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 principle of energy transfer between the impeller and the fluid, moreover, the effect of the blade shape on the performance characteristics of the pump.

DESCRIPTION OF THE SETUP:

The flowrate is measured by means of a venturi meter placed at the top of the setup. A mercury manometer is connected to the venturi meter directly gives the flowrate on a calibrated scale. Pressures at four different points on the impeller and at the pump discharge line are measured with respect to the inlet pressure by a U-tube 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 pressurised air on the water tank until the system is full of water until water issues out from the vent.
2. Make sure that the valve A is closed before starting the pump.
3. Set the selector switch to pump operation and adjust the speed.
4. Read torque, flow rate and pressure head at inlet and across the pump, namely, P_{ref}, and $(P_5 – P_{ref})$.
5. Insert the yaw meter into position 3, note the angular position of the yaw meter, and dynamic head for V_2.
6. Repeat steps 3 and 4 for seven more times each at different openings of valve A. The last reading should be taken when valve A is full open.
7. Stop the pump, release the pressure and insert the yaw meter into the inlet of the impeller.

8. Adjust the speed to the value set in step 2, and take at least 5 yaw angle measurements starting from shut off to the full flow condition. Record yaw angle and corresponding dynamic head for V_2.

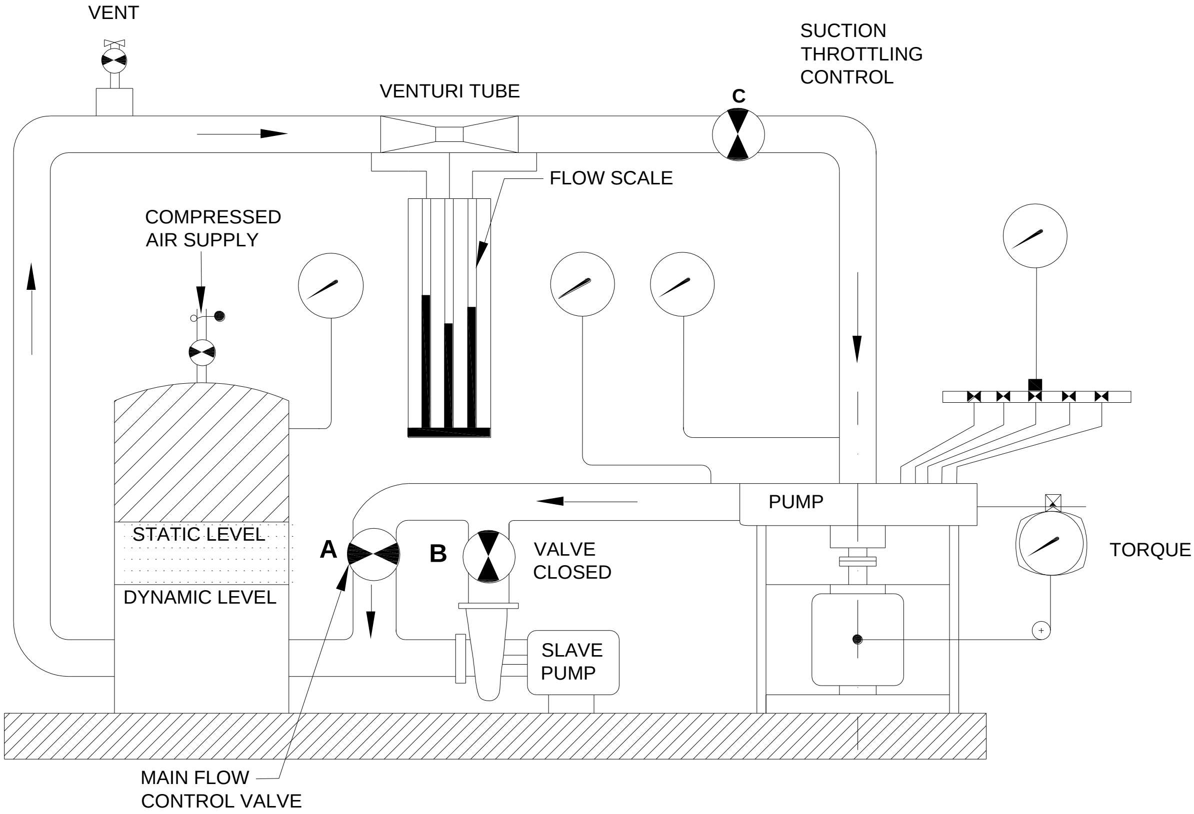

Figure 1 Radial flow water pump flow diagram.

CALCULATIONS:

After completing all readings, calculate:

1. The head developed by the pump,
2. Hydraulic and mechanical power (IP_h and IP_s),
3. Efficiency, η,
4. Specific speed, N_s,

After completing the calculations above and using the data gathered for each three different types of impeller:

a) Draw h-Q, η-Q, IP_s-Q, α_1-Q, α_2-Q on the same graph paper,

b) Determine actual V_1 and V_2 from data and construct actual velocity polygons at point of maximum efficiency point for each of the different impellers,

c) Calculate V_{m1} and V_{m2} from data and draw theoretical velocity polygons at maximum efficiency point.

QUESTIONS:

1. Why does the actual characteristics deviate from Euler line,
2. What are the basic differences in the actual characteristics of the three impellers tested? Explain by referring to graphs plotted, actual velocity polygons, 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 do backward curved blades are preferred in practise 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, to determine the operational characteristics using similarity rules and compare them with those experimentally determined characteristics. In addition it is aimed to investigate the performance characteristics of parallel and series operation of radial flow pumps.

DESCRIPTION OF THE SETUP:

The setup consists of two similar pumps of different sizes together with a water tank, the pipe network, necessary fitting elements, valves and measurement instruments. The pumps can be operated in series, parallel or each one individually. Smaller pump can also be operated as a turbine. It can rotate in both directions whereas Pump 1 can only rotate in one direction.

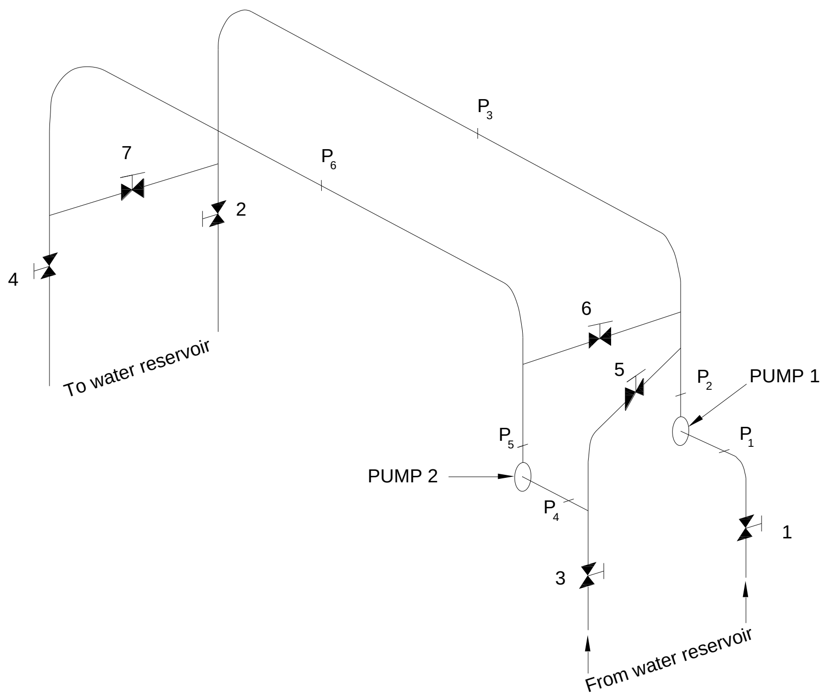

Figure 2 Pipe network schematic for Experiment 2.

Each pump has its own control panel. Motor speeds are adjustable and there is a flow meter on both piping circuit. There are 6 pressure gages on control panels. The locations of these pressure measurements are shown in Fig. 2. Valve settings corresponding to different modes of operation are given in the following table:

OPERATION MODE	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

PROCEDURE:

For each step adjust the valves for the particular operation mode as specified in the table 1. Open the inlet valve of the pump slowly (designated in the table as O*) for proper operation after priming the pump before starting. Closely monitor the pump speed during the experiment.

1. Set the speed of the pump 1 to $\mathbf{N_1}$. Record $\mathbf{P_1}$, $\mathbf{P_2}$ and **F** for eight different values of **Q** by adjusting valve 2 starting from closed to fully open position.
2. Change the speed of the pump 1 to $\mathbf{N_2}$ and repeat the same sequence of measurements as stated in step 1.
3. First close valve 2, then valve 1 and stop the pump.
4. Set the speed of the pump 2 to $\mathbf{N_2}$. Record $\mathbf{P_4}$, $\mathbf{P_5}$ and **F** for eight different values of **Q** by adjusting valve 4 starting from closed to fully open position.
5. For parallel operation: set the speed of the both pump to $\mathbf{N_2}$. Record $\mathbf{P_3}$ for eight different values of **Q** by adjusting valve 2 starting from closed to fully open position.
6. For series operation: set the speed of the both pump to $\mathbf{N_2}$. Record $\mathbf{P_6}$ for eight different values of **Q** by adjusting valve 4 starting from closed to fully open position.

CALCULATIONS:

After completing all readings:

1. Calculate the net head across the pump, h, hydraulic power, IP_h, shaft power, IP_s, and efficiency, η.

2. Plot h-Q, IP_s-Q, and η–Q on the same graph paper (pump characteristics).
3. Using the similarity rules and the data obtained for pump 1 at N_2 calculate and plot h and IP_s versus Q for:
    a) Pump 1 running at N_1,
    b) Pump 1 running at 4000 rpm,
    c) Scale pump 2 running at N_2.
4. Calculate flow coefficient Π_Q, head coefficient Π_h for pump 1 running at N_1 and N_2.
5. Plot h-Q characteristic on the same graph paper for pump 1 and 2 at N_2 together with the parallel and series operation modes.

QUESTIONS:

1. Explain the effect of rotor speed and impeller dimensions on the performance characteristics.
2. Explain the importance of the design point.
3. Explain why specific speed of pump 1 at N_1, N_2, and 4000 rpm remains constant although operating conditions are different.
4. Label critical points on the series and parallel operation curves and discuss their importance as well as their location of occurrence.
5. What has been achieved by parallel and series operation of pumps, explain.

EXPERIMENT NO: 3

AXIAL FLOW PUMP PERFORMANCE CHARACTERISTIC: INFLUENCE OF BLADE SHAPE AND CONFIGURATION

OBJECT:

It is aimed to reveal the effect of two important design parameters on the performance characteristics, namely, the blade stagger and camber angles on the pump performance. A comparison regarding to the actual and theoretical velocity polygons will be made to see these effects in detail.

DESCRIPTION OF THE SETUP:

An axial flow pump with changeable rotor and stator is used. Different rotor and stator combinations can be tested such that in each case blades having different camber angle may be tested and stagger angle can be adjusted to different values.

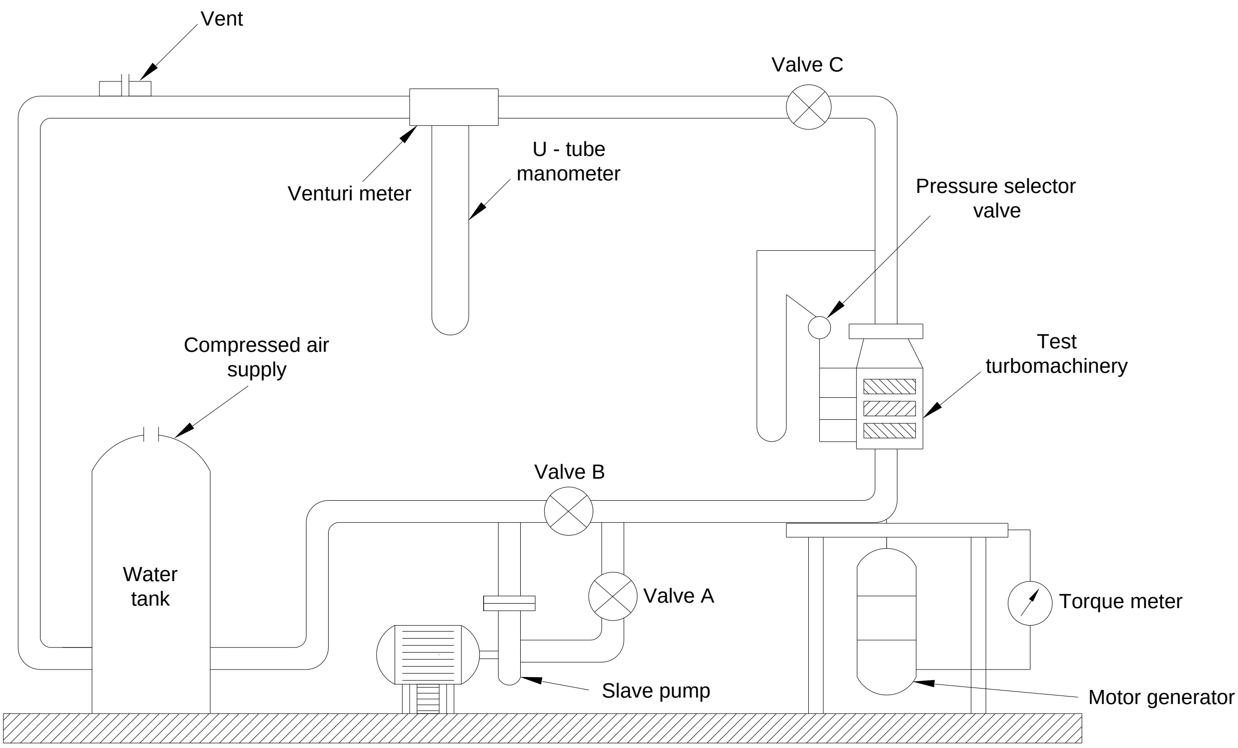

Figure 3 Axial flow pump test rig diagram.

Flow rate is measured through a venture meter which is connected to a calibrated U-tube manometer. Torque is directly read from a calibrated torque meter. Another manometer is used selectively to measure the pressure difference between the pump inlet and at different points along the pump axis.

PROCEDURE:

1. Set the water reservoir pressure to 0.4.
2. Fully open valves B, C, and fully close valve A
3. Switch on the mains supply, turn operation mode selector switch to pump and set the torque meter to zero spring balance.
4. Bring the rotational speed to the required value by turning the knob of the speed control unit and record it.
5. Read torque, flowrate, reference pressure, overall head rise (pressure selector switch set to position 5), flow direction, and dynamic head at the exit.
6. Repeat step 5 for 14 different flowrate by adjusting valve B from fully open to fully closed position.
7. Upon completion of data collection bring rotational speed to zero, turn off mains supply, and relieve pressure.
8. Set the stagger angle of the rotor to a different value or change rotor having blades with different camber angle.
9. Repeat steps 1-7 for the same configuration

CALCULATIONS:

After completing all readings:

1. Calculate the net head across the pump, h, hydraulic power, IP_h, shaft power, IP_s, and efficiency, η.
2. Plot h-Q, IP_s-Q, and η–Q on the same graph paper (pump characteristics).
3. Caclulate, N_s.
4. Calculate degree of reaction, R, at the point of maximum efficiency.
5. Draw the actual velocity polygons at maximum point at the inlet and exit.
6. By using the affinity laws calculate h-Q characteristic at 3000 rpm, and plot results in terms of Π_h vs Π_Q.

QUESTIONS:

1. Is there a difference between the maximum efficiency obtained from experiment, and the efficiency corresponding to N_s value for this class of

pump obtained from graph given in lecture note. If there is a difference what is the reason for it?

2. Discuss the effect of stagger and camber angles on the pump performance.
3. State why an inlet guide vane in the direction of rotor rotation does not improve the pump performance?
4. State why a stator following a rotor better combination than an inlet guide vane is preceding a rotor.

EXPERIMENT NO: 4

PELTON TURBINE

OBJECT:

The aim of the experiment is to reveal the performance characteristics of an impulse turbine.

DESCRIPTION OF THE SETUP:

Pelton wheel inside the casing is coupled to a manually controlled prony brake to mimic the varying loading conditions on the turbine and the torque produced in doing so is measured via a scale attached to it. Flow rate is measured by the V-notch type flow meter as shown in the figure. The static pressure at the inlet of the nozzle is measured by a bourdon gage type manometer. Some of the specifications are already carved on a plate attached to the setup and will be repeated here:

Prony brake torque arm:	381 mm,	Prony brake drum dia.:	304 mm
Runner mean diameter:	241 mm,	Runner outer diameter:	304 mm
Number of buckets:	18,	Nozzle outlet diameter:	27.58 mm
Nozzle inlet diameter:	63.5 mm	Spear included angle:	57^0
Normal spear travel:	17.62 mm	Maximum spear travel:	21.96 mm
Supply pipe diameter:	76.2 mm		

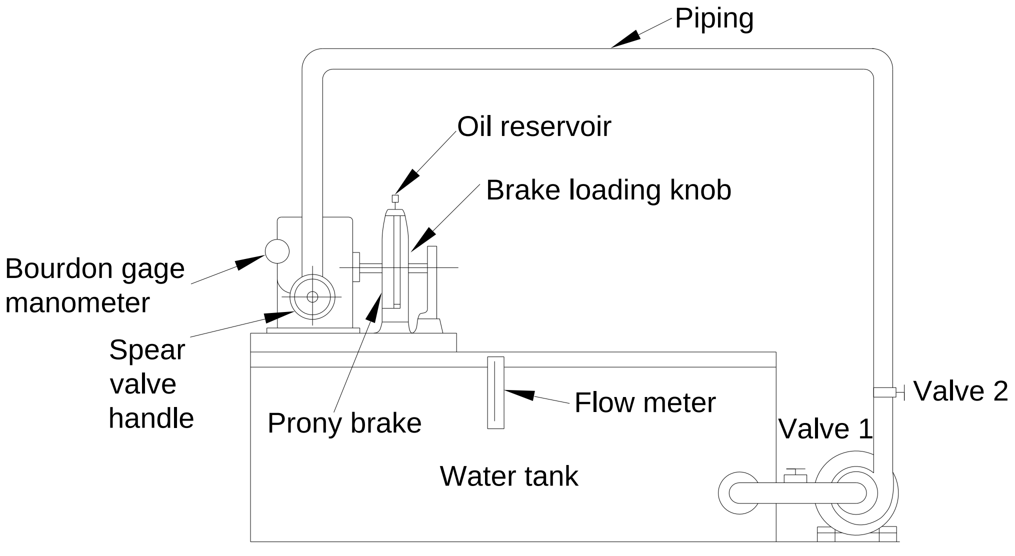

Figure 4 Pelton turbine setup diagram.

PROCEDURE:

1. Open valve 1 and close valve 2.
2. Adjust the spear valve position to 1/4.

3. Open the oil flow on the brake drum.
4. Adjust the valve 2 until the desired head at the nozzle inlet is obtained.
5. Record head, h and flowrate, Q.
6. Gradually decrease the loading by means of brake loading knob until the wheel reaches to runaway speed. This should be completed approximately in 10 steps.
7. Record F, and N at each loading condition set in step 6.
8. Repeat steps from 4 to 7 for spear valve opening positions of 1/2, 3/4, and 4/4.

CALCULATIONS:

After completing all readings:

5. Calculate the shaft power, IP_s, hydraulic power, IP_h, and efficiency for all data points taken.
6. Calculate IP_s, and Q corresponding to a head of 100 m by using the data obtained and affinity laws.
7. Plot efficiency, η and reduced IP_s versus N for all valve opening settings.
8. Determine the most efficient speed (speed at the maximum efficiency point) and the corresponding flowrate, Q and load IP_s form the plot created at step 3.
9. Calculate the percentage of the most efficient speed for all measurements.
10. Calculate the percentage of the load corresponding to the load at the most efficient speed for all points.
11. Calculate the percentage of the discharge corresponding to the discharge at the most efficient speed for all points.
12. Plot % of maximum efficiency versus % of the most efficient speed.
13. Plot % of maximum efficiency and % of rated full load discharge versus % of rated discharge on the same graph paper.
14. Plot % of full load discharge vs % of most efficient speed (Iso-efficiency curves).

QUESTİONS:

1. Outline the procedure of plotting iso-efficiency curves for % of full load output vs % of the 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, will the prototype be suitable to do this job when coupled to a generator with 10 poles. Assume that 1/3 of the total head lost to the frictional losses.

   N = 60 f / # of poles, f = 50 Hz.

EXPERIMENT NO: 5

PERFORMANCE CHARACTERISTIC OF RADIAL FLOW TURBINE

OBJECT:

The performance characteristics of a radial flow turbine will be obtained by running the unit under constant head, h, by varying the load and observing load /speed characteristics at various guide vane settings.

DESCRIPTION OF THE SETUP:

The basic features of the setup are shown on the figure given below. The head is adjusted to the required value by means of discharge valve and this head is maintained by the use of by pass valve. Force is read from the gage and the related torque is calculated using the torque arm length of 240 mm. Flowrate is monitored via a V-notch weir. As the load changes the speed of the turbine will vary and it is read from the rpm reading gage. Two pressure gages are used to measure the turbine inlet and outlet pressures. Loading is varied by turning the generator control knob.

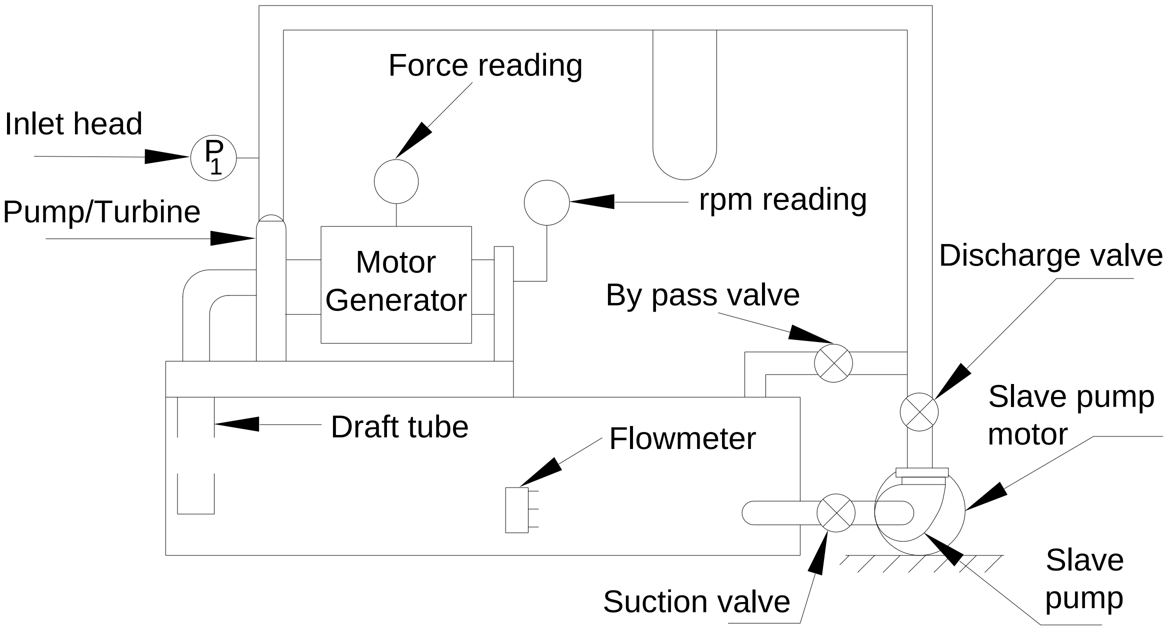

Figure 5 Radial flow water turbine setup diagram.

PROCEDURE:

1. Set the wicket gate position to full open position (1/1).
2. Start slave pump with discharge valve fully closed and suction valve fully open.
3. Switch on control cabin power and set the selector switch to generator, notice that load controller switch set at 100 %.
4. Slowly open the discharge valve of slave pump and fine tune the head by using the bypass valve to the required value of h.
5. Record speed, N, force, F, flowrate, Q, and h_s (the level between the centre of the runner and tailrace water level) for at least 10 different loading condition starting from full load to no load condition (runaway speed state).
6. Repeat the steps 4 and 5 for wicket gate settings of 3/4, 1/2, 1/4.

CALCULATIONS:

After completing all readings, calculate:

1. Calculate the velocity at the exit of the draft tube, V_{dte}.
2. Calculate the head across the turbine, h, shaft power, IP_s, hydraulic power, IP_h, and overall efficiency, η.
3. Calculate N_{st}, Q_{st}, and IP_{st} corresponding to a head of 1 m across the turbine by using the data obtained and affinity laws.
4. Plot η-N_{st} for all wicket gate settings on the same graph paper and determine the most efficient speed.
5. Plot % of maximum efficiency versus % of the most efficient speed.
6. Plot % of maximum efficiency and % of rated full load discharge versus % of rated discharge on the same graph paper.
7. Plot % of full load discharge vs % of most efficient speed (Iso-efficiency curves).

QUESTIONS:

1. What is the influence of wicket gate setting on the turbine performance?
2. What is the function of the draft tube?
3. Why the measured values are transformed to the standard state?

4. State the significance of % of full load discharge versus % of most efficient speed and % of maximum efficiency and % of rated full load discharge versus % of rated discharge plots.
5. State giving reasons whether the design of this turbine is acceptable.
6. Although the inlet head is kept constant explain why flowrate, Q changes as the load on the turbine runner is varied.

EXPERIMENT NO: 6

PERFORMANCE CHARACTERISTIC OF AN AXIAL FLOW TURBINE

OBJECT:

Basic performance characteristics of an axial flow turbine will be investigated.

DESCRIPTION OF THE SETUP:

The setup is already described in experiment 3.

PROCEDURE:

1. Open all valves and fill up the system with water and then apply 0.4 bar pressure to the water tank.
2. Close valve A and B, and fully open valve C.
3. Set the pressure selector switch to position 5.
4. Switch on mains supply and turn selector switch to turbine mode.
5. Adjust torque meter to zero spring balance position.
6. Open valve A until the required flow Q_1 is obtained. Make sure that the pressure level in the pressure measuring manometer does not exceed 800 mmHg. Fine tuning of the flow is achieved by means of valve B.
7. Record Q_1.
8. Write down speed, N, torque, τ, P_{ref}, and overall head drop across the turbine ($P_5 - P_{ref}$).
9. Change the speed by changing the load on the turbine and readjust flow rate Q_1 by means of valves A and B.
10. Repeat steps through 8 and 9 for a series of speeds from full load to runaway speed.
11. Repeat steps 7 to 10 for flow rates Q_2, Q_3, and Q_4.
12. After completing data collection plot η versus N, and determine the speed, N at the point of maximum efficiency for Q_1.
13. Adjust flowrate once again to Q_1 while running the machine at the speed found in step 12.

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

CALCULATIONS:

After completing experiment, calculate:

1. Shaft power, IP_s, hydraulic power, IP_h, and overall efficiency, η.
2. Plot η - N, IP_s – N, and h – N for every Q.
3. Plot η - Q, IP_s – Q, and h – Q for every N.
4. Calculate N_{sp}.
5. Draw the inlet and exit velocity triangles to the rotor for flowrate Q_1 at the point of maximum efficiency.
6. Calculate degree of reaction, R, for flowrate Q_1 at the point of maximum efficiency.

QUESTIONS:

1. Comment on the N_{sp} values of particular turbine used in the experiment.
2. Compare flowrate and head of this turbine with those of a pelton turbine.
3. Discuss the effect of degree of reaction on turbine performance.
4. What happens to the leakage losses of an axial flow turbine, compared with the energy transfer to the runner, as its size increased?
5. What is the effect of stator at the inlet on degree of reaction?
6. Explain how the maximum efficiency can be attained in a Kaplan turbine at all loads.

EXPERIMENT NO: 7

INVESTIGATION OF CAVITATION CHRACTERISTIC OF A RADIAL FLOW PUMP

OBJECT:

Cavitation behaviour of a radial flow pump is to be examined.

DESCRIPTION OF THE SETUP:

The same setup used in experiment 2 will be used.

PROCEDURE:

1. Prime pump 1.
2. Start pump 1 with valve 2 fully closed and set the rotational speed to N_1.
3. Fully open valve 1 slowly.
4. Partly open valve 2 so that Q_1 L/s 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 torque.
7. Repeat steps 6 for different inlet throttling conditions.
8. Repeat steps 4 to 7 for another different value of flow rate Q_2.
9. Repeat steps 4 to 8 for a different pump speed N_2.

CALCULATIONS:

1. Calculate NPSH.
2. Plot NPSH versus h and NPSH versus η.
3. Determine critical NPSH.
4. Calculate σ_{cr} and S_{cr}.

QUESTIONS:

1. Explain and comment on the NPSH versus h and NPSH versus η.

2. Explain the importance of σ_{cr} and S_{cr}.in pump selection for a given operating condition.
3. What is the influence of the following factors on cavitaiton:
    a) Suction elevation, h_s.
    b) Barometric pressure, P_{atm}.
    c) Rotational speed, N.
    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.

EXPERIMENT NO: 8

PERFORMANCE CHRACTERISTIC OF A PISTON PUMP

OBJECT:

It is aimed to investigate the performance characteristic of a piston pump.

DESCRIPTION OF THE SETUP:

The setup is shown in Fig. 6 has been designed as a simple self-contained unit to permit to reveal the important features of performance characteristic of a piston pump.

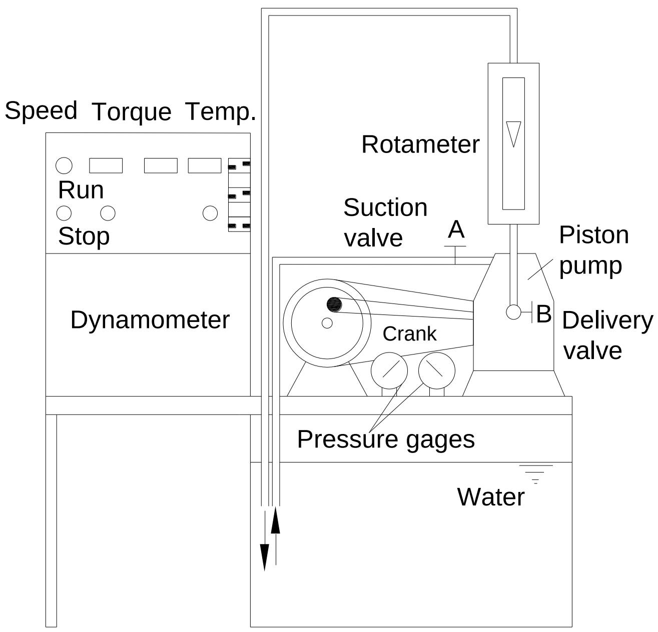

Figure 6 Schematic diagram of the setup

The pump is a horizontal single-cylinder, double acting machine which is the most frequently used type of pump to supply water in isolated homesteads for domestic use. A sectional view of the pump is given in Fig. 7.

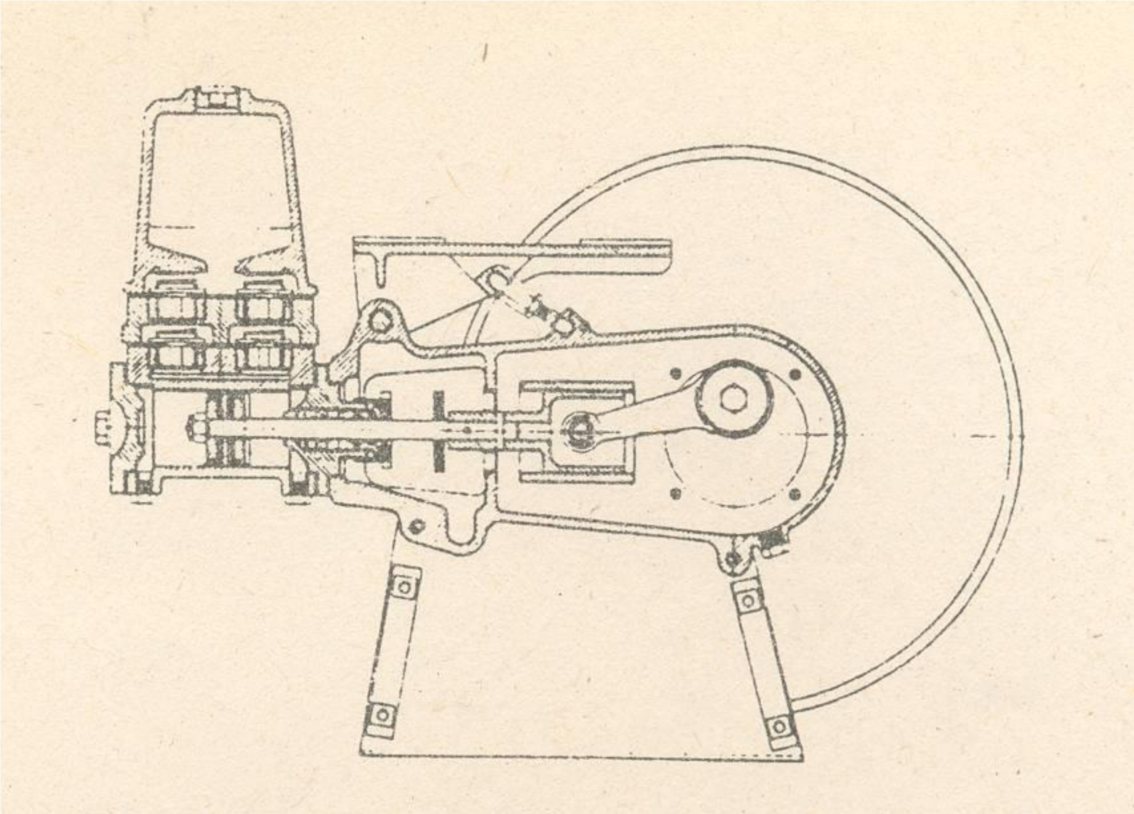

Figure 7 Sectional view of the pump.

The pump is driven by a 0.75 kW – two speed ac dynamometer motor. The power is transmitted from the motor to the pump by a toothed belt. The motor is mounted on trunnion bearings and is fitted with a spring balance for the measurement of torque and a counter for the measurement of rotational speed.

The pump takes water from a reservoir by means of a control valve. Water is delivered by the pump through a second control valve and, after passing through a flowmeter, is returned to the reservoir. The pump delivery and suction pressures are regulated by means of delivery and suction valves which are monitored from a pressure gage and a vacuum gage, respectively. Both gages are provided with needle valves to damp out pulsations. The pump is fitted with an air vessel to reduce pressure fluctuations in the delivery pump and with a relief valve to prevent overloading of the pump.

PROCEDURE:

1. Make sure that both inlet (A) and outlet (B) control valves are fully closed prior to start up.
2. Switch on the motor and run it at a low speed to let the system to exhale any trapped air and reach to steady operation mode.

3. Set the delivery pressure using delivery valve and speed to the required values.
4. Record suction pressure P_1, delivery pressure P_2, torque τ, rotational speed N, and flow rate Q.
5. Repeat steps 3 – 4 until the maximum delivery pressure is reached such that incrementing the whole range should yield at least 10 different states.

Note: Adjust the needle valves below the suction and delivery pressure gages such that the amplitude of the oscillations of the gage pointers are kept at minimum.

If the air vessel at the delivery becomes filled with water, the noise produced by the pump will increase and the indicator diagram will be distorted by showing a marked variation in the delivery pressure.

On the other hand, if a shortage of air is suspected then the vessel may be recharged by running the pump for a short time whit the air inlet valve beneath the pump cylinder slightly open.

CALCULATIONS:

Upon completion of the data collection, calculate:

1. Pump outlet and inlet pressure difference, $(P_2 – P_1)$.
2. Shaft power IP_s, and hydraulic power IP_h.
3. Overall efficiency η, and volumetric efficiency η_v.
4. Indicated power, IP_i.
5. Mechanical efficiency η_m.
6. Plot The flowrate Q, shaft power IP_s, overall and volumetric efficiencies η.& η_v.against the head difference across the pump on the same graph paper.

QUESTIONS:

1. List the basic differences between rotodynamic and positive displacement pumps.
2. Compare h-Q characteristic of the piston pump and gear pump.
3. Compare h-Q characteristic of the piston pump and rotodynamic pump.

EXPERIMENT NO: 9

PERFORMANCE CHRACTERISTIC OF A GEAR PUMP

OBJECT:

It is aimed to investigate the performance characteristic of a gear pump.

DESCRIPTION OF THE SETUP:

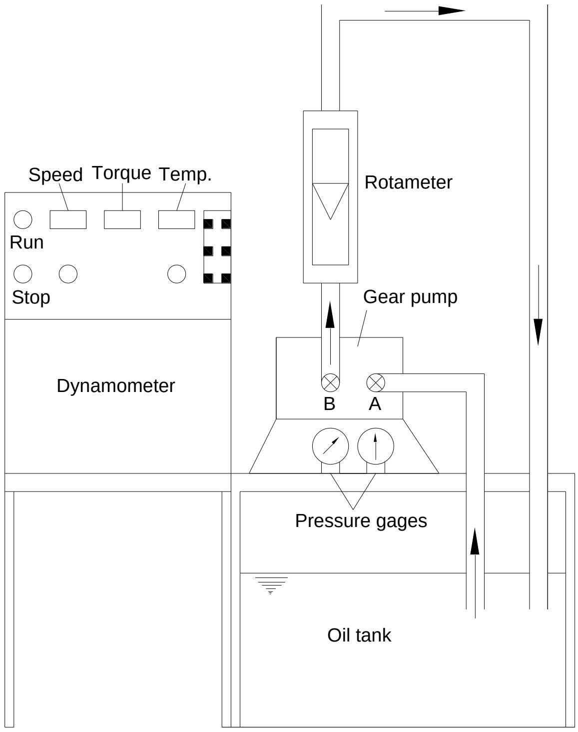

Figure 8 Diagram of the gear pump setup.

The setup is he type comprising two meshing helical gears.

The pump is flange-mounted and driven by a 0.75 kW two speed dc dynamometer motor. The motor is mounted un trunnion bearings and is fitted with a spring balance for the measurement of torque and a tachometer for the measurement of speed.

The pump and driving motor are mounted on a common steel base plate which is placed on a steel tank that forms the oil reservoir for the apparatus.

The pump draws oil from the reservoir by a suction pump provided with a throttling valve by means of which the pressure at the inlet of the pump is varied. The pump discharges via a throttling valve to a diverter which directs the oil either back to the reservoir or to a calibrated measuring tank equipped with a gauge glass.

A spring loaded relief valve bypasses the pump delivery back to the suction side should the delivery control valve somehow be completely closed, and thus protecting the pump and motor from overloading.

PROCEDURE:

1. Ensure that both control valves at the inlet (A) and outlet (B) are fully open.
2. Start the motor running at the low speed and run for a short period to allow the air to escape from the system.
3. After setting the motor speed set the delivery pressure by means of valve B to the required value. Note that the relief valve opens at 800 kPa, approximately.
4. Before starting the data collection let the system to run and to reach steady operating condition. Afterwards, fully close the delivery valve (B)
5. Register the suction pressure P_1, the delivery pressure P_2, torque τ, rotational speed N.
6. Set the level of oil in the measuring tank by means of drain valve and record this level.
7. Direct oil by means of diverter in to measuring tank and at the start of the action run the stopwatch.
8. Return the diverter to the waste position and stop the stopwatch when the tank is nearly full.
9. Record the new level in the tank and drain ready for next test.
10. Adjust delivery valve opening to a different position at given motor speed and repeat steps 5 – 9.

CALCULATIONS:

When the data acquisition finished, calculate:

1. Pump outlet and inlet pressure difference, $(P_2 - P_1)$.
2. Volumetric rate offlow, Q.
3. Shaft power IP_s, and hydraulic power IP_h.
4. Overall efficiency η, and volumetric efficiency η_v.
5. Plot The flowrate Q, shaft power IP_s, overall and volumetric efficiencies η.& η_v.against the head difference across the pump on the same graph paper.

QUESTIONS:

1. Compare h-Q characterisitic of the gear pump with that of the piston pump.
2. Compare h-Q characteristic of the gear pump with that of the rotodynamic pump

PROPERTIES OF IMPELLERS

Number of Blades : 8

Blade Height : 15 mm

Outside Diameter : 200 mm

Inside Diameter : 100 mm

N : (rpm)

	Angle from Radial Direction	
	Inlet Angle	Outlet Angle
Backward	70^0	65^0
Forward	70^0	0^0
Straight	0^0	0^0

Impeller Type	BACKWARD										FORWARD										STRAIGHT									
τ (N.m)																														
P_s-P_{ref} (mmHg)																														
P_{ref} (m WC)																														
P-P_{ref} (ΔH) (m WC)																														
IP_s (W)																														
IP_{hyd} (W)																														
η (%)																														
m (kg/s)																														
α_2																														
Dynamic Head for V_2 (mmHg)																														
α_1																														
Dynamic Head for V_1 (mmHg)																														

	EXPERIMENTAL									USING SIMILARITY LAWS					
	Q L/s	P_1 (P_4) m H_2O	P_2 (P_5) m H_2O	F (N)	h m H_2O	τ N.m	IP_h W	IP_s W	η (%)	Q L/s	h m H_2O	IP_h W	IP_s W	η (%)	
Pump 1 at N_1															Pump 2 at N_1 using the data of Pump 1 at N_2
Pump 1 at N_2															Pump 1 at 4000 rpm using the data of Pump 1 at N_2
Pump 2 at N_2															Pump 2 at N_2 using the data of Pump 1 at N_2

Series operation both pumps at N_2	Q (L/s)	P_6 (m H_2O)	Parallel operation both pumps at N_2	Q (L/s)	P_3 (m H_2O)

Torque arm = 0.165 m

D_1 = 0.140 m

D_2 = 0.076 m

Rotational speed N_1 = rpm

Rotational speed N_2 = rpm

D = 89 mm

Mean dia. = 77.5 mm

Number of blades = 12

C = 20 mm

S/C = 1

STATOR γ:

θ:

N = rpm

		ROTOR γ: θ:														γ: θ:														γ: θ:													
	m (kg/s)																																										
	τ (N.m)																																										
	h (cm Hg)																																										
	h (m H_2O)																																										
	P_{ref} (m H_2O)																																										
	IP_s (W)																																										
	IP_h (W)																																										
	η (%)																																										
	α_2																																										
	Exit dynamic head																																										
Affinity Laws	m (kg/s)																																										
	h (m H_2O)																																										
	Π_Q																																										
	Π_h																																										

	Valve position: 1/4	1/2	3/4	4/4
h, Q, IP_h	h = Q = P_h =	h = Q = P_h =	h = Q = P_h =	h = Q = P_h =
N (rpm)				
Brake force (N)				
τ (N.m)				
P_s (W)				
η (%)				
% of N at η_{max}				
% of η at η_{max}				
% of rated Q				
% of rated P_s				

Draft tube d_e = mm

h = m

h_{st} = 1 m

	Wicket gate setting: 1/1										3/4										1/2										1/4									
N (rpm)																																								
F (gr)																																								
Q (m^3/hr)																																								
h_s (cm)																																								
τ (N.m)																																								
ω (rad/s)																																								
IP_s (W)																																								
V_{dt} (m/s)																																								
h (m)																																								
IP_h (W)																																								
η (%)																																								
IP_{st} (W)																																								
Q_{st} (m^3/hr																																								
N_{st} (rpm)																																								
% N																																								
% Q																																								
% IP_s																																								

For both runner and stator: $\theta : 70^0$ $\gamma : 45^0$

Flowrate (kg/s)	N (rpm)	τ (N.m)	P_{ref} (m H_2O)	P_5 - P_{ref} (m H_2O)	ω (rad/s)	IP_s (W)	h (m)	P_h (W)	η (%)
Q_1									
Q_2									
Q_3									
Q_4									

Yaw meter measurements at the point of maximum efficiency, η_{max}:

α_2 = h_{total} = h_{static} =

P_{atm} :　　　　　　T_{water}

D_{inlet} :　　　　　　h_s :

		P_1 (m)	P_2 (m)	τ (N.m)	h (m)	IP_s (W)	IP_h (W)	η (%)	NPSH (m)
N_2 : rpm	Q_1 :								
	Q_2 :								
N_2 : rpm	Q_1 :								
	Q_2 :								

	P_1 (kPa)	P_2 (kPa)	$P_2 - P_1$ (kPa)	τ (N.m)	N_m (rpm)	IP_s (W)	N_p (rpm)	h (m)	Q (L/min)	IP_h (W)	η (%)	η_h (%)	P_i (kPa)	IP_i (W)	η_m (%)
1															
2															
3															
4															
5															
6															
7															
8															
9															
10															

Technical data	Nominal motor power	0.75 kW
	Nominal motor speed	750/1500 rpm
	Pump/motor speed ratio	18/120
	Max. delivery pressure	600 kPa
	Maximum flow rate	27 L/min

Double acting	Cylinder bore	44.5 mm
	Piston rod dia.	11.1 mm
	Piston stroke	41.3 mm
Swept volume, V		0.1245 L/rev
Torque arm length		179 mm
Brake constant, K		53.35

"