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Subject: Turbomachinary

Unit: Fourth

Topic: Centrifugal and axial flow machines, Vane shape and stresses, Velocity triangle, degree of reaction, Size and speed of machines, efficiency, Fans law and characteristics.

Lecture 2

As we have studied in previous class, the following observations may be noted from figure 4.5.

(i) $V_{w2} < U_2$, if $\beta_2 < 90^0$, backward swept blades

(ii) $V_{w2} = U_2$, if $\beta_2 = 90^0$, radial blades

(iii) $V_{w2} > U_2$, if $\beta_2 > 90^0$, forward swept blades

The velocity triangles at entries and exits for three types of impellers are shown in fig 4.5. In all three cases for inlet diagram, it is assumed that the absolute velocity at inlet is axial, which means velocity of whirl at inlet, V_{w1} is zero, and absolute velocity at inlet is equal to the velocity of flow, i.e. $V_1=V_{f1}$. The axial portion of the vanes must be curved so that the air can pass smoothly into the eye. During the passage in the impeller, the flow of fluid is turned through some angle. The angle turned depends upon the shape of the impellers. The distance from the centre of rotation is much greater at outlet than at inlet, and therefore different blade velocities u_1 and u_2 must be used at inlet and outlet.

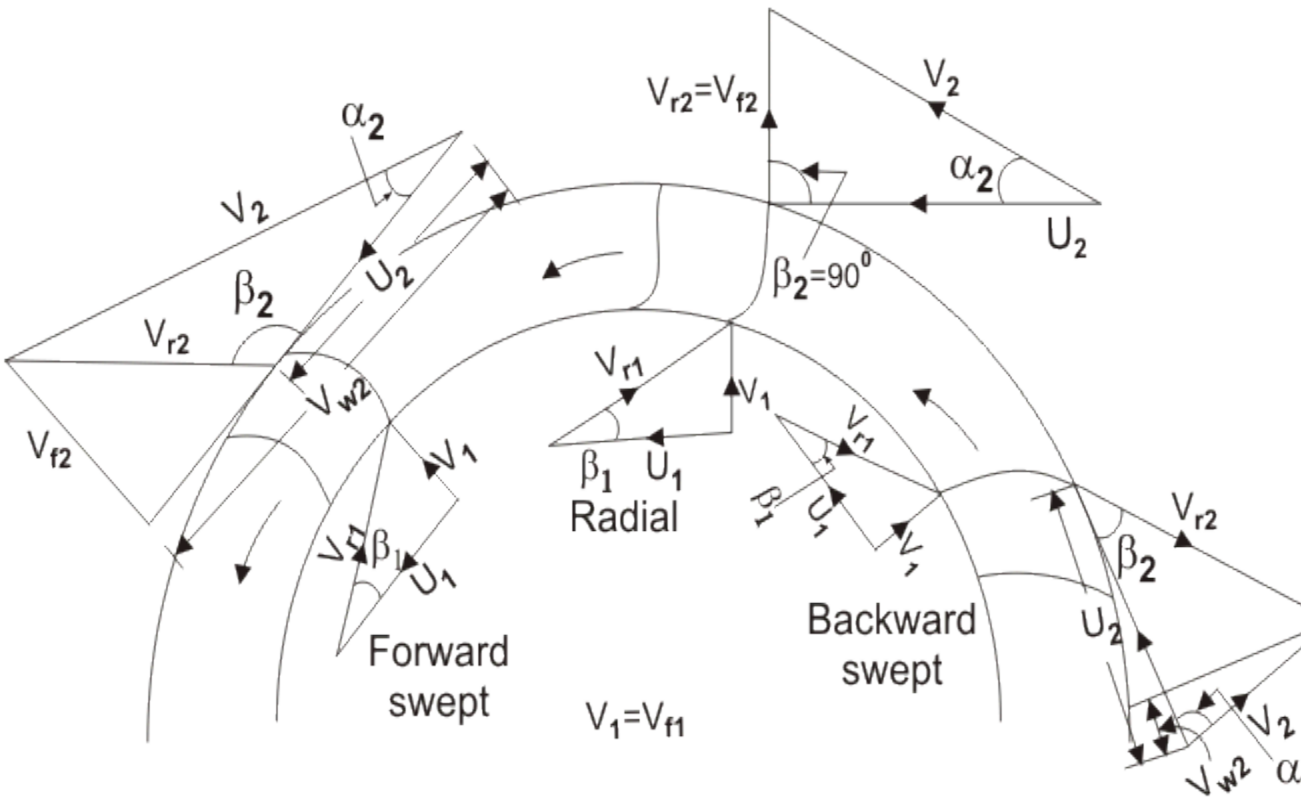

Figure 4.5 Velocity triangles at inlet and outlet of different types of blades of an impeller of a centrifugal blower

Since $V_{w1}=0$, therefore, energy transfer or work is given by Euler's equation, i.e.

$$W=mV_{w2}u_2$$

Energy or work transfer per unit mass is called as the change of head of fluid and is designated by H. For unit mass flow,

$$H=V_{w2}\,u_2 = u_2\,(u_2 - V_{f2}\cot\beta_2) \quad \ldots\ldots..(i)$$

As we know,

$$\text{Volume rate flow} = \text{Flow area} \times \text{Flow Velocity}$$

$$Q = A_2\,V_{f2}$$

Or $$V_{f2} = Q/A_2$$

Substituting values of V_{f2} in equation (I), we get

$$H = u_2\left(u_2 - \frac{Q\cot\beta_2}{A_2}\right) \quad \ldots(ii)$$

For a given machine running at fixed,u_2,β_2 and A_2, are all fixed.

Thus, equation (ii) can be written as

$$H= C_1 - C_2Q \quad \ldots(iii)$$

Where $C_1=u_2^2$

And $C_2 = u_2/A_2\cot\beta_2$

From equation (iii), it can be inferred for three types of blades-

(i) For facing vanes (with $\beta_2 > 90°$), Euler head goes on increasing with an increase in flow rate.

(ii) For radial vanes (with $\beta_2=90°$), Euler head remains constant with variation in mass flow rate.

(iii) For backward facing vanes (with $\beta_2<90°$), Euler head decreases with an increase in flow rate.

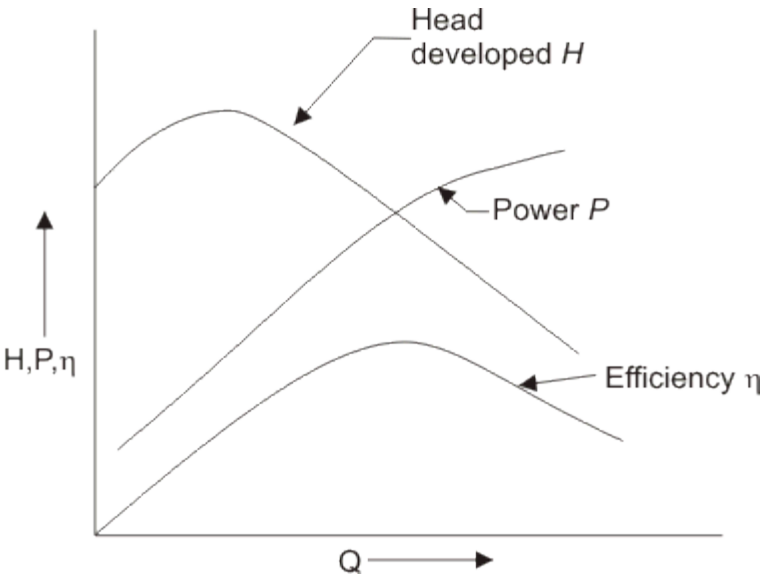

Figure 4.6 Performance characteristic curves of a centrifugal blower or fan

The advantages of the different types of vane shapes used in centrifugal flow machines.

- Forward facing vanes have higher pressure ratios, but lower efficiency owing to large slip factor. Further, operating range is closer to the surge line even under normal running conditions thus narrowing the stable operating range.
- Radial vanes have higher efficiency as compared to forward facing vanes. The vanes of slip factor is 0.9 for radial vanes, thus they give a large pressure rise in small space. They are usually preferred because of following advantages-
    1. Ease of manufacturing.
    2. Lowest unit blades stress for a given diameter and rotational speed.
    3. Equal energy conversion in impeller and diffuser, which gives higher pressure ratios with good efficiency.
- Backward facing vanes gives better efficiencies than with radial vanes, but the pressure ratio is lower. Therefore, the backward facing vanes are used where a high compressor efficiency is desired.

Vanes shapes affect the size and speed of a turbomachine

The size of a turbomachine determines its power output. The speed of the machine must be adjusted so that a minimum of gearing is required to drive it.

For a turbomachine, employing an incompressible fluid, an increase in β_2 increases the exit whirl velocity V_{w2} if the mass flow and V_{f2} are maintained constant.

$$\text{Torque} = (V_{w2}r_2 - V_{W1}r_1) \text{ N-m}$$

For the same inlet condition, torque must increase as β_2 Increases if the machines speed is to remain constant. If fans and blowers are to produce a certain pressure rise in the air, one must design units that absorb specified amount of power.

For a machine with a specified speed, absorbing a specified amount of energy, the diameter and V_{w2} can be varied to maintain u_2V_{w2} at the required value. Hence if β_2 is large, V_{w2} is also large, and u_2 has to be small. Similarly, if β_2 is small (as in backward curved machines), the diameter should be enlarged to give the same output. For these reasons, machines with backward curved blades are larger in size than those with radial and forward curved blades of the same capacity. A suitable change in vane configuration helps in change either the size or speed of the machine as required.

Relation between vane shape and stresses in a centrifugal flow machine

- Rotor vanes of a centrifugal flow machine subjected to stresses due to pressure, viscous, shear and other fluid forces acting upon them. A non-radial blade is subjected, to bending stresses, and these add to the existing levels. Machines like compressors producing high pressure differences are usually more stress limited in their capacity than fans and blowers that causes only a small pressure rise.
- Tip speed also affects the stresses induced in the vanes. A small diameter is helpful in producing a high rotational speed without unduly increasing radial stress.
- Compressors handling high pressure differences and running at low speeds usually have radial vanes. A lower rotational speed considerably reduces the operating stress levels and ensures a long compressor life.

Consider the flow of incompressible fluids through machines running at the same speed u_2 and with the same mass flow rate, V_{f2} = constant. Let V_{w1} =0 in the machines to be compared. As shown in fig. 4.7, with increase in β_2, V_2 increases.

For example, consider V_{f2} = 60 m/s while u_2 = 120 m/s, then for the different values of β_2, the value of V_2 will be different.

(i) If $\beta_2 = 45°$, then

$$V_2^2 = V_{w2}^2 + V_{f2}^2$$
$$= (120 - 60)^2 + 60^2 = 7200 \text{ m}^2/\text{s}^2$$

(ii) If $\beta_2 = 90°$, then

$$V_2^2 = u^2 + V_{f2}^2 = (120)^2 + 60^2 = 18000 \text{ m}^2/\text{s}^2$$

(iii) If $\beta_2 = 135°$ then

$$V_2^2 = (120 + 60)^2 + 60^2 = 36000 \text{ m}^2/\text{s}^2$$

Thus, the exits kinetic energy, $\frac{V_2^2}{2g}$, increase quite rapidly as β_2 increases. Consequently, an extremely efficient diffuser is needed to obtain a pressure rise utilizing all the kinetic energy at the exit. Since in most of the cases, the diffuser efficiencies are low because of adverse pressure gradients and the resulting thick boundary layers, a complete utilization of kinetic energy with a corresponding pressure rise is impossible. It is therefore, to be expected that machines with large exit angle, will be less efficient than the machine with the small exit blade angle.

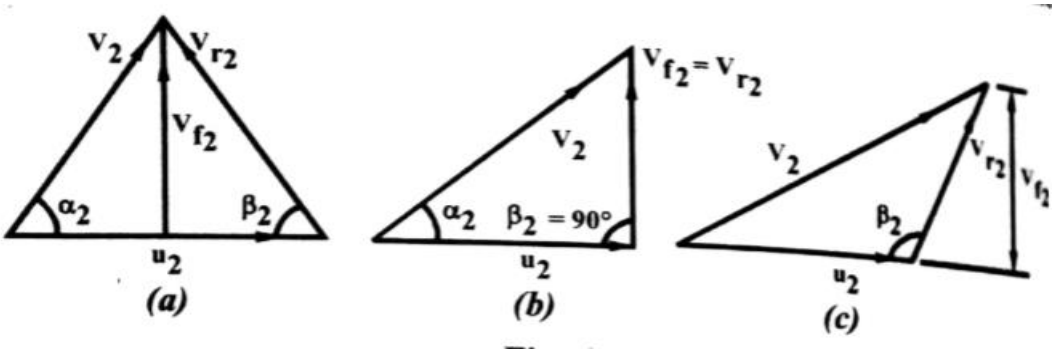

Fig.4.7

When a high blower efficiency is desired, machines with backward curved vanes must be used. In some cases, where a large pressure rise is needed with a machine of small size, radial blades are used though the efficiency may not be high. But in rare case, the machines with forward curved blades are used.

Characteristics of Vane shape

Head developed by a turbomachine is given by

$$H = C_1 - C_2Q \qquad ...(i)$$

Where $C_1 = u_2^2$

$$C_2 = \frac{u_2}{A_2} cot\beta_2$$

The constant C_2 determines whether the slope of the H-Q curve is positive, Zero or negative. The only way C_2 can be varied, is by constructing vanes of different angles β_2 when u_2 and A_2 are maintained constant. If β_2 lies between 0° to 90° (backward curved vane), cot β_2 is always positive and for the value of β_2 between 90° and 180°, cot β_2 is negative. For β_2 =90° or radial vanes $H = U_2^2$ = constant, i.e. head is constant at all flow rates.

The rising characteristic of forward curved blades, the falling characteristic of backward curved blades and the neutral characteristic for radial blades are shown in fig. 4.8

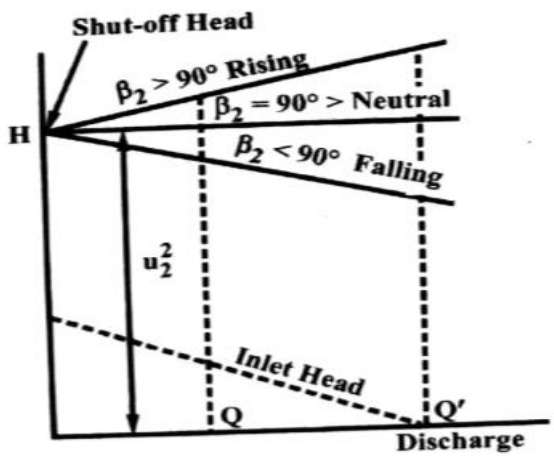

Fig.4.8 Theoretical Head-discharge Characteristic for Incompressible flow Turbo Machines

For a machine with a constant tip speed, the diagram show that an increase of Q decreases V_{w_2} since V_{f_2} is raised. Thus Δh_o is decreased when the flow rate increases. The characteristic is therefore falling. Similarly, for a forward curved vane, an increase of Q rise Δh_o and the characteristic will then be risisng. For a radial vane, an increase of Q makes no difference to Δh_o and the characteristic is therefore neutral.

If an initial whirl speed exist at the blower inlet so that so u_1 V_{w_1} is not zero, then

$$H = u_2\left(u_2 - \frac{Q\cot\beta_2}{A_2}\right) - u_1\left(u_1 - \frac{Q\cot\beta_1}{A_1}\right) \qquad ...(ii)$$

In equation (i) if Q = 0, $H = C^1 = U_2^2$. This head which is independent of the vane shape is called the shut-off head. The actual measured head at shut-off is much less than u_2^2 due to the high turbulence losses that occur when there is no flow.

Fan laws and characteristic curves of fans

If the fluid flowing through the fan or blower is considered to be incompressible, it is possible to show that certain simple relations exist among quantities like speed, mass flow rate and power. These relations are called the fans laws.

For any series of geometrically similar fans and for any point on the characteristic curve, the following relationships hold-

$$\frac{Q}{ND^3} = constant, \ \frac{g}{n^2D^2} = constant \ and \ \frac{P_t}{N^2D^2\,\rho} = Constant.$$

Where Q = Discharge in m^3/sec

P_t = fan total pressure

N = Rotational speed

g = Acceleration due to gravity

H = Fan total head

P = Power consumed

D = Impeller diameter

ρ = Fluid density.

In any consistent system of units, the above dimensionless relationships hold good and the co-relations are known as fan laws. Thus

(i) Q α N_3 (ii) Q α D3(iii) p_t α **ρ**N^2 (IV) power P α **ρ**N^3.

An axial flow fan has a casing which is cylindrical and the air enters substantially parallel to the axis of the cylinder and also leaves parallel to the axis. For centrifugal fan, the direction of air leaving from the impeller is at right angle to the axis.

Fan power- It is given by

$$P = 9.81\ Q_1 P_t K_p \text{ Watts}$$

Where Q_1 = Discharge in mm of water

P_t = Pressure in mm of water

K_p = Compressibility co-efficient.

Fan Efficiency – It is given by

$$\eta_{fan} = \frac{Air\ power}{Shaft\ Power}$$

The characteristic curves for fans are plotted in fig. 4.8. The fan characteristics are curves of stagnation pressure and static pressure rise, power input and fan efficiency, plotted against volume flow as a percentage of maximum flow rates, the stagnation pressure plotted is the sum of the static pressure change and the quantity, $\frac{\rho V_2^2}{2g_c}$, $\boldsymbol{\rho}$ being the gas density and V_2 the gas velocity at the fan exit.

In conclusion, the forward curved fans have large volume discharge and pressure rise but they demand higher power. However, forward curved fans are unstable for off-design operating conditions.

Backward curved fans are very efficient and the drooping power characteristic makes them suitable for a better off-design performance

(i) Radial curved fans are preferred for dust-laden fluids. Due to their shape, the solid particles are not stuck and deposited on the blade surface.

(ii) The relationships of discharge Q , head H and Power P with the diameter D and rotational speed N of a centrifugal fan can easily be expressed from the dimensionless performance parameters determined from the principle of similarity of rotodynamic machines as described before . These relationships are known as Fan Laws described as follows

$$Q = K_q D^3 N \qquad (40.1)$$

$$H = \frac{K_h D^2 N^2 \rho}{g} \qquad (40.2)$$

$$P = \frac{K_p D^5 N^3 \rho}{g} \qquad (40.3)$$

where $K_q, K_h,$ and K_p are constants.

For the same fan, the dimensions get fixed and the laws are

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$$

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2 \quad \text{and} \quad \frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3$$

For the different size and other conditions remaining same, the laws are

$$\frac{Q_1}{Q_2} = \left(\frac{D_1}{D_2}\right)^3, \quad \frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2 \quad \text{and} \quad \frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^5 \qquad (40.4)$$

These relationships are known as the Fan-laws. The Fan-laws can be summarized as

For the same fan:

Discharge $\propto$ Speed

Head developed $\propto$ (Speed)^2

Power $\propto$ (Speed)^3

For the fans of different sizes:

Discharge $\propto$ (Diameter)^3

Head developed $\propto$ (Diameter)^2

Power $\propto$ (Diameter)^5

The performance characteristics of fans having backward curved, radial and forward curved vanes are demonstrated as follows-

(i) A fan with backward curved vanes has to run at a higher speed than the other two types of fans, but its efficiency is much higher if properly designed. The maximum efficiency of a fan with backward curved vanes can reach 90% while a radial vanes fan reaches 80% and a fan with forward vanes about 70%.

(ii) For the same capacity and pressure rise, a fan with backward curved vanes consumes less power at maximum efficiency.

(iii) Forward curved fans $(\beta_2 > 90^0)$ develop the highest pressure for a given impeller diameter and speed.

(iv) Power requirement of a forward curved fan increases steeply for a small change in flow rate.

(v) Pressure developed decreases fast with increasing flow rate in a backward curved fan.

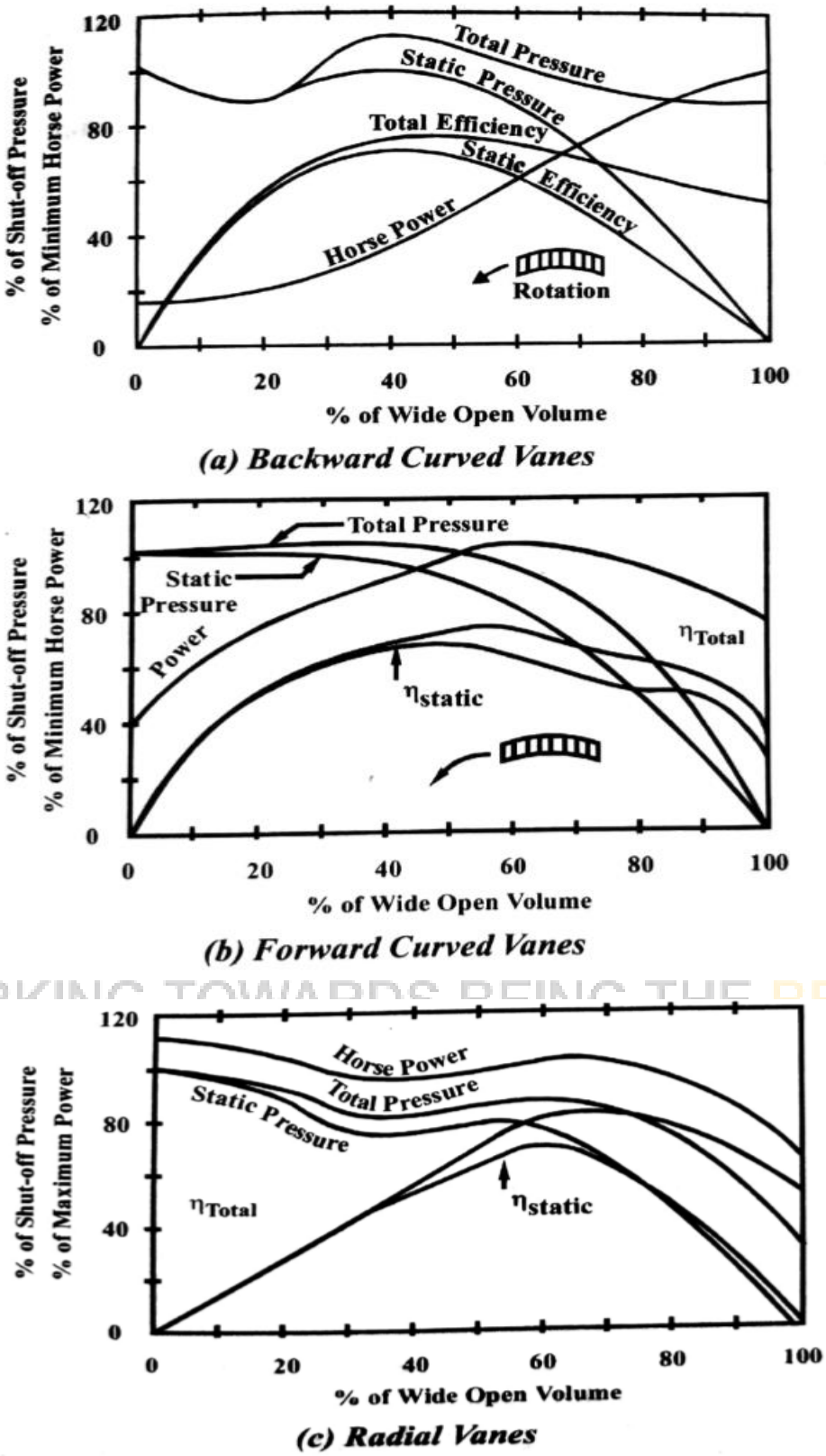

Fig- 4.9 Performance characteristics of Fans