SOLUTION

B. Tech (Fifth Semester) Mechanical Engg.
Subject:- Fluid Machinery
Subject Code:- ME355
Paper Code:- AS-4146

Section-A

1. Multiple choice questions

(i) (d) at which the runner can be allowed to run freely without load and with gates wide open

(ii) (b) Kaplan

(iii) (b) curved backward

(iv) (c) $u=0$ at $y=0$

(v) (d) 60 m/s

(vi) (a) $\frac{N}{\sqrt{H}}$

(vii) (a) 2.5 m

(viii) (a) relative velocity and direction of motion of vane

(ix) (c) Reduce acceleration heads to minimum

(x) (b) $\frac{1}{2}$

2. Answer in brief

(i) * Specific speed is used as a guide to select a type of turbine/pump under the given condition of head and flow (i.e. site condition).

* It is the same for geometrically similar machines.

(ii) The energy thickness may be conceived as the transverse distance by which the boundary layer should be displaced to compensate for the reduction in energy of the flowing fluid on account of the boundary layer formation.

$$\delta^{**} = \int_0^\delta \frac{u}{U_\infty}\left[1 - \left(\frac{u}{U_\infty}\right)^2\right] dy$$

(iii) Dimensionless quantities have the following advantages:-

* The prediction of a prototype performance from the test conducted on a scale model.
* To find the most suitable type of machine on the basis of maximum efficiency for a specified range of head, speed and flow rate.
* A large no. of variables are involved in describing the performance characteristics of a fluid machine. Because of these large no. of variables, more no. of experiments have to be conducted in a performance test. In order to reduce time and cost, these variables are grouped into dimensionless quantities.

(iv) Slip of a reciprocating pump is defined as the difference between the theoretical discharge and actual discharge of the pump.

$$\text{Slip} = Q_{th} - Q_{act}$$

$$\% \text{ Slip} = \frac{Q_{th} - Q_{act}}{Q_{th}} \times 100 = \left(1 - \frac{Q_{act}}{Q_{th}}\right) \times 100$$

$$= (1 - C_d) \times 100 \qquad C_d = \text{coeff. of discharge.}$$

2(V) Unit Power (P_u):–

It is defined ~~by~~ as the power developed by a turbine, working under unit head.

$$\eta_o = \frac{P}{\rho g Q H}$$

$$P = \eta_o \times \rho g Q H$$

$$P \propto QH \propto \sqrt{H}\, H \propto H^{3/2}$$

$$P = K\, H^{3/2}$$

If $H = 1\,m$, $P = P_u$

$$K = P_u$$

or $P = P_u H^{3/2}$

or $\boxed{P_u = \frac{P}{H^{3/2}}}$

Section - B

3. (a) $\frac{u}{U_\infty} = 2\left(\frac{y}{\delta}\right) - \left(\frac{y}{\delta}\right)^2$

Displacement thickness $\delta^* = \int_0^\delta \left(1 - \frac{u}{U_\infty}\right) dy$

$$\delta^* = \int_0^\delta \left[1 - \left\{2\left(\frac{y}{\delta}\right) - \left(\frac{y}{\delta}\right)^2\right\}\right] dy$$

$$= \left[y - \frac{2y^2}{2\delta} + \frac{y^3}{3\delta^2}\right]_0^\delta$$

$$= \frac{\delta}{3}$$

Momentum thickness $\cancel{\delta^*}$ $\theta = \int_0^\delta \frac{u}{U_\infty}\left\{1 - \frac{u}{U_\infty}\right\} dy$

$$\theta = \int_0^\delta \left(\frac{2y}{\delta} - \frac{y^2}{\delta^2}\right)\left[1 - \left(\frac{2y}{\delta} - \frac{y^2}{\delta^2}\right)\right] dy$$

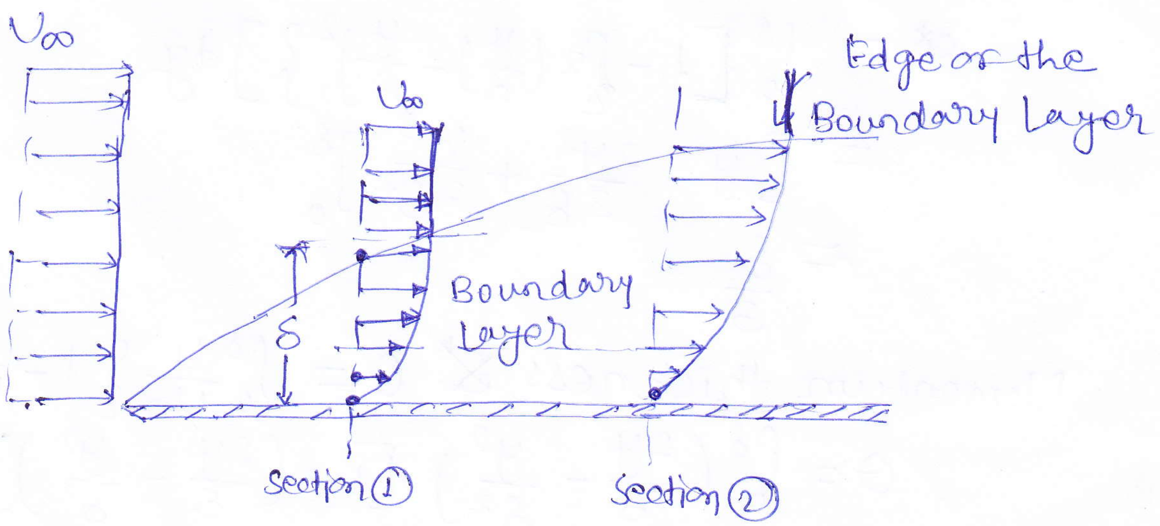
$$= \int_0^\delta \left[\frac{2y}{\delta} - \frac{4y^2}{\delta^2} + \frac{2y^3}{\delta^3} - \frac{y^2}{\delta^2} + \frac{2y^3}{\delta^3} - \frac{y^4}{\delta^4}\right] dy$$

$$= \left[\frac{2y^2}{2\delta} - \frac{5y^3}{3\delta^2} + \frac{4y^4}{4\delta^3} - \frac{y^5}{5\delta^4}\right]_0^\delta = \frac{2\delta}{15}$$

(b) When a real fluid flows past a solid wall, the fluid particles adhere to the boundary and condition of no slip occurs. This means that the velocity of fluid close to the boundary will be same as that of the boundary. The velocity of fluid increases from zero velocity on the stationary boundary to free stream velocity (U_∞) of the fluid in the direction normal to the boundary. This variation of velocity from zero to free stream velocity in the direction normal to the boundary takes place in the narrow region is called boundary layer.

If we join the tip of the velocity vector ~~so what~~ we get a velocity profile at ~~this~~ location.

If we draw the same velocity profile at a section further away from the leading edge. Velocity is zero at the surface, uper layer velocity should be even less than, that of the previous section. The region is obvious that now more and more fluid is in contact with the plate so the effect of slowing down by the wall is more and more severe. Here also the U_∞ condition will be reached, the distance over which it will rich, will be more than the distance it was reached in the previous section.



3 (C) Dia. of cylinder $D = 50\,mm = 0.05\,m$

Length of cylinder $L = 1.0\,m$, Density of air $\rho = 1.25\,kg/m^3$

$\therefore$ Projected area $A = L \times D = 0.05\,m^2$

velocity of air $U_\infty = 0.1\,m/s$

Total drag coefficient $C_{DT} = 1.5$

Shear drag coefficient $\cancel{C_{DS}} = C_{DS} = 0.2$

Total drag is given by, $F_D = C_{DT} \times A \times \frac{\rho U_\infty^2}{2}$

$$F_D = 1.5 \times 0.5 \times \frac{1.25 \times (0.1)^2}{2} = 0.000468\,N$$

Shear drag is given by $F_{DS} = C_{DS} \times A \times \frac{\rho U_\infty^2}{2}$

$$F_{DS} = 0.2 \times 0.05 \times \frac{1.25 \times (0.1)^2}{2} = 0.0000625\,N$$

Total drag, F_D = Pressure drag + shear drag

$\therefore$ Pressure Drag $= 0.000468 - 0.0000625$

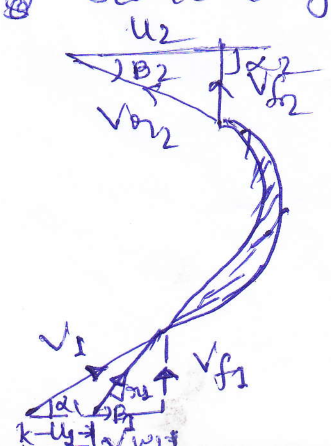
$= 0.0004055\,N.$

~~5~~5(a) Degree of Reaction R is defined as the ratio of pressure energy change inside the runner (energy transit by means of static pressure) to the total energy transfer.

The energy transfer per unit weight (the Euler head) in a turbine is

$$H_e = \frac{V_{w_1} U_1 - V_{w_2} U_2}{g} \quad \text{--- (1)}$$

From the inlet velocity triangle corresponding to the blade angle at inlet $B_1 < 90°$



$V_{f_1}^2 = V_1^2 - V_{w_1}^2$ and $V_{f_1}^2 = V_{r_1}^2 - (V_{w_1} - U_1)^2$

Combining these two expressions

$$V_1^2 - V_{w_1}^2 = V_{r_1}^2 - (V_{w_1} - U_1)^2$$

$$V_{w_1} U_1 = \frac{1}{2}(V_1^2 + U_1^2 - V_{r_1}^2)$$

Like wise the general outlet-velocity triangle would yield

$$V_{w_2} u_2 = \frac{1}{2}\left(V_2^2 + u_2^2 - V_{r_2}^2\right)$$

upon substitution in equation (i) we get

$$H_e = \frac{V_1^2 - V_2^2}{2g} + \frac{u_1^2 - u_2^2}{2g} + \frac{V_{r_2}^2 - V_{r_1}^2}{2g}$$

The term $\frac{V_{r_2}^2 - V_{r_1}^2}{2g}$ represents change in K.E. due to acceleration of fluid relative to runner; it represents conversion of K.E. within the runner itself.

The term $\frac{u_1^2 - u_2^2}{2g}$ represents the change in static pressure that is accomplished due to movement of rotating fluid from one radius of rotation to another.

Therefore the total energy change inside the runner is $\frac{(V_{r_2}^2 - V_{r_1}^2)}{2g} + \frac{(u_1^2 - u_2^2)}{2g} = H_e - \frac{V_1^2 - V_2^2}{2g}$

$\therefore$ Degree of Reaction $R = \frac{H_e - (V_1^2 - V_2^2)/2g}{H_e}$

$$\boxed{R = 1 - \frac{V_1^2 - V_2^2}{2g H_e}}$$

$$\boxed{R = 1 - \frac{\cot\alpha_1}{2(\cot\alpha_1 - \cot\beta_1)}}$$

5(b)

5(b) Flow ratio, $\frac{V_{f1}}{\sqrt{2gH}} = 0.20$

$V_{f1} = 0.20 \times \sqrt{2g \times 60} = 6.862$ m/s

Breadth Ratio $\Rightarrow \frac{B_1}{D_1} = 0.1$

Outer diameter $D_1 = 2 \times D_2$

velocity of flow $V_{f1} = V_{f2} = 6.862$ m/s

Actual area of flow $= 0.95 \times \pi D_1 \times B_1$

$V_{w2} = 0$ and $V_{f2} = V_2$, $\eta_0 = \frac{S.P.}{W.P} \Rightarrow 0.84 = \frac{294.3}{W.P.}$

$W.P. = 350.357$ kW

$W.P. = \frac{\rho g Q H}{1000} \Rightarrow Q = 0.5952$ m³/sec

$Q = 0.95 \times \pi D_1 \times B_1 \times V_{f1}$

$= 0.95 \times \pi D_1 \times 0.1 D_1 \times V_{f1}$

$D_1 = 0.54$ m

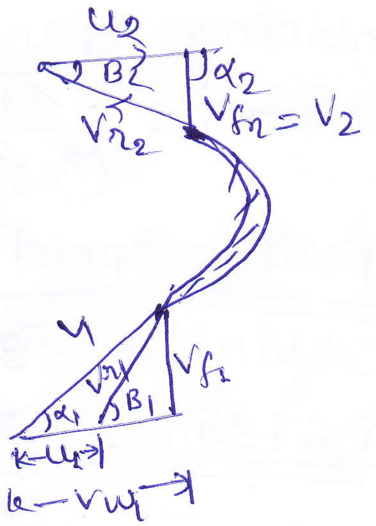
$\frac{B_1}{D_1} = 0.1$, $B_1 = 0.054$ m $= 54$ mm

$u_1 = \frac{\pi D_1 N}{60} = 19.79$ m/s, $\eta_h = \frac{V_{w1} u_1}{gH}$; $V_{w1} = 27.66$ m/s

(i) Guide Blade angle

$\tan\alpha = \frac{V_{f1}}{V_{w1}} = 0.248$

$\alpha = 13.92°$



(ii) Blade angles (B_1, B_2)

$\tan B_1 = \frac{V_{f1}}{V_{w1} - u_1}$, $B_1 = \cancel{37.74}$ $41.09°$

$\tan B_2 = \frac{V_{f2}}{u_2} \Rightarrow u_2 = \frac{\pi D_2 N}{60} = 9.896$ m/s

$B_2 = 34.74°$

$D_1 = 0.54$ m , $D_2 = 0.27$ m

$B_1 = 54$ mm

5 (c) Following are the variables considered

Flow rate $Q\ (m^3/s)$

Speed $N\ (rpm)$

Power $P\ (Watt)$

Head or Pressure $H\ (meters)$

Diameter $(d)\ (meters)$

Fluid Density $\rho\ (kg/m^3)$

Fluid viscosity $\mu\ (N\text{-}s/m^2)$

$P = f(\rho, N, \mu, d, Q, H)$

$f(P, \rho, N, \mu, d, Q, H) = 0$

Repeating variables

d, N, ρ

$\therefore m = 3$

Total variables $n = 7$

π-terms:-

$(n-m) = 4\ \pi$-terms

π_1-term $= d^{a_1} N^{b_1} \rho^{c_1} Q$

$M^0 L^0 T^0 = [L]^{a_1} [T^{-1}]^{b_1} [ML^{-3}]^{c_1} [L^3 T^{-1}]$

$$\boxed{\pi_1 = \frac{Q}{Nd^3}}$$

π_2-term $= d^{a_2} N^{b_2} \rho^{c_2} H = \dfrac{H}{N^2 d^2}$

π_3-term $= d^{a_3} N^{b_3} \rho^{c_3} P = \dfrac{P}{\rho N^3 d^5}$

π_4 term $= d^{a_4} N^{b_4} \rho^{c_4} \mu = \dfrac{\rho N d^2}{\mu}$

Specific Speed for Pumps

Combining π_1 and π_2

$$\pi\text{-}5 \text{ term} = \frac{\sqrt{\pi_1}}{(\pi_2)^{3/4}} = \frac{Q^{1/2} N^{3/2} d^{3/2}}{N^{1/2} d^{3/2} H^{3/4}} = \boxed{\frac{N\sqrt{Q}}{H^{3/4}} = N_{sp}}$$

Specific speed for turbines

Combining π_3 and π_2

$$\pi\text{-}6 \text{ term} = \frac{(\pi_1)^{1/2}}{(\pi_2)^{5/4}} = \frac{P^{1/2} N^{5/2} d^{5/2}}{\rho^{1/2} N^{3/2} d^{5/2} H^{5/4}}$$

$$\boxed{N_{sT} = \frac{N\sqrt{P}}{H^{5/4}}}$$

4(a) Let us consider a large number of plates are mounted on the circumference of a wheel at a fixed distance as shown in fig.

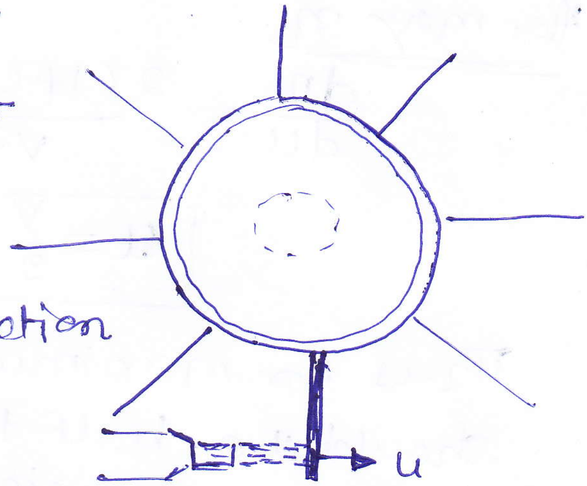
Let V = velocity of jet

d = dia. of jet

a = cross-section area of jet

$= \frac{\pi}{4} d^2$

u = velocity of vane



Force exerted by the jet in the direction of motion of plate

$$F_x = \rho a V[(V-u) - 0]$$
$$= \rho a V (V-u)$$

W.D. by the jet on the ~~surface~~ series of plates per sec

$$= F_x \times u = \rho a V[V-u] \times u$$

K.E. of the jet per sec $= \frac{1}{2} \dot{m} V^2 = \frac{1}{2}(\rho a V) V^2$

$$\therefore \eta = \frac{\text{W.D. per sec}}{\text{K.E. per sec}} = \frac{\rho a V[V-u] \times u}{\frac{1}{2} \rho a V^3}$$

$$\eta = \frac{2u[V-u]}{V^2} \quad \text{—} \quad (1)$$

Condition for maximum efficiency; for a given jet velocity the efficiency will be max when

$$\frac{d\eta}{du} = 0 \Rightarrow \frac{d}{du}\left[\frac{2u(V-u)}{V^2}\right] = 0$$

$$u = \frac{V}{2}$$

Force exerted on a series of Radial Curved Vanes:-

(Jet strikes at the centre):-

$$F = \rho a V[(V-u) - \{-(V-u)\cos\alpha\}]$$

$$F = \rho a V[(V-u)(1+\cos\alpha)]$$

$$\eta = \frac{\text{W.D. per sec on the vane}}{\text{K.E. supplied by the jet}}$$

$$\eta = \frac{\rho A V (V-u)[1+\cos\alpha] \times u}{\frac{1}{2}(\rho a V) \times V^2}$$

$$\eta = \frac{2u(V-u)(1+\cos\alpha)}{V^2}$$

For max η

$$\frac{d\eta}{du} = \frac{2(1+\cos\alpha)}{V^2}(V-2u) = 0$$

$$\boxed{u = \frac{V}{2}}$$

Thus for maximum efficiency, speed of the vane should be half the velocity of jet, so the corresponding maximum efficiency

$$\eta_{max} = \frac{2u(2u-u)}{(2u)^2}(1+\cos\alpha) = \frac{1+\cos\alpha}{2}$$

Thus a maximum efficiency of 100% is attained when $\alpha = 0°$, i.e. when the vanes are of semicircular configuration.

4(b) velocity of jet $V_1 = k_v\sqrt{2gH} = 0.97\sqrt{2g \times 650}$

$V_1 = 109.6$ m/s, $u_1 = u_2 = u$, $V_{r1} = V_1 - u$

$V_{r2} = 0.85 V_{r1} = 0.85(109.6 - u)$ — (I)

$\beta_2 = 15°$, $V_{r2}.\cos\beta_2 = u$ — (II)

From equation (i)

$u = 0.85(109.6 - u)\cos 15°$

$u = 49.41$ m/s $= \frac{\pi D N}{60}$, $N = 786.8$ rpm

$\frac{u}{V_1} = \frac{49.41}{109.6} = 0.4508$

$Q = \frac{\pi}{4}d^2 \times V_1 = \frac{\pi}{4}d(0.1)^2 \times 109.6 = 0.86$ m³/s

available power $= \rho g Q H = 5455$ kW

power input to buckets $= \frac{1}{2}\rho Q V_1^2 = 5165$ kW

Power Developed $= \rho Q V_{w1} u_1 = 4657$ kW

6 (c)

6 (c) $u_1 = \frac{\pi D_1 N}{60} = R_1 \times \omega = 7.2\ m/s$

$u_2 = R_2 \times \omega = 14.4\ m/s$

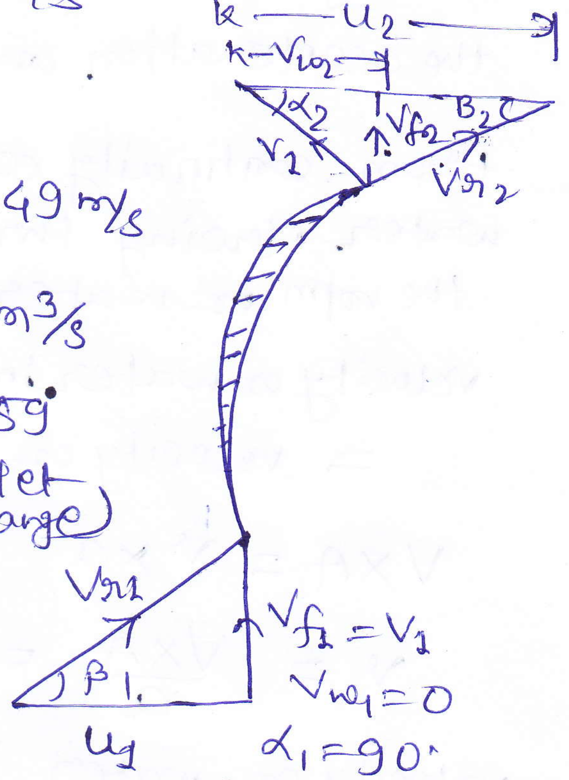
$\tan \beta_1 = \frac{V_{f1}}{u_1} \Rightarrow V_{f1} = 3.49\ m/s$

$Q = \pi D_1 B_1 V_{f1} = 0.087\ m^3/s$

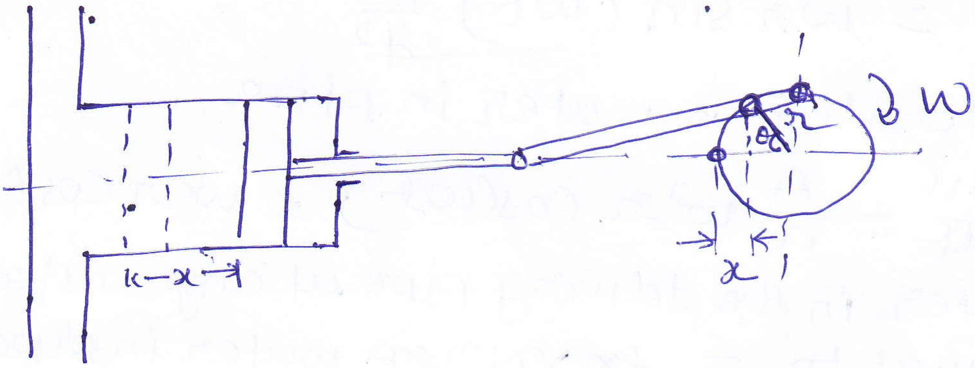
$H_m = \frac{V_{w2} u_2}{g}$, $V_{w2} = 7.59$ (from outlet velocity triangle)

$H_m = \frac{7.591}{9.81} = 11.14\ m$

$R_1 = 90mm$
$R_2 = 320mm$
$D_1 = 160mm$
$D_2 = 320mm$



7 (b)



Let the piston motion to be simple harmonic in nature

ω = angular speed of the crank in rad/s

A = area of the piston/cylinder, D dia

a = area of the pipe (suction or delivery)

l = length of the pipe (suction or delivery)

r = radius of the crank

The displacement of the piston in time 't' is 'x'

θ = angle turned by crank in radians in time 't'

$\theta = \omega t$

The horizontal distance moved by the piston is given by

$x = r - r\cos\theta = r - r\cos(\omega t) = r[1 - \cos(\omega t)]$

velocity of the piston $\dot{x} = \frac{dx}{dt} = \omega r \sin\theta = \omega r \sin(\omega t)$

the acceleration of the piston $\ddot{x} = \frac{d^2x}{dt^2} = \omega^2 r \cos(\omega t)$

from continuity equation, At any instant volume of water flowing through cylinder per sec is equal to the vol of water flowing from the pipe per sec.

velocity of water in cylinder × area of cylinder
= velocity of water in pipe × area of pipe

$$V \times A = v \times a$$

$$v = \frac{V \times A}{a} = \frac{A}{a} \times V$$

velocity of water in pipe $v = \frac{A}{a_d} \cdot \omega r \sin(\omega t)$

$$v = \omega r \sin(\omega t) \frac{D^2}{d^2}$$

and acceleration of water in pipe

$$\frac{du}{dt} = \frac{A}{a} \omega^2 r \cos(\omega t) = \omega^2 r \cos(\omega t) \frac{D^2}{d^2}$$

Mass of water in the delivery pipe at any instant of time is equal to = $\rho \times$ vol of water in pipe
$= \rho \times [a \times l]$

force required to accelerate the water in the pipe
= mass of water in the pipe × acceleration of water in the pipe

$$= \rho \times a \times l \times \frac{A}{a} \omega^2 r \cos(\omega t)$$

Intensity of pressure due to acceleration in the pipe
$= \frac{\text{Force required to accelerate the water}}{\text{area of the pipe}}$

$$= \frac{\rho a l \cdot \times \frac{A}{a} \omega^2 r \cos(\omega t)}{a}$$

$$= \rho \cdot l \frac{A}{a} \omega^2 r \cos(\omega t)$$

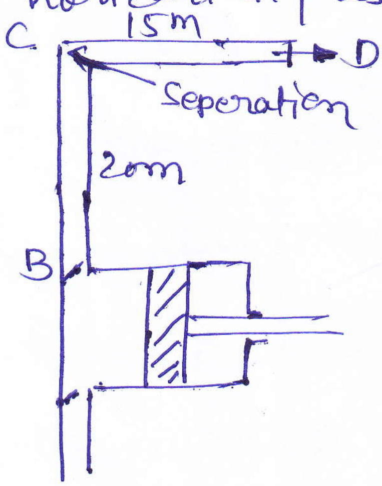
Pressure head due to acceleration in the delivery pipe

$h_a = \frac{\rho l \frac{A}{a} \omega^2 r \cos(\omega t)}{\rho g} = \frac{l}{g} \times \frac{A}{a} \omega^2 r \cos(\omega t)$

7(C) The separation on delivery side can occur only at the end of delivery stroke as the pressure head during delivery stroke is minimum. The acceleration head at the end of delivery stroke

$$h_{ad} = \frac{l_d}{g} \times \frac{A}{a_d} \times \omega^2 r = \frac{35}{9.81} \times \frac{\frac{\pi}{4}D^2}{\frac{\pi}{4}d_d^2} \times \omega^2 \times 0.20$$

1st Case:- the pipe rises first vertically and then horizontally as shown. The possibility of separation is at the point C at the end of delivery stroke.



Pressure head at B = $H_{atm} + h_d - h_{ad}$

—— " —— at C = $H_{atm} - h_{ad}$

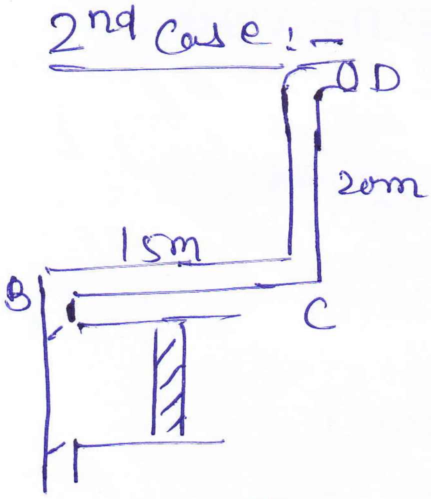
For separation to take place at C, then the pressure head at C is ~~2m~~ 2.5m

$2.5 = H_{atm} - h_{ad}$

$h_{ad} = 10.3 - 2.5 = 7.8$

$\omega = 1.85$ rad/s, $N = \frac{60 \times \omega}{2\pi} = 17.68$ rpm

2nd Case:-



The pressure head at B is equal to

$H_{atm} + h_d - h_{ad} = h_{sep}$

$2.5 = 10.3 + 20 - h_{ad}$

$\omega = 3.495$ rad/s

$N = \frac{60 \times \omega}{2\pi} = 33.37$ rpm.

6 (a)(i) $\frac{\sqrt{H_m}}{D_m N_m} = \frac{\sqrt{H_p}}{D_p N_p} \Rightarrow \sqrt{H_m} = \frac{D_m N_m}{D_p N_p}\sqrt{H_p} = 3.71$

$\boxed{H_m = 13.78m}$

(ii) $\frac{Q_m}{N_m D_m^3} = \frac{Q_p}{N_p D_p^3} \Rightarrow Q_m = Q_p\left(\frac{N_m}{N_p}\right)\cdot\left(\frac{D_m}{D_p}\right)^3$

$= 0.05 \times \left(\frac{5000}{1500}\right) \times \left(\frac{1}{4}\right)^3$

$\boxed{Q_m = 0.0026\ m^3/s}$

4 (c) **Difference between Impulse and Reaction turbine**

Impulse turbine	Reaction turbine
The entire available energy of the water is converted into kinetic energy.	Only a portion of the fluid energy is converted into kinetic energy before the fluid enters the turbine runner.
The work is done only by the change in the kinetic energy of the jet.	The work is done partly by the change in the velocity head, but almost entirely by the change in pressure head.
Flow regulation is possible without loss.	It is not possible to regulate the flow without loss.
Unit is installed above the tailrace.	Unit is entirely submerged in water below the tailrace.
Casing has no hydraulic function to perform, because the jet is unconfined and is at atmospheric pressure. Thus, casing serves only to prevent splashing of water.	Casing is absolutely necessary, because the pressure at inlet to the turbine is much higher than the pressure at outlet. Unit has to be sealed from atmospheric pressure.
It is not essential that the wheel should run full and air has free access to the buckets.	Water completely fills the vane passage.

6 (b)

19. CHARACTERISTIC CURVES OF CENTRIFUGAL PUMPS

Characteristic curves of centrifugal pumps are defined those curves which are plotted from the results a number of tests on the centrifugal pump. These curves are necessary to predict the behaviour and ...rmance of the pump when the pump is working under different flow rate, head and speed. The followings ...e important characteristic curves for pumps :

1. Main characteristic curves,
2. Operating characteristic curves, and
3. Constant efficiency or Muschel curves.

19.10.1. Main Characteristic Curves. ...main characteristic curves of a centrifugal ...p consists of variation of head (manometric ..., H_m), power and discharge with respect to ...d. For plotting curves of manometric head ...s speed, discharge is kept constant. For ...ing curves of discharge *versus* speed, ...ometric head (H_m) is kept constant. And for ...ing curves of power *versus* speed, the ...metric head and discharge are kept con... Fig. 19.14 shows main characteristic cur... a pump.

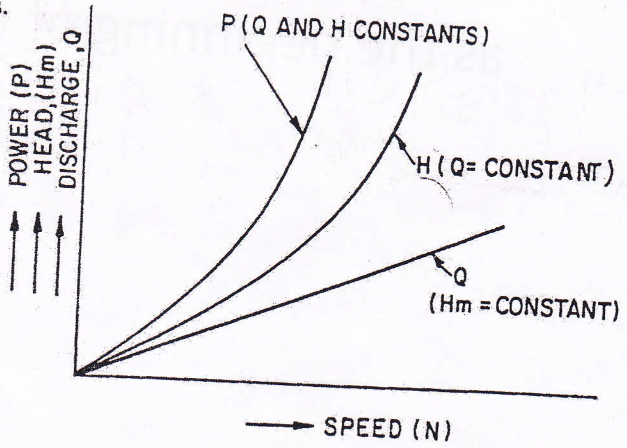


Fig. 19.14. Main characteristic curves of a pump.

19.10.2. Operating Characteristic Curves. If the speed is kept constant, the variation of [illegible] head, power and efficiency with respect to discharge gives the operating characteristics of the pu[illegible] shows the operating characteristic curves of a pump.

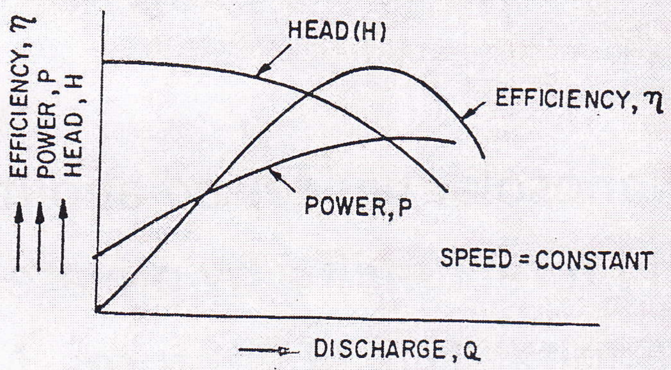


Fig. 19.15. Operating characteristic curves of a pump.

19.10.3. Constant Efficiency Curves. For obtaining constant efficiency curves for a p[illegible] *versus* discharge curves and efficiency *versus* discharge curves for different speeds are used. Fi[illegible] shows the head *versus* discharge curves for different speeds. The efficiency *versus* discharge c[illegible] different speeds are as shown in Fig. 19.16 (*b*). By combining these curves ($H \sim Q$ curves and $\eta -$ [illegible] constant efficiency curves are obtained as shown in Fig. 19.16 (*a*).

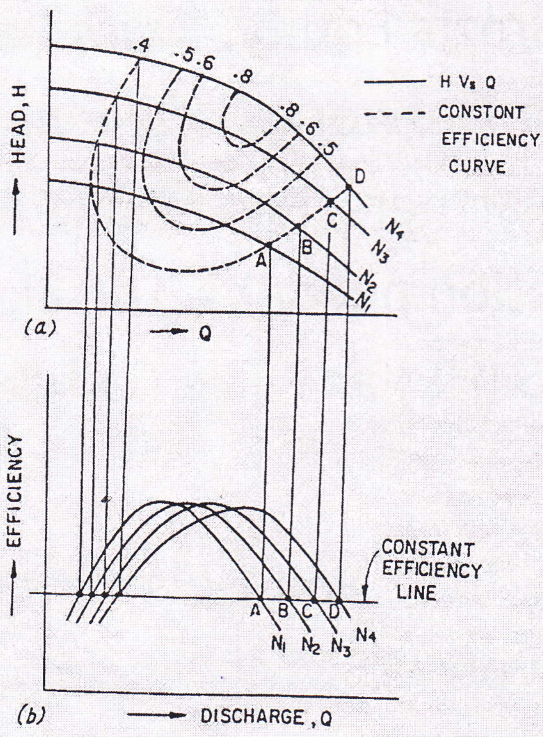


Fig. 19.16. Constant efficiency curves of a pump.

7 (a) Hydraulic Jack :-

This device is commonly used to lift the automobiles through a small height for romaving tyres when punctured. It also works on the same principle as that of hydraulic press.

When the plunger is operated by a ~~padel~~ handle, the liquid pressure on the ram is increased and ram moves upward moving the automobile up. The pressurised oil is passed to the ram chamber through the delivery valve D.

The ram is lowered to bring to its lower original position by unscrewing the lowering screwe C. When screw C is un screwed, the liquid from the ram chamber rushes to the main reservoir releasing the pressure acting on the ram.

The force F applied on the lever to lift the automobile is also given by the equation $W = F\left(\frac{A}{a}\right)^{2}$