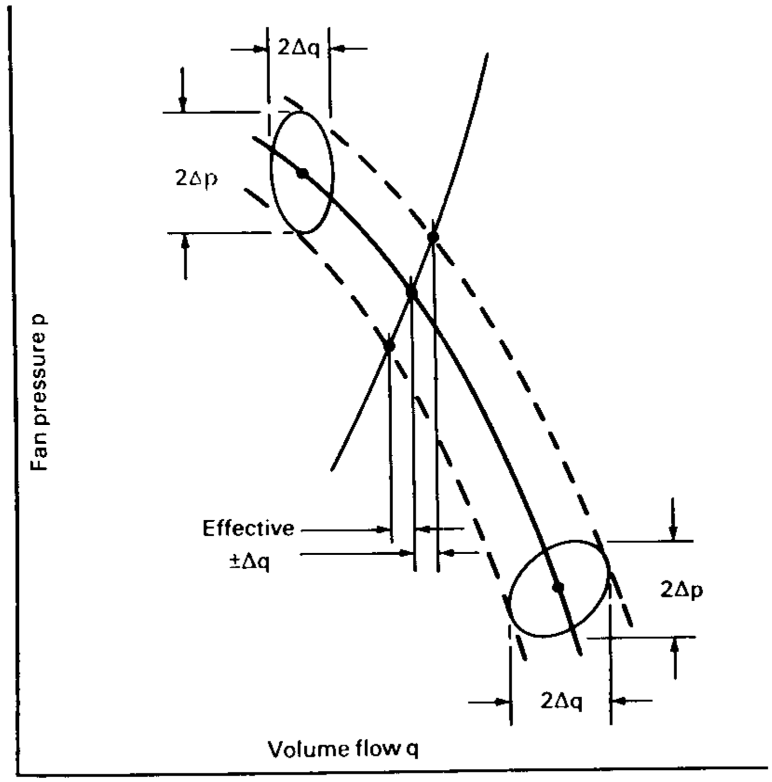


Fig. 11.13

the volume flow error, which is allowed for by canting the ellipse at high volumes.

It will be seen that the broken lines touching the outer edges of all the error ellipses will be two fan characteristics between which the true characteristic will be in 95% of cases. Crossing these has been drawn a system resistance line-here a "square law" line, but see Chapter 9 for other possibilities-representing exactly the user's estimate. The intersections with the broken lines now give the range of volume flow within which the fan performance will be. Often this will be one half or less of the basic uncertainty range of volume flow.

CHAPTER 12

Use of air in heat exchange and drying

In many industries the product requires to be heated, cooled or dried at some stage in the manufacturing process. The techniques involved are numerous and varied, and it would be quite outside the scope of this book to attempt to cover such a broad field. However, atmospheric air is often a partner with the product in an exchange process in which heat or moisture is transferred from one to the other. The effectiveness and economy of this process is greatly dependent on the way in which the airflow is organised and there are plenty of examples of gross waste of fan power. This chapter deals in a very general way with the principles of heat and moisture transfer.

12.1 Heat Transfer

12.1.1 Heat capacity of air

For ordinary heat transfer processes, which take place without significant compression or expansion of the air, the appropriate constant is c_p the *specific heat at constant pressure.* This is the heat in joules (or more conveniently kilojoules, kJ) required to raise the temperature of 1 kg of air by 1 °C. It is practically independent of pressure and varies only slightly with temperature, thus:

Temperature, °C	20	100	200	300	400	500
c_p - kJ/kg, °C	1.006	1.01	1.02	1.04	1.06	1.09

For water vapour c_p is about 2.01 kJ/kg °C, which means an effective value for c_p of approximately (1.01 + m) in the case of humid atmospheric air containing m kg of moisture per kg mixture - a correction that can usually be neglected. m is greater in hot air drying processes, but the much greater heats of evaporation or condensation are then the predominant factors.

Example. Ambient air at a maximum temperature of 40°C is passed at a rate of 0.60m^3/S through an electrical machine to remove losses estimated at 6.0 kW. What is the average temperature of this air as it leaves the machine?

Assuming a barometric pressure of 100,000 Pa the density at inlet is:

$$1{\cdot}20 \times \frac{289}{273 + 40} = 1{\cdot}10 \text{ kg/m}^3$$

The airflow is thus 0.66 kg/s. Adding heat at the rate of 6.0 kW = 6.0 kJ/s produces a temperature rise of:

$$\frac{6{\cdot}0 \text{ (kJ/s)}}{1{\cdot}01 \text{ (kJ/kg °C)} \times 0{\cdot}66 \text{ (kg/s)}} = 9{\cdot}0\text{°C}$$

12.1.2 Heat transfer by convection

Convection implies the continuous movement of successive masses of air to and from a hot surface, extracting and taking away heat as they pass (strictly speaking the first stage of heat transfer takes place by *conduction* across a thin stagnant film of gas which never leaves the surface). The movement may be due simply to the tendency of the heated air, being less dense, to rise and make room for fresh, cold air. This is *natural convection* as found in many familiar space heating appliances. The converse passage of heat from the air to a colder surface follows similar laws, but for simplicity we shall speak as if the air were always the colder.

Natural convection is too slow and uncertain for most industrial processes, and is replaced by *forced convection*. In this the surface from which heat is to be extracted is placed in a relatively high velocity air stream generated by a fan. The air stream is preferably confined by a casing which ensures that none is wasted, passing too far away to take part in the heat extraction.

Calculation for both natural and forced convection is made from the relationship:

$$H = h_c \,.\, A \,.\, \Delta t \qquad (121)$$

H = heat transfer rate in watts, that is joules/sec.

A = area of hot surface in m^2.

Δt = temperature difference between the hot surface and the air stream in °C.

h_c = convection heat transfer coefficient in W/m^2 °C.

The difference is that, in forced convection, h_c is dependent on the air velocity and the pattern of forced air flow over the surface, whereas in natural convection ho is dependent on the temperature difference and the geometry of the surface which together determine the pattern of air movement.

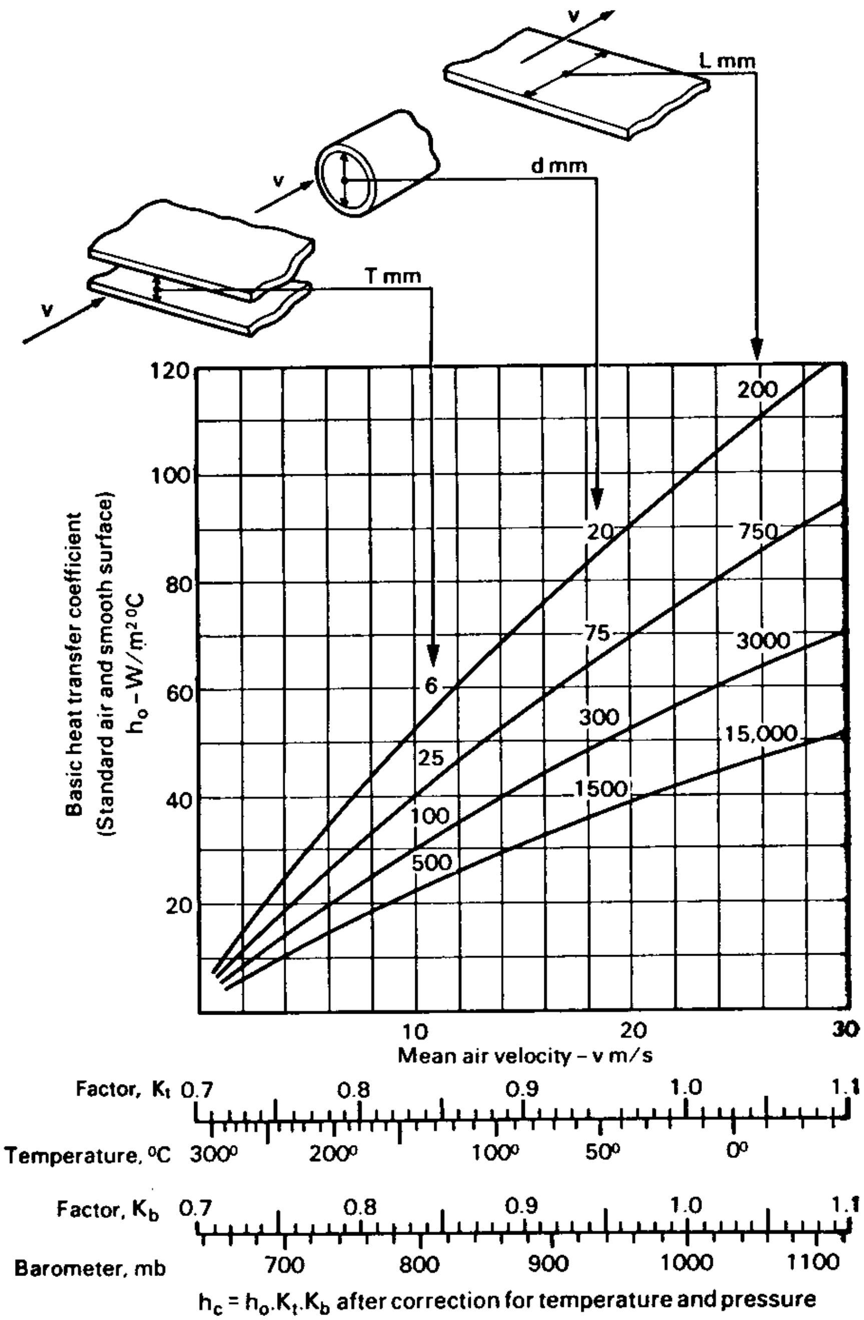


Fig. 12.1

Forced convection with air flowing over smooth surfaces.

12.1.3 Forced convection

For a high rate of heat transfer, turbulent flow must be established over most of the hot surface. This is necessary for the thorough mixing required to carry away the heat-in laminar, streamline, flow only the layers next to the surface will be heated. In fact there is a strong parallelism between the friction factor K of Chapter 6 and h_c - high values of one tend to accompany high values of the other.

With turbulent flow parallel to smooth surfaces as in Fig. 12.1, h_c is found to be proportional to (mass flow)$^{0.8}$. With separated flow, however, as when air flows at right-angles to a bank of tubes, hc is more nearly proportional to (mass flow)$^{0.6}$. The constant of proportionality can only be guessed at in a complex assembly such as a finned tube battery, for which the maker's heat transfer data should in any case be consulted.

Estimates for simple cases with flow parallel to smooth surfaces can sometimes be built up from the established elements plotted in Fig. 12.1. The graph applies to air at 16°C and 100,000 Pa, and corrections are given allowing for density and viscosity changes at other temperatures and pressures. Surface roughness will increase h_o, by a factor of about 2 at 500 μm for example.

Example. In an accelerated paint drying enclosure wet sheets each 1m x 2m are suspended at 100mm spacing. Air at 80°C dry-bulb, 38°C wet-bulb flows between them at 8m/s. At what rate is heat supplied to each sheet?

The 4m^2 surface (both sides) will be at the wet-bulb temperature of 38°C so that Δt = 80 - 38 = 42°C. From Fig. 12.1 the appropriate diagram for T = 100mm and v = 8m/s gives h_o = 25 W/m^2 °C. At 38°C, K_t = 0.96.

$$\begin{aligned} H &= h_o \,.\, K_t \,.\, k_b \,.\, A\Delta t \\ &= 25 \times 0{\cdot}96 \times 1{\cdot}00 \times 4{\cdot}0 \times 42 \\ &= 4000 \text{ W} \end{aligned}$$

12.1.4 Natural convection

As in the case of forced convection, the natural heat loss from objects of complex form can only be found by trial. However, for certain simple shapes, the data are well established, and some of these are summarised in Fig. 12.2. The figures should be used with caution, noting the following limitations on their reliability.

The hot surface should be sufficiently isolated in space for air to flow freely to the base and away at the top under the small gravitational forces generated by changes in density. Thus the vertical surface curve would not be accurate for a wall heated all over, still less for a ceiling where the inability of the heated air to get away would make nonsense of the assumptions. On the other hand both faces of a domestic "radiator' are effective natural convection heat sources, even if one is spaced only a few centimetres from the wall.

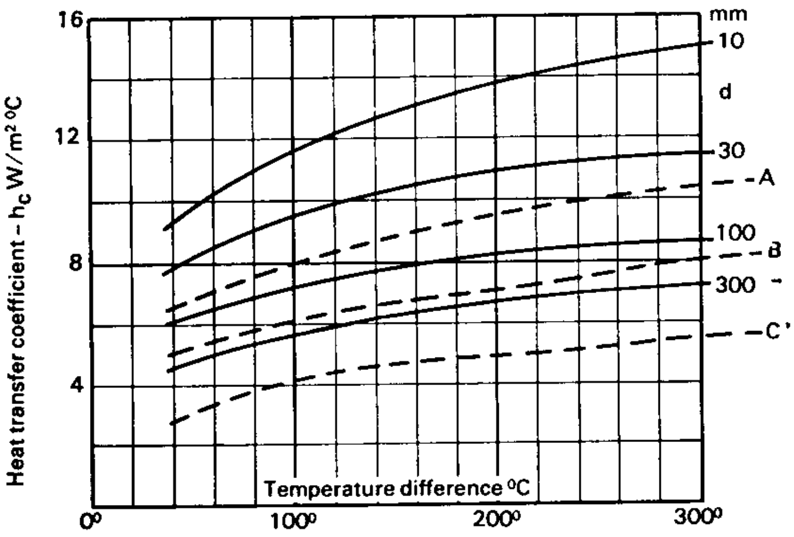


Fig. 12.2 Natural convection to the ambient atmosphere.

A draught in the 0-. to 1 m/s range can increase the natural convection heat transfer of a small source by 50% to 100%, or more if the temperature difference is low. On the other hand a large, hot, surface creates its own draught pattern, which is much less sensitive to casual air movement.

12.1.5 Radiation

All bodies radiate heat at a rate which is proportional to the fourth power of their absolute temperature - (273 + t°C). On the other hand all the surfaces around radiate heat back, and if they are at the same temperature there is precise balance and no heat is transferred. The appropriate temperature to take for the surroundings is the *mean radiant temperature*, t_r°C (see 1.9) and if the hot surface is at t°C the net heat emission will be:

$$5{\cdot}67 \times \epsilon \left[\left(\frac{273 + t}{100}\right)^4 - \left(\frac{273 + t_r}{100}\right)^4 \right] \text{ W/m}^2$$

The factor ε is called the emissivity and ranges from near 1.0 for a matt black surface to nearly zero for a brightly polished mirror-like one, which not only reflects back most of the radiant heat that reaches it, but also retains precisely the same fraction (1 - ε) of the radiant heat it could emit as a *black* body, thereby preserving balance in surroundings of equal temperature.

Fig. 12.3 is based on the above formula, with t_r at room temperature level, and Table 12.1 records the current recommendations of the IHVE for the emissivity of surfaces found in heating practice. The radiant heat transfer is additional to, and completely independent of, forced or natural convection transfers. Fig. 12.3 applies only when the walls of the room or enclosure are at a relatively uniform temperature. They then become effectively one "black body" ($\varepsilon = 1$) since they absorb all the radiation in the room, some after repeated reflection. When there is a strong radiant heat source, or a sink such as a large window, more detailed analysis is necessary.

The heat transfer coefficient h_r W/m^2 °C is directly comparable with h_c for convection, and in the following formula for the heat transfer rate H, the area A m^2 is that of all exposed surfaces. In re-entrant pockets of surface, however, only the visible, projected area across the mouth of the recess should be counted.

$$\mathbf{H = h_r\, A\, (t - t_r)\ watts} \qquad \mathbf{(122)}$$

Example. A white-painted panel in a hot water heating system is 2.0m long by 0.5m high and has a surface temperature of 75°C. What heat output might be expected in a room with an air temperature of 25°C and mean radiant temperature 20 °C ?

From curve B in Fig. 12.2 and a temperature difference of 75 - 25 = 50°C, h_c = 5.0 while the surface area of both sides of the panel is 2.0 m^2. The heat transfer by natural convection is thus:

$$5{\cdot}0\ (\mathrm{W/m^2\ °C}) \times 2{\cdot}0\ (\mathrm{m^2}) \times 50\ (°\mathrm{C}) = 500\ \mathrm{W}$$

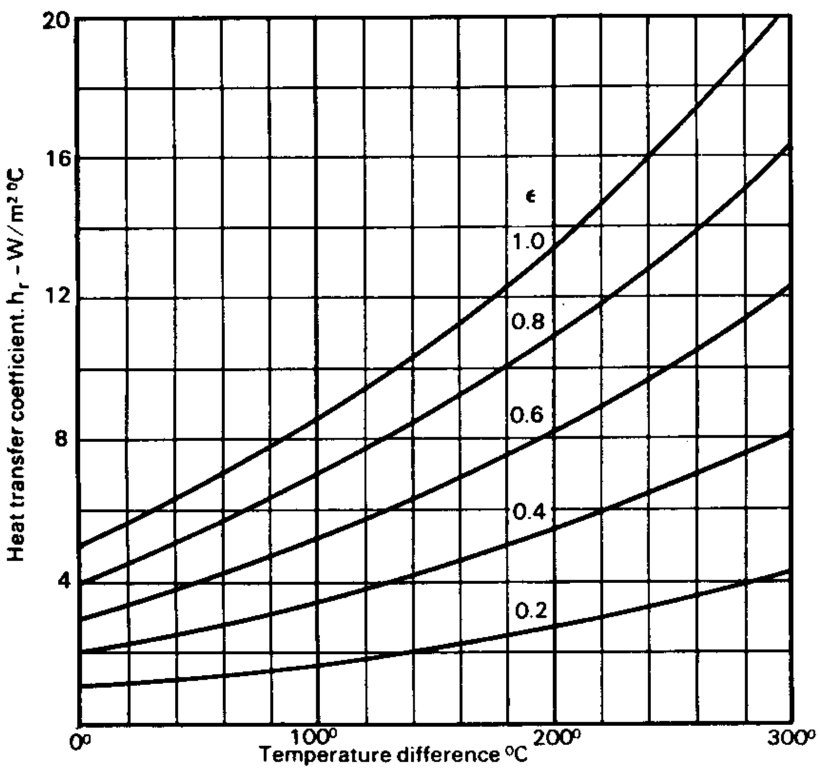


Fig. 12.3 Radiant heat transfer to an enclosure at C.20°

From Fig. 12.3 with a temperature difference of 75 - 20 = 55 °C, and an emissivity assessed at 0.95, h_c = 6.5, and the radiant heat transfer rate is:

$$6{\cdot}5\ (\mathrm{W/m^2\ ^\circ C}) \times 2{\cdot}0\ (\mathrm{m^2}) \times 55\ (^\circ\mathrm{C}) = 715\ \mathrm{W}$$

This gives an estimate of just over 1200 watts, but it must be emphasised that an actual radiator will be rated on the basis of tests under controlled conditions of installation, and in terms of the entering water temperature and flow rate which set the surface temperature distribution over the whole unit.

Table 12.1

Emissivities of various surfaces (after IHVE Guide)

Type of surface	Emissivity ϵ at 50°C	ϵ at 250°C
Painted, gloss white	0·95	0·91
Painted, matt black	0·96	0·97
Steel, oxidised	0·79	0·79
Steel, galvanised, new	0·23	0·42
Steel, galvanised, weathered	0·28	0·90
Steel, stainless, polished	0·15	0·18
Steel, stainless, weathered	0·85	0·85
Aluminium, polished	0·04	0·05
Aluminium, oxidised	0·11	0·12
Aluminium, anodised	0·72	0·46
Copper, polished	0·04	0·05
Copper, oxidised	0·87	0·83

12.2 Heat Exchangers

12.2.1 Temperature distribution

Consider a heat exchange battery designed to transfer heat from hot water flowing through a bank of tubes to an air stream flowing over them. The water will be cooled and the air heated progressively as each passes through the battery. The temperature at which each stream leaves the battery can be calculated from the quantity of heat transferred and the heat capacity of each stream as in the following example:

Heat transferred: 100 kW

Entering air flow: 1.20m^3/s at 0°C: 1.53 kg/s

c_p = 1 -01 kJ/kg 'C temperature rise = 100/(1 -53 x 1 -01) = 65 'C

Entering water flow: 1.20 kg/s at 100°C

c_p = 4.20 kJ/kg °C temperature fall = 100/(1.20 x 4.20) = 20°C

Clearly the local temperature difference across the tube wall, on which the local heat transfer depends, will vary widely from point to point through the battery. It might range from as much as 100°C if the entry points of air and water coincide to as little as (100 - 20) - 65 = 15°C if the exit points coincide. A simple average of the entry difference and the exit difference will always overestimate the effective value. The best estimate is usually the quantity known as the *logarithmic mean temperature difference* Δt_{lm}. If, of the air entry and air exit differences, the larger is Δt_{mx} and the smaller Δt_{mn}

$$\Delta t_{lm} = \frac{\Delta t_{mx} - \Delta t_{mn}}{\log_e (\Delta t_{mx}/\Delta t_{mn})} = \Delta t_{mn} + K_{lm} (\Delta t_{mx} - \Delta t_{mn})$$

taking K_{lm} from Fig. 12.4 for convenience in calculation.

Fig. 12.4 also illustrates the case we have just calculated in the diagram marked "parallel flow". In this it is assumed that air and water enter together at one end of the battery, travel along its length, and emerge together. Clearly, in this case:

$$\Delta t_{mx} = 100 - 0 = 100°C$$

$$\Delta t_{mn} = 80 - 65 = 15°C$$

$$\Delta t_{mx}/\Delta t_{mn} = 100/15 = 6{\cdot}7$$

$$\text{Therefore } K_{lm} = 0{\cdot}351 \text{ from Fig. } 12{\cdot}4$$

$$\text{and } \Delta t_{lm} = 15 + 0{\cdot}351\ (100 - 15) = 45°C$$

If in this same battery we were to reverse the flow of water so that hot water enters where hot air leaves, we should have a "counter flow" system, and would find that the temperatures became those shown in the second diagram. Then:

$$\Delta t_{mx} = 78 - 0 = 78°C$$

$$\Delta t_{mn} = 100 - 71 = 29°C$$

$$\Delta t_{mx}/\Delta t_{mn} = 78/29 = 2{\cdot}7 \quad \text{Therefore } K_{lm} = 0{\cdot}418$$

$$\text{and } \Delta t_{lm} = 29 + 0{\cdot}418\ (78 - 29) = 49{\cdot}5°C$$

We see that the log mean temperature rise has gone up by about 10%, and so have the temperature rise of the air and the temperature fall of the water. This equality of percentage changes is in fact the condition that determines the new values, and it means also that the heat exchange rate of the battery has gone up 10% to 110 kW. Indeed a fully counterflow system quite generally makes the most effective use of a given battery if it can be achieved.

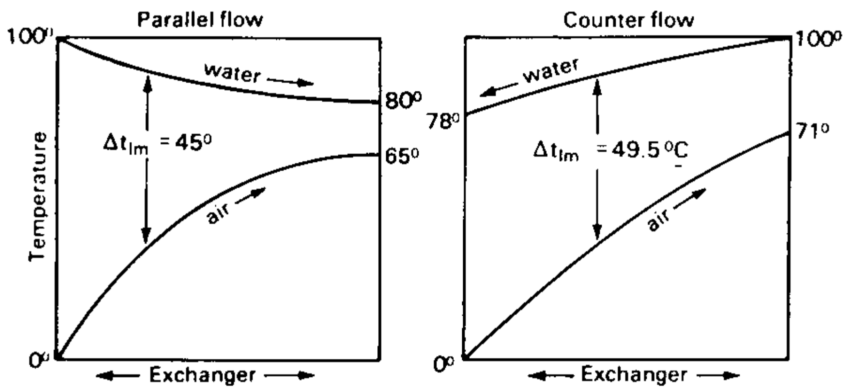
12.2.2 Effectiveness of heat exchangers

In most heat exchangers involving two fluids such as air and water the objective is to change the temperature of one of them, the primary fluid, only. The other, secondary, fluid is there merely as a means to an end. From one point of view we might suppose that the exchanger was

100% effective if it brought the primary fluid right to the entering temperature of the secondary. Not only is this strictly impossible, however, but it would require a grossly oversized exchanger and excessive secondary flow to get close to it. Nevertheless, the *heat exchange effectiveness* defined as:

$$E = \frac{\text{Temperature change of primary fluid}}{\text{Temperature difference of fluids at entry}}$$

is a valuable index of performance. It also has the property of remaining the same for geometrically similar heat exchangers whatever the temperatures of the two fluids, provided the flow rates are kept in the same proportions.



$$\Delta t_{lm} = \Delta t_{mn} + K_{lm}\,(\Delta t_{mx} - \Delta t_{mn})$$

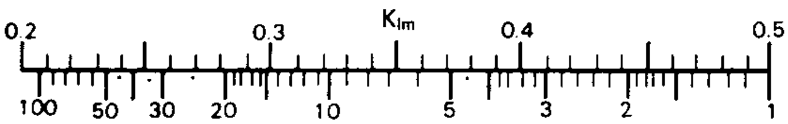


Fig. 12.4 Logarithmic mean temperature difference Δt_{lm}.

If we have a flow of M_P kg/s of the primary fluid with specific heat c_p kJ/kg °C it will have a heat capacity of $G_P = M_P\, c_p$ kW/°C, this being the heat transfer rate needed to change its temperature by 1 °C. The heat exchange effectiveness E now depends on two factors:

$$G_H/G_P \text{ and } G_s/G_P$$

where G_H is the heat exchange capacity of the exchanger in kW per °C log mean temperature difference and G_s is the heat capacity of the flow of secondary fluid. $G_s = M_s c_s$, kW/°C. The curve marked m is used for steam. Because steam supplies heat by condensing at substantially a constant temperature throughout the battery tubing it is equivalent to an almost infinite flow of hot water which could give up heat without any drop in temperature.

The general nature of the dependence of E on these two factors is shown in Fig. 12.5. The cross-flow chart applies to the form of heat exchanger typified by the air heater battery in which a narrow bank of tubes spans a duct through which the air flows. This particular chart is enlarged in Fig. 12.6 to demonstrate its use in predicting the result of changes in temperatures and flow rates when the performance of a heat exchanger is known for one set of conditions only. Changes in flow rate will change G_s or G_P with resulting change in G_H. For a cross-flow battery with most of the temperature difference on the air side G_H will be roughly proportional to $(G_s)^{0.6}$ - or $(G_P)^{0.6}$ if air is the primary fluid. The linear chart in Fig. 12.7 follows this relationship.

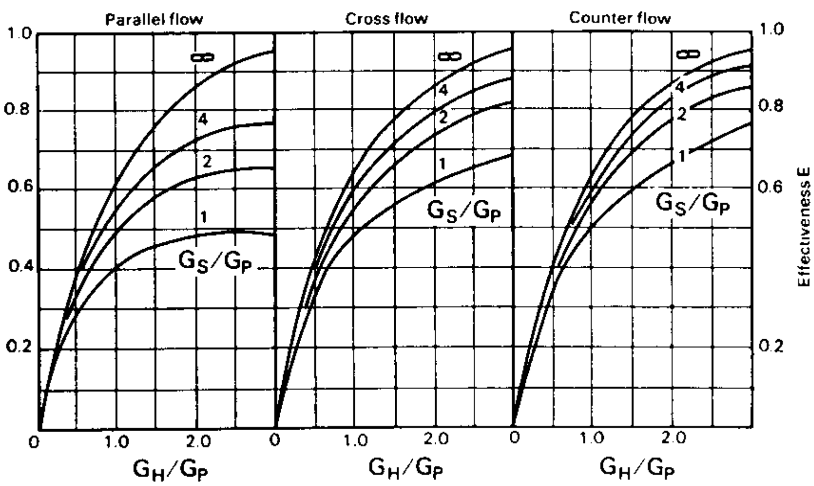


Fig. 12.5 Effectiveness of different flow arrangements.

12.2.3 Example-Use of effectiveness chart

A process liquid circulating at 110°C is to be cooled to 50°C before discharge, to limit flammability. This is done in a cross-flow heat exchange battery using atmospheric air as a secondary fluid. The battery is designed to deal with 4 kg/sec of liquid having a specific heat of 1.80 kJ/kg °C with an air flow of 32m³/s at 35°C.

$$G_P = 4\,(\mathrm{kg/s}) \times 1{\cdot}80\,(\mathrm{kJ/kg\,^\circ C}) = 7{\cdot}2\,\mathrm{kW/^\circ C}$$

$$G_S = 32\,(\mathrm{m^3/s}) \times 1{\cdot}12\,(\mathrm{kg/m^3}) \times 1{\cdot}01\,(\mathrm{kJ/kg\,^\circ C}) = 36\,\mathrm{kW/^\circ C}$$

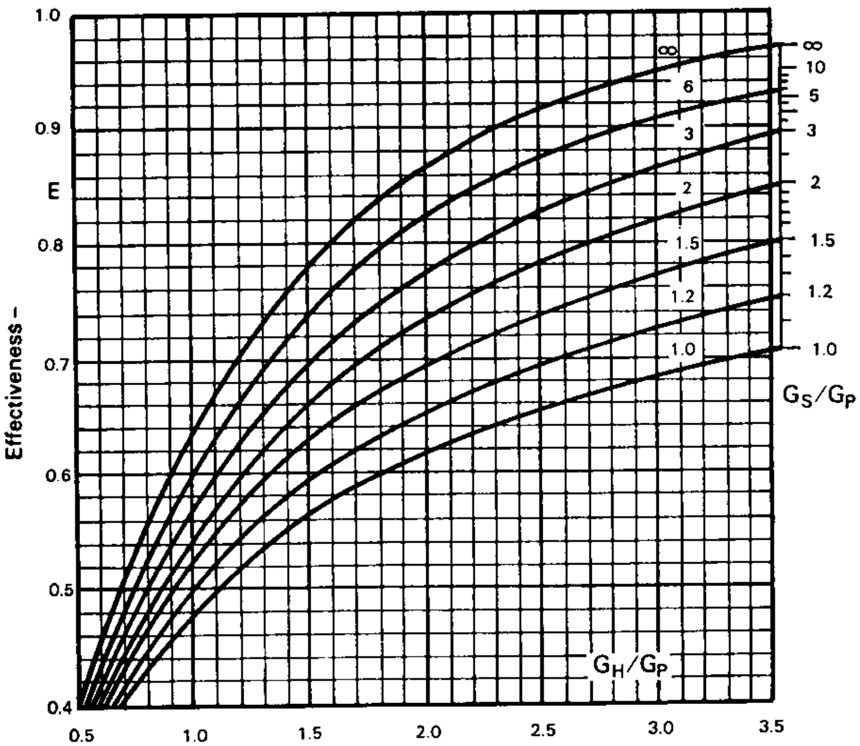
$$\Delta t_p = 110 - 50 = 60\,^\circ\mathrm{C}\ \text{(primary fluid)}$$

$$H = 60\,(^\circ\mathrm{C}) \times 7{\cdot}2\,(\mathrm{kW/^\circ C}) = 432\,\mathrm{kW}$$

$$\Delta t_s = 432\,(\mathrm{kW})/36\,(\mathrm{kW/^\circ C}) = 12\,^\circ\mathrm{C}\ \text{(secondary fluid)}$$

$$\text{Effectiveness E must be } \frac{110-50}{110-35} = 0{\cdot}80$$

Combining this with $G_s / G_P = 36/7.2 = 5.0$ we see from Fig. 12.6 that G_H / G_P should be 2.0. Thus the temperature difference across the exchanger should be $\Delta t_H = At_P \times (G_P / G_H) = 30°C$. This checks with a log mean temperature difference derived on the assumption that the



G_P = Heat capacity of primary flow, kW per °C change.
G_S = Heat capacity of secondary flow, kW per °C change.
G_H = Heat transfer capacity of battery, kW per °C log mean difference.

Fig. 12.6 Heat exchange effectiveness of a cross-flow battery.

air temperature is constant at its mean value of 41 °C - fair enough for a cross-flow exchanger.

Suppose now that a variable pitch fan is used, set to control the volume flow over the range 32 to 12m^3/s in order to maintain the liquid discharge at 50°C irrespective of air temperature. At 12m^3/s G_s will be reduced to 37% of its original value.

$$\text{New } G_s / G_P = 0.37 \times 5.0 = 1.9$$

From the linear chart in Fig. 12.7 a reduction to 37% in G_S produces a reduction to 55% in G_H.

$$\text{New } G_H / G_P = 0.55 \times 2.0 = 1.10$$

Entering these values on Fig. 12.6 we obtain a new effectiveness E = 0.57. In order to cool the liquid from 110°C to 50°C we now need a temperature difference at entry of (110 - 50)/0.57 = 105 °C, i.e. an entering air temperature of 110 - 105 = 5°C.

We can therefore conclude that the variable pitch fan can be used to control the liquid discharge temperature at 50°C while the atmospheric temperature varies over the range 5 °C to 35 °C. The potential for energy saving can be seen from completion of the exercise:

Volume flow	**32**	**26**	**20**	**16**	**12**	**m³/s**
Air temperature	**35°**	**30°**	**25°**	**16°**	**5°**	**C**
Fan power input	**10**	**5·5**	**3·0**	**2·0**	**1·5**	**kW**

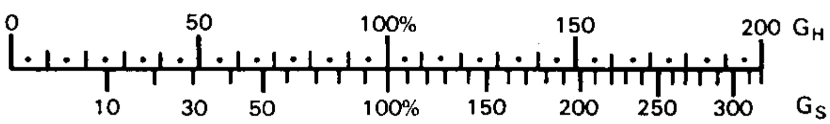


Fig. 12.7 Dependence of G_H on G_S when G_S is a secondary flow of air across tubes.

12.3 Industrial Drying

12.3.1 General principles

Metals apart, almost all the materials used in industry and commerce contain significant amounts of free moisture - that is water which is not in chemical combination. The quantity of this *hygroscopic* moisture varies throughout the life of the material, being in equilibrium with the relative humidity of the atmosphere at levels indicated in Fig. 12.8 for a few natural organic materials. A number of important properties of the material are dependent on the moisture content, notably mechanical strength, handling properties, and electrical and thermal resistance. Also life may be greatly affected through the biological attack of fungus and mould, when damp.

In service, paints, varnishes and impregnants will slow down, though they cannot altogether stop, deterioration, and sensitive materials must be protected from too damp or too dry an environment - particularly one which swings from one extreme to the other. When first produced, however, many natural and some synthetic products contain much too much free moisture. These must be dried to a moisture content which experience has shown to be satisfactory for transport, storage, further processing or marketing, but which may substantially exceed the equilibrium moisture content.

Quality is often greatly influenced by the drying process, and slow, traditional, sun-and-air, methods have tended to set the standards. In engineering applications drying implies the acceleration and control of the natural process to secure a cheaper and more consistent product.

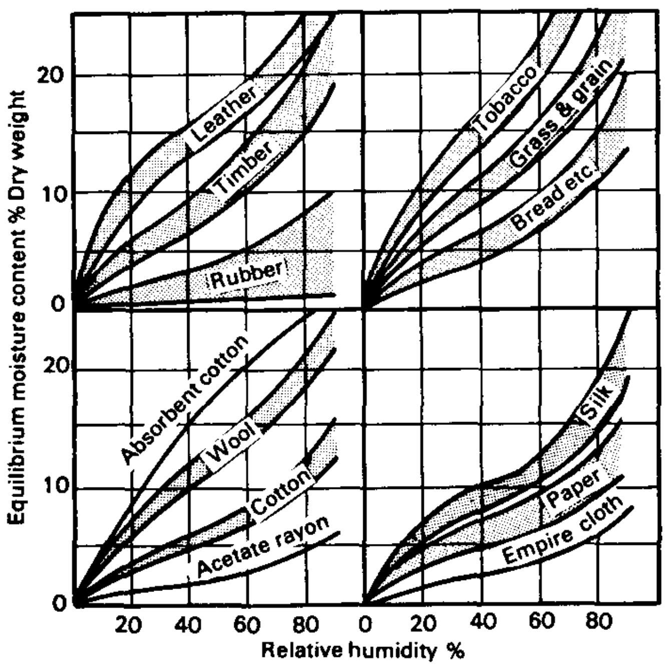


Fig. 12.8 Typical equilibrium moisture contents.

Drying times will vary from minutes to weeks, and rates must be carefully scheduled to suit the product. Too rapid drying causes damage in several ways:

Parts already dried may be overheated.

Over-rapid evaporation may injure the fibres as water forces its way out of the interior.

If the surface layers are allowed to dry out while the interior is still wet, they will shrink and may crack or warp the whole piece.

12.3.2 Moisture content

The weighing of a sample before and after the drying process will clearly give the moisture removed as a percentage of the initial wet weight. To find the dry weight it will be necessary to raise the temperature and allow time for all the hygroscopic moisture to be expelled (but not that which is chemically combined, as in crystalline hydrates). As this complete desiccation may damage the substance, it is likely to be carried out on a small specimen only. In drying calculations it is usually more convenient to express the *moisture content on the basis of dry weight* thus:

$$\frac{\text{weight of moisture present}}{\text{weight of dry stock}} \times 100 - \%$$

It is important always to distinguish this quantity from the moisture content on the basis *of* wet weight which is:

$$\frac{\text{weight of moisture present}}{\text{weight of dry stock + moisture}} \times 100 - \%$$

The relationship between the two is given in Fig. 12.9.

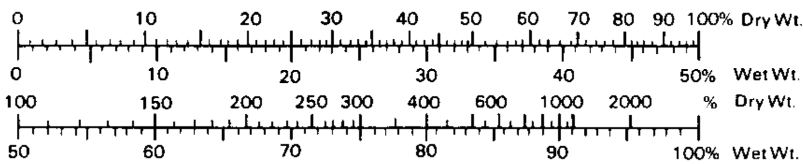


Fig. 12.9 Comparison of moisture content expressions.

12.3.3 Drying processes

Bulk liquid and large amounts of surface water are most economically removed by mechanical means before the stage of drying proper begins. Sedimentation in settling tanks, *filtration*, *drainage by gravity* on mesh support, or by *centrifuging* which is accelerated draining, and mechanical *pressing* between rolls or otherwise are common methods.

Hygroscopic moisture on the other hand can be removed only by evaporation, which is the process we are concerned with in this chapter.

Two things are necessary to promote evaporative drying:

Heat must be supplied-almost as much heat is needed to evaporate a given weight of moisture at room temperature as would be needed to boil it.

The air immediately in contact with the surface is quickly saturated, and must be continually replaced if evaporation is to continue.

Industrial driers may be classified according to the way in which the air is brought into contact with the moist surface and replaced as it saturates.

Drying sheds or lofts are naturally ventilated buildings in which the product is stacked, hung or festooned for comparatively long periods. Air movement may be promoted by circulating fans.

Batch driers are heated ovens in which the product is spread on trays or racks. Sufficient air change must be ensured to prevent saturation, and an internal air circulating system may be incorporated. Air temperature and humidity may be adjusted to a programme covering the drying period.

Conveyer driers are similar in principle to batch driers, but provided with a continuously or intermittently moving conveyer on which wet stock enters at one end and dried stock leaves at the other. Air conditions will be steady with time, but may be adjusted along the length of the drier, particularly by a counter flow system.

Through flow driers are used for a granular or porous product supported on a perforated base so that air may be passed directly through it. Hop kilns and in-sack grain driers are examples.

Rotary driers are rotating drums through which the stock is cascaded or tumbled so that it comes into intimate contact with air flowing through.

Spray driers inject a solution or slurry through nozzles forming a fine spray which evaporates rapidly, leaving the product in powder form.

Heat may be supplied wholly through the air stream, which is preheated before entering the drier. If the drying temperature is high economy depends on a close approach to saturation in the air leaving the drier. At low drying temperatures, although drying times are longer, heating may be dispensed with altogether, particularly in a hot, dry climate. The ambient air itself is then the source of heat, which will flow to the stock by virtue of the low, wet -bulb, temperature of the surface.

Alternatively, the heat may be applied directly to the stock, generally by radiation which is particularly suitable for sheets and similar forms with a high ratio of exposed surface to mass. In special cases heat may be generated within the body of the product, either by resistance heating from an electric current flowing through, or by electro-magnetic induction or dielectric heating in a high frequency field.

12.3.4 The physical mechanism of drying

In the evaporative drying of any product, four main phases may be distinguished.

1. The *rising rate* period during which the heat supplied is used to raise the temperature of the wet stock to the designed maximum, with a correspondingly rising evaporation rate.
2. The *constant rate* period during which temperature is steady, the surface of the stock remains wet, and the evaporation rate is constant, and at its maximum.
3. The *first falling rate* period, which commences at the critical moisture content for the material, when dry patches begin to appear on the surface and the evaporation rate falls.
4. The *second falling rate* period, which is governed by the rate of evaporation, migration and diffusion of moisture within the stock. The surface appears dry, and air humidity no longer influences the rate.

From the point of view of the fan engineer the constant rate period is the significant one. The air flow must remove the moisture generated at a planned humidity level, and the heat required for the evaporation must be supplied, both being at their maximum in this period. Control to lower levels of air flow and heat supply will be appropriate to the falling rate periods.

12.4 Drying Calculations

12.4.1 Heat required for evaporation

During the constant rate period the rate of heat supply to the stock must balance the latent heat of evaporation multiplied by the rate at which water is evaporated. The latent heat in kilojoules (kJ) per kg of water is a function of temperature only and this temperature will be the wet-bulb temperature of the air in contact with the surface. Fig. 12.10 gives this relationship.

Example. At what rate must heat be supplied to evaporate 100 kg per hour of water from a wet surface at 80°C ?

$$\frac{100}{3600}\ (\text{kg/s}) \times 2310\ (\text{kJ/kg}) = 64\ \text{kW}$$

It does not matter whether this heat is applied via the air or directly to the stock. There will, however, be additional heat required in practice to cover:

Heat lost through the walls of the drier.

Heat wasted in so far as the whole of the air does not reach saturation.

Heat needed during the rising rate period to raise the temperature of the stock and its moisture content to 80 °C; this will not have been available for evaporation.

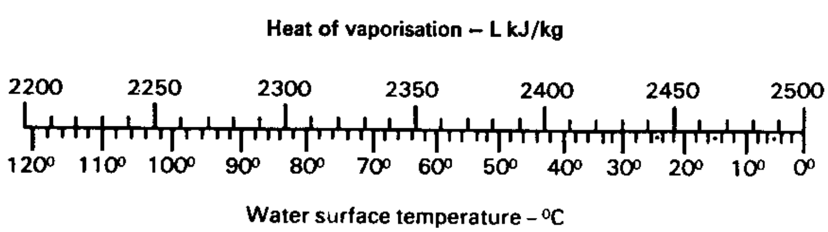


Fig. 12.10 Latent heat or heat of vaporisation for water.

12.4.2 Evaporation rate

The convective heat transfer rate h. W/m^2 °C discussed earlier in this chapter is directly linked with the evaporation rate. It has already been pointed out that the steady state, constant-rate, temperature of the wet surface is equal to the wet-bulb temperature of the air. The temperature difference available for heat transfer is the difference between this and the dry-bulb temperature, that is the wet-bulb depression Δt_w - provided the air is well mixed by turbulent flow. The heat supplied to A m^2 of wet surface is therefore:

$$H\ (kW) = \frac{h_c}{1000}\ (kW/m^2\ °C) \times A\ (m^2) \times \Delta t_w\ (°C)$$

This is now equated to the heat required to evaporate M kg/s of vapour of latent heat L kJ/kg:

$$H\ (kW) = M\ (kg/s) \times L\ (kJ/kg)$$

It follows that the evaporation rate in kg per sec per m^2

$$\frac{M}{A} = \frac{h_c\,\Delta t_w}{1000L} \qquad \textbf{(123)}$$

Example. A bed of wet sand 2m x 2m in area is dried by an air stream of 4m/s with a temperature of 90°C dry-bulb, 35°C wet-bulb. What is the maximum evaporation rate to be expected ?

From Fig. 12.1, with L = 2000mm and v = 4m/s $h_o = h_c = 16$ W/m^2 °C. But this is for a smooth surface; a substantial multiple should be used for a rough sand surface - say 2.5. We therefore take $h_c = 40$ W/m^2 °C.

Working in the more practical units kg/hr, the evaporation rate from the expression just quoted is:

$$M = \frac{3600\ h_c\ A\ \Delta t_w}{1000L}$$

$$= \frac{3{\cdot}6 \times 40\ (W/m^2\ °C) \times 4\ (m^2) \times (90 - 35)\ (°C)}{2420\ (kJ/kg\ at\ 35°C\ from\ Fig.\ 12.10)}$$

$$= 13\ kg/hour$$

12.4.3 Evaporation rate with direct heating

When additional heat is fed directly to the wetted surface by radiation or conduction, the surface temperature will rise above the wet-bulb temperature of the air. This will reduce the rate at which heat is drawn from the air, but will increase the rate of evaporation.

The new rate is dependent on the partial pressure of the water vapour, which is the independent contribution made by the water molecules to the total pressure of the air-vapour mixture.

Two values of vapour pressure are significant, and may be obtained from the relationship with temperature in Fig. 12.11

VPws (Pa) the saturation vapour pressure at the temperature of the wet surface.

VPDP (Pa) the vapour pressure in the main air stream, which is derived from its dew point, that is the temperature at which it would saturate, if cooled.

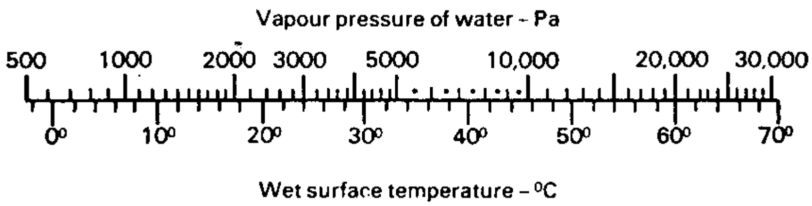


Fig. 12.11 Saturation vapour pressure of water.

The difference measured in Pa between these two vapour pressures is the motive power which drives the water vapour out into the main stream, making room for fresh evaporation to maintain saturation at the wet surface. The situation is analogous to the convective transfer of heat due to the temperature difference between a hot surface and the temperature of the air stream. The same constant, h_cW/m^2 °C, applies, in a new expression for the evaporation rate, in kg per hour:

$$M = \frac{A \times h_c \times (VP_{WS} - VP_{DP})}{45{,}000} \qquad (124)$$

Example. What would be the evaporation rate in the last example if the wet sand surface were independently heated to 50°C in the same air stream ?

From the psychrometric chart, Fig. 12.13, the dew point of air at 90°C dry-bulb and 35°C wet-bulb is 19°C, giving VP_{DP} = 2200 Pa from Fig. 12.11. At 50°C VP_{ws} = 12,300 Pa from Fig. 12.11.

As before h. will be taken as 40 W/m^2 °C - roughness allowance is too uncertain to justify minor correction for K_t.

$$\text{Therefore } M = \frac{4\ (\text{m}^2) \times 40\ (\text{W/m}^2\ °\text{C}) \times (12{,}300 - 2200)\ (\text{Pa})}{45{,}000}$$

$$= 36 \text{ kg/hour}$$

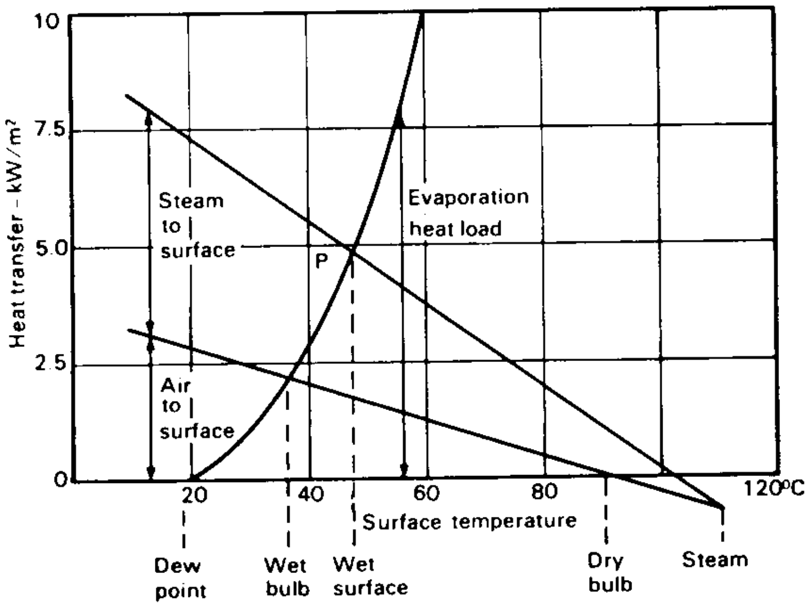


Fig. 12.12 Equilibrium temperature (at P) of a wet surface. (see example 12.4.4)

12.4.4 Estimation of actual wet surface temperature

The, last calculation assumes a surface temperature of 50°C, but generally speaking this will not be known in advance. Organised trial and error is needed as the following treatment indicates.

Suppose the wet sand bed of the previous examples to be supported on a plate heated by steam at 110°C. The heat transfer rate between the steam and the wet surface of the sand is estimated at 50 W/m^2 °C.

The heat required per square metre to evaporate the quantity of moisture just calculated for a surface temperature of 50°C is:

$$\frac{36\ (\text{kg/hr}) \times 2380\ (\text{kJ/kg})}{3600\ (\text{s/hr}) \times 4{\cdot}0\ (\text{m}^2)} = 5{\cdot}9\ \text{kW/m}^2$$

at other surface temperatures the evaporation heat rate will be proportional to (VP_{ws} - 2200) as plotted in Fig. 12.12. Also plotted are the heat transfer rates to the surface from:

the air, which will be zero when the surface is at dry-bulb temperature and increase by h_c = 40 W/m^2 for each °C below;

the steam, which will be zero when the surface is at steam temperature, and increase by 50 W/m^2 for each °C below.

Point P where the total heat transfer equals the evaporation heat rate, will be the probable surface temperature: 47°C.

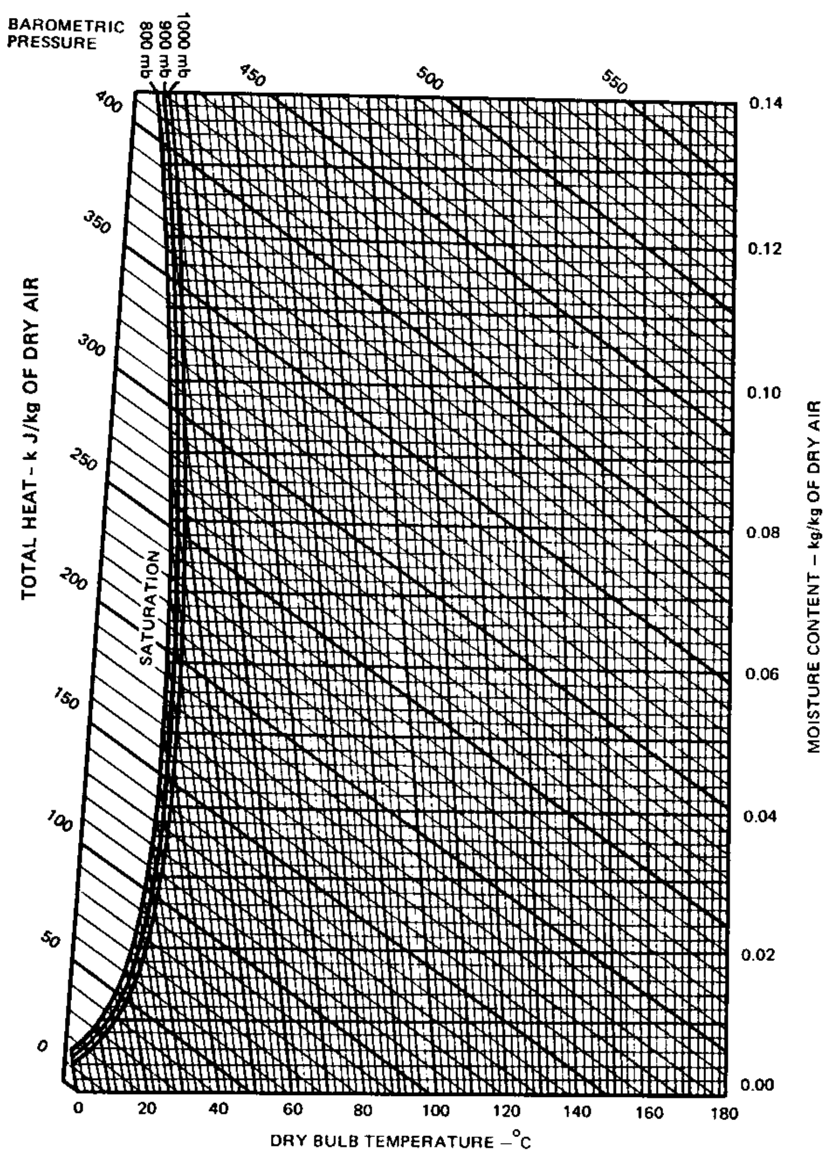
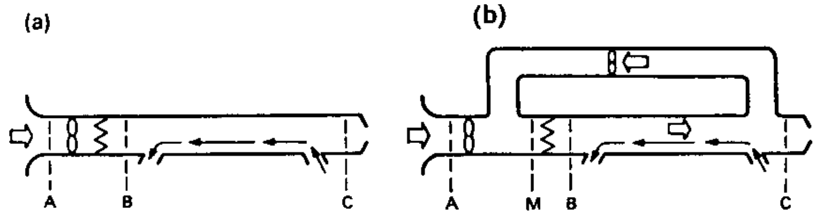
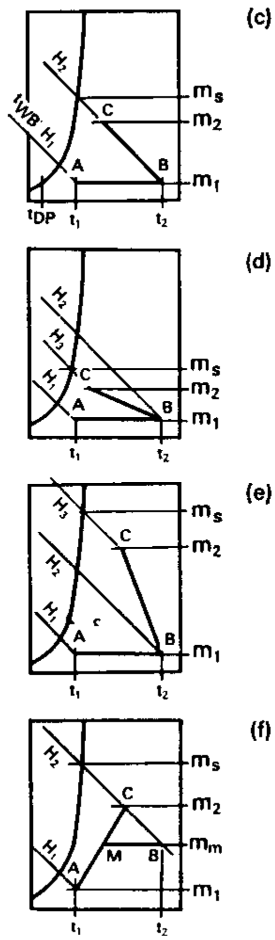


Fig. 12.13 Psychrometric chart for drying.





Figs. (a) and (c) Constant rate period with airborne heat only.

A—Entering air condition
Dry-bulb t_1. Wet-bulb t_{WB}.
Moisture content m_1. Dew point t_{DP}.

B—Air condition after heating
Heat added $H_2 - H_1$. Moisture content m_1.
Dry-bulb t_2.

C—Air condition leaving drier.
Total heat H_2. Evaporative efficiency E.
Evaporative rate $(m_2 - m_1) = E\,(m_s - m_1)$.

Figs. (a) and (d) Allowing for heat lost through drier walls or absorbed by stock during rising rate period.

A and B—as previously.

C—Air condition leaving drier.
Heat lost $H_2 - H_3$. Evaporative efficiency E.
Evaporative rate $(m_2 - m_1) = E\,(m_s - m_1)$.

Figs. (a) and (e) Allowing for heat supplied directly to stock in addition to airborne heat.

A and B—as previously.

C—Air condition leaving drier.
Additional heat $H_3 - H_2$. Evaporative efficiency E.
Evaporation rate $(m_2 - m_1) = E\,(m_s - m_1)$.

Figs. (b) and (f) Operation with air recirculation.

A—as previously. Heat supply $H_2 - H_1$.

$$R = \frac{\text{kg dry air recirculated}}{\text{kg fresh dry air supplied}} = \frac{AM}{MC}$$

Locate B (air condition over stock) from:

$$(m_m - m_1) = \frac{RE}{1 + RE}(m_s - m_1)$$

Locate C from $(m_2 - m_m) = E\,(m_s - m_m)$.
Evaporation rate $= m_2 - m_1$

Note. All heat and moisture quantities are per kg of dry air entering and leaving drier.

Fig. 12.14 Use of psychrometric chart for drying problems.

At P the evaporation heat rate is 4.8 kW per m2 giving a total *of* 19 kW over the 4M2. This corresponds to an evaporation rate *of:*

$$M = \frac{3600 \times 19\ (\text{kW})}{2390\ (\text{kJ/kg at } 47^{\circ}\text{C})} = 29\ \text{kg/hour}$$

12.5 Psychrometric Chart for Drying

Many Psychrometric charts contain a great deal *of* valuable data, but are liable to confuse those unfamiliar with them just because of their rather intimidating completeness. Fig. 12.13 has been deliberately drawn to include only those quantities which are needed for the solution of drying problems.

The key diagram, Fig. 12.14, should be self-explanatory. At each point, A, B, C or M, in a conveyer drying system the first line or two of text gives the information available, the last line or two the information obtained from the chart. All quantities are per kilogram of dry air entering and leaving the drier thus:

H = kJ of total heat per kg of dry air
m = kg of moisture per kg of dry air

they may also be used as flow rates however:

H = kJ/s or kW per kg/s of dry air
m = kg/s evaporation rate per kg/s of dry air
m = kg/hour evaporation rate per kg/hour of dry air

Note that each point defines an air condition at some stage in the drying process, and that:

The condition point moves horizontally to the right as heat is added to the air without change of moisture content.

The condition point moves diagonally up one of the constant H lines as moisture is absorbed by the air with no addition or subtraction of heat. This is very nearly a line of constant wet-bulb temperature.

The evaporative efficiency E is comparable in value to the heat exchange effectiveness of a system of similar geometry. It is called an "efficiency" rather than an *"effectiveness" because* it really does give the fraction of the heat supplied to the air which is used in evaporating moisture, the rest being wasted. At least that would be the situation if the surface temperature of the stock equalled the wet-bulb temperature of the incoming air; and all the heat were delivered to the air with none lost. Heat delivered directly to the stock is used at 100% efficiency.

In an air heated drier *of* economical size the air velocities required to maintain a good drying rate may be so high that an excessive quantity *of* air flows through the drier. This means a low moisture content in the air leaving the drier with a low evaporative efficiency and consequent loss of heat in the excess air.

This difficulty may be overcome by recirculating a substantial proportion of the air flowing over the stock back to a point upstream of the heater. The bigger the ratio R of recirculated air to fresh air the higher the efficiency, though to go too far would reduce the evaporation rate because the circulating air would be saturated.

Similar results arise from simple accelerated circulation within a batch oven.

CHAPTER 13

Ventilation of tunnels and mines

Effective ventilation is an essential requirement for life support in underground workings of all kinds, both in operation and during construction. Toxic and inflammable gases must be diluted to a safe level; dust and smoke must be dispersed; excessive heat should be relieved; and escape routes from fire or other emergency must be kept clear of smoke and flame.

Clearly these vital matters must be the responsibility of a professional consultant in the interests of public safety. Here we can only touch on the salient points of tunnel and mine ventilation technology, but the treatment does serve to introduce the jet fan, together with aspects of system design which are of importance to the fan engineer.

13.1 Pollutants in Road Tunnels

13.1.1 Carbon monoxide

Both the time of exposure and the concentration are significant, the progressive effects being loss of alertness, headache and unconsciousness. In view of the short time the driver is exposed in normal traffic quite high concentrations are tolerated, but there should be provision for a "switch off engines" instruction in the event of a hold up. Typical recommendations in parts of CO per million by volume are:

250 ppm: a common maximum for extreme traffic conditions.

150 ppm: suitable for normal peak traffic flow.

100 ppm :control level for ventilation rate reduction.

50 ppm: maximum average exposure for full-time tunnel staff.

13.1.2 Other products of car exhausts

Toxic oxides of nitrogen and sulphur are also produced by spark ignition engines, the recommended maximum 8-hour day exposures being: nitric oxide, 25 ppm NO; nitrogen dioxide, 5 ppm NO_2; sulphur dioxide, 5 PPM $S0_2$. With present day engine designs these limits will not be reached until the CO content is as high as 500 ppm so that ventilation design may be based on carbon monoxide dilution. Unburnt hydrocarbons in the exhaust are responsible for bad smells and eye irritation, but once again these should be negligible provided the carbon monoxide is adequately controlled.

13.1.3 Diesel engine exhaust smoke

Compression ignition engines produce much less carbon monoxide and other toxic gases, but a successful tunnel ventilation system must limit the unpleasant smoke and smell characteristic of heavy vehicle exhausts. The standard test is of visibility which can be monitored by photo-electric devices calibrated in Dmg/m^3 - milligrams of standard diesel smoke particles per cubic metre of air. This measure is less than the total solid content (mostly unburnt carbon) in the air since large heavy particles play relatively little part in the obscuring effect of the smoke haze.

1 Dmg/m^3 : clear for practical purposes.

2 Dmg/m^3 : 90m visibility at 2 cd/m^2.

4 Dmg/m^3 : 90m visibility at 4 cd/m^2.

8 Dmg/m^3 : foggy ; traffic must move slowly.

90m visibility is satisfactory for 60 km/hr driving speed. 2 and 4 cd/m^2 = are measures of the luminance of the surface seen.

13.1.4 Estimation of pollution load

The total load is clearly equal to the total numbers of cars in the tunnel at any one time multiplied by the average emission per car. The letter is dependent on the speed and gradient and, of course, on the car and its condition. Various national transport authorities have published estimates of average vehicle performance, those deduced by the ASF from actual trials in Swiss road tunnels being summarised in Figs. 13.1 and 13.2. These charts are for sea level, and more pollutant is generated in mountain tunnels where the engine must produce the same powerudren supplied with air of reduced density. Table 13.1 gives the factor by which the sea level value should be multiplied for various altitudes.

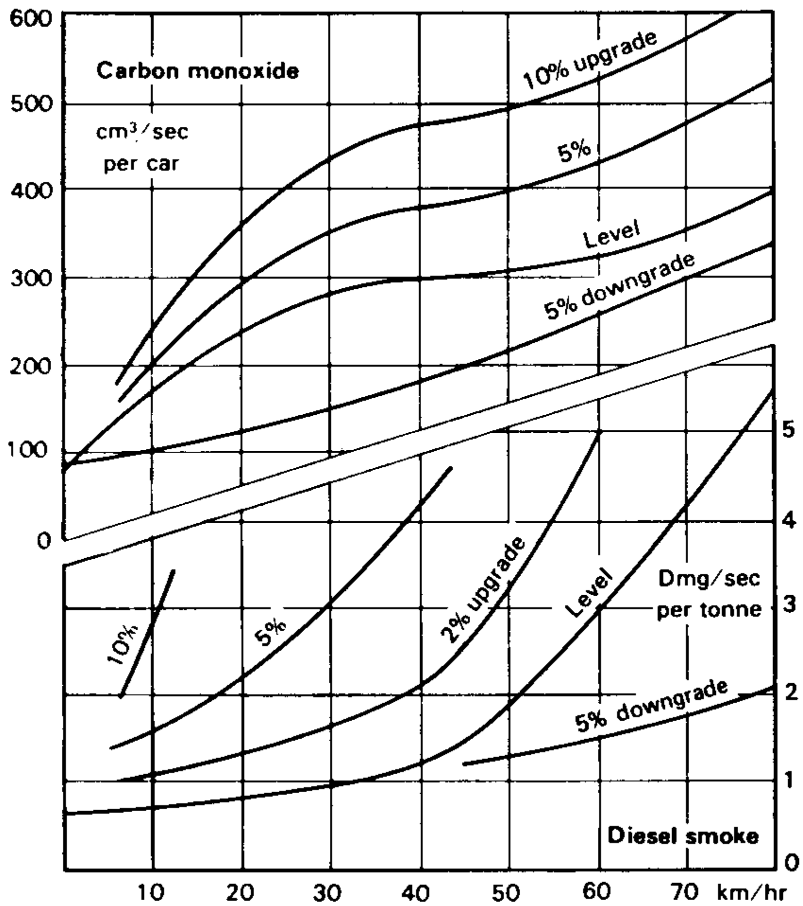


Fig. 13.1 Typical pollution rates at sea level.

Table 13.1
Altitude factors

Altitude—m	0	400	800	1200	1600	2000
Density—kg/m³	1·20	1·15	1·11	1·07	1·03	0·98
Factor for carbon monoxide	1·00	1·25	1·55	1·90	2·30	2·70
Factor for diesel smoke	1·00	1·20	1·45	1·65	1·90	2·10

Traffic statistics will indicate the normal number of cars and lorries to be dealt with, but on trunk roads there will always be occasions when each lane is used to its maximum capacity. Many studies have shown that there is a minimum spacing which the average driver considers safe, and the way in which this varies with speed is indicated in Fig. *13.2* for the case where the traffic is wholly private cars of European size (4m long). It is interesting to note that there is a maximum capacity of the order of 1800 cars per hour per lane which occurs around 60 km/hour with reduction at both lower and higher speeds.

Multiplying together the number of cars per km (Fig. 13.2) by the length of the tunnel the number of traffic lanes and the pollution rate per vehicle gives the total to be dealt with by the ventilation system. This

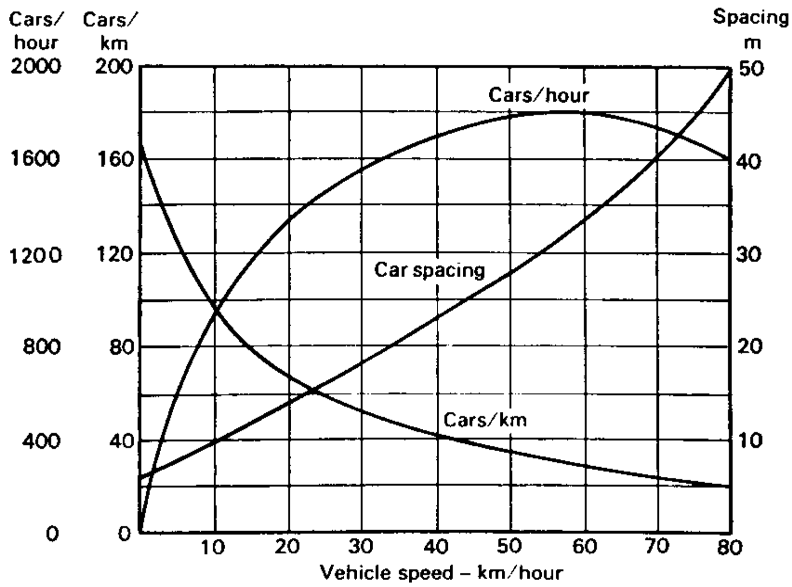


Fig. 13.2 Typical maximum traffic per lane (private cars).

reaches a maximum with congested, crawling traffic in the 10 to 30 km/ hour speed range, and maxima derived from Figs. 13.1 and 13.2 are given in Table 13.2. The importance of avoiding steep gradients in long tunnels is obvious - particularly at high altitudes. For the Diesel smoke load traffic statistics must be relied on to indicate the proportion of heavy lorry or coach vehicles and their average loaded weight.

Table 13.2
Maximum pollution rates with congested traffic

Pollution	Gradient – 5%	level	+ 5%	+ 10%	Altitude
cm^3 CO/sec per metre per lane with all private-car traffic	14	17	21	25	Sea level
	24	30	35	45	1000m
	40	45	55	70	2000m
Dmg/sec per metre per lane with 10% diesel vehicles of 15 tons average weight	0·15	0·15	0·25	0·45	Sea level
	0·15	0·25	0·40	0·70	1000m
	0·20	0·30	0·60	1·00	2000m

13.2 Ventilation of Road Tunnels

13.2.1 Ventilation rate

In a good ventilation system the maximum pollution concentration anywhere in the tunnel is likely to equal the total pollutant generated, divided by the total volume of air exhausted. Since 1 m^3 = $10^6 cm^3$ the concentration of carbon monoxide in parts per million will be:

$$\text{ppm CO} = \frac{\text{cm}^3\ \text{CO/s.m} \times \text{m length, all lanes}}{\text{m}^3\text{/s exhaust rate of polluted air}}$$

and of diesel smoke:

$$\text{Dmg/m}^3 = \frac{\text{Dmg/s.m} \times \text{m length, all lanes}}{\text{m}^3\text{/s exhaust rate of polluted air}}$$

To determine the ventilation rate required it will be necessary to set Inrpet pollution levels for various traffic conditions which might be, for example

250 ppm CO: congested, crawling, flow; all private cars.

100-150 ppm CO: expected maximum flow at 40-60 km/hr; all private cars.

2-3 Dmg/m3: congested flow with maximum forecast proportion of heavy goods vehicles and buses.

1-171 Dmg/m3: expected maximum tonnage flow of diesel vehicles at 40-60 km/hr.

Each of these targets will set a separate value for the ventilation rate, and the largest of these will be the m^3/s peak capacity to be installed, plus an allowance for plant out of service for maintenance, etc.

13.2.2 Sectionalising of tunnel length

Ventilation cost is greatly influenced by the section length between access points at which fresh air may be supplied and polluted air exhausted. Thus, for a given traffic flow and cross-sectional area of tunnel and ducts:

Pollution rate is proportional to: (length)

Ventilation rate is proportional to: (length)

Pressure gradient is proportional to: $(\text{length})^2$

Pressure drop is proportional to: $(\text{length})^3$

Power required is proportional to: $(\text{length})^4$

Doubling the section length thus multiplies the fan power required by 1111. Most long tunnels have ventilation sections of the order of 1 km long where they can be conveniently provided, but with tunnels under wide rivers, straits or high mountains this is difficult and costly. If such tunnels are very long it is usual to limit the maximum traffic density by control at entry. Minimum speed and vehicle spacing may also be controlled.

Even with strict capacity limitation large powers may be involved Thus, the 16 km long tunnel now under construction to replace the St. Gotthard Pass between Switzerland and Italy has one 5.5 km long ventilating section and the installed fan power will be over 20,000kw. Duty control to relate the ventilation rate to the pollution level produced by the number of vehicles actually in the tunnel can reduce the average power to a fraction of the installed capacity - see the example of the Lion Rock tunnel at Hong Kong described in Section 9.1.5, with an average measured power about one-eighth of capacity.

13.2.3 Ventilation systems

Fig. 13.3 illustrates a number of ways in which air can be supplied and exhausted in a road tunnel. To facilitate comparison of air flow and pollution levels each system, a to f, is applied to a common ventilation problem

Tunnel: length 1 km; area $40m^2$; two-lane; one-way.

Target pollution level: 100 ppm CO.

Cars in tunnel: 80 at 40 km/hr or 300 stationary.

CO generation: 80 cars at $300Cm^3/s$ or 300 cars at $80cm^3/s$, either producing 24,OOO cm^3/s CO and requiring $240m^3/s$ total air supply and extraction for 100 ppm CO.

The diagrams under the heading "arrangement" indicate the tunnel with traffic flow from left to right, together, where appropriate, with air supply and exhaust ducts. These are branched at frequent intervals to supply and exhaust ports situated at road and roof level respectively in the tunnel walls. Arrows coded according to volume flow indentify the points where air enters and leaves the system.

The "tunnel velocity" curves show the longitudinal air velocity averaged over the 40M2 cross-section, the full line applying to the stationary car case. When the cars are moving they will induce air to flow with them along the tunnel. The broken line shows the velocity resulting from an increase from $240m^3/s$ to $320m^3/s$ in system (a) or an induced flow of $160m^3/s$ in the remainder, where there is no net fan-induced flow.

Finally, the variation of "pollution level" along the tunnel is shown again in full line for the stationary and broken line for the moving cg cases.

13.2.4 Longitudinal ventilation

Longitudinal ventilation is a system without ducts in which the whale required volume flow moves through the tunnel at constant velocity from one end to the other. In tunnels up to 200m or 250m long *natural ventilation* is often relied on, the air movement being induced by natural wind and thermal forces supplemented by traffic drag. These forces may

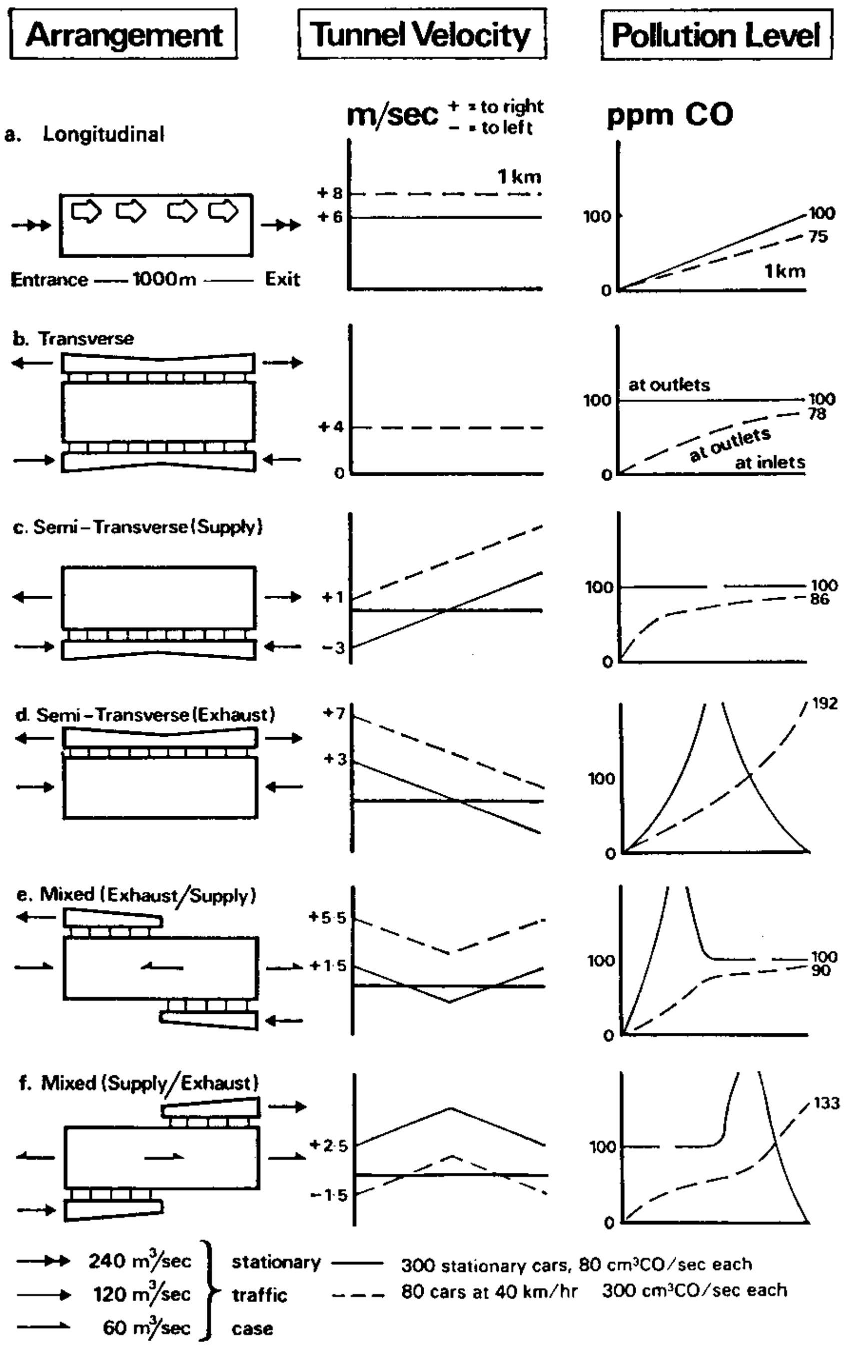
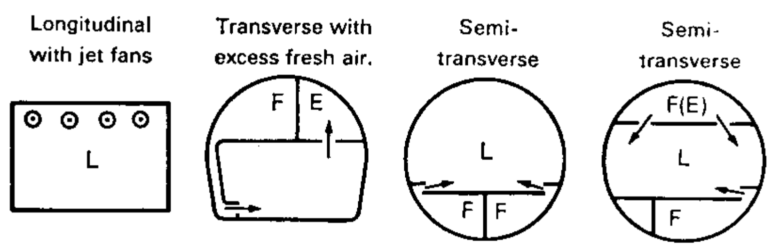


Fig. 13.3 Road tunnel ventilation systems.

always become momentarily equal and opposite resulting in ventilation failure, but if there is much traffic (which must be one-way) and therefore appreciable pollution the drag forces are likely to prevail.

In tunnels of greater length, or subject to rush-hour congestion, axial jet fans are used to provide positive longitudinal ventilation. The principle of the jet fan is discussed in Section 13.6; in practice they are mounted in the space between the maximum traffic height gauge and the tunnel roof, repeating at intervals of 100 metres or more. They blow in the same direction as the traffic (normally one-way) though they are capable of reversal in emergency. They are particularly well suited to cut-and-cover tunnels with flat roofs; a line of axial fans in parallel across the whole width of the tunnel will occupy minimum headroom and provide the necessary thrust with minimum power.

The fact that the tunnel itself is used as the airway gives the longitudinal system notable economy both in first cost and energy consumption. However, the volume flow and air velocity must increase in proportion to the length. which is therefore limited to about 1 km for roads with heavy traffic, corresponding to an average air velocity around 8m/s. As Fig. 13.3 shows, the average pollution level is low, and the brisk turbulent flow in the tunnel eliminates high local concentrations of exhaust fumes.



F: Fresh air supply airway. E: exhaust airway. L: longitudinal flow.
F(E): Fresh air supply, reversible for emergency extraction.

Fig. 13.4 Examples of road tunnel cross-sections.

13.2.5 Transverse ventilation

Transverse ventilation is a fully ducted system with a fresh air supply duct and a polluted air extract duct each running the full length of the tunnel. Closely spaced inlet jets (e.g. every 5m) connect the supply duct with the tunnel, usually at roadway level to limit the formation of stagnant pockets with high exhaust gas content if the traffic is stalled. Similarly spaced outlets connect to the exhaust duct at ceiling level, so that there is a continuous slow upward drift of fresh air across the tunnel.

Both supply and exhaust ducts are generally fitted with fans at both ends of each ventilating section with obvious saving of duct cost and energy consumption - half the volume flow to be conveyed half the distance. Transverse systems are likely to be highest in both first and running cost, though they are virtually essential for very long tunnels, since they can be readily sectionalised through supply and exhaust access shafts. Some consulting engineers also consider them best at restricting the spread of a fire, and extracting the smoke. This may imply supply fans reversible to extract in an emergency, and capable of continuing to function for 15 to 30 minutes in very hot gases.

Large, high efficiency, fans are likely to be needed, installed in plant rooms with the duty and layout generally favouring the axial type. Variable pitch or other control means are important to save energy by matching ventilating rate to need. In urban surroundings towers may be needed, both to obtain fresh unpolluted air, and to dissipate the exhaust gases by chimney action so that they will not constitute a nuisance to neighbouring buildings.

13.2.8 Semi-transverse ventilation

Semi-transverse ventilation uses the same supply duct as a transverse system, but the outgoing air travels along the tunnel as in the longitudinal system. If the traffic is not moving or if it is two-way, the outflow will be equally towards each end of the tunnel, and in the middle there will be no longitudinal flow. At this point-indicated by a break in the 100 ppm CO line in Fig. 13.3c - the supply jets must be relied on to stir up any exhaust gas pockets, which will then drift, adequately and uniformly diluted, towards the ends of the tunnel.

The corresponding pollution level diagram Fig. 13.3d shows the unsatisfactory result of operating a semi-transverse system with an exhaust, rather than a supply, duct. There is again a point of no longitudinal flow, at the centre with stationary traffic, and displaced towards the exit end as the traffic speeds up. This time, however, instead of fresh air flowing away from the null point, polluted air will be flowing towards it, resulting in a very high local level. Furthermore, the exhaust ports produce no jets so that local mixing is not promoted.

Semi-transverse systems are intermediate in cost and are most frequently found for tunnels in the 1 to 3 km range. Where the tunnel has a circular cross-section the supply duct can be conveniently accommodated under the roadway. Mixed systems are sometimes used with a supply duct along one half of the tunnel length and an exhaust duct along the other. As Fig. 13.3e shows it is better to have the supply duct at the exit end if there is one-way traffic. Both semi-transverse and mixed systems are often sectionalised with reversible fans so that it is possible to direct smoke and flames from an accident away from traffic trapped on one side.

13.3 Ventilation of Rail Tunnels

13.3.1 Main line tunnels

Main line tunnels hardly ever require mechanical ventilation. Whereas the drag of one-way traffic induces air velocities in the 4 to 8m/s range in road tunnels, the frontal area of such vehicles is usually 5% to 10% of the tunnel cross-section. Railway rolling stock, on the other hand, runs from 20% to 30% (double track) to 50% to 60% (single track) of tunnel area; this introduces a "piston effect" which further increases the propulsive force, providing adequate ventilation by displacing air from entry to exit portal.

The tunnel should nevertheless be sectionalised by means of intermediate vent shafts. By limiting the volume of air to be moved these reduce the pressure required and the corresponding extra tractive effort. It is also important to limit the sudden rise or fall of pressure, within the train, proportionate to the drag force, which occurs when the train enters or leaves the tunnel or passes a vent shaft. A change of 3% in absolute pressure (± 3000 Pa) is unpleasant to the passengers, and may harm the ear of someone whose pressure equalising mechanism (the Eustachian tube) is partially blocked, or who has some other special physiological sensitivity.

13.3.2 Underground railways

Underground railways forming part of an urban "rapid transit" system involve long tunnels, have a far higher density of traffic, and have frequent underground station platforms. The comfort of the passengers requires better air conditions on the platforms than are necessary in the tunnels, and excessive air velocities in the access subways should be avoided. The heat generated underground will equal practically the whole of the (normally electrical) energy supplied to the rolling stock and services. Bearing in mind the repeated acceleration and braking this is very large, and cannot be absorbed by the earth around the tunnel without building up an excessive temperature.

Consequently a sufficient volume of air must pass through all the underground workings to carry away the whole of the heat generated with a limited rise of temperature. The air drawn in will be at the atmospheric temperature on the surface, so that climate and season have a strong influence on the volume flow required. Taking the specific heat of air as 1 kJ/kg °C

$$\mathrm{m^3/s} = \frac{\text{Heat load (kW)}}{\text{Temperature rise (°C)} \times \text{Air density (kg/m}^3\text{)}}$$

Fans will be used both to supplement the air movement produced by the trains, and to control the distribution of supply and exhaust between the stations, the tunnels, and the plant areas. Extraction at the centres of tunnel sections is usual, with supply and pressure relief vents near the ends, controllable so that cooling air can be drawn or blown into the stations in hot weather, limiting the platform temperature to, say, 3°C above the outside level. Higher exhaust temperatures may be permissible elsewhere; for example there may be fan-powered extraction points under station platforms to remove heat stored in the brakes with exhaust air well above platform temperature.

The fan powers are large so that energy economy calls for a sophisticated control system. This will operate through damper and fan duty control, secured by on-off switching, variable-pitch or speed control as discussed in Chapter *9.* Reliable estimates of flow resistance are also necessary, obtained by the methods of Chapter 6, supplemented by the treatment of friction drop in long airways to be found later in this chapter.

13.4 Mine Ventilation

13.4.1 Ventilation requirements

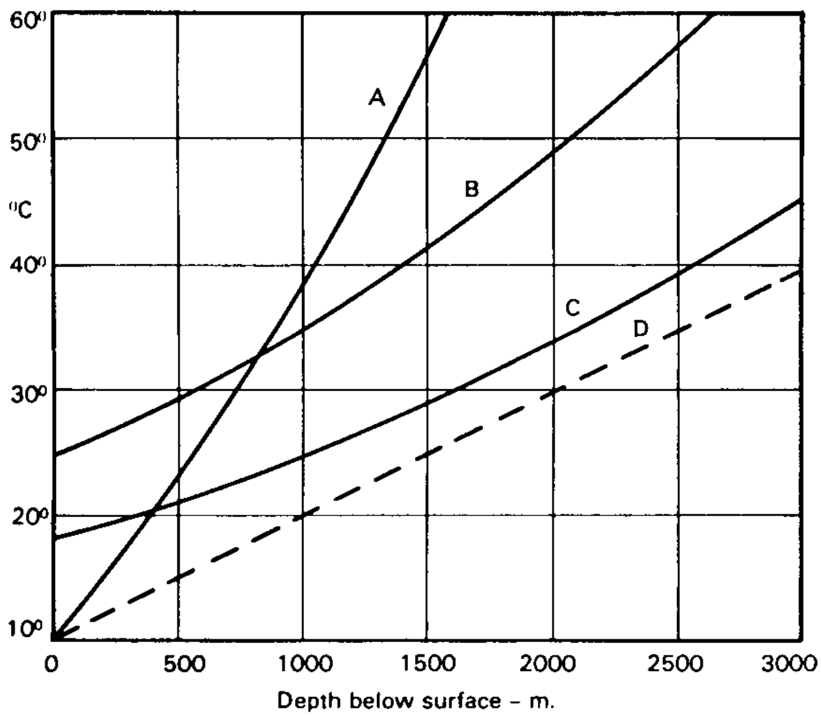
The first responsibility of the mine ventilation engineer is to maintain at all times, including emergencies, the air needed for life-support underground. With this requirement satisfied he will want to improve the quality of the air in the workings to limit the stress imposed on the miners by an inevitably hostile environment. Dust suppression is important because of the danger of respiratory diseases grouped under the general heading of pneumoconiosis. Finally, some mines, notably British coal mines, are "fiery", introducing the need to limit the concentration of gases such as methane below the flammable or explosive level. In most countries these requirements are specified as precise statutory obligations.

In deep mine workings heat and humidity are the principal enemies of good working conditions. Even in dry strata humidity is high when water is used to lay the dust at the working face. Heat will arise from the energy supplied to the mining machinery and also from two natural sources:

Heat generated in the earth's core by radioactive processes flows to all parts of the surface where it is dissipated by radiation. There is, therefore, a *geothermic temperature gradient,* determined by geological factors, which makes the deep rock temperature higher than the ground temperature at the surface. Three examples are given in Fig. 13.5; the British coal-bearing strata have a particularly low thermal conductivity (unless they are water-bearing) leading to the high geothermic gradient shown.

As air flows down a mine shaft it is gradually compressed by the increasing weight of atmosphere above it. As we saw in Section 11.3.10 compression heats the air, and the *adiabatic temperature gradient* isolates this effect by ignoring any other heat given up or acquired by the air during its descent. The broken line in Fig. 13.5 shows this gradient, starting from a typical UK surface temperature. When the air returns to the surface its adiabatic temperature will follow this line in reverse, there being in fact an average adiabatic temperature appropriate to each depth below sea level.

When the rock underground is wet, or when water is used for dust suppression, there will be evaporation which will substantially lower the dry-bulb temperature of the air. This will not improve the working conconditions much, however, the heat stress being primarily dependent on the wet-bulb temperature. In Europe 28°C WB is regarded as the maximum for effective working, but in the deep mines of South Africa selected and acclimatised miners can operate at over 30°C WB. Air movement at 1 to 2m/s is helpful if the dry-bulb temperature is not too high, as shown in Section 1.8.



A : UK. Lancashire coalfield.
B : India. Kilar goldfield.
C : South Africa. Witwatersrand goldfield.
D : Adiabatic air temperature.

Fig. 13.5 Variation of temperature below ground.

As an example of heat balance an investigation may be quoted from the Nottinghamshire coalfield, U.K. The mine was 850m deep with workings extending 5 km from the main shafts and the air extracted was 150m^3/s. Heat flow rates are averaged over a 24 hour period.

Heat flow from strata	5500 kW
Heat from electrical equipment	1100 kW
Other underground heat sources	450 kW
Total heat load:	7050 kW

This was balanced by the following heat absorbed by means of the ventilation system:

Raising air temperature from 0°C to 19°C	3350 kW
Evaporating 5300 kg/hour of water	3700 kW
	7050 kW

13.4.2 Main mine fans

The main fans are large machines, ranging from a few hundred to as much as 5000 kW. They run for 24 hours a day so that efficiency is important - 85% or so would be expected. Both axial and backwardcurved centrifugal types come into consideration with lower pressures favouring the former and higher pressures the latter. Emergency reversibility is often called for to deal with an underground fire; reversal of rotation suffices for an axial fan, whereas a centrifugal requires the installation of by-pass ducts and dampers. Periodic adjustment of fan volume flow and pressure as the workings are developed and expanded is another requirement, usually met by pitch-angle adjustment, inlet vane angle adjustment, or speed change.

Main mine fans are nearly always installed on the surface, for reasons of cost and ease of access for maintenance and control. The majority are used as exhaust fans at the top of an *upcast shaft so* that the whole mine is under suction. This is safer in a gassy mine since ventilation failure would cause a rise in underground pressure, inhibiting rather than encouraging the outflow of gas from the rock face. The shaft for access and mineral winding, for which the fresh air side is preferred, may then provide the air intake, and need not be air-locked as is necessary if the supply fan is at the top of the *downcast shaft.*

13.4.3 Booster fans

The initial plan of a mine embodies a network of intake airways leading from the downcast shaft to the working faces and return airways leading to the upcast shaft. The intake airways commonly serve as haulage roadways as well. As the mine develops, following the coal seams or other strata to be worked new loops must be added. When there is insufficient pressure to circulate the required air volume an underground *booster fan* is added, handling the whole volume in the loop and topping up the pressure as required. These are normally axial fans, of a few hundred kW rating, which are naturally "in line" machines and will develop the necessary pressure in a single stage. When the airway is also a roadway the booster fan must be in a by-pass passage with an air-lock in the roadway to prevent short-circuiting.

13.4.4 Auxiliary fans

Working faces may be advanced a considerable distance without a return airway. A duct of 400 to 800mm diameter is used to supply fresh air to the face, which then returns along the roadway/airway itself. The fan, invariably of the axial type, is fitted at the inlet to this duct, adjacent to the fresh air main roadway. It is portable, from 5 to 50 kW rating, and is slung from the roof of the roadway. The ducts are also slung from the roof, and must be readily extensible as the work progresses. When the length is great (100 to 1000m) every precaution must be taken against leakage and excessive friction, otherwise the fan power will become prohibitive. A flexible PVC design, with a helical steel reinforcing wire attached externally, is good in both these respects.

Contra-rotating axial fans of the same diameter as the duct and with each impeller mounted on the shaft of its own 2-pole electric motor, are particularly suited to auxiliary fan duty. They may be stocked in 2-stage units, and built up to 4- or 6-stages as required by the growing length of the airway. In coal-mining all stages are installed at inlet, building up to a pressure of several thousand Pa. In some circumstances, however, it may be preferable to string them at intervals along the duct, thereby limiting the pressure rise above the airway - though this should never be allowed to become negative, with possible ingress of contaminated air.

13.4.5 Tunnel drivage fans

A ventilating system similar to the auxiliary mines fan and duct is used during the construction of road and rail tunnels, and in the initial driving of mine shafts and roadways. As well as providing heat removal and ventilation for the workers, the air stream must serve to disperse the dust and fumes arising from the explosive charges used to split the rock.

Increasing the volume flow will shorten the time lost in shot-firing and may enable longer shifts to be worked. Axial fans with high tip speeds, e.g. 800mm diameter at 59 rev/s (3540 rev/min) may therefore be used. Even in the conditions of a construction site it is advisable to fit such fans with silencers, which can take the simple form of a metre or two of acoustically lined duct. Silencers are also increasingly employed for much less noisy booster and auxiliary fans in mines; mine roadways can be very reverberant, and the value of good working conditions is becoming more widely recognised.

13.5 Estimation of Pressure Requirements

13.5.1 Items of pressure loss

The high volume flows and substantial power requirements of tunnel ventilation make it important that each element in the airway system should be designed for minimum pressure drop compatible with reasonable first cost. The fan duty will be determined by the summation of items from a list such as the following:

Inlet from atmosphere: loss can be made small by good entry design.

Outlet to atmosphere: loss = $\frac{1}{2}\rho$ (outlet velocity m/s)^2 - Pa.

Bends, diffusers, screens and louvres: see Chapter 6.

Acoustic treatment and silencers: see proprietary data.

Intake and discharge registers and dampers: a good part of the pressure drop will arise from the restrictions necessary to equalise flow.

Duct and roadway friction: see Sections 13.5.2. and 13.5.4.

Traffic drag and obstructions: see Section 13.5.5.

Influence of duct leakage: see Section 13.5.6.

Wind pressure: tunnel portals in approach cuttings on level ground will be insensitive to wind pressure provided they are clear of the pressure fields surrounding large buildings. In hilly country the wind can have a substantial influence on the longitudinal flow velocity in a tunnel. Some Japanese tests on a tunnel piercing a mountain range showed a pressure difference as high as 1.6 times the velocity pressure corresponding to the wind speed quoted in meteorological reports.

Thermal pressure difference: if the average temperature of the air in a tunnel is Δt °C above that outside, and if there is a difference of altitude h metres between the tunnel portals, then there will be an upwardly directed pressure difference of:

$$\rho_1 gh \frac{\Delta t}{273 + t_1} \text{ Pa} \qquad (131)$$

where t_1 (°C) and ρ_1 (kg/m^3) are the temperature and density at the lower portal and g = 9.81 m/s^2.

Barometric pressure difference: differences due to changes in altitude along the tunnel are self-cancelling. Barometric pressure gradients represented by the closeness of the isobars on a meteorological chart are not significant-hurricane conditions apart.

13.5.2 Friction drop in long airways

Fig. 6.17 gave the pressure gradient in a duct of normal metal construction with sufficient accuracy for most purposes in air conditioning and process work. Airways in tunnels and mines are generally formed in situ by civil engineering methods. Furthermore, they absorb the greater part of the overall pressure required, so that the estimation should be as realistic as possible.

Fig. 13.6 is a version of the well-known Moody chart summarising the researches and formulations of Prandtl, van Karman, Nikuradse and Colebrook. f is the *friction factor* relating the pressure drop, Δp Pa, to the velocity pressure, $\frac{1}{2}\rho v^2$ Pa, corresponding to the average velocity in the airway, v m/s, and the air density, ρ kg/m^3, and to the length, Lm, and diameter, Dm, of the airway if cylindrical.

$$\Delta p = f \frac{L}{D} \tfrac{1}{2}\rho v^2 \text{ Pa} \qquad (132)$$

If the airway is not cylindrical, D is replaced by the hydraulic diameter D_h defined by:

$$D_h = \frac{2ab}{a + b} \text{ for a rectangular airway, } a \times b$$

$$D_h = \frac{4 \text{ (cross-sectional area} - \text{m}^2)}{\text{(length of section periphery} - \text{m)}} \text{ in general}$$

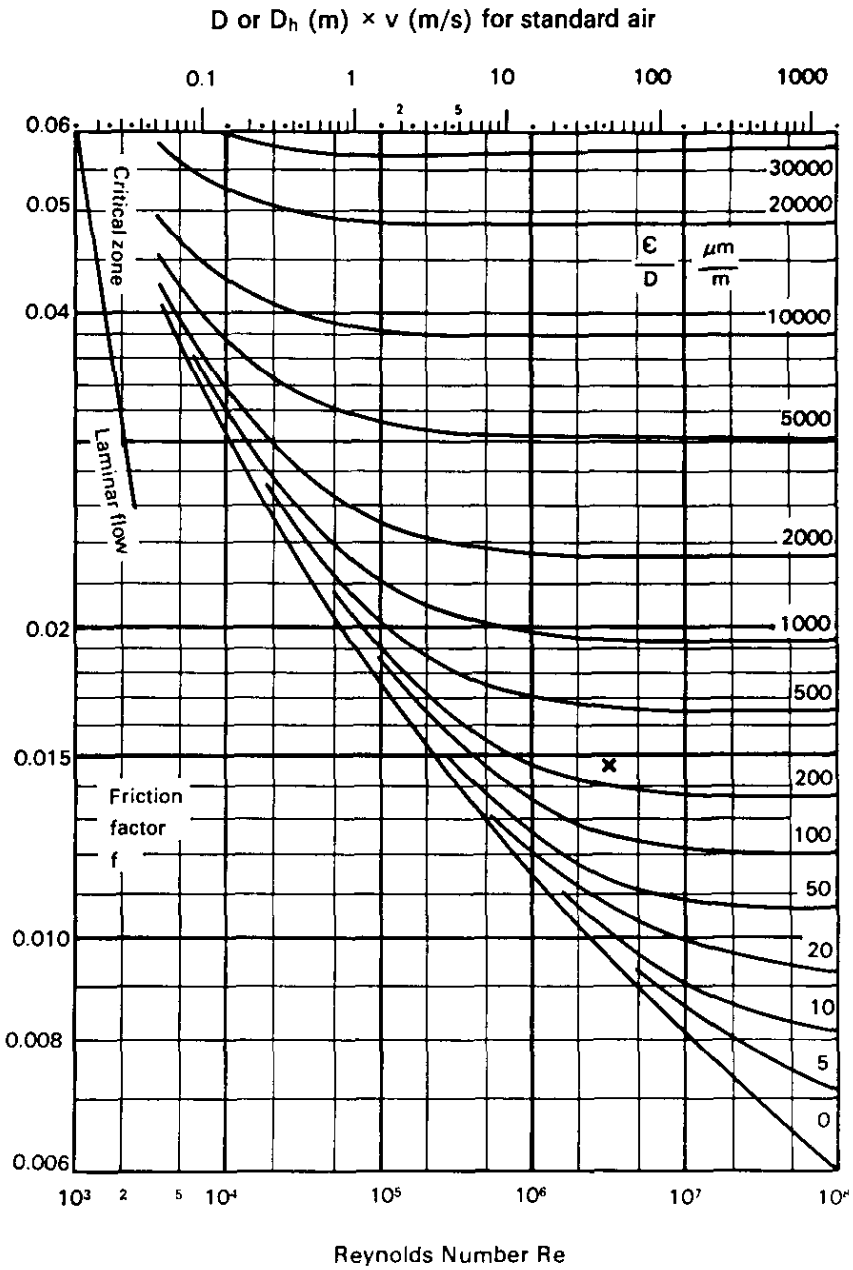


Fig. 13.6 Friction factors in rough airways.

It will be seen that f varies over quite a range according to the values of the two quantities:

Reynolds number—Re
Roughness ratio—ϵ/D

Re is a dimensionless number representing the ratio of the inertial to the viscous forces. The shear forces in the boundary layers near the walls are viscous, and cause the drag of the air on the airway walls which results in the pressure drop. With air (or any gas) of viscosity μPas.

$$Re = \frac{Dv\rho}{\mu} \text{ for a cylindrical duct, diameter D}$$

$$Re = \frac{D_h v\rho}{u} \text{ for the general case} \qquad (133)$$

We cannot expect to get closer than f 10% or so in our estimation of f, and consequently a rough approximation to Re is quite sufficient. When the fluid flowing is atmospheric air it is good enough to take ρ as 1.2 kg/m^3 except perhaps at high altitude, and μ as 18 x 10^{-6} Pa s.

Then:

$$Re = \frac{Dv}{15} \times 10^6$$

This approximation is used to construct the scale at the top of Fig. 13.6, so that the chart may be entered directly with the product of D or D_h in metres and v in m/s.

ε is the *equivalent roughness* of the airway walls here measured in micrometres, μm (1000μm = 1 mm). It is not necessarily the same as the surface roughness of the material used, because account is taken of imperfectly fitted joints and similar factors representative of normal installation practice. Fig. 13.6 shows that, for any given value of rough-

Table 13.3
Equivalent roughness

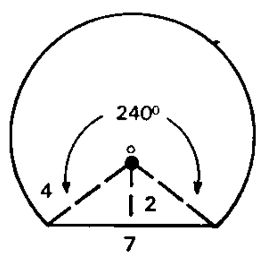
Material, etc.	ϵ – μm
Plastic and non-ferrous drawn tubing	1·5
Asbestos cement piping	13
Black steel piping	46
Aluminium ducting	50
Spirally wound galvanised ducting	75
Galvanised steel jointed ductwork	150
Cast iron pipes	200
Cement- or plaster-faced airways	250
Welded and painted ductwork	500
Fair-faced brickwork or concrete	1300
Rough brick or concrete	5000
Lined *in situ* ducts for road tunnels	400–1000
Road tunnels designed for longitudinal flow	1400–2000
Airways not free from obstructions such as transverse ribs and fillets, or pipes and pipe hangers	5000–20000

ness ε, and diameter, D metres, the friction factor falls from a value close to that for a completely smooth duct ($\varepsilon = 0$), through a transitional curve to a constant value as v and Re increase.

Values of ε quoted by various authorities as representative of ducts made from particular materials are quoted in Table 13.3. Also quoted are ranges corresponding to experimental values of pressure drop reported for a number of actual road tunnels, mainly in the European Alpine area.

The formulae and the chart are quite general and apply to any duct, pipe or airway of any (constant) cross-section. Also to any gas or liquid provided the appropriate values of ρ and μ (see Chapter 14) are used. No practical airflow absorbing enough power to be worth consideration will fall into the uncertain area marked *critical flow* but viscous liquid flows may. At some Re value between 3500 and 2000 the f value for such a flow will fall to the *laminar flow* line where turbulence is entirely suppressed and f = 64/Re.

13.5.3 Example of airway friction calculation



A road tunnel 1 km long of the crosssection illustrated is to carry 300m^3/s longitudinal ventilation. What will the system pressure drop be?

$$\text{Area} = \frac{240°}{360°} \times \pi \times (4\text{m})^2 + \tfrac{1}{2} \times 2\text{m} \times 7\text{m} = 33{\cdot}6 + 7{\cdot}0 = 40{\cdot}6\text{m}^2$$

$$\text{Periphery} = \frac{240°}{360°} \times 2\pi \times 4\text{m} + 7\text{m} = 16{\cdot}8 + 7 = 23{\cdot}8\text{m}$$

$$\text{Hydraulic diameter, } D_h = \frac{4 \times 40{\cdot}6}{23{\cdot}8} = 6{\cdot}8\text{m}$$

$$\text{Mean velocity, } v = 300\text{m}^3/\text{s}/40{\cdot}6\text{m}^2 = 7{\cdot}4\text{m/s}$$

$$\text{Reynolds number, Re} = \frac{6{\cdot}8\text{m} \times 7{\cdot}4\text{m/s}}{15 \times 10^{-6}\,\text{m}^2/\text{s}} = 3{\cdot}3 \times 10^6$$

Assume roughness, $\epsilon = 1700\mu\text{m}$ $\quad \epsilon/D_h = 1700/6{\cdot}8 = 250$

Reading these into Chart 13.6 at the point X: f = 0·015

(to estimate at 0·0147 would be to overstate the accuracy of the data)

$$\text{Pressure drop } \Delta p = 0{\cdot}015 \times \frac{1000\text{m}}{6{\cdot}8\text{m}} \times \tfrac{1}{2} \times 1{\cdot}2\,\text{kg/m}^3 \times (7{\cdot}4\text{m/s})^2$$

$$= 72\text{ Pa}$$

Outlet loss = $\frac{1}{2}\rho v^2$ = ½ x 1.2 kg/m^3 x (7.4m/S)^2 = 33 Pa

System loss (ignoring wind pressure) = 72 + 33 = 105 Pa

That is K = 32 in $\Delta p = K.\ \frac{1}{2}\rho v^2$

13.5.4 Airways with distributed intake or discharge

In the transverse and semi-transverse ventilation systems the ducts will be connected to the roadway through a series of ports which are usually equally spaced and adjusted to admit or extract equal volume flows of air. Both duct and roadway will thus experience a gradual rise or fall in volume flow and therefore in velocity and pressure gradient.

If the airway is of constant cross-section throughout it can be shown that the pressure required to overcome the airway friction will be reduced to approximately one-third of the pressure that would be required if the whole volume flow travelled the full length of the airway:

$$\Delta p = \tfrac{1}{3}.f.\frac{L}{D}\ \tfrac{1}{2}\rho v^2$$

Supply and exhaust ducts are sometimes stepped down in size as they progress from the fan end to the far end. In any one section with mean velocity v, at the end nearer the fan and v_2 at the other end the pressure loss will be approximately:

$$\Delta p = \tfrac{1}{3}.f.\frac{L}{D}\ \tfrac{1}{2}\rho v_1^2 \left[1 + \frac{v_2}{v_1} + \left(\frac{v_2}{v_1}\right)^2\right] \qquad (134)$$

In addition to duct friction there is bound to be some "mixing" loss as the air passing through each port joins or is extracted from the main stream. This subject was discussed in Chapter 6, and detail design is important to minimise loss of energy at these points. Loss of energy is inevitable at the dampers needed in each port to maintain equality of flow in spite of the greater pressure difference available at some of them.

Fig. 13.7 illustrates a basic difference in this respect between the supply and exhaust sides. While the distributions of total pressure will bathe same if mixing losses are neglected, the static pressure changes far more along the exhaust duct and relatively large damper losses will be incurred towards the fan end on this side.

13.5.5 Traffic drag

Any vehicle travelling with a velocity v m/s relative to the air experiences a drag force - F Newtons. An equal and opposite propulsive force + F is applied to the air. If the vehicle is in a tunnel of crosssectional area A_tm^2, the propulsive force will cause a rise in pressure (F/A_t) Pa tending to move the air in the same direction as the vehicle. Conversely, if the vehicle is stationary and the air flowing along the tunnel, additional pressure (F/A_t) Pa will be needed to maintain the air movement against the vehicle drag.

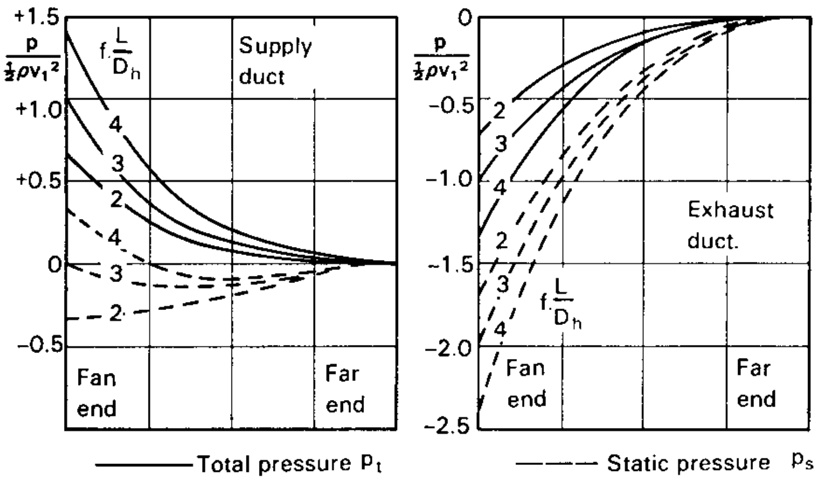


Fig. 13.7

Ducts with uniformly distributed supply and exhaust ports.

If the vehicle is travelling with velocity a and the air with velocity v in the same direction their relative velocity is (u - v) m/s. The force F is then proportional to the relative velocity pressure $\frac{1}{2}\rho$ (u - v)^2, to the frontal area of the vehicle A_v and to a *drag coefficient* C_v.

$$F\ (N) = C_v.A_v\ (m^2).\tfrac{1}{2}\rho.(u - v)^2 \qquad (135)$$

A typical value of A_v for a private car is 2.3m^2. Cars in convoy benefit from the "slipstream" effect, and suggested typical values for C_v per car with current models are:

Car spacing:	large	30m	15m	close
C_v	0·8	0·6	0·5	0·4

Vehicles which are large compared with the tunnel add to the drag by "piston effect'. One model test of double deck buses side by side in a two-lane tunnel indicated a C_v in the order of 5.

The total drag of a line of stationary vehicles can thus be estimated, and the corresponding pressure added to that required to maintain longitudinal flow. If they are moving faster than the air they will add to the other pressures supplied by jet fans or a semi-transverse system. Trial and error calculations are then needed to find the tunnel air velocity which will result in balance between the propulsive and the drag forces.

An example of the results of such a calculation is given in Fig. 13.8 which relates to a system of natural longitudinal ventilation without forced draught or external wind or other forces. Two-lane, one-way traffic is assumed with the maximum traffic density obtained according to speed from Fig. *13.2.* If the pollution data of Section *13.1* for a level roadway at sea level are applied to this situation it will be found that the CO levels are satisfactory for speeds above *30* km/hr in a 1000m long tunnel or above *20* km/hr in a 500m long tunnel. At lower traffic densities the situation would be better. Jet fans or other emergency provision would be necessary, however, unless the possibility of congested, crawling, traffic can be rigorously excluded.

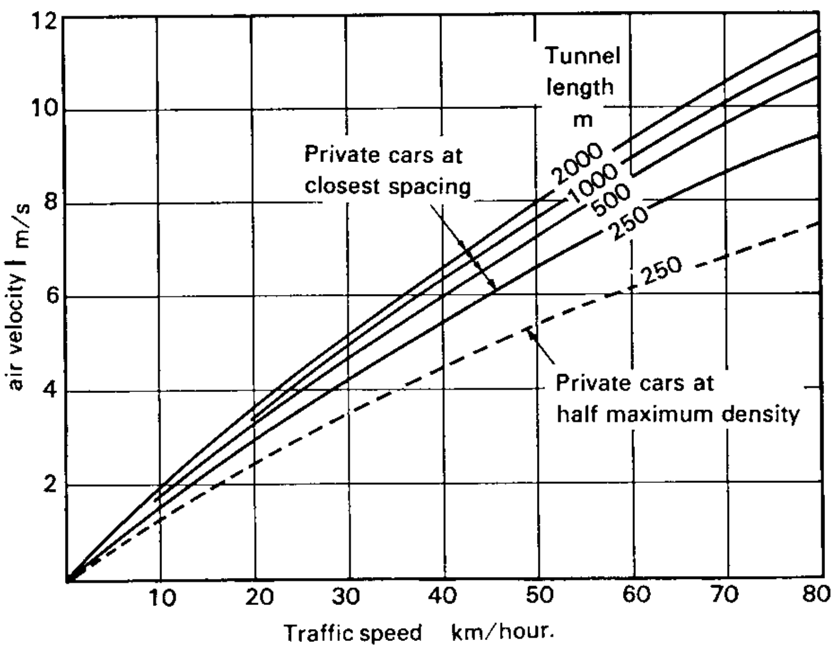


Fig. 13.8 Induced velocity in two-lane 45m² road tunnel.

13.6 Leakage from Long Ducts

When air has to be delivered through ductwork to a distant location great precautions must be taken to minimise leakage. Even when every care is taken a surprisingly large proportion of the air supplied fails to arrive at the far end and some estimate of the loss needs to be made. Ducted air supply during tunnel construction is one example, and auxiliary mines fan systems another.

Leakage can be measured simply by sealing both ends of a test length and maintaining pressure through a supply pipe containing a flow measurement orifice. Tests on ducts in service in mines and tunnel construction show the effects of installation strains and service damage. Both bolted joints and welded seams are liable to be less airtight in service than under laboratory conditions. The following table broadly corresponds to the recorded experience of the British National Coal Board and other observers.

Table 13.4
Leakage coefficients

Laboratory test duct with welded seams and asbestos, rubber or rubberised cork joint rings	$\lambda = 0{\cdot}1$
As above, very well installed and maintained in service	$\lambda = 0{\cdot}15$
As above, in fair average condition	$\lambda = 0{\cdot}5$
As above, in poor condition	$\lambda = 1{\cdot}5$
Plasticised PVC seamless tubing high pressure grade	$\lambda = 0{\cdot}2$
Plasticised PVC seamless tubing normal grade	$\lambda = 0{\cdot}7$
Flexible ducts of treated canvas	$\lambda = 5$
Metal ducts with joints and seams not stopped up, sealed or gasketted	$\lambda = 5$ to 20

λ is the leakage of standard air ($\rho = 1{\cdot}2$ kg/m^3) in m^3/s per 1000m^2 of duct surface at a pressure difference of 1000 Pa.

13.6.1 Leakage calculations

If a test duct of diameter D and length L is under uniform pressure Δp, λ can be found from the measured leakage flow Q_λ taken as:

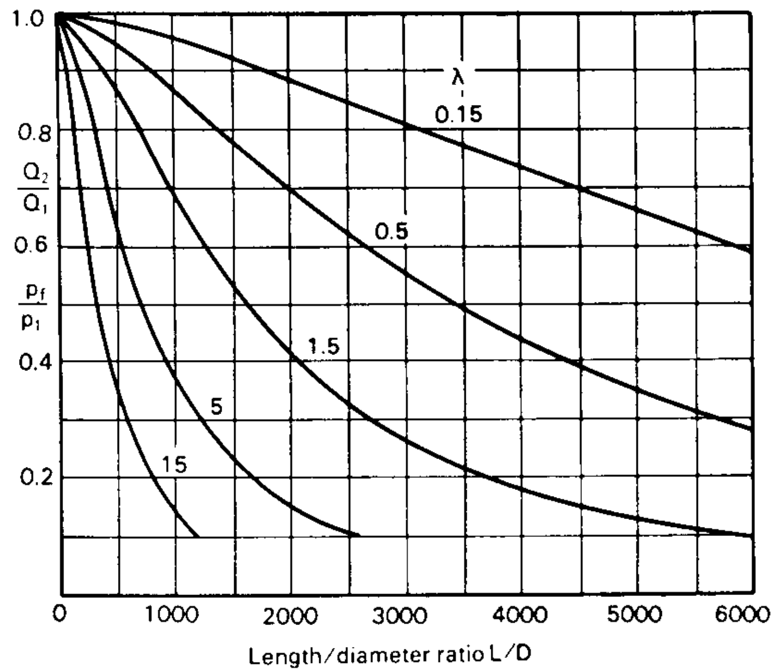
$$Q_\lambda = \lambda \frac{\pi\, DL\,(m^2)}{1000} \sqrt{\frac{\Delta p\,(Pa)}{1000} \frac{1{\cdot}2}{\rho\,(kg/m^3)}}\ m^3/s \qquad (136)$$

Consider a long duct coupled to a supply fan at the inlet end and intended to deliver Q_2m^3/s of fresh air at the far end. If the leakage were zero, the appropriate friction factor f, and the duct cross-section A m^2, there would be a positive static pressure p_f at the fan end equal to:

$$p_f = f.\frac{L}{D}.\tfrac{1}{2}\rho \left(\frac{Q_2}{A}\right)^2 \text{ Pa}$$

A greater volume flow Q, will be required at the fan end if there is leakage ($Q_1 - Q_2$), in order that Q_2 may be maintained. The higher velocities towards the fan end will increase the pressure drop; requiring a greater pressure, p_1. Mathematical analysis* leads to the practical data of Fig. 13.9; some approximation is involved and f is taken as 0.015 throughout, but precision is out of place when only a crude assessment of the leakage coefficient can be made.

* $\frac{Q_2}{Q_1}$ and $\frac{p_f}{p_1}$ are closely similar functions of the dimensionless variable $f^{\frac{1}{3}} \lambda^{\frac{2}{3}} \frac{L}{D}$



$\frac{Q_2}{Q_1} = \frac{\text{outlet volume flow}}{\text{inlet volume flow}}$ $\frac{p_f}{p_1} = \frac{\text{pressure drop at } Q_2 \text{ without leakage}}{\text{pressure to deliver } Q_2 \text{ with leakage}}$

Fig. 13.9 Influence of leakage on volume flow and pressure.

13.8.2 Example

An 800mm diameter duct is required to deliver 5m^3/s over a distance of 2 km. Welded seam duct lengths are to be used, properly gasketted and supported at each joint to minimise strain. What fan duty should be specified ?

Duct area $= \pi\,(0{\cdot}4)^2 = 0{\cdot}50\text{m}^2$ $\quad v_2 = 10\text{m/s}$

Velocity pressure at outlet $= \frac{1}{2} \times 1{\cdot}2\ (\text{kg/m}^3) \times 10^2 = 60$ Pa

From Fig. 13.6 f $= 0{\cdot}014$

$$p_f = 0{\cdot}014 \times 60 \times \frac{2000}{0{\cdot}8} = 2100 \text{ Pa}$$

Assume $\lambda = 0{\cdot}5$ as an average service condition. $\quad L/D = 2500$

From Fig. 13.9: $Q_2/Q_1 = 0{\cdot}62$

Therefore $Q_1 = 5{\cdot}0/0{\cdot}62 = 8{\cdot}1\text{m}^3/\text{s}$

Also from this Fig. $p_f/p_1 = 0{\cdot}62$

Therefore $p_1 = 2100/0{\cdot}62 = 3400$ Pa

Thus, a suitable fan duty would be:

8m^3/s at 3500 Pa with (8×3500) W $= 28$ kW air power

If extra care in installation secured an initial value of $\lambda = 0.15$, Q_2/Q_1 and p_f/p_1 would become 0.84. The corresponding fan requirement for $Q_2 = 5m^3/s$ would be

$5/0.84 = 6.0m^3/s$ $\quad$ $2100/0.84 = 2500$ Pa $\quad$ 15kW

The fan specified will exceed this duty, moving to a slightly different operating point on its characteristic. However, the air power is likely to remain fairly constant at 28 kW, so to estimate new values of Q_1 and

Q_2 multiply by $\sqrt[3]{28/15}$:

Fan duty:	**$7{\cdot}4m^3/s$ at 3800 Pa = 28 kW**
Discharged volume flow:	**$6{\cdot}2m^3/s$**

If the duct installation were allowed to deteriorate to the poor condition $\lambda = 1.5$, $Q_2/Q_1 = p_f/p_1 = 0.33$:

Fan duty required:	**$15m^3/s$ at 6300 Pa = 95 kW**
Actual fan duty:	**$10m^3/s$ at 2800 Pa = 28 kW**
Discharged volume flow:	**$3{\cdot}3m^3/s$**

Clearly there are practical limits to the length over which air may be conveyed by a single fan. These limits can be extended by employing a number of fans in series, spaced along the duct length. In fact, if the airway is divided into a number of sections of equal length, with a fan at the centre of each section the volume flow discharged will equal that at entry however long the airway.

This is rarely a practical solution, however. Equal leakage volumes will flow, outwards from the downstream, positive pressure, half of each section and inwards to the upstream, negative pressure, half. This local recirculation will be liable to contaminate the fresh air supply in the duct, if that is its purpose. If, on the other hand, the system is used to extract contaminated air, then the outward leakage will constitute an escape of contamination.

A more satisfactory method is to locate the fans at the beginning of each section, perhaps with an additional stage in the first one, to ensure a positive pressure everywhere in a supply duct, or negative pressure in an extraction duct. There will then be leakage, but the very high pressure required from a single fan, and the corresponding excessive leakage, will be avoided. Axial fans are best suited to such a system, but it is important that they should be guide vane types, or contra-rotating pairs. This is to avoid excessive outlet swirl, which is maintained for long distances in a cylindrical duct. Swirl at the inlet of a later fan stage would reduce its performance, and swirl in the airway increases the pressure drop.

13.7 Jet Fans

13.7.1 Use in tunnel ventilation

Jet fans may provide the whole propulsive force in a longitudinal ventilation scheme, or they may supplement a semi-transverse or transverse system in sensitive locations, e.g. at the tunnel entrance or exit. The fan is an axial machine with open inlet and outlet, designed for maximum efficiency in this condition rather than when operating against pressure. The rating is commonly in terms of the thrust applied to the air - with an equal and opposite reaction force borne by the fan supports. It is mounted above the traffic suspended from the tunnel roof, but must not be too close (less than 1 to 1.5 fan diameters) or there will be loss of thrust.

The basic thrust rating is equal to the *momentum flux* at fan outlet, that is the product of the mass flow and average velocity:

$$\begin{aligned}\text{Thrust} &= \rho\ (\text{kg/m}^3).Q_2\ (\text{m}^3/\text{s}).v_2\ (\text{m/s}) \\ &= \rho\, Q_2\, v_2\ \text{kg.m/s}^2 \text{ or Newtons} \\ &= \rho\, v_2^2\, A_2 \text{ where } A_2 \text{ is the outlet area}\end{aligned}$$

Acting on the tunnel area, A_1 m^2, this would produce a pressure rise:

$$\Delta p = \frac{A_2}{A_1}.\rho v_2^2 \text{ Pa}$$

The total pressure rise due to a number of fans, n, is simply the sum of the individual pressure rises. The fans may be located in parallel groups, but should be spaced 10 or more tunnel diameters apart lengthwise to ensure complete mixing of the jet from each.

$$\Delta p = \frac{n\,A_2}{A_1}.\rho v_2^2 \text{ Pa} \qquad (137)$$

These formulae are only exact if the mean tunnel velocity is very small compared with the fan outlet velocity. As much energy can be saved by employing a lower fan velocity and larger volume flow for the same thrust a more thorough treatment is justified.

13.7.2 Principles of jet action

Fig. 13.10 has been drawn (exaggerated for clarity) to assist in understanding the operation. The important factors are the average air velocities, V_1 in the tunnel, V_2 at the fan outlet, and V_3 in that section of the tunnel which is by-passed by the flow through the fan. V_3 and the corresponding velocity pressure p_{v3} will be less than V_1 and p_{v1}. As there is very little energy loss where flow is diverted into the fan, the total pressure in the by-passed tunnel section will remain virtually unchanged, which means that the static pressure here must rise-as shown by the heavy lines in the diagram. This static rise, Δp_s is the first contribution to the effective performance.

The broken line shows the course of the total pressure through the fan and on through the outlet jet until it rejoins the tunnel total pressure line when the jet is completely mixed. The fan total pressure, P_t is the total pressure rise after the impeller, but since the static pressure in the casing must equalise at outlet with the static pressure in the tunnel by-pass section, it will be seen that the fan velocity pressure, P_{v2}, exceeds the fan total pressure. The difference constitutes a *negative* fan static pressure, which will be:

$$P_s = -P_{v3}$$

The high outlet velocity of the fan creates a local excess of momentum flux in the jet. In dropping back to the normal tunnel value as the jet slows down this will exert a forward force on the air ("force = rate of change of momentum"). The reaction to this force is provided by a rise in tunnel static pressure, Ap_j, which is the second contribution to the

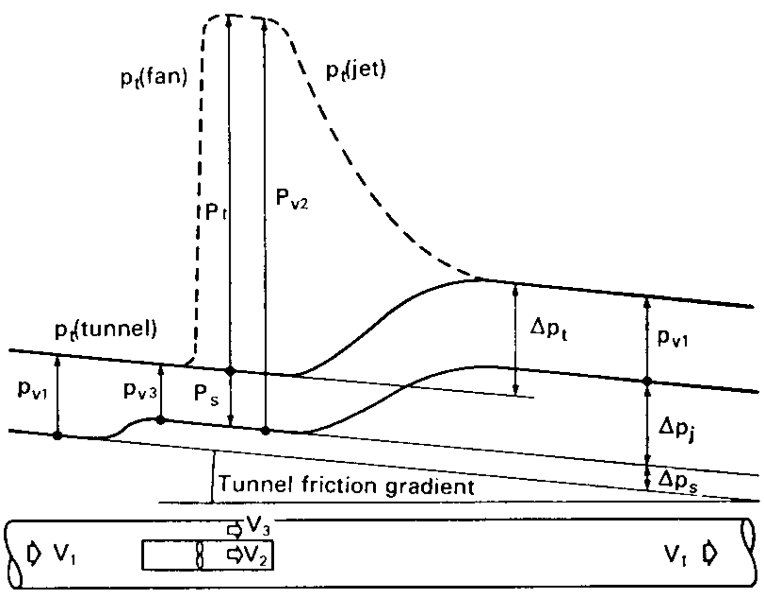


Fig. 13.10 Pressure diagram for jet fan in tunnel.

effective performance. The drop in the broken total pressure line represents the loss of fan outlet *energy* which takes place in the turbulent decay of the outlet jet.

The static and total pressure lines have been drawn at a slope which corresponds to the pressure gradient required to keep the air moving against tunnel wall friction. The effective pressure changes are measured from the downstream projections of these gradients, and it will be seen that the effective boost in total pressure due to the fan:

$$\Delta p_t = \Delta p_s + \Delta p_j$$

13.7.3 Calculation and design

The formulae that are needed for the full assessment of a tunnel ventilation system can be derived from these principles in terms of the quantities:

Q_1 Q_2 m^3/s volume flow	Q_1 A_1 V_1 measured in the
A_1 A_2 m^3 cross-section	tunnel cross-section,
V_1 V_2 m/s air velocity	Q_2 A_2 V_2 at the fan outlet

In terms of the mean tunnel velocity pressure: **(138)**

Static pressure component $\Delta p_s = \frac{1}{2}\rho V_1^2 . \frac{Q_2}{Q_1} . 2 \left(1 - \frac{V_1}{V_2}\right)$

Jet action component $\Delta p_j = \frac{1}{2}\rho V_1^2 . \frac{Q_2}{Q_1} . 2 \left(\frac{V_2}{V_1}\right) \left(1 - \frac{V_1}{V_2}\right)^2$

Total pressure rise $\Delta p_t = \frac{1}{2}\rho V_1^2 . \frac{Q_2}{Q_1} . 2 \left(\frac{V_2}{V_1} - 1\right)$

Fan static pressure $P_s = -\frac{1}{2}\rho V_1^2$

Fan total pressure $P_t = -\frac{1}{2}\rho V_1^2 \left[\left(\frac{V^2}{V_1}\right)^2 - 1\right]$

In terms of the fan outlet velocity pressure: **(139)**

Static pressure component $\Delta p_s = \frac{1}{2}\rho V_2^2 . \frac{A_2}{A_1} . 2 \left(\frac{V_1}{V_2}\right) \left(1 - \frac{V_1}{V_2}\right)$

Jet action component $\Delta p_j = \frac{1}{2}\rho V_2^2 . \frac{A_2}{A_1} . 2 \left(1 - \frac{V_1}{V_2}\right)^2$

Total pressure rise $\Delta p_t = \frac{1}{2}\rho V_2^2 . \frac{A_2}{A_1} . 2 \left(1 - \frac{V_1}{V_2}\right)$

Fan static pressure $P_s = -\frac{1}{2}\rho V_2^2 . \left(\frac{V_1}{V_2}\right)^2$

Fan total pressure $P_t = \frac{1}{2}\rho V_2^2 \left[1 - \left(\frac{V_1}{V_2}\right)^2\right]$

The jet action efficiency η_j is the ratio of the work done against tunnel friction to the air power output of the fan Q_2P_t.

$$\eta_j = \frac{2V_1}{V_2 + V_1}$$

The overall efficiency will depend on the fan efficiency η_t, also:

$$\text{overall} = \eta_j \, \eta_t$$

These formulae involve residual errors of the order of $n_P \, A_2/A_1$ per unit where n_p is the number of fans in a parallel group.

The resistance Δp_1 to the required volume flow Q_1 through the tunnel will be estimated as the sum of the items listed in 13.5.1. It can be expressed in terms of the velocity $V_1 = Q_1/A_1$ as:

$$\Delta p_1 = K \cdot \tfrac{1}{2}\rho V_1^2$$

The number of fans, n, and total pressure rise per fan, Δp_t must be chosen so that:

$$n \cdot \Delta p_t = \Delta p_1$$

From formula (138) for Δp_t

$$K \cdot \tfrac{1}{2}\rho V_1^2 = \tfrac{1}{2}\rho V_1^2 \cdot \frac{nQ_2}{Q_1} \cdot 2\left(\frac{V_2}{V_1} - 1\right)$$

$$\text{Therefore } nQ_2 = a\,K\,Q_1$$

$$\text{where } a = \frac{V_1}{2\,(V_2 - V_1)}$$

$$\text{while } \eta_j = \frac{2V_1}{V_2 + V_1}$$

a and η_j are plotted against V_2/V_1 in Fig. 13.11 showing how the jet action efficiency can be increased by reducing V_2/V_1 which, in turn, requires a higher value of a and hence of nQ_2 for a given ventilation load expressed by KQ_1.

The designer must, therefore, judge how far to reduce running costs (higher η_j)at the expense of increase in capital cost (more and larger, lower speed, jet fans). In any case a substantial reserve jet thrust is desirable to deal with possible excess in uncertain factors such as wind pressure and traffic drag.

Two-speed operation (e.g. 2/4 pole motors) is worth careful consideration. If the low speed suffices for normal operation the low value of V_2/V_1 increases η_j and saves running cost. For heavy traffic some of the fans can be switched to top speed, and in emergency more than four times normal thrust is available when all fans are at full speed.

The selection of jet fans for a given tunnel duty is covered by Fig. 13.11. The inverse problem can be solved with the aid of the linear chart, Fig. 13.12. This enables the volume flow generated in the tunnel by a given installation of jet fans to be determined. The method is best explained by an example.

13.7.4 Example.

Tunnel data: $L = 900\text{m}$ $\quad A_1 = 50\text{m}^2$ $\quad D_h = 7{\cdot}5\text{m}$

$f = 0{\cdot}015$ $\quad \Delta p_f/\tfrac{1}{2}\rho V_1^2 = f \cdot L/D_h = 0{\cdot}015 \times 120 = 1{\cdot}8$

Traffic: 300 stationary private cars in two lanes.

Drag $\Delta p_v/\tfrac{1}{2}\rho V_1^2 = nC$ $\quad A_v/A_1 = 300 \times 0{\cdot}4 \times 2{\cdot}3/50 = 5{\cdot}5$

CO production at $80\text{cm}^3\text{/s}$ per car $= 80 \times 300 = 24{,}000\text{cm}^3\text{/s}$

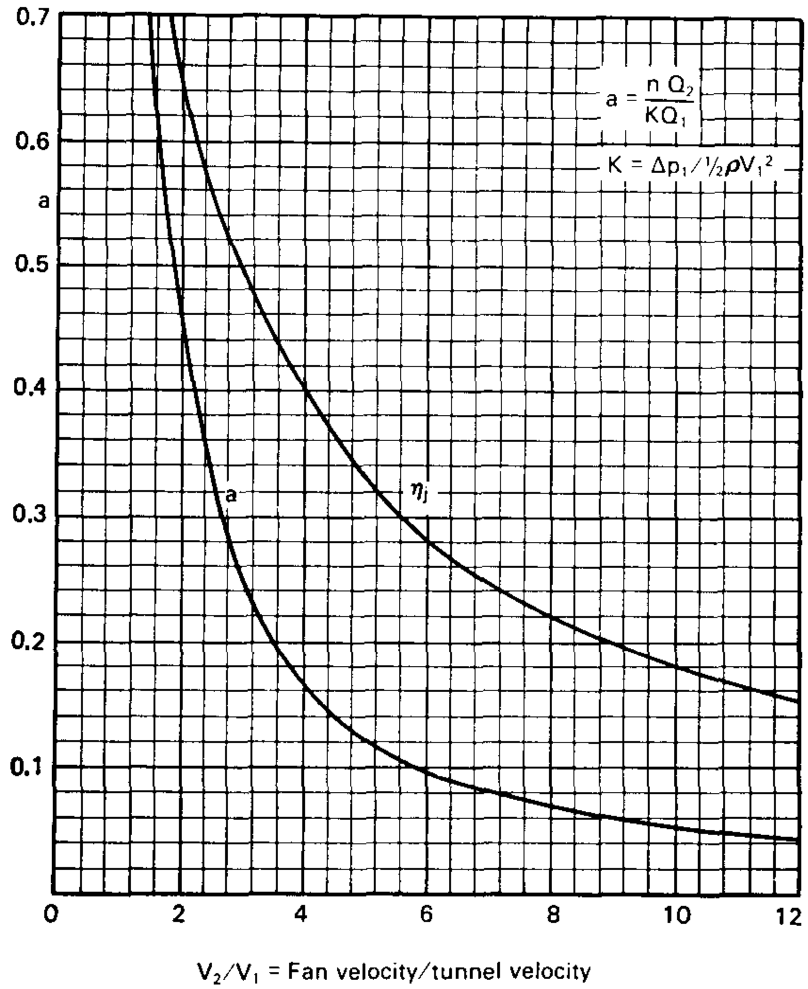


Fig. 13.11 Jet fan performance.

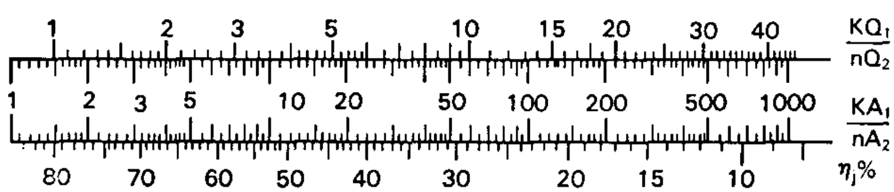


Fig. 13.12

Derivation of volume flow and efficiency from area ratio.

Loss factors:

$$\begin{aligned} \text{Entrance loss} &= 0{\cdot}2 \times \tfrac{1}{2}\rho V_1^2 \\ \text{Friction loss} &= 1{\cdot}8 \times \tfrac{1}{2}\rho V_1^2 \\ \text{Outlet loss} &= 1{\cdot}0 \times \tfrac{1}{2}\rho V_1^2 \\ \text{Drag loss} &= 5{\cdot}5 \times \tfrac{1}{2}\rho V_1^2 \\ \hline K &= 8{\cdot}5 \end{aligned}$$

$\Delta p_1 = K \,.\, \tfrac{1}{2}\rho V_1^2$

Jet fans: 18 × 500mm × 2900 rev/min spaced 100m in pairs.

$Q_2 = 7{\cdot}0\text{m}^3/\text{s} \qquad A_2 = 0{\cdot}20\text{m}^2 \qquad V_2 = 35\text{m/s}$

$\tfrac{1}{2}\rho V_2^2 = 730 \text{ Pa} \qquad \eta_t = 70\%$

$W_i = 7{\cdot}0 \times 730/0{\cdot}70 = 7300 \text{ W per fan}$

$$\frac{KA_1}{nA_2} = \frac{8{\cdot}5 \times 50}{18 \times 0{\cdot}2} = 118 \quad \text{Therefore from Fig. 13.12 } \frac{KQ_1}{nQ_2} = 14{\cdot}5$$

$$\text{Therefore } Q_1 = \frac{18\,(n) \times 7{\cdot}0\,(Q_2) \times 14{\cdot}5}{8{\cdot}5\,(K)} = 215\text{m}^3/\text{s} \quad V_1 = 4{\cdot}3\text{m/s}$$

$$\Delta p_1 = K \,.\, \tfrac{1}{2}\rho V_1^2 = 8{\cdot}5 \times 0{\cdot}5 \times 1{\cdot}2 \times (4{\cdot}3)^2 = 93 \text{ Pa}$$

Maximum CO content = 24,000/215 = 110 ppm

Total power input = 18 × 7·3 = 131 kW

The following table compares this performance with that of twelve 800mm, 1450-970 rev/min two-speed fans spaced 150m in pairs, calculated in a similar manner.

Jet Fans					Whole Tunnel			
Number n	Diam. mm	Speed rev/min	Q_2 m^3/s	Input kW	Q_1 m^3/s	Δp_1 Pa	Total kW	CO ppm
18	500	2900	7·0	7·3	215	93	131	110
12	800	1450	14·0	9·4	215	93	113	110
12	800	970	9·3	2·8	145	43	34	165

Clearly the twelve 800mm fans have the better performance, though their cost is likely to be about 25% greater - offset by savings in wiring and installation cost. The low speed performance should meet the stationary traffic requirement; when in motion the drag would change to a propulsive force, and Q_2 would increase substantially. The power saving is 12% at high speed, 75% at low, and the low speed fans would be 10 dB quieter.

The margin to deal with the drag of large vehicles and with adverse winds might be considered inadequate. An installation of eighteen 800mm fans would produce 150 Pa - a margin of 100 Pa over that required for $150\text{m}^3/\text{s}$ on the above basis of calculation.

CHAPTER 14
Useful data

Table 14.1
SI basic and derived units

Quantity	Unit	Symbol	Dimensions
Base Units			
Length	metre	m	L
Mass	kilogram	kg	M
Time	second	s	T
Temperature, absolute	degree kelvin	K	Θ
Electric current	ampere	A	QT^{-1}*
Derived Units			
Area	square metre	m^2	L^2
Volume	cubic metre	m^3	L^3
Density	kilogram per cubic metre	kg/m^3	$L^{-3}M$
Frequency	hertz	Hz	T^{-1}
Velocity	metre per second	m/s	LT^{-1}
Volume flow rate	cubic metre per second	m^3/s	L^3T^{-1}
Mass flow rate	kilogram per second	kg/s	MT^{-1}
Acceleration	metre per second per second	m/s^2	LT^{-2}
Force	newton ($kg.ms^{-2}$)	N	LMT^{-2}
Pressure, stress	pascal (Nm^{-2})	Pa	$L^{-1}MT^{-2}$
Viscosity, dynamic	pascal-second	Pa s	$L^{-1}MT^{-1}$
Viscosity, kinematic	metre squared per second	m^2/s	L^2T^{-1}
Energy, work, heat	joule (N.m)	J	L^2MT^{-2}
Power, heat flow	watt (Js^{-1})	W	L^2MT^{-3}
Heat flow intensity	watt per square metre	W/m^2	MT^{-3}
Calorific value	joule per kilogram	J/kg	L^2T^{-2}
Specific heat capacity	joule per kilogram—degree	J/kg deg C	$L^2T^{-2}\Theta^{-1}$
Heat transfer coefficient	watt per sq. metre–degree	W/m^2 deg C	$MT^{-3}\Theta^{-1}$
Thermal conductivity	watt per metre–degree	W/m deg C	$LMT^{-3}\Theta^{-1}$
Thermal resistivity	metre–degree per watt	m deg C/W	$L^{-1}M^{-1}T^3\Theta$
Quantity of electricity	coulomb (A.s)	C	Q*
Electromotive force	volt (WA^{-1})	V	$L^2MT^{-2}Q^{-1}$
Electric resistance	ohm (VA^{-1})	Ω	$L^2MT^{-1}Q^{-2}$
Capacitance	farad (CV^{-1})	F	$L^{-2}M^{-1}T^2Q^2$

*The choice of electric charge, Q, as the fundamental electrical dimension leads to one of several equally valid alternative systems.

Multiples and submultiples of SI basic units

Prefix	Multiple	Name	Prefix	Submultiple	Name
da –	10	deca –	d –	0·1	deci –
h –	100	hecto –	c –	0·01	centi –
k –	1000	kilo –	m –	0·001	milli –
M –	10^6	mega –	μ –	10^{-6}	micro –
G –	10^9	giga –	n –	10^{-9}	nano –
T –	10^{12}	tera –	p –	10^{-12}	pico –
			f –	10^{-15}	femto –
			a –	10^{-18}	atto –

The factors defined by deca- hecto- deci- and centi- are best confined to well-established usages such as centimetre or hectare.

The original European definitions of a billion (10^{12}) and a trillion (10^{18}) appear to be dying out in favour of the current American usage with a billion equal to 10^9 and a trillion equal to 10^{12}. A milliard is unambiguously equal to 10^9.

Some mathematical constants

π	= 3·14159	$\pi/4$	= 0·785398
π^2	= 9·86960	e	= 2·71828
$\log_e x$	= 2·30259 $\log_{10} x$	$\log_{10} x$	= 0·434294 $\log_e x$
1 radian	= 57·2958 degrees	1 degree	= 0·0174533 radians

The decibel scale

When two powers (measured in W, μW, pW, etc.) differ by x decibels the ratio of the larger power to the smaller is given by the following table. The approximate value is sufficient for most of the purposes for which decibels are used.

x dB =	0	1	2	3	4	5	6	7	8	9	10
$(10)^{0.1x}$ =	1·000	1·259	1·585	1·995	2·512	3·162	3·981	5·012	6·310	7·943	10·00
approx. =	1·0	1·25	1·6	2·0	2·5	3·2	4·0	5·0	6·3	8·0	10

Normal statistical distribution

The results of research are often reported with a statement of the remaining uncertainty in statistical terms. The arithmetic mean value M of the quantity determined is quoted together with the *standard deviation S*. Given the *normal distribution* of random errors in the measurements there will then be a y% probability that the true value lies within the range M - xS to M+ xS.

Probability y =	25%	50%	68%	80%	90%	95%	98%	99%
when x =	0·32	0·68	1·00	1·28	1·65	1·96	2·33	2·57

Engineering tolerances are commonly set at ± 2S (i.e. ± 200 S/M per cent) and are classed as having 95% *probability.*

Table 14.2

Conversions to and from SI units

LENGTH

1 inch	= 25·4 mm*	1 m	= 39·3701 in
1 foot	= 304·8 mm*	1 m	= 3·28084 ft
1 yard	= 0·9144 m*	1 m	= 1·09361 yd
1 mile (statute, 1,760 yards)	= 1·60934 km	1 km	= 0·621371 mile
1 nautical mile (international)	= 1·852 km*	1 km	= 0·539957 n. mile
1 fathom (6 ft)	= 1·8288 m*	1 m	= 0·546807 fathom
1 "thou" (0·001 in)	= 25·4 μm*	1 μm	= 0·03937 "thou"

AREA

1 square inch	= 645·16 mm^2*	1 m^2	= 1550·00 in^2
1 square foot	= 0·0929030 m^2	1 m^2	= 10·7639 ft^2
1 square yard	= 0·836127 m^2	1 m^2	= 1·19599 yd^2
1 square mile	= 2·58999 km^2	1 km^2	= 0·386102 sq mile
1 acre (4,840 sq yards)	= 0·404686 ha	1 ha	= 2·47105 acre
1 hectare (ha)	= 10000 m^2	1 km^2	= 100 ha

*Exactly

Table 14.2—*continued*

Conversions to and from SI units

	VOLUME		
1 cubic inch	= 16·3871 cm^3	1 l (litre)	= 61·0237 in^3
1 cubic foot	= 0·0283168 m^3	1 m^3	= 35·3147 ft^3
1 cubic yard	= 0·764555 m^3	1 m^3	= 1·30795 yd^3
1 pint	= 0·568261 l	1 l	= 1·75975 pint
1 gallon (UK)	= 4·54609 l	1 l	= 0·219969 gall (UK)
1 gallon (U.S. = 231 in^3)	= 3·78541 l	1 l	= 0·264172 gall (US)
1 barrel (oil = 42 gall U.S.)	= 0·1590 m^3	1 m^3	= 6·290 bbl
1 litre (1 l = 1 dm^3, exactly)	= 0·001 m^3	1 m^3	= 1000 l
1 cc (cubic centimetre)	= 10^{-6} m^3	1 l	= 1000 cc
	MASS		
1 grain	= 64·7989 mg	1 g (gram)	= 15·4324 gr
1 ounce (oz avoirdupois)	= 28·3495 g	1 kg	= 35·2740 oz
1 pound (lb avoirdupois)	= 0·453592 kg	1 kg	= 2·20462 lb
1 stone (14 lb)	= 6·35029 kg	1 t (tonne)	= 157·473 stone
1 hundredweight (112 lb)	= 50·8023 kg	1 t	= 19·6841 cwt
1 ton (2,240 lb)	= 1016·05 kg	1 t	= 0·984207 ton
1 short ton (U.S. = 2,000 lb)	= 907·185 kg	1 t	= 1·10231 short ton
1 slug	= 14·5939 kg	1 kg	= 0·0685218 slug
1 tonne (metric)	= 1 000 kg		

Table 14.2—*continued*

Conversions to and from SI units

TIME

1 hour (60 min)	= 3 600 s*		
1 day (24 hours)	= 86 400 s*		
1 year of 365·25 days	= 31 557 600 s*		

ROTATIONAL SPEED		*FREQUENCY*	
1 radian per second	= 0·159155 rev/s	1 cycle per second	= 1 Hz (hertz)
1 rev/sec (60 rev/min)	= 6·28319 rad/s	1 kilocycle (kc)	= 1000 Hz
1 radian per second	= 9·54930 rev/min		

VELOCITY

1 foot per second	= 0·3048 m/s*	1 m/s	= 3·28084 ft/s
1 foot per minute	= 0·00508 m/s*	1 m/s	= 196·850 ft/min
1 mile per hour	= 0·44704 m/s	1 m/s	= 2·23694 mile/h
1 knot (international kn)	= 0·514444 m/s	1 m/s	= 1·94384 knot
1 kilometre per hour	= 0·277778 m/s	1 m/s	= 3·6 km/h*

*Exactly

Table 14.2—*continued*

Conversions to and from SI units

VOLUME FLOW			
1 cubic foot per minute (cfm)	= 0·471947 l/s	1 m^3/s	= 2118·88 ft^3/min
1 cubic foot per second (cusec)	= 0·0283168 m^3/s	1 m^3/s	= 35·3147 ft^3/s
1 gallon (UK) per minute	= 0·0757682 l/s	1 $m^3/hour$	= 0·588578 ft^3/min
1 cubic metre per hour	= 0·277778 l/s	1 ft^3/min	= 1·69902 $m^3/hour$
MASS FLOW			
1 pound per second	= 0·453592 kg/s	1 kg/s	= 2·20462 lb/s
1 ton (UK) per hour	= 0·282235 kg/s	1 kg/s	= 3·54314 ton/hour
ACCELERATION			
1 foot per second per second	= 0·3048 m/s^2	1 m/s^2	= 3·28084 (ft/s^2)
32·174 ft/s^2 (std. gravity)	= 9·80665 m/s^2	1 m/s^2	= 100 Gal (galileo)
DENSITY			
1 pound per cubic foot	= 16·0185 kg/m^3	1 kg/m^3	= 0·0624280 lb/ft^3
1 pound per cubic inch	= 27·6799 g/cm^3	1 g/cm^3	= 0·0361273 lb/in^3
1 gram per cubic centimetre (g/cm^3)	= 1000 kg/m^3	1 g/cm^3	= 10·0224 lb/gall (UK)

Table 14.2—*continued*

Conversions to and from SI units

	FORCE		
1 pound-force (lbf)	= 4·44822 N	1 N	= 0·224809 lbf
1 ton-force	= 9·96402 kN	1 MN	= 100·361 ton-f
1 poundal	= 0·138255 N	1 N	= 7·23301 pdl
1 kilogram-force (kgf)	= 9·80665 N	1 N	= 0·101972 kgf
1 dyne	= 10^{-5} N	1 N	= 100 000 dyn
	TORQUE		
1 pound-force foot	= 1·35582 Nm	1 Nm	= 0·737562 lbf ft
1 kilogram-force metre	= 9·80665 Nm	1 Nm	= 0·101972 kgf m
1 dyne centimetre	= 10^{-7} Nm	1 Nm	= 10^{7} dyn cm
	PRESSURE, STRESS		
1 inch of water	= 249·089 Pa	1 kPa	= 4·01463 in H_2O
1 millimetre of water (=1 kgf/m^2)	= 9·80665 Pa	1 Pa	= 0·101972 mm H_2O
1 inch of mercury	= 3·38639 kPa	1 kPa	= 0·295300 in Hg
1 millimetre of mercury (torr)	= 133·322 Pa	1 kPa	= 7·50062 mm Hg
1 millibar	= 100 Pa	1 kPa	= 10 mb
1 standard atmosphere	= 101·325 kPa	10^{5} Pa	= 0·986923 atmos

Table 14.2—*continued*

Conversions to and from SI units

PRESSURE, STRESS

1 pound-force per sq. ft.	= 47·8803 Pa	1 kPa	= 20·8854 lbf/ft^2
1 pound-force per sq. inch	= 6·89476 kPa	1 MPa	= 145·038 lbf/in^2
1 ton (UK) per sq. inch	= 15·4443 MPa	1 MPa	= 0·0647490 ton f/in^2
1 kilogram-force per sq. mm	= 9·80665 MPa	1 MPa	= 0·101972 kgf/mm^2

DYNAMIC VISCOSITY

1 pound (mass) per second per ft	= 1·48816 Pa s	1 Pa s	= 0·671969 lb/s ft
1 pound-force second per sq ft	= 47·8803 Pa s	1 Pa s	= 0·0208854 lbf s/ft^2
1 kilogram-force per sq m	= 9·80665 Pa s	1 Pa s	= 0·101972 kgf s/m^2
1 kilogram per metre second	= 1 Pa s	1 Pa s	= 10 poise, P
1 centipoise (= 0·01 dyn s/cm^2)	= 0·001 Pa s	1 Pa s	= 1000 cP

KINEMATIC VISCOSITY

1 foot squared per second	=0·0929030 m^2/s	1 m^2/s	= 10·7639 ft^2/s
1 centistokes (=0·01 cm^2/s)	10^{-6} m^2/s	1 mm^2/s	= 1 cSt

For details of the Redwood Engler and Saybolt scales see Table 14.15.

Table 14.2—*continued*

Conversions to and from SI units

	ENERGY, HEAT		
1 foot pound-force	= 1·35582 J	1 J	= 0·737562 ft lbf
1 metre kilogram—force	= 9·80665 J*	1 J	= 0·101972 m kgf
1 kilowatt hour	= $3{\cdot}6 \times 10^{6}$ J	1 MJ	= 0·277778 kWh
1 erg	= 10^{-7} J	1 μ J	= 10 erg
1 British thermal unit (Btu)	= 1055·06 J	1 kJ	= 0·947817 Btu
1 therm (of gas = 10^{5} Btu)	= 105·506 MJ	1 GJ	= 9·47817 therm
1 (gram) calorie	= 4·1868 J*	1 J	= 0·238846 cal
1 kilcalorie (kcal)	= 4186·8 J*	1 kJ	= 0·238846 kcal
	POWER, HEAT FLOW		
1 horse-power (550 ft lbf/s)	= 745·700 W	1 kW	= 1·34102 hp
1 cheval-vapeur (metric hp)	= 735·499 W	1 kW	= 1·35962 ch
1 Pferdestärke (= 75 kgf m/s)	= 735·499 W	1 kW	= 1·35962 Pf
1 British thermal unit per hour	= 0·293071 W	1 W	= 3·41214 Btu/h
1 calorie per second	= 4·1868 W*	1 W	= 0·238846 cal/s
1 kilocalorie per hour	= 1·163 W*	1 W	= 0·859845 kcal/h
	HEAT FLOW INTENSITY		
1 British thermal unit per ft^2 per hr	= 3·15459 W/m^2	1 W/m^2	= 0·316998 Btu/ft^2h
1 kilocalorie per square metre per hr	= 1·163 W/m^2*	1 W/m^2	= 0·859845 $kcal/m^2h$

*Exactly.

Table 14.2—*continued*

Conversion to and from SI units

ABSOLUTE TEMPERATURE				*TEMPERATURE*	
t K (Kelvin)		= 1·8t °R (Rankine)		t°C	= $^9/_5$t + 32 °F
At t°C (Celsius)	abs. temp.	= (t + 273·15) K		t°F	= $^5/_9$ (t −32) °C
At t°F (Fahrenheit)	abs. temp.	= (t + 459·67) °R		1 deg C	= 1·8 deg F
At t°F (Fahrenheit)	abs. temp.	= ($^5/_9$t + 255·37) K			(temperature difference)

SPECIFIC HEAT CAPACITY

1 British thermal unit per pound deg F	= 4·1868 kJ/kg deg C	1 kJ/kg deg C	= 0·238846 Btu/lb deg F
1 kilocalorie per kilogram per deg C	= 4·1868 kJ/kg deg C	1 kJ/kg deg C	= 0·238846 kcal/kg deg C
1 British thermal unit per ft³ per deg F	= 67·0661 kJ/m³ deg C	1 kJ/m³ deg C	= 0·0149107 Btu/ft³ deg F
1 kilocalorie per cubic metre per deg C	= 4·1868 kJ/m³ deg C	1 kJ/m³ deg C	= 0·238846 kcal/m³ deg C

HEAT TRANSFER COEFFICIENT

1 Btu per ft² per hour per deg F	= 5·67826 W/m² deg C	1 W/m² deg C	= 0·176110 Btu/ft²h deg F
1 kilocalorie per m² per hour per deg C	= 1·163 W/m² deg C	1 W/m² deg C	= 0·859845 kcal/m²h deg C

THERMAL CONDUCTIVITY

1 Btu per ft per hour per deg F	= 1·73073 W/m deg C	1 W/m deg C	= 0·577789 Btu/ft h deg F
1 kilocalorie per m per hour per deg C	= 1·163 W/m deg C	1 W/m deg C	= 0·859845 kcal/m h deg C

Table 14.3 Conversion of inches to millimetres

in	0	1	2	3	4	5	6	7	8	9
0		25·4	50·8	76·2	101·6	127·0	152·4	177·8	203·2	228·6
10	254·0	279·4	309·8	330·2	355·6	381·0	406·4	431·8	457·2	482·6
20	508·0	533·4	558·8	589·2	609·6	635·0	660·4	685·8	711·2	736·6
30	762·0	787·4	812·8	383·2	863·6	889·0	914·4	939·8	965·2	990·6
40	1016·0	1041·4	1066·8	1092·2	1117·6	1143·0	1168·4	1193·8	1219·2	1244·6
50	1270·0	1295·4	1320·8	1346·2	1371·6	1397·0	1422·4	1447·8	1473·2	1498·6
60	1524·0	1549·4	1574·8	1600·2	1625·6	1651·0	1676·4	1701·8	1727·2	1752·6
70	1778·0	1803·4	1828·8	1854·2	1879·6	1905·0	1930·4	1955·8	1981·2	2006·6
80	2032·0	2057·4	2082·8	2108·2	2133·6	2159·0	2184·4	2208·8	2235·2	2260·6
90	2286·0	2311·4	2336·8	2362·2	2387·6	2413·0	2438·4	2463·8	2489·2	2514·6

All conversions on this page are exact, since 1 inch = 25·4 mm exactly by definition.

The table can be extended indefinitely by dividing or multiplying the in and mm entries by 10, 100, 1000, etc., and also by addition. Example:

3·9 in	=	99·06 mm	dividing 39 in entry by 10
0·058 in	=	1·4732 mm	dividing 58 in entry by 1000
0·0005 in	=	0·01270 mm	dividing 5 in entry by 10000
3·9585 in	=	100·54590 mm exactly	

Since measurement is seldom closer than the nearest 0·0001 in or 0·001 mm the result might be read as 100·546 mm.

Table 14.4
The Beaufort scale of wind

Beaufort number	Description of wind	Wind speeds up to (approximately)			Approx. maximum velocity pressure
		m/s	km/hr	mph	Pa
0	Calm	0·5	2	1	0·2
1	Light air	1·5	5	3	1·2
2	Light breeze	3	11	7	6
3	Gentle breeze	6	22	12	20
4	Moderate breeze	8	30	18	40
5	Fresh breeze	11	40	24	75
6	Strong breeze	14	50	30	120
7	Moderate gale	17	60	38	170
8	Fresh gale	21	75	47	260
9	Strong gale	24	87	56	350
10	Whole gale	28	100	66	500
11	Storm	32	115	76	650
12	Hurricane	36+	130+	82+	800+

Note. The scale originally established in miles per hour (as above) differs a little from the current values quoted in kilometres per hour. Meteorologists report wind speeds for an altitude of 10 metres in open country.

Table 14.5
Variations of air conditions with altitude

Altitude m	Pressure kPa	Temperature °C	Density kg/m^3
−250	104·4	17	1·25
Sea level	101·3	15	1·22
250	98·4	13	1·20
500	95·5	12	1·17
750	92·6	10	1·14
1000	89·9	8	1·11
1500	84·6	5	1·06
2000	79·5	2	1·00
3000	70·1	−4	0·91
4000	61·6	−11	0·82
6000	47·2	−24	0·66
8000	35·6	−37	0·53
10000	26·4	−50	0·41
20000	5·5	−56	0·088
30000	1·2	−46	0·018

Based on the standard atmosphere of the International Civil Aviation Organisation (ICAO) and representative of average atmospheric conditions in temperate latitudes.

Table 14.6

Fahrenheit (°F) to Celsius (°C) conversion

Deg. F	Deg. C	Deg. F	Deg. C	Deg. F	Deg. C	Deg. F	Deg. C
−40	−40·0	0	−17·8	40	4·4	80	26·7
−39	−39·4	1	−17·2	41	5·0	81	27·2
−38	−38·9	2	−16·7	42	5·6	82	27·8
−37	−38·3	3	−16·1	43	6·1	83	28·3
−36	−37·8	4	−15·6	44	6·7	84	28·9
−35	−37·2	5	−15·0	45	7·2	85	29·4
−34	−36·7	6	−14·4	46	7·8	86	30·0
−33	−36·1	7	−13·9	47	8·3	87	30·6
−32	−35·6	8	−13·3	48	8·9	88	31·1
−31	−35·0	9	−12·8	49	9·4	89	31·7
−30	−34·4	10	−12·2	50	10·0	90	32·2
−29	−33·9	11	−11·7	51	10·6	91	32·8
−28	−33·3	12	−11·1	52	11·1	92	33·3
−27	−32·8	13	−10·6	53	11·7	93	33·9
−26	−32·2	14	−10·0	54	12·2	94	34·4
−25	−31·7	15	−9·4	55	12·8	95	35·0
−24	−31·1	16	−8·9	56	13·3	96	35·6
−23	−30·6	17	−8·3	57	13·9	97	36·1
−22	−30·0	18	−7·8	58	14·4	98	36.7
−21	−29·4	19	−7·2	59	15·0	99	37·2
−20	−28·9	20	−6·7	60	15·6	100	37·8
−19	−28·3	21	−6·1	61	16·1	101	38·3
−18	−27·8	22	−5·6	62	16·7	102	38·9
−17	−27·2	23	−5·0	63	17·2	103	39·4
−16	−26·7	24	−4·4	64	17·8	104	40·0
−15	−26·1	25	−3·9	65	18·3	105	40·6
−14	−25·6	26	−3·3	66	18·9	106	41·1
−13	−25·0	27	−2·8	67	19·4	107	41·7
−12	−24·4	28	−2·2	68	20·0	108	42·2
−11	−23·9	29	−1·7	69	20·6	109	42·8
−10	−23·3	30	−1·1	70	21·1	110	43·3
−9	−22·8	31	−0·6	71	21·7	111	43·9
−8	−22·2	32	0	72	22·2	112	44·4
−7	−21·7	33	0·6	73	22·8	113	45·0
−6	−21·1	34	1·1	74	23·3	114	45·6
−5	−20·6	35	1·7	75	23·9	115	46·1
−4	−20·0	36	2·2	76	24·4	116	46·7
−3	−19·4	37	2·8	77	25·0	117	47·2
−2	−18·9	38	3·3	78	25·6	118	47·8
−1	−18·3	39	3·9	79	26·1	119	48·3

Table 14.6—*continued*

Fahrenheit (°F) to Celsius (°C) conversion

Deg. F	Deg. C	Deg. F	Deg. C	Deg. F	Deg. C	Deg. F	Deg. C
120	48·9	160	71·1	200	93·3	400	204·4
121	49·4	161	71·7	201	93·9	410	210·0
122	50·0	162	72·2	202	94·4	420	215·6
123	50·6	163	72·8	203	95·0	430	221·1
124	51·1	164	73·3	204	95·6	440	226·7
125	51·7	165	73·9	205	96·6	450	232·2
126	52·2	166	74·4	206	96·7	460	237·8
127	52·8	167	75·0	207	97·2	470	243·3
128	53·3	168	75·6	208	97·8	480	248·9
129	53·9	169	76·1	209	98·3	490	254·4
130	54·4	170	76·7	210	98·9	500	260·0
131	55·0	171	77·2	211	99·4	510	265·6
132	55·6	172	77·8	212	100·0	520	271·1
133	56·1	173	78·3	215	101·7	530	276·7
134	56·7	174	78·9	220	104·4	540	282·2
135	57·2	175	79·4	225	107·2	550	287·8
136	57·8	176	80·0	230	110·0	560	293·3
137	58·3	177	80·6	235	112·8	570	298·9
138	58·9	178	81·1	240	115·6	580	304·4
139	59·4	179	81·7	245	118·3	590	310·0
140	60·0	180	82·2	250	121·1	600	315·6
141	60·6	181	82·8	255	123·9	610	321·1
142	61·1	182	83·3	260	126·7	620	326·7
143	61·7	183	83·9	265	129·4	630	332·2
144	62·2	184	84·4	270	132·2	640	337·8
145	62·8	185	85·0	275	135·0	650	343·3
146	63·3	186	85·6	280	137·8	660	348·9
147	63·9	187	86·1	285	140·6	670	354·4
148	64·4	188	86·7	290	143·3	680	360·0
149	65·0	189	87·2	295	146·1	690	365·6
150	65·6	190	87·8	300	148·9	700	371·1
151	66·1	191	88·3	310	154·4	710	376·7
152	66·7	192	88·9	320	160·0	720	382·2
153	67·2	193	89·4	330	165·6	730	387·8
154	67·8	194	90·0	340	171·1	740	393·3
155	68·3	195	90·6	350	176·7	750	398·9
156	68·9	196	91·1	360	182·2	760	404·4
157	69·4	197	91·7	370	187·8	770	410·0
158	70·0	198	92·2	380	193·3	780	415·6
159	70·6	199	92·8	390	198·9	800	426·7

Table 14.7 Velocity pressure
in pascals for air at 1·2 kg/m³ density

Velocity m/s	0·0	0·1	0·2	0·3	0·4	0·5	0·6	0·7	0·8	0·9
0	0·00	0·01	0·02	0·05	0·10	0·15	0·22	0·29	0·38	0·49
1	0·60	0·73	0·86	1·01	1·18	1·35	1·54	1·73	1·94	2·17
2	2·40	2·65	2·90	3·17	3·46	3·75	4·06	4·37	4·70	5·05
3	5·40	5·77	6·14	6·53	6·94	7·35	7·78	8·21	8·66	9·13
4	9·60	10·09	10·58	11·09	11·62	12·15	12·70	13·25	13·82	14·41
5	15·00	15·61	16·22	16·85	17·50	18·15	18·82	19·49	20·18	20·89
6	21·60	22·33	23·06	23·81	24·58	**25·35**	26·14	26·93	27·74	28·57
7	29·40	30·25	31·10	31·97	32·86	33·75	34·66	35·57	36·50	37·45
8	38·40	39·37	40·34	41·33	42·34	43·35	44·38	45·41	46·46	47·53
9	48·60	49·69	50·78	51·89	53·02	54·15	55·30	56·45	57·62	58·81
10	60·00	61·21	62·43	63·65	64·90	66·15	67·42	68·69	69·98	71·29
11	72·60	73·93	75·26	76·61	77·98	79·35	80·74	82·13	83·54	84·97
12	86·40	87·85	89·30	90·77	92·26	93·75	95·26	96·77	98·30	99·85
13	101·40	102·97	104·54	106·23	107·74	109·35	110·98	112·61	114·26	115·93
14	117·60	119·29	120·98	122·69	124·42	126·15	127·90	129·65	131·42	133·21
15	135·00	136·81	138·62	140·45	142·30	144·15	146·02	147·89	149·78	151·69
16	153·60	155·53	157·46	159·41	161·38	163·35	165·34	167·33	169·34	171·37
17	173·40	175·45	177·50	179·57	181·66	183·75	185·86	187·97	190·10	192·25
18	194·40	196·57	198·74	200·93	203·14	205·35	207·58	209·81	212·06	214·33
19	216·60	218·89	221·18	223·49	225·82	228·15	230·50	232·85	235·22	237·61
20	240·00	242·41	244·82	247·25	249·70	252·15	254·62	257·09	259·58	262·09

Example in **heavy type** velocity pressure at 6·5m/s = 25·35 Pa. At density ρkg/m³ the velocity pressure = (tabulated Pa) × ρ/1·2.
To extend to higher velocities note that at 10 × (tabulated m/s) the velocity pressure equals 100 × (tabulated Pa).

Table 14.7 Velocity pressure
in pascals for air at 1·2 kg/m³ density—*continued*

Velocity m/s	0·0	0·1	0·2	0·3	0·4	0·5	0·6	0·7	0·8	0·9
21	264·60	267·13	269·66	272·21	274·78	277·35	279·94	282·53	285·14	287·77
22	290·40	293·05	295·70	298·37	301·06	303·75	306·46	309·17	311·90	314·65
23	317·40	320·17	322·94	325·73	328·54	331·35	334·18	337·01	339·86	342·73
24	345·60	348·49	351·38	354·29	357·22	360·15	363·10	366·05	369·02	372·01
25	375·00	378·01	381·02	384·05	387·10	390·15	393·22	396·29	399·38	402·49
26	405·60	408·73	411·86	415·01	418·18	421·35	424·54	427·73	430·94	434·17
27	437·40	440·65	443·90	447·17	450·46	453·75	457·06	460·37	463·70	467·05
28	470·40	473·77	477·14	480·53	483·94	487·35	490·78	494·21	497·66	501·13
29	504·60	508·09	511·58	515·09	518·62	522·15	525·70	529·25	532·82	536·41
30	540·00	543·61	547·22	550·85	554·50	558·15	561·82	565·49	569·18	572·89
31	576·60	580·33	584·06	587·81	591·58	595·35	599·14	602·93	606·74	610·57
32	614·40	618·25	622·10	625·97	629·86	633·75	637·66	641·57	645·50	649·45
33	653·40	657·37	661·34	665·33	669·34	673·35	677·38	681·41	685·46	689·53
34	693·60	697·69	701·78	705·89	710·02	714·15	718·30	722·45	726·62	730·81
35	735·00	739·21	743·42	747·65	751·90	756·15	760·42	764·69	768·98	773·29
36	777·60	781·93	786·26	790·61	794·98	799·35	803·74	808·13	812·54	816·97
37	821·40	825·85	830·30	834·77	839·26	843·75	848·26	852·77	857·30	861·85
38	866·40	870·97	875·54	880·13	884·74	889·35	893·98	898·61	903·26	907·93
39	912·60	917·29	921·98	926·69	931·42	936·15	940·90	945·65	950·42	955·21
40	960·00	964·81	969·62	974·45	979·30	984·15	989·02	993·89	998·78	1003·69

Table 14.8

Relative humidities—per cent

Dry-bulb temp. °C	Wet-bulb depression (°C DB–°C WB)														
	1°	2°	3°	4°	5°	6°	7°	8°	9°	10°	12°	14°	16°	18°	20°
0°	82	64	48	32	16	2									
2°	84	69	54	39	24	11	0								
4°	85	70	56	43	32	19	7								
6°	87	72	59	46	34	23	15	4							
8°	87	75	62	50	39	28	18	8	2						
10°	88	76	65	54	44	33	23	14	5						
12°	89	78	68	57	48	38	29	20	11	3					
14°	90	79	70	60	51	42	33	25	17	9					
16°	90	81	71	62	54	46	37	30	22	14	0				
18°	91	82	73	65	56	49	41	34	27	20	6				
20°	91	83	74	66	59	51	44	37	30	24	11	0			
22°	92	83	76	68	61	54	47	40	34	28	16	5			
24°	92	84	77	69	62	56	49	43	37	31	20	9	0		
26°	92	84	77	70	63	57	51	45	39	34	24	14	5		
28°	92	85	78	72	65	58	53	47	42	36	27	17	8	0	
30°	93	86	79	73	67	61	55	49	44	39	29	21	12	4	
32°	93	87	80	73	67	62	57	51	46	41	32	23	15	8	1
34°	93	87	81	74	69	63	58	53	48	43	35	26	19	11	4
36°	94	87	81	75	70	64	59	54	50	45	37	29	21	14	8
38°	94	88	81	76	71	66	61	56	51	47	38	31	24	17	11
40°	94	88	82	77	72	67	62	57	53	48	40	33	26	19	13
42°	94	89	83	78	72	67	63	58	54	50	42	34	28	22	16
44°	94	89	83	78	73	68	64	59	55	51	43	36	30	24	18
46°	94	89	84	78	74	69	64	60	56	52	45	38	32	26	20
48°	94	89	84	79	74	70	65	61	57	53	46	39	33	27	22
50°	94	89	84	79	75	71	66	62	58	54	47	40	34	29	23

The wet-bulb depressions are for sling-type or other hygrometers in which the air is in motion over the thermometer bulbs.

$$\text{Relative humidity} = \frac{100}{p_{sdb}}\left[p_{swb} - 66{\cdot}6\left(\frac{p_o}{100{,}000}\right)\left(t_{db} - t_{wb}\right)\right]\%$$

t_{db} and t_{wb} are the dry-bulb and wet-bulb temperatures in °C p_{sdb} and p_{swb} are the saturation vapour pressures in Pa at t_{db} and t_{wb} respectively. p_o is the barometric pressure in Pa. The table is correct for a barometric pressure of about 10^5 Pa.

Table 14.9 Properties of air and water

At a pressure of 100 kPa, i.e. atmospheric pressure (1,000 mb)

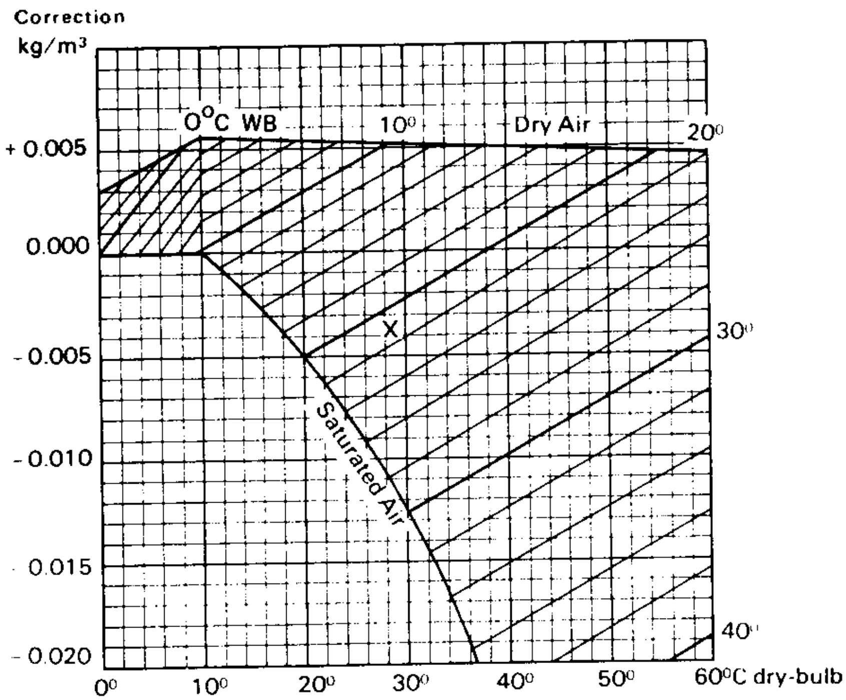
	Temp. °C	Density $kg\ m^3$	Ratio of specific heats C_p/C_v	Specific heat at constant pressure kJ/kg °C	Thermal conductivity W/m °C	Dynamic viscosity Pa s
Dry air	−100°	2·014	1·406	1·003	0·016	$10·0 \times 10^{-6}$
	−50°	1·562	1·404	1·004	0·020	$14·9 \times 10^{-6}$
	−20°	1·377	1·402	1·004	0·022	$16·2 \times 10^{-6}$
	0°	1·276	1·402	1·005	0·024	$17·3 \times 10^{-6}$
	20°	1·189	1·401	1·006	0·026	$18·2 \times 10^{-6}$
	50°	1·079	1·399	1·007	0·028	$19·6 \times 10^{-6}$
	100°	0·934	1·397	1·011	0·032	$22·0 \times 10^{-6}$
	200°	0·737	1·391	1·025	0·039	$26·1 \times 10^{-6}$
	300°	0·608	1·380	1·046	0·046	$29·8 \times 10^{-6}$
	400°	0·518	1·370	1·070	0·053	$33·2 \times 10^{-6}$
	500°	0·451	1·357	1·092	0·060	$36·4 \times 10^{-6}$
	600°	0·399	1·347	1·116	0·067	$39·4 \times 10^{-6}$
	Temp. °C	**Density kg/m^3**	**Vapour pressure k Pa**	**Specific heat at constant pressure kJ/kg °C**	**Thermal conductivity W/m °C**	**Dynamic viscosity Pa s**
Ice	−10°	920	0·26	2·05	2·3	10^{13}
Water	0°	999·8	0·61	4·217	0·56	1782
	10°	999·7	1·23	4·192	0·57	1306
	20°	998·2	2·34	4·182	0·59	1002
	30°	995·6	4·24	4·178	0·60	798
	40°	992·2	7·38	4·178	0·62	653
	50°	988·0	12·33	4·180	0·63	547
	60°	983·2	19·92	4·184	0·64	467
	70°	977·7	31·16	4·189	0·66	404
	80°	971·8	47·36	4·196	0·67	355
	90°	965·3	70·11	4·205	0·69	314
	100°	958·3	101·33	4·217	0·70	281
Steam	100°	0·581	101·3	2·01	0·023	$12·1 \times 10^{-6}$
	120°	0·552	198·5	2·01	0·025	$12·9 \times 10^{-6}$
	140°	0·525	361·2	2·00	0·027	$13·8 \times 10^{-6}$
	160°	0·501	617·7	1·98	0·028	$14·6 \times 10^{-6}$
	180°	0·479	1001·9	1·98	0·029	$15·4 \times 10^{-6}$
	200°	0·458	1553	1·97	0·031	$16·2 \times 10^{-6}$
	240°	0·423	3344	1·98	0·034	$17·8 \times 10^{-6}$
	280°	0·392	6413	2·00	0·036	$19·4 \times 10^{-6}$
	320°	0·366	11279	2·02	0·039	$21·0 \times 10^{-6}$
	360°	0·343	18656	2·04	0·041	$22·5 \times 10^{-6}$

Ratio of specific heats for steam C_p/C_v = 1·334 at 100°C.

Table 14.10
Density of atmospheric air with 0·75% moisture by weight

°C dry-bulb	Density in kg/m³ at a barometric pressure in kPa (kilopascals) of									
	94	95	96	97	98	99	100	101	102	103
0*	1·197	1·210	1·222	1·235	1·248	1·260	1·273	1·286	1·298	1·312
1*	1·192	1·205	1·218	1·231	1·243	1·256	1·269	1·281	1·293	1·306
2*	1·088	1·200	1·213	1·226	1·238	1·251	1·264	1·276	1·288	1·301
3*	1·083	1·196	1·208	1·221	1·234	1·246	1·259	1·271	1·284	1·297
4*	1·179	1·191	1·204	1·216	1·229	1·241	1·254	1·267	1·279	1·292
5*	1·174	1·187	1·199	1·212	1·224	1·237	1·249	1·262	1·274	1·287
6*	1·170	1·182	1·195	1·207	1·219	1·232	1·244	1·257	1·269	1·282
7*	1·166	1·178	1·190	1·203	1·215	1·227	1·240	1·252	1·264	1·277
8*	1·161	1·173	1·185	1·198	1·210	1·222	1·235	1·247	1·259	1·272
9*	1·156	1·169	1·181	1·193	1·206	1·218	1·230	1·242	1·255	1·267
10	1·152	1·164	1·176	1·189	1·201	1·213	1·225	1·238	1·250	1·262
11	1·148	1·160	1·172	1·184	1·197	1·209	1·221	1·233	1·245	1·258
12	1·144	1·156	1·168	1·180	1·193	1·205	1·217	1·229	1·241	1·254
13	1·140	1·152	1·164	1·177	1·189	1·201	1·213	1·225	1·237	1·249
14	1·136	1·148	1·160	1·172	1·184	1·196	1·208	1·220	1·232	1·244
15	1·132	1·144	1·156	1·168	1·180	1·192	1·204	1·216	1·228	1·240
16	1·128	1·140	1·152	1·164	1·176	1·188	1·200	1·212	1·224	1·236
17	1·124	1·136	1·148	1·160	1·172	1·184	1·196	1·208	1·220	1·232
18	1·120	1·132	1·144	1·156	1·168	1·180	1·192	1·204	1·216	1·227
19	1·116	1·128	1·140	1·152	1·164	1·176	1·188	1·199	1·211	1·223
20	1·113	1·124	1·136	1·148	1·160	1·172	1·184	1·195	1·207	1·219
21	1·109	1·121	1·132	1·144	1·156	1·168	1·180	1·191	1·203	1·215
22	1·105	1·117	1·128	1·140	1·152	1·164	1·176	1·187	1·199	1·211
23	1·101	1·113	1·125	1·136	1·148	1·160	1·172	1·183	1·195	1·207
24	1·097	1·109	1·121	1·133	1·144	1·156	1·168	1·179	1·191	1·203
25	1·094	1·105	1·117	1·129	1·140	1·152	1·164	1·175	1·187	1·199
26	1·090	1·101	1·113	1·125	1·136	1·148	1·160	1·171	1·183	1·195
27	1·087	1·098	1·110	1·121	1·132	1·144	1·156	1·167	1·179	·1191
28	1·083	1·094	1·106	1·117	1·129	1·141	1·152	1·164	1·175	1·187
29	1·080	1·091	1·103	1·114	1·125	1·137	1·148	1·160	1·171	1·183
30	1·076	1·087	1·099	1·110	1·121	1·133	1·145	1·156	1·167	1·179
31	1·072	1·084	1·095	1·106	1·118	1·129	1·141	1·152	1·163	1·175
32	1·069	1·080	1·091	1·103	1·114	1·126	1·137	1·148	1·160	1·171
33	1·065	1·077	1·088	1·099	1·111	1·122	1·133	1·145	1·156	1·167
34	1·062	1·073	1·084	1·096	1·107	1·118	1·130	1·141	1·152	1·163
35	1·058	1·070	1·081	1·092	1·103	1·115	1·126	1·137	1·148	1·160
36	1·055	1·066	1·077	1·089	1·100	1·111	1·122	1·134	1·145	1·156
37	1·051	1·063	1·074	1·085	1·096	1·107	1·119	1·130	1·141	1·152
38	1·048	1·059	1·070	1·082	1·093	1·104	1·115	1·126	1·137	1·148
39	1·045	1·056	1·067	1·078	1·089	1·100	1·111	1·122	1·134	1·145
40	1·041	1·052	1·063	1·075	1·086	1·097	1·108	1·119	1·130	1·141
mm Hg	705	712·5	720	727·5	735	742·5	750	757·5	765	772·5
mb	940	950	960	970	980	990	1000	1010	1020	1030

*Saturated below 10°C.



Density correction to be added to (+) or subtracted from (–) the density given by Table 14:10.

Fig. 14.1 Density corrections for humidity of atmospheric air.

Intemperate climates the density of atmospheric air maybe determined from measurements of barometric pressure and dry-bulb temperature alone, using Table 14.10, provided errors up to ±0.5% are permissible. The table is calculated for a representative moisture content of 0.75% by weight except below 10°C where the saturation content is less than 0.75% and saturation density is used for the tabulation.

In hot, humid climates (and in any case where greater accuracy is required) an additional measurement of wet-bulb temperature should be made, using a sling or other type of forced-convection thermometer. The correction derived from Fig. 14.1 will then be applied to the value obtained from Table 14.10, and the result will be within ± 0.1% of the true value over the tabulated range of pressure.

Example. The measured air condition is 29°C dry-bulb, 986 mb. From Table 14.10 the density is 1.125 at 98 kPa, 1.137 at 99 kPa. Density at 98.6 kPa = 1.125 + 0.6 (1.137 – 1.125) = 1.132 ±0.006 kg/m^3.

Suppose an additional measurement gives a wet-bulb temperature of 21 °C. The point marked X on Fig. 14.1 corresponds to 29°C DB, 21 °C WB and gives a correction of – 0.003 kg/m^3. The revised air density is:

$$1{\cdot}132 - 0{\cdot}003 = 1{\cdot}129 \pm 0{\cdot}001 \text{ kg/m}^3$$

Table 14.11 Properties of gases

At atmospheric temperature and pressure (20 °C 100 kPa) except where otherwise stated.

Gas or vapour		Relative density Air=1	Boiling temperature at one atmosphere °C	Specific heat at constant pressure kJ/kg °C	Ratio of specific heats $\gamma = C_p/C_v$	Thermal conductivity W/m °C	Dynamic viscosity Pa s	Velocity of sound in gas m/s
Air (dry)		1·000		1·01	1·401	0·026	$18{\cdot}2\times10^{-6}$	344
Ammonia	NH_3	0·597	−33	2·10	1·336	0·024		430
Butane	C_4H_{10}	2·015	0	1·64		0·015		
Carbon dioxide	CO_2	1·53	−78	0·84	1·300	0·016	$14{\cdot}7\times10^{-6}$	268
Carbon monoxide	CO	0·971	−192	1·04	1·401	0·025	$17{\cdot}5\times10^{-6}$	350
Chlorine	Cl_2	2·41	−34	0·48	1·23	0·009	$13{\cdot}2\times10^{-6}$	219
Hydrogen	H_2	0·070	−253	14·30	1·407	0·177	$8{\cdot}8\times10^{-6}$	1330
Methane	CH_4	0·556	−161	2·22	1·313	0·032	$11{\cdot}0\times10^{-6}$	445
Nitrogen	N_2	0·971	−196	1·04	1·401	0·025	$17{\cdot}6\times10^{-6}$	350
Oxygen	O_2	1·108	−183	0·92	1·400	0·026	$20{\cdot}4\times10^{-6}$	326
Propane	C_3H_8	1·53	−42	1·67	1·130	0·016		246
Water vapour	H_2O	0·622	+100	1·86	1·334	0·023*	$12{\cdot}1\times10^{-6}$*	405*
Town gas (typical)		0·47						
Natural gas (N. Sea)		0·62						

*At 100 °C.

The ratio of gas density to air density remains constant over a wide range of pressure and temperature.
All the above quantities except the boiling temperature are independent of pressure over a wide range.
Specific heats increase by 1% to 5% for the permanent gases and up to 25% for propane and butane over range 20 °C to 120 °C.
Thermal conductivities increase by 25% to 50% between 20 °C and 120 °C.
Dynamic viscosities increase by 25% to 35% between 20 °C and 120 °C.
Velocities of sound increase by 16% approximately between 20 °C and 120 °C.

Table 14.12 Properties of liquids

At 20°C and atmospheric pressure

Liquid		Density kg/m³	Cubical expansion coeff. per °C	Freezing point temp. °C	Boiling point temp. °C	Heat of vaporisation kJ/kg	Specific heat kJ/kg °C	Thermal conductivity W/m °C	Viscosity Pa s
Alcohol (ethyl)	C_2H_6O	789	$1{\cdot}08\times10^{-3}$	−117°	78°	868	2·50	0·17	$1{\cdot}19\times10^{-3}$
Benzene	C_6H_6	879	$1{\cdot}21\times10^{-3}$	+5°	80°	394	1·54	0·15	$0{\cdot}65\times10^{-3}$
Ether (diethyl)	$C_4H_{10}O$	714	$1{\cdot}63\times10^{-3}$	−116°	34°	358	2·34	0·14	$0{\cdot}24\times10^{-3}$
Glycerine	$C_3H_8O_3$	1 260	$0{\cdot}47\times10^{-3}$	+20°	290°	243	2·43	0·29	$1{\cdot}49\times10^{-3}$
Mercury	Hg	13 546	$0{\cdot}18\times10^{-3}$	−99°	357°	284	0·14	8·10	$1{\cdot}55\times10^{-3}$
Turpentine (pinene)	$C_{10}H_{16}$	861	$0{\cdot}96\times10^{-3}$	−40°	155°	290	1·76	0·12	$1{\cdot}49\times10^{-3}$
Water (pure)	H_2O	998	$0{\cdot}21\times10^{-3}$	0°	100°	2257	4·18	0·59	$1{\cdot}00\times10^{-3}$
Ethylene Glycol/water	50%	1 065		−35°			3·20	0·20	$4{\cdot}0\times10^{-3}$
Kerosene		790						0·16	$2{\cdot}4\times10^{-3}$
Gas oil		830							$4{\cdot}0\times10^{-3}$
Light fuel oil		930							
Heavy fuel oil		970							

Table 14.13. Properties of plastics on the following page, is a generalised data summary for the principal groups of structural plastics. Each type is available in a number of grades designed to optimise one property or another and there may be further modifications provided by the use of fillers, plasticisers or fibre reinforcement.

Coefficients of linear expansion lie between 2 and 20 × 10^{-5} per °C, and may in some cases be selected by grade.

Thermal conductivities lie mainly between 0·12 and 0·25 W/m °C, but may be lower (polystyrene) or higher (polythene).

Specific heat capacities lie between 1·0 (PTFE and epoxy resin) and 1·9 (polypropylene) kJ/kg °C.

Electrical resistivities are typical quotations for clean, dry specimens at room temperature, but may vary widely in practice.

Table 14.13 Properties of plastics

See footnote on previous page.

Polymer type (common or trade name)	Typical density kg/m^3	Tensile strength MPa	Electrical resistivity		Maximum working °C†	*Resistance to		
			Volume ohm.m	Surface ohm/sq		Acids	Alkalis	Solvents
Acetal resins	1420	65–80	10^{13}		80	3	3	1
Acrylonitrile-butadiene-styrene (ABS)	1070	48–55			90–110	2	2	3
Amino plastics (melamine)	1500	45–90	10^{9}	10^{12}	100–140	3	3	1
Cellulose acetate	1300	30–50	10^{9}	10^{11}	60–110	3	3	4
Epoxy resin—cast	1300	25–85	10^{12}	10^{8}+	80	2	1	2
Phenol-formaldehyde (bakelite)	1350	45–52	10^{9}	10^{8}+	120–170	2	3	2
Polyamides—Nylon 6/6	1150	60–80	10^{10}	10^{11}+	80–150	2	1	2
Polycarbonates	1200	50–65	10^{14}		140	2	3	3
Polyesters—glass reinforced	1500/ 2000	200–350	10^{9}	10^{13}	120–200	3	3	3
Polyethylenes—low density	920	6–15	10^{14}	10^{14}+	60–75	2	1	2
(polythene)—high density	950	20–35	10^{14}	10^{12}+	90–100	2	1	2
Polypropylene, copolymer	905	30–40	10^{14}	10^{15}+	190–240	2	1	1
Polystyrene	1060	30–70	10^{15}	10^{14}+	65–80	2	1	3
Polytetrafluoroethylene (PTFE)	2170	13–33	10^{16}	10^{13}+	260	1	1	1
Polyvinylchloride, rigid (PVC)	1350	40–60	10^{13}	10^{12}	70–75	2	3	3
Silicone resin, mineral filled	1800/ 2800		10^{11}	10^{11}+	290	2	3	1

*1. Unaffected by acids/alkalis/solvents.
*2. Resistant except to some strong acids/alkalis/solvents.
3. Resistant only to weak acids/alkalis/solvents.
4. Readily attacked by acids/alkalis/solvents.

† Unloaded. Distortion or creep under load may occur at much lower temperatures.

Table 14.14 Properties of structrural materials

Material (purity or percentage composition)	Density kg/m³	Linear expansion coefficient $\times 10^{-6}$ per °C	Melting point temperature °C	Specific heat kJ/kg °C	Thermal conductivity W/m °C	Modulus elasticity kN/mm²	Ultimate strength N/mm²	Yield of proof stress N/mm²
Aluminium (99·5% Al) $\frac{1}{2}$ hard	2700	24	660	0·93	220	70	100	75
Aluminium (12 Si) chill cast	2630	20	580	0·88	150	72	200	75
Aluminium (10 Mg) cast and heat treated	2550	24	500	0·88	85	72	310	200
Copper, high conductivity strip, $\frac{1}{2}$ hard	8890	16	1080	0·38	380	110	250	200
Brass (63 Cu 37 Zn) $\frac{1}{2}$ hard	8500	20	940	0·37	120	103	400	360
Bronze, phosphor (7 Sn 0·3 P) hard	8850	18	1050	0·38	80	103	600	450
Iron, mid-grade grey cast, 20mm thick	7200	11	1100	0·50	45	120	250	—
Iron, malleable cast	7300	12	1150	0·51	45	170	400	260
Iron (3·6 C, 2·0 Si) hard grade S.G.*	7100	12	1200	0·50	45	160	720	460
Magnesium (4 Al 1·3 Mn) forged	1770	28	650	1·04	95	45	260	160
Titanium (5 Al 2·5 Sn)	4500	10	1600	0·44	16	110	800	750
Tungsten, hard drawn 0·1 mm wire	19250	4	3380	0·14	185	410	3500	—
Steel, structural carbon	7850	11	1500	0·48	50	200	400	250
Steel, stainless (18 Cr 8 Ni) $\frac{1}{2}$ hard sheet	7930	17	1450	0·50	16	200	1000	720
Steel (0·3C 4Ni 1·2Cr 0·3Mo) heat treated	7820	12	1500	0·48	12	205	1550	1300
Lead (4Sb) sheet	11100	28	300	0·13	30	16	40	3
Tin (99·5 Sn)	7300	21	232	0·23	64	4	14	1·4
Zinc (99Zn 1 Cu) $\frac{1}{2}$ hard, rolled	7140	28	419	0·38	120	110	180	70
Timber, white deal	610	L5 a 40	—	1·70	a 0·13	L14	L30–70	—
Timber, oak, beech	740	L5 a 40	—	1·70	a 0·16	L11	L60–110	—
Glass (soda-lime) crown sheet	2500	8·5	1100	0·70	0·70	70	30–90	—
Concrete (1 : $2\frac{1}{2}$: $3\frac{1}{2}$)	2000–2400	10	—	1·00	~1·5	20	c 17	—
Brick, medium grade	1900–2300	5	—	0·80	~1·0	16	c 24	—
Granite	2700	6	—	0·80	~2·5	50	c170	—

*SG = spheroidal graphite. a = across the grain. L = along the grain. c = compressive.

Table 14.15
Kinematic viscosity grades for liquids

ISO viscosity grade	Kinematic viscosity cSt	Engler degrees	Redwood No. 1 seconds	Redwood No. 2 seconds	ASTM grade (Saybolt) universal seconds	Saybolt Furol seconds
	1·0	1·00	28·5			
	1·5	1·06	29·7			
VG 2	2·2	1·14	31·4		32	
VG 3	3·2	1·24	33·5		36	
VG 5	4·6	1·35	37·0		40	
VG 7	6·8	1·55	42·5		50	
VG 10	10	1·83	52		60	
VG 15	15	2·32	68		75	
VG 22	22	3·12	94		105	
VG 32	32	4·34	133		150	
VG 46	46	6·12	190		215	24
VG 68	68	8·95	277		315	33
VG 100	100	13·2	405		465	47
VG 150	150	19·7	605		700	70
VG 220	220	29·0	890		1000	102
VG 320	320	42·0	1300	127	1500	148
VG 460	460	60·5	1870	182	2150	213
VG 680	680	89·5	2750	268	3150	315
VG 1000	1000	132	4060	395	4650	465
VG 1500	1500	197	6090	590	7000	700
	2200	290	8950	870		1020
	3200	420	13000	1260		1480

1 cSt (centistokes) = 1mm^2/s = 10^{-6} m^2/s (Kinematic viscosity)
1 cP (centipoise) = 10^{-3} Pa s (Dynamic viscosity)
Dynamic viscosity (Pa s) = Kinematic viscosity (m^2/s) × Density (kg/m^3)
Dynamic viscosity (cP) = Kinematic viscosity (c St) × Relative density (water = 1)

The VG viscosity grades specified in BS *4231 :1967* and ISO *3448: 1976* are equivalent to the corresponding American ASTM grades (Saybolt universal seconds) and are particularly suitable for the classification (with a ± 10% tolerance) of lubricating oils.

The VG grade of a liquid is now established from a centistokes measurement at *40°C.* Formerly the temperature was 100°F *(37.8°C)* which is still the basis of the ASTM. Saybolt and Redwood scales; the difference is only a few per cent. The equivalences in the table are substantially true at either temperature.

The SAE grades of crankcase and gearbox oils are not directly comparable; being determined at 0 °F and 210°F.

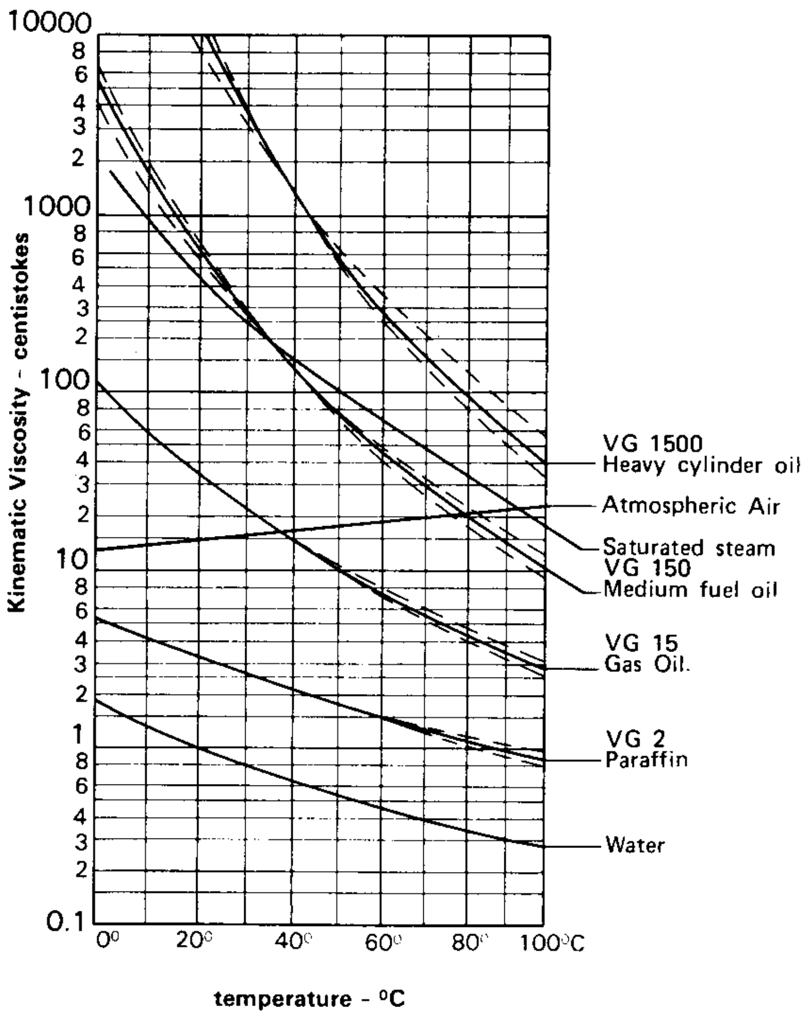


Fig. 14.2

Kinematic viscosities. Variation with temperature.

Values are for a pressure of one atmosphere except in the case of saturated steam where the pressure is that of saturated vapour over water at the temperature quoted (air being absent). See Fig. 12.11.

The full lines for VG grade oils are at a viscosity index (VI) of 50. The broken lines are for 0 VI (steeper slope) and 95 VI (shallower slope).

The temperature gradient for most pure gases is similar to that for atmospheric air.

Table 14.16 Climatic data

The temperatures quoted are those generally accepted for the design of heating and air conditioning systems. They are not the recorded extremes and are quite likely to be exceeded on a few occasions in an average year. The data are mainly extracted from the IHVE Guide, Book A, which should be consulted for comprehensive information.

The altitude is that of the meteorological station. Sea-level temperatures in the same area would be expected to be higher by about 1 °C per 165m difference in altitude. The column headed "% Moist." is the percentage moisture by weight at the design condition.

Location and latitude		Altitude	Cold season	Hot season			Annual rainfall
		m	°C D-B	°C D-B	°C W-B	% Moist.	mm
United Kingdom:							
Aberdeen	57°N	14	−1	22	16	0·9	750
Belfast	54°N	66	−1	24	17	0·9	
Birmingham	52°N	163	−1	27	19	1·1	
Bristol	51°N	10	−1	26	19	1·1	
Glasgow	56°N	9	−1	24	18	1·0	
Liverpool	53°N	60	−1	26	18	1·0	
London	51°N	5	−1	28	19	1·0	580
Northern Europe:							
Amsterdam	52°N	2	−7	28	19	1·0	650
Copenhagen	56°N	13	−12	28	20	1·1	592
Helsinki	60°N	9	−27	27	19	1·0	701
Oslo	60°N	94	−17	27	19	1·0	683
Reykjavik	64°N	23	−18	19	14	0·8	861
Stockholm	59°N	44	−19	27	19	1·0	569
Western Europe:							
Brussels	50°N	100	−10	31	21	1·1	838
Bordeaux	45°N	48	−2	34	23	1·3	
Geneva	46°N	404	−12	32	22	1·2	861
Lisbon	39°N	95	+3	34	22	1·1	686
Luxembourg	50°N		−14	31	21	1·1	
Marseilles	43°N	75	−2	33	22	1·2	575
Paris	49°N	50	−5	32	21	1·1	566
Central Europe:							
Berlin	52°N	57	−18	32	21	1·1	587
Budapest	48°N	120	−15	34	22	1·2	615
Hamburg	54°N	20	−15	28	19	1·0	
Munich	48°N	529	−18	29	19	1·0	
Prague	50°N	202	−16	32	19	0·8	490
Vienna	48°N	202	−15	31	21	1·1	650

Table 14.16 Climatic data—*continued*

Location and latitude		Altitude m	Cold season °C D-B	Hot season °C D-B	Hot season °C W-B	Hot season % Moist.	Annual rainfall mm
Southern Europe:							
Athens	38°N	107	− 1	37	22	1·0	401
Barcelona	41°N	95	+ 2	32	24	1·6	
Belgrade	45°N	138	− 13	37	23	1·2	625
Madrid	40°N	666	− 4	36	22	1·0	419
Milan	45°N	142		34	23	1·3	803
Nicosia	35°N	213		40	24	1·2	363
Rome	42°N	115	− 1	36	23	1·2	653
Palermo	38°N	108		37	25	1·5	709
Middle East:							
Ankara	40°N	861	− 14	36	21	0·9	345
Baghdad	33°N	34	+ 2	47	24	0·9	140
Beirut	34°N	34	+ 4	33	26	1·8	892
Istanbul	41°N	18	− 4	34	23	1·3	800
Jerusalem	32°N	757	− 1	36	21	0·9	528
Kuwait City	29°N	5	+ 4	45	31	2·3	130
Tehran	36°N	1220	− 4	40			246
North Africa:							
Alexandria	31°N	32		37	24	1·3	
Algiers	37°N	59	+ 3	37	26	1·7	762
Benghazi	32°N	25		38	25	1·4	267
Cairo	30°N	116	+ 3	41	22	0·8	28
Casablanca	33°N	240		33	25	1·7	404
Dakhla Oasis	25°N	122	0	45	25	1·1	Nil
Tunis	37°N	66		42	27	1·6	419
Aden	13°N	7		39	29	2·1	23
Addis Ababa	9°N	2445	+ 1	27	19	1·0	1237
Djibouti	12°N	7		44	31	2·3	130
Khartoum	16°N	390	+ 8	45	24	1·0	157
Central Africa:							
Freetown	8°N	11	+ 18	33	27	2·0	3495
Kampala	0°	1310		32	23	1·4	
Kinshasa	4°S	325		35	28	2·1	
Lagos	6°N	3	+ 18	34	28	2·2	1836
Mombasa	4°S	16		33	26	1·8	
Nairobi	1°S	1819	+ 3	28	18	0·9	958

Table 14.16 Climatic data—*continued*

Location and latitude		Altitude m	Cold season °C D-B	Hot season °C D-B	Hot season °C W-B	Hot season % Moist.	Annual rainfall mm
Southern Africa:							
Capetown	34°S	17	+1	34	22	1·1	508
Dar-es-Salaam	7°S	14	+16	33	28	2·2	1064
Durban	30°S	5	+7	31	24	1·6	1080
Johannesburg	26°S	1668		31	21	1·1	709
Luanda	9°S	59	+15	32	27	2·1	315
Salisbury	18°S	1470	0	32	21	1·1	810
Windhoek	23°S	1728	0	33	19	0·8	363
E. Europe, Siberia							
Bucharest	44°N	82	−20	36	22	1·1	579
Irkutsk	52°N	466	−40	33	22	1·2	380
Leningrad	60°N	5	−25	28	19	1·0	488
Moscow	56°N	154	−26	31	21	1·1	630
Vladivostok	43°N	29	−25	29	22	1·4	599
Warsaw	52°N	120	−20	32	21	1·1	559
Far East:							
Hong Kong	22°N	33	+7	33	28	2·2	2162
Manila	15°N	14	+17	36	28	2·1	2083
Seoul	38°N	87		35	26	1·8	1250
Shanghai	31°N	7		37	28	2·0	1135
Tientsin	39°N	4		38	28	2·0	533
Tokyo	36°N	6	−2	33	26	1·8	1565
Southern Asia:							
Bombay	18°N	11	+15	34			1808
Calcutta	23°N	6	+8	39			1600
Colombo	7°N	7	+17	33	27	2·0	2365
Karachi	25°N	4		39	30	2·3	196
Lahore	32°N	214		46	26	1·3	503
Madras	13°N	16		42			1270
New Delhi	29°N	218	+4	44			640
Rangoon	17°N	5	+14	39			2616
South-East Asia							
Bangkok	14°N	2		38	29	2·2	1397
Jakarta	1°S	8	+20	33	26	1·8	1840
Port Moresby	9°S	38	+20	34	28	2·1	1011
Saigon	11°N	9		37	29	2·2	1984
Singapore	1°N	10	+20	33	28	2·2	2413

Table 14.16 Climatic data—*continued*

Location and latitude		Altitude m	Cold season °C D-B	Hot season °C D-B	°C W-B	% Moist.	Annual rainfall mm
Australia:							
Adelaide	35°S	43	+2	42	23	1·0	536
Brisbane	27°S	42	+5	36	25	1·5	1135
Canberra	35°S	559	0	36	21	0·9	
Hobart	43°S	54	0	33	20	0·9	600
Perth	32°S	60	+2	39	24	1·2	881
Sydney	34°S	42	+7	35	24	1·4	1181
Pacific Area:							
Auckland	37°S	26	+7	26	19	1·1	1247
Christchurch	44°S	10	−1	36	19	0·7	638
Dunedin	46°S	73	+2	28	19	1·0	
Honolulu	21°N	12	+12	30	24	1·6	800
Suva, Fiji	18°S	6	+15	33	27	2·0	2974
North America:							
Chicago	42°N	251	−24	34	24	1·5	836
Edmonton	54°N	680	−40	28	20	1·1	430
Los Angeles	34°N	95	+5	36	23	1·2	381
New York	41°N	96	−15	36	24	1·4	1092
San Francisco	38°N	16	+2	29	20	1·1	561
Toronto	44°N	115	−23	33	22	1·2	815
Vancouver	49°N	14	−12	29	21	1·0	1458
Central America:							
Hamilton	32°N	46	+12	32	26	1·9	1463
Houston, Texas	30°N	19	−7	36	27	1·9	
Kingston	18°N	33	+15	34	28	2·1	800
Mexico City	19°N	2307	0	28	19	1·0	747
Miami, Florida	26°N	7	0	32	27	2·1	
South America:							
Buenos Aires	35°S	27	−1	36	26	1·7	950
Georgetown	7°N	2	+20	32	26	1·9	2253
La Paz	16°S	3655		23	14	0·6	574
Lima	12°S	120	+15	31	24	1·6	41
Maracaibo	11°N	6		37	29	2·2	577
Rio de Janeiro	23°S	61	+13	34	26	1·8	1082
Santiago	33°S	519		33	21	1·0	360

Table 14.17

Circumferences and areas of circles

Diam. mm	Circum. mm	Area m^2	Diam. mm	Circum. mm	Area m^2	Diam. mm	Circum. mm	Area m^2
100	314·2	0·00785	135	424·1	0·01431	170	534·1	0·02270
101	317·3	0·00801	136	427·3	0·01453	171	537·2	0·02297
102	320·4	0·00817	137	430·4	0·01474	172	540·4	0·02323
103	323·6	0·00833	138	433·5	0·01496	173	543·5	0·02351
104	326·7	0·00849	139	436·7	0·01517	174	546·6	0·02378
105	329·9	0·00866	140	439·8	0·01539	175	549·8	0·02405
106	333·0	0·00882	141	443·0	0·01562	176	552·9	0·02433
107	336·1	0·00899	142	446·1	0·01584	177	556·1	0·02461
108	339·3	0·00916	143	449·2	0·01606	178	559·2	0·02488
109	342·4	0·00933	144	452·4	0·01629	179	562·3	0·02516
110	345·6	0·00950	145	455·5	0·01651	180	565·5	0·02545
111	348·7	0·00968	146	458·7	0·01674	181	568·6	0·02573
112	351·9	0·00985	147	461·8	0·01697	182	571·8	0·02602
113	355·0	0·01003	148	465·0	0·01720	183	574·9	0·02630
114	358·1	0·01021	149	468·1	0·01744	184	578·1	0·02659
115	361·3	0·01039	150	471·2	0·01767	185	581·2	0·02688
116	364·4	0·01057	151	474·4	0·01791	186	584·3	0·02717
117	367·6	0·01075	152	477·5	0·01815	187	587·5	0·02746
118	370·7	0·01094	153	480·7	0·01838	188	590·6	0·02776
119	373·8	0·01112	154	483·8	0·01863	189	593·8	0·02805
120	377·0	0·01131	155	486·9	0·01887	190	596·9	0·02835
121	380·1	0·01150	156	490·1	0·01911	191	600·0	0·02865
122	383·3	0·01169	157	493·2	0·01936	192	603·2	0·02895
123	386·4	0·01188	158	496·4	0·01961	193	606·3	0·02925
124	389·6	0·01208	159	499·5	0·01986	194	609·5	0·02956
125	392·7	0·01227	160	502·7	0·02011	195	612·6	0·02986
126	395·8	0·01247	161	505·8	0·02036	196	615·8	0·03017
127	399·0	0·01267	162	508·9	0·02061	197	618·9	0·03048
128	402·1	0·01287	163	512·1	0·02087	198	622·0	0·03079
129	405·3	0·01307	164	515·2	0·02112	199	625·2	0·03110
130	408·4	0·01327	165	518·4	0·02138	200	628·3	0·03142
131	411·5	0·01348	166	521·5	0·02164	202	634·6	0·03205
132	414·7	0·01368	167	524·6	0·02190	204	640·9	0·03268
133	417·8	0·01389	168	527·8	0·02217	206	647·2	0·03333
134	421·0	0·01410	169	530·9	0·02243	208	653·5	0·03398

Table 14.17

Circumferences and areas of circles—*continued*

Diam. mm	Circum. mm	Area m^2	Diam. mm	Circum. mm	Area m^2	Diam. mm	Circum. mm	Area m^2
210	659·7	0·03464	280	879·6	0·06157	350	1099·6	0·09621
212	666·0	0·03530	282	885·9	0·06246	352	1105·8	0·09731
214	672·3	0·03597	284	892·2	0·06335	354	1112·1	0·09842
216	678·6	0·03664	286	898·5	0·06424	356	1118·4	0·09954
218	684·9	0·03732	288	904·8	0·06514	358	1124·7	0·10066
220	691·2	0·03801	290	911·1	0·06605	360	1131·0	0·10179
222	697·4	0·03871	292	917·3	0·06697	362	1137·3	0·10292
224	703·7	0·03941	294	923·6	0·06789	364	1143·5	0·10406
226	710·0	0·04011	296	929·9	0·06881	366	1149·8	0·10521
228	716·3	0·04083	298	936·2	0·06975	368	1156·1	0·10636
230	722·6	0·04155	300	942·5	0·07069	370	1162·4	0·10752
232	728·8	0·04227	302	948·8	0·07163	372	1168·7	0·10869
234	735·1	0·04300	304	955·0	0·07258	374	1175·0	0·10986
236	741·4	0·04374	306	961·3	0·07354	376	1181·2	0·11104
238	747·7	0·04449	308	967·6	0·07451	378	1187·5	0·11222
240	754·0	0·04524	310	973·9	0·07548	380	1193·8	0·11341
242	760·3	0·04600	312	980·2	0·07645	382	1200·1	0·11461
244	766·6	0·04676	314	986·5	0·07744	384	1206·4	0·11581
246	772·8	0·04753	316	992·7	0·07843	386	1212·7	0·11702
248	779·1	0·04830	318	999·0	0·07942	388	1218·9	0·11824
250	785·4	0·04909	320	1005·3	0·08042	390	1225·2	0·11946
252	791·7	0·04988	322	1011·6	0·08143	392	1231·5	0·12069
254	798·0	0·05067	324	1017·9	0·08245	394	1237·8	0·12192
256	804·2	0·05147	326	1024·2	0·08347	396	1244·1	0·12316
258	810·5	0·05228	328	1030·4	0·08450	398	1250·4	0·12441
260	816·8	0·05309	330	1036·7	0·08553	400	1256·6	0·12566
262	823·1	0·05391	332	1043·0	0·08657	405	1272·3	0·12882
264	829·4	0·05474	334	1049·3	0·08762	410	1288·1	0·13202
266	835·7	0·05557	336	1055·6	0·08867	415	1303·8	0·13526
268	841·9	0·05641	338	1061·9	0·08973	420	1319·5	0·13854
270	848·2	0·05726	340	1068·1	0·09079	425	1335·2	0·14186
272	854·5	0·05811	342	1074·4	0·09186	430	1350·9	0·14522
274	860·8	0·05896	344	1080·7	0·09294	435	1366·6	0·14862
276	867·1	0·05983	346	1087·0	0·09402	440	1382·3	0·15205
278	873·4	0·06070	348	1093·3	0·09511	445	1398·0	0·15553

Table 14.17

Circumferences and areas of circles—*continued*

Diam. mm	Circum. mm	Area m^2	Diam. mm	Circum. mm	Area m^2	Diam. mm	Circum. mm	Area m^2
450	1413·7	0·15904	600	1885·0	0·28274	750	2356·2	0·44179
455	1429·4	0·16260	605	1900·7	0·28747	755	2371·9	0·44770
460	1445·1	0·16619	610	1916·4	0·29225	760	2387·6	0·45365
465	1460·8	0·16982	615	1932·1	0·29706	765	2403·3	0·45963
470	1476·5	0·17349	620	1947·8	0·30191	770	2419·0	0·46566
475	1492·3	0·17721	625	1963·5	0·30680	775	2434·7	0·47173
480	1508·0	0·18096	630	1979·2	0·31172	780	2450·4	0·47784
485	1523·7	0·18474	635	1994·9	0·31669	785	2466·1	0·48398
490	1539·4	0·18857	640	2010·6	0·32170	790	2481·9	0·49017
495	1555·1	0·19244	645	2026·3	0·32674	795	2497·6	0·49639
500	1570·8	0·19635	650	2042·0	0·33183	800	2513·3	0·50265
505	1586·5	0·20030	655	2057·7	0·33696	810	2544·7	0·51530
510	1602·2	0·20428	660	2073·5	0·34212	820	2576·1	0·52810
515	1617·9	0·20831	665	2089·2	0·34732	830	2607·5	0·54106
520	1633·6	0·21237	670	2104·9	0·35256	840	2638·9	0·55418
525	1649·3	0·21647	675	2120·6	0·35785	850	2670·4	0·56745
530	1665·0	0·22062	680	2136·3	0·36317	860	2701·8	0·58088
535	1680·8	0·22480	685	2152·0	0·36853	870	2733·2	0·59447
540	1696·5	0·22902	690	2167·7	0·37393	880	2764·6	0·60821
545	1712·2	0·23328	695	2183·4	0·37937	890	2796·0	0·62211
550	1727·9	0·23758	700	2199·1	0·38484	900	2827·4	0·63617
555	1743·6	0·24192	705	2214·8	0·39036	910	2858·8	0·65039
560	1759·3	0·24630	710	2230·5	0·39592	920	2890·3	0·66476
565	1775·0	0·25072	715	2246·2	0·40151	930	2921·7	0·67929
570	1790·7	0·25518	720	2261·9	0·40715	940	2953·1	0·69398
575	1806·4	0·25967	725	2277·7	0·41282	950	2984·5	0·70882
580	1822·1	0·26421	730	2293·4	0·41854	960	3015·9	0·72382
585	1837·8	0·26878	735	2309·1	0·42429	970	3047·3	0·73898
590	1853·5	0·27340	740	2324·8	0·43008	980	3078·8	0·75430
595	1869·2	0·27805	745	2340·5	0·43591	990	3110·2	0·76977

Table 14.18

Preferred sizes and cross-sectional areas of ducts

Circular Renard series		Circular HVCA series		Rectangular HVCA	
Diam. mm	Area m^2	Diam. mm	Area m^2	Section mm × mm	Area m^2
50	0·00196	75	0·00442	150 × 100	0·0150
56	0·00246	100	0·00785	250 × 100	0·0250
63	0·00312	125	0·0123	200 × 150	0·0300
71	0·00396	150	0·0177	250 × 150	0·0375
80	0·00503	175	0·0241	400 × 150	0·0600
90	0·00636	200	0·0314	200 × 200	0·0400
100	0·00785	225	0·0398	300 × 200	0·0600
112	0·00985	250	0·0491	500 × 200	0·100
125	0·0123	275	0·0594	600 × 200	0·120
140	0·0154	300	0·0707	250 × 250	0·0625
160	0·0201	325	0·0830	300 × 250	0·075
180	0·0254	350	0·0962	500 × 250	0·125
200	0·0314	375	0·110	600 × 250	0·150
224	0·0394	400	0·126	500 × 300	0·150
250	0·0491	450	0·159	700 × 300	0·210
280	0·0616	500	0·196	400 × 400	0·160
315	0·0779	550	0·238	600 × 400	0·240
355	0·0990	600	0·283	700 × 400	0·280
400	0·126	650	0·332	600 × 500	0·300
450	0·159	700	0·385	700 × 500	0·350
500	0·196	750	0·442	700 × 600	0·420
560	0·246	800	0·503	800 × 600	0·480
630	0·312	900	0·636	700 × 700	0·490
710	0·396	1000	0·785	800 × 700	0·560
800	0·503	1100	0·950	800 × 800	0·640
900	0·636	1200	1·131		
1000	0·785	1300	1·327		
1120	0·985	1400	1·539		
1250	1·227	1500	1·767		
1400	1·539	1600	2·011		
1600	2·011	1700	2·270		
1800	2·545	1800	2·545		
2000	3·142				

The first table gives the first choice (**heavy type**, R10 Renard series) and second choice (R20, remainder) which are most advantageous for a fan size range (inlet and outlet connections). It is also used in some European countries for standard duct sizes.

The second and third tables give the arithmetic series adopted as standard for circular and rectangular ducts by the Heating and Ventilating Contractors' Association in the UK.

Table 14.19

Preferred thicknesses and weights of sheet material

Thickness mm	Steel	Aluminium	Unplasticised PVC	Polypropylene	Polyester/ glass-fibre
	Mass in kg/m²				
0·3	**2·35**	**0·81**	0·42	0·27	0·54
0·35	2·75	0·94	0·49	0·32	0·63
0·4	**3·14**	**1·08**	0·56	0·36	0·72
0·45	3·53	1·21	0·63	0·41	0·81
0·5	**3·93**	**1·35**	0·70	0·45	0·90
0·55	4·31	1·48	0·77	0·50	0·99
0·6	**4·71**	**1·62**	0·84	0·55	1·08
0·7	5·50	1·89	0·98	0·64	1·26
0·8	**6·30**	**2·16**	1·12	0·73	1·44
0·9	7·06	2·43	1·26	0·82	1·62
1·0	**7·85**	**2·70**	**1·40**	**0·91**	**1·80**
1·1	8·64	2·97	1·54	1·00	1·98
1·2	**9·42**	**3·24**	1·68	1·09	2·16
1·4	11·0	3·78	1·96	1·27	2·52
1·6	**12·6**	**4·32**	**2·24**	**1·45**	**2·88** ($\frac{1}{16}$")
1·8	14·1	4·86	2·52	1·63	3·24
2·0	**15·7**	**5·40**	2·80	1·82	3·60
2·2	17·3	5·95	3·08	2·00	3·96
2·5	**19·6**	**6·75**	3·50	2·27	4·50
2·8	22·0	7·55	3·92	2·54	5·04
3·0	**23·5**	**8·10**	**4·20**	**2·72**	**5·40** ($\frac{1}{8}$")
3·5	27·5	9·44	4·90	3·18	6·30
4·0	**31·4**	**10·8**	5·60	3·64	7·20
4·5	35·3	12·1	**6·30**	**4·10**	**8·10** ($\frac{3}{16}$")
5·0	**39·3**	**13·5**	7·00	4·55	9·00
5·5	43·1	14·8	7·70	5·00	9·90
6·0	**47·1**	**16·2**	**8·40**	**5·45**	**10·8** ($\frac{1}{4}$")
7·0	55·0	18·9	9·80	6·36	12·6
8·0	**63·0**	**21·6**	11·2	7·28	14·4
9·0	70·6	24·3	12·6	8·20	16·2
10·0	**78·5**	**27·0**	**14·0**	**9·10**	**18·0** ($\frac{3}{8}$")

Thicknesses quoted in **heavy type** are the first choice standards for steel and aluminium sheet, the remainder being second choice (R20 series).

The **heavy type** entries for plastic sheets are for the nearest metric thickness to the inch sizes currently available in the UK, and quoted alongside. Internationational standards are not yet established (1975).

For steel structures hot-dip galvanised after manufacture add 0·6 kg/m².

Table 14.20 Construction of sheet metal ducts

Size of duct	Low velocity ducts 0 to 10 m/s −500 to +500 Pa						Ducts for plant connections 0 to 10 m/s −1,000 to +1,000 Pa						High velocity ducts 10 to 40 m/s −500 to +2,500 Pa		
	Steel			Aluminium			Steel			Aluminium			Steel		
Rectangular Longside—mm	T	A	S	T	A	S	T	A	S	T	A	S	T	A	S
Up to 300	0·6	25	—	0·8	25	—	1·0	40	0·8	1·2	40	0·8	0·8	25	—
301 to 400	0·6	25	—	0·8	25	—	1·0	40	0·8	1·2	40	0·8	0·8	25	1·2
401 to 600	0·8	25	1·5	0·8	25	1·5	1·0	40	0·8	1·2	40	0·8	0·8	25	1·2
601 to 800	0·8	25	1·5	1·0	25	1·5	1·0	40	0·8	1·2	40	0·8	0·8	40	1·0
801 to 1000	0·8	25	1·2	1·0	40	1·2	1·0	40	0·8	1·2	50	0·8	0·8	40	0·8
1001 to 1500	1·0	40	0·8	1·2	40	0·8	1·2	50	0·6	1·6	60	0·6	1·0	50	0·6
1501 to 2250	1·0	40	0·8	1·2	50	0·8	1·2	50	0·6	1·6	60	0·6	1·2	50	0·6*
2251 to 3000	1·2	50	0·6	1·6	60	0·6	1·6	60	0·6	2·0	60	0·6	1·2	50	0·6*

Circular Diameter—mm	T	A	S	T	A	S	T	A	S	T	A	S	T†	T‡	A	S
Up to 500	0·6	25	—	0·8	25	—	0·6	25	—	0·8	25	—	0·8	1·0	25	—
501 to 750	0·8	25	—	1·0	25	—	0·8	25	—	1·0	25	—	1·0	1·2	25	—
751 to 1250	1·0	30	—	1·2	30	1·0	1·0	30	—	1·2	30	1·0	1·2	1·2	30	—
1251 to 1750	1·2	35	1·0	1·6	40	1·0	1·2	35	1·0	1·6	40	1·0	1·2	1·2	40	1·0
1751 to 2000	1·2	40	1·0	1·6	50	1·0	1·2	40	1·0	1·6	50	1·0	1·2	1·2	40	1·0
2001 to 2500	1·2	40	1·0	1·6	50	1·0	1·2	40	1·0	1·6	50	1·0				

T = thickness in mm. A = angle size, mm × mm. S = maximum stiffener spacing, m. * With tie-rod bracing.
† = Seamed. ‡ = Welded.

This table summarises the recommendations of the Heating and Ventilating Contractors' Association, UK, for ventilating ducts constructed from sheet metal without stiffening beading, etc. Their publications should be studied for the full specifications.

Table 14.21 Preferred sizes of metric bolts and screws

Size (Max diam of thread mm)	Pitch of thread mm	Precision hexagon screws				Socket head screws max diam mm	Core stress area mm^2	Proof load at strength grade quoted kilonewtons		
		Max width		Preferred lengths within range						
		across flats mm	across corners mm	min mm	max mm			4·6	8·8	12·9
M 1·6	0·35	3·2	3·7	5	12		1·27	0·27	0·72	1·18
M 2	0·40	4·0	4·6	5	16		2·07	0·46	1·18	1·93
M 2·5	0·45	5·0	5·8	5	25		3·39	0·75	1·93	3·16
M 3	0·50	5·5	6·4	5	30	5·5	5·03	1·11	2·87	4·69
M 4	0·70	7·0	8·1	5	70	7·0	8·78	1·95	5·02	8·18
M 5	0·80	8·0	9·2	6	80	8·5	14·20	3·15	8·10	13·20
M 6	1·00	10·0	11·5	6	90	10·0	20·10	4·46	11·50	18·70
M 8	1·25	13·0	15·0	8	110	13·0	36·60	8·13	20·90	34·10
M 10	1·50	17·0	19·6	8	160	16·0	58·00	12·80	33·10	54·00
M 12	1·75	19·0	21·9	10	180	18·0	84·30	18·70	48·20	78·50
M 16	2·00	24·0	27·7	12	220	24·0	157	34·80	89·50	146
M 20	2·50	30·0	34·6	16	220	30·0	245	54·40	140	228

Table 14.21 Preferred sizes of metric bolts and screws—*continued*

Size (Max diam of thread mm)	Pitch of thread mm	Precision hexagon screws: Max width across flats mm	Max width across corners mm	Preferred lengths within range min mm	Preferred lengths within range max mm	Socket head screws max diam mm	Core stress area mm^2	Proof load at strength grade quoted kilonewtons 4·6	8·8	12·9
M 24	3	36·0	41·6	16	220	36·0	353	78·4	201	329
M 30	3·5	46·0	53·1	35	400	45·0	561	124·5	320	523
M 36	4	55·0	63·5	35	400	54·0	817	181	466	761
M 42	4·5	65·0	75·1	40	400	63·0	1120	248	640	1080
M 48	5	75·0	86·6	40	400		1470	326	840	1370
M 56	5·5	85·0	98·1	110	400		2030	451	1160	1890
M 64	6	95·0	109·7	110	400		2680	595	1530	2500

Coarse-pitch series to BS 3692 : 1967 and ISO Recommendations

Strength grade designation		4·6	4·8	5·6	5·8	6·6	6·8	8·8	10·9	12·9	14·9
Tensile strength—min.	N/mm^2	392	392	490	490	588	588	785	981	1177	1373
Yield or permanent set—min.	N/mm^2	235	314	294	392	353	471	628	883	1059	1236
Proof load stress	N/mm^2	222	286	276	357	333	429	571	777	932	1089

At proof load the permanent extension must not exceed 12·5 μm. The working load must be further reduced by an appropriate safety factor.
1 kN (kilonewton) = 102 kgf (kilogrammes force) = 225 lbf (pound force).
1 N/mm = 1 MN/m^2 = 0·102 kgf/mm^2 = 145 lbf/in^2.

Table 14.22

Ball and roller bearings: load capacity

Dimensions d × D × B mm × mm × mm	Base speed rev/min	Ball bearing	40° angular contact	Roller bearing
Series 2 (light)	n_o	P_B	P_A	P_R
10 × 30 × 9	2800	260 N	260 N	600 N
12 × 32 × 10	2800	350	360	600
15 × 35 × 11	2500	400	420	720
17 × 40 × 12	2400	520	540	900
20 × 47 × 14	2100	720	750	1850
25 × 52 × 15	1600	860	910	2150
30 × 62 × 16	1600	1200	1200	2900
35 × 72 × 17	1400	1650	1750	3800
Series 3 (med.)				
10 × 35 × 11	2200	440	490	900
12 × 37 × 12	2200	530	500	950
15 × 42 × 13	2200	640	650	1100
17 × 47 × 14	2000	770	840	1500
20 × 52 × 15	2000	920	1010	2100
25 × 62 × 17	1800	1270	1350	3000
30 × 72 × 19	1600	1720	1900	3800
35 × 80 × 21	1500	2100	2400	5200
40 × 90 × 23	1400	2650	2900	6600
45 × 100 × 25	1300	3500	3900	7600
50 × 110 × 27	1200	4200	4500	9400
55 × 120 × 29	1200	4800	5200	10500
60 × 130 × 31	1100	5700	6300	13000
65 × 140 × 33	1100	6500	7000	15000
70 × 150 × 35	1000	7500	8000	17500
75 × 160 × 37	900	8300	9400	21000
80 × 170 × 39	800	9600	10400	25000
85 × 180 × 41	750	10600	11600	29000
90 × 190 × 43	700	11500	12700	32000
95 × 200 × 45	600	12700	14000	37000
100 × 215 × 47	500	16000	17000	43000
Radial (journal) capacity F_J =		$P_B K_B K_J$	$P_A K_B K_J$	$P_R K_R$
Axial (thrust) capacity F_T =		$P_B K_B K_T$	$P_A K_B K_T$	Nil

P_A, P_B and P_R are approximate journal capacities for 20,000 hours' life at n_o rev/min. Consult makers' catalogues for exact ratings. See Chapter 8.

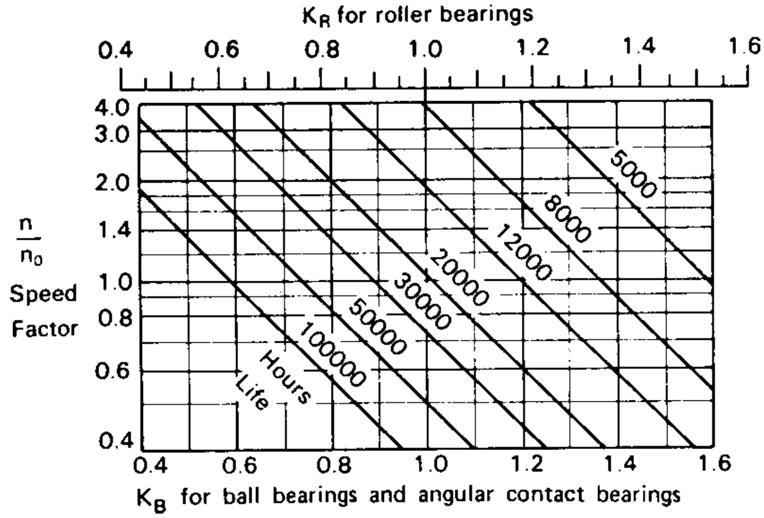


Fig. 14.3 Combined speed and life factors.

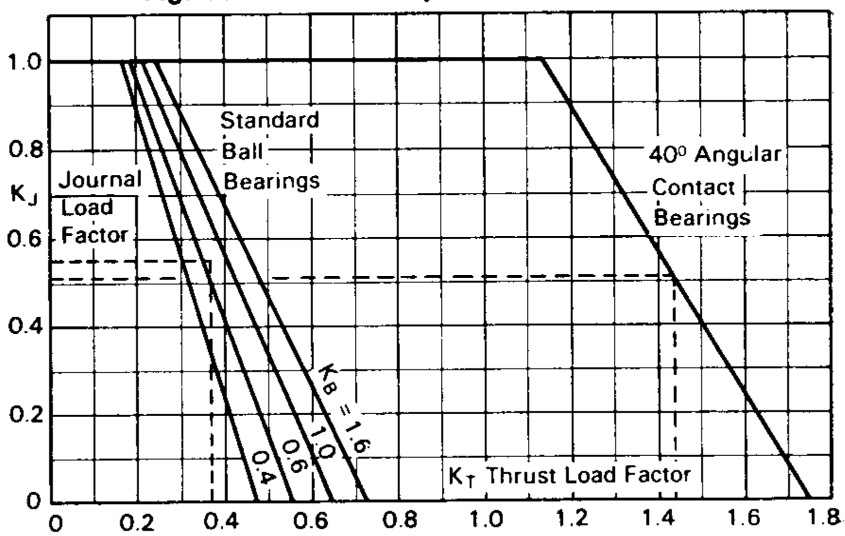


Fig. 14.4 Factors for combined thrust and journal loading.

Example. A 40mm medium ball bearing runs at 1760 rev/min with a weight (journal) load of 100 kg = 980 N. What is the thrust capacity for 50,000 hours life?

n/n_o = 1760/1400 = 1·26
Therefore K_B = 0·67 (from Fig. 14.3).
$K_J = F_J/P_B\ K_B$ = 980/2650 × 0·67 = 0·55 (from Table 14.22).
Therefore K_T = 0·37 (from Fig. 14.4). Therefore thrust capacity
F_T = 2650 (P_B) × 0·67 (K_B) × 0·37 (K_T) = 650 N (Newtons)

Alternatively (if protected against reverse thrust) a 40° angular contact bearing could be used:

$K_J = F_J/P_AK_B$ = 980/2900 × 0·67 = 0·51
Therefore K_T = 1·44 (from Fig. 14.4)
Therefore thrust capacity, F_T = 2900 (P_A) × 0·67 (K_B) × 1·44 (K_T) = 2800 N.

Table 14.23
Approximate full load currents of electric motors

Synchronous Speed		3000 at 50 Hz	1500 at 50 Hz	1000 at 50 Hz	750 at 50 Hz	600 at 50 Hz	500 at 50 Hz
Output rating		3600 at 60 Hz	1800 at 60 Hz	1200 at 60 Hz	900 at 60 Hz	720 at 60 Hz	600 at 60 Hz
kW	*hp*	*Amps at 230 volts, single phase. Capacitor run motors*					
0·06	1/12	0·5	0·6	0·7			
0·09	1/8	0·7	0·8	0·9	1·0		
0·12	1/6	0·9	1·0	1·1	1·2	1·4	
0·18	1/4	1·3	1·4	1·5	1·7	2·0	2·5
0·25	1/3	1·7	1·8	1·9	2·1	2·4	3·0
0·37	1/2	2·5	2·6	2·7	2·9	3·2	3·5
0·55	3/4	3·5	3·6	3·8	4·0	4·3	4·6
0·75	1	4·8	5·0	5·2	5·4	5·7	6·0
1·1	1·5	6·6	6·8	7·0	7·2	7·5	7·8
1·5	2	9·0	9·0	9·2	9·5	10·0	10·5
2·2	3	12·5	12·5	13·0	13·5	14·0	14·5
3·7	5	20·5	20·5	21·0	21·5	22·0	23·0
		Amps at 400 volts, three phase squirrel-cage motors					
0·12	1/6	0·4	0·5	0·5	0·6	0·7	
0·18	1/4	0·5	0·6	0·8	1·0	1·3	1·8
0·25	1/3	0·7	0·9	1·0	1·2	1·4	2·0
0·37	1/2	0·9	1·1	1·3	1·6	1·8	2·3
0·55	3/4	1·3	1·5	1·8	2·1	2·4	2·9
0·75	1	1·8	2·0	2·3	2·7	3·2	3·7
1·1	1·5	2·6	2·8	3·1	3·6	4·2	4·8
1·5	2	3·3	3·6	4·0	4·7	5·4	6·0
2·2	3	4·6	5·0	5·6	6·3	7·0	8·0
3·7	5	7·5	8·0	8·8	9·5	11	12
5·5	7·5	11	11·5	12·5	13·5	15	17
7·5	10	14	15	16·5	18	20	22
11	15	21	22	23	25	27	30
15	20	28	30	31	33	35	38
18·5	25	34	36	38	40	43	47
22	30	40	43	45	47	50	54
30	40	54	57	60	63	67	71
37	50	65	68	71	75	80	86
45	60	80	84	86	90	96	102
55	75	96	100	104	108	115	122
75	100	130	135	140	145	155	165
90	125	157	165	170	175	185	195
110	150	190	200	205	210	220	230
132	175	225	235	240	250	260	275
150	200	260	270	275	280	300	310
160	220	270	280	290	300	320	330
185	250	310	325	330	350	360	380
200	270	340	350	360	370	390	410
220	300	370	375	390	400	420	440
250	350	420	440	450	460	480	500

For other supplies multiply amps by $\frac{230}{\text{1 ph voltage}}$ or $\frac{400}{\text{3 ph line voltage}}$

Based on typical efficiencies and power factors from Figs. 14.8 and 14.9.

Table 14.24

Typical full load speeds of induction motors

kW output	Number of poles—50 Hz						
	2	4	6	8	10	12	16
	rev/min	rev/min	rev/min	rev/min	rev/min	rev/min	rev/min
0·1	2700	1380	900	680	550	460	
0·2	2750	1400	910	680	550	460	
0·5	2800	1410	920	690	560	470	
1·0	2825	1420	930	700	560	470	350
2	2850	1430	940	700	570	475	355
5	2875	1440	950	710	575	480	360
10	2900	1450	960	720	580	485	360
20	2920	1460	970	730	580	485	
50	2940	1470	980	735	585	490	
100	2960	1480	985	740	590	490	
200	2970	1485	990	742	594	494	
500	2980	1490	993	744	595	496	
1000	2985	1492	995	745	596	496	

kW output	Number of poles—60 Hz						
	2	4	6	8	10	12	16
	rev/min	rev/min	rev/min	rev/min	rev/min	rev/min	rev/min
0·1	3300	1660	1080	800	660	550	
0·2	3350	1680	1100	810	670	555	
0·5	3400	1700	1110	825	675	560	
1·0	3420	1710	1120	840	680	565	420
2	3440	1720	1130	850	685	570	425
5	3460	1730	1140	860	690	575	430
10	3480	1740	1150	870	695	580	435
20	3500	1750	1165	875	700	585	
50	3525	1765	1175	880	705	585	
100	3550	1775	1180	886	710	590	
200	3560	1780	1185	890	712	592	
500	3575	1785	1190	892	714	595	
1000	3580	1790	1194	894	715	595	

Table 14.25

Current ratings for electric cables with copper conductors

PVC insulated. Single core. Low and medium voltage. 30°C ambient
Circuits protected against sustained overload exceeding 50%

Derived from the *Regulations for the Electrical Equipment of Buildings* published by the Institution of Electrical Engineers UK.

	Nominal cross-sectional area of conductor mm^2	Number and diam of wires No/mm	Single phase and DC in conduit		Single phase and DC on surface		Three phase in conduit		Three phase on surface	
			Current rating amp	Max* length m	Current rating amp	Max* length m	Current rating amp	Max* length m	Current rating amp	Max* length m
Copper conductors	1	1/1·13	14	10	17	8	11	24	16	17
	1·5	1/1·38	17	12	21	10	15	26	20	20
	2·5	1/1·78	24	14	30	11	21	31	27	24
	4	7/0·85	32	16	40	13	28	36	36	29
	6	7/1·04	42	19	50	16	37	42	46	34
	10	7/1·35	56	24	68	20	52	52	61	44
	16	7/1·70	75	28	90	24	70	59	81	51
	25	7/2·14	97	33	119	27	94	66	106	59
	35	19/1·53	120	36	145	30	115	78	130	70
	50		145	40	175	33	125	95	160	75
	70		185	44	220	37	160	105	200	85
	95		230	48	270	42	195	115	240	95
	120		260	54	310	45	220	125	280	100

Table 14.25—*continued*

Current ratings for electric cables with aluminium conductors

	Nominal cross-sectional area of conductor mm^2	Single phase and DC in conduit		Single phase and DC on surface		Three phase in conduit		Three phase on surface	
		Current rating amp	Max* length m	Current rating amp	Max* length m	Current rating amp	Max* length m	Current rating amp	Max* length m
Aluminium conductors	16	60	21	72	18	52	50	65	40
	25	78	25	94	21	67	60	85	47
	35	96	28	115	24	83	67	105	52
	50	120	30	140	27	100	71	125	61
	70	150	32	175	30	125	80	155	69
	95	175	35	210	35	150	83	185	78
	120	205	35	240	38	175	82	215	83
	150	235	34	275	41	200	78	245	85

*The distance from supply to load for maximum allowed voltage drop (2½%) at rated current on 230V, 1 phase or 400V, 3 phase. For greater lengths the current must be proportionately reduced.

Rating factors for temperature and insulating material

Ambient temperature (°C)	30	40	55	65	75	100	115	130	140
PVC insulation	1·00	0·87	0·61	0·35	—	—	—	—	—
Butyl rubber insulation	1·16	1·03	0·82	0·66	0·45	—	—	—	—
Silicone rubber insulation	1·16	1·16	1·16	1·16	1·16	0·98	0·81	0·59	0·40

Rating factors for groups of single or three phase circuits bunched in conduit or trunking

Number of circuits	1	2	3	4	5	6	8	10	12	14
Rating factor	1·00	0·80	0·69	0·62	0·59	0·55	0·51	0·48	0·43	0·41

Table 14.26

Standard power ratings for V-belt drives

Belt section and width	Smaller pulley pitch diam.	Speed of smaller pulley	Standard kW rating at speed ratio 1 to 1	1·1 to 1	1·3 to 1	2 to 1 and over
	mm	*rev/min*	W_1 *kW per belt*			
Y	20		0·02	0·02	0·02	0·03
	25	1440	0·04	0·04	0·04	0·05
6·5mm	28		0·05	0·05	0·05	0·06
	32		0·06	0·06	0·06	0·07
	36		0·07	0·07	0·07	0·08
	40		0·08	0·08	0·08	0·09
	50		0·11	0·11	0·11	0·12
Z	40		0·09	0·10	0·11	0·12
	50	1440	0·16	0·17	0·18	0·19
10mm	63		0·25	0·26	0·27	0·28
	71		0·30	0·31	0·32	0·33
	80		0·35	0·36	0·37	0·38
	90		0·36	0·37	0·38	0·39
A	75		0·73	0·79	0·85	0·91
	80	1440	0·86	0·92	0·98	1·04
13mm	85		0·99	1·05	1·11	1·17
$\frac{1}{2}$in	90		1·12	1·18	1·24	1·30
	100		1·38	1·44	1·50	1·56
	112		1·63	1·69	1·75	1·81
	125		2·00	2·06	2·12	2·18
B	125		2·25	2·40	2·55	2·70
	140	1440	2·80	2·95	3·10	3·25
17mm	160		3·60	3·75	3·90	4·05
$\frac{21}{32}$in	180		4·20	4·35	4·50	4·65
	200		5·25	5·40	5·55	5·70
C	200		4·9	5·2	5·5	5·8
	224	960	6·0	6·3	6·6	6·9
22mm	250		7·4	7·7	8·0	8·3
$\frac{7}{8}$in	280		8·8	9·1	9·4	9·7
	315		10·2	10·5	10·8	11·1

Table 14.26

Standard power ratings for V-belt drives—*continued*

Belt section and width	Smaller pulley pitch diam.	Speed of smaller pulley	Standard kW rating at speed ratio			
			1 to 1	1·1 to 1	1·3 to 1	2 to 1 and over
	mm	*rev/min*	*W_1 kW per belt*			
D	355		13·4	14·1	15·0	15·7
	400	720	17·1	17·8	18·7	19·4
32mm	450		20·8	21·5	22·4	23·1
$1\frac{1}{4}$in	500		24·1	24·8	25·7	26·4
	560		27·9	28·6	29·5	30·2
	630		31·2	31·9	32·8	33·5
E	500		24·2	25·4	26·6	27·8
	560	580	29·1	30·3	31·5	32·7
38mm	630		34·3	35·5	36·7	37·9
$1\frac{1}{2}$in	710		39·7	40·9	42·1	43·3
	800		46·2	47·4	48·6	49·8
	900		51·0	52·2	53·4	54·6

Abridged and derived from BS 1440 and BS 3548. These standards and makers' catalogues should be consulted for comprehensive data.

The ratings in Table 14.26 give the power supplied by the driving motor. The power transmitted to the load is commonly taken as 95% of the power supplied. The ratings are for Standard Belts to BS 1440 and BS 3548 in V-V speed reduction drives. For speed-up drives take the factor K_1 for reversing motors. Premium grade belts are available to the same dimensions with 30% to 40% higher rating. Special proprietary belts to different dimensions can transmit two or more times the power in the same space.

The kW quoted in Table 14.26 must be multiplied by each of the factors below to obtain the kW rating, W, of the complete drive with N belts in parallel. Factor K_1 is given below and K_2, K_3 and K_4 on the next page.

$$W = N W_1 K_1 K_2 K_3 K_4$$

Service Factor **K_1**

Driven machine	Driving motor	Operational hours per day: 0–10	10–16	16–24
Fans and blowers to 7·5 kW Centrifugal pumps (A to E belts) Centrifugal compressors	Normal torque	1·00	0·91	0·83
	High torque	0·91	0·83	0·77
	Reversing	0·80	0·72	0·66
Fans over 7·5 kW Centrifugal pumps (Y and Z belts) Positive displacement and rotary pumps	Normal torque	0·91	0·83	0·77
	High torque	0·83	0·77	0·72
	Reversing	0·72	0·66	0·62
Reciprocating pumps and compressors Positive displacement blowers	Normal torque	0·83	0·77	0·72
	High torque	0·72	0·67	0·60
	Reversing	0·66	0·62	0·58

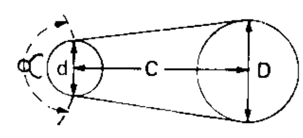
Factors K_2, K_3 and K_4 for use with Table 14.26

Motor speed		580	700	720	870	960	1160	1440	1740	2880	3480
K_2	Y	K_2 =				0·73	0·84	1·00	1·15	1·7	1·9
Speed factor	Z					0·73	0·84	1·00	1·15	1·6	1·8
for belt	A			0·58	0·67	0·73	0·84	1·00	1·14	1·6	1·8
section:	B			0·60	0·69	0·75	0·85	1·00	1·12	1·4	
	C	0·69	0·78	0·81	0·92	1·00	1·12	1·25			
	D	0·87	0·96	1·00	1·10	1·15					
	E	1·00	1·08	1·10							

Use the same ratings and factors for motor speeds nearer synchronism.

K_3		0·80	0·84	0·88	0·92	0·96	1·00	1·04	1·08	1·12	1·16	1·20
Length factor		*Inside length in metres. These are not standard lengths, but guides for estimating* **K_3**										
Belt lengths	Y	0·21	0·26	0·31	0·36	0·41	0·46	0·53				
for belt	Z	0·40	0·44	0·49	0·53	0·58	0·64	0·73	0·84	0·94	1·04	1·14
sections:	A	0·61	0·77	0·97	1·18	1·42	1·70	2·08	2·50	2·95	3·70	4·60
	B	0·81	1·02	1·27	1·55	1·88	2·28	2·72	3·22	3·80	4·50	5·50
	C	1·30	1·65	2·03	2·44	3·00	3·65	4·40	5·30	6·25	7·40	8·70
	D	2·25	2·75	3·40	4·05	5·00	6·00	7·25	8·80	10·50	12·40	14·80
	E				5·30	6·10	7·20	8·70	10·30	12·20	14·60	17·60

K_4 Arc of contact factor



K_4	1·00	0·98	0·96	0·94	0·92	0·90	0·88	0·86	0·84	0·82
α	180°	171°	163°	155°	148°	142°	136°	130°	125°	120°
(D–d)/C	0·00	0·15	0·29	0·43	0·55	0·65	0·75	0·85	0·92	1·00

Table 14.27

Circular equivalents of rectangular ducts for equal pressure drop at same volume flow

LONG SIDE	SHORT SIDE													LONG SIDE
	325	**350**	**400**	**450**	**500**	**550**	**600**	**650**	**700**	**750**	**800**	**900**	**1000**	
1000	609	635	682	727	770	810	848	884	918	952	984	1044	1101	**1000**
900	581	605	650	693	732	770	806	840	873	904	934	991	556	**900**
800	551	573	616	655	693	728	761	793	824	853	881	503	527	**800**
750	535	556	597	636	672	706	738	769	798	826	463	488	512	**750**
700	518	539	578	615	650	682	713	743	771	424	449	473	496	**700**
650	500	520	558	594	627	658	687	716	384	410	434	457	479	**650**
600	482	501	537	571	603	632	661	350	371	395	419	441	462	**600**
550	462	481	515	547	577	606	315	336	356	380	402	423	443	**550**
500	442	459	492	522	551	291	302	322	341	364	385	405	424	**500**
450	420	436	467	496	268	278	288	307	325	346	366	385	403	**450**
400	397	412	440	244	254	264	273	291	308	328	346	367	381	**400**
350	371	385	220	229	239	248	257	273	289	308	325	341	357	**350**
325	358	202	212	222	231	240	248	264	279	297	313	329	344	**325**
	100	**110**	**120**	**130**	**140**	**150**	**160**	**180**	**200**	**225**	**250**	**275**	**300**	

LONG SIDE	SHORT SIDE													LONG SIDE
	100	**110**	**120**	**130**	**140**	**150**	**160**	**180**	**200**	**225**	**250**	**275**	**300**	
300	185	195	205	214	223	231	239	254	269	286	301	316	330	**300**
275	178	188	197	205	214	222	229	244	258	274	289	303	168	**275**
250	171	180	188	196	204	212	219	233	246	261	275	151	161	**250**
225	163	171	179	187	194	201	208	221	234	248	139	144	153	**225**
200	154	162	170	177	184	190	197	209	220	127	132	136	145	**200**
180	146	154	161	168	175	181	187	198	116	121	126	130	139	**180**
160	139	146	152	159	165	171	176	105	110	115	119	123	131	**160**
150	134	141	148	154	160	165	97·6	102	107	111	115	119	127	**150**
140	130	136	143	149	154	89·8	94·6	99·2	104	108	112	116	123	**140**
130	125	132	138	143	82·0	86·9	91·5	95·9	100	104	108	112	119	**130**
120	121	126	132	74·1	79·1	83·7	88·1	93·2	96·4	100	104	107	114	**120**
110	115	121	66·2	71·3	76·0	80·4	84·7	88·7	92·5	96·1	99·7	103	109	**110**
100	110	58·3	63·5	68·2	72·7	77·0	81·0	84·8	88·4	91·8	95·2	98·4	104	**100**
		30	**35**	**40**	**45**	**50**	**55**	**60**	**65**	**70**	**75**	**80**	**90**	
	SHORT SIDE													

References in the vertical columns must not cross the zigzag line. Examples: 700 × 600 = 713mm diameter; 700 × 250 = 449mm diameter. To extend the table all the entries may be multiplied or divided by 10, example: 70mm × 60mm = 71·3mm diameter

Based on the formula: $D_q = 1{\cdot}265\, a^{0{\cdot}6}\, b^{0{\cdot}6}/(a+b)^{0{\cdot}2}$

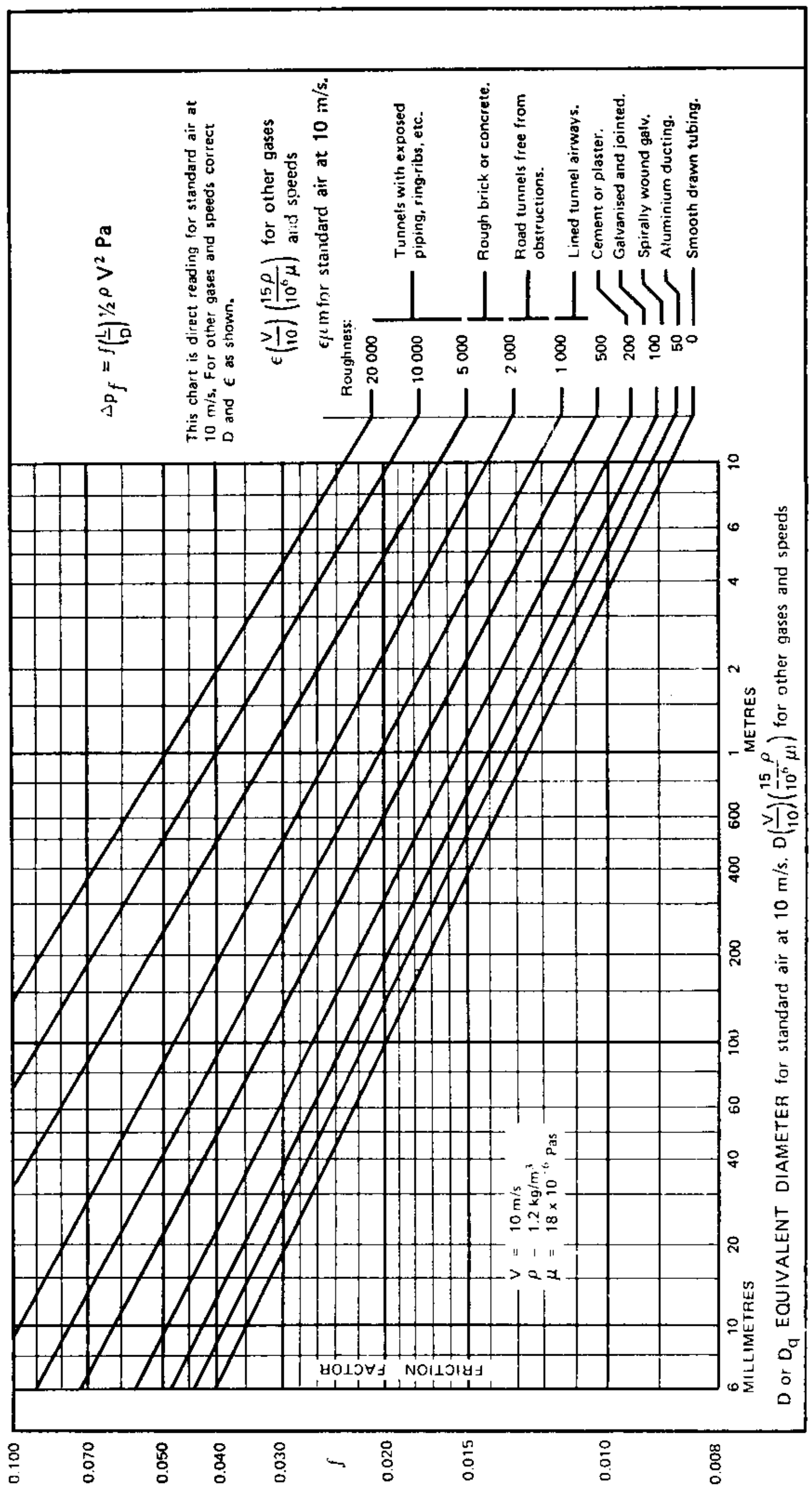


Fig. 14.5 Friction factors for rough-walled ducts.

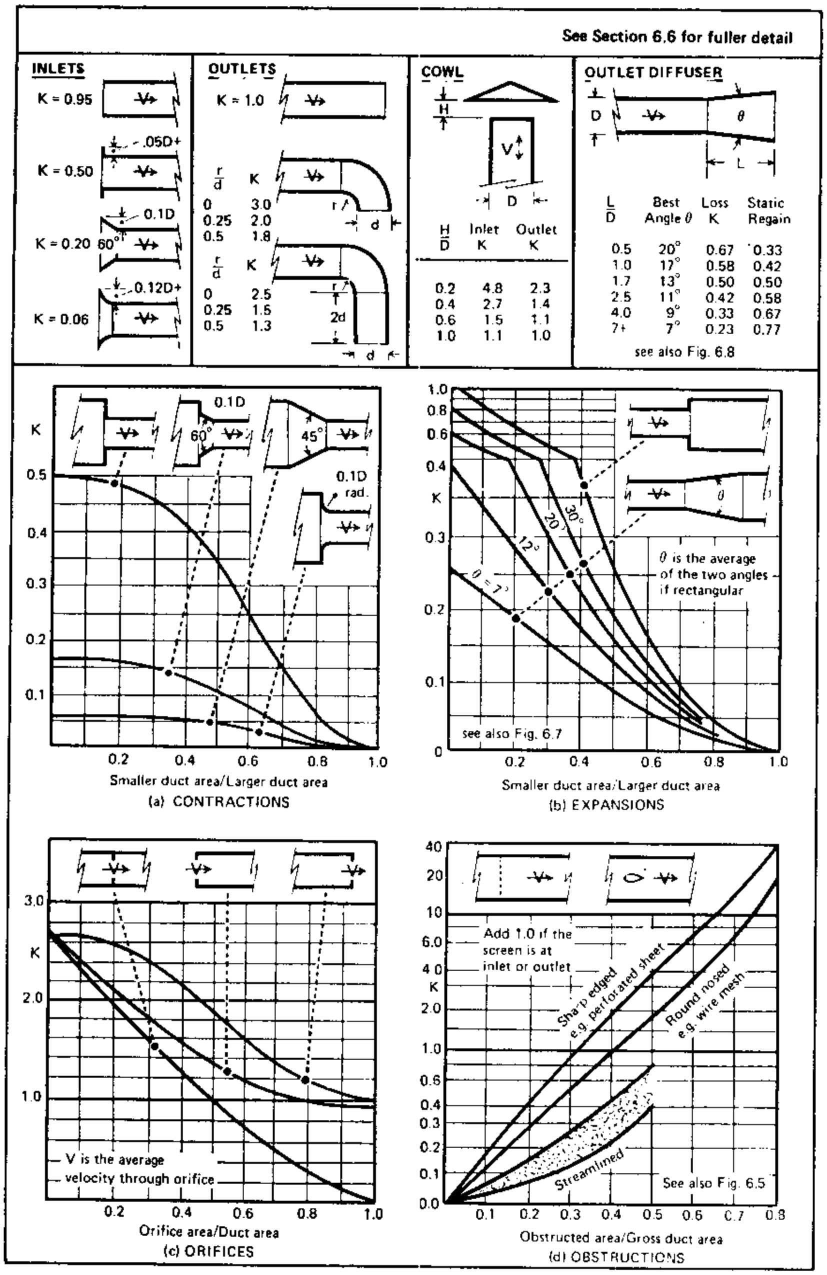
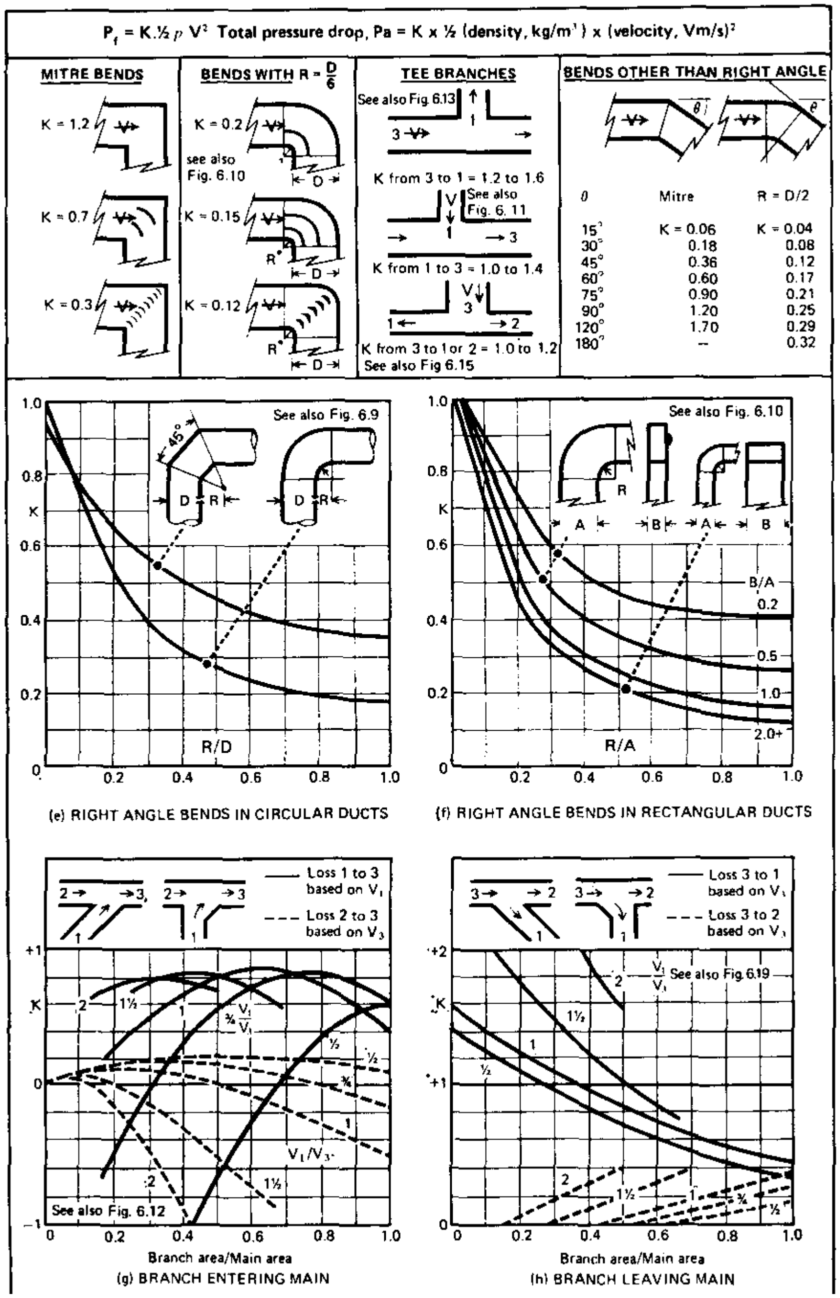
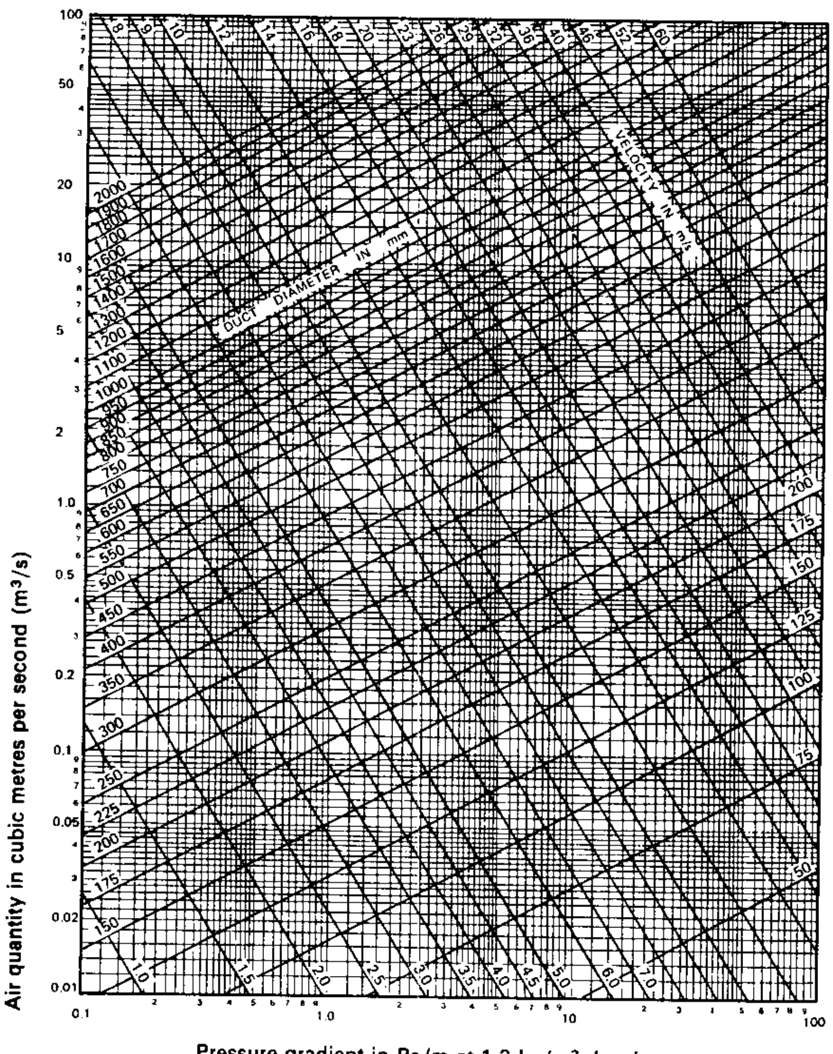


Fig. 14.6 Summary of pressure drop data.

Fig. 14.6—*continued*





For normally jointed galvanised steel ducts, with acknowledgements to the I.H.V.E. Guide to Current Practice.

Fig. 14.7 Resistance to air flow in straight ducts for ventilating systems.

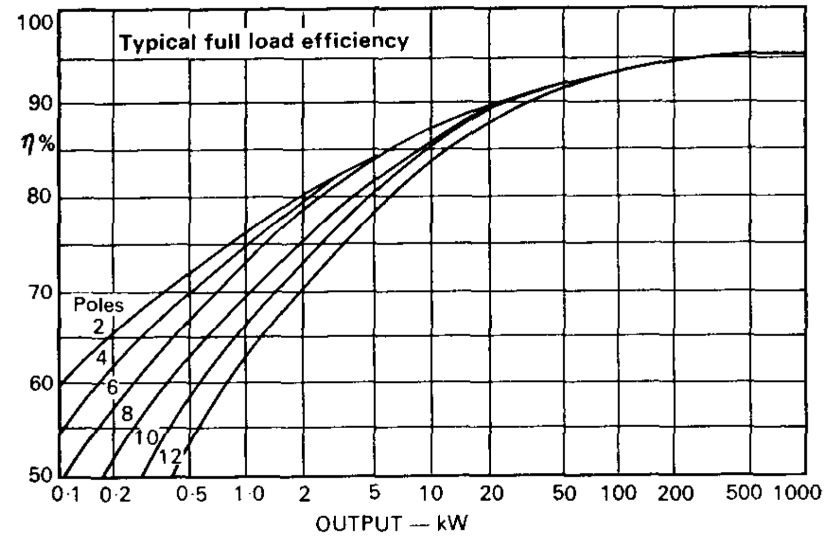


Fig. 14.8

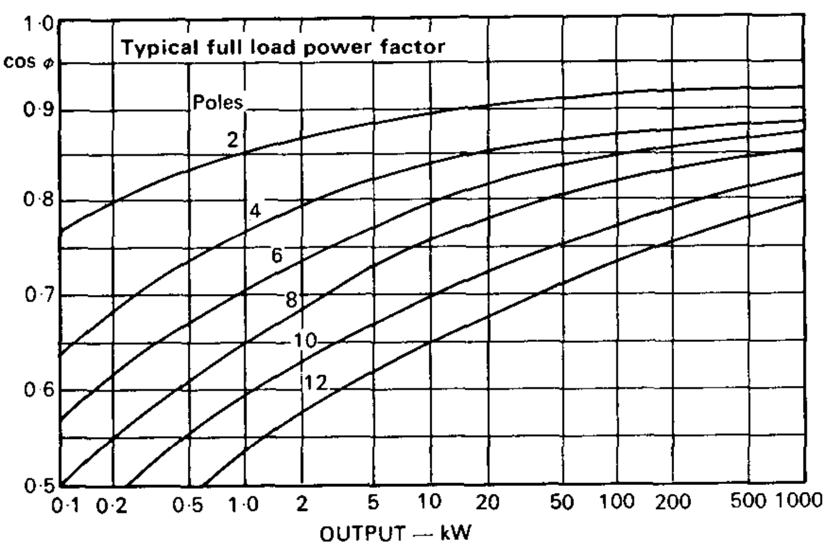


Fig. 14.9

Note that individual motor designs may vary by from ± 3% at 90% or 0·9 to ± 10% at 60% or 0·6.

Typical performance of three-phase induction motors.

Bibliography

There are many books on the topics covered by this Practical Guide, most of them written to meet the needs of the examination student. A few works of general reference are listed here which themselves contain extensive bibliographies of books and papers for more detailed study.

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Figures in italics denote illustrations.