TRAINING MANUAL

Guide from Elta

A MEMBER OF eg ELTA GROUP

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1 FAN TERMINOLOGY

These notes are designed to explain some of the terms which are used in describing the characteristics of fans and the relationship between the fan performance and the duct system curve.

1.1 Pressure

The term 'Pressure' is used in a number of ways that can, at times, be confusing. These are described below.

1.1.1 Static Pressure

In Fan Engineering, Static Pressure is the difference between the pressure at a point and the atmospheric pressure. A simple example of Static Pressure is the pressure that is inflating a balloon or a tyre; it acts in all directions. However, in a duct system, Static Pressure is a measure of the resistance to the flow of air caused by the duct and components in the duct and it acts in all directions.

For air to travel along a duct it requires an amount of energy, or Static Pressure, to overcome the friction generated by the air. The value of that energy is governed by the size of the duct and the amount of air travelling along it. The more air being handled the greater the amount of Static Pressure required. Also, if a damper, cooling coil, filter, air grille, series of bends etc. is added to the duct system the Static Pressure will have to be increased to overcome the resistance to airflow of these components.

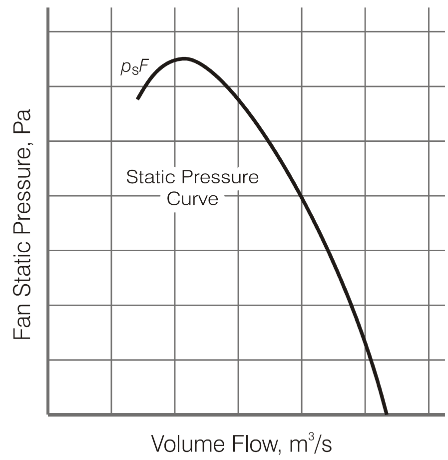


Fig. 1.1 – Static Pressure Fan Curve

Static Pressure can be a positive or a negative value. A negative value for pressure would be found at the intake, or upstream side of the fan, where the fan is sucking in the air.

The value of the Static Pressure at the fan inlet is the measure of the energy required to overcome the resistance to flow to this point. There would be a positive value on the discharge, or downstream side of the fan, where the fan now has to have enough energy, or Static Pressure, to blow the air through the ductwork. Whether it is on the suction or discharge side of the fan, pressure will be required to move the air along the duct.

The value of Fan Static Pressure is expressed as ps F and measured in Pascals, Pa.

Static Pressure is the value used most commonly to select fans but, in a 'purists' sense, Total Pressure should be used. Using Static Pressure gives a small additional factor of safety.

1.1.2 Velocity Pressure

Air flows naturally from a region of high pressure to one of lower pressure; it is this difference in pressures that cause winds. When these differences are extreme, we get cyclones and similar extreme climatic conditions. Its speed of flow, or velocity, depends on the resistance met with by the air stream. Air, like everything else that moves, exerts pressure against obstructions relative to its speed. This is apparent from the way trees bend in a strong wind. The pressure due to wind is called Velocity Pressure.

In a duct system, the Velocity Pressure is determined from the amount of air being handled and the cross-section of the ductwork.

Fig. 1.2 illustrates the velocity curve on the discharge of a fan.

Velocity Pressure is very important, as its value is used to determine the pressure losses throughout the system.

The value of Velocity Pressure is expressed as pd and measured in Pascals, Pa.

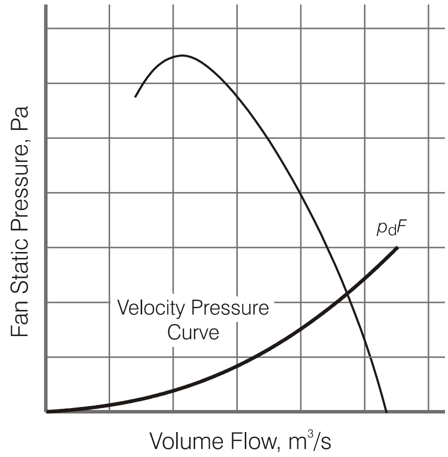


Fig. 1.2 - Velocity Pressure Curve of a fan discharge

Fan Velocity Pressure is determined using the same formulae, but is based on the velocity at the fan discharge as opposed to somewhere along the duct system where its value could have changed due to changes in the duct dimensions etc, which could impact on the velocity of the air.

Velocity Pressure is determined by the following formulae: -

Velocity Pressure, pd $= 0.5\varrho V^2$, or

$= 0.6V^2$, (when handling Standard Air)

Where: -

$\varrho = 1.2$ kg/m^3 (Standard air density at 20°C and 1013mb atmospheric pressure)

V = duct air velocity, m/s

To derive the fan velocity curve using the above formula, the fan diameter or dimensions of the fan discharge need to be known, so that the velocity at various airflows can be determined. By then entering the velocities in the formula, a series of values in Pascals (Pa) will be obtained and plotted.

1.1.3 Total Pressure

The Static Pressure developed by a fan is the pressure it can build up in order to move air against resistance as explained above. In all air movement there is some Velocity Pressure and some Static Pressure according to the resistance of the system.

The Total Pressure developed by a fan is simply the sum of the Static Pressure and Velocity Pressure.

The value of Total Pressure is expressed in Pascals, Pa and by the letters $p_t F$.

Hence, Total Pressure, $p_t F = p_s F + p_d F$

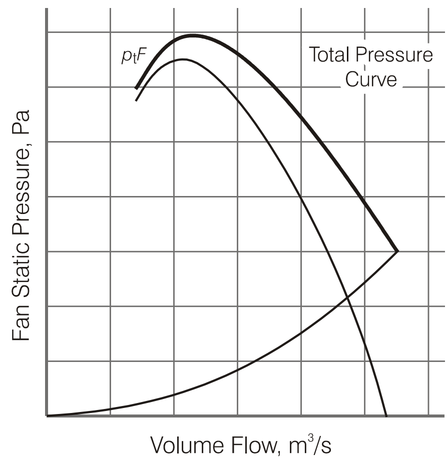


Fig. 1.3 – Total Pressure Curve

1.1.4 Peak Pressure

The term Peak Pressure is used to determine the maximum pressure a fan is capable of developing. This value is determined by testing the fan in a suitable test rig.

Peak Pressure is measured in Pascals, Pa.

The graph shows the performance curve of a fan. The Peak Pressure of the fan is the highest point of the curve.

If a fan is selected above the Peak Pressure point it will not perform satisfactorily, in fact it will run in a condition called ‘Stall’. In practice we do not recommend any selections above 80% of Peak Pressure without great care being taken. The reason for this is that the science of estimating the pressure requirements of a duct system is not precise. In addition, variations occur between the duct design and what is actually installed making accurate estimates quite difficult.

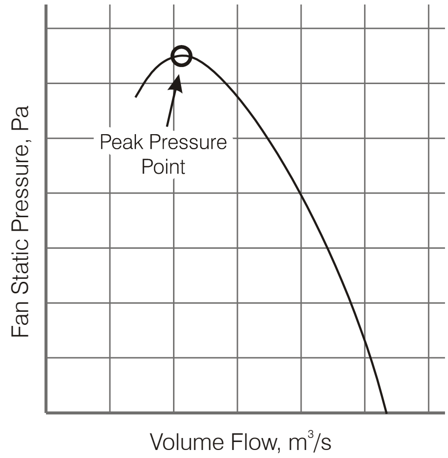


Fig. 1.4 – Peak Pressure point

1.1.5 Stall

In an aircraft, when Stall occurs, the smooth flow of air over the aircraft wings breaks down, turbulence is generated, the wings are unable to maintain lift (pressure) and the aircraft falls from the sky. If an aircraft is high enough in the air it may recover sufficient speed as it falls for lift to be re-established and for it to continue to fly and land safely.

It is not quite as dramatic with a fan, but the principle is exactly the same. Stall will occur when the fan can no longer develop the pressure (lift) required to deliver the required volume of air. Put another way, the smooth flow of air over the fan blades breaks down, turbulence is generated and the fans performance is degraded. Axial fans with medium to high pitch angles are particularly prone to stall.

The area of Stall on a fan curve is anywhere to the left of the Peak Pressure point, which is one reason why fans should not be selected to operate near this point.

When in Stall, a fan, apart from not performing as required, generates more noise, usually with a characteristic rumble. Occasionally there can be a tendency for the performance from the fan to ‘run’ up and down the curve, creating a condition called ‘hunting’, which can be quite acoustically unpleasant and can cause physical damage to the impeller.

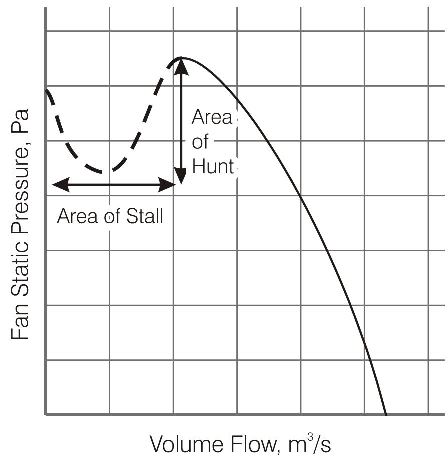


Fig. 1.5 – Stall Region of curve

1.1.6 System Pressure

System Pressure, also known as System Resistance, is the total of all the pressure losses which occur in a ducted system before and after the fan by the flow of air through the duct system. The airflow is determined by the application e.g. kitchen or toilet exhaust or office ventilation etc.

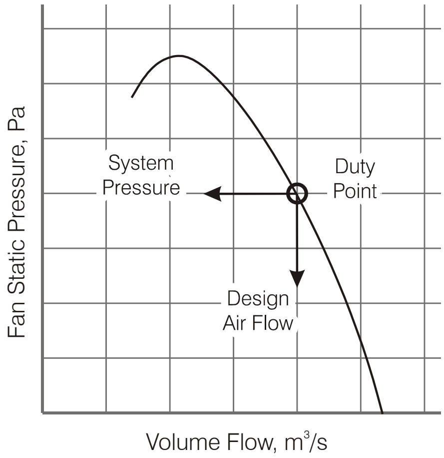


Fig. 1.6 – showing the Duty Point on fan curves

The estimation of the System Pressure takes into account pressure losses through all the components in the system including filters, silencers, heating/cooling coils, inlet and outlet grilles and louvres, bends in the duct and the duct itself. System Pressure is expressed in Pascals, Pa, but ALWAYS in conjunction with the airflow that was used to determine the pressure losses through the system. The importance of this will be explained in greater detail later.

As we now know the Airflow and the System Pressure we can put these values onto a graph as shown in Fig. 1.6. These two values, when put together, give us a point which is called the Duty Point, and it is this combination which is used to select the fan.

Although the System Pressure figure that has been calculated is actually a Total Pressure, in practice the value is taken as the Fan Static Pressure. The difference between the Fan Static Pressure and the System Pressure is the difference in the velocity pressures at the fan discharge and at the end of the duct system. As the velocity on the discharge of a system is generally quite low the difference provides a negligible safety margin.

1.1.7 System Curve

There are a number of laws that govern how fans and duct systems perform and the one governing Ducted Systems is called the square law. For any duct system, the resistance to flow of the air will increase as the velocity increases or decrease as the velocity decreases; the velocity of the air flow is directly related to the amount of air being handled. Once the System Pressure is determined, (see 1.6) a system curve can be produced using the square law. From this curve the pressure loss for any airflow in this particular system can be determined.

The square law formula for pressure change with velocity is: -

$$p_2 = p_1 \times \left(\frac{q_{v2}}{q_{v1}} \right)^2$$

Where:

p_1 = known pressure drop for the system at air quantity, qv1
p_2 = pressure drop at new air quantity, qv2
q_{v1} = known airflow
q_{v2} = new airflow

The System Curve will intersect with the fan curve at the Duty Point, which is always the fan operating point. To draw a System Curve you need to know the pressure drop through the system at a particular airflow and, by then substituting other airflows into the formula, you can then determine the pressure drop for the other airflows. Plotting these points on a graph, and joining the points together, creates the System Curve.

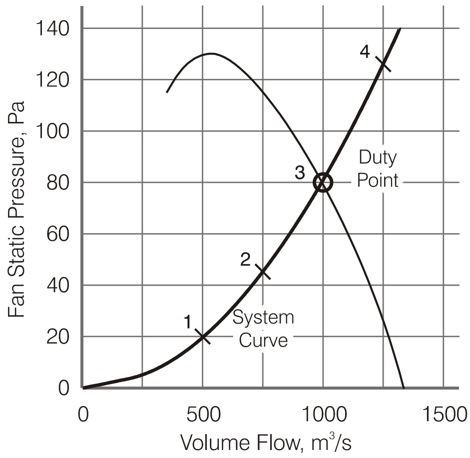


Fig. 1.7 – Creation of System Curve

Examples on the application of this formula: -
Assume a duty requirement of 1000L/s @ 80Pa.

1/ Determine the pressure loss with 500L/s

$$p_2 = p_1 \times \left(\frac{q_{v2}}{q_{v1}}\right)^2 = 80 \times \left(\frac{500}{1000}\right)^2 = 80 \times 0.5^2 = 80 \times 0.25 = 20 \text{ Pa}$$

2/ Determine the pressure loss with 750L/s

$$p_2 = 80 \times \left(\frac{750}{1000}\right)^2 = 80 \times 0.75^2 = 80 \times 0.5625 = 45 \text{ Pa}$$

3/ Determine the pressure loss with 1000L/s

$$p_2 = 80 \times \left(\frac{1000}{1000}\right)^2 = 80 \times 1^2 = 80 \times 1 = 80 \text{ Pa}$$

4/ Determine the pressure loss with 1250L/s

$$p_2 = 80 \times \left(\frac{1250}{1000}\right)^2 = 80 \times 1.25^2 = 80 \times 1.5625 = 125 \text{ Pa}$$

These points are now drawn onto the graph.

1.1.8 Peak Power

Peak Power is the term used to describe the maximum motor power a fan of a set size will require under a specific set of circumstances. In most instances, as far as we are concerned, the fan rotational speed is the basis, but, in the case of axial flow fans, it would also be at a specific pitch angle of the fan blade (see Axial Fans 2.2 for an explanation of pitch angle).

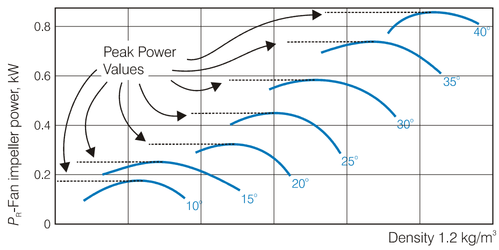


Fig. 1.8 – Peak Power Points

Peak Power is expressed in Watts (W) or Kilowatts (kW). The diagram, Fig 1.8, shows the absorbed power curves for an axial flow fan. Each curve is for a specific pitch angle of the fan blade and the highest absorbed power value for each curve is what is termed the Peak Power.

With some fan types, the power curve from free air to peak pressure is fairly flat; the variation may be as low as 10-20%. These fans are commonly called 'non-overloading' as, with only a small sacrifice, the motor can be selected to accommodate the power requirements at all pressures at the selected fan speed.

Non-overloading fan types are most axial flow, backward-curved centrifugal and mixed-flow fans. Forward-curved fans have a steep power curve with the maximum power being absorbed at the free air condition; i.e. when the fan is not working against any resistance. These are considered to be overloading fans and the motor is generally selected to accommodate the duty power requirements plus a safety margin.

1.1.9 Fan Efficiency

Some sources refer to both Fan Total Efficiency and Fan Static Efficiency but only the former is a true value.

Fan Total Efficiency is based on the total pressure and Static Efficiency is based on the static pressure. So, if a fan, such as a propeller fan, were delivering air against zero Static Pressure, its efficiency would be zero. This would obviously be wrong as the fan is working but, nevertheless, the term is used.

The efficiency of a fan is a measure of the quality of the design of the impeller and the fan casing. It is based on the airflow being handled, the pressure being developed and the amount of energy required to do that work. It is worth noting that the efficiency value quoted does not, generally speaking, take into account the efficiency of the motor driving the fan yet this can have a significant impact on the energy consumed, particularly with small motors which can be very inefficient.

The formula to determine Fan Total Efficiency is as follows: -

$$\text{Fan Total efficiency, \%} = \frac{q_v \times p_t F}{10 \times P_R}$$

Where: -

q_v = volume flow m^3/sec
$p_t F$ = fan total pressure, Pa
P_R = power absorbed by the fan, kW

To calculate Static Efficiency simple change the Fan Total Pressure, $p_t F$, value for the fan static pressure, $p_s F$ value.

Selecting a fan on the basis of Total Efficiency can be misleading as illustrated by the example shown on the next page.

Example: -

If we have a duty of $40m^3/s$ (40,000 l/s) @ 500Pa, limit the selections to axial flow fans between 1400 and 1800mm diameter and to speeds between 960r/min and 1440r/min we get the selections as shown in Fig. 1.9 from our selection software.

Product	Type	Dia.	Spd	AkW	Volt.	TEff%	dBA	Cost%	SFP
LC140R4-A9/30	Long-Cased A...	1400	1440	46.31	400	82%	85	100	1.23
LC140R4-A12/27	Long-Cased A...	1400	1440	47.01	400	78%	87	117	1.23
LC140T4-A6/28	Long-Cased A...	1400	1440	53.88	400	68%	88	120	1.42
LC140X4-A6/27	Long-Cased A...	1400	1440	56.25	400	65%	87	121	1.48
LC140X4-A9/25	Long-Cased A...	1400	1440	60.74	400	60%	88	163	1.59
LC140X4-A12/24	Long-Cased A...	1400	1440	63.14	400	58%	89	165	1.65
LC160X6-A9/30	Long-Cased A...	1600	960	44.49	400	68%	83	167	1.18
LC160X6-A12/27	Long-Cased A...	1600	960	44.49	400	68%	84	169	1.18
LC180X6-A9/20	Long-Cased A...	1800	960	33.92	400	77%	86	133	0.90
LC180T6-A9/21	Long-Cased A...	1800	960	35.22	400	77%	86	131	0.93
LC180T6-A6/26	Long-Cased A...	1800	960	37.06	400	71%	85	128	0.98
LC180X6-A6/25	Long-Cased A...	1800	960	37.59	400	70%	85	130	1.00
LC180X6-A12/19	Long-Cased A...	1800	960	38.23	400	74%	91	181	1.01

If we were to focus our selection solely on the Total Efficiency the obvious selection would be the LC140R4-A9/30, which has a Total Efficiency of 82%. This fan absorbs 46.31kW and has a noise level of 85dB(A) at 3 metres. However, if we look at the absorbed power column (AkW), the fan consuming the least energy is the LC180X6-A9/20 as it absorbs only 33.92kW, yet its Total Efficiency is only 77%. Its noise level is 1dB(A) higher at 86dB(A) and it is 33% more expensive.

If this fan were to run 12 hours/day, 5 days a week and 52 weeks a year the running cost saving with the LC180 fan, with electricity at £0.11/kWh, would be: -

$$\text{Cost saving} = (46.31-33.92) \times 12 \times 5 \times 52 \times 0.11 = £4,252.25$$

As the price difference between the two fans is only £2045.00 the difference would be recovered in 6 months.

So far we have focused our attention on the Fan Total Efficiency.

Although the Fan Static Efficiency is not a strictly correct term we can look at what these figures would be to illustrate what impact the fan velocity pressure has on the efficiency percentage value. To determine the Static Efficiency simply transpose the static pressure for the total pressure in the Efficiency formula as shown on the following page: -

LC140R4-A9/30

$$\text{Fan Static Efficiency, \%,} = \frac{40 \times 500}{10 \times 46.31} \quad 43.2\%$$ The Fan Total Efficiency is 82%

AP1806DA9/21

$$\text{Fan Static Efficiency, \%,} = \frac{40 \times 500}{10 \times 33.92} \quad 60.0\%$$ The Fan Total Efficiency is 77%

So the Static Efficiency of the LC180 is 28% higher than the LC140 yet its Total Efficiency is 6.4% less and that is simply because the velocity pressure in the LC140 fan is approximately 420Pa whereas it is only 155Pa with the LC180.

1.2 Sound Power and Sound Pressure

These are quite difficult terms to understand, but they will become clearer as we progress through the sessions.

1.2.1 Sound Power

The ‘sound energy’ created by a noise source is defined as its Sound Power. The shock wave from a sonic boom, or the thumping feeling of the bass drum at a rock concert, is caused by the sound energy behind them. This energy is quantified in Watts and is converted to differing sound pressure fluctuations depending on the environment. The ear cannot hear Sound Power and we cannot directly measure it.

However, Sound Power is important because it is independent of its surroundings. We can calculate the Sound Power of a machine, for example, and use it as a reference point as it will not change if we move the machine to another location.

Sound Power is expressed in decibels, dB.

1.2.2 Sound Pressure

Sound Pressure is defined as fluctuations in air pressure caused by sound waves. The ear converts these fluctuations into sound and, while the ear cannot hear Sound Power, it can hear Sound Pressure.

Sound Pressure is what is measured by a sound level meter. Sound Pressure is quantified in Pa and is dependent on Sound Power and also on its interaction with the environment.

To understand what this means imagine taking a radio at full volume into the smallest room in your house - the Sound Pressure would be unbearable. However, imagine the same radio in the centre of a football stadium - a person in the stands would barely hear it. Now the Sound Power has remained constant but the environment, and therefore the Sound Pressure, has altered dramatically. In the small room the sound has

reflected from the walls, ceiling and floor. In the stadium the listener is much further away from the source of the sound, the radio, and the sound is not reflected to any great extent.
Sound Pressure is expressed in decibels, dB.

So Sound Power is an energy value and Sound Pressure is what you hear. The further you are away from the source the lower the Sound Pressure. Both are expressed in dB and both are quoted in mid-octave band frequencies ranging from 63Hz through 125, 250, 500, 1000, 2000, 4000 to 8000Hz.

Hertz, Hz, is the modern unit for what used to be called ‘cycles/second’.

An Octave is a range of frequencies where the higher frequency is twice that of the lower.

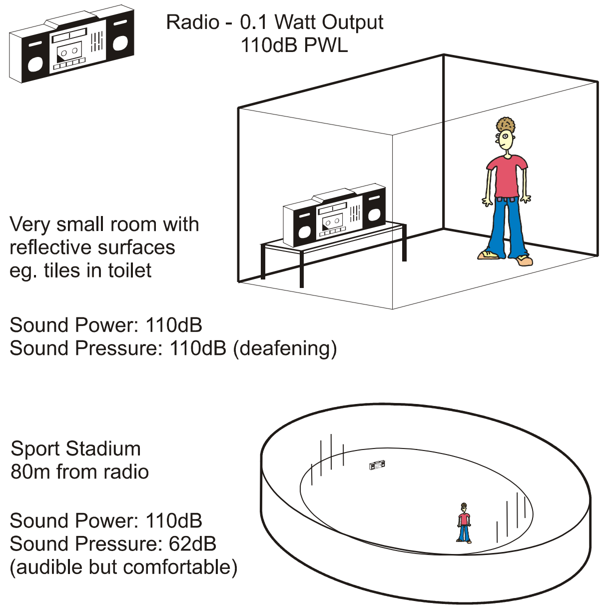


Fig. 1.10 – Illustration of Sound Pressure

An Octave Band is defined by its mid-octave frequency and is approximately 2 × the lower frequency value. These values have been rounded to 63, 125, 250, 500, 1000 (1k), 2000 (2k), 4000 (4k) and 8000 (8k) for ease of use. Obviously there are frequencies below 63Hz and above 8kHz but the range of 63 to 8K is sufficient for fan engineering. Indeed, most noise problems with fans are in the octave bands having mid-octave frequencies of 125 to 500 Hz.

2 FAN TYPES

A fan is a device that provides a continuous flow of air. They differ from air compressors, which usually give a pulsating flow of air.

There are numerous fan types, each of which has different qualities in terms of their ability to deliver air at low, medium or high pressure. This section is designed to explain the qualities (and faults) of commonly used fans.

The performance of a fan is determined by testing it on special test facilities. Generally tests are conducted at one speed and, using formulae called the Fan Laws, the performance at other speeds and atmospheric conditions can be predicted. In special circumstances the performance of a fan at different diameters can also be predicted accurately.

There is a wide range of fan types, many of which can do similar jobs. Some of these are listed below:

Fan Types		Cost	Volume	Pressure	Sound
Propeller Fans		Lowest	High	Lowest	Loud
Axial Flow Fans	single-stage	Low	High	Low	Loud
	multi-stage	Medium	High	Med/High	Loud
	guide-vane units	Medium	High	Medium	Loud
Centrifugal Fans Forward-curved	single-width	Medium	High	Medium	Medium
	double-width	Medium	High	Medium	Medium
Centrifugal Fans Backward-curved	single-width	High	Low	Med/High	Low
	double-width	High	Low	Med/High	Low
	laminar blade	High	Low	Med/High	Low
	aerofoil blade	Highest	Low	Med/High	Low
Mixed Flow Fans		Medium	Medium	Medium	Medium
Roof Ventilators	Axial Flow	Low	High	Low	Loud
	Centrifugal	Medium	Low	Med/High	Low
	Mixed Flow	High	Medium	Medium	Medium
In-Line Centrifugal Fans		Medium	Low	Med/High	Low
Tangential or Cross-flow		OEM Market			

There are other fan types such as Radial or Paddle Bladed, Mill Exhaust, High Pressure Blowers etc, which are all variations of Centrifugal Fans, but these are not covered here as we rarely come across a need for them.

2.1 Propeller Fans

Propeller Fans are excellent at moving large volumes of air relatively economically and at noise levels that are generally acceptable.

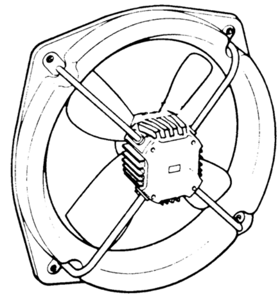


Fig. 2.1 - ‘Old’ style Propellor Fan

Fig. 2.2 - Square Plate Axial Fan

Traditionally they have an impeller with 3-6 fixed pressed metal blades mounted directly onto the motor shaft. This has begun to change in recent years and the modern product is capable of developing higher static pressures. The product shown in Fig. 2.1 is the older and traditional style of Propeller Fan; these were commonplace into the 1980’s and 1990’s but, with the advent of the ‘Square Plate Fan’, their use, although still substantial, is declining.

Fig. 2.2 shows the modern version, which is usually called a ‘Square Plate Fan’. This has the impeller running in a short tube with a bell-mouth inlet. The unit shown has pressure die-cast aluminium blades but they can also be plastic or even pressed metal. When die-cast, the blades have an airfoil cross-section, like aircraft wings and as used with axial flow fans. Propeller fans serve a wide range of applications where resistance to airflow is generally low. Examples of this would be for general ventilation work where they would be wall mounted and simply moving air from one space to another. Generally there would be very little ductwork, if any at all. In the modern configuration they are also frequently seen on air-cooled condensing units or cooling towers.

2.1.1 Propeller Fan Performance Curves

Propeller Fans, whether the old or new style, tend to have a fixed blade pitch angle for each diameter; with some sizes there may be a couple of pitch settings to choose from.

Fig. 2.3 illustrates the airflow performances available from a range of Propeller Fans. The absorbed power curves, for the product range shown, would have curves the same as that shown in Fig. 2.5 below for axial flow fans.

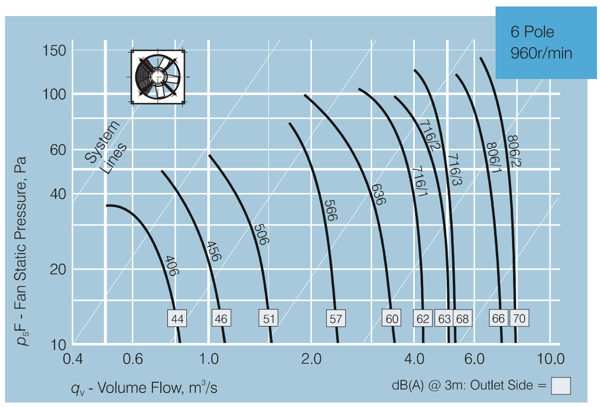


Fig. 2.3 - Propellor Fan Performance Curves

2.2 Axial Flow Fans

Axial Flow Fans have enjoyed a rapid increase in their acceptance in recent years and, in many instances, their designs have improved resulting in higher efficiencies and lower noise levels. In addition, they are very compact, simple to install and are generally very competitively priced when compared to the more traditional centrifugal fan. The products used in the HVAC market range in size from 100mm to 2000mm although there are occasions where larger diameters are required but these are rare.

The use of small fans is becoming increasingly common, particularly for the exhaust of air from small rooms such as laundries and bathrooms.

Axial Flow Fans generally have a non-overloading power characteristic.

The flow of air through an axial flow fan is straight through the impeller i.e. in an axial direction. The impeller of an axial flow fan is usually made up of a hub with a number of airfoil shaped blades fixed to it. The impeller is then mounted onto the motor shaft and, in turn, assembled into a steel cylinder. In most modern designs the angle of the impeller blade can be adjusted to vary the performance from the fan and, in addition, the number of blades that are fitted can also be varied. This means that by varying the number of blades and the blade pitch angle, any one fan size can be used to meet a wide range of performances.

Fig. 2.4 - Single-stage Axial Flow Fan

Fig. 2.4 shows a standard single-stage Axial Flow Fan and Fig. 2.5 shows typical performance curves for an axial fan with adjustable pitch blades. The lower set of curves illustrate the power absorbed by the fan across the range of pitch angles.

In the middle of Fig 2.5, there are 3 scales described as Type A, Type B and Type C; these are beside a diagram illustrating different fan installations. If the required installation is one of these, rather than the Type D shown on the airflow performance graph, corrections to the Fan Static Pressure have to be made. The value of the correction is obtained by coming vertically down from the airflow to the appropriate scale and reading off the value. This value is then added to the previously determined Fan Static Pressure and the fan can then be selected.

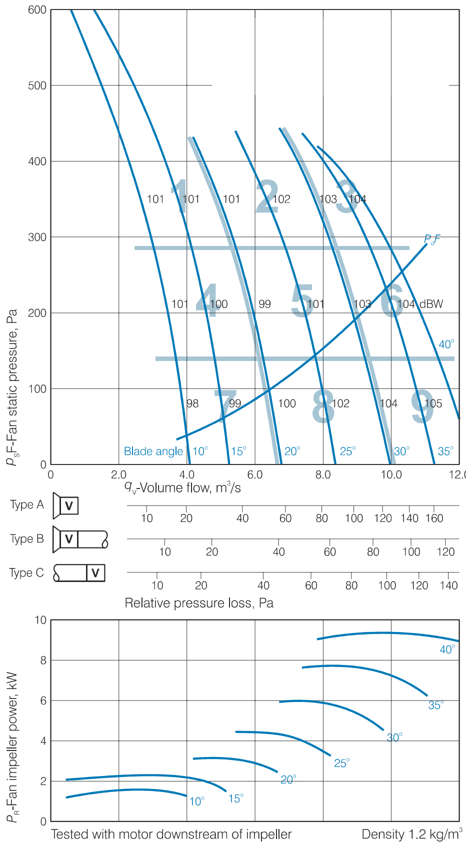


Fig. 2.5 - Axial Flow Fan Curves

Comprehensive information on how to select axial fans can be obtained via our website and catalogues.

2.2.1 Short-cased and Belt-Driven Axial Flow Fans

There are many variations in the designs of axial flow fans. Some, called Short-Cased Units, have a short casing that results in the motor protruding out of the casing.

Another variant is Belt-Driven Axial Flow Fans where the motor is mounted outside the fan casing. This arrangement is sometimes used when the gases being handled by the fan require that the motor be mounted outside to protect it.

Fig. 2.6 - Belt-driven Axial Flow Fan

A belt-driven arrangement also enables any fan speed to be selected, within the limits of the impeller, by varying the pulley dimensions, and hence the fan speed to suit. Also, adjusting the pulley diameters enables relatively simple capacity adjustments to be made by adjusting the fan speed. The performance of belt-driven fans is very similar to standard direct-driven axial fans although there is a loss in the pressure development capability of the fan, of the order of 5%, if a belt-guard is fitted within the housing, thereby restricting airflow.

2.2.2 Bifurcated Axial Fans

Bifurcated Axial Fans, as shown in Fig 2.7, are a variant of direct-driven axial flow fans, and frequently used as an alternative to belt-driven units. They are designed to handle toxic, noxious, abrasive and hot gases.

Fig. 2.7 Bifurcated Axial Flow Fan

To make this possible, the motor is mounted in a tunnel built into the fan casing; this keeps the motor out of the air stream and in a cooler environment. However, as the tunnel obstructs the airflow, the fans suffer a loss of efficiency.

If the fan is handling hot gases of the order of 200ºC continuously, special construction of the tunnel may be necessary as well as the possible addition of a cooling fan to blow cooler ambient air down the tunnel and over the motor to cool it.

In all installations, the fan should be mounted so that the tunnel is vertical; this will ensure a flow of air over the motor is always available by natural convention to assist in cooling the motor.

The performance curves of Bifurcated Axial Fans is very similar to standard axial flow fans although there is a loss in the pressure developing capability of the fan, of the order of 10-30%, caused by the tunnel in the housing restricting the airflow.

2.2.3 Multi-Stage Contra-Rotating Axial Flow Fans

Another variant to the axial fan is the Multi-Stage Unit, as shown in Fig. 2.8. By bolting single-stage axial flow fans together the pressure development capability of the assembly is increased dramatically.

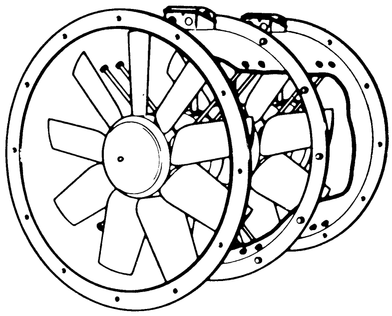


Fig. 2.8 – Contra-rotating Axial Flow Fans

A 2-stage assembly would be expected to develop at least 2.5 times the pressure of a single-stage unit. However, to do this, it is essential the impellers rotate in different directions, which means both left and right handed impellers are required. The pitch angles of the impeller blades should be set so that the power from each motor is the same; doing this will automatically ensure the output airflow is virtually free from swirl.

The addition of further stages is possible, but care should be taken to ensure the temperature of the air being handled doesn't get too high; the temperature rises from heat given off by the motors.

2.2.4 Axial Fans with Guide Vanes

Guide Vanes can be fitted either on the upstream or downstream side of the axial flow fan and effectively change the shape of the performance curve.

2.2.4.1 Upstream Guide Vanes

Upstream Guide Vanes are fitted to the air inlet side of the fan and cause the air to swirl in a manner that assists the fan by introducing the air to the fan blade at an optimum angle. This action enables the fan to generate substantially more pressure for a given airflow. To be effective, they have to be very close to the fan blade and this can result in a dramatic increase in noise levels. An increase in pressure development from 20 to 60% is achievable with a corresponding increase in power absorbed.

A well-designed unit should ensure the air leaves the impeller in an axial direction, i.e. there is no swirl.

Although the potential increase in pressure development is very attractive, the noise level increase generally means they are avoided. They can also be more expensive than going one fan size larger.

2.2.4.2 Downstream Guide Vanes

Downstream Guide Vanes, Fig 2.9, are more commonly used than Upstream Guide Vanes. They improve the pressure development by 10 to 25% with 20% being the usual average increase. There is no increase in the power absorbed by the fan as the guide vanes do their work after the impeller has done its, therefore there is a corresponding increase in fan efficiency. However, Downstream Guide Vanes, to be really effective, require a considerable swirl component imparted by the rotor to enable the guide vanes to do their work. As a result, guide vanes should not be used for pitch angle selections below 20º.

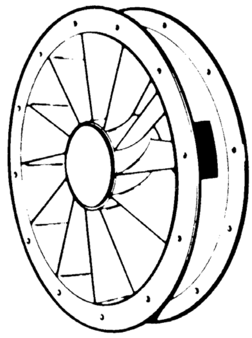


Fig. 2.9 – Downstream Guide Vane Unit

The noise level increase is generally only 1-2dB above the rotor only selection. In addition to improving the pressure development and efficiency of the fan, Downstream Guide Vane units remove the swirl component generated by the axial impeller on the air; this is particularly important when the fan is discharging into a long high velocity duct. Swirl tends to persist for a considerable distance in such situations thereby generating substantially higher pressure losses through the system. As with upstream guide vanes, a fan size larger can sometimes meet the duty at a lower cost.

2.2.5 Reversal of airflow

If the rotation of an axial flow fan impeller is reversed, the direction of the airflow will also be reversed. This will mean that the blade of the impeller will be operating tail first and have the camber, or airfoil shape, in the wrong direction. If this is done with a standard axial flow fan the airflow will be reduced by approximately 30% of the normal airflow.

In addition, the pressure development capability will be reduced by approximately 55% and the power absorbed by 25%.

A Truly Reversible impeller can be built by rotating every other blade through 180 degrees. Half the blades will then be running correctly and half in reverse. With this arrangement the airflow will be about 85% of normal in each direction.

Do not do this with guide vane units as the guide vane performance with the airflow in reverse is poor. A more effective truly reversible impeller is also available with a blade specifically designed for this purpose.

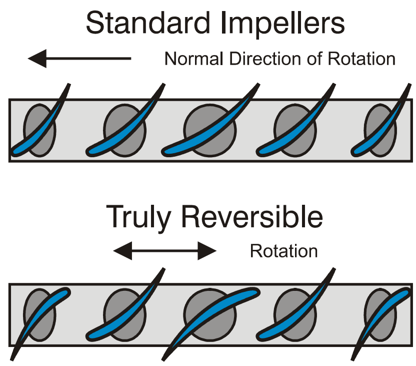


Fig. 2.10 - Standard and Truly Reversible Impellers

2.2.6 Fractional Solidity

The term Fractional Solidity means that the number of blades on an impeller can be varied to suit the needs of the application, as shown in Fig. 2.11.

With fewer blades the peak pressure development is reduced and the peak efficiency point moves to a lower pressure. The characteristics of the noise levels generally alter as well.

By fitting fewer blades there is a saving in cost and, on occasions, a motor with a lower kW output can be fitted as the lower solidity impeller may be a more efficient selection for the particular duty than an impeller with a greater number of blades.

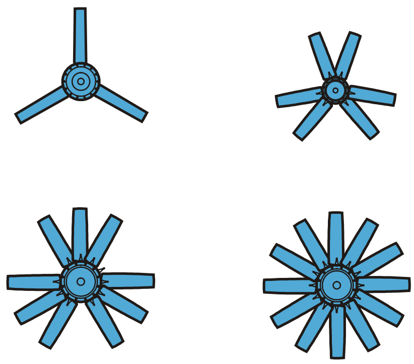


Fig. 2.11 – Variations in blade solidity

2.3 Centrifugal Fans

Forward-curved

- Single-width, single inlet (SWSI)
- Double-width, double inlet (DWDI)

Backward-curved

- Single-width, single inlet (SWSI)
- Double-width, double inlet (DWDI)
- Laminar blade
- Aerofoil blade

2.3.1 Forward-Curved Centrifugal Fans

Forward-Curved Centrifugal Fans are also called Multi-Vane Fans because of the large number of blades on the impeller, usually somewhere between 30 and 60. The impeller is cylindrical in shape and the blades are short, only 8-10% of the diameter. As the name suggests, they are curved in the direction the fan rotates, see Fig. 2.12. Diameters range from around 50mm to 1000mm although most would be in the small end of the range up to approximately 450mm diameter.

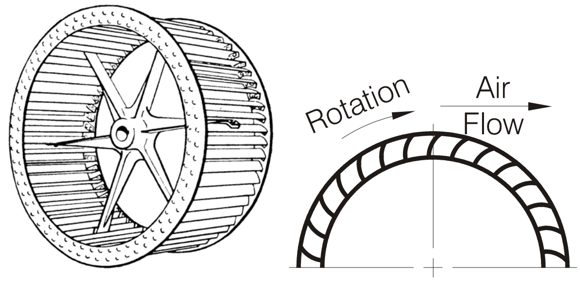


Fig. 2.12 Forward-curved impeller and side elevation

This fan type provides the most compact and competitively priced fan for a given duty; for one revolution of the impeller this fan type handles much more air than a fan of the same diameter but of a different type. However, their efficiency is limited to 60-70%. The absorbed power characteristic curve rises sharply towards free airflow; this may necessitate the fitting of a motor with a higher kW output than is necessary for the required duty.

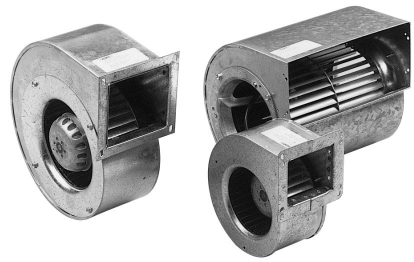


Fig. 2.13 - Single and double-width Forward-Curved Centrifugal Fans in volute shaped casing

This fan type is mounted in a volute or scroll shaped housing, Fig. 2.13, which is essential for the fan to perform. The air enters the housing at the side, turns 90º then goes through the impeller and out the fan discharge. Single-width fans have an inlet on one side of the housing. Double-width fans are generally made up of two single-width impellers back to back, or a long single impeller with a central support plate, and have an inlet on both sides of the housing.
Double-width fans generally deliver a little less than double the volume flow of the single-width unit at the same pressure.

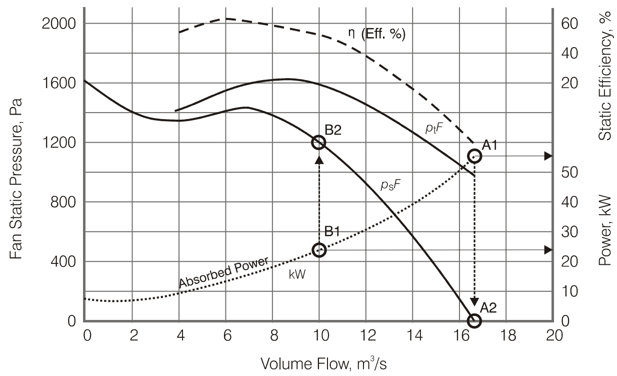


Fig. 2.14 - Typical Forward-Curved Centrifugal Fan Performance curv

The double-width Forward-Curved Fans are best suited to free inlet and ducted outlet installations. In smaller sizes, up to approximately 450mm diameter, most are direct-driven but larger units are generally belt-driven. The maximum diameter is around 1000mm.

Fig. 2.14 shows a typical performance curve for a Forward-Curved Centrifugal Fan and clearly illustrates how the absorbed power of the fan rises sharply as the pressure decreases.

In this particular example, a 55kW motor (Pt. A1) would be required if there was no static pressure (Pt.A2) or if the fitted motor had to cope with all possible performances. However, if the duty required is 10m³/s @ 1200Pa (Pt.B2), a 30kW motor (Pt.B1) would be sufficient.

2.3.2 Backward-Curved Centrifugal Fans

Backward-Curved Centrifugal Fans have three basic blade types: -

- Backward-inclined
- Backward-curved and
- Birfoil bladed

The term Backward-Curved Centrifugal Fans is the generic name for this product type.

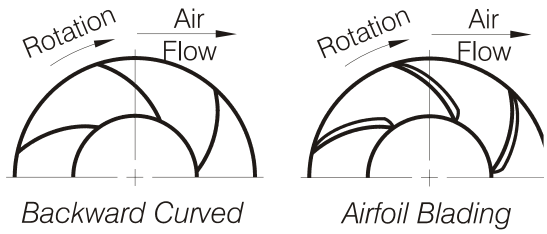


Fig. 2.15 – Side elevations of Backward-Curved and Airfoil Bladed Impellers

All of them have the outer trailing edges of the blades inclined backwards, Fig. 2.15, rather than forward as with the forward-curved centrifugals. They also have a much greater radial depth than the forward-curved impeller blade. Sizes range from 130mm to around 2500mm although larger units can be found in specialist applications such as mine exhaust.

These fans generally have a non-overloading power characteristic; see 1.18 and Fig. 2.18. The backward-inclined blades are flat and single thickness. The backward-curved blades are slightly curved and also single thickness. The airfoil blades have a cross-section similar to axial flow fans and are generally formed from hollow sheet metal.

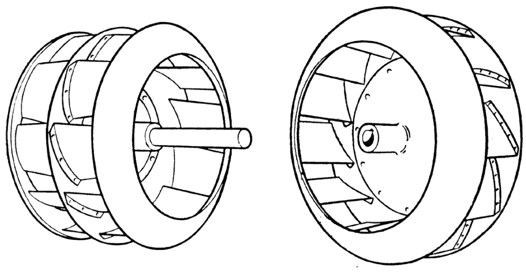


Fig. 2.16 – Single and Double Width Backward-Curved impellers

Because of the blade design usually only 6-12 blades are required and efficiencies as high as 80-85% are possible; the airfoil bladed units achieve efficiencies as high as 90%. Airfoil Backward-Curved Centrifugal Fans do not illustrate an improved efficiency when static pressures are relatively low, and generally speaking, they have to rotate slightly faster to achieve the same performance than the backward-inclined or backward-curved fans, although they will do this more efficiently and at about the same noise level.

Fig. 2.17 – Single and Double-Width Backward-Curved Centrifugal Fans

Mechanically, these fan types have impellers which are much more robust than the forwardcurved centrifugals enabling them to run at much higher tip speeds. Fan drives can be either direct or belt-drive, the latter being by far the more popular.

Housings are of the volute or scroll types, Fig. 2.17, and tend to be quite robust, although this aspect is changing in modern designs. In more recent times there have been developments of backward-curved impellers which perform satisfactorily without the conventional volute or scroll shaped fan casing. This development has enabled the production of many products, in particular Centrifugal Roof Ventilators and In-Line Centrifugal Fans, which are capable of developing higher pressures than is possible with axial flow fans. Also, the development of In-Line Centrifugal Fans has undergone some radical change resulting in very compact and low cost housings for products that perform well. These products are covered later.

Fig. 2.18 illustrates a typical performance curve for backward-curved centrifugal fans. This particular example is of a single-width backward-curved fan and illustrates the higher efficiency achievable than from a forward-curved centrifugal fan as well as the non-overloading power characteristics.

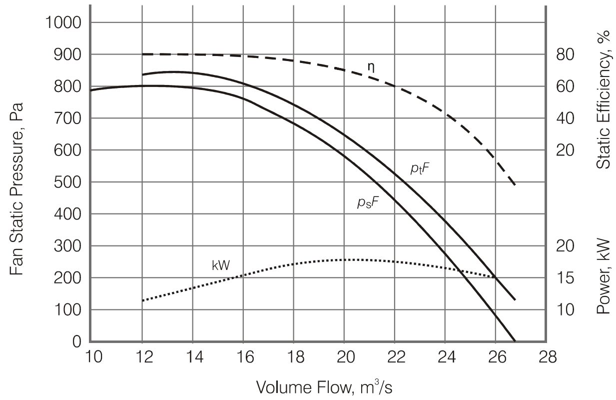


Fig. 2.18 - Backward-Curved Centrifugal Fan performance curves

2.3.2.1 Inlet-Guide Vanes

Inlet-Guide Vanes are frequently fitted to the inlet of centrifugal fans and can have either fixed or variable pitch vanes. They perform the same function as Axial Flow Fan Upstream Guide Vanes by inducing a swirl in the air, which increases the efficiency of the impeller.

The variable pitch guide vanes are used where the capacity required from the fan has to be varied to meet varying airflow requirements in an air-conditioned building or in a manufacturing process. As they are rotated they alter the flow rate, pressure development and the amount of energy required to drive the fan.

Inlet-Guide Vanes are often fitted to backward-curved fans to achieve lower energy consumption at reduced flow rates much more efficiently than simple dampers.

The inlet conditions for air entering a fan of any type is important to ensure the fan is able to perform as tested. The fitting of guide-vanes can assist in overcoming any adverse inlet swirl in the air, which could have been created by upstream disturbance and act as air straighteners.

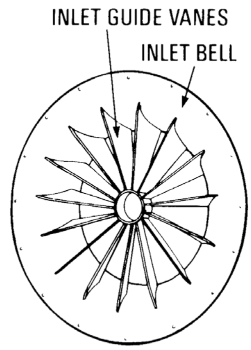


Fig. 2.19 – Inlet Guide Vanes for Centrifugal Fans

2.3.3 Other Centrifugal Fan Types

There are a number of other types of centrifugal fans but these tend to be for specialist applications in the industrial sector and usually for special processes. They are not covered here.

2.4 Mixed Flow Fans

Mixed Flow Fans take advantage of certain aspects of axial flow fans, particularly volume flow and axial direction of airflow, and from backward-curved centrifugal fans, particularly pressure development, and incorporating them into the one fan. The final product compares favourably to the axial flow fan for efficiency, noise and pressure development and to the centrifugal fan for airflow and compactness.

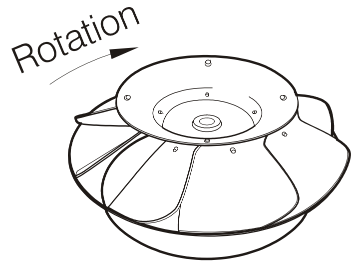


Fig. 2.20 – Mixed-Flow Impeller

2.5 Comparison of Fan Performances

The notes above describe each fan type but the graph below, Fig. 2.21, illustrates the relative performances from the four basic fan types each of the same size and speed.

This illustrates the different pressure development capabilities and airflow performances only.

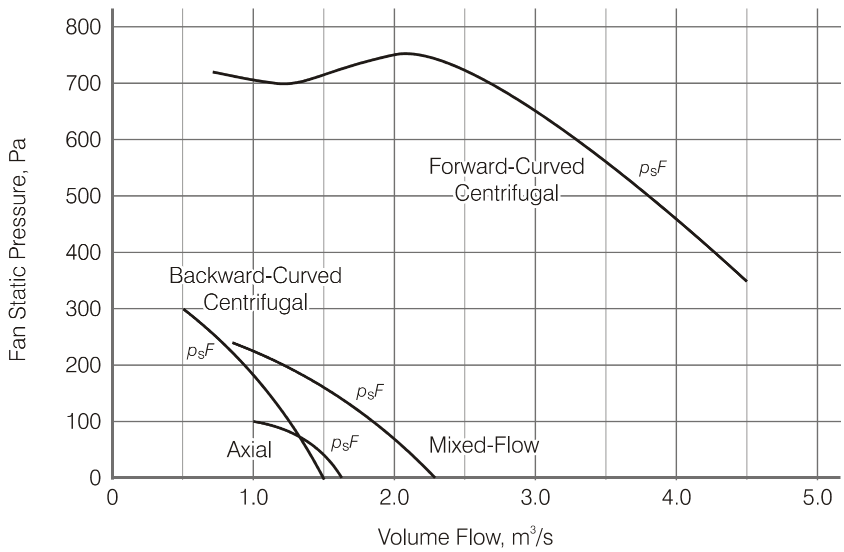


Fig. 2.21 – Fan Performance Comparison
Based on 500 diameter fans running at 6 pole speeds

2.6 Roof Ventilators

Roof Ventilators can be fitted with either axial flow, backward-curved centrifugal and, less commonly, mixed-flow impellers thereby providing a wide range of performances to suit a wide range of applications. There are also some Roof Ventilators which have forward-curved centrifugal fans fitted, but these are rare.

For most applications, the housings are made of moulded plastic materials such as injection moulded, rotationally moulded and vacuum formed plastic as well as fibreglass. These materials allow a greater degree of flexibility in design, as they are easily moulded. This makes it possible to design products for aesthetic appeal as well as for purpose. In addition, and principally for larger fans or smoke-spill applications, the housings are made of steel.
Roof Ventilators are designed for either vertical or downflow discharge, sometimes called side discharge, and are more frequently used for exhaust applications. However, there are also products designed specifically for supply air applications.

There are advantages and disadvantages in the use of Roof Ventilators. Amongst the advantages is that they do not take up space within the building and access to, and ease of maintenance, is generally good. A disadvantage can be ‘out-of-sight, out-of-mind’ and particularly so in buildings where there is a poorly managed program of maintenance.

2.6.1 Axial Roof Ventilators

Axial Roof Ventilators come in a wide range of sizes and styles, Fig. 2.22, with impeller diameters extending from 100mm to 1800mm. Impellers are generally of the fixed pitch type up to 800mm diameter, although adjustable pitch impellers may also be fitted from 310mm diameter.

Applications range from simple exhaust (or supply) to ones requiring specialist construction and motors built for hazardous applications such as handling toxic or inflammable fumes to smoke-spill, where high temperatures may be involved. Unit housings can be of all the materials detailed above.

Apart from sizes up to around 630mm diameter, most Axial Roof Ventilators are driven by conventional squirrel cage induction motors. For the smaller units the motors can be of the external rotor type. Motors are described in a later section.

2.6.2 Centrifugal Roof Ventilators

Centrifugal Roof Ventilators come in sizes ranging from 195mm to 710mm diameter. Impellers are generally of the backward-curved design. Applications are similar to the Axial Roof Ventilators although the Centrifugal Roof Ventilators develop much higher pressures for a given fan size and speed. All sizes are driven by external rotor motors, although standard squirrel cage induction motors may be used from size 310mm diameter.

These can also be used for smoke-spill applications.

2.6.3 Mixed-Flow Roof Ventilators

Not yet a product developed by Fantech but they are available from competitors; sizes range from 200mm to 760mm. The Mixed-Flow Impeller design makes it very suitable for incorporating into roof ventilators because of the direction the air leaves the impeller, Fig. 2.24. These are generally driven by standard squirrel cage induction motors.

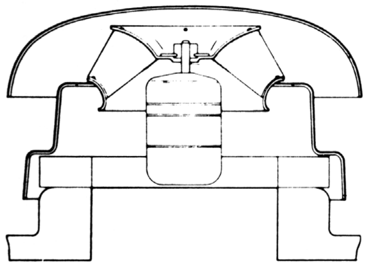


Fig. 2.24 – Mixed-Flow Roof Unit

2.7 In-Line Centrifugal Fans

This product type began as cylindrically cased units, like axial flow fans, but with mixed flow impellers and driven by standard squirrel cage induction motors. They tended to be over 450mm diameter ranging up to 1500mm and were very expensive. Recent developments of this fan utilize mainly backward-curved centrifugal impellers, although mixed-flow impellers are also used. At the small end of the range, i.e. with impellers from 130mm to around 310mm diameter,

Fig. 2.25 – Small Centrifugal and Mixed-Flow In-Line Fans

the products are often called 'Pipe Fans', Fig. 2.25. Their housings are usually of pressed steel or moulded plastic.

Fig. 2.26 – Centrifugal and Mixed-Flow Fans in In-Line steel housings

From around 310mm diameter the product is generally housed in a square sheet steel box, Fig. 2.26, although cylindrically cased units are also available. Impeller sizes range up to 710mm for the backward-curved centrifugal units and 800mm for mixed-flow. The fans are usually driven by external rotor motors, with standard motors being an option for size 310 upwards for the backwardcurved centrifugal fans. The mixed-flow impellers are usually driven by standard motors. There is a range of mixed-flow impellers driven by external rotor motors.

In-line fans provide excellent performances at very competitive prices.

2.8 Cross-flow Fans

Cross-Flow Fans, also known as Tangential Fans, look similar to forward-curved centrifugal fans although the ratio of impeller width to diameter is much greater, i.e. the impeller is like a long tube.

When the impeller rotates, a vortex is generated within it, causing air to enter through the back of the impeller and it then exits at the other side. Air doesn't enter at the sides, or ends, as with other centrifugal fans.

With a few exceptions, the pressure developed by this fan type is generally quite low, around
50-60 Pa.
This type of fan is usually purchased by Original Equipment Manufacturers (OEM's), and incorporated into their equipment such as fan heaters and air curtains.

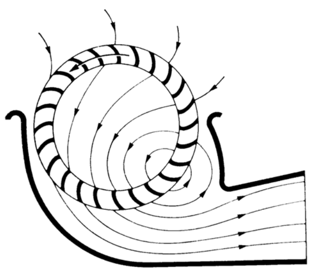


Fig.2.27 - Side elevation of Cross-Flow Fan

Fig.2.28 - Cross-Flow Fan

3 SELECTING FANS

How to Select a Fan

There is much advice about selecting fans but the most basic rule to go by is to ensure the fan is suitable for the application. To do that all the factors shown below may come into play so it is essential to have a broad understanding of them and how they may interact with each other.

The **Air/Pressure** performance is an essential requirement but, for a variety of reasons, the application usually necessitates other parameters being considered. The other parameters to consider are as follows (listed in alphabetical order):

- **Efficiency** – there is a strong move towards the selection of fans at their optimum efficiency. If using the selection program, don't just look at the percentage efficiency, look at the Absorbed Power column. Refer to 1.1.9 for an example of how to select for optimum efficiency.
- **EU Legislaition** – The EU legislation on Ecodesign and energy labelling is an effective tool for improving the energy efficiency of products, helping to eliminate the worst performing products from the market. It also supports industrial competitiveness and innovation by promoting better environmental performance of products throughout Europe.
  The Ecodesign Directive is implemented through product-specific regulations, directly applicable in all EU countries. A number of non-EU countries also have legislation similar to the Ecodesign and Energy Labelling Directives.
- **Electrical Supply** – what voltage, single or three-phase and what frequency.
- **Fan Speed**
- **Fan Type** – e.g. axial, centrifugal, in-line fan, roof unit etc.
- **Noise levels** – this could be very important because you are ventilating a TV Studio, an air inlet/outlet is very close to a neighbouring building or it is an environment where specific noise levels have to be met.
- **Operating Temperature** – could be smoke spill, high or low temperature application.
- **Size** – there may be space limitations.
- **Toxic, noxious or explosive fumes** – these may require special motors or impellers and casings of special materials.
- **Weight** – the weight of the fan(s) could impact on the building structure. This applies to roof ventilators in particular, but not exclusively.

3.1 Duty

The Duty is the beginning of the process of selecting a fan, or, for that matter, almost any other product.

In the case of a fan, the Duty is generally the air/pressure performance required to meet the ventilation requirements of the particular application involved. However, other considerations such as noise level, size etc may also have to be considered.

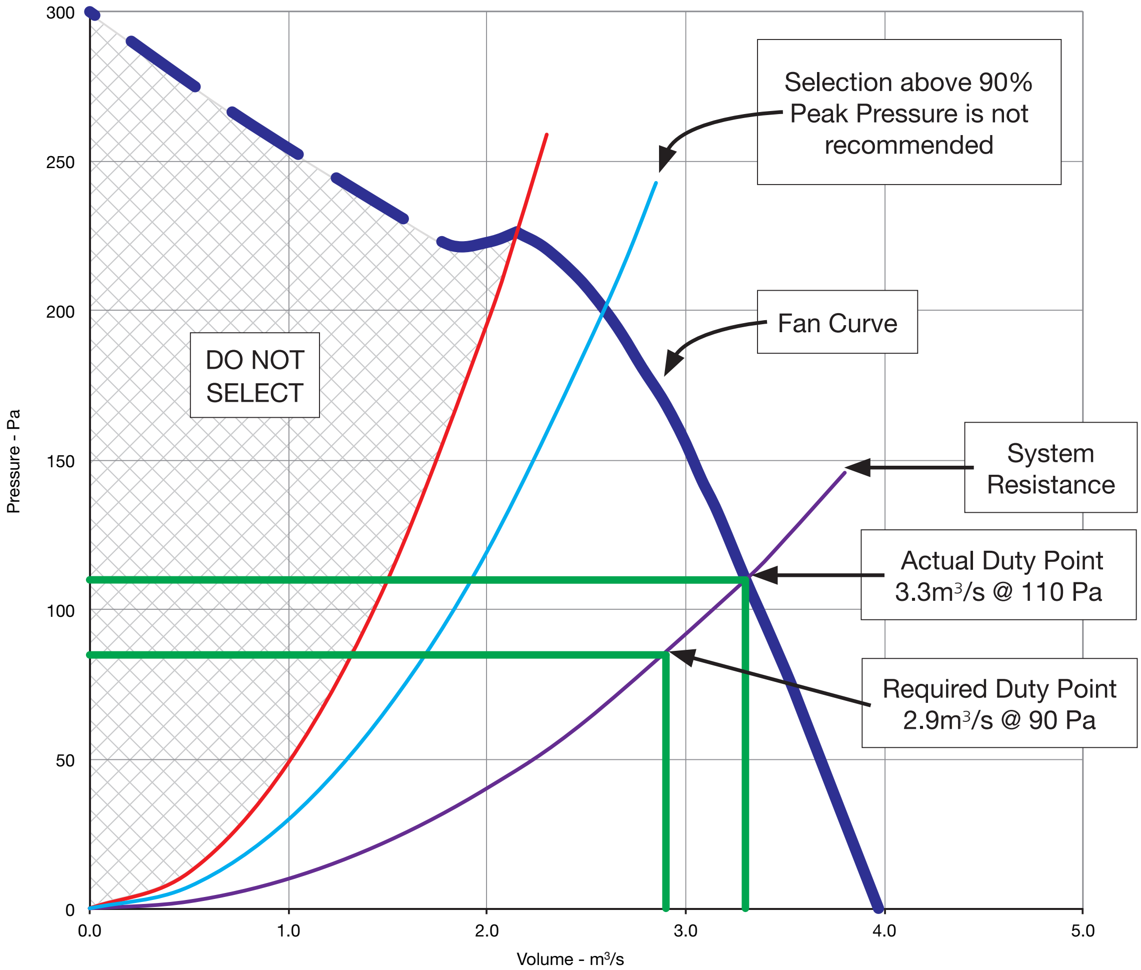
To determine the amount of air to be handled for any application a working knowledge of local regulations, in which the ventilation rates for many specific applications is defined, is essential. If you don't have this knowledge get in contact with someone who does.

The airflow may be defined as a minimum flow rate per person or a number of air changes/hour (ac/hr), although this latter term is less frequently used nowadays. Either figure would be dependent upon the

application and local regulations.
Air change rates can vary quite dramatically and range from 10L/s to 50L/s/person, depending on the application. Interestingly, the highest ventilation rate is for an Autopsy Room although it is assumed the high exhaust rate is not for the benefit of the deceased!
Generally speaking the design engineer determines the amount of air that has to be supplied or exhausted and he determines this from the application and how it is defined in the Standards.
Once the airflow is determined, the positions of the fan and exhaust/supply points can be finalised.
Positioning the fan is important as access for cleaning and general maintenance has to be considered – although it frequently isn't. The design engineer will then connect these points with ducting. The duct size will be determined using the velocity of the air as one parameter although other parameters, such as space availability, will have an impact as well.
Once the duct size has been fixed he can then calculate the pressure loss through the duct and associated fittings such as filters etc. and nominate an appropriate fan for the job.

3.1.1 Selecting Fans from a Performance Curve

The following graph shows the recommended method of how to select a fan using the performance curves provided within this Product Guide.



The Actual Duty Point is the intersection of the System Resistance and the Fan Performance Curve.

4 THE FAN LAWS

4.1 General Notes

The Fan Laws, which are referred to in Section 1, are more fully explained and detailed below. This is a 'long handed' explanation on their use.

The 'Fan Laws' take known data for a fan and determine the performance of the same fan or another fan under different conditions; as a result the Laws are basically a ratio of known data to the data at another condition.

The main Laws are centred on the airflow, pressure development and power absorbed figures and there are 4 elements, which have an impact on these values, these are:

- Fan rotational speed
- Impeller diameter
- Air temperature and
- Barometric pressure

The Fan Laws can only be used to their full extent under special circumstances. For example, to determine the performance of a fan at a different diameter the new fan must be an exact scale model of the original. Terms used for fans in this category are 'Geometrically Similar" or, of a 'Homologous Series'. However, if only the speed, air temperature or barometric pressure vary, the Laws can be applied fully and the 'diameter' element deleted from the equations.

In practice, the Fan Laws are sometimes used for diameter variations of 10-15% if the new diameter is simply a reduction or extension of the original impeller blade. Even such a minor change will introduce an error of a few percent but it is generally acceptable. An example of where this could occur would be with the Elta axial impellers. These are made up of a hub with blades attached and different diameters are achieved by simply trimming the blade length to suit requirements. Therefore, a new diameter would not be a scale model of the original. Nowadays, it is much more common to have a range of backward-curved centrifugal fans being geometrically similar than a range of axial flow fans, although the latter does exist.

4.2 The Fan Laws

The main Fan Laws are listed below but the following should be noted: -

- Formulae 4.2.1 to 4.2.4 can only be applied to fans, which are geometrically similar if there is a change in the diameter. It doesn't matter what units are used as long as the units for speed, diameter etc. are the same.
- In formulae 4.2.5 to 4.2.9 the units shown in the nomenclature must be used to satisfy the formulae.

It should be noted that any variable in the following formulae that is used as a ratio (e.g. $\left(\frac{n_2}{n_1}\right)$) but for which there is no change, can safely be ignored.

4.2.1 Volume Flow

Volume Flow, $q \propto n \times d^3$

The symbol $\propto$ means 'varies as'. So, in the case of the above, it means that the airflow from a fan varies directly with the speed, e.g. double the speed and you get twice the air. It also varies with the cube of the diameter, e.g. double the diameter and you get 2 x 2 x 2 = 8 times the amount of air.

To be able to apply this Law it is formed into a formula as follows: -

Air volume, $q_{v2} = q_{v1} \times \left(\frac{n_2}{n_1}\right) \times \left(\frac{d_2}{d_1}\right)^3$

4.2.2 Pressure

Pressure, $p \propto \varrho \times n^2 \times d^2$

In the form of a formula, Pressure, $p_2 = p_1 \times \left(\frac{\varrho_2}{\varrho_1}\right) \times \left(\frac{n_2}{n_1}\right)^2 \times \left(\frac{d_2}{d_1}\right)^2$

4.2.3 Absorbed Power

Absorbed Power, $P_R \propto \varrho \times n^3 \times d^5$

In the form of a formula, Absorbed Power, $P_{R2} = P_{R1} \times \left(\frac{\varrho_2}{\varrho_1}\right) \times \left(\frac{n_2}{n_1}\right)^3 \times \left(\frac{d_2}{d_1}\right)^5$

4.2.4 Sound Power Level

Sound Power Level, $PWL_2 = PWL_1 + 70\log_{10}\left(\frac{d_2}{d_1}\right) + 55\log_{10}\left(\frac{n_2}{n_1}\right) + 20\log_{10}\left(\frac{p_2}{p_1}\right)$

4.2.5 Air Density

Air Density, $\varrho \propto \frac{B}{T}$

In the form of a formula, Air Density, $\varrho_2 = \varrho_1 \times \left(\frac{B_2}{B_1}\right) \times \left(\frac{T_1}{T_2}\right)$

4.2.6 Fan Total Efficiency

Fan Total Efficiency, %, $= \frac{q_v \times p_t F}{10 \times P_R}$

or, Fan Static Efficiency, %, $= \frac{q_v \times p_s F}{10 \times P_R}$

4.2.7 Fan Total Pressure

Fan Total Pressure, $p_t F = p_s F + p_d F$

or, Fan Static Pressure, $p_s F = p_t F - p_d F$

4.2.8 Fan Velocity Pressure

Fan Velocity Pressure, pd $= 0.5\varrho V^2$, or
$= 0.6V^2$, (when handling Standard Air with density ϱ = 1.2 kg/m^3)

4.3 Nomenclature

q_v = volume flow of air, m3/sec
n = rotational speed of the fan
d = diameter of the fan
p = pressure developed by the fan
ϱ = density of air, kg/m3
P_R = power absorbed by the fan, kW
B = barometric pressure of the atmosphere
T = absolute temperature of the air, °K (°K = °C + 273)
$p_t F$ = fan total pressure, Pa
$p_s F$ = fan static pressure, Pa
$p_d F$ = fan dynamic/velocity pressure, Pa
V = velocity of air, m/sec
PWL = sound power level

4.4 Examples

In one of the examples shown below, the fan speed is increased by 10% and, in the other, the diameter of the fan is doubled. The impact of these changes on the airflow, pressure developed and power absorbed are then determined.

4.4.1 Volume Flow

10% increase in fan speed: -

Volume Flow, $q_{v2} = q_{v1} \times \left(\frac{n_2}{n_1}\right) \times \left(\frac{d_2}{d_1}\right)^3$

In the example, only the fan speed has changed. The diameter component of the formula has been removed, as it has no impact on the outcome.

Example: $q_{v2} = q_{v1} \times \left(\frac{n_2}{n_1}\right) = q_{v1} \times \left(\frac{1.1}{1.0}\right) = 1.1q_{v1}$

Double the fan diameter: -

Example: $q_{v2} = q_{v1} \times \left(\frac{d_2}{d_1}\right)^3 = q_{v1} \times \left(\frac{2}{1}\right)^3 = 8q_{v1}$

In this example, the fan speed components can be ignored as only the diameter has changed.

Now, if you don't understand the above you should get it explained to you as all the other Laws are similar and require the same level of mathematical understanding.

4.4.2 Pressure

10% increase in fan speed: -

Pressure, $p_2 = p_1 \times \left(\frac{\varrho_2}{\varrho_1}\right) \times \left(\frac{n_2}{n_1}\right)^2 \times \left(\frac{d_2}{d_1}\right)^2$

Example: $p_2 = p_1 \times \left(\frac{n_2}{n_1}\right)^2 = p_1 \times \left(\frac{1.1}{1.0}\right)^2 = 1.21p_1$

In this example, the elements of density, ϱ and diameter have been removed, as the only change applicable in this instance is the speed, n.

Double the fan diameter: -

Example: $p_2 = p_1 \times \left(\frac{d_2}{d_1}\right)^2 = p_1 \times \left(\frac{2}{1}\right)^2 = 4p_1$

In this example the only variable was the diameter, so the elements of density and speed have been removed from the equation.

4.4.3 Absorbed Power

10% increase in fan speed: -

Absorbed Power, $P_{R2} = P_{R1} \times \left(\frac{\varrho_2}{\varrho_1}\right) \times \left(\frac{n_2}{n_1}\right)^3 \times \left(\frac{d_2}{d_1}\right)^5$

Example: $P_{R2} = P_{R1} \times \left(\frac{n_2}{n_1}\right)^3 = P_{R1} \times \left(\frac{1.1}{1.0}\right)^3 = 1.33P_{R1}$

Again the elements of density and diameter have been removed.

Double the fan diameter: -

Example: $P_{R2} = P_{R1} \times \left(\frac{d_2}{d_1}\right)^5 = P_{R1} \times \left(\frac{2}{1}\right)^5 = 32P_{R1}$

In this example the only variable is the diameter, so the elements of density and speed have been removed from the equation.

4.5 Summary

The table below summarises the above calculations showing the impact on the airflow, pressure and absorbed power if, in one instance, the fan speed is increased by 10% and, in the other, the fan diameter is doubled.

	Speed +10%	**Diameter x 2**
Airflow	10%	8 times
Pressure	21%	4 times
Absorbed Power	33%	32 times

5 FAN PERFORMANCE AT VARIOUS TEMPERATURES

A fan will always operate at the intersection of the fan performance curve and the system curve. (See Section 1.1.7) The fan performance curve is catalogued at 'normal' or 'standard conditions' and these are defined as 20ºC and 1013mb barometric pressure; at these conditions the density of air is 1.2kg/m^3.

If the temperature and/or barometric pressure are different to 'standard conditions' then the density of the air will change and so will the fan characteristics. In addition the 'system pressure' or 'system curve' will change because of the change in density.

To illustrate this a couple of examples are shown below where the temperature has changed; for simplicity we have assumed the barometric pressure is the same in each instance.

In the example illustrated in Fig. 5.1, for a duty of 3.25m3/s @ 220Pa, the performance of a 630mm diameter axial flow fan with 10 blades, set at a pitch angle of 25º, absorbed power of 1.35 kW and running at 1440r/min, is plotted. This curve is based on tests conducted with 'standard air'.

Curves for the fan running in temperatures of –40ºC and 250ºC have then been computed using the Fan Laws and plotted on the same graph. From these curves of the fan running in 'non-standard' conditions we can determine the impact on the airflow, pressure and power absorbed.

For the sake of simplicity, and an understanding of Section 4, it is assumed you know how to work out the curves. So the figures shown below simply illustrate the application of the Fan Law formulae and how the results of these calculations are shown on the graph.

In a real situation, where you are asked for performance details of a fan running at a different temperature and/or barometric pressure, the impact can simply be calculated; there is no need to draw the fan curve.

5.1 The Fan Laws

1. The Airflow from a fan, $q \propto n \times d^3$

Where: q = air volume
n = fan rotational speed
d = fan diameter

As air density is not involved in this Law we can state the air volume will be the same irrespective of the air temperature or the barometric pressure. Also, as there is no change to the fan diameter or speed, we can ignore this Law completely.

$$p \propto \varrho \times n^2 \times d^2$$

2. The Pressure,

Where: p = fan pressure, total or static
ϱ = density of air, kg/m3
n = fan rotational speed
d = fan diameter

Density is involved here but we can delete the fan diameter and speed elements, as they are not changing.

2.1 So the Pressure, $p \propto \varrho$

3. The Power Absorbed, $P_R \propto \varrho \times n^3 \times d^5$

Where: P_R = power absorbed
ϱ = density of air, kg/m3
n = fan rotational speed
d = fan diameter

Density is also involved here but we can again delete the fan diameter and speed elements as they are constant.

3.1 So the Power Absorbed, $P_R \propto \varrho$

4. The density of air, at conditions other than Standard, $\varrho \propto \frac{B}{T}$

Where: B = barometric pressure
T = absolute temperature, ºKelvin = ºC + 273

To compare one density to the Standard density the formula becomes: -

4.1 $\varrho_2 = \varrho_1 \times \left(\frac{B_2}{B_1}\right) \times \left(\frac{T_1}{T_2}\right)$

As we are assuming the barometric remains constant in these examples we can delete that element from the equation, which now becomes: -

4.2 $\varrho_2 = \varrho_1 \times \left(\frac{T_1}{T_2}\right)$

Where: ϱ_2 = new density
ϱ_1 = old density, in this case 1.2kg/m^3
T_1 = original absolute temperature, in this case 20ºC + 273 = 293ºK (Kelvin)
T_2 = new absolute temperature.

You can see from the above that the Fan Laws have been simplified because we only have to consider changes to the air temperature in this example.

5.2 Applying the Formulae

Once we have determined the new relative densities at –40ºC and 250ºC we can apply that to the Pressure and Power Absorbed formulae and obtain the answers we are looking for.

At –40°C: -

$$\varrho_2 = \varrho_1 \times \left(\frac{20+273}{-40+273}\right) = \varrho_1 \times \left(\frac{293}{233}\right) = \varrho_1 \times 1.258$$

At 250°C: -

$$\varrho_2 = \varrho_1 \times \left(\frac{20+273}{250+273}\right) = \varrho_1 \times \left(\frac{293}{523}\right) = \varrho_1 \times 0.56$$

Applying the relative density values for –40°C of 1.258 ρ1 and 0.56 ρ1 at 250°C to Laws 2.1 and 3.1 as follows: -

At – 40°C: -

Pressure, $p \propto \varrho$ only in this example:

$$p_2 = p_1 \times \frac{\varrho_2}{\varrho_1} = 220 \times \frac{1.258}{1.0} = 277 \text{ Pa}$$

Power Absorbed, only in this example:

$$P_{R2} = P_{R2} \times \frac{\varrho_2}{\varrho_1} = 1.35 \times \frac{1.258}{1.0} = 1.7 \text{ kW}$$

At 250°C: -

Pressure, $p \propto \varrho$ only in this example:

$$p_2 = p_1 \times \frac{\varrho_2}{\varrho_1} = 220 \times \frac{0.56}{1.0} = 123.2 \text{ Pa}$$

Power Absorbed, only in this example:

$$P_{R2} = P_{R2} \times \frac{\varrho_2}{\varrho_1} = 1.35 \times \frac{0.56}{1.0} = 0.756 \text{ kW}$$

5.3 Summary of Results

Temperature	Airflow	System Pressure	Absorbed Power
-40°C	3.25m³/s	277Pa	1.7kW
20°C (Std. Air)	3.25m³/s	220Pa	1.35kW
250°C	3.25m³/s	123.2Pa	0.756kW

These results are now plotted onto the fan curves as shown in Fig. 5.1

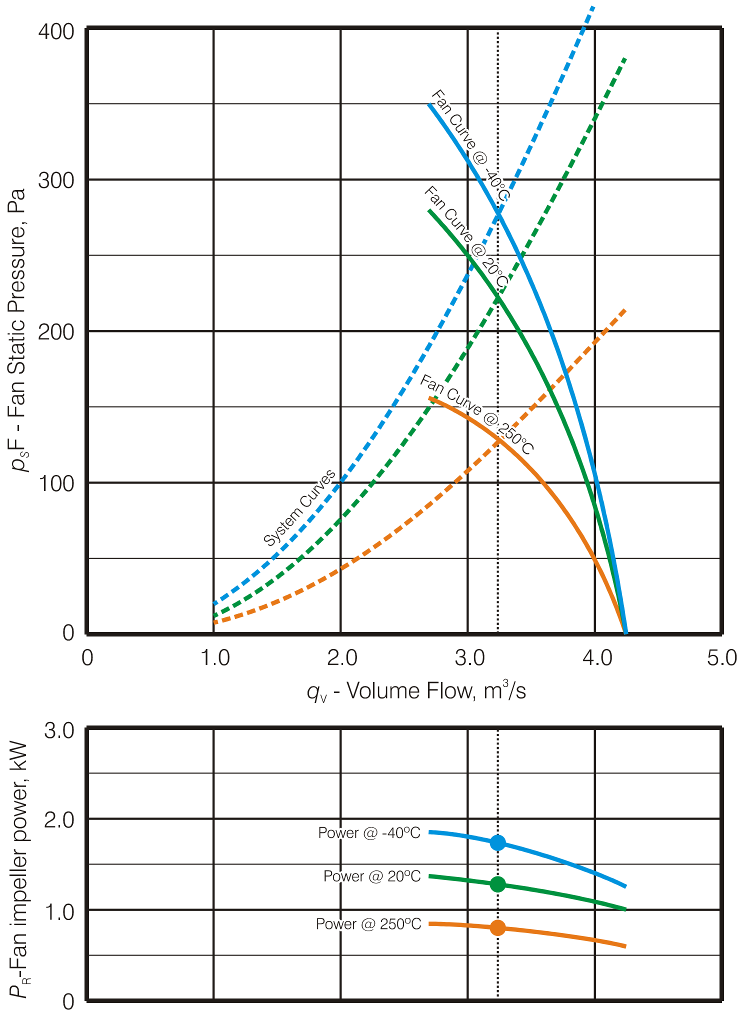


Fig. 5.1 – Summary of Results

To fully comprehend the above, working out the figures and drawing the curves yourself would assist.

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