FLUID MACHINERY LABORATORY INSTRUCTION MANUAL

Prepared by

Prof. Dr. Ömer T. GÖKSEL

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Published by the University of Gaziantep
Gaziantep - TÜRKİYE

ISBN : 975-7375-06-3

Printed at the printing
Department of the University of
Gaziantep - TURKEY
Tel : (00) (342) 3601200 / 1363

1. INTRODUCTION

In general fluid machinery may be classified under two headings:

1. Fluid machinery producing head or pressure, like pumps and compressors.

2. Fluid machinery producing power, like turbines.

The basic type of phenomenon occuring in any type of fluid machinery is the energy transfer between the fluid and the moving parts of the machinery in contact with the fluid. In basic analysis of such machinery one usually considers the idealized cases where mechanical and hydraulic losses are neglected.

Only the roto-dynamic type of fluid machinery will be considered in this manual. The actual characteristics of these machinery and the reasons of deviations from the ideal characteristics will be studied.

Since roto-dynamic hydraulic machinery are extensively used in industry and in practical everyday life, understanding of their operation and performance evaluation are very important for practicing engineers. The experiments that are discussed in this manual aim to teach the performance evaluation of these machinery under different design and operation conditions. These experiments are:

1. Performance evaluation and modes of operation of radial flow pumps and rules of similarity.

2. Effects of blade shape on the performance of a radial flow pump.

3. Effects of inlet flow conditions and cavitation on the performance of a radial flow pump.

4. Determination of performance characteristics of an axial flow pump.

5. Determination of performance characteristics of a Francis turbine.

6. Determination of performance characteristics of an axial flow

reaction turbine.

7. Determination of performance characteristics of an impulse turbine (Pelton Wheel).

2. THEORY AND PERFORMANCE OF RADIAL FLOW MACHINERY

2.1. Basic Centrifugal Pump Characteristics

The centrifugal pump consists basically of three components, an inlet duct, an impeller and volute.

The inlet duct conducts the fluid into the center of the impeller. Due to the impeller rotation the fluid is centrifuged towards the outside diameter of the impeller.

The high velocity of the fluid at the exit from the impeller is converted into pressure energy by velocity reduction in the volute which acts basically as fluid collection channel. The high pressure fluid then passes through the pump discharge to the system.

When the exit valve is closed there is no flow through the pump, the head developed tends to be the highest for a given speed. As the flow is increased developed head reduces.

It can be shown that head generated by a centrifugal pump is given by the expression:

$$H_i = \frac{1}{g} (U_2 C_{u2} - U_1 C_{u1}) \tag{1}$$

where,

H_i : Ideal head developed.

U_1: Impeller I. D. tangential velocity.

U_2: Impeller O. D. tangential velocity.

C_{u2}: Tangential component of fluid velocity at impeller outlet.

C_{u1}: Tangential component of fluid velocity at impeller inlet due to prerotation.

Equation (1) is known as the Euler's turbine equation, and is derived from the velocity triangles in fig.1.

If it is assumed that there is no prerotation of fluid at inlet, then $C_{u1}= 0$ and equation (1) reduces to:

$$H_i = \frac{1}{g} (U_2 C_{u2}) \quad (2)$$

By analysis of the exit velocity triangle, it can be shown that,

$$C_{u2} = U_2 - (C_{m2} / \tan \beta_2)$$

Where C_{m2} is the mean meridional through-flow velocity and β_2 is the discharge blade angle. The value of C_{m2} is proportional to flow rate and is equal to:

$$C_{m2} = (Q/(2\pi r_2 b_2 - n b_2 t))$$

where,

r_2 : Impeller radius at discharge.

b_2 : Blade height at discharge.

t : Blade thickness at discharge.

n : Number of blades on the impeller.

Therefore equation (2) can be rewritten as:

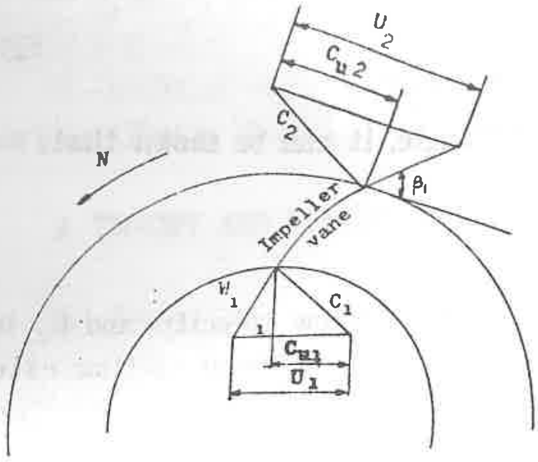
$$H_i = \frac{1}{g} U_2 (U_2 - KQ) \quad (3)$$

This relationship gives the H-Q characteristics shown in fig.1.

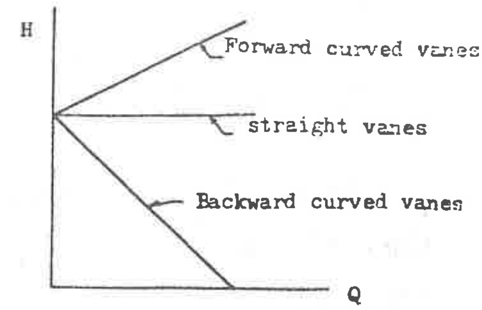
As was mentioned previously, the analysis used to produce fig.1, is wery idealized and the performance of the actual pump is modified from the Euler line by the following effects.

1. Prerotation
2. Entrance losses
3. Diffuser or volute losses
4. Off design entrance losses
5. Off design diffuser losses
6. Leakage losses

With regard to items (4) and (5) the impeller blading is designed for efficient use with a given flow quantity. At any other flow quantity, it is obvious that the blades will not be as efficient. Euler's turbine equation is founded upon certain ideal assumptions, namely, frictionless non-turbulent flow through an impeller with infinite



Theoretical flow pattern in impeller



Euler lines

Fig 1. Velocity triangles and Euler Line

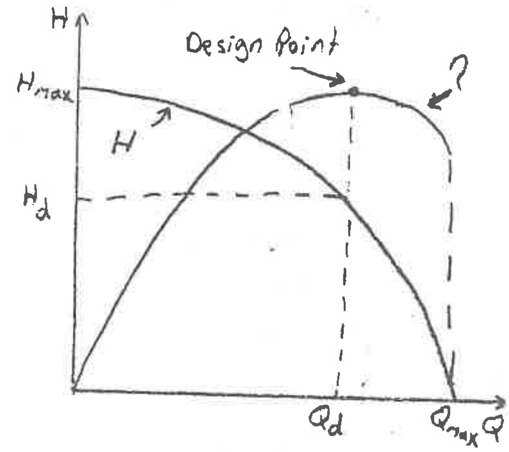


Fig 3. H-Q and η-Q curves

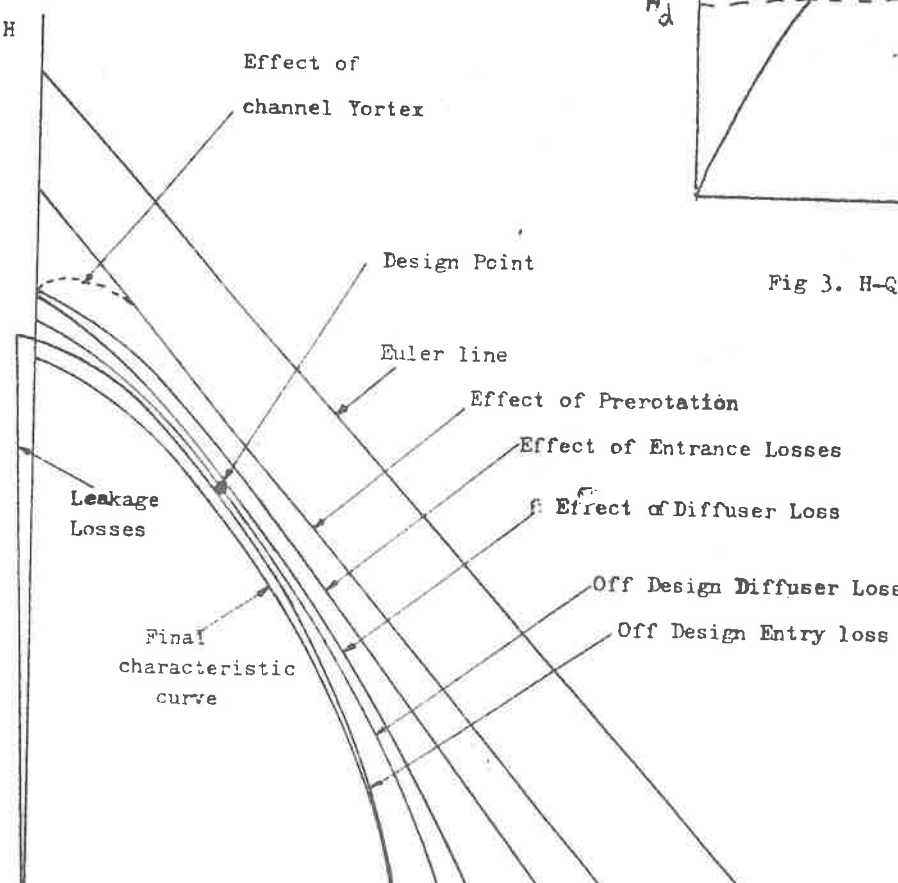


Figure 2. Centrifugal Pump Characteristic 4

number of vanes. The effect of all these factors are shown in fig.2.

Since the blade angles are designed for operation with a given flow quantity, the overall pump efficiency will be maximum at this point, as illustrated in fig.3. It will be seen from fig.3 that although the efficiency is zero at shut off head (i.e. when Q=0), the efficiency at maximum flow (i.e. when H=0) is not zero.

The power required to derive a pump is defined as:

$$P = 9.81 \frac{\rho Q H}{\eta} \tag{4}$$

where,

P = Power W (Watts)

Q = Flow rate (m^3/sec)

H = Total head in meters of fluid column.

ρ = Density of fluid (kg/m^3)

η = Efficiency

Therefore the maximum power required to derive the pump will occur as the flow rate aproaches maximum. The minumum power required will therefore occur at shut off head. This explains why it is necessary to start centrifugal pumps with the discharge valve closed.

2.2 Modes of Operation of Centrifugal Pumps

a) Paralell Operation:

Fig.5. shows the pump characteristic curves for two pumps A and B of different characteristics. With the two pumps in paralell operation the overall characteristic is shown as the curve 1-5-6. It will be noticed that the pump B does not contribute to the pumped quantity until the head has dropped to that at point 5 of pump B characteristic.

b) Series Operation:

Fig.5. shows the characteristic curves for pump A and for pump B. It is assumed that the discharge from pump A is fed into the inlet of pump B. The combined characteristic is shown by the characteristic curve 5-6-7. As is seen from fig.5, a change of characteristic occurs between the point 6 and 7. This is due to the fact that flow passing through pump B is greater than its normal maximum pumping quantity.

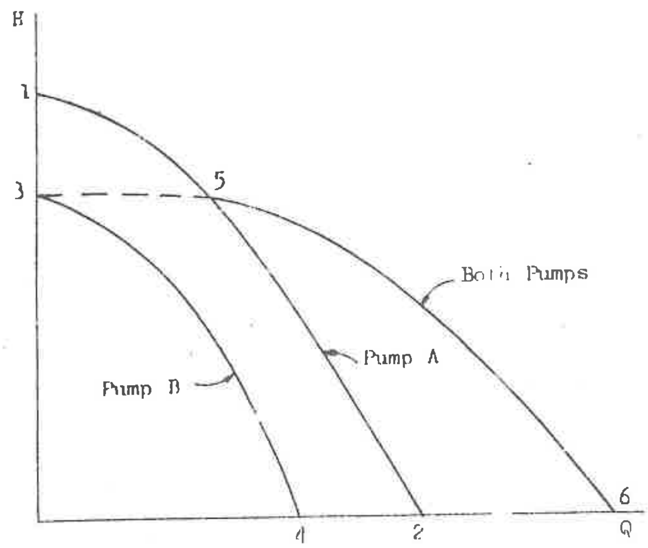


Fig 4. Parallel Operation

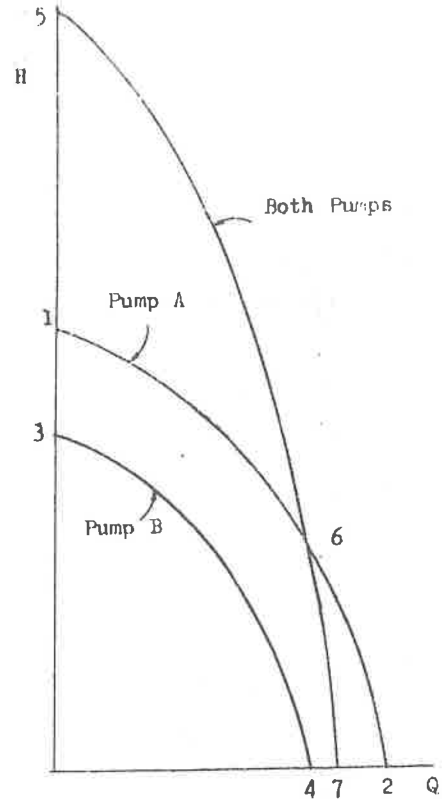


Fig.5. Series Operation

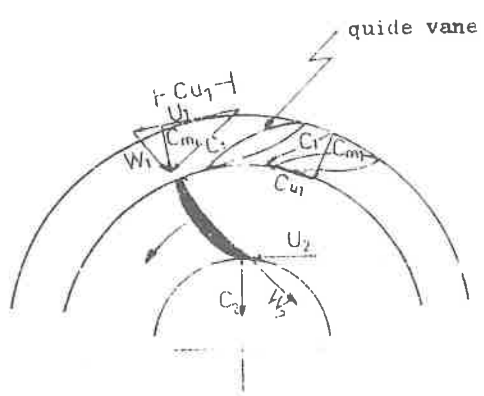


Fig: 6- Inlet and outlet velocity diagram of a Francis Turbine

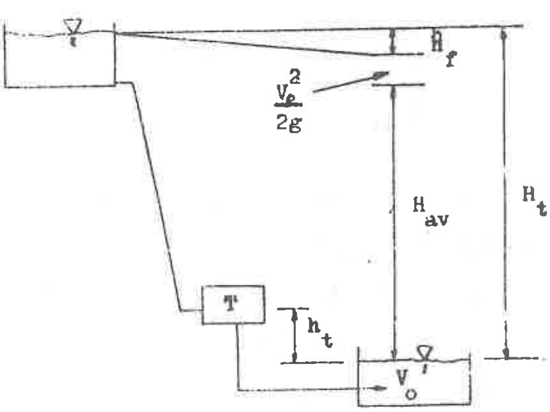


Fig:7-Line diagram of a reaction turbine plat

2.3 Basic Characteristics of Inward Flow Turbines

The principle of operation of the Francis type reaction turbines is similar to a centrifugal pump working under reversed flow conditions, namely the fluid entering the 'discharge' end and leaving at inlet. In centrifugal pumps a whirl is developped in the impeller and subsequently transformed into pressure, but in reaction turbines one potion of the head available is first transformed into a whirl by stationary guide vanes and is then converted into power inside the runner. The other portion of the head is used for the acceleration of the relative velocity of the water passing through the channels formed by the runner vanes.

Velocity diagrams for the low specific speed turbines of Francis type are similar in construction to those shown in the section on centrifugal pumps. Because the small whirl improves the efficiency of the draft tube, it was found in practice that best efficiency is obtained when a small whirl is retained at the exit. However assuming $C_{u2}=0$ the ideal power developed by the runner becomes (see figures 6 and 7):

$$H_{av}=\frac{U_1 C_{u1}}{g}$$

Introducing the hydraulic efficiency, η_h, the effective head is:

$$H_e = H_{av}\eta_h \tag{5}$$

Therefore actual whirl velocity at inlet is:

$$C_{u1}=\frac{H_{av} g \eta_h}{U_1} \tag{6}$$

As shown in fig.7. the available head can be calculated, considering the total difference in elevation, H_t minus the frictional losses in the ducting and velocity head lost in the tailrace at the point of discharge.

$$H_{av}=H_t - h_f - \frac{V_0^2}{2g} \tag{7}$$

The overall efficiency of the turbine may be written as:

$$\eta = \frac{\text{Brake Power}}{\text{Hydraulic Power}} = \frac{P_s}{P_h}$$

where $P_h = \rho gQH$ and $P_s = T2\pi N/60$. If H is in meters, Q is in m^3/sec., ρ is in kg/m^3, T in Newton-meters, and N is in RPM, then units of P_s, and P_h are Watts.

In inward flow radial reaction turbines, the operating characteristics can be expressed as a function of N, H_{av}, P_s, Q, η, and the guide vane angle α. In general it is possible to draw two sets of characteristics. In the first set N, H_{av} and in the second set α, H_{av} are kept constant, and in this way the whole operating region of the turbine can be investigated. Since the first set is related to the second set, it is sufficient to plot the second set of characteristics to determine the overall characteristics of the turbine.

Typical characteristic curves are shown in fig.8, 9, and 10. From the above explanation it is apparent that for any value of H_{av} it is possible to draw a new set of curves, therefore it is possible to draw infinite number of curves for determining the characteristicsof the turbine within the operating range.

However by means of similarity analysis all results of the tests at different H_{av} values are reduced to standart values corresponding to a constant standart head, H_{st} of 1 meter. The characteristic curves in fig.8, 9 and 10 show plot of results for standart Q, P_s, and N for constant standart head of 1 meter; therefore:

$$Q_{st} = Q(H_{st}/H)^{0.5}$$

$$N_{st} = N(H_{st}/H)^{0.5}$$

$$P_{st} = P(H_{st}/H)(H_{st}/H)^{0.5}$$

Refering to curve 8 first a series of η-N plots are made from run away speed to zero speed (stalled speed) each for a constant value of α. From this characteristic it is easy to determine the speed

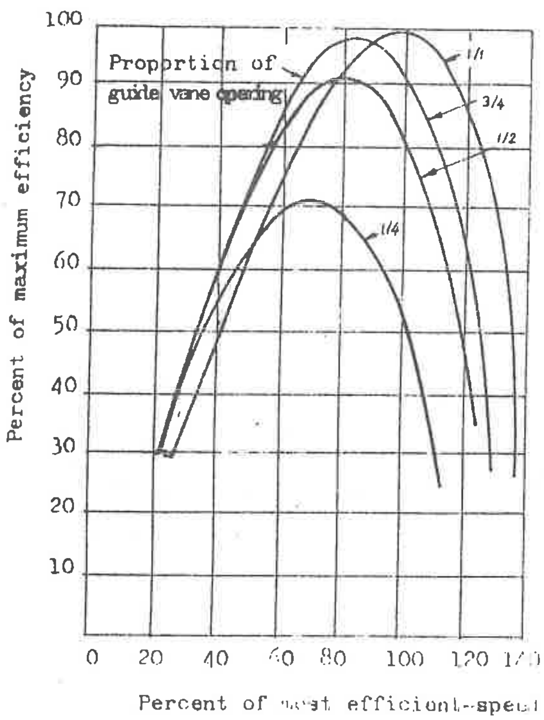


Fig 8. Efficiency-speed at different guide vane openings

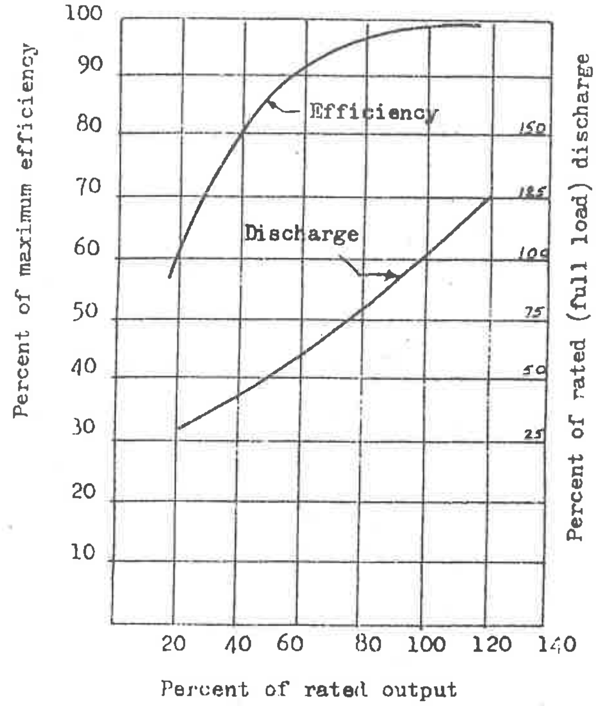


Fig 9. Output - Efficiency and Output-Discharge Curves

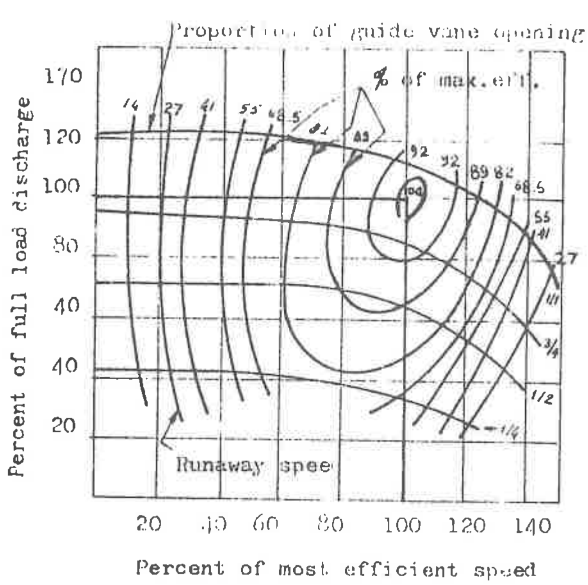


Fig 10. Iso-efficiency or General Characteristic Curve

(based on Discharge)

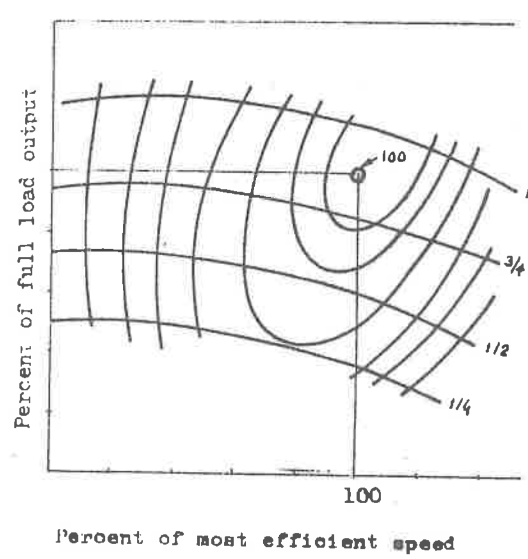


Fig 11. Iso - efficiency Curve

(based on Output)

coresponding to maximum efficiency for every value of α. For this determined value of N for each value of α, it is possible to plot P_s vs η or Q vs η.

The η-P_s or Q-P_s curves shown in fig.9 are those of most interest to the purchaser of the turbine. The purchaser will want to know how much power he is going to get for a given discharge of water.

Finally full characteristic or iso-efficiency curves shown in fig.10 can be plotted. These curves are essentially the tool of the designer and manufacturer and are seldom of much use to the purchaser.

At each guide vane angle, plotting a line of N against Q and writting down the value of η on each point plotted, fig.10 can be constructed by joining the contours of points of equal efficiency using reasonable judgement.

Similarly the iso-efficiency curve for P_s vs N can also be drawn as shown in fig.11.

The Q vs N curves shown in fig.10 drops for low specific speed turbines, is nearly horizontal for medium specific speeds and rise for propeller and Kaplan turbines.

3.THEORY AND PERFORMANCE EVALUATION OF AXIAL FLOW MACHINERY

3.1. Blade Terminology and Sign Conventions:

a) Blade Nomenclature:

The blade nomenclature is clearly shown in fig.12 and 13. The definition of symbols for this figure is as follows:

α_1 : Inlet flow angle

α_2 : Outlet flow angle

β_1 : Inlet blade angle

β_2 : Outlet blade angle

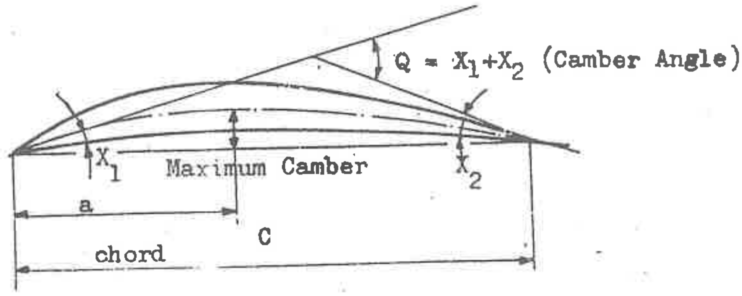
θ : Camber angle ($X_1 + X_2$)

i : Angle of incidence ($\alpha_1 - \beta_1$)

δ : Deviation angle ($\alpha_2 - \beta_2$)

Υ : Stagger angle

All angles are measured relative to axial direction with the exception of incidence and deviation angles which are measured relative to the line tangent to the camber.



Definitions relating to camber

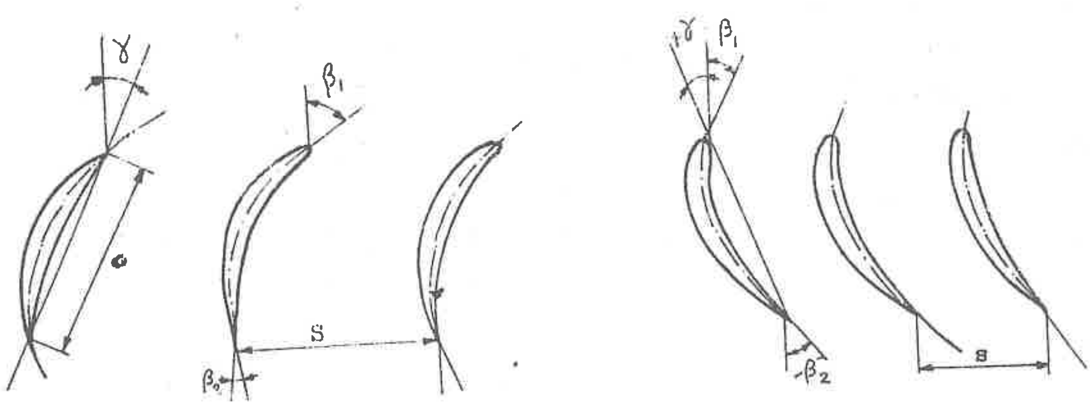


Fig.12 - Definitions for a cascade of blades

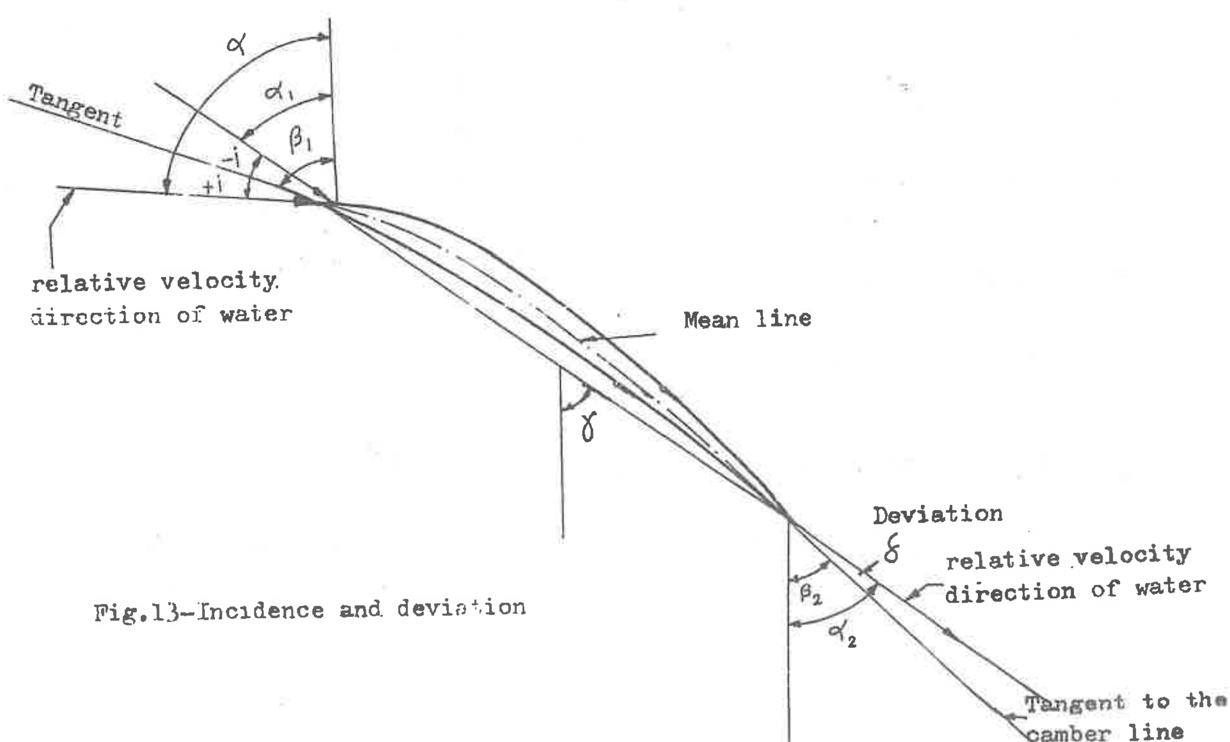


Fig.13-Incidence and deviation

b) Sign Convention for Angles:

All angles with the exception of i and δ measured in the clockwise direction from the axial direction have negative (-) and if measured in the opposite sense have positive (+) signs. The signs of incidence and deviation angles are deduced from their calculation. For incidence and deviation angles measured clockwise from line tangent to camber shows positive direction, and counter clockwise direction shows negative direction.

c) Definitions Related to Camber and Setting of a Row of Blades:

Chamber line may be a circular or parabolic arc or combinations of circles of parabolas. The axial flow set up in the laboratory has blades with circular arc profiles where $X_1=X_2$ and $e/2=X_1=X_2$.

The setting of a row of blades may be expressed by the stagger angle γ. The spacing or pitch S of the blades is the distance between corresponding points of adjaccent blades and is expressed by the pitch chord ratio S/c. When the blades are evenly spaced around the rotor, the pitch is the circumference at any radius divided by the number of blades.

d) Effect of Rotor Stagger and Camber:

As the blades are set more in the axial direction, degree of reaction will tend to zero and decreasing the stagger angle causes the head and the flow to increase; however the working range is reduced and the efficiency tend to be lower.

By using rotor blades of different cambers but with the same stagger, the performance is increased as the camber is made greater, but the effect is less than expected because of the increased deviation of the flow from blades. Thus an increase of 10° in chamber does not produce 10° of extra turning of the flow. Therefore the total fluid deflection is:

$$\varepsilon = \alpha_1 - \alpha_2 = (\beta_1 \pm i) - (\beta_2 \pm \delta)$$

The Euler's turbine equation for axial flow machinery is:

$$H = \frac{UC_a}{g} (\tan \alpha_1 - \tan \alpha_2)$$

As is seen from the above equation, there is infinite value of α_1 and α_2 which will give a fixed value of H. Thus a blade of given camber will give values of H dependent on the setting of the blade with respect to the reference direction, that is the blade stagger. Not only the energy transfer but also loss and drag coefficients vary with stagger, because the setting of a given group of blades affects the relative amount of diffusion or acceleration of fluid between the blades. For fixed stagger angle losses increase with increasing camber or with increasing fluid turning angle.

3.2. Performance Characteristics of Axial Flow Pumps:

The axial velocity in an axial flow machinery may be taken as constant.As can be seen from the lecture notes:

$$E = \frac{U}{g} (C_{u2} - C_{u1})$$

or $$H = \frac{U}{g} (U - C_a \tan \alpha_2 - U + C_a \tan \alpha_1)$$

$$H = \frac{UC_a}{g} (\tan \alpha_1 - \tan \alpha_2)$$

and the degree of reaction may be expressed as:

$$R = \frac{C_a}{2U} (\tan \alpha_1 + \tan \alpha_2)$$

Typical pump and turbine blading is shown in fig.15. As it is seen from fig.15 when the flowrate is reduced by discharge throttling the head rise suddenly decreases and keeps on decreasing untill the flow is nearly zero. The efficiency has maximum value at flow rate just above that at stall.

The stalling occurs when the incidence of the flow on to the rotor blades exceeds a certain critical value and the flow become seperated from the back of the blade. The blade no longer provide lift. Running in the stalled region may lead to blades breaking and due to large fluctuating forces on the blades near stall pump operation is unstable. The stable region of the charecteristic is shown in fig.15 and practice axial pumps should only be operated in this region.

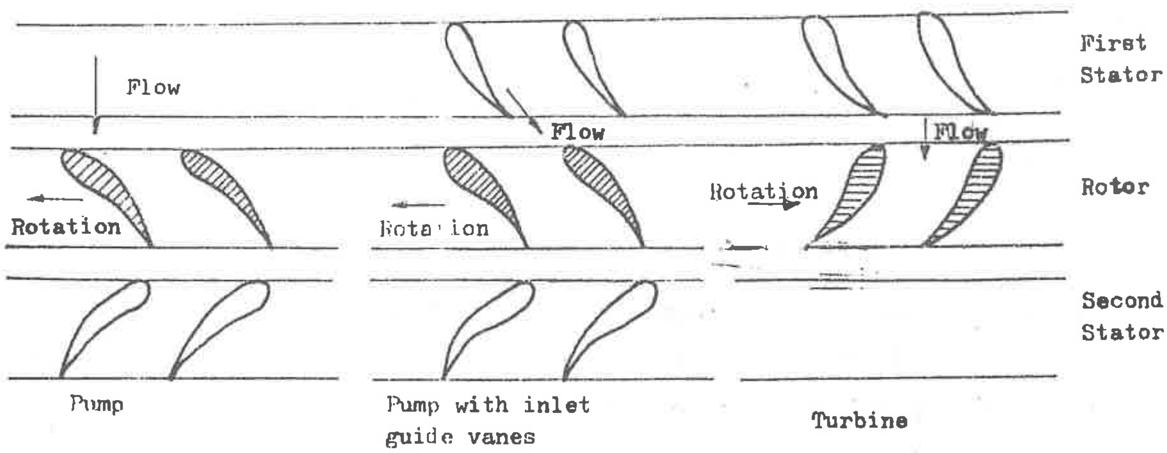


Fig 14. Typical blading of axial Flow machinary

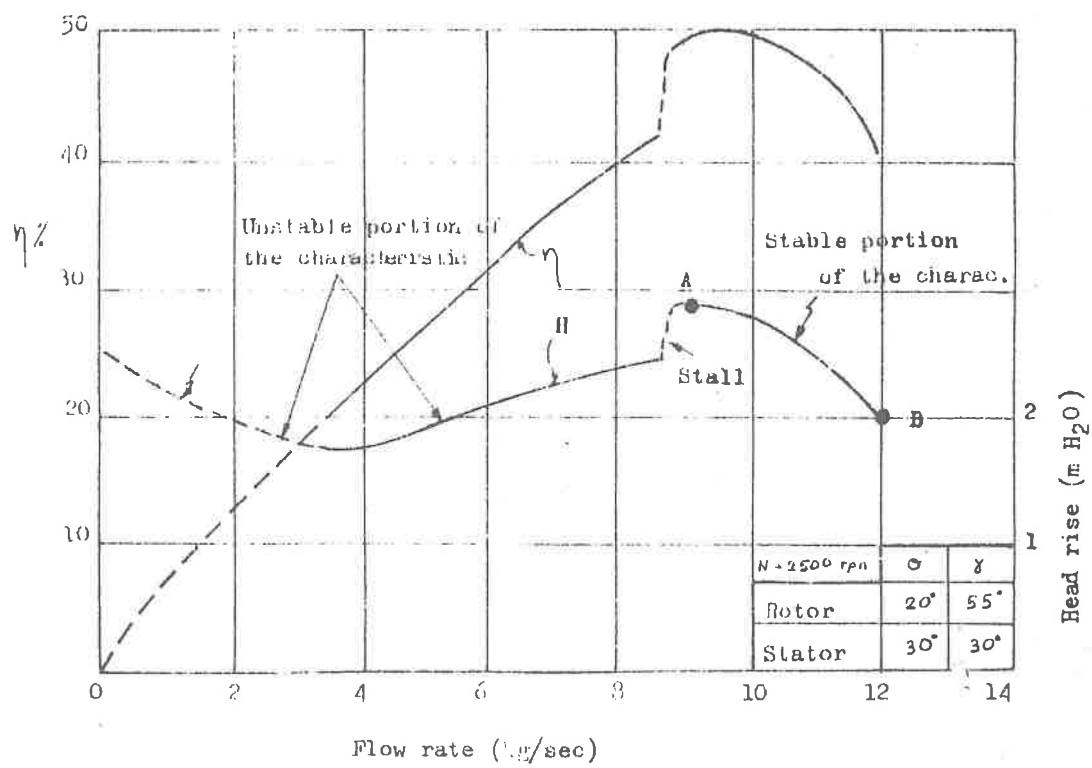


Fig 15. Axial flow pump characteristics of constant speed

It will be seen that the isolated rotor produces a poor head rise with low efficiency because of high outlet swirl. Therefore normal pump configuration is a rotor followed by a stator which removes the swirl and provides an axial exit for the flow.

The performance may be changed considerably by swirling the inlet flow by means of a row of inletguide vanes. When the swirl is in the same direction as the rotor rotation, the head-flow characteristicis moved down as though the speed had been reduced. In addition the efficiency is reduced due to the reduced performance of the stator which is subjected to increased incidence. With swirl in the opposite direction to the rotor motion, the characteristic is moved up as though the speed had increased. The efficiency however is essentially is unchanged. In this case the stator incidence is decreased and the performance is maintained.

Due to the blade blokage and presence of boundary layers and non uniform velocity distribution in the flow passeges the true velocity at the mid-blade height is greater than the mean value based on flow rate and area by about 10%.

If the root diameter of the blade is D and the tip diameter is D , the flow area becomes:

$$A = \frac{(D_o^2 - D_h^2)\pi}{4}$$

mean blading diameter is defined as:

$$D_m = \frac{(D_o + D_h)}{2}$$

Therefore for a rotational speed of N mean tangential blade speed, U becomes:

$$U = \frac{\pi D_m N}{60}$$

3.3. Performance Characteristics of Axial Flow Turbines:

a) Reaction Turbines

The performance evaluation of these turbines is exactly the same as was explained for the Francis turbine with only an other variable, namely the blade stagger of the rotor. Therefore to obtain the complete characteristics of an axial flow turbine measurements described for the radial flow turbine must be repeated for different values of stagger giving values of reaction between zero and unity. As was explained in class the prefered value of reaction should be about 50 %.

The governing equations for these type of turbine is the same as that for axial flow pumps. It should be noted that the speed for maximum output power and efficiency is about half the unloaded or 'run-away' speed. This is typical of a wide range of turbines. It is a consideration which must be born in mind when stressing any turbine which may be accidentally run unloaded.

b) Impulse Turbine (The Pelton Wheel):

The Pelton turbine is used in very high head installations, and developes efficiencies very close to those of the Francis and Kaplan types. In fig. 16 a Pelton turbine installation diagram is shown. The gowerning equations for the Pelton turbine are derived with reference to fig. 17. Euler's Turbine equation for these turbine may be written as:

$$E = H = \frac{U}{g}(C_{u1} - C_{u2})$$

from fig. 17

$$C_{u1} = C_1 = U_1 + W_1$$

$$C_{u2} = (U - W_2 \cos\alpha_2)$$

Therefore for ideal flow $W_1 = W_2$

$$H = \frac{UW_1}{g}(1+\cos\alpha_2) = \frac{U}{g}(C_1 - U)(1+\cos\alpha_2)$$

From above equation and the definition of utilization for turbines, utilazation factor for the impulse turbine becomes maximum when $U/C_1 = 0.5$. This value is obtained for a zero angle turbine;

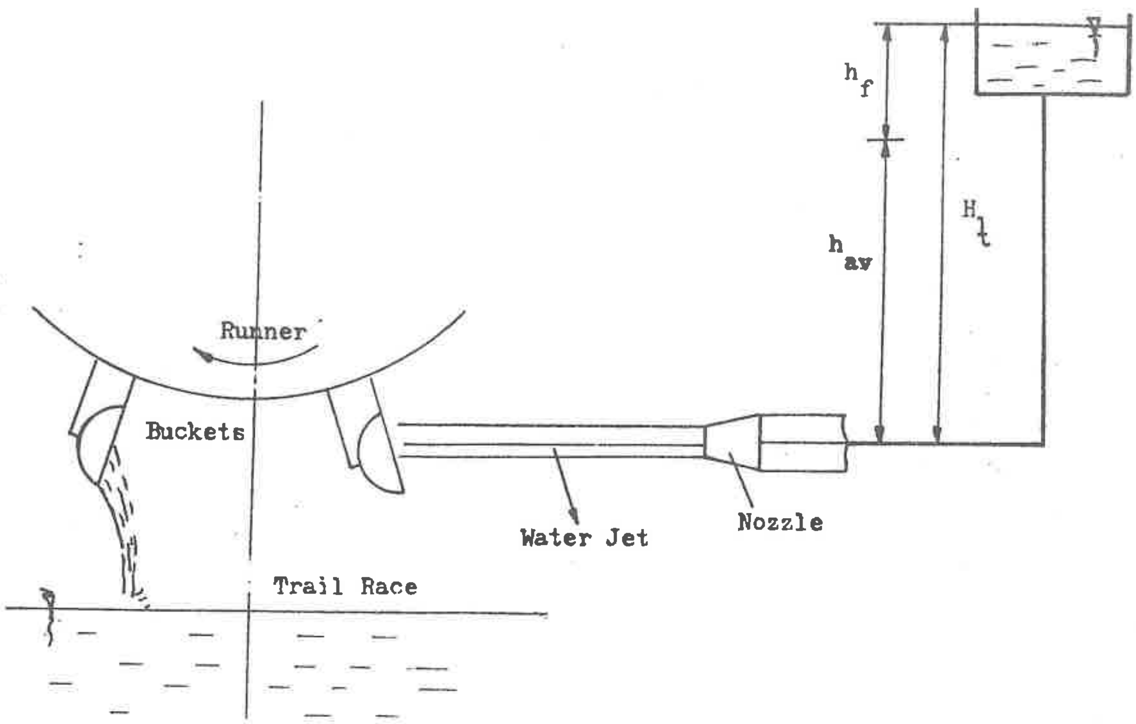


Fig 16. Elements of a Pelton Wheel

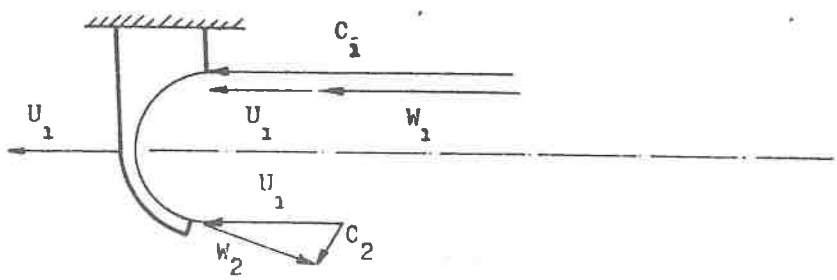


Fig 17. Velocity Diagram for a pelton wheel bucket

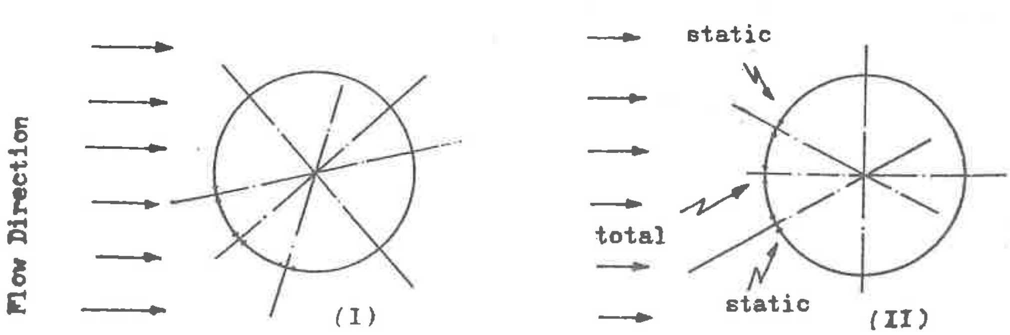


Fig 18. Yawmeter

however in practice α_2 is made about 15° to prevent interference of splashing water with buckets. It is found that for actual operating conditions U/C_1 is about 0.47 for maximum utilization.

The utilization factor, is:

$$\varepsilon = \frac{E}{C_1^2 / 2g} = 2\,(1-\cos\alpha_2)\,(U/C_1)\,(1 - U/C_1)$$

The flow rate is regulated by means of the spear valve. Instead of the guide vane angle of the Francis turbine percent opening of the spear valve is placed and the performance characteristics are obtained exactly the same way as that for a Francis turbine. However it must be noted that for the Pelton turbine H is:

$$H_{av} = H_1 - h_f$$

where H_1 is the total head at the spear valve centerline.

4. DETERMINATION OF THE MAGNITUDE AND DIRECTION OF ACTUAL VELOCITY:

4.1. The Yaw Meter:

The yaw meter shown in fig. 18, is used to measure velocity and direction of flow at the required point. The flow direction is determined by rotating the yaw tube until the levels of manometer fluid in the two static pressure measuring manometer tubes are the same. In this position the dynamic head and thus the velocity is given by the difference between the total and static pressure readings. The direction of velocity is measured by a protractor, attached to the yaw meter tube, with respect to a reference direction.

If h_v : Dynamic head indicated by yaw meter

q : True local dynamic head

v : Local velocity

Then

$$v = K(2gh_v)^{0.5} \quad \text{and} \quad q/h_v = K^2$$

where K is a constant for the yaw meter determined by calibration. For the yaw meters in our laboratory K=0.99.

5. SIMILARITY ANALYSIS

5.1. Dimensionless Parameters

In fluid machinery design and application certain dimensionless parameters are used. The flow, head parameters and the specific speed can be expressed as:

Flow parameter : $\phi = Q/ND^3$

Head parameter : $\psi = gH/N^2D^2$

Specific speed (kinematic) : $N_{sq} = NQ^{0.5}/H^{0.75}$

Specific speed (dynamic) : $N_{sp} = NP_s^{0.5}/H^{1.25}$

Kinematic specific speed is used for pumps and for turbines dynamic specific speed is used. The following table lists the proportionalities for scaling one pump size to another and scaling of pump charecteristic to another speed.

Constant	Q	H	Power	Comments
N	D^3	D^2	D^5	scaling of one pump size to another
D	N	N^2	N^3	scaling of pump characteristic to another speed

6. CAVITATION

6.1. Net Positive Suction Head:

Here, we will only consider the cavitation in centrifugal pumps. Cavitation occurs when the static pressure of the fluid at inlet is approximately equal to the vapor pressure of the fluid. Essentially cavitation is local vaporization of the fluid due to the dynamic conditions. Saturation conditions usually occur at room temperature and low pressure. Therefore cavitation is likely to occur where the fluid velocity is high and the general pressure level is low, such as inlet side of a pump and discharge side turbine. The actual mechanisms of cavitation is still in argument. The theory generally accepted now is that a liquid contains a number of small, undisolved gas nuclei, non-wetting solid particles, such as dust, and cavitation start first by the appearance or the liberation of this gas nuclei in the flow. Although a homogenous nuclei free liquid, starts to evaporate only at saturation temperature and pressure, presence of nuclei initiates cavitation much earlier. A bubble of vapor formed by local reduction of pressure eventually collapses when carried a region of high pressure. This collapse is sudden, leading to a pressure wave which is trasmitted to the material surface and may cause pressure up to 300 atmospheres.

6.2. Signs of Cavitation:

Cavitation occur with one or several of the following signs all of which adversely effects the pump performance and may physically damage the pump.

a) Noise and Vibration: This is caused by the sudden collapse of vapor bubbles as soon as they reach the high pressure zones within the pump. The bigger the pump, the greater is the noise and vibration. Although noise and vibration are cavitation symptoms they are present in all pumps to a varying degree when they are operated at points far away from best efficiency point, because of bad angle attack at the entrance to the impeller, and this must not be mixed with cavitation. To reduce cavitation shock and vibration, small amounts of air is admitted through the suction eye. In this way air serves as a cushion when the vapor bubbles collapse. This completly eliminates noise and vibration, and also prevents impeller vane pitting to a great extent.

Some large hydraulic equipment such as valves or pumps are generally designed such that automaticaly let air in at high throtling ratios.

b) Impeller vane pitting and corrosion fatique failure of metals:

Vapor or gas pockets exert great forces on the surfaces where they collapse, and causes erosion and crack propagatio. With time, errosion causes pittings on the impellers or on other surfaces. In addition metal distruction occurs by the fatique of the metal surface as aresult of repeated hammering effect of collapsing bubbles. Wear speeds up with the liquid penetration into and escapting from the pores of the metal under successive pressure waves, and causing the metal particles to be torn off from the surface. It can be said that more porous materials are more effected by such distruction.

Cavitation pitting should be distinguished from corrosion and errosion. The first is completely a chemical process, and is dependent on the liquid pumped. Errosion is the removal of materials from the surfaces by the abrasion of foreign particles carried by the pumped liquid such as sand, coke, coal, and some sediments.

c) Drop in Head-Flow and Efficiency-Flow charecteristic: When cavitation starts sharp drop of head and efficiency occur at constant flow rate as shown in fig.19. After this point it is impossible to increase the flow rate.

Net Positive Suction Head (NPSH): For any centrifugal pump net positive suction head required means: "The total suction head in meter of liquid (absolute) determined at pump inlet and corrected to datum at sea level, minus the vapor pressure of liquid in meters (absolute)."
Thus if:

$(NPSH)_r$: Net Positive Suction Head Required

H_a : Absolute pressure prevailing at the surface of liquid in pump suction reservoir. This will be atmospheric pressore if suction reservoir is open to atmosphere.

H_s : Total suction head in meters, positive or negative and including $V^2/2g$ at the suction.

H_g : Gage pressure reading at inlet in meters of liquid.
Therefore,

$$H_s = H_g + V^2/2g$$

If H is the absolute vapor pressure of the liquid in meters, then:

$$(NPSH)_r = H_a + (H_g + V^2/2g) - H_v$$

Again if: $(NPSH)_a$, the net positive suction head available, which

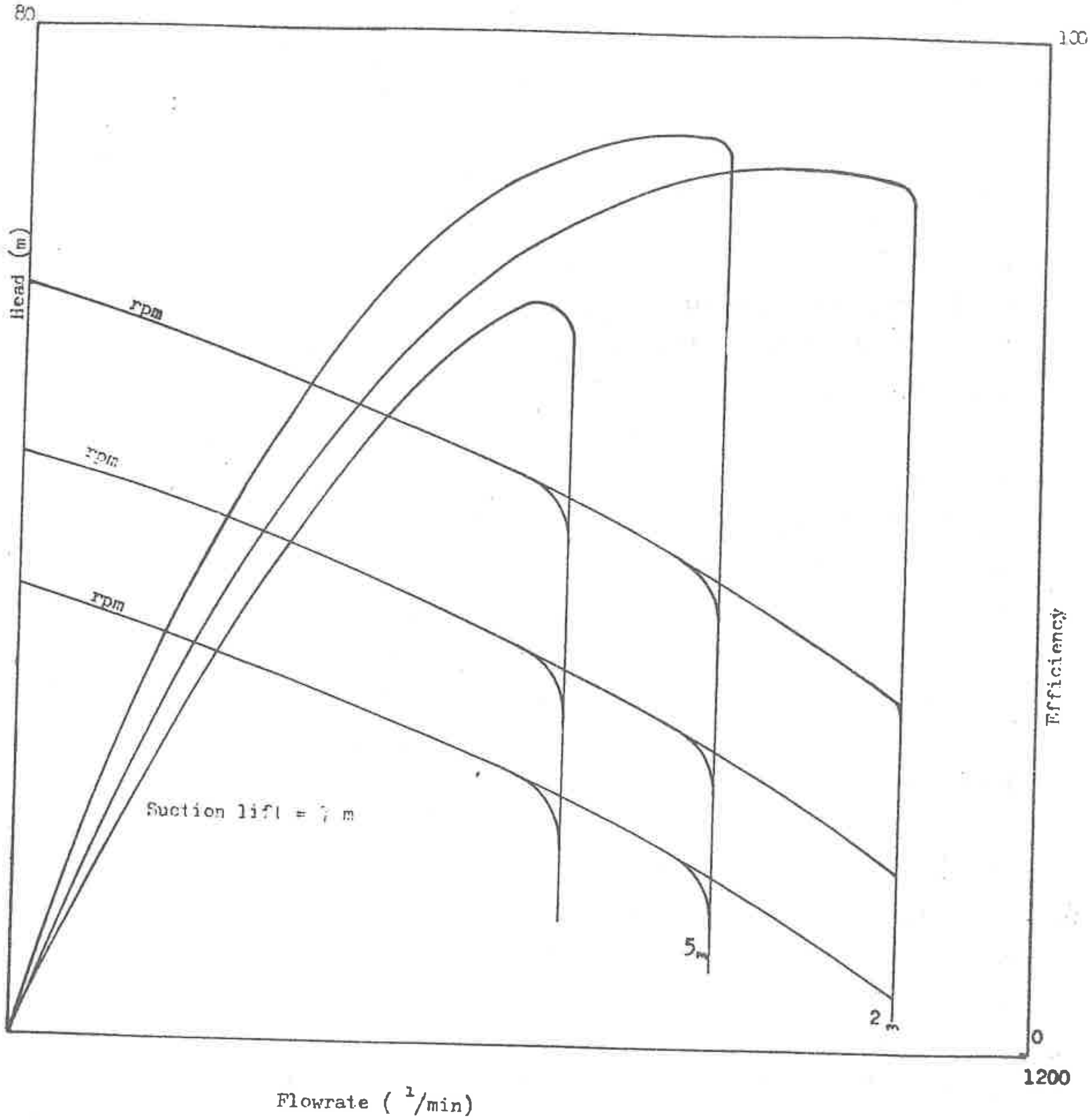


Fig. 19 - Effect of speed and suction lift on cavitation in a centrifugal pump ($n_s = 15$)

is the total suction head under working condition minus the absolute vapour pressure as in the $(NPSH)_r$.

H_{sa}=Total suction pressure "available" which again can be positive or negative and must include $V^2/2g$ at the suction

$$(NPSH)_a = H_a + H_{sa} - H_v$$

but

$$H_{sa} = H_s + V^2/2g$$

where H_s is the suction elevation head which may be positive or negative. Therefore

$$(NPSH)_a = H_a + (H_s + V^2/2g) - H_v$$

For a pump to operate safely and be free from cavitation $(NPSH)_a$ must be greater than the $(NPSH)_r$ required by the pump.

There are many experimental data available on NPSH and some empirical relations derived from them; among them Thoma takes NPSH as a certain fraction of total head H and states that at certain critical value of this fraction cavitation starts. Therefore,

$$NPSH = \sigma H$$

where σ is Thoma's cavitation constant and is a function of specific speed.

$$\sigma = \frac{6.3(N_{sq})_{us}^{1.33}}{10^6} \quad \text{for single suction pumps} = \frac{1194.4}{10^6} N_s^{1.33}$$

$$\sigma = \frac{4.0(N_{sq})_{us}^{1.33}}{10^6} \quad \text{for double suction pumps} = \frac{758.4}{10^6} N_s^{1.33}$$

Suction condition of pumps in respect to cavitation can be expressed by another criterion known as suction specific speed, S.

$$S = \frac{RPM\ (GPM)^{0.5}}{(h)^{0.75}} = (N_{sq})_{us}/(\sigma)^{0.75}$$

where h is the dynamic head at inlet including relative velocity effect. For safe operation, free from cavitation, the practical values of σ is between 0.17 and 0.09. For all practical purposes NPSH value of 1 meter of water is reasonable for small and medium sized pumps. A typical cavitation pump characteristic is shown in fig. 20.

$(N_{sp})_{us} = 0.0159\,(N_{sq})_{us}$

$(N_{sq})_{us} = 14.15\,N_{sp} = 51.6\,N_{sq}$

$(N_{sp})_{us} = 0.225\,N_{sp} = 0.822\,N_{sq}$

N_{sp} & N_{sq} $\Rightarrow Q\ m^3/s,\ h(m),\ \&\ P\ (H.P.),\ N$ rpm.

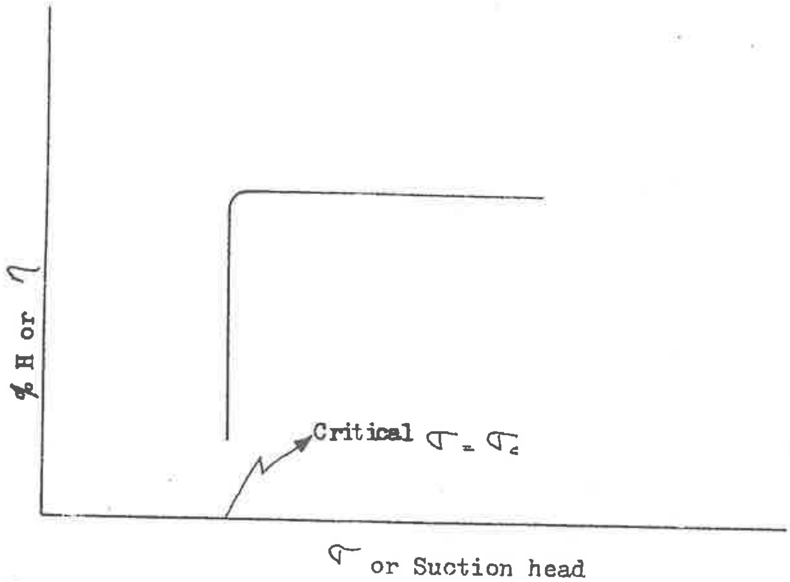


Fig 20. Cavitation characteristic of a centrifugal pump and determination of σ_c

EXPERIMENT NO : 1

EFFECTS OF BLADE SHAPE ON THE PERFORMANCE OF A RADIAL FLOW PUMP.

OBJECT :

The object of this experiment is to investigate the priciple of energy transfer between the impeller and the fluid, and the effect of the blade shape on the performance characteristics of the pump.

DESCRIPTION OF THE SET UP :

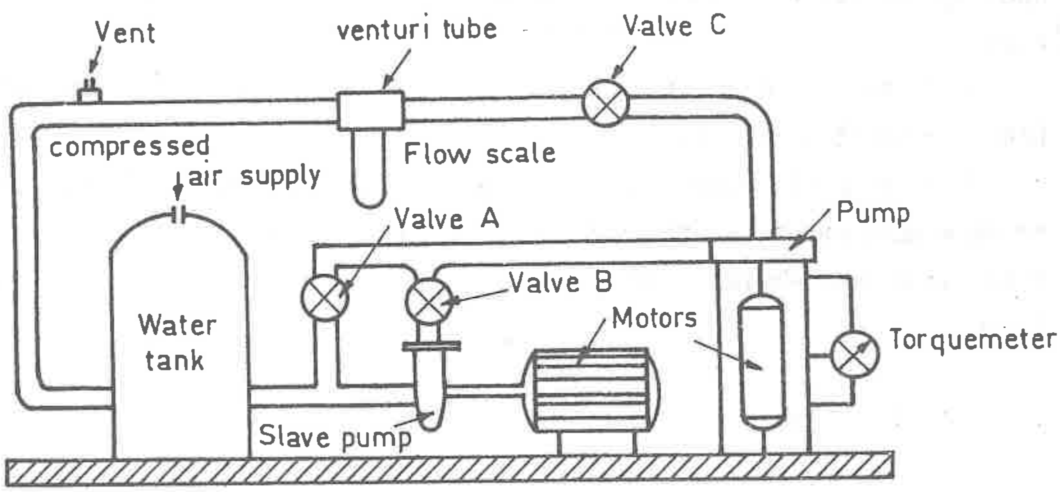


Fig. 1. Schematic diagram of the set up

The flowrate is measured by means of a venturi meter placed at the top of the set up. A mercury manometer is connected to the venturi meter which directly gives the flowrate on a calibrated scale. Pressures at four differnt points on the impeller under investigation and at the pump discharge line can be measured with respect to the inlet pressure by a 'U' tube mercury manometer. For yaw meter measurements there is a water manometer at the side of the control panel.

PROCEDURE :

1. Close valve B, open valve A and C completely. Apply 0.4 atm. pressurized air on the water tank until the system is full of water and water issues out from the vent.

2. Set the selector switch to pump operation and adjust the speed.

3. Make sure that you have closed valve A before starting the pump.

4. Read torque, flow rate and pressure head at inlet and across the pump, namely P_{ef}, and $(P_5 - P_{ef})$.

5. Insert the yaw meter into position 3, note the angular position of the yaw meter, and dynamic head for C_2

6. Repeat steps 3 and 4 seven more times each at different openings of valve A. Last reading should be taken when valve A is full open.

7. Stop the pump, relive the pressure and insert the yaw meter into the inlet of the impeller.

8. Adjust the speed to the value in 2, and take at least 5 yaw angle measurements starting from shut off and the last reading at full flow condition. Record only yaw angle and corresponding dynamic head for C_1.

CALCULATIONS :

1. Calculate the head developed by the pump.

2. Calculate hydraulic and mechanical power (P_h and P_s)

3. Calculate efficiency.

4. Calculate N_{sq}.

5. Using above calculations and data taken plot for each of the three different types of impeller H-Q, η-Q, P_s-Q, α_1-Q, α_2-Q on the same graph paper.

6. For each of the different impellers at maximum efficiency point, determine C_1 and C_2 actual from data. construct actual velocity polygons for maximum efficiency point.

7. Calculate $C_{m1}$and $C_{m2}$from data and draw theoretical velocity

polygons at maximum efficiency point.

QUESTIONS :

1. Why does the actual characteristics deviate from the Euler line?

2. What are the basic differences in the actual characteristics of the three impellers tested ? Explain by refering to graphs plotted and actual and theoretical velocity polygons, and specific speeds.

3. Why does α_1 and α_2 vary with Q. What can you say about the value α_1 at design point ?

4. Why is backward curved blades are prefered in practice to radial blades although radial blades give a higher head for a given Q ?

EXPERIMENT NO : 2

RULES OF SIMILARITY AND PARALLEL/SERIES OPERATION OF RADIAL FLOW PUMPS.

OBJECT :

The object of this experiment is to investigate the dynamic and kinematic similarity of two geometrically similar pumps, and to determine the operational characteristics using similarity rules and compare them with those experimentally determined characteristics. In adition it is aimed to investigate the performance characteristics of parallel and series operation of radial flow pumps.

DESCRIPTION OF EXPERIMENTAL SET UP :

The set up is consisted of two similar pumps of different sizes a wter tank and the necessary piping, valves and measuring instruments. The pumps can be operated in series, parallel or each one individually. Pump 2 is a smaller pump and can operate as a turbine. It can rotate in both directions whereas pump 1 can only rotate in one direction.

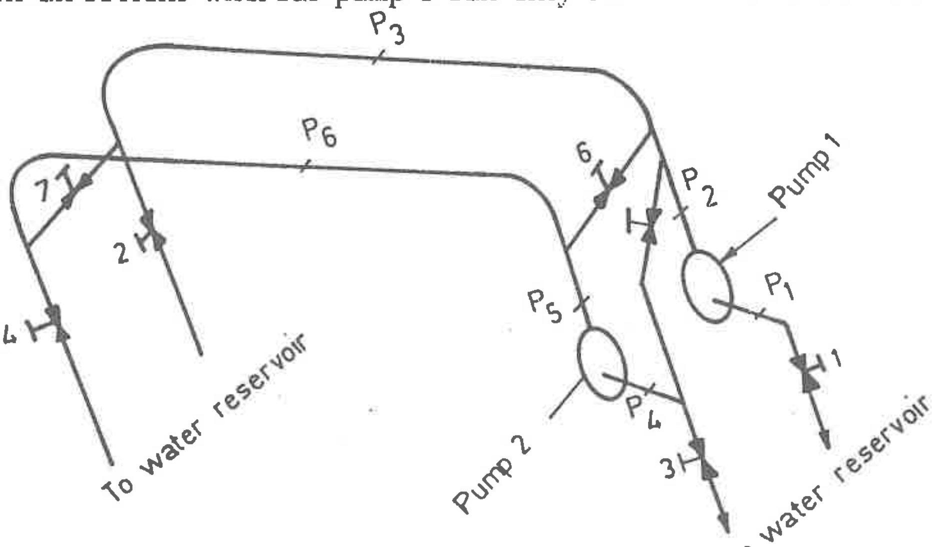


Fig. 1. Schematic diagram of the set up

Each machine has its own control panel. Motor speeds are adjustable and there is a flow meter on both piping. There are 6 pressure gauges on control panels and the points of measurement of these pressure are shown in fig. 1. In the table below valve positions for different operating conditions are shown :

OPERATION	VALVE POSITION						
	1	2	3	4	5	6	7
Pump 1 only	O	O	C	C	C	C	C
Pump 2 only	C	C	O	O	C	C	C
Parallel operation	O	O	O	C	C	O	C
Series operation	O	C	C	O	O	C	C

KEY

O : Open

C : Closed

_ : Control Valve

TAable 1. Valve positions for different operating conditions.

PROCEDURE :

For each step adjust the valves for the particular operation mode using table. Prime the pump before starting, and open the inlet valve of the pump slowly for proper operation. Check the pump speed frequently during experiment.

1. To determine the characteristic curves of pump 1 at N_1 RPM record the values of P_1 , P_2 and F for 8 operating values of Q. Start from full closed position of the control valve to full open position.

2. Determine the operating characteristic of pump 1 at N_2 RPM repeat the procedure in 1 for this speed.

3. To determine the characteristic curves of pump 2 at N_2 RPM record P_4 , P_5 and F for 8 operating values of Q.

4. To determine the characteristic curves for series operation of pumps 1 and 2 both running at N_2 RPM, record P_6 and Q for 8 operating conditions. Note that the discharge of large pump is fed into the inlet of small pump.

5. To determine the characteristic for parallel operation of pumps 1 and 2 both running at N_2 RPM, record P_3 and Q for 8 operating conditions.

CALCULATIONS :

1. Calculate the net head across the pump, H, hydraulic power, P_h, mechanical power, P_s, and efficiency.

2. Plot H-Q, P_s-Q, and η-Q on the same graph paper.

3. Using the similarity rules calculate and plot H and P_s vs Q for:

a) N_1 RPM using the data of pump 1 at N_2 RPM.

b) 4000 RPM using the data of pump 1 at N_2 RPM.

c) Scale pump 2 running at N_2 RPM to pump 1 at N_2 RPM.

4. Calculate flow and head coefficients Q/ND^3 and gH/N^2 for pump 1 running at N_1 and N_2 RPM. Plot the results. (Q: m^3/sec, H: m, N: RPS, D: m)

5. Calculate N_{sq} for maximum efficiency points for N_1, N_2 and 4000 RPM rotational speed of pump 1.

6. Plot the parallel and series operational modes H-Q characteristics on the same graph paper. Also plot H-Q curves of pump 1 and 2 at N_2 RPM on the same graph paper.

QUESTIONS :

1. Explain the effect of rotor speed and the impeller dimensions on the performance characteristics.

2. Explain the importance of maximum efficiency point.

3. Explain why specific speed of pump 1 at N_1, N_2 and 4000 RPM remained constant although operating conditions are differnt.

4. Label the critical points on the series and parallel operation curves and discuss their importance as well as their locations.

5. What has been achieved by parallel and series operation of pumps, explain.

EXPERIMENT : 3

PERFORMANCE CHARACTERISTICS OF AXIAL FLOW PUMP AND EFFECT OF BLADE SHAPE AND CONFIGURATION ON PERFORMANCE :

OBJECT :

The object is to investigate the effect of camber and blade stagger on pump performance, and to determine the actual velocity polygons and comparision of these with the ideal ones.

DESCRIPTION OF THE EXPERIMENTAL SET UP :

An axial flow pump with changeable rotor and stator is used. The blades of the machine can be changed and adjusted. It can be used as a pump or a turbine. Flow rate is measured by a venturi meter and it is directly read from the calibrated scale of the 'U' tube mercury manometer. Torque is read directly from calibrated torque meter. Another 'U' tube mercury manometer measures the pressures between the inlet and various points on the test set up, and is controlled by a pressure selector valve.

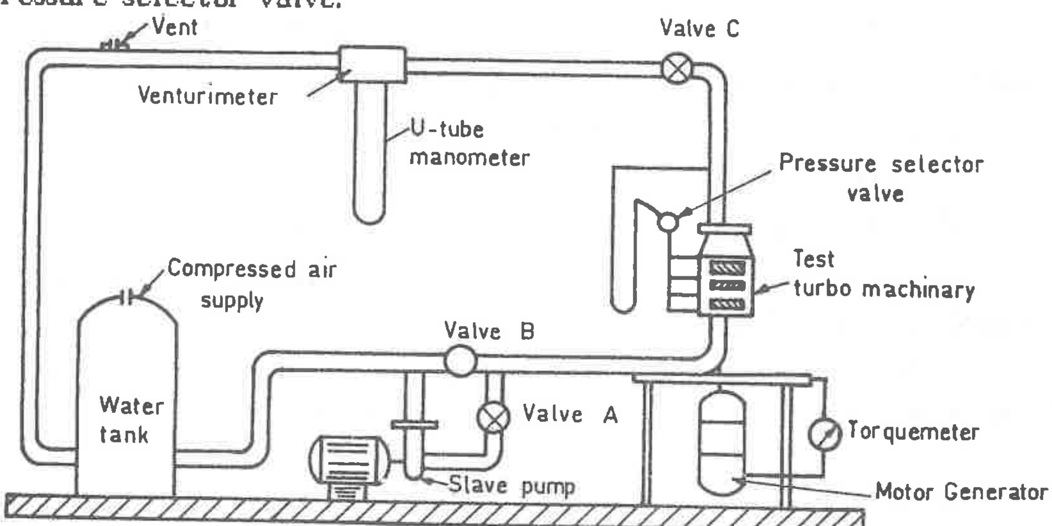


Fig. 1. Schematic diagram of the set up

PROCEDURE :

1. Set the reservoir air pressure to 0.4 bar.

2. Set water circuit valves to the following condition : Open valves B and C fully, close valve A.

3. Switch on main supply, turn selector switch to pump. Zero spring balance.

4. Rotate the pump speed control untill the desired speed is indicated on the tachometer.

5. Record speed , torque, flow rate, reference pressure, overall head rise, (set selector switch to position 5), flow direction and dynamic head at the exit.

6. Take about 14 readings of parameters mentioned in 5 by closing valve B from fully open to fully closed state in 14 step intervals.

7. After completing data slow down pump and turn of f power. Relieve pressure.

8. Change or reset the blades of rotor.

9. Do 1-7 again for new rotor.

CALCULATIONS :

1. Calculate the net head across pump, H, hydraulic power, P_h , mechanical power, P_s , efficiency η.

2. Plot P_s , H, and η vs Q.

3. Calculate N_{sq}

4. Calculate reaction at best efficiency point.

5. Determine the actual velocity polygons at maximum efficiency point for rotor inlet and exit.

6. By similarity analysis calculate H-Q characteristic at 3000 RPM and plot all results in terms of flow coefficient Q/ND^3 vs gH/N^2D^2.

QUESTIONS :

1. Is there a difference between maximum efficiency obtained from experiment and the efficieny corresponding to N_{sq} value for this class of pump obtained from chart in lecture notes. If there is a difference

what is the reason for it ?

2. Discuss the effect of stagger and camber to pump performance.

3. Discus the difference between actual and ideal velocity polygons.

4. State why an inlet guide vane in the direction of rotor rotation does not improve the pump performance ?

5. State why a stator following a rotor is a better combination than an inlet guide vane preceding a rotor.

EXPERIMENT NO : 4

PELTON TURBINE

OBJECT :

The object of this experiment is to investigate the basic performance characteristics of an impulse turbine.

DESCRIPTION OF THE SET UP :

The experimental set up is shown in fig. 1. Cast bronze Pelton Wheel inside the protective shielding is coupled externally to a manually loaded Prony Brake. Torque is measured by the Prony Brake. Flow rate is measured by the 'V' notch Weir and the static pressure at the inlet by a Bourdon type pressure gauge. Torque arm of the Prony Brake is 381 mm.

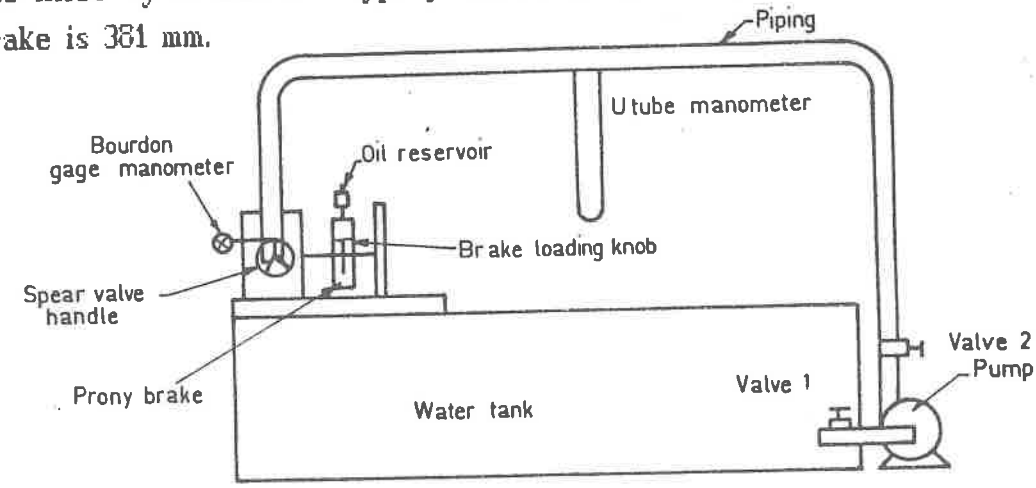


Fig. 1. Schematic diagram of the experimental set up.

Other important technical specifications of the set up are as follows :

Runner :

Mean diameter : 241 mm, No of buckets : 18, Outside diameter : 304 mm.

Prony Brake :

Brake drum diameter :304 mm, Brake arm radius : 381 mm.

Nozzle and Spear :

Nozzle diameter at outlet : 27.58 mm, Nozzle diameter at inlet : 63.5mm.
Included angle : 57°, Normal spear travel : 17.62 mm.
Maximum spear travel : 21.96 mm, Supply pipe diameter :76.2 mm.

PROCEDURE :

1. Open valve 1 and close valve 2. Start the pump.
2. Adjust the spear valve to 1/4 open position.
3. Open the oil flow on the brake drum.
4. Adjust valve 2 slowly until the desired head at the nozzle inlet is obtained.
5. Decrease the load until the wheel reaches the run away speed.
6. Take Q, H, F and RPM measurements.
7. Decrease the brake load a little and take F and RPM measurements.
8. Do step 7 ten more times until turbine stalls.
9. Do steps 4-8 again for 1/2, 3/4, and 4/4 spear valve opening positions.
10. Experiment is over shut off the pump.

CALCULATIONS :

1. Calculate the mechanical power, P_s , hydraulic power, P_h , and efficiency for the four spear valve positions.
2. Reduce calculated P_s and measured Q to standart values correspondig to a head of 100 m. (use similarity analysis p.p. 9 of lab. manual).
3. Plot efficiency and reduced P_s vs RPM for four spear valve settings.
4. Find most efficient speed from 3 which may correspond to any one of the four spear valve settings.
5. Calculate % of most efficient speed for all the measured values of RPM.
6. Calculate % of full load (load corresponding to maximum efficiency point) for maximum values of P corresponding to four

spear valve settings.

7. Calculate the % of full load discharge (discharge corresponding to maximum efficiency) for all values of measured Q.

8. Plot % maximum efficiency vs % of most efficient speed.

9. Plot % maximum efficiency and % rated full load discharge vs % of rated output on the same graph paper.

10. Plot % of full load discharge vs % of most efficient speed as a parameter. (Iso-Efficiency Characteristics).

QUESTIONS :

1. Outline the procedure of plotting iso-efficiency curves for % of full load output vs % of most efficient speed.

2. What is the specific speed of this turbine and how does it compare with the value obtained from the chart in the lecture notes ?

3. If this turbine is a model of a turbine to be placed in a hydraulic power station, where the water elevation from turbine inlet is 200 m and the required power output is 10000 kW. 1/3 of the total head is assumed to be lost as friction losses, will the prototype be suitable to do this job when coupled to a generator with 10 poles.

N = 60f/number of poles, and f : 50 cycles/sec.

EXPERIMENT NO : 5

PERFORMANCE CHARACTERISTIC OF A RADIAL FLOW TURBINE

OBJECT :

The object of the experiment is to investigate on the basic performance characteristics of a Francis Turbine.

DESCRIPTION OF THE EXPERIMENTAL SET UP :

Experimental set up is shown in fig. 1. A force gauge, a V-nitch weir and two pressure gauges are used to measure force, flow rate and pressure at inlet and outlet of the turbine. The torque arm is 240 mm.

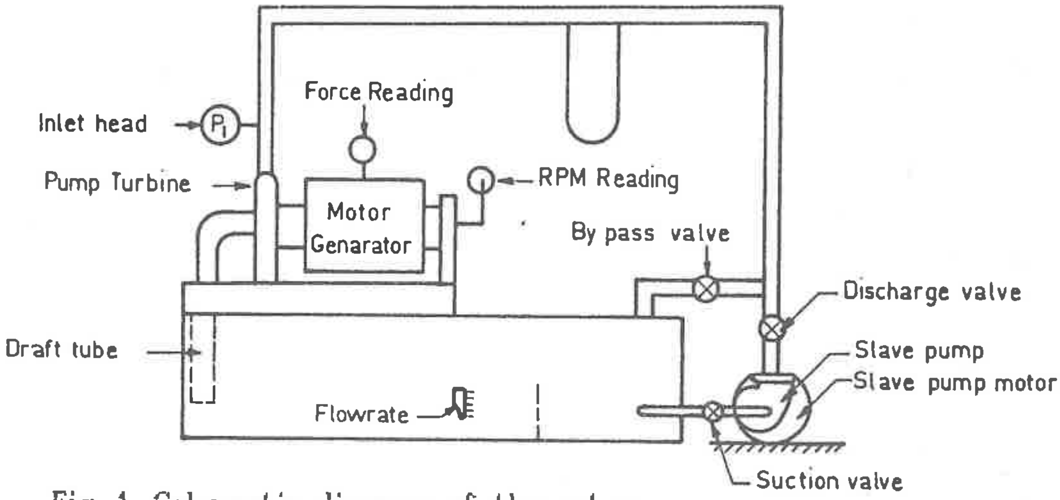


Fig. 1. Schematic diagram of the set up.

PROCEDURE :

1. Start slave pump with discharge valve fully closed and suction valve fully open.

2. Set the wicket gate position to full open (1/1)

3. Switch on Control cabin power and set thye selector switch to Generator , notice that load controller switch set at 100 %.

4. Slowly open the discharge valve of slave pump and adjusting by pass valve set the inlet head to H meters.

5. Record N, F, Q and h_s (tail race water level from the center of the runner) for at least 10 different load position starting from full load to run away speed.

6. Set wicket gate position to 3/4, 1/2, 1/4 opening and repeat steps 4 and 5 for each gate opening position.

CALCULATIONS :

1. Calculate V_2, the velocity at the exit of Draft tube.

2. Calculate ΔH across the turbine, P_s , P_{hyd} and η.

3. Reduce the values of N, Q and P_s to standart values. N_{st}, Q_{st}, $(P_s)_{st}$ corresponding to 1 meter of head across the turbine.

4. Plot η-N_{st} for 4 wicket gate opening on the same graph paper and determine the most efficient speed.

5. Plot similar graphs as shown in laboratory manual figures 8, 9 and 10.

QUESTIONS :

1. What is the influence of wicket gate setting on turbine performance.

2. What is the influence of Draft tube.

3. why do we reduce the measured values to standart values.

4. State the significance of load-η and load-Q plots.

5. State giving reasons whether the design of this turbine is acceptable.

6. Although the inlet head is kept constant state and explain why Q changes as the load on turbine runner is varied.

EXPERIMENT NO : 6

PERFORMANCE CHARACTERISTICS OF AN AXIAL FLOW TURBINE

OBJECT :

The object is to investigate the basic performance characteristics of an axial flow turbine.

DESCRIPTION OF THE EXPERIMENTAL SET UP

Set up described in third experiment is used.

PROCEDURE :

1. Open all the valves and fill the system with water applying 0.4 bar air pressure on water tank.

2. Set the valves to following condition; valves A and B closed, valve C fully open. Set pressure selector switch to position 5.

3. Switch on mains supply and turn selector switch to "turbine". Adjust torque meter to zero spring balance.

4. Open valve A until the required flow Q_1 is obtained, check that mercury level in pressure measuring manometer does not exceed 800 mm Hg. Fine flow control can be obtained by opening valve B.

5. Record Q_1.

6. Take reading of speed, torque, P_{ref} and overall head drop across the turbine ($P_5 - P_{ref}$).

7. Change speed by changing the load on turbine and readjust flowrate to Q_1 by adjusting valves A and B.

8. Do steps 6 and 7 for a series of speeds from full load to run away speed.

9. Repeat steps 5 to 8 for flowrates Q_2, Q_3 and Q_4.

10. After you have taken all of the data and after you have plotted η-RPM graph, determine the speed at maximum efficiency for Q_1

11. Adjust flowrate to Q_1 while running the machine at the speed corresponding to maximum efficiency.

12. Measure flow angles and velocity at entrance to the runner.

CALCULATIONS :

1. Calculate P_s , P_{hyd}, η.

2. Plot η - RPM, P_s - RPM, and ΔH - RPM for every Q.

3. Plot η - Q, P_s - Q and ΔH - Q for N RPM.

4. Calculate N_{sp}

5. Draw the inlet and exit velocity triangles to the rotor for Q_1 flowrate at maximum efficiency point.

6. Calculate reaction for Q_1 flowrate at maximum efficiency point.

QUESTIONS :

1. Comment on the N_{sp} values of particular turbine used in experiment as an axial flow turbine.

2. Compare flowrate and head of this turbine with those of a Pelton turbine.

3. Discuss the effect of reaction on turbine performance.

4. What happens to leakage losses of an axial turbine, compared with the energy transfered to the runner, as its size increased.

5. What is the effect of stator at the inlet on reaction.

6. Explain how the maximum efficiency can be attained in a Kaplan turbine at all loads.

EXPERIMENT NO : 7

INVESTIGATION OF CAVITATION CHARACTERISTICS OF A RADIAL FLOW PUMP

OBJECT :

The object of this experiment is to investigate the cavitation characteristics of a radial flow pump.

DESCRIPTION OF EXPERIMENTAL SET UP :

The set up described in experiment sheet of second experiment is used.

PROCEDURE :

1. Prime the pump 1.
2. Start pump with valve 2 fully closed and set pump rotational speed to N_1 RPM.
3. Fully open valve 1 slowly.
4. Partly open valve 2 so that Q_1 lt/sec flowrate passes through the pump.
5. Record P_1 , P_2 and torque.
6. Decrease P_1 by throttling valve 1 and adjust valve 2 to maintain the constant flowrate Q_1. Take the values P_1 , P_2 and T.
7. Repeat step 6 for different inlet throttling conditions.
8. Repeat steps 4 to 7 for a different value of pump discharge Q_2.
9. Repeat steps 4 to 8 for a different pump speed N_2.

CALCULATIONS :

1. Calculate NPSH and
2. Plot NPSH versus ΔH and NPSH versus η
3. Find critical NPSH.
4. CAlculate σ_c and S

QUESTIONS :

1. Explain and comment on the NPSH versus ΔH and NPSH versus η graphs.

2. Explain the importance of σ_c or S in pump selection for a given operation condition.

3. What are the influences of the following factors oc cavitation

a) Suction elevation

b) Barometric pressure

c) Pump rotational speed

d) Suction pipe size

e) Altitude from sea level of the geographical location of pump station

f) Temperature of the water being pumped

g) Dissolved air or gas content of pumped liquid.

D1

FOR ALL IMPELLER TYPES
NO OF BLADES : 8mm
BLADE HEIGHT : 15 mm
OUTSIDE DIAMETER : 200 mm
INSIDE DIAMETER : 100 mm
N : rpm

	Angle from radial	
	Inlet ang	Outlst ang
Straight	0°	0°
Forward	70°	0°
Backward	70°	65°

IMPELLER TYPE	T (N-M)	P_s - Pref (mm Hg)	Pref (mW-c)	P - Pref (ΔH) (m-Wc)	P_o (W)	Phyd (W)	η %	Q (kg/sec)	α_2	Dynamio head for C_2(mm Hg)	Q (kg/sec)	α_1	Dynamio head for C_1(mm Hg)		
STRAIGHT															
FORWARD															
BACKWARD															

D2

Series Operation	Pump II	Pump I	Pump I
Pump 1 at rpm Pump 2 at rpm	N_2: rpm	N_2: rpm	N_1: rpm

Q (lt/sec)

P_6 (mWC)

Parallel operation

Pump 1 at rom

Pump 2 at rom

Q (lt/sec)

P_3 (mWC)

Torque Arm = 0.165 m
D_1 = 0.140 m
D_2 = 0.076 m

EXPERIMENTAL

Q lt/sec	$P_1(P_4)$ mWC	$P_2(P_5)$ mWC	F N	H mWC	T N-M	P_{hyd} W	P_{mech} W	η %

Calculated using similarity

Q	H	P_{hyd}	P_{mech}	η %

Pump 2 at N_2 rom using the data of Pump 1 at N_2 rom	Pump 1 at 4000 rom using the data of Pump 1at N_2 rom	Pump 2at N_1 rom using the data of Pump 1at N_2 rom

D3

D = 89 mm
d = 66 mm
lean Dia : 77.5 mm
Na of blades : 12
C = 20 mm S/C 1 = 0^{0}

STATOR
θ =
γ =

RPM :

ROTOR	Q(kg/sec)	T(N-m)	H(cm Hg)	H(m WC)	Pref(m WC)	P_s (W)	P_{hyd} (W)	η (%)	α_2	Dynamic Head for C2		Similarity			
												Q(kg/sec)	H (m)	Q/ND^3	$H/(ND)^2$
γ : θ :															
γ : θ :															
γ : θ :															

D4

SpoarValve Opening	1/4	1/2	3/4	1/1
Head Flow	H : Q :	H : Q :	H : Q :	H : Q :
RPM				
BrakeForos F (N)				
T (N - M)				
P_s (W)				
Phyd				
η %				
% of most effient speed				
% of maximum η				
% of rated discharge				
% of rated output				

D5

Diameter at the exit of Draft tube :

Hinlet =

$(\Delta H)_{st}$ = 1m

Valve Position	N RPN	F	Q	h_s m	T N-m	W l/sec	P_h W	V_2 m/sec	H mWC	P_{hyd}	η %	(Ps_{st}) W	Q_{st}	N_{st}	%N	%Q	%P_s	
1/1																		
3/4																		
1/2																		
1/4																		

Q: 70

γ: 45 for both runner and stator.

D6

	N (RPN)	T (N-M)	Pref (mwc)	Ps-Pref (mm Hg)	W (1/sec)	P_s (W)	H (m)	P_{hyd} (W)	η %
a_1 :									
a_2 :									
a_3 :									
a_1 :									

α_2

H_{toticl} :

H_{static} :

$Y_{awmeter}$ measurement for point O_1 at η_{max}

07

P_{atm} :

D_{inlet} :

T_{H_2O} :

Suction elevation :

	N_2 : RPM		N_1 : RPH	
	Q_2 :	Q_1 :	Q_2 :	Q_1 :
P_1 (m)				
P_2 (m)				
T (N-m)				
H (m)				
P_S (w)				
P_{hyd} (w)				
η (%)				
NPSH				