Axial Flow or Propeller Pump

The axial flow or propeller pump is the converse of axial flow turbine and is very similar to it an appearance. The impeller consists of a central boss with a number of blades mounted on it. The impeller rotates within a cylindrical casing with fine clearance between the blade tips and the casing walls. Fluid particles, in course of their flow through the pump, do not change their radial locations. The inlet guide vanes are provided to properly direct the fluid to the rotor. The outlet guide vanes are provided to eliminate the whirling component of velocity at discharge. The usual number of impeller blades lies between 2 and 8, with a hub diameter to impeller diameter ratio of 0.3 to 0.6.

The Figure 37.1 shows an axial flow pump. The flow is the same at inlet and outlet. an axial flow pumps develops low head but have high capacity. the maximum head for such pump is of the order of 20m.The section through the blade at X-X (Figure 37.1) is shown with inlet and outlet velocity triangles in Figure 37.2.

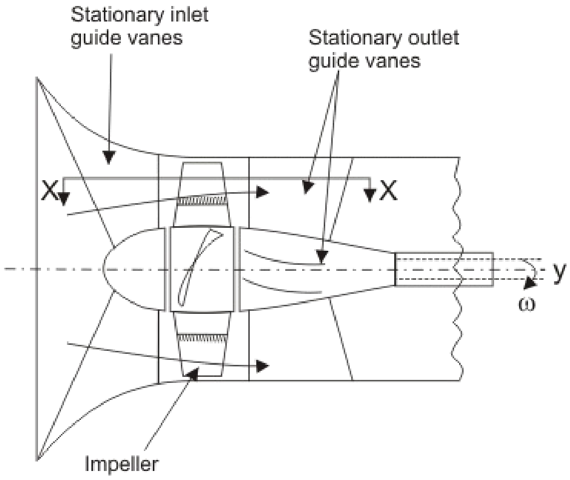

Figure 37.1 A propeller of an axial flow pump

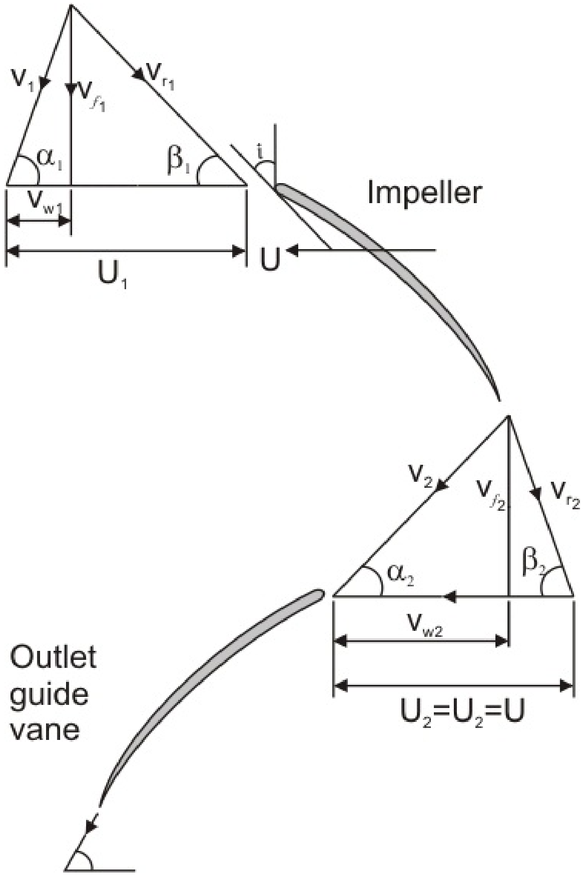

Figure 37.2 Velocity triangles of an axial flow pump

Analysis

The blade has an aerofoil section. The fluid does not impinge tangentially to the blade at inlet, rather the blade is inclined at an angle of incidence
(i) to the relative velocity at the inlet V_1 . If we consider the conditions at a mean radius r_m then

$$u_2 = u_1 = u = wr_m$$

where w is the angular velocity of the impeller.

Work done on the fluid per unit weight = $u(V_{w2} - V_{w1})/g$

For maximum energy transfer , $V_{w1} = 0$, i.e $\alpha_1 = 90^0$ .Again , from the outlet velocity triangle,

$$V_{w2} = u - V_{f2} \cot \beta_2$$

Assuming a constant flow from inlet to outlet

$$V_{f1} = V_{f2} = V_f$$

Then, we can write

Maximum energy transfer to the fluid per unit weight

$$= u(u - V_f \cot \beta_2)/g \qquad (37.1)$$

For constant energy transfer over the entire span of the blade from hub to tip , the right hand side of Equation (37.1) has to be same for all values of r . It is obvious that u^2 increases with radius r , therefore an equal increase in $uV_f \cot \beta_2$ must take place, and since V_f is constant then $\cot \beta_2$ must increase. Therefore , the blade must be twisted as the radius changes.

Matching of Pump and System Characteristics

The design point of a hydraulic pump corresponds to a situation where the overall efficiency of operation is maximum. However the exact operating point of a pump, in practice, is determined from the matching of pump characteristic with the headloss-flow, characteristic of the external system (i.e. pipe network, valve and so on) to which the pump is connected.

Let us consider the pump and the piping system as shown in Fig. 15.18. Since the flow is highly turbulent, the losses in pipe system are proportional to the square of flow velocities and can, therefore, be expressed in terms of constant loss coefficients. Therefore, the losses in both the suction and delivery sides can be written as

$$h_1 = fl_1 V_1^2 / 2gd_1 + K_1 V_1^2 / 2g \qquad (37.2a)$$

$$h_2 = fl_2 V_2^2 / 2gd_2 + K_2 V_2^2 / 2g \qquad (37.2b)$$

where, h_1 is the loss of head in suction side and h_2 is the loss of head in delivery side and *f* is the Darcy's friction factor, l_1, d_1 and l_2, d_2 are the lengths and diameters of the suction and delivery pipes respectively, while V_1 and V_2 are accordingly the average flow velocities. The first terms in Eqs. (37.1a) and (37.1b) represent the ordinary friction loss (loss due to friction between fluid ad the pipe wall), while the second terms represent the sum of all the minor losses through the loss coefficients K_1 and K_2 which include losses due to valves and pipe bends, entry and exit losses, etc. Therefore the total head the pump has to develop in order to supply the fluid from the lower to upper reservoir is

$$H = H_s + h_1 + h_2 \qquad (37.3)$$

Now flow rate through the system is proportional to flow velocity. Therefore resistance to flow in the form of losses is proportional to the square of the flow rate and is usually written as

$$h_1 + h_2 = \text{system resistance} = KQ^2 \qquad (37.4)$$

where *K* is a constant which includes, the lengths and diameters of the pipes and the various loss coefficients. System resistance as expressed by Eq. (37.4), is a measure of the loss of head at any particular flow rate through the system. If any parameter in the system is changed, such as adjusting a valve opening, or inserting a new bend, etc., then *K* will change. Therefore, total head of Eq. (37.2) becomes,

$$H = H_s + KQ^2 \qquad (37.5)$$

The head *H* can be considered as the total opposing head of the pumping system that must be overcome for the fluid to be pumped from the lower to the upper reservoir.

The Eq. (37.4) is the equation for system characteristic, and while plotted on *H-Q* plane (Figure 37.3), represents the system characteristic curve. The point of intersection between the system characteristic and the pump characteristic on *H-Q* plane is the operating point which may or may not lie at the design point that corresponds to maximum efficiency of the pump. The closeness of the operating and design points depends on how good an estimate of the expected system losses has been made. It should be noted that if there is no rise in static head of the liquid (for example pumping in a horizontal pipeline between two reservoirs at the same elevation), H_s is zero and the system curve passes through the origin.