ME 223
ELEMENTS OF FLUID MECHANICS & MACHINERY

CENTRIFUGAL PUMP

LECTURE 5

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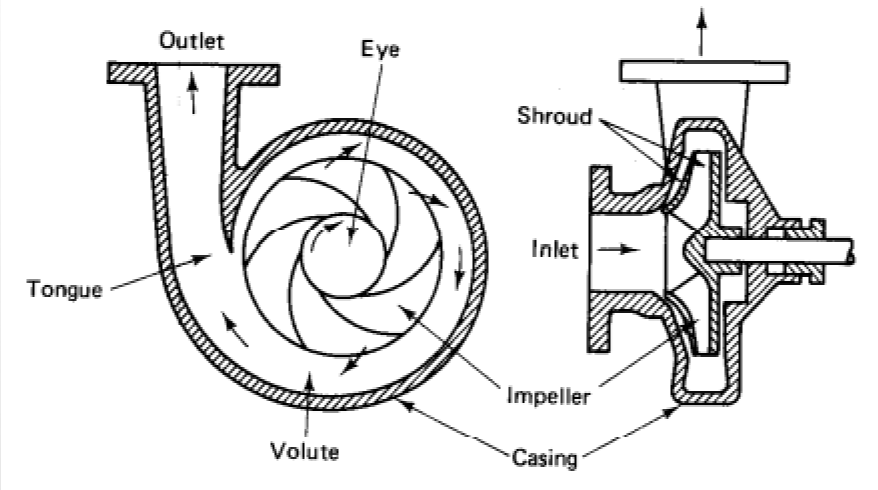
CENTRIFUGAL PUMPS

- A centrifugal pump consists of two principal parts: an impeller, which imparts a rotary motion to the liquid, and the pump housing, or casing, which directs the liquid into the impeller region and transports it away under a higher pressure



CENTRIFUGAL PUMPS

- In a typical single-suction radial flow centrifugal pump, the impeller is mounted on a shaft and is often driven by an electric motor. The casing includes the suction and discharge nozzles and houses the impeller assembly
- The portion of the casing surrounding the impeller is termed the volute. Liquid enters through the suction nozzle to the impeller eye and travels along the shroud, developing a rotary motion due to the impeller vanes. It leaves the volute casing peripherally at a higher pressure through the discharge nozzle

MAIN PARTS OF CENTRIFUGAL PUMP

- Impeller
- Casing
- Suction pipe
- Delivery pipe
- Impeller shaft
- Engine/Motor driver

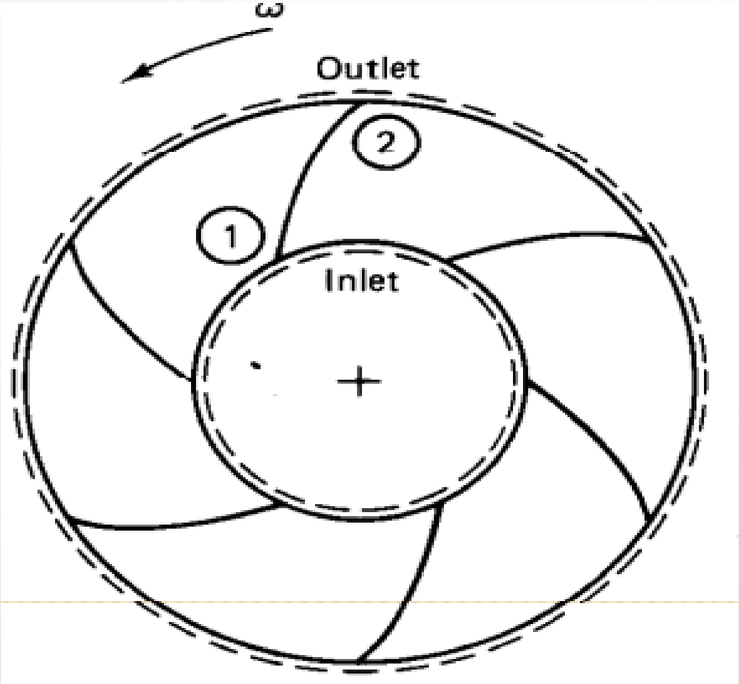
APPLICATIONS

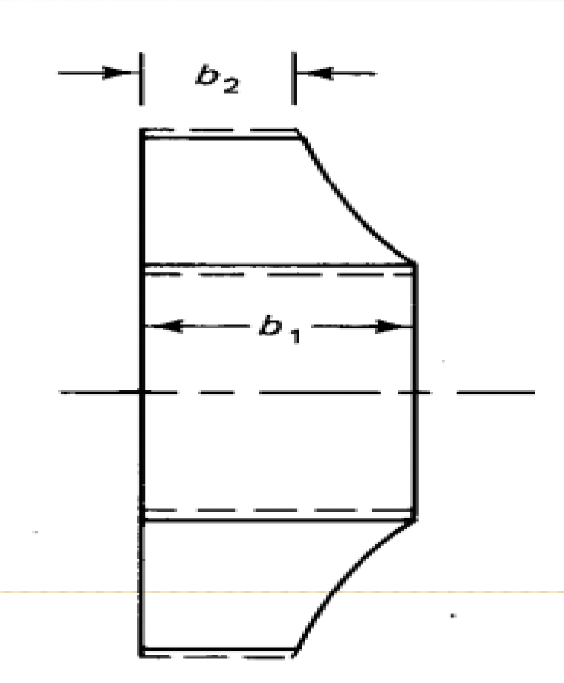
Centrifugal pumps(radial-flow pumps) are the most used pumps for the hydraulic purposes where high discharge with relatively low head is required such as irrigation, cooling tower of central A/C systems, Power plant etc.

CENTRIFUGAL PUMPS: ELEMENTARY THEORY

- The actual flow patterns in a turbo pump are highly three dimensional with significant viscous effects and separation patterns taking place. To construct a simplified theory for the radial-flow Pump, it is necessary to neglect viscosity and to assume idealized Two-dimensional flow throughout the impeller region
- Consider a control volume that encompasses the impeller region. Flow enters through the inlet control surface and exits through the outlet surface. Note that a series of vanes exists within the control volume, and that they are rotating about the axis with an angular speed ω

CENTRIFUGAL PUMPS: ELEMENTARY THEORY





Pump theory

Using subscripts 1 for inlet and 2 for outlet quantities:

D_1 and D_2 =Diameters at inlet of outlet of the impeller

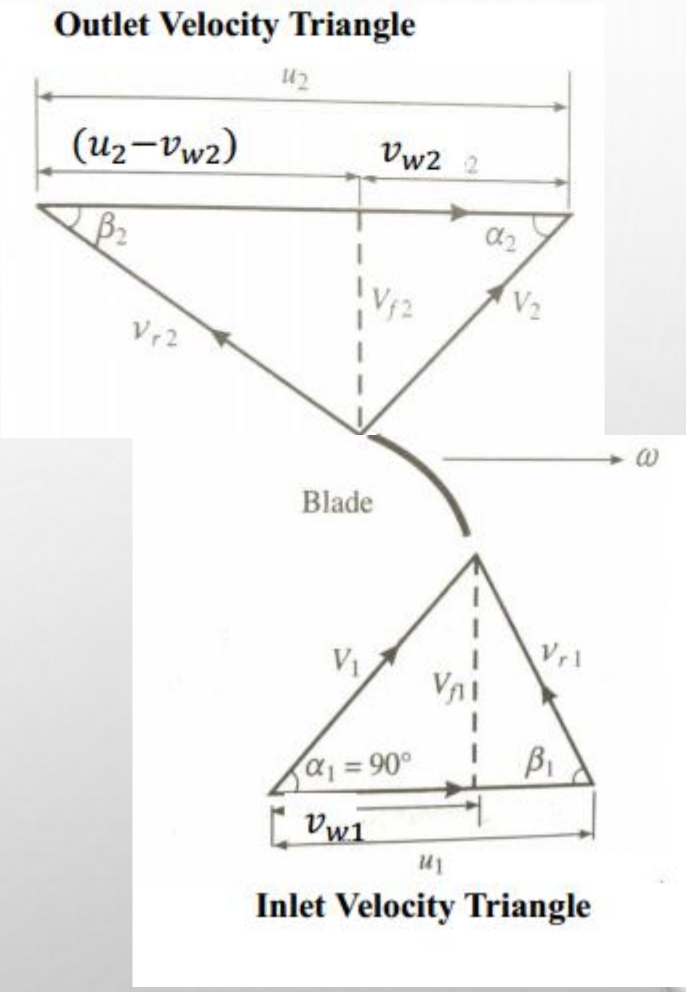
N =Rotational Speed of the impeller in rpm

V_1 and V_2 =Absolute velocities at inlet and outlet

$u_1 \left(= \frac{\pi D_1 N}{60}\right)$ and $u_2 \left(= \frac{\pi D_2 N}{60}\right)$ =Tangential velocities at inner and outer periphery of the impeller

v_{r1} and v_{r2} =Relative velocities at inlet and outlet

α_1 and α_2 =Direction of absolute velocities at inlet and outlet. α is the angle made by the absolute velocity vector V with the positive direction of the peripheral velocity u.



PUMP THEORY

β_1 and β_2 =Vane (blade) angle at inlet and outlet made by the relative velocity vector v_r with the negative direction of the peripheral velocity u.

$v_{w1}$and v_{w_2} =Whirl velocities at inlet and outlet (Tangential components of V_1 and V_2 respectively).

$v_{f1}$and v_{f_2} =Flow velocities at inlet and outlet (Radial components of V_1 and V_2 respectively) .

PUMP THEORY

- The relative velocity is assumed to be always tangent to the vane; that is, perfect guidance of the fluid throughout the control volume takes place
- Since perfect guidance along the vane is assumed, β designates the blade angle as well
- Equating the torque T acting on the fluid to the flux of angular momentum

(= mass x tangential velocity x radius) through the control volume we get for steady flow

$$T = \rho Q \left(r_2 v_{w2} - r_1 v_{w1}\right)$$

PUMP THEORY

- The power delivered to the fluid is

$$\omega T = \rho Q\,(u_2 v_{w2} - u_1 v_{w1})$$

- From the velocity vector diagrams $v_w = V cos\alpha$ so the above Eq. can be written as

$$\omega T = \rho Q\,(u_2 V_2 cos\alpha_2 - u_1 V_1 cos\alpha_1) \qquad [1]$$

- For the idealized situation in which there are no losses, the delivered power must be equal to $\gamma Q H_e$, in which H_e is the theoretical pressure head rise across the pump

PUMP THEORY

- Then from eq. [1], we get Euler's turbo machine relation

$$H_e = \frac{\omega T}{\gamma Q} = \frac{\rho Q\ (u_2 V_2 cos\alpha_2 - \ u_1 V_1 cos\alpha_1)}{\gamma Q} = \frac{u_2 V_2 cos\alpha_2 - \ u_1 V_1 cos\alpha_1}{g} \qquad [2]$$

- Insight on the nature of flow through an impeller region can be obtained using this Eq. From the law of cosines we can write

$$v_{r1}^2 = \ u_1^2 + \ V_1^2 - 2u_1 V_1 cos\alpha_1$$
$$v_{r2}^2 = \ u_2^2 + \ V_2^2 - 2u_2 V_2 cos\alpha_2$$

PUMP THEORY

- These can be substituted into the Euler's relation to provide

$$H_e = [\frac{V_2^2 - V_1^2}{2g} + \frac{v_{r1}^2 - v_{r2}^2}{2g} + \frac{u_2^2 - u_1^2}{2g}]$$

Here:

$\frac{V_2^2 - V_1^2}{2g}$ represents the increase in absolute kinetic energy of fluid;

$\frac{u_2^2 - u_1^2}{2g}$ represents the increase in static pressure due to centrifugal action;

$\frac{v_{r1}^2 - v_{r2}^2}{2g}$ represents the change in the kinetic energy due to retardation of flow.

PUMP THEORY

- Assuming radial entry

$$\text{Euler Head, } H_e = \frac{u_2 v_{w2}}{g}$$

Actual head developed by the impeller, H_e

Manometric Head (H_m) of a pump is the gross head that must be provided by the impeller for the liquid to flow from the sump to the delivery point.

Manometric (Hydraulic) Efficiency, $\eta_m = \frac{H_m}{H_e}$

MINIMUM STARTING SPEED

At the time of start, the fluid velocities are zero and the only head that is operating is the centrifugal head . this centrifugal force must overcome the manometric head for the fluid to move, i.E.,

$$\frac{u_2^2 - u_1^2}{2g} \geq H_m$$

Here, $u_1 = \frac{\pi D_1 N}{60}$, $u_2 = \frac{\pi D_2 N}{60}$

So that equating, $\frac{u_2^2 - u_1^2}{2g} = H_m$

MINIMUM STARTING SPEED

$$N_{min} = 84.596\sqrt{\frac{H_m}{D_2^2 - D_1^2}}$$

Again, $\frac{u_2^2 - u_1^2}{2g} \geq \eta_m H_e$

By equating, $\frac{u_2^2 - u_1^2}{2g} = \eta_m \left(\frac{u_2 v_{w2}}{g}\right)$

$$N_{min} = \frac{120\eta_m}{\pi}\left(\frac{v_{w2} D_2}{D_2^2 - D_1^2}\right)$$

MINIMUM IMPELLER DIAMETER

Usually the impeller outer diameter is designed as twice the inner (inlet) diameter, i.e., $D_2 = 2D_1$. Using this relationship, the minimum diameter of the impeller for fluid to move,

$$\frac{u_2^2 - u_1^2}{2g} \geq H_m$$

So that equating, $\frac{u_2^2 - u_1^2}{2g} = H_m$

$$D_{2/min} = 97.68 \frac{\sqrt{H_m}}{N}$$

This equation is used in practical situations to design impeller for liquid pumping at a given speed

PRIMING

When the pump casing and the suction conduit are completely filled with water, as the impeller rotates, the pressure at the pump suction side becomes lower than the atmospheric pressure. Due to this difference in pressure head between the water surface of the sump and the inlet of the pump, the atmospheric pressure pushes the water from the sump to the pump casing. However, an impeller running in air would produce only a small head. This cannot create the necessary differential head of water between the sump and the pump inlet as the density of air is much less than that of water. Consequently, the pump does not do its work of pumping of water.

PRIMING

Further, dry running of the pump may damage several parts of the pump. This is, therefore, necessary to ensure that the pump casing, impeller, suction pipe and the portion of the pump delivery pipe up to the delivery valve are always filled with water before the start of the pump. Filling is done by pouring water into the funnel or priming-cup provided for this purpose. An air vent in the casing is provided for the air to escape. This air vent must be closed after filling. This filling process is called the “priming” of the pump. Most centrifugal pumps are not self-priming, so they always need priming.

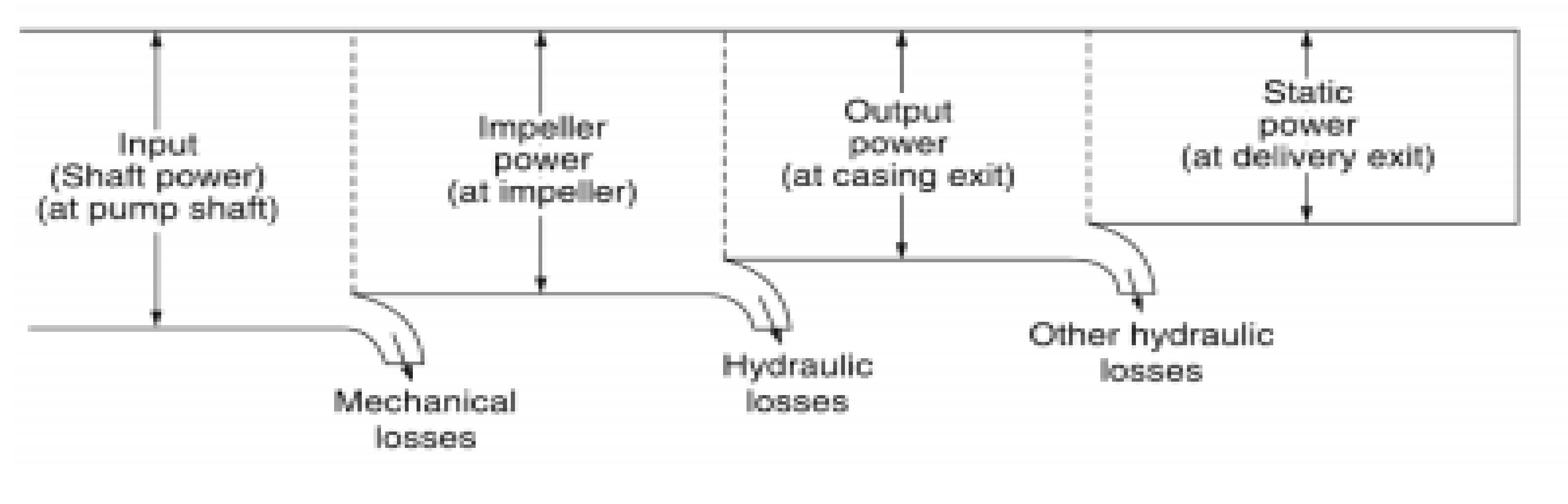
Problem 1:

A centrifugal pump impeller runs at 950 rpm (N). Its external and internal diameters are 500 mm (D_2) and 250 mm (D_1) respectively . The vanes are set back at an angle of 35^0 (β_2) to the outer rim. If the radial velocity of water through the impeller is maintained constant at 2 m/s ($V_{f1} = V_{f2}$). Find the angle of the vanes at inlet (β_1), the velocity and direction of water at outlet(v_2, α_2) and the euler head (H_e).

Problem 2:

A centrifugal pump delivers 0.20 m^3/s water against a head of 26 m while running at 950 rpm. The constant velocity of flow is 2.9 m/s and the vanes are curved backward at an angle of $30^0\,(\beta_2)$. If the manometric efficiency is 77%, find the diameter and the width (or breadth) of the impeller at outlet. Also find the power required to run the impeller. Neglect the effect of vane thickness and mechanical friction and leakage losses.

HEAD LOSSES IN CENTRIFUGAL PUMP



HEAD LOSSES IN CENTRIFUGAL PUMP

- Mechanical losses are the frictional losses in bearings, glands, packages, etc. And the disc friction between the impeller and the liquid which fills the clearance space between the impeller and the casing
- Some leakage loss also take place between impeller and casing, at mechanical seals, glands, etc.
- Hydraulic losses are due to:
  - Circulatory flow at the passages of the impeller and independent of the discharge

HEAD LOSSES IN CENTRIFUGAL PUMP

- Fluid friction at the flow passage: this loss depends on the fluid contact area and the roughness of the surface and hence equal to $K_1 Q^2$ Where K_1 is a coefficient.
- Shock losses at the entrance to impeller: this loss occurs due to improper entry angle of the flow with respect to the blade angle. At design condition, this loss is practically zero and increases at reduced or increased flow from normal values.

HEAD LOSSES IN CENTRIFUGAL PUMP

To account for various losses, several efficiencies are defined.

- Volumetric efficiency , $\eta_v = \frac{Q}{Q+Q_L}$

where Q is Discharge reaching the pump outlet, Q_L is the leakage flow which does not reach the pump outlet and $Q + Q_L$ is Discharge entering the eye of the impeller.

- Mechanical efficiency, $\eta_{mech} = \frac{\gamma(Q+Q_L)H_e}{p}$

Where P is the mechanical power input to the impeller shaft by the prime mover and H_{mech} is the mechanical head losses.

HEAD LOSSES IN CENTRIFUGAL PUMP

- Overall efficiency, $\eta_o = \eta_v \times \eta_m \times \eta_{mech}$

$$\eta_o = \frac{Q}{Q + Q_L} \times \frac{H_m}{H_e} \times \frac{\gamma (Q + Q_L) H_e}{p}$$

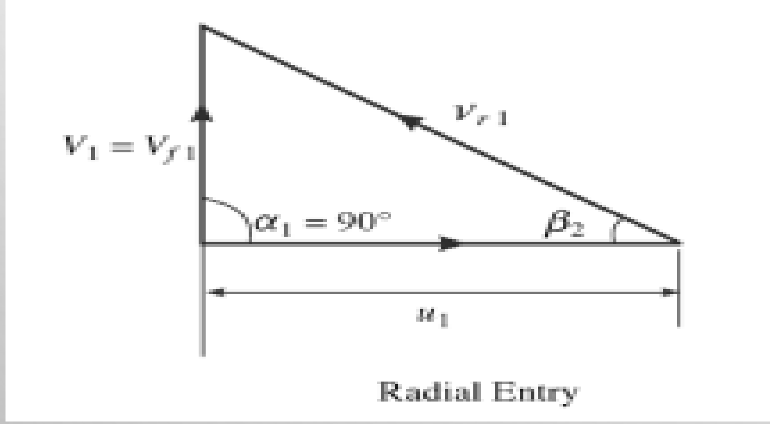
$$\eta_o = \frac{\gamma Q H_m}{p}$$

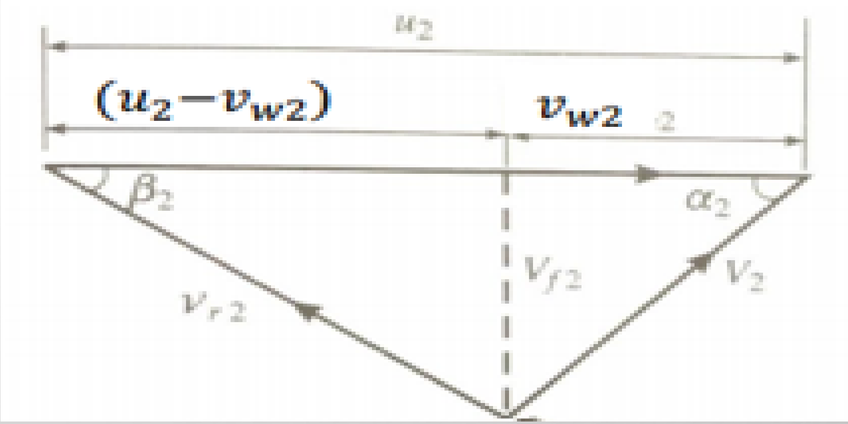
THEORETICAL HEAD-DISCHARGE RELATIONSHIP

Consider the theoretical head (Euler head) as given by

$$H_e = \frac{u_2 v_{w2}}{g}$$

Assumption: Radial entry at the inlet. This implies that $\alpha_1 = 90^0\ and\ v_{f1} = v_1$





THEORETICAL HEAD-DISCHARGE RELATIONSHIP

Consider the velocity triangle at outlet

$$\tan\beta_2 = \frac{v_{f2}}{u_2 - v_{w2}}$$

$$v_{w2} = u_2 - v_{f2}\cot\beta_2$$

By neglecting the blade thickness, the discharge at the impeller outlet

$$Q = \pi b_2 D_2 v_{f2}$$

Where b_2 = width of the impeller at the outlet and D_2 = Diameter of the impeller at the outlet

THEORETICAL HEAD-DISCHARGE RELATIONSHIP

$$v_{f2} = \frac{Q}{\pi b_2 D_2} \quad \text{and} \quad v_{w2} = u_2 - \frac{Q}{\pi b_2 D_2} \cot \beta_2$$

Hence, $H_e = \frac{u_2 v_{w2}}{g} = \frac{u_2}{g}\left(u_2 - \frac{Q}{\pi b_2 D_2} \cot \beta_2\right)$

For a given pump running at constant speed, the above equation can be written as

$$H_e = A - BQ \cot \beta_2$$

Where A and B are constants for a given impeller at a constant speed.

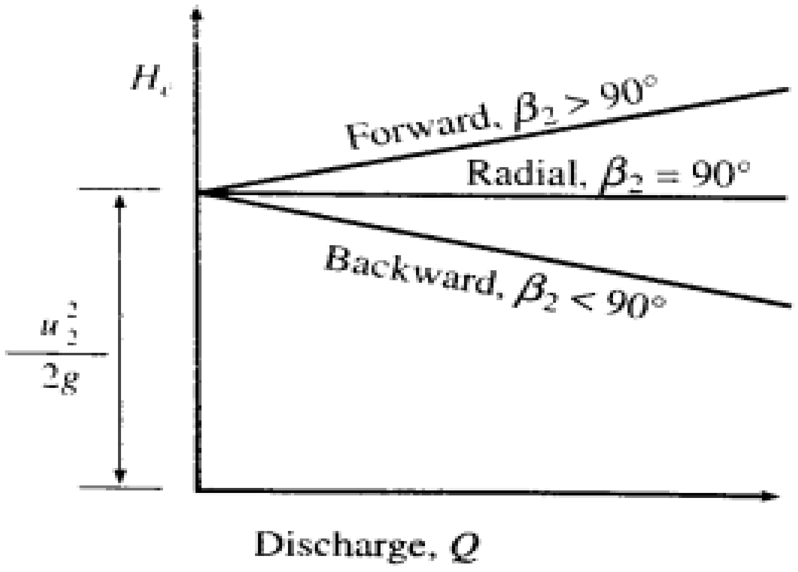
THEORETICAL HEAD-DISCHARGE RELATIONSHIP

The above equation shows that for a given vane angle β_2, the theoretical head H_e Varies linearly with the discharge Q. When $Q = 0$, there is a finite positive value of H_e. This indicates that if the delivery valve of a running pump is completely closed shut and the pump is kept running, a positive pressure head is produced by the pump. This is known as shut-off head. The ideal value of shut-off head is

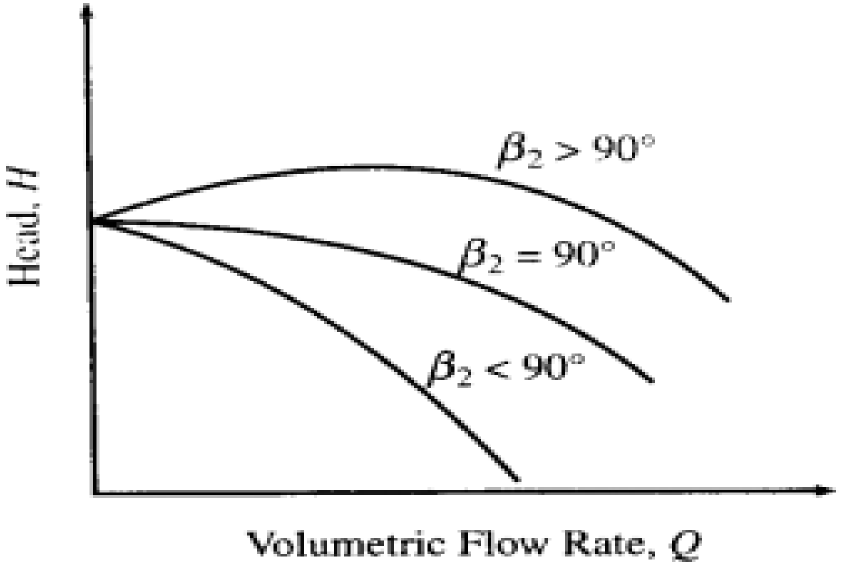
$$H_{e,shut-off} = \frac{u_2^2}{g}$$

The actual value will be smaller than this value, roughly about 60%, due to some unrecoverable losses in the impeller.

THEORETICAL HEAD-DISCHARGE RELATIONSHIP



Head-Discharge (H_e-Q) relationship for ideal pumps



Head-Discharge (H_e-Q) relationship for actual pumps

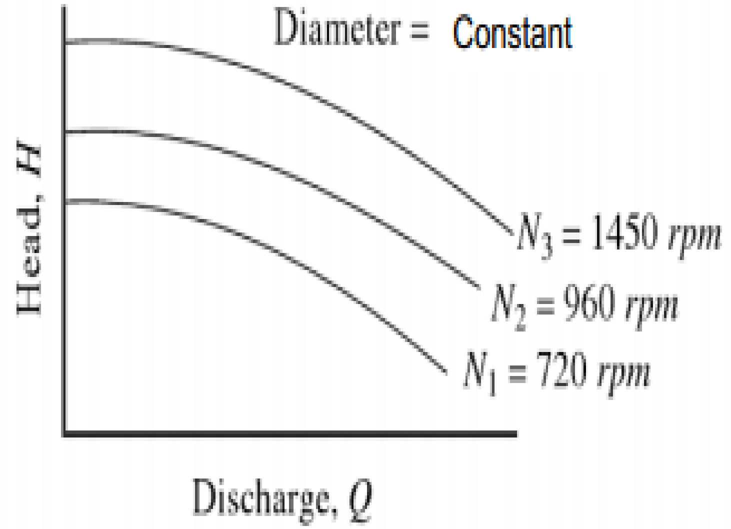
THEORETICAL HEAD-DISCHARGE RELATIONSHIP

From the ideal $H - Q$ curve on the left, it can be inferred that forward curved vanes produce higher head at higher discharge. However, both forward-curved and radial vane pumps result in poor efficiency. Forward-curved vane produce larger absolute velocities that require very efficient diffusers to convert the exit kinetic energy into pressure energy. So the energy losses are high.

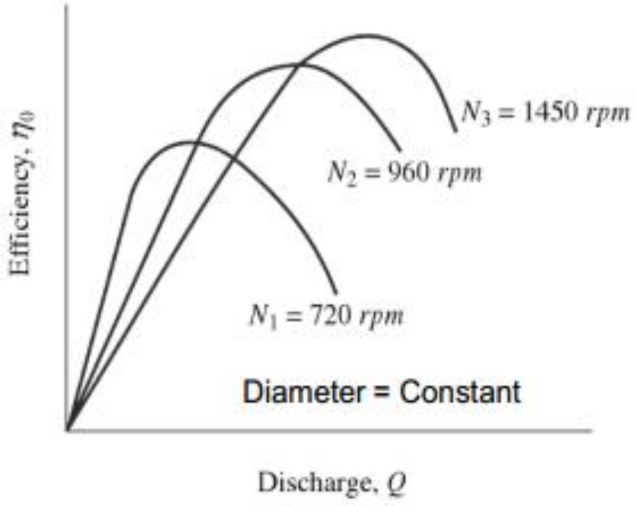
Therefore, in actual practice, backward-curved vanes in the range of $20° - 40°$ are of common use.

In actual $H - Q$ curve shown on the right the head decreases with increase in discharge for all types of vanes due to hydraulic losses present in real-world applications.

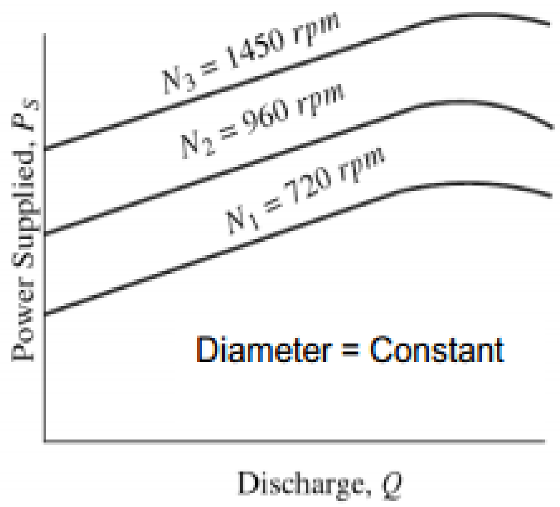
TYPICAL CHARACTERISTICS CURVES



Head-Discharge ($H - Q$) characteristics



Variation of Efficiency with discharge

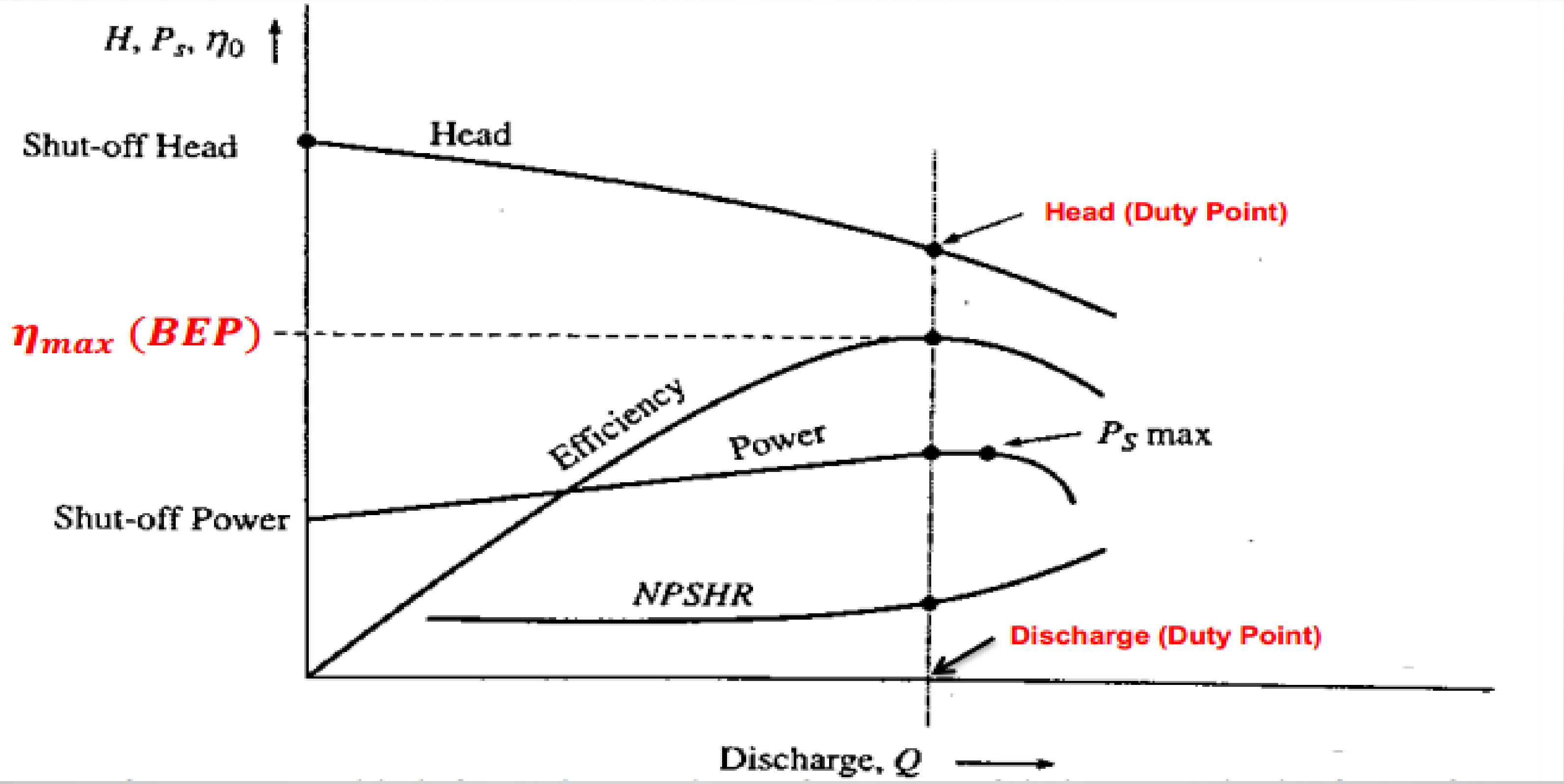


Variation of Power with discharge

TYPICAL CHARACTERISTICS CURVES

- From the $H - Q$ curve, it can be observed that for the same pump (same impeller/diameter) as the speed increases, the head also increases ($H \propto N^2$) at the same discharge, or the discharge increases ($Q \propto N$) at the same head. The higher the pump speed, the higher will be the head or discharge.
- From the $\eta - Q$ curve, it can be observed that for the same pump (same impeller/diameter) as the speed increases, both the maximum efficiency and the maximum discharge increase. The higher the speed, the higher will be the efficiency and discharge.
- However, the $P - Q$ curve shows that for a given discharge, higher speed needs higher power input.

MAIN CHARACTERISTICS CURVE



MAIN CHARACTERISTICS CURVE

Pump manufacturers provide information on the performance of their pumps in the form of curves, commonly called pump characteristic curve (or simply pump curve). In pump curves the following information may be given:

(i) the discharge on the x-axis, (ii) the head on the left y-axis, (iii) the pump efficiency as a percentage on the right (or left) y-axis, (iv) the pump power input on the left (or right) y-axis, (v) the NPSH of the pump on the y-axis (vi) the speed of the pump one the y-axis.

The discharge-head ($\boldsymbol{Q}$, $\boldsymbol{H}$) values corresponding to BEP (η_{max}) is called the 'Duty Point' of the pump.

SPECIFIC SPEED

- ❖ Specific speed of a pump is defined as the speed of an imaginary pump which will produce the unit discharge under unit head.
- ❖ This is a numerical engineering tool for the selection of the type of the pump for installation.

$$\text{Specific speed, } N_s = \frac{N\sqrt{Q}}{H_m^{3/4}}$$

- ❖ The values N, Q and H_m are taken at Best Efficiency Point (BEP)

SPECIFIC SPEED

Please note the usage of units while calculating Specific Speed. Different units will result in different values for the same pump.

N_s = Specific Speed $N_s = \frac{N\sqrt{Q}}{H^{3/4}}$ N = Speed in rpm Q = Discharge in US gallons per minute H = Head in ft [1 US gallon = 3.785 Liters] In US Customary Unit (FPS) Reciprocating Pump : 50 to 500 **Centrifugal Pump: 500 to 10000** Radial Flow Pump : 500 to 4000 Mixed Flow Pump : 2000 to 8000 Axial Flow/Propeller Pump: 7000 to 20000	$N_s = c\,\frac{N\sqrt{Q}}{H^{3/4}}$ N = Speed in rpm Q = Discharge in m^3/hr H = Head in m $c = 0.861$ if Q in m^3/hr, $c = 6.67$ if Q in m^3/min and $c = 51.66$ if Q in m^3/s If the factor c is omitted, the calculated specific speeds will be different than the US customary values, for example, specific speed of **Centrifugal Pump: 10 to 220 if Q is in m^3/s** The factor c (might be omitted) is used to compare the value obtained with that obtained from US customary unit which are prevalent in usage among practicing professionals.

SPECIFIC SPEED

- For instance, in an application, if the required discharge and head are known, the prime mover (motor or engine) rpm is also known, then using these values N_s can be calculated and a particular type of pump suitable in this N_s range can be selected for installation.

CAVITATION OF PUMPS

- Cavitation is rapid formation and collapse of vapor bubbles within a liquid. In general, cavitation occurs when the liquid pressure at a given location is reduced to the vapor pressure of the liquid
- For a piping system that includes a pump, cavitation occurs when the absolute pressure at the inlet falls below the vapor pressure of the water
- This phenomenon may occur at the inlet to a pump and on the impeller blades, particularly if the pump is mounted above the level in the suction reservoir

CAVITATION OF PUMPS

- Under this condition, vapor bubbles form (water starts to boil) at the impeller inlet and when these bubbles are carried into a zone of higher pressure, they collapse abruptly and hit the vanes of the impeller (near the tips of the impeller vanes). Causing:
    - Damage to the pump (pump impeller)
    - Violet vibrations (and noise)
    - Reduce pump capacity
    - Reduce pump efficiency

CAVITATION OF PUMPS

- To avoid cavitation, the pressure head at the inlet should not fall below a certain minimum which is influenced by the further reduction in pressure within the pump impeller. The parameter used for the determination of cavitation is called 'Net Positive Suction Head (NPSH)'.

NET POSITIVE SUCTION HEAD

- NPSH is the difference between the total head at the pump inlet and the water vapor pressure head $(H_v = 25\ m)$, i.e. $NPSH = H_{pi} + \frac{v_s^2}{2g} - H_v$

  the datum is taken through the centerline of the pump impeller inlet (eye).

- There are two values of NPSH of interest. The first is the required NPSH, denoted $(NPSH)_R$, that must be maintained or exceeded so that cavitation will not occur and usually determined experimentally and provided by the manufacturer.

NET POSITIVE SUCTION HEAD

- The second value for NPSH of concern is the available NPSH, denoted $(NPSH)_A$, which represents the absolute pressure at the suction port of the pump. For proper pump operation (no cavitation) :

$$(NPSH)_A > (NPSH)_R$$

$$\boldsymbol{(NPSH)_{available\ (at\ the\ installation\ site)} > (NPSH)_{required\ (for\ pump)}}$$

As stated above, $(NPSH)_R$ is usually given for a particular pump by the manufacturer for its installation without cavitation. $(NPSH)_A$ is calculated at the installation site.

NET POSITIVE SUCTION HEAD

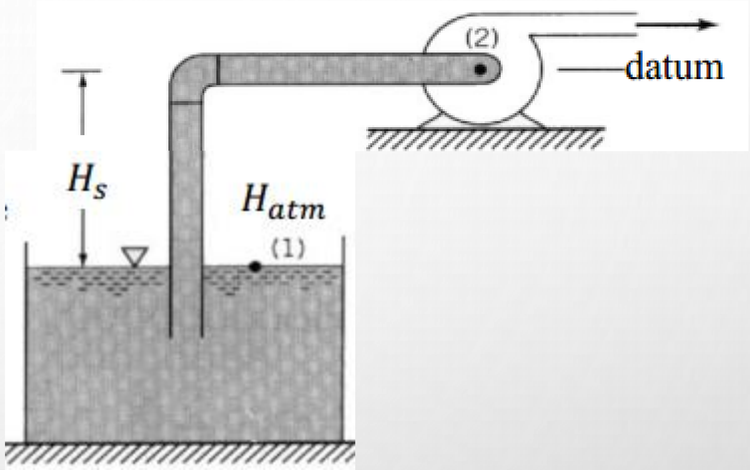
$$NPSH = H_{pi} + \frac{v_s^2}{2g} - H_v$$

Applying the Bernoulli's equation between point (1) and (2), datum at pump center line

$$H_{atm} - H_s - H_L = H_{pi} + \frac{v_s^2}{2g}$$

$$NPSH_A = H_{atm} - H_s - H_L - H_v$$

Where, $H_L = f\frac{L}{D}\frac{v^2}{2g} + \sum k\frac{v^2}{2g}$



Problem 3:

The $NPSH_R$ of a centrifugal pump is given by the manufacturer as 7.5 m abs. The pump is employed to pump water at 0.3 $\frac{m^3}{s}$ from a sump whose water level is 2.05 m below the pump inlet. The atmospheric pressure at the site is 97 kPa abs and the vapor pressure at the relevant temperature is 2.35 kPa abs. Total head loss in the suction pipe is estimated to be 0.95 m. Determine the $NPSH_A$ and comment on the suitability of the installation against the cavitation.

ONE who is contented with
what he/she has
is the happiest man in the
universe.