SHORT - CUT METHOD OF DETERMINING INDUCTION - MOTOR HORSEPOWER from AMMETER READINGS

The following is a simple method whereby the horse-power loadings on 3 Phase induction motors can be determined with practical accuracy by means of a hook-on ammeter.
When an induction motor is operated at other than full load, the percentage efficiency and the power factor will vary from the figures applicable to the full-load condition. Hence the current taken by the motor does not vary in direct proportion to the load, and the horsepower output cannot be determined accurately from ammeter readings alone.
The curves given, which are based on average motor performance, enable ammeter readings to be translated into the corresponding horsepower outputs.
For the most accurate result, disconnect the motor from the load and measure the no-load current, and this, when expressed as a percentage of the full-load current (which is stamped on the motor name plate), determines the curve which should be used. Greater accuracy may be obtained by interpolation between the curves shown on Page 23.
If it is not convenient to disconnect the motor from the load in order to measure the no-load current, use the curve appropriate to the number of poles in the motor, as given in the table, which applies to motors of average characteristics. However, the accuracy of results so obtained may be much lower than if the no-load current of the particular motor is measured.
The accuracy of this method is subject, of course, to both the ammeter accuracy and the accuracy of maintained voltage at the motor terminals.

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FAN SELECTION

FOREWORD

The following notes on "Fan Selection" are presented to you only as a very brief guide to assist in the problems associated with selection of a Fan for industrial purposes. No attempt has been, or indeed could be, made in this catalogue to cover the vast field of knowledge required by the completely qualified Fan Engineer, if such a person exists. There are a great many very informative text and reference books available on various aspects of Fan Engineering some of which are referred to in the following articles, we have found them very helpful and we accordingly recommend them to you for your reference.
We have included tables, charts and formulae which are in every day use in this field of engineering and should be useful in calculating solutions to your problems or those of your customer.
We must once again emphasize that this information is only of the most elementary nature and where there is any doubt in your mind regarding a Fan problem, that you refer back to our Technical Representatives or to the factory where every assistance possible will be given enthusiastically.

Recommended Reference Books:

Axial-Flow Fans by R. A. Wallis.

Fans by T. Baumeister, Jnr.

Design of Industrial Exhaust Systems by J. L. Alden.

Fan Engineering by R. D. Madison.

Industrial Ventilation by the American Conference of Governmental Industrial Hygienists.

Woods Practical Guide to Fan Engineering by W. C. Osborne and C. G. Turner.

Axial-Flow Fans by Curt Keller and Lionel S. Marks.

Air Conditioning and Engineering by American Blower Corporation.

The various publications of British Standards Institution. (See Table No. 5).

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INDEX (See Page 24)

PRESSURE – ENERGY

A Fan is a machine for imparting to a volume of air, an increase in Velocity Pressure and also an increase in Static Pressure. The Static Pressure increase is the increase in the Potential energy of the air in contrast to the Velocity Pressure increase which is the increase in kinetic energy of the air

British Standard 848 : Part 1 : 1963 defines a Fan as "A Rotary machine which propels air continuously."

POTENTIAL ENERGY can be defined as "Energy possessed by a body because of its position in a field of force."

KINETIC ENERGY can be defined as "Energy possessed by a body because of its motion."

It follows then that we can think of STATIC PRESSURE as POTENTIAL ENERGY and VELOCITY PRESSURE as KINETIC ENERGY.

STATIC PRESSURE may be more simply defined as "A Pressure measurement of the ability of air to overcome resistance to flow."

VELOCITY PRESSURE may also be simply defined as "A Pressure measurement of the linear velocity of Air Flow."

TOTAL PRESSURE is the sum of both STATIC PRESSURE and VELOCITY PRESSURE or as an equation $P_t = P_s + P_v$
Note: All symbols, Units and Parameters used in this data are identical with those used in BS-848 Part 1:1963.

EXTRACT OF BS•848 PART I:1963

SPECIFICATION

SECTION ONE. GENERAL

SCOPE

1. This British Standard deals with the determination of the performance and efficency of Fans, and sets out methods of tests to provide a basis of comparison of the performance and efficiency of particular types. It also covers the terms and definitions, symbols and formulae recommended for use in fan testing. The standard also specifies the instruments to be used in carrying out the standard tests.

The standard does not cover ceiling type and desk type fans.

TERMS AND DEFINITIONS

2. For the purpose of this British Standard the following definitions apply:

a. AIR. The term "air" is used as an abbreviation for the expression "air or other gas," except where referred to as "atmospheric air."

b. FAN. A rotary machine which propels air continuously.

In this standard the term "fan" shall be taken to mean the fan as supplied, without any addition to the inlet or outlet, except where such addition is specified.

NOTE: The function of a fan as envisaged by this standard is that of moving air rather than compressing it although corrections for compressibility are given for pressure ratios up to 1.5.

c. IMPELLER. That part of a fan which, by its rotation, imparts movement to the air.

d. AXIAL-FLOW FAN. A fan having a cylindrical casing in which the air enters and leaves the impeller in a direction substantially parallel to its axis.

e. CENTRIFUGAL FAN or RADIAL-FLOW FAN. A fan in which the air leaves the impeller in a direction substantially at right angles to its axis.

f. PROPELLER FAN. A fan having an impeller, other than of the centrifugal type, rotating in an orifice, the air flow into and out of the impeller not being confined by any casing.

g. CASING. Those stationary parts of the fan which guide air to and from the impeller.

h. MULTI-STAGE FAN. A fan having two or more impellers working in series.

j. EXPANDER. That part of an airway which enlarges gradually in the direction of the air stream.

k. REDUCER. That part of an airway which reduces gradually in the direction of the air stream.

TERMS RELATING TO FAN PERFORMANCE

3. a. STANDARD AIR. Atmospheric air having a weight per unit volume of 0.075 lb/ft³.
NOTE: The value of 0.075 lb/ft³ corresponds to atmospheric air at a temperature of 68°F (20°C), a barometric pressure of 30 inches (762 mm) of mercury and a relative humidity of 62 per cent.

b. ABSOLUTE PRESSURE. That pressure which is exerted equally in all directions at a point.
NOTE: The absolute pressure of the ambient atmosphere is the barometric pressure.

c. STATIC PRESSURE The difference in consistent units between the absolute pressure at a point, and the absolute pressure of the ambient atmosphere, being positive when the pressure at the point is above the ambient pressure and negative when below.

d. VELOCITY PRESSURE. The pressure-equivalent of the air velocity at any particular point. This is always positive.

e. TOTAL PRESSURE. The algebraic sum of the static pressure and velocity pressure at any particular point.

f. FAN TOTAL PRESSURE. The algebraic difference between the mean total pressure at the fan outlet and the mean total pressure at the fan inlet.

g. FAN VELOCITY PRESSURE. The velocity pressure corresponding to the average velocity at the fan outlet based on the total outlet area without any deductions for motors, fairings, or other bodies.

h. FAN STATIC PRESSURE. The difference between the fan total pressure and the fan velocity pressure.

j. INLET VOLUME. The volume per unit time entering the fan.

k. FAN DUTY (Total). The inlet volume dealt with by a fan at a stated fan total pressure.

l. FAN DUTY (Static). The inlet volume dealt with by a fan at a stated fan static pressure.

m. AIR POWER (Total). That part of the energy per unit time, imparted by the fan to the air in increasing its total pressure from that at the inlet to that at the outlet, as obtained from the formula in Clause 6(vii).

n. AIR POWER (Static). The air power (total) less the nominal kinetic energy* per unit time at the outlet, as obtained from the formula in Clause 6(viii).

o. IMPELLER POWER. The energy input, per unit time, to the fan impeller(s).

p. SHAFT POWER. The energy input, per unit time, to the fan shaft(s) including the power absorbed by such parts of the transmission system as constitute an integral part of the fan, e.g. fan shaft bearings.

q. NET FAN TOTAL EFFICIENCY. The ratio of the air power (total) to the impeller power.

r. NET FAN STATIC EFFICIENCY. The ratio of the air power (static) to the impeller power.

s. FAN TOTAL EFFICIENCY. The ratio of the air power (total) to the shaft power.

t. FAN STATIC EFFICIENCY. The ratio of the air power (static) to the shaft power.

TERMS RELATING TO PRESSURE MEASUREMENTS

4. a. SIDE TUBE or STATIC PRESSURE TUBE. A tube which allows the air to flow without disturbance past one or more small orifices having their axes at right angles to the direction of the air stream in which it is placed.

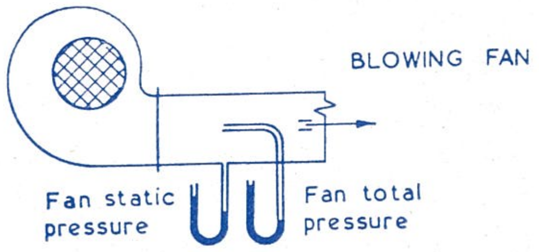
b. SIDE TAPPING or STATIC PRESSURE TAPPING. A small opening in the wall of an airway having its axis at right angles to the wall and so constructed as to allow the air to flow past it without disturbance.

c. FACING TUBE or TOTAL PRESSURE TUBE. An open-ended tube the axis of which is coincident with the direction of the air stream in which it is placed, the open end facing up stream, i.e. against the direction of flow.

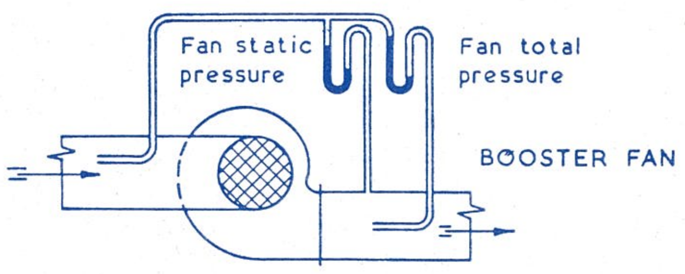
d. PITOT TUBE†. A combination of side tube and facing tube as one unit.

*The nominal kinetic energy is that calculated from the mean velocity of discharge based on the total area of the fan outlet.
†The term "pitot tube" orinally referred to a facing tube only, but for the purpose of this British Standard the above definition has been adopted.

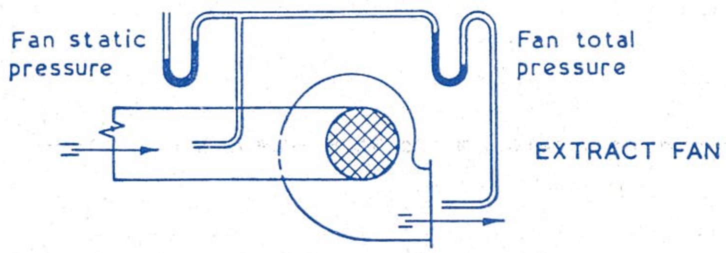
DIAGRAMMATIC ILLUSTRATIONS of FAN PRESSURE DEFINITIONS



Fan ducted at outlet only



Fan ducted at inlet and outlet



Fan ducted at inlet only

SYMBOLS AND UNITS

5. For the purpose of this British Standard, the following symbols and units apply.

Symbol	Term	Unit
p_s	Static pressure at a point	inches* of water
$\bar{p}_s$	Average static pressure at test section	
p_v	Velocity pressure at a point	
$\bar{p}_v$	Velocity pressure corresponding to the average velocity at test section	
p_t	Total pressure at a point	
Δp	Differential pressure across an orifice or conical inlet	
p_f	Loss of pressure due to friction	
P_s	Fan static pressure	
P_v	Fan velocity pressure	
P_t	Fan total pressure	
HP_{at}	Air power (total)	horsepower
HP_{as}	Air power (static)	horsepower
HP_{im}	Impeller power	horsepower
HP_{sh}	Shaft power	horsepower
η_t	Fan total efficiency	per cent
η_s	Fan static efficiency	per cent
N	Rotation speed	rev/min
D	Diameter of airway at test section	ft.
d	Diameter of orifice	ft.
D_i	Diameter of inlet	ft.
D_{im}	Diameter of impeller	ft.
A	Cross-sectional area of airway at test section	sq. ft.

Symbol	Term	Unit
a and b	Dimensions of rectangular airway at test section	ft.
C_D	Coefficient of discharge	–
C_S	Corrected orifice coefficient	–
v	Velocity of air at a point	ft/min
$\bar{v}$	Average velocity of air at test section	ft/min
Q	Volume of air per unit time (air flow) at test section	ft³/min
Q_i	Inlet volume	ft³/min
L	Length of airway from fan to test section	ft.
t	Average temp. of air at test section	°F
B	Barometric pressure	inches of mercury
w	Weight of air or other gas per unit volume	lb/ft³
w_a	Weight of atmospheric air per unit volume	lb/ft³
w_i	Weight of air or other gas per unit volume at fan inlet	lb/ft³
γ	Ratio of specific heat of air at constant pressure to specific heat at constant volume	–
r	Ratio of absolute static pressure at fan outlet to absolute static pressure at fan inlet	–
K_p	Compressibility coefficient	–

*Based on water having a weight of 62.3lb/ft³ as measured in standard air defined in Clause 3.

FORMULAE

6. (i) Total pressure at a point, in inches of water.

$$p_t = p_s + p_v$$

(ii) Weight per unit volume of atmospheric air at test section in pounds per cubic foot.

$$w_a = 0{\cdot}0750 \times \frac{B + 0{\cdot}0737\,\bar{p}_s}{30} \times \frac{528}{460+t}$$

where 0.0750=weight of standard air in pounds per cubic foot.

NOTE 1 For the purpose of this British Standard, the formula in Clause 6(ii) may be used for all works testing without correction for humidity. Under site conditions, if the moisture content exceeds 2 per cent by weight, the weight per unit volume of the moist air should be determined by reference to psychrometric data. The value of 0.0750 for standard air corresponds to atmospheric air at 68°F (20°C) and 30 inches of mercury (762mm), having a moisture content of 0.09 per cent by weight (62 per cent relative humidity at 68°F (20°C)). The value of the "weight per unit volume" to be used for other gases or mixtures of gases and vapours shall be the subject of agreement between the manufacturer and the purchaser

(iii) Velocity of air at a point, in feet per minute.

$$v = 1097\sqrt{\frac{p_v}{w}}$$

(iv) Velocity pressure corresponding to the average velocity in inches of water.

$$\bar{p}_v = w \times \left(\frac{\bar{v}}{1097}\right)^2$$

(v) Air flow at test section in cubic feet per minute.

$$Q = \bar{v} \times A$$

(vi) Inlet volume in cubic feet per minute.

$$Q_1 = Q \times \frac{w}{w_1}$$

(vii) Air power (total), in horse-power.

$$HP_{at} = \frac{Q_1 \times P_t}{6360} \times K_p$$

(viii) Air power (static), in horse-power.

$$HP_{as} = \frac{Q_1 \times P_s}{6360} \times K_p$$

NOTE 2 For the purpose of this British Standard the compressibility coefficient K_p in the formulae given in Clause 6(vii) and (viii) may be taken as unity, provided that the fan static pressure does not exceed 10 inches of water under normal atmospheric conditions.

At higher pressures K_p should be determined by reference to Fig. 2 calculating the pressure ratio r by means of the formula in Clause 6(x) below. The value of γ may be taken as 1.4 for atmospheric air; for other gases refer to B.S. 2009, "Code for acceptance tests for turbo-type compressors and exhausters."

(ix) Allowance for friction loss in airway, in inches of water.

$$\text{For circular airway} \quad p_f = 0{\cdot}02\frac{L}{D} \times \bar{p}_v$$

$$\text{For rectangular airway:} \quad p_f = 0{\cdot}01 \times L\left(\frac{a+b}{a \times b}\right) \times \bar{p}_v$$

(x) Pressure ratio for use in Fig. 2

In general the pressure ratio for use in Fig. 2 equals the absolute static pressure at the fan outlet divided by the absolute static pressure at the fan inlet.

In particular cases and for the purposes of this standard the following formulae may be used.

A. Blowing conditions

$$r = \frac{B + 0{\cdot}0737P_s}{B}$$

B. Exhausting conditions

$$r = \frac{B}{B - 0{\cdot}0737P_s}$$

THE WORK DONE BY A FAN

A Fan, either Centrifugal or Axial-Flow, moves a quantity of air (Volume) in unit time (CFM). As the speed of the fan is increased the quantity of air is also increased (In direct ratio to the speed $\frac{(RPM_2)}{(RPM_1)}$)

At any constant speed a certain Total increase of energy (Total Pressure) will be added to the air stream and this will be distributed between Static Pressure (Potential Energy), and Velocity Pressure (Kinetic Energy). The Velocity Pressure being the pressure corresponding to the Linear Velocity (FPM) of the flow of the air leaving the Fan.

For any given Fan operating, (a) against a given resistance, (b) on the same unchanged duct system, (c) at the same point on the static efficiency curve, or (d) on the same point on any of the other efficiency curves; always with air or other gas at the same density, then: As the speed, (depending upon design) of a Centrifugal Fan is increased the maximum possible Static Pressure of the Fan is also increased (By the square of the speed ratio $\frac{(RPM_2)^2}{(RPM_1)}$)

This Static Pressure imparted to the air is available to overcome resistance to flow in ducts either on the inlet, if the Fan is used as an Exhauster, or on the outlet if the Fan is used as a Blower.

Similarly as the speed is increased the Velocity Pressure will also be increased (In direct ratio to the speed $\frac{(RPM_2)}{(RPM_1)}$)

At constant speed any Fan requires a certain Horsepower (Input) to drive it, the Input power (Horsepower) being equal to the Output power plus mechanical and aerodynamic efficiency losses

As Fan speed is increased the Horsepower input required to drive it must also increase, because more air is moved per unit time (CFM) and more energy (Total Pressure) is added to the air. (Horsepower increases as the cube of the speed ratio $\frac{(RPM_2)^3}{(RPM_1)}$)

STATIC PRESSURE

Before explaining the methods of Fan Selection, it will be helpful to describe in a little more detail Static Pressure, a term that is in constant use in Fan engineering and applications.

Static Pressures are generally measured in the form of Head. With water for example, 1 foot Head is equivalent to 62.35 lb. per square foot pressure. The Head theoretically is measured in feet of "fluid under test," but with Air the density is low and the Head would be so large that there would be physical difficulty in measuring it with a manometer, so water is substituted as the measuring fluid. Therefore the Head is measured in inches of water ("wg). A Fan is accordingly rated as producing a Static Pressure of "So many" inches static water gauge ("swg). This means that the Air delivered from the Fan, in addition to velocity, exerts an equal static pressure in all directions and this pressure is sufficient to support a column of water "So many" inches high.

1" of water pressure is equivalent to 69.4 ft. of standard Air, ie; Air at 68°F, Barometric Pressure of 30"Hg, 62% Relative Humidity, and having a Density of .075 Lb/Cu Ft.

On the basis of Feet of Fluid, a small Forge Blower developing a pressure of 7"swg is producing a Head of Air 485.8' which becomes comparable with water pump performance when related to Head in Feet of Fluid. As a comparison of pressure it should be noted that 27.74"swg is equal to 1 Lb/Sq Inch, (1psi). (See "Pressure Conversion Table," Table No 4).

THE BASIC PROBLEMS IN FAN SELECTION

(a) What TYPE of Fan is required to do the job?
(b) What VOLUME of Air is required?
(c) What STATIC PRESSURE is required?
(d) What SIZE of Fan?
(e) At what SPEED must the Fan be run to obtain (b) and (c)?
(f) What HORSEPOWER will be required to economically maintain (e)?
(g) Is Air LINEAR VELOCITY an important consideration? If so what is minimum requirement?
(Linear Velocity is a function of (b) Volume, and is usually taken into consideration at the same time)
(h) Is NOISE a factor to be considered?

To briefly explain the previous questions:

(a) First it must be decided which one of the three main Industrial Fan applications is involved; CONVEYING, VENTILATING or COMBUSTION. So that when other factors have been calculated, it will only be necessary to turn to the relevent Performance Tables in that section of the Catalogue to select the Fan required for the job.
The type of Fan will generally be determined by the type of job to be done, e.g. If the job requires the transporting of Sawdust from a Saw-Bench to a Cyclone, the Fan required is of a type which is capable of CONVEYING solid material This would then be the "DAWN" Cast Iron Exhauster or Mill Exhaust type. If the job requires VENTILATING of an enclosed area, then the Fan required will be the "DAWN" Axial-Flow, Propeller or Multi-Vane type.
If the job requires a Fan for the supplying of Air for fuel COMBUSTION, then the Fan will be found in the "DAWN" Pressure Blower range.

(b) For the CONVEYING of solid materials the Linear Velocity multiplied by the duct cross-sectional area, in square feet, will give the Volume required (CFM), or $Q = \bar{v} \times A$

For VENTILATING, the Volume of air required is equal to the cubic capacity of the enclosed area, in cubic feet, multiplied by the number of air changes per hour, divided by 60.

For COMBUSTION, the Volume of Air required is generally available from the Burner manufacturers or from the Fuel suppliers. There are a great many variable factors in this application and some are detailed on page 14 of this Data.

(c) Static Pressure required is determined by a calculation of the "System Resistance" and is covered in more detail on page 11.

(d) When the Volume and Pressure required are known it will only be necessary to check Rating Tables of relevant type to select size of the most suitable Fan.

(e) (f) The Rating Table which correlates the Volume and Pressure required, will also show the necessary Speed and Horsepower for the particular Fan.

(g) Linear Velocity is particulary important in applications requiring the CONVEYING of Solid Materials where certain Minimum Velocities are necessary to keep the Material "Airborne." (A selection of these Velocities are shown in Table No. 6).

(h) Noise is an important factor in VENTILATING applications and is mostly the result of Air Velocity being too high. It will therefore be necessary to use bigger Ducting to carry the required Volume of Air, and this will generally demand a larger and slower Fan. (It is worth remembering that High Air Velocity will produce High Noise Level).

OTHER PROBLEMS OF FAN SELECTION

There are a number of other factors which should be considered in selecting a Fan, and these include such things as:

Air Temperatures (above or below Standard Air), Corrosive or Abrasive Air or Material, Potentially Explosive or Inflammable Air or Material, Space Limitations where Fan is to be installed, Direct Coupled Motorised or Belt Driven Fans, etc.

We consider that these are not really BASIC Problems and for that reason they will not be expanded upon in this section.

More information is available in other pages of this data or from our Technical Representatives.

ECONOMICAL SELECTION

It is important to note that no Fan should be selected from arbitrary choice. A Fan must be selected because it will MOST ECONOMICALLY PRODUCE THE VOLUME OF AIR AND STATIC PRESSURE REQUIRED TO DO A PARTICULAR JOB, and of course it must also be constructed so that it has long life under the conditions of service.

Therefore it will be necessary to consider at least five (5) other factors:

(1) Primary Costs. (Initial Expenditure — Fan & Installation).

(2) Running Costs. (Horsepower required — Fan Efficiency — Maintenance).

(3) Replacement Cost. (As related to length of economical life).

(4) Plant Extensions. (Possibility of more Air being required later).

(5) Noise Level. (Remembering that this increases with increase in speed).

For small Fans the purchase price is generally the prime consideration in the customers mind, but as Fan size is increased the power cost spread over a period of years becomes more and more important.

Factors 3, 4 and 5 all point to the larger Fan being selected because it will run at a slower speed for the same performance, it will have a greater reserve capacity, and the noise level will be less.

Generally, the larger Fan is also indicated by factor 2, because the same performance at a slower speed will require less horsepower.

It should be noted that the above statement is only applicable to what is known as a homologous series of Fans, ie; a series of Fans of similar geometric proportions increasing in size.

CHARACTERISTICS OF COMMON INDUSTRIAL FAN APPLICATIONS

CONVEYING — COMBUSTION — VENTILATION

CONVEYING: Relatively high Air Velocities are required to convey solid materials in the air stream and relatively high Static Pressure is required to overcome the resistance to flow in the ducts confining the moving air, and the pressure losses in the Hoods, Canopies, Elbows, Bends, Branches, Cyclones etc; which will be necessary in conveying systems. The solid material (often abrasive) being conveyed in the air stream means that a robust impeller will be essential.

These factors point to a centrifugal Exhaust Fan. "DAWN" Exhaust Fans are manufactured in an extensive range which includes two (2) series, "DAWN" CAST IRON EXHAUSTERS, which cover the lower end of the range, and "DAWN" MILL EXHAUST FANS, which adequately cover the middle and upper end. Rating Tables and other data concerning these Fans will be found in the fourth or Blue section of the catalogue.

COMBUSTION: Generally, Fans for this application require a high Static Pressure characteristic, however the number of applications are so wide and varied that "DAWN" Pressure Blowers are being produced in a range of design duties which cover any required performance from 120 CFM @ 7"SWG to multi-stage units rated up to 5000 CFM @ 5 PSI. For more information refer to the sixth or Green section of the catalogue.

VENTILATION: Fans for this application are required to produce Large air Volume at low air Velocity and low Static Pressure. "DAWN" Fans for Ventilating duties include Axial-Flow, Propeller, Diaphram and Multi-Vane types. Rating Tables and other data can be referred to in the third or Yellow section of the catalogue.

FAN APPLICATIONS

As a brief guide, we include here, descriptions of the selection of Fans for specific cases of the common industrial fan applications referred to above, Conveying, Combustion and Ventilation.

1. CONVEYING

1 (A) EXHAUSTING SOLID MATERIALS

For this purpose a cast iron exhauster or a mill exhaust fan will be required.

A convenient basis for exhaust fan selection is pipe or duct size, because any standard machine, for example a saw bench, a grinding wheel, a belt sander etc; will have a certain duct size necessary to contain the necessary air volume to convey the solid residue produced by the operation of the machine, at a velocity known as suitable from previous application (See Velocity Table for conveying, Table No. 6) and (Table of Duct sizes for various machinery, Table No. 2).

If there is only one source of material to be exhausted the fan will normally be that having inlet diameter closest to duct diameter. (An exception might be where an abnormally long length of duct is necessary and the system resistance is so high that a larger fan than normal is required).

Generally there will be two or more machines in the one exhaust system and in such cases the selection of fan size depends on duct area.

This is important. Where any two branch ducts join to form a main duct, the main should be of such a size that its area is 10% to 20% greater than the sum of the areas of the two individual branch ducts.

This is understood if thought is given to the air flow. The volume flowing per minute is equal to the velocity of flow multiplied by the pipe area.

ie; $Q = \bar{v} \times A$

where, Q = Volume in cubic feet per minute,

$\bar{v}$ = Velocity in feet per minute,

A = Area of duct in square feet.

Obviously to keep the same velocity of flow, (reason for necessity of constant velocity will be given shortly) where two ducts of equal flow meet, the area of the resulting main duct must be the sum of the areas of each individual duct.

In practice it is found that there are reductions of flow at junctions, hence the statement above that the main should be 10% to 20% larger in area than the sum of the seperate branches.

In such cases the fan size should have inlet size equal to the size of main Duct necessary to cater for the sum of all individual branch ducts. This fixes fan size.

The size of the fan is determined as above and the speed and horsepower to drive for the particular system have to be determined.

The speed of the fan has to be such as to give the required volume of air for conveying in the ducts against the resistance to air flow of the duct system, measured in inches water gauge.

The volume of air depends on the velocity necessary to convey the particular solid material. Obviously the lower the velocity required in any duct, the smaller is the volume required.

For any material a certain minimum velocity is required. (See Velocity table for conveying. Table No. 6).

These velocities assume that at the origin of the material to be removed a closely fitting hood of appropriate shape and good design can be fitted to catch the material as produced.

If because of some peculiaraity of operation, an efficient hood cannot be used then a higher velocity may be necessary.

Higher velocity than necessary means larger volume and larger resistance hence higher horsepower. Higher velocities also mean more noise. For these reasons the air velocities passing through the ducts are kept as little as possible above the minimum velocities necessary.

CONSTANT VELOCITY:

If Velocity is decreased below the minimum requirement, then the material being conveyed will "settle" or fall out of the air stream. This can occur if the change in main duct size, to allow for an additional branch entry, is made too large.

If Velocity is increased above the maximum requirement, then the material being conveyed can be damaged and/or excessive abrasive action can be destructive to the system. This can occur if the change in main duct size to allow for an additional branch entry, is made too small. This will also require additional Static Pressure to compensate for greater friction loss and will require higher Fan speed and hence higher Horsepower.

The volume necessary is of course the area of the main duct in square feet multiplied by the necessary conveying velocity in feet per minute. This fixes air volume.

The area of the main duct has of course been decided from the areas of the branch ducts needed for each machine as shown in the "Table of Duct sizes for various Machinery." (Table No. 2).

(It will be noted in rating tables for fan performance that figures for horsepower needed, to drive a selected fan at a speed necessary to obtain a certain volume, are given at varying resistance pressures. The volume delivered by a fan driven at a fixed speed falls off as resistance is increased, and of course the speed and horsepower necessary to deliver a certain volume are increased when the resistance against which the volume is to be delivered is increasd).

From the above, fan size and volume have been found, but to read off necessary speed (in revs. per minute) and horse power the system resistance must be known, and calculation of this is a problem in itself.

CALCULATION OF SYSTEM RESISTANCE

Consider only that part of the duct system through which the air will flow from the farthest machine to the Fan.

Among other things the system resistance depends on:

The Inlet characteristics of the duct including Hood shape if any.

The characteristics of the Outlet duct including Elbows, Cyclones etc., if any.

The number of Elbows in the run from the farthest machine to the Fan.

The actual length and sizes of duct in this run.

TO EXPLAIN THE ABOVE FACTORS

For Inlet and Outlet allow a total 2.5"swg resistance. (This does vary with particular duct velocity used).

For each section of the duct of varying diameter proceed as follows:

1. Measure the straight length.
2. Count the number of elbows.
3. Add to straight length an equivalent feet of straight length for elbows, assuming 15 times diameter (in feet) of straight length, for each elbow.

Example: 12" Saw Bench. Branch 20 ft. straight 4" pipe and 3 elbows.

Each elbow equivalent to $\frac{4 \times 15}{12} = \frac{60\text{ft.}}{12} = 5\text{ft.}$

and 3 elbows are equivalent to 15ft. and so total equivalent straight pipe = 20 + 15 = 35 feet.

4. Find resistance from chart "Friction Losses in Duct." (See Table No. 1) for each length of varying diameter.
5. Add original 2.5"swg plus all seperate sectional resistance.
6. This final figure is an approximation to the system resistance. You now have System Resistance, Volume and Fan Size, and from the rating table for the particular Fan, a value can be arrived at by direct reading or interpolation for the necessary Speed and Horsepower required to drive the Fan so that it will produce the required Volume against the calculated System Resistance.

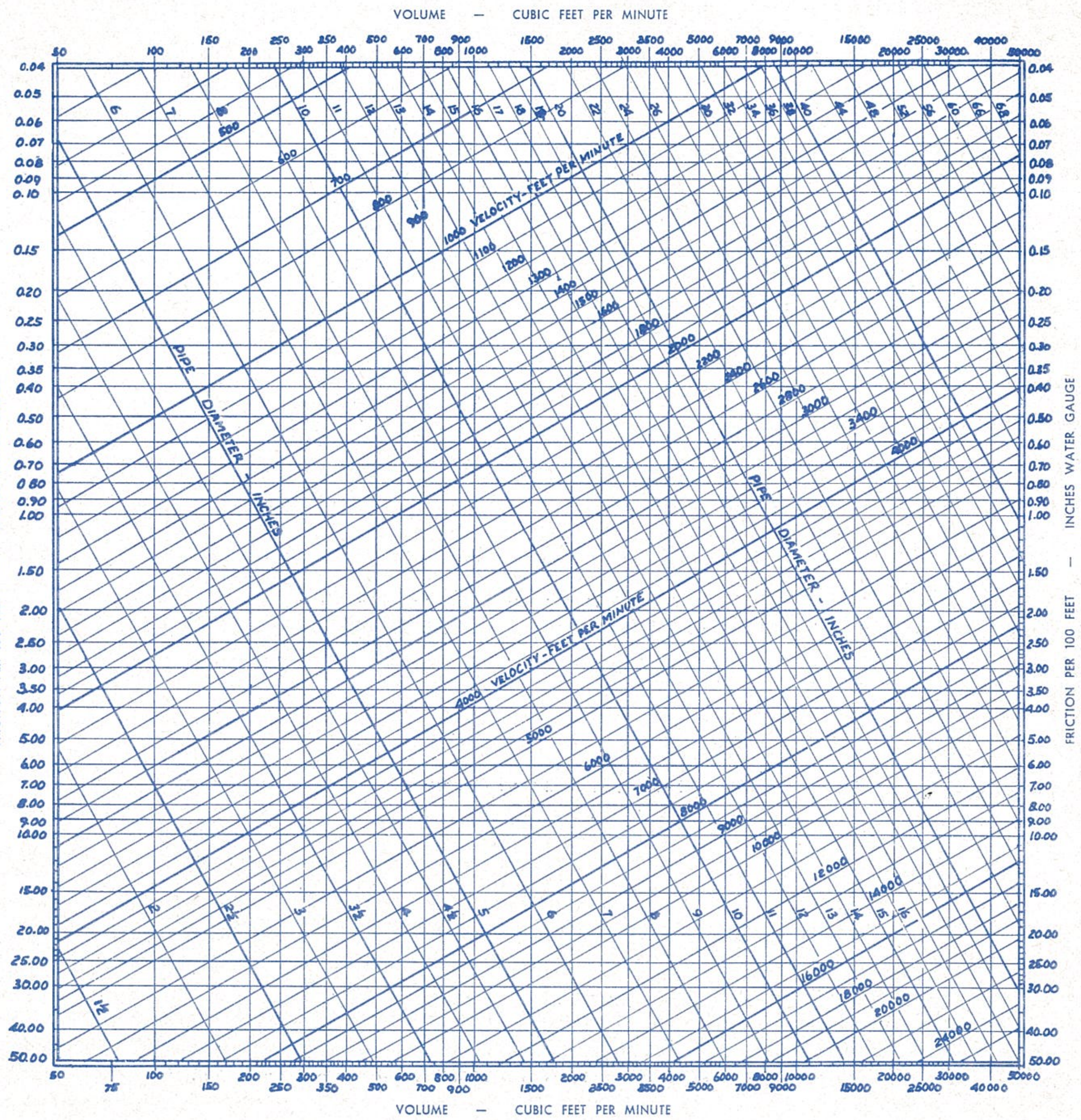
There is no accurate basis for the method of System Resistance calculation suggested above and it will only give a very approximate answer. It is not recommended for determining the actual Horsepower required but short of a rather involved explanation with further tables, it will give a rough first approximation to the System Resistance.

For a more comprehensive treatment of this subject we recommend your reference to an American publication available in Australian book stores:

"INDUSTRIAL VENTILATION" (A Manual of Recommended Practice).
by the Committee on Industrial Ventilation, P.O. Box 453, Lansing, Michigan, 48902, U.S.A.
American Conference of Governmental Industrial Hygienists.

FRICTION CHART

Table No. 1



FRICTION LOSS IN INCHES OF WATER PER 100 FT.

THIS CHART APPLIES TO SMOOTH ROUND GALVANIZED IRON DUCTS. USE TABLE BELOW FOR CORRECTIONS TO APPLY WHEN USING OTHER PIPE.

TYPE OF PIPE	DEGREE OF ROUGHNESS	ROUGHNESS FACTOR (Use as Multiplier)
CONCRETE	MEDIUM ROUGH	1.4
RIVETED STEEL	VERY ROUGH	1.9
TUBING	VERY SMOOTH	0.9

TABLE OF DUCT SIZES FOR VARIOUS MACHINERY

Table No. 2

N.B. Duct sizes are given in inches. Note well that when using these sizes to calculate volumes required duct areas must be calculated as square feet.

WOODWORKING MACHINERY	DUCT SIZE DIA. INCHES	DUCT AREA SQ. FEET
Saw Benches		
Up to 16" Saw	4"	0873
16" to 24"	4½"	.1104
Large	5"	.1364
Large Saws wet timber	6"	.1964
Jointers		
Up to 6"	4"	.0873
6" to 12"	4½"	.1104
Above 12"	5"	.1364
Thicknessers		
Up to 20"	6"	.1964
20" to 36"	7½"	.3068
Above 36"	9"	.4418
Tripple Drum Sanders		
Up to 36" Long	7"	.2673
36" to 48" Long	9"	.4418
Over 48" Long	11"	66

WOODWORKING MACHINERY	DUCT SIZE DIA. INCHES	DUCT AREA SQ. FEET
Swing Saws		
Up to 24"	4"	.0873
Over 24"	4½"	.1104
METALWORKING MACHINERY		
Grinding Wheels		
Up to 9" Diameter x 1½" width	3"	.0491
9" to 19" Diameter x 4" width	5"	.1364
19" to 30" Diameter x 5" width	6"	.1964
GENERAL		
Abrasive Belts		
Up to 3" width	3"	.0491
For each extra 2" width add ½" to duct diameter e.g. 5"	3½"	.0668
SHOE MACHINERY		
General	5"	.1364

1(B) PAINT SPRAY BOOTHS

The Victorian Factories Act specifies that every Booth shall be fitted with a system of exhaust ventilation of sufficient capacity to exhaust all spray laden or contaminated air from the booth and to prevent its escape into the work area in which the booth is situated.

Where the operator works within the booth, an air flow of not less than 100 linear feet per minute is maintained within the breathing zone of the operator.

Where the operator works outside the booth, an air flow of not less than 100 linear feet per minute is maintained across the open front of the booth.

Where practicable, the position of the exhaust is opposite the position where air enters the booth, so as to prevent the formation of pockets of vapour in the booth.

For example, with the operator standing at or about the front end of the booth, the minimum air volume required is of course the 100 ft. per minute linear velocity multiplied by the cross sectioned area of the mouth of the booth in square feet, i.e. width in feet times height in feet (of the front of the booth).

It should be carefully noted that for fast and efficient removal of paint spray laden air, exhausting of a room booth can set problems because the most convenient position for the exhaust fan (an enclosed propeller fan on a wall of the room) may not draw a uniform value from all parts of the room and some corners of the room may hold dead pockets of air which, because of the position of the actual spraying, build up paint spray concentration only diluted by the natural settling rate of the spray.

"DAWN" Enclosed Propeller Fans are used for this class of work and the size required and drive horsepower can be found from the capacity tables, knowing the volume from consideration outlined above.

It should be noted that any discharge duct should be as short as possible and at least the same diameter as the fan and that any bends in the duct should be of a generous radius.

The spray painting regulations of the Factories Act lays down certain specifications for the construction and installation of such ducts which must be observed.

With Propeller Fans the resistance to air flow in the duct is low because air velocities are low and duct diameter is large. The duct length is usually short.

Generally the volume figures from the chart are accepted and no account taken for variations in resistance to flow.

If resistance to flow increases above about ½" static water gauge pressure, it generally means that Propeller Fans are not suitable for the application and either a "DAWN" Axial-Flow Fan, which is based on aerodynamic design to overcome system resistance, or a centrifugal Fan of the "DAWN" Multi-Vane type is required.

1(C) CANOPY HOODS

Canopy hoods should always be designed to extend beyond the edge of the vat, stove, furnace etc. that the hood serves, by a distance dependent upon the height of the canopy above the top of vat or furnace and with the amount of cross draught adjacent to the installation.

"Industrial Ventilation" gives a figure of O.4D for the extension of the canopy on every side of the object covered, where D is the height between face of canopy and top of vat or furnace.

The actual air volume is dependent on experience because of the importance of draughts and type of fumes being collected.

An approximate formula is:

$Q = 1.4\ PDV.$

where Q = volume of air, cubic feet per minute.
P = perimeter of hood in feet.
D = height between face of canopy and top of vat, furnace etc. in feet.
V = control velocity across the edge of the hood and varies from 50ft. per minute, for conditions of no draught and active liberation of hot gas lighter than air, up to 500ft. per minute with strong cross draughts and gases not rising freely.

This formula is suggested in "Industrial Ventilation."

The duct size above the canopy depends on the exhaust volume and should be of such an area that the linear velocity in the duct is not more than 3000 feet per minute.

Generally a Multi-Vane or an Axial-Flow Fan is used for these applications.

The volume through the fan is of course the volume through the duct. The resistance pressure to be overcome by the fan is very approximately for inlet and outlet 1"swg plus the duct resistance.

The duct resistance can be calculated as for ducts in "Exhausting Solid Materials" and from the volume, total system resistance and duct diameter, the correct Fan size, speed and horsepower can be selected from the Rating Tables.

2. COMBUSTION

Oil Burner Air Supply

The volume of air to be supplied by a fan to any oil burner depends on the type of burner and the quantity of oil to be burnt. It is usually available from the Burner manufacturer.

Certain burners obtain most of the combustion air from the fan. The air through the burner is sufficient to atomise the oil and also for complete combustion.

Other burners require only portion of the combustion air for atomisation and the balance of the combustion air is introduced round the outside of the burner by an induced or forced draught as in steam boiler applications. In these cases the fan only has to supply the atomising air.

For normal small burner operation on furnaces of various types it can be assumed that the air to be supplied by the fan is the total combustion air.

This means that for any burner the quantity of air in cubic feet per minute required from the fan should be 35 cubic ft. per minute for each gallon of oil per hour to be burnt. (35 CFM/GAL/HR).

The required static pressure against which the calculated volume must be delivered depends upon the characteristics of the burner. It is usually high to overcome the resistance in the system which generally consists of, small pipes, valves, nozzles etc. all of which offer high resistance to Air flow.

Some low volume burners operate against a pressure of about 7 inches SWG. Others operate against a pressure of 10" - 14"SWG. Others again operate against a pressure of 18" - 21"SWG and others in the 24" - 28"SWG range.

The capacity of "DAWN" Oil Burner Blowers can be found from "DAWN" Oil Burner Blower and "DAWN" High Pressure Blower brochures in the Green section of the catalogue, and for any particular oil burner the Fan having both capacity and delivery pressure closest to, but above, the burner requirements will naturally be chosen.

3. VENTILATION

Ventilation is basically the process of supplying fresh air to an enclosed area, and although there are many aspects which should be considered, it is fundamentally a matter of changing the air in the area, for fresh air, a given number of times in a period in accordance with accepted practices.

The main points to be considered when planning the ventilation of a building are:

(a) Volume of air content (stationary air) i.e. L x W x H in cubic feet.
(b) Purpose for which building is used, i.e. Toilet, Kitchen etc.

The following table shows a range of recommended air changes per hour for various types of buildings:

AIR CHANGES FOR VENTILATION **Table No. 3**

	Min.	Max.		Min.	Max.
Auditoriums	4	10	Hospitals	4	8
Assembly Halls	4	10	Hotels	**6	10*
Bakeries	20	40	Kitchens	15	25
Boiler Rooms	20	30	Library	4	6
Cafes	10	15	Laboratories	4	8
Canteens	4	8	Museums	3	4
Dance Halls	10	20	Offices	4	6
Dyers & Dry Cleaners	20	30	Painting Shops	30	50
Engine Rooms	20	30	Residence	2	4
Factories	8	12	Retail Houses	6	12
Foundries	10	20	School Rooms	4	6
Garages	6	10	Toilets	12	15

From the table above and from the dimensions of the Room to be ventilated we can calculate the volume of air required per minute by the Fan from the following equation.

$$\frac{\text{Length x Width x Height x Air Changes / hour}}{60} = \text{CFM}$$

e.g. Find Max volume of air required to ventilate a Hotel Bar 60' long x 40' wide x 15' high:

$$\frac{60 \times 40 \times 15 \times 10^{*}}{60} = 6000\ \text{CFM}$$

NOTE: The Minimum air volume for this Hotel Bar would be,

$$\frac{60 \times 40 \times 15 \times 6^{**}}{60} = 3600\ \text{CFM}$$

PRESSURE CONVERSION TABLE

Table No. 4

	Oz/Sq In.	"Hg	"wg	Lb/Sq In.	Lb/Sq Ft.
1 Oz/Sq In.	1.0	.1272	1.733	.0625	9.0
1" Hg.	7.86	1.0	13.63	.491	70.56
1" Wg.	.5774	.074	1.0	.0362	5.196
1 PSI	16	2.036	27.74	1.0	144
1 PSF	.111	.0142	.192	.00694	1.0
1 Atmosphere	234.5	29.92	407	14.694	2116.3

STANDARDS USED in Manufacture & Testing of "DAWN" Fans

Table No. 5

STANDARD No. MOTORS	DESCRIPTION
MOTORS	
BS.170:1962	Performance. Up to 2HP
BS.2048:Pt1 1961	Dimensions. Up to 2HP
BS.2613:1957	Performance Class "E" Insulation
BS.2960:Pt 1&2	Dimensions 3HP up to 100HP
AS.No. C98-1961	Flameproof Motors
BS.229:1957	Flameproof Enclosure of Electrical Apparatus
BS.2083:1956	Dimensions. Single Phase Intergral HP Motors
TESTING	
BS.848:1963	Testing of Fans for general purposes
BS.1042:1943	Flow Measurement
FMA. Code 3 1952	Fan performance Tests
BS.380:1958	Performance of Desk-Type Electric Fans
BS.367:1941	Ceiling Type Electric Fans
BS.726:1957	Air Flow Measurement (Compressors & Exhausters)
BS.269:1927	Methods of declaring efficiency of Electrical M/C.
BS.89:1954	Electrical Indicating Instruments
MATERIALS	
BS.1490	Aluminium Casting Alloys
BS.1452:1956	Grey Iron Castings
BS.970:1955	Wrought Steels
BS.1449:1956	Steel Plate, Sheet & Strip
BS.46:Pt1:1958	Keys & Keyways
BS.84:1956	Whitworth Screw Threads
BS.10:1962	Steel or Cast Flanges

VELOCITIES (FPM) for EXHAUST SYSTEMS & PNEUMATIC CONVEYING

Table No. 6

MATERIAL	Velocity (FPM)	MATERIAL	Velocity (FPM)
Asbestos Dust	2000	Jute Dust	2500
Ashes, Small Clinkers	8000	Lead Dust	4500
Barley	6000	Leather Dust	2500
Bakelite Powder	3250	Limestone (Powdered)	5500
Bakelite Dust	2250	Metal Dust	4500
Brass Turnings	5000	Napping Machine Dust	2500
Buffing Wheel Dust	4000	Nylon (Teased)	2500
Castor Beans	5000	Oats	4500
Cast Iron Filings	7000	Paper	5000
Cement	7500	Paper Lint (Fine)	2000
Coal (Fine)	4000	Rags	4500
Coffee Beans	3250	Rubber	4500
Cork (Ground)	4000	Rubber Dust	2000
Corn	6000	Salt	6000
Cotton	4000	Sand	7000
Cotton Lint	2000	Sander Machine Dust	2000
Cotton Seed	4500	Shoe Dust	3500
Dry Dust & Powder	2500	Sugar	6000
Emery Grinding Dust	3500	Vegetable Pulp (Dry)	4500
Flour	3500	Wood Sawdust (Dry)	3500
Gas & Fumes	2000	Wood Sawdust (Wet)	4000
Grain Dust	2500	Wood Shavings (Dry)	3500
Granite Dust	3500	Wood Shavings (Wet)	4000
Hops	5000	Wood Knots	4500
Iron Oxide	6500	Wool	4500
Jute	5000	Wheat	5500

The above Velocities are given only as a guide. The speeds shown against the more common materials are known to be adequate for normal circumstances, the others are taken from a variety of published data and would need to be confirmed through trials or actual practice.

There are several factors which must be considered when planning an Exhaust or Conveying system, of these probably the most important are:

(a) Risk or Damage to the Material being Conveyed.
e.g., Materials such as Coffee Beans, Wheat, Sugar etc., can be broken or otherwise seriously damaged, particularly if conveyed through the Fan.

(b) Risk of Fire or Explosion.
e.g., Airborne finely divided solids having high calorific value should be treated as Explosive. These include Powdered Aluminium, Flour, Grain Dust, Pulverised Fuel, Hard Rubber Dust etc. (See page 18).

"DAWN" Fans used for Exhaust Systems and Pneumatic Conveying include Cast Iron Blowers & Exhausters No's. 2½B to 6B, and Mill Exhaust Fans No's 9 to 24. These have Inlet sizes from 3" to 24" and Capacities from 100 CFM @ .25" wg to 20,000 CFM @ 10" wg.

REDUCTION OF FIRE and EXPLOSION HAZARD

(John L. Alden, M.A.S.M.E. 1949)

The following rules for the design of exhaust systems will minimize the likelihood of disasterous fires or explosions.

1. Do not connect spark-producing process such as grinding to exhaust systems handling combustibles.
2. Install adequate traps to take out heavy metallic objects near the source.
3. Do not run pipes through fire walls unless absolutely unavoidable.
4. Install automatic fire dampers on both sides of a wall through which an exhaust pipe passes.
5. Do not depend on solder for structural strength of any joint.
6. Allow at least 6" clearance between exhaust pipes and floors, ceilings or combustible materials.
7. Electrically ground all Fans, piping, dust arresters and motors.
8. Treat as explosive all air-borne finely divided solids having considerable calorific value.
9. When handling explosive substances, locate collectors on the roof or otherwise safely isolated.
10. Non-Ferrous Fans are an added safety precaution against fire or explosion.

    This includes Flour, Starch, Grain Dust, Sugar, Milk Powder, Ground Spices, Phenolic Moulding Powder, Pulverized Fuel Wood Flour, Aluminium Buffings, Hard Rubber Dust and Sulphur Dust.

The above rules are offered as suggestions only and may not be acceptable in the district in which the system is being installed. We therefore recommend your reference to Government Regulations controlling Fire and Health risks in your state.

VELOCITIES of STANDARD (Dry) AIR at VARIOUS VELOCITY PRESSURES

Calculated from formula according to BS. 848 : 1963.

Air Temperature 68°F. Barometric Pressure 30"Hg. Density .075 Lb/Ft³

Table No. 7

P_v "wg	0	.1	.2	.3	.4	.5	.6	.7	.8	.9
0		1260	1790	2190	2530	2830	3100	3350	3580	3800
1	4000	4200	4380	4560	4730	4900	5060	5220	5370	5520
2	5660	5800	5940	6070	6200	6330	6450	6580	6700	6820
3	6930	7050	7160	7270	7380	7490	7590	7700	7800	7910
4	8010	8100	8200	8300	8400	8490	8590	8680	8770	8860
5	8950	9040	9130	9220	9300	9390	9470	9560	9640	9720
6	9810	9890	9970	10050	10130	10210	10280	10360	10440	10520
7	10590	10670	10740	10820	10890	10970	11040	11110	11180	11250
8	11320	11390	11470	11530	11600	11670	11740	11810	11870	11940
9	12010	12080	12140	12210	12270	12340	12400	12470	12530	12590
P_v "wg	+0	+1	+2	+3	+4	+5	+6	+7	+8	+9
10	12660	13280	13870	14440	14980	15510	16020	16510	16990	17450
20	17910	18350	18780	19200	19620	20020	20420	20810	21190	21560
30	21940	22290	22650	23000	23350	23690	24030	24360	24680	25010
40	25330	25640	25970	26260	26560	26860	27160	27450	27750	28030

Table above is of Slide Rule accuracy only.

For more accurate results use formula as per BS. 848 : 1963.

$$v = 109\sqrt{\frac{P_v}{w}} = \text{ft/min.}$$

TABLE OF AREAS OF CIRCLES

Table No. 8

For use with Velocity Table to calculate Air Volume in Duct

DIAMETER IN INCHES	AREA SQUARE FT.	DIAMETER IN INCHES	AREA SQUARE FT.	DIAMETER IN INCHES	AREA SQUARE FT.	DIAMETER IN INCHES	AREA SQUARE FT.
½	.00136	3½	.0668	7	.2673	14	1.069
¾	.00307	3¾	.0767	7½	.3068	15	1.227
1	.00545	4	.0873	8	.3491	16	1.396
1¼	.00852	4¼	.0986	8½	.3942	17	1.576
1½	.0123	4½	.1104	9	.4418	18	1.767
1¾	.0167	4¾	.1231	9½	.4923	19	1.969
2	.0218	5	.1364	10	.5454	20	2.182
2¼	.0276	5¼	.1504	10½	.6010	21	2.405
2½	.0341	5½	.165	11	.66	22	2.64
2¾	.0412	5¾	.184	11½	.7215	23	2.885
3	0491	6	.1964	12	.7854	24	3.142
3¼	.0576	6½	.2304	13	.9218	25	3.409

TEMPERATURE CONVERSION

Temperature for calculation of Air Density in Fan engineering is based on the Rankine scale. This is an extension of the Fahrenheit scale calculated from Absolute Zero; a point where research in physics has proved that molecular kinetic energy is zero, and beyond which further decreases in temperature is reckoned impossible, this point is variously given at between −459.2°F and −459.7°F. For the sake of simplicity and in line with BS-848 : 1963, we use 460°F as Absolute Zero.

Absolute Zero using the Centigrade scale is --273°C (Kelvin Scale).
Therefore ABSOLUTE TEMPERATURE on the Fahrenheit scale = °F + 460 = °R and Absolute Temperature on the Centigrade scale = °C + 273 = °K.
It will often be necessary and useful to convert Centigrade Temperatures to Fahrenheit and vice versa. There are many published tables in trade reference books for converting temperature and these are very useful, however they cannot give every possible figure and for that reason we use the following simple equations:

To convert degrees Centigrade to degrees Fahrenheit,

$$°F = \frac{9 \times °C}{5} + 32$$

To convert degrees Fahrenheit to degrees Centigrade,

$$°C = \frac{5 \times (°F - 32)}{9}$$

SPEED - PERFORMANCE FACTORS

Table No. 9

Factors to calculate Fan Performance at a particular Point of Rating with change of Fan Speed.

SPEED RPM	VOLUME CFM	STATIC PRESSURE Ps" wg	HORSE POWER BHP	SPEED RPM	VOLUME CFM	STATIC PRESSURE Ps" wg	HORSE POWER BHP
750	.263	.069	.018	2750	.965	.93	.9
970	.34	.116	.039	2850	1.0	1.0	1.0
1000	.35	.123	.043	3000	1.05	1.11	1.16
1250	.438	.192	.084	3250	1.14	1.3	1.48
1440	.505	.255	.123	3500	1.23	1.5	1.84
1500	.526	.276	.145	3750	1.32	1.73	2.27
1750	.614	.376	.23	4000	1.4	1.96	2.75
2000	.7	.49	.345	4250	1.49	2.22	3.3
2250	.787	.62	.486	4500	1.58	2.49	3.92
2500	.88	.77	.675	5000	1.75	3.08	5.4

The above Table is based on known Fan Performance at 2850 RPM, (Direct Coupled 2 Pole 50 Cycle Motor Speed), and Fan Laws which state:

Volume varies directly as the Speed Ratio.
Pressure varies as the square of the Speed Ratio.
Horsepower varies as the cube of the Speed Ratio.

eg. A Fan rated to supply 1000 cfm at 28"wg at 6hp at 2850 rpm, when driven at 3500 rpm would supply 1230 cfm at 42"wg at 11 hp.

ie. 1000 x 1.23 = 1230 cfm.
28 x 1.5 = 42"wg.
6 x 1.84 = 11 bhp.

STANDARD FLANGES AS USED ON "DAWN" FABRICATED STEEL CASE PRESSURE BLOWERS

BS.10:1962 (Table A)

Table No. 10

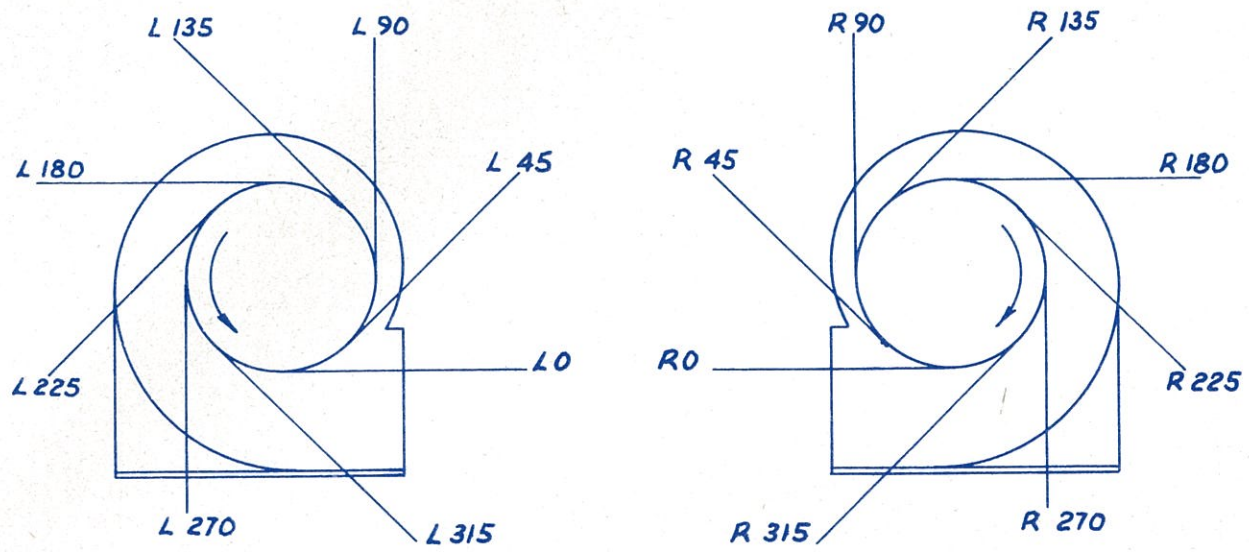
Flange size Designation (nominal bore of pipe)	Approx. O/D of Steel Pipe	Diameter of Flange	Bolt PCD.	Number of Bolts	Dia. of Bolts	Flange Thickness	
						Cast Iron	Steel
2"	2-3/8"	6"	4½"	4	5/8"	5/8"	5/16"
2½"	3"	6½"	5"	4	5/8"	11/16"	5/16"
3"	3½"	7¼"	5¾"	4	5/8"	11/16"	3/8"
3½"	4"	8"	6½"	4	5/8"	3/4"	3/8"
4"	4½"	8½"	7"	4	5/8"	3/4"	3/8"
5"	5½"	10"	8¼"	4	5/8"	3/4"	1/2"
6"	6½" or 6-5/8"	11"	9¼"	4	5/8"	13/16"	1/2"
7"	7-5/8"	12"	10¼"	8	5/8"	13/16"	1/2"
8"	8-5/8"	13¼"	11½"	8	5/8"	7/8"	1/2"
9"	9-5/8"	14½"	12¾"	8	5/8"	7/8"	5/8"
10"	10¾"	16"	14"	8	3/4"	15/16"	5/8"
12"	12¾"	18"	16"	8	3/4"	15/16"	3/4"
13"	14"	19¼"	17¼"	8	3/4"	15/16"	3/4"
14"	15"	20¾"	18½"	8	7/8"	1"	7/8"
15"	16"	21¾"	19½"	8	7/8"	1"	7/8"

Holes for 5/8" Bolts are 11/16" ϕ, 3/4" Bolts are 7/8" ϕ, for 7/8" Bolts are 1" ϕ

Table No. 11

DESIGNATION of CASED CENTRIFUGAL FANS

BS. 848 : PART 1 : 1963

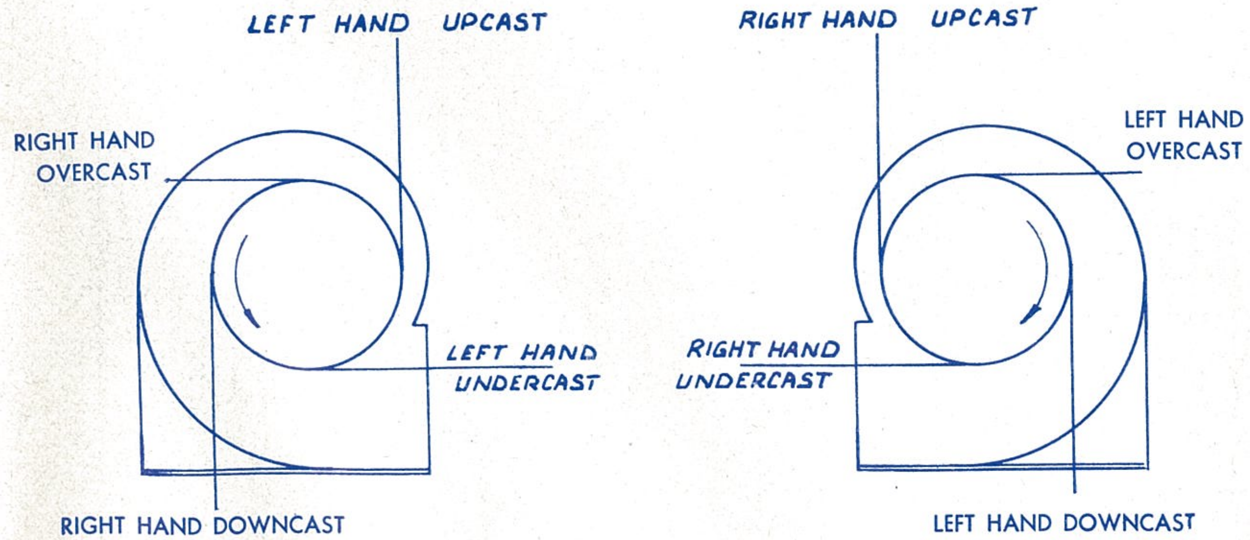


b. COUNTER - CLOCKWISE

a. CLOCKWISE

VIEWED FROM DRIVING SIDE

ORIGINAL DAWN DESIGNATION CHART

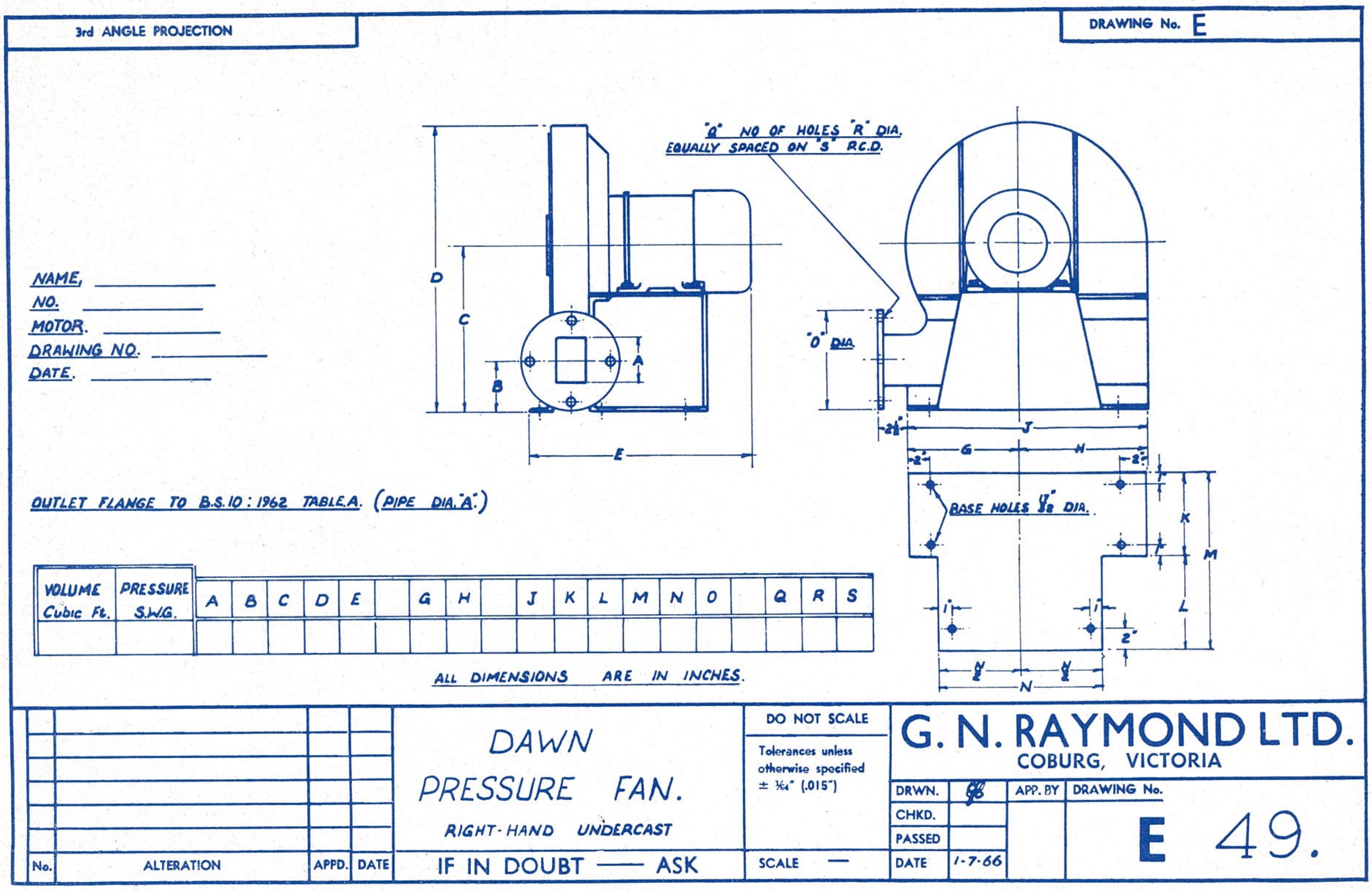


b. COUNTER - CLOCKWISE

a. CLOCKWISE

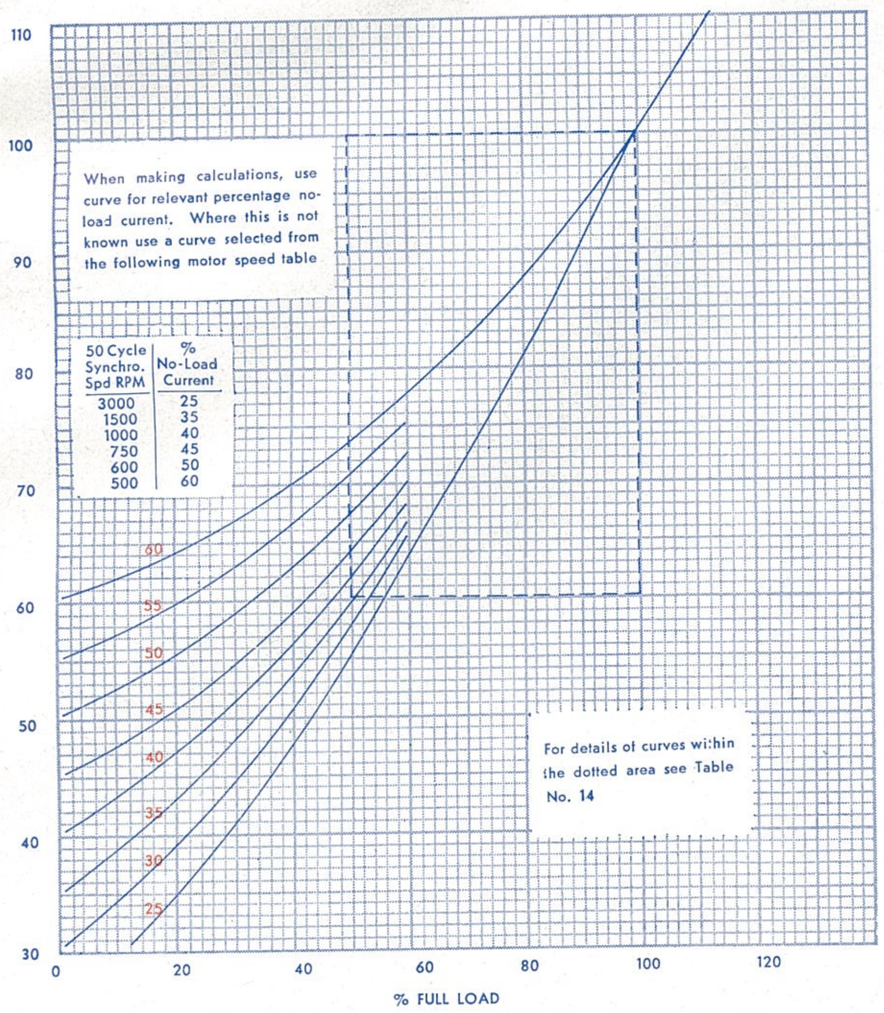
VIEWED FROM DRIVING SIDE

Table No. 12



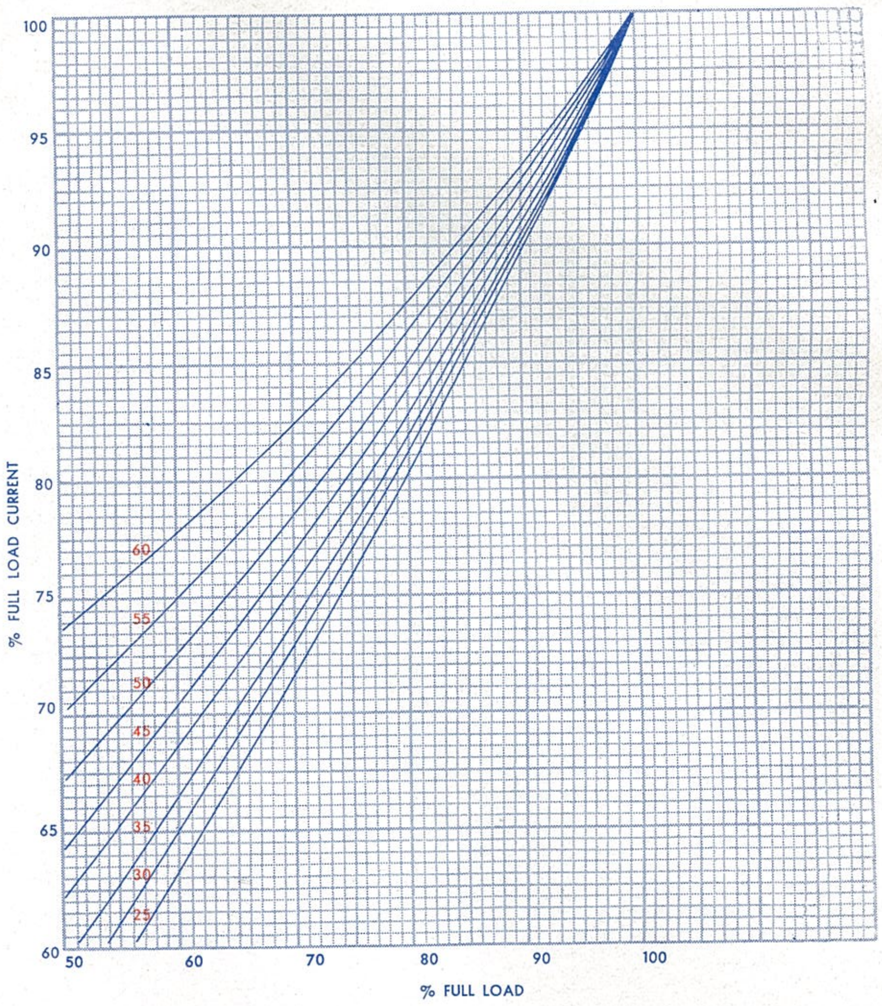
Example of Certified Dimension Drawing available to customers for Fabricated Steel Fans.

LOAD-CURRENT CURVES for 3 PHASE INDUCTION MOTORS



Typical load-current curves for 3-phase induction motors.

TABLE No. 13



Enlarged part-curves from Table No. 13

TABLE No. 14