EXPERIMENTS IN FLUID POWER ENGINEERING

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DEPARTMENT OF MECHANICAL ENGINEERING

SUBJECT : FLUID POWER ENGINEERING

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INDEX

PRACTICAL NO: - 01

AIM: - Explanation of Hydropower station.

Quiz:

1. Explain in brief about Hydro power plant
2. Classification of Hydro power plant.
3. Site selection for Hydro power plant.
4. Basic Component of Hydro power plant.

Answers:

PRACTICAL NO: - 02

AIM: - Explanation of Impact of Jet.

Quiz:

1. Forced exerted by a jet on a stationary inclined plate.
2. Forced exerted by a jet on a stationary curved plate.
3. Forced exerted by a jet on stationary unsymmetrical curved plate when jet strikes tangentially at one end.
4. Forced exerted by a jet on Hinged plate.
5. Forced exerted by a jet on inclined moving flat plate in the direction of jet.
6. Forced exerted by a jet on an unsymmetrical moving curved plate when jet strikes tangentially at one of tips.

PRACTICAL NO: - 03

AIM: - To perform the practical on Pelton Wheel Turbine and determine its operating characteristics

APPARATUS:-

The test rig mainly consists of (1) A Pelton Turbine, (2) A Supply pump unit to supply water to the above Pelton Turbine, (3) Flow Measuring unit consisting of a Venturi meter and Differential "U" tube Manometer (4) Piping system and (5) Sump.

GENERAL DESCRIPTION:-

The unit essentially consists of casing, with a circular transparent window kept at the front for the visual inspection of the impact of the Jet on buckets, a bearing pedestal, a rotor assembly of shaft, runner and brake drum, all mounted on a suitable sturdy cast iron base plate, A rope brake arrangement is provided to load the turbine. The input to the turbine can be controlled by adjusting the spear position by means of a hand wheel fitted with indicator arrangement. The water inlet pressure is measured by a pressure gauge and for the measurement of speed; a Digital tachometer is to be used. An Optimum size sump is provided to store sufficient water from independent circulation through the unit for experimentation.

CONSTRUCTIONAL SPECIFICATION:

1. **CASING:-** It is made of a close grained cast iron having a large circular transparent window.
2. **RUNNER:-** It is made of cast gunmetal disc fitted with accurately finished gun metal buckets and electroplated
3. **SHAFT:-** It is made of Stainless, steel for just free special steel operation and of sample size for high strength
4. **NOZZLE:-** It is made of gunmetal designed for smooth flow.
5. **SPEAR:-** of stainless steel designed for efficient operation.
6. **SPEAR SPINDLE:-** It is made of stainless steel of liberal size.
7. **INLET BEND:-** It is made of cast iron, which accommodates the indicator bracket.

8. **BALL BEARINGS:-** It is made of double row deep groove rigid type in the casing and double rows self aligning type in the pedestal both of liberal size.
9. **BRAKE ARRANGEMENT:-** It consists of Machined and polished brake drum 200mm dia. cooling water pipes, internal water scoop, discharge pipe, standard cast iron dead weights, spring balance, rope brake etc., arranged for loading the turbine.
10. **FINISHING:-** It is of high standard suitable for the laboratory used in technical institutions

INTRODUCTION:

Hydraulic machines are grouped into two categories, namely those which absorb power and the other which develop power. In the first class of machines, the energy transfer takes place between the rotor and the fluid and thus the work is done on the fluid. Such machines are called **Pumps**. However, in the second category of machines, the work is done by the fluid. These machines are called **Turbines**. If the above class of machines have the primary part rotating and there is a free passage of fluid between the inlet and outlet, these are called **Rotodynamic machines**. These machines are thus different from another class of machines known as **Positive Displacement** machines in which category we have reciprocating pumps, screw, gear, vane pumps and hydraulic rams.

In positive displacement machine, ordinarily the primary part has a reciprocating motion and fluid flow is fluctuating.

Hydraulic Turbines are classified into Impulse and Reaction turbines on the basis of their working principle. In the impulse turbines (Pelton wheel), all the hydraulic energy available is first converted to kinetic energy in the form of a water jet by passing water through one or more nozzles before it strikes the buckets fixed on a disc. Due to the change in the angular momentum, the energy is transferred between the fluid and the rotor. In these turbines, the pressure is atmospheric throughout.

In the Reaction turbines (Francis, Propeller and Kaplan) the energy at the inlet is in the form of kinetic as well as pressure. As the water passes through the guide ring and rotor, change in velocity and pressure takes place. Ultimately water is discharged through a draft tube into the tail race.

Water turbines are mainly used in power houses for power generation. A comparison of the main types of turbines in use is given below.

Pelton Turbine:

It is a tangential flow impulse turbine and was invented by Lester A. Pelton (1829-1908). It consists of a rotating disc on which buckets having a sharp edged splitter in the centre giving them double hemispherical shapes are fixed. Water is supplied to the buckets in the form of a jet which comes out of a nozzle as shown in the figure.

Item	Pelton Wheel	Francis	Kaplan
Specific Speed	4-35 per nozzle	120-430	300-1000
Operating Head	200-1700 m	80-500 m	4-200 m
Max. Efficiency (full load)	93%	94%	94%
Speed Regulation	Dual (spear valve and deflector)	Guide Vanes	Movable runner blades
Part Load Eff.	85 % at 40 % load	70% at 40% load	Max. eff. at about 40% load
Runaway Speed (N is the design speed)	1.8N to 1.9N	2N to 2.2N	2.5N to 3.0N

The turbine may have more than one nozzle to increase the power of the unit but keeping in the specific speed under limits. If the net head available at the nozzle is H_n, the absolute velocity of the jet coming out of the nozzle is

$$v_1 = C_v \left(2gH_n\right)^{1/2} \quad \textbf{(1)}$$

Where C_v is the velocity coefficient or velocity ratio, which has a value ranging between 0.97 to 0.99.

If it is assumed that the jet is deflected through an angle θ as shown in the figure. And the relative velocity at the outlet is k times the relative velocity at the inlet, it can be shown by applying the Euler's equation of turbine that the specific energy transfer

$$E = (v_i - u_i)(1 - k\cos\theta)u_i / g \quad \textbf{(2)}$$

Further, it can also be shown that the maximum efficiency of the turbine

$$\eta_{\max} = (1 - k\cos\theta)/2 \quad \textbf{(3)}$$

It is observed from the above equation that in the ideal case when the k= 1.0 and θ = 180°, the efficiency becomes 100 %.
Since, in practical situation k is in the region of 0.9 and the angle of deflection is usually 165° to avoid the returning of the jet striking the back of the oncoming bucket, the efficiency is in the range of 92-94%.

As turbines are mainly used to drive the electrical generators, it is imperative that these rotate at constant speed irrespective of the load. In Pelton turbines, there is dual speed regulation through spear valve and the deflector as shown in the figure.

As soon as there is a change in the load, the spear moves forward or backward in the nozzle thus reducing or increasing the rate of flow respectively (the head on the turbine remains constant). Since the spear can not be moved suddenly because of the serious risk of water hammer effect which may result in bursting the penstock, a deflector plate is used. The deflector plate deflects the jet away from the buckets and in the mean time the spear adjusts itself to the correct position gradually. Once, the correct opening of the nozzle is achieved, the deflector returns to its original position. The movement of the spear and the deflector is controlled automatically through a servo motor mechanism operated by a governor.

Performance of Turbines:

Turbines are designed and constructed to work under optimum conditions. However, it is seldom that these work under designed conditions and fro a major period, work at regimes other than designed regimes. It is therefore, necessary to determine the variation of different parameters with the change in working regimes for which it is imperative to carry out tests in the laboratory on model turbines. But, before performance of turbines is discussed in detail, it is necessary to define the relevant terms like head, power, speed and efficiency.

HEAD:

In assessing the performance of a turbine, correct meaning of the term gross head and the net head should be understood. The gross head is defined as the difference in levels of head race and tail race. The net or effective head has different definitions for impulse and reaction turbines. For impulse turbine, the net head is the head at the entrance to the nozzle reckoned above a datum plane passing through the lowest point of the bucket pitch circle. In reaction turbines, the net head is the difference between the specific energy just before the water enters the turbine and the specific energy at the tail race.

POWER:

There are three ways in which the power of a turbine can be expressed, namely hydraulic power, brake power and input power. The hydraulic power is that which is determined from the velocity triangles. Thus,

$$P_{hyd} = \rho\, Q\, (u_i\, v_{wi} - u_o\, v_{wo}) \qquad \textbf{(3)}$$

(It may be noted that it can also be expresses as $P_{hyd} = \rho\, Q\, (h_n - h_f)$, where, h_n is the net head and h_f is the fluid friction loss as the flow takes place through the turbines including the exit loss)

The brake power or output power, P_o is defined as the power available at the shaft of the turbine. Thus if T represents the torque delivered to the shaft and ω is the angular velocity,

$$P_o = \omega\, T \qquad \textbf{(4)}$$

The input power is defined as the power taken from the water. Thus

$$P_i = \rho\, g\, Q\, h_n \qquad \textbf{(5)}$$

SPEED:

The term speed is defined in many ways studying performance of turbines, such as designed or normal speed, synchronous speed, runaway speed, unit and specific speed. Designed speed, N, which is some times also called the normal speed is that speed of the turbine at which its efficiency is maximum. As the turbines are invariably coupled to generators which must be run at constant speed decided upon by the frequency of electric power and the number of poles of the generator, these should also run at the constant speed. This speed is known as the synchronous speed. The runaway speed is defined as the maximum speed at which turbine would run, when there is no load on the turbine but operates under designed head and discharge. Unit speed is that hypothetical speed of turbine which operates under unit head. It is given by the formula:

$$N_1 = N / \sqrt{H_n} \qquad \textbf{(6)}$$

The specific speed N_s of a turbine defined as the speed of a geometrically similar turbine working under unit head and developing unit power. It is given by the expression:

$$N_s = NP_o^{1/2} / H_n^{5/4} \tag{7}$$

It should be noted that all the parameters, N, P_o and H_n should correspond to maximum power. (Although SI units are used universally now, the units of N and P_o in the above expression are RPM and metric Horse Power.)

EFFICIENCY:

There are four types of efficiencies which are associated with the performance of turbines, namely the overall efficiency, η_o, volumetric efficiency, η_v, hydraulic efficiency, η_h and mechanical efficiency, η_m. The overall efficiency is defined as the ratio between the shaft power and the input power and is given by the expression:

$$\eta_o = P_o / P_i = \omega T / \rho g Q H_n \tag{8}$$

Where T is the torque delivered.
The volumetric efficiency takes care of any possible loss due to leakage in the runner of the turbine. Thus,

$$\eta_v = (Q - Q_1)/ Q \tag{9}$$

where Q_1 represents the leakage loss. IN most of the circumstances the volumetric efficiency is taken as 100%. However, in some unfavourable conditions, the leakage loss may have to be accounted for.

The hydraulic efficiency is defined as the ratio of hydraulic power transferred to the runner and the input power. Thus,

$$\eta_h = (V_{wi}u_i - V_{wo}u_o) /g \; H_n \tag{10}$$

The above efficiency can also be expressed as $\eta_h = (H_n - H_h)/ H_n$ where H_n represent hydraulic head loss in flow including the exit loss.

The mechanical efficiency is defined as the ratio of shaft power to the power exerted by the water on the runner. Thus,

$$\eta_m = \omega T / \omega (T + T_f) \tag{11}$$

Where T_f represents the torque required to overcome mechanical friction. It may be noted that mechanical friction will include the loss in bearings and disc friction.

The behaviour of water turbine is studied through the following three types of characteristics, namely:

(a) Main Characteristics
(b) Operating Characteristics
(c) Universal Characteristics

Main Characteristics:

These are also called the characteristics at constant head. The condition of constant head is not often found out side the laboratory and therefore these characteristics have only a theoretical importance. Such curves are drawn by performing experiment at various nozzle/ guide vane openings and carrying out the brake test. All the measured parameters are reduced to unit quantities. Usually, the curves are plotted between unit discharge, unit power and efficiency versus unit speed indicating nozzle/ guide vane opening as a percentage.
Typical main characteristics for Pelton Turbine, Francis Turbine and Kaplan Turbine are shown in Figure 3 and 4

It can be seen that as far as the main characteristics of Francis and Kaplan turbines are concerned, the difference occurs only in the unit discharge – unit speed curves, the other two characteristics (P_1 vs. N_1 and η Vs. N_1 are more or less of the same shape.

Operating Characteristics:

These characteristics are also known as constant speed curves as these are drawn for values obtained at constant speed. These curves have a much more practical importance as normally the turbines are supposed to run at constant speed when other parameters may change according to their availability. For example the gross head may change with varying level in the reservoir at in-take and the net head varies because of changed flow conditions in the penstock. Normally, these characteristics show the variation of efficiency with percentage of power output at maximum efficiency. Typical operating with percentage of power output at maximum efficiency. Typical operating characteristics of various turbines are shown in the figure 5.

Universal Characteristics:

These are also known as constant efficiency or Muschell curves. These are a much more important graphical statement of turbine performance. These are in fact obtained from main characteristics as follows:

(1) Efficiency versus unit speed curves are drawn for various nozzle/ guide blade openings.

(2) On the same graph sheet and with the same scale for unit speed curves are drawn for the same nozzle/ guide blade openings between unit power and unit speed.

(3) On the first graph parallel horizontal lines are drawn for different efficiencies.

(4) Vertical projections are drawn from the points where the above straight lines intersect the efficiency curves on to the other graph drawn between unit power and unit speed.

(5) Points are marked where ever these projections intersect the corresponding curves in respect of nozzle/ guide vanes openings.

(6) Smooth curves are drawn representing the same efficiency.

OBJECT: To determine the Performance characteristics of a Pelton Turbine.

General Remarks:

All machines in which the input is in the form of hydraulic energy and the out put as mechanical energy are termed turbines or motors. If the action is roto dynamic, the machines are classified as turbines. However, in case of positive displacement action, the machines are classified as hydraulic motors. Hydraulic turbines are further classified on the basis of working principle as impulse or reaction turbines. Pelton turbines which is popularly known as Pelton wheel is an impulse turbine in which buckets of double hemispherical shape are attached to the rotating wheel. Water under pressure is supplied to the turbine through one or more nozzles in the form of jet which strikes the buckets in the centre, spills and leaves symmetrically on both the sides of the bucket.

The tests on the Pelton wheel are performed to:

1. Determine the performance characteristics
2. Determine the forces acting on different components of the turbine
3. Determine the vibrations produced in various components of the turbine.
4. Study certain special phenomenon in the turbine.

The above tests help in understanding the application of momentum principle, significance of universal characteristics (Isoefficiency or Muschel Curves) and model study of turbines.

Working Formulae:

(1) $h_i = p_i/ \rho g + v_i^2/ 2 g + z_i + z$

(2) $h_o = p_o/ \rho g + v_o^2/ 2 g + z_o$

(3) $h = h_i - h_o$

$= p_i/ \rho g + v_i^2/ 2 g + z_i$

$= p_i/ \rho g + 16 Q^2/ (\pi^2 d_i^4 2g) + z_i$

(4) $P_i = \rho g Q h$

(5) $P_o = 2 \pi N W L$

(6) $\eta_o = P_o / P_i$

(7) $N_1 = N/ h^{1/2}$

(8) $Q_1 = Q/ h^{1/2}$

(9) $P_1 = P_o / h^{3/2}$

Where,

h_i = the total energy of water per unit weight at inlet of the turbine

h_o = the total energy of water per unit weight at outlet of the turbine

$p_i/\rho g$ = pressure head at the inlet of the turbine

$p_o/\rho g$ = pressure head at the outlet of the turbine (neglected as it represents atmospheric head)

$v_i^2/2g$ = velocity head at the turbine inlet

$v_o^2/2g$ = velocity head at the turbine outlet (usually neglected as it is a very small quantity)

z_i & z_o = datum heads of the entry and exit sections of the turbine which are equal.

z = height of pressure gauge centre above the inlet section of the turbine

d_i = diameter of the supply pipe line

P_i = input power of the turbine

P_o = output power of the turbine

N = the speed of the turbine

W = difference in the tension between the two sides of the brake rope

L = radius of the brake drum

η_o = efficiency of the turbine

N_1, Q_1 and P_1 are the unit speed, discharge and power respectively.

Experimental Setup:

The set up consist of a centrifugal pump which draws water from the sump through a foot valve and supplies it to the Pelton wheel. On the supply line, venturimeter is installed for flow measurement. The pressure gauge at the entry gives the supply pressure. Gate opening is varied by the spear valve arrangement. The torque is measured by the brake drum arrangement, which has the facility for varying the load by putting on known weights on the weighing pan.

START UP:

- Make sure all the pipe lines are free from the foreign matter and all the joins are water tight and leak proof.
- Make sure all the bearings in the set up are properly lubricated.
- Start the pump with by pass valve partially closed condition.
- Open the ball valve provided on the pressure gauge to read the pressure of the supply line.
- Slowly open the spear valve and close the bypass valve so as to develop the necessary head at the turbine inlet.
- The stuffing box of the pump should occasionally show the drip of water. If the gland is over tightened, the dripping stops but the packing will heat up and damage the shaft.
- Start the cooling water supply to the rope brake pulley with the help of the valve provided.
- Do not suddenly load the turbine, load the turbine gradually and at the same time open the spear to run the turbine.

SHUT DOWN:

Before switching off the supply of the pump set, first remove the entire load gradually.

- Slowly close the spear to its fully close position and at the same time open the by pass valve, so as to avoid the loading of the pump.

OBSERVATION:

Diameter of nozzle	:	0.022 m
Diameter of Delivery pipe	:	0.065 m
Diameter of suction pipe	:	0.075 m
Radius of brake drum	:	0.15m (r)
Distance from turbine centre to jet	:	0.170m

Observation Table:

Sr. no.	Flow rate Q (m^3/s)	Disc. Pre. (m)	Manometer reading h (m)		Applied weight w = mg ($kg\ m/s^2$)	Spring force S = mg ($kg\ m/s^2$)	RPM (Brake drum)	Pi= ρghq (HP)	Po (HP)	Eff. η (%)
			h_1	h_2						
1										
2										
3										
4										
5										

Venturi meter:

Diameter of converging		:	0.065m
Area of converging /pipe inlet A_1		:	0.00331 m^2
Diverging section		:	0.065m
Diameter of Throat		:	0.039m
Area of throat A_2		:	0.00119 m^2
Coefficient of Discharge	C_d	:	0.96
Area Ratio	A_2/A_1	:	0.0000039487

CALCULATION:

1. $Q = CA_2 \sqrt{\dfrac{2g(h_1 - h_2)}{1-(A_2 / A_1)^2}}$

2. Pi = Input power =ρ*g*h*q

   ρ = Density of fluid, kg / m^3

   g = 9.81 m / s^2

   h = Head developed, m

   q = Flow rate of fluid,m^3 / s

3. P_o=Output power = W*T

   W = 2*3.14*N / 60

   T = ($w_{d\,-}$ s) *9.81*r

   R = Radius of drum=0.15m

   N = r.p.m of Drum,

   w_d = Weight of (Dead weight+rope+hanger),Kg

   s = Spring loaded weight, Kg

4. Efficiency of turbine: Po / Pi

Conclusion:

PRACTICAL NO: - 04

AIM : To determine operating characteristics of Francis Turbine

INTRODUCTION:

The hydraulic turbines are classified according to the type of energy available at the inlet of the turbine, direction of flow through the vanes. The Hydraulic turbines mainly of two types

1. Impulse Turbine
2. Reaction Turbine

Francis Turbine is axial flow reaction turbine. If the water flows parallel to the axis of the rotation of the shaft the turbine is known as axial flow turbine. And of the head at the inlet of the turbine is the sum of pressure energy and kinetic energy and during the flow of water through runner at part of pressure energy is converted into kinetic energy, the turbine is known as reaction turbine

The main parts of a Francis Turbine are :

1. Scroll casing
2. Guide vanes mechanism
3. Hub with vanes or runner of the turbine
4. Draft tube EXPERIMENTAL SET-UP

This apparatus i.e. Francis Turbine is designed for laboratory experiments and study purpose and to conduct tests in metric units for the following specifications:

Net head : 20 meters Approx

Discharge : 2000 LPM

Normal speed : 1500 RPM

Motor & Pump: 15 HP Monoblock Pump Kirloskar Make

Turbine Impeller: Gun Metal

The turbine consists of a spiral casing a rotor assembly, shaft and brake drum all mounted on a sturdy support pedestal. A straight conical draught tube is provided vertically after the runner. Rope brake arrangement with suitable pulleys is provided for loading the turbine. The input into the turbine controlled by a set of guide vanes. The net supply head is measured by means of a differential manometer. For the measurement of speed tachometer is to be used.

PROCEDURE

1) Clean the apparatus and make tank free from dust.
2) Close the drain valve provided.
3) Fill sump tank ¾ with clean water and ensure that no foreign particles are there.
4) Fill manometer fluid i.e. Hg in manometer by opening the valves of manometer and one PU pipe from pressure measurement point of pipe.
5) Connect the PU pipe back to its position and close the valves of manometer.
6) Ensure that there is no load on the brake drum.
7) Switch ON the pump with the help of the starter.
8) Open the air release valve provided on the manometer, slowly to release the air from manometer. (This should be done very carefully.)
9) When there is no air in the manometer, close the air release valves.
10) Now turbine is in operation.
11) Apply load on hanger and adjust the spring balance load by hand wheel just to release the rest position of the hanger.
12) Note the manometer reading, pressure gauge reading and vacuum gauge reading.
13) Measure the RPM of the turbine.
14) Note the applied weight and spring balance reading.
15) Repeat the same experiment for different load.
16) Regulate the discharge by regulating the guide vanes position.
17) Repeat the experiment for different discharge.

OBSERVATION TABLE

Sr. No.	RPM N	Pr. Gauge Read-ing P	Vacuum Gauge Read-ing V	Differential Pressure of Manometer reading			(Weight + Hanger with Rope) Reading w1	Adjustable Spring Balance Reading w2	W = (w1-w2)	HEAD H	Velocity V	Q = AV	INPUT Kw	OUTPUT Kw	η TURBINE Output = ---------- x 100 Input
				h 1	h 2	H = (h1-h2)									
1.															
2.															
3.															
4.															
5.															

CALCULATION

PRACTICAL NO: - 05

AIM: To determine operating characteristics of Kaplan Turbine

INTRODUCTION :

The hydraulic turbines are classified according to the type of energy available at the inlet of the turbine, direction of flow through the vanes. The Hydraulic turbines mainly of two types :

1. Impulse Turbine
2. Reaction Turbine

Kaplan Turbine is axial flow reaction turbine. If the water flows parallel to the axis of the rotation of the shaft the turbine is known as axial flow turbine. And of the head at the inlet of the turbine is the sum of pressure energy and kinetic energy and during the flow of water through runner at part of pressure energy is converted into kinetic energy, the turbine is known as reaction turbine. When the vanes are fixed to hub and they are not adjustable the turbine is known as propeller turbine. But if the vanes of the hub are adjustable the turbine is known as Kaplan Turbine after the name of V. Kaplan an Austrian Engineer. This turbine is suitable where a large quantity of water at low heads is available.

The main parts of a Kaplan Turbine are :

1. Scroll casing
2. Guide vanes mechanism
3. Hub with vanes or runner of the turbine
4. Draft tube

Procedure

1. Unload the turbine if there is any load on it.
2. Fill the sump tank with sufficient amount of water
3. Switch on the power supply to start the centrifugal pump.
4. Now open the delivery valve slowly and adjust the required head at the inlet of turbine, simultaneously manipulate by pass valve, so as to adjust the required head.
5. Now open the supply of the water to cool load drum.
6. Adjust the load on load drum and note the corresponding reading of spring balance. Also measure the rpm. Increase the load slowly and note the corresponding values of spring balance.

Repeat the above procedure for different inlet pressures.

OBSERVATION TABLE

S. No.	RPM	Pr. Gauge Read-ing	Vacuum Gauge Read-ing	Differential Pressure of Manometer reading			(Weight + Hanger with Rope) Reading	Adjustabl e Spring Balance Reading		HEAD	FLOW RATE		INPUT Kw	OUTPUT Kw	η TURBINE Output = ---------- x 100 Input
	N	P	V	h 1	h 2	H = (h1-h2)	w1	w2	W = (w1-w2)	H	V	Q = AV			
1.															
2.															
3.															
4.															

A= 0.034619 m2

STANDARD DATA

g	=	Acceleration due to gravity	= 9.81 m/sec^2
ρ_w	=	Density of water	= 1000 kg/m^3
ρ_m	=	Density of manometerfluid i.e. Hg	= 13550 kg/m^3
D	=	Dia meter of brake drum	= 350 mm = 0.35 meter
C_d	=	0.69	

Venturi meter:

Diameter of converging		:	0.065m
Area of converging /pipe inlet A1		:	0.00331 m^2
Diverging section		:	0.065m
Diameter of Throat		:	0.039m
Area of throat A2		:	0.00119 m^2
Coefficient of Discharge	C_d	:	0.96
Area Ratio	A_2/A_1	:	0.0000039487

FORMULA USED

1. **TOTAL HEAD**

$$\mathbf{H} = 10\left(P_d + \frac{P_s}{760}\right) \text{ m of } H_2O$$

P_d = Pressure Gauge reading

P_s = Vacuum of Hg.

2. **VELOCITY**

$$v = c_d\sqrt{2gh/100*((\rho_m/\rho_w)-1)} \text{ m/s}$$

$$v = c_d\sqrt{2gh/100*12.6}$$

h = Manometric difference in cm

ρ_m = density of manometer fluid i.e. Hg = 13550 kg/m^3

ρ_w = Density of water = 1000 kg/m^3

g = 9.81 m/sec^2

3. DISCHARGE

$$Q = CA_2 \sqrt{\frac{2g(h_1 - h_2)}{1-(A_2/A_1)^2}}$$

4. H.P. Hydraulic (Input) = ρgHQ **KW** = **KW**

1. B.H.P. (Output)

$$BHP = \frac{2\pi N T}{60000} \text{ KW}$$

$$BHP = \frac{2\pi N (w_1 - w_2) * D * g}{6000} \text{ KW} = \text{ KW}$$

T = ($w_1 - w_2$) * D * g

Rope with Hanger Weight = 1.0 Kg

w_1 = Weight + Hanger with Rope Reading

w_2 = spring balance

D = Diameter of the brake drum

N = Number of R.P.M. of brake drum

2. Efficiency of Turbine

$$\eta = \frac{\text{BHP (Output)}}{\text{HP Hydraulic (Input)}} * 100\ \%$$

PRACTICAL NO: - 06

AIM: To draw the operating characteristics of the pump for different flow rates and find the optimum conditions for operating the gear pump

Experimental Procedure:

- Check the gate valve provided on the delivery pipe and keep it in fully open position.
- Fill the storage tank with oil and empty the measuring tank.
- Switch on the pump and set the desired speed with the help of variable speed D.C. drive and note down the speed of the pump in RPM with the help of digital RPM indicator.
- Also note down the power drawn by the pump with the help of Watt meter.
- Operate the gate valve on the delivery line to increase the delivery head of the pump and note down both Vacuum gauge reading and Pressure gauge reading.
- Initially allow flow from the delivery line to drain directly into the sump tank with the help of liquid diverting arrangement, then accumulate the predetermined volume in the measuring tank and also the time required for the same with the help of self graduated level indicator and stop watch.
- Repeat the experiment for different positions of delivery line valve (delivery head) while maintaining the speed of the pump constant.
- Change the speed of the pump and repeat the above procedure.

TECHNICAL SPECIFICATIONS OF GEAR PUMP TEST RIG:

The test rig should consist of following:

- The set up consists of gear pump of size 25 mm (1”) X 25 mm (“1”) coupled to 3 HP 3 Phase, 440 V. Motor.
- M.S. Collecting tank with anti corrosive lining of size 0.3m X 0.3m X 0.6m.
- Reservoir of size 0.5m X 0.5m X 0.5m made from 10 gauge M.S. sheet.
- All components should be mounted on strong and sturdy M.S. stand.
- Instruction manual should be furnished.

Observation Table:

RPM of the Pump:

Sr. No.	Vacuum Gauge Reading (kg/cm^2)	Pressure Gauge Reading (kg/cm^2)	Total Static Head (m)	Initial Level of Liquid (cm)	Final Level of Liquid (cm)	Time θ (sec)	Volum. Flow Rate (Q) (m^3/s)	Power Input P (W)
1								
2								
3								
4								
5								

Calculations:

For each run, calculate

(1) Actual Capacity of the pump, Q_a , m^3 / sec

Q = [(Final level of liquid - Initial level of liquid)*C/s area of the measuring tank] / Time

=

=

(2) Theoretical Capacity of the pump, Q_{th} , m^3 / sec

(3) Total head developed H_s, m

H_s = (Pressure Gauge Reading + Vacuum Gauge Reading) / ρ

=

=

(4) Volumetric Efficiency, η_Q

(5) Power Output = $Q_a * \rho * H_s$

=

=

(6) Overall Efficiency of Pump & motor, η

= (Power Output/ Power Input) *100

=

=

Graph:

Plot Total head developed (H_s), as a function of capacity (Q_a).

Plot Efficiency (η), as a function of capacity (Q_a).

Plot Theoretical Power Input (P), as a function of capacity (Q_a).

Double Acting Reciprocating Pump

Pump bore	44.5 mm = 0.0445m
Pump stroke	35mm = 0.035m
Flow capacity	1590 lit/h
Total suction	25ft = 7.62m
Total head	100ft = 30.48m
Max. Pressure capacity	6-7 kg/cm2
Suction pipe	25 mm = 0.025m
Delivery pipe	25 mm = 0.025m
Pump speed	250 RPM
HP required	1/3 to ½ hp
Motor speed	1440 RPM
V-belt size	A48
Material used	CI, SS, brass

Sump Tank

Dimension – 100 *100 * 50 mm

Max. Storage – 500lit

Material – M.S

Measuring Tank

Dimension – 40 * 40 *50 mm

Max. Storage – 80lit

Material – M.S

PRACTICAL NO: - 07

AIM: To draw the operating characteristics of the pump for different flow rates and find the optimum conditions for operating the reciprocating pump.

EXPERIMENTAL PROCEDURE:

⇒ Check the valves of the experimental setup.

⇒ Fill the storage tank with liquid. Switch on the pump and by operating the valve on the delivery line adjust the flowrate.

⇒ Note down the initial level of the liquid in the tank at time zero (i.e. at the time of starting the stop watch).

⇒ When steady conditions are reached note the readings of the pressure gauge, vacuum gauge, final liquid level and time. Also note the energy input for this time from energy meter reading.

⇒ Take watt meter reading under no flow conditions for the same time.

⇒ Repeat the experiment for different positions of delivery line valve.

OBSERVATION TABLE:

1. RPM :______________
2. Area of measuring tank : A = m^2

Sr. No.	Vacuum Gauge Reading (kg/ cm^2)	Pressure Gauge Reading (kg/ cm^2)	Total Static Head (H_p) (m)	Initial Level of Liquid (cm)	Final Level of Liquid (cm)	Time θ (sec)	Power Input P (kWh1)	Power Input (P) (kWh2)
1								
2								
3								
4								
5								

CALCULATIONS:

For each run, calculate

(1) Actual Capacity of the pump, $\mathbf{Q_a}$, m^3 / sec

Q = [(Final level of liquid - Initial level of liquid)*C/S of the measuring tank] / Time

(2) Theoretical Capacity of the pump, $\mathbf{Q_{th}}$, m^3 / sec

$\mathbf{Q_{th}}$ = 2ALN/ 60

Where,

A	=	Area of piston (m^2)
L	=	Stroke Length of piston (m)
N	=	RPM of pump

(3) Total head developed $\mathbf{H_p}$, m

H_p = (Pressure Gauge Reading + Vacuum Gauge Reading)

(4) Slip = $Q_{th} - Q_a$

(5) Percentage Slip = $\left[1 - \frac{Q_a}{Q_{th}}\right] * 100$

(6) Volumetric Efficiency of the Pump η,

= Q_a / Q_{th} * 100

(7) Power Output

$$= \rho\left(\frac{2ALN}{60}\right) g \left(h_s + h_d\right)$$

(8) Power Input

= (kWh2 – kWh1)*3600*1000/time(sec)

(9) Overall Efficiency of Pump & Motor

= (Power Output/ Power Input) * 100

CONVERSIONS

- TO CONVERT PRESSURE GAUGE READING FROM kg/cm^2 TO MWC FOLLOW AS BELOW:
  1 kg/cm^2 = 393.70 in water

  1 inch = 0.0254 meter

- TO CONVERT VACUUM GAUGE READING FROM mmHg TO MWC FOLLOW AS BELOW:
  1 mmHg = 0.5352 in water

  1 inch = 0.0254 meter

- 1cm = 0.01meter
- 1 liter = 0.01 m^3
- TO CONVERT M^3/SEC TO LPH FOLLOW AS BELOW:

1M^3/SEC	=	3600000LITRE/HOUR
	=	60000LITRE/MIN
	=	1000LITRE/SEC
	=	3600M^3/HOUR
	=	60 M^3/MIN

PRACTICAL NO: - 08

AIM: To draw the operating characteristics of the pump for different flow rates and find the optimum conditions for operating the centrifugal pump.

INTRODUCTION:

Transport of fluid through closed conduit is a common feature in all chemical industries. It may be necessary to move a liquid against gravity force i.e. into a pressure vessel,; or pump it out from a vessel under vacuum as in the case of evaporators. In all these cases, there will be additional loss of energy due to friction as the liquid flows through conduits, fittings and valves. To ensure fluid movement, energy has to be supplied to fluid from an external source. The centrifugal pumps are the most widely used in chemical industries. It has many advantages. It is simple to operate, gives an uniform flow rate, occupies small floor space and has low maintenance cost. It can be used either with a motor or with turbine drive.

The capacity of the pump is defined as the volume of the fluid handled per unit time. For incompressible fluids it is given in liter per minute. For compressible fluids, the capacity is given at the inlet temperature and pressure of the fluid.

The total head is the energy added by the pump to unit mass of the flowing stream. The head is expressed in units of length. For a steady incompressible flow the total head H is given by,

$$H = \frac{P_2 - P_1}{\rho g} + \frac{V^2{}_2 - V^2{}_1}{2g} + (Z_2 - Z_1)$$

where point 1 is taken as any point before pump on the suction line and point 2 is any point on the delivery line.

Theoretical energy supplied is given by the product (multiplication) of capacity per unit time expressed as kg/ s, the total head, H_1 expressed in meter; g accelerations due to gravity in m/ s^2 and θ, the time of pumping in sec.

Efficiency is the ratio of theoretical energy to the actual energy measured by a watt meter.

The operating characteristics of a pump are shown by plotting the head developed H, the power supplied W and the efficiency η against the flow rate Q. The optimum conditions for operating a pump are at the conditions of maximum efficiency.

APPARATUS:

Centrifugal pump, D. C. motor, rpm meter, vacuum gauge, pressure gauge, energy meter, sump tank, measuring tank.

PROCEDURE:

⇒ Check that all valves of the experimental setup are fully open.

⇒ Fill the storage tank with liquid. Switch on the pump and operate it at selected value of RPM.

⇒ By operating the valve on the delivery line note down the pressure at fully open and shut-off condition of valve.

⇒ Take five to six points between this pressure range for experiment.

⇒ For particular set pressure, note down the initial and final level of the liquid in the tank for particular time interval (use stopwatch for recording time).

⇒ Also note down the readings of the vacuum gauge.

⇒ Also note the energy input for this time from energy meter.

⇒ Repeat the experiment for different positions of delivery line valve.

⇒ Also repeat experiment for different value of RPM.

PRECAUTIONS:

⇒ Keep rpm regulator at zero before starting the pump.

⇒ Never shut off delivery valve more than a minute.

OBSERVATION

Area of measuring tank = 0.16 sq. m.

OBSERVATION TABLE:

RPM of motor = ___________

Sr. No.	Initial Level of Liquid (m)	Final Level of Liquid (m)	Time θ (sec)	Pressure Gauge Reading (kg/cm^2)	Vacuum Gauge Reading (mmHg)	Initial Energy Meter Reading (E_0) kWh	Final Energy Meter Reading (E) kWh
1							
2							
3							
4							
5							

CALCULATIONS:

For each run, calculate

(1) Capacity of the pump, Q, m^3 / sec

Q = [(Final level of liquid - Initial level of liquid)*C/S area of the calibration tank] / Time

(2) Total head developed **H**, metre of water column

H = Pressure Gauge Reading + Vacuum Gauge Reading

(3) Theoretical energy supplied $E_T = (Q * \rho * H * / 3.67 \times 10^5)$ kw

Where,

Q = Flow rate in m^3 / hr

ρ = Density in kg/m^3

H = Head in meter of water

(4) Actual Energy supplied $E_A = (E - E_0)$ kWh * 3600 / θ

(5) Efficiency (%) $\eta = E_T / E_A * 100$

CONVERSIONS

- TO CONVERT PRESSURE GAUGE READING FROM kg/cm^2 TO MWC FOLLOW AS BELOW:

  1 kg/cm^2 = 393.70 in water

  1 inch = 0.0254 meter

- TO CONVERT PRESSURE GAUGE READING FROM mmHg TO MWC FOLLOW AS BELOW:

  1 mm of Hg = 0.5352 in water

  1 inch = 0.0254 meter

  - 1cm = 0.01meter

- 1 liter = 0.01 m^3

TO CONVERT M^3/SEC TO LPH FOLLOW AS BELOW:

1M^3/SEC	=	3600000 LITRE/HOUR
	=	60000 LITRE/MIN
	=	1000 LITRE/SEC
	=	3600 M^3/HOUR
	=	60 M^3/MIN

PRACTICAL NO: - 09

AIM:. Explanation on centrifugal and axial flow compressor

Quiz:

Centrifugal compressor

(1) Construction and working of centrifugal compressor,
(2) Derive ideal energy transfer (Euler's equation) with velocity diagram,
(3) Sleep and sleep factor,
(4) Define pre-whirl ,and power input factor,
(5) Define surging and chocking,

Axial flow compressor

(1) Construction and operation of axial flow compressor,
(2) Draw velocity diagram and define work-done factor,
(3) Drag and lift and co-efficients
(4) Define stalling,
(5) Explain forced vertex and free vertex

Ans: -

PRACTICAL NO: -10

AIM: Explanation on various Hydraulic systems.

Quiz:

(1) Explain the Hydraulic Press with construction and working,

(2) Explain the Hydraulic Accumulator,

(3) Explain Fluid Coupling,

(4) Explain the Torque convertor,

(5) Explain the Air lift pump,