

Experimental Investigation of Centrifugal Pump Characteristics

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Abstract – The pumps are divided into two main groups as positive displacement namely volumetric pumps and dynamic i.e. centrifugal pumps. Volumetric pumps are generally used to transport hydraulic oils, while centrifugal pumps are used to transport clean or dirty water. Pumps are the tools that are used to move liquid fluids from one place to another in many areas like as cars, industry, construction sites, agriculture, and homes. Therefore, determination of the hydraulic pump's characteristics under various working conditions, including manometric head, overall efficiency, specific speed, and net positive suction head, is crucial for development, operational limitations, and durability. Experimental investigation is a more realistic method, although it is possible to determine the hydraulic pump characteristics by means of mathematical methods. Therefore, an experimental investigation is performed for the determination of the centrifugal pump characteristics in this study by using an experimental setup. The findings indicate that the hydraulic power given to the water and the electrical power used by the pump both increase up to a particular flow rate and then decline. Additionally, it is found that manometric head of the pump falls even as the overall flow losses rise as the flow rate rises. On the other hand, overall efficiency of the pump enhances at the specific flow rate and then it reduces with the increased flow rate while specific speed of the pump varies about linearly. Moreover, it is finally noticed that the tested centrifugal pump works in cavitations at all flow rates.

Keywords: Hydraulic Machines, Hydraulic Turbines, Centrifugal Pumps, Pump Characteristics, Cavitations

I. INTRODUCTION

Hydraulic machines are separated into two main classes as hydraulic turbines and pumps. While smaller hydraulic turbines are used to generate mechanical power using hydraulic lubricants in industries and vehicles, larger hydraulic turbines are often used in hydroelectric power plants to generate energy from the hydraulic power of water [1–3]. On the other hand, pumps are the tools that are used to move liquid fluids from one place to another in many areas like as cars,

industry, construction sites, agriculture, and homes. The pumps are divided into two main groups as positive displacement i.e. volumetric pumps and dynamic i.e. centrifugal pumps. While volumetric pumps operate under constant torque, centrifugal pumps operate in various rotational speeds, heads and flow rates because of variable torque capacity. Volumetric pumps are generally used to transport hydraulic oils, while centrifugal pumps are used to transport clean or dirty water. A centrifugal pump is a rotor dynamic pump that uses a rotating impeller to

increase the pressure of a fluid. This pump is constituted by rotor inside a carcass. The fluid enters the pump impeller along or near the rotating axis and is accelerated by the impeller flowing radially outward into a diffuser or volute chamber, and subsequently into the downstream piping system. It works by converting the kinetic energy into potential energy measurable as the static fluid pressure at the outlet of pump. Centrifugal pumps are used for large discharge through smaller heads. To achieve larger specific work, more pumps are can connected in series; however, if the aim is larger flow, they can be connected in parallel [4–11]. Moreover, centrifugal pumps can be employed as a turbine with reverse operation [12–13]. Therefore, determination of the characteristics of centrifugal pumps under various working conditions is critical. This study aims to investigate experimentally the characteristics of a centrifugal pump such as manometric head, overall

efficiency, specific speed and net positive suction head.

II. EXPERIMENTAL SETUP AND MEASUREMENT

The experimental setup is a closed-circuit system consisting of a centrifugal pump driven by an electric motor, a water tank, connecting pipes, measuring devices and a control unit. The schematic layout of the test setup is as in Fig. 1. The setup has a centrifugal pump driven by with an electric motor power of 3 kW, a maximum manometric head of 45 m, a speed of 2850 rpm and a maximum flow rate of 18 m³/h, a flow meter measuring capable of between 5–25 m³/h, pressure measuring elements located in pump inlet and outlet, a 250 liter water tank, diameter of 2 inches connection pipes, valves and strainer. In the experiment, the flow rate was adjusted with the valve in the discharge line and measurements were made at different flow rates.

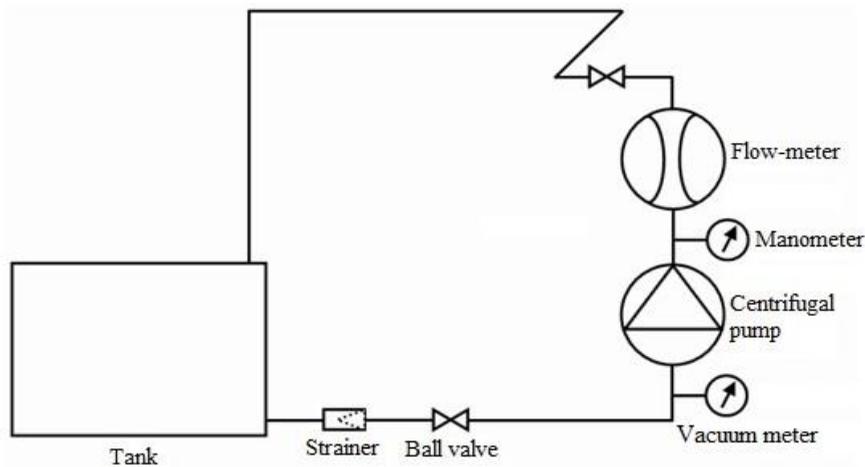


Fig. 1. Experimental setup

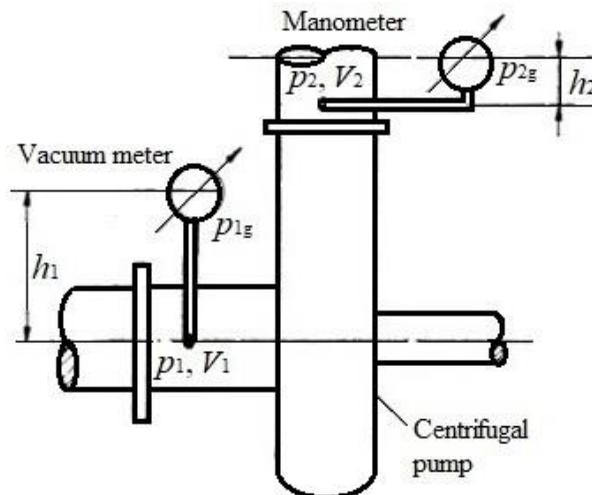


Fig. 2. Measurement of gauge pressure

Measurement of gauge pressure on inlet and outlet of the pump in the setup is shown in Fig. 2 and numerical values of the setup are given in Table 1. In experiments; ambient pressure and temperature (p_{amb} and T_{amb}), pump inlet and outlet pressures (p_{1g} and p_{2g}), the flow rate of the water (Q), the rotational speed of the pump (N) and the current and voltage (I and U) of consumed electricity by the electric motor that runs the pump are measured at different operating conditions. The measured values in the experiments are given in Table 2.

Table 1. Numerical values of experimental setup

Suction pipe diameter (D_1)	0.0525 m	
Discharge pipe diameter (D_2)	0.0525 m	
Suction pipe length (L_1)	0.3 m	
Discharge pipe length (L_2)	1.25 m	
Elevation of pump inlet and outlet (Δz)	0.3 m	
Height of vacuum meter (h_1)	0.05 m	
Height of manometer (h_2)	0.05 m	
Local loss element	Number	Loss coefficient (K_{LL})
Tank outlet	1	0.6
Strainer	1	3
Threaded elbow-90°	4	1.2
Threaded T-spur	2	1.2
Threaded T-branching	1	2
Conical union	2	0.3
Plain sleeve	1	0.3
Ball valve	2	10
Flow meter	1	12

Table 2. Measured values in the experiments

Ambient conditions	p_{amb} [kPa]	98.2				
	T_{amb} [°C]	19				
Experiment no	1	2	3	4	5	
p_{1g} [bar]	-0.1	-0.2	-0.4	-0.6	-0.7	
p_{2g} [bar]	3.6	3.4	3	2.5	0.5	
Q [m ³ /h]	6	9	12	15	18	
N [rpm]	2850	2850	2850	2850	2850	
U [volt]	380	380	380	380	380	
I [amper]	6.5	7.1	8.6	8.2	7.2	

III. CALCULATION OF PUMP CHARACTERISTICS

Gauge pressure is measured in the experiments, but absolute static pressure is used in the calculations. The absolute pressures from the

measured gauge pressures are determined as follow.

$$p_1 = p_{amb} \pm p_{1g} + h_1 \cdot \gamma_{water} \text{ [Pa]} \quad (1)$$

$$p_2 = p_{amb} + p_{2g} + h_2 \cdot \gamma_{water} \text{ [Pa]} \quad (2)$$

Flow rate of the pump in the system is measured with a flow meter and the average velocity of the water circulating in the suction and discharge pipes (V_1 and V_2) is determined as follow.

$$Q = AV \text{ [m}^3\text{/s]} \quad (3)$$

$$V_1 = Q/A_1 \text{ [m/s]} \quad (4)$$

$$V_2 = Q/A_2 \text{ [m/s]} \quad (5)$$

Here; A_1 and A_2 are the cross-sectional areas of the pipes in the pump suction and discharge lines. Manometric head (H_m) is defined as the energy gained by a unit weight of liquid between the inlet and outlet of the pump or the difference between the energies of the unit weight of liquid at the inlet and outlet of the pump. According to this definition, manometric height is determined as follow.

$$H_m = H_2 - H_1 = \frac{p_2 - p_1}{\gamma_{water}} + (z_2 - z_1) + \frac{V_2^2 - V_1^2}{2g} \text{ [m]} \quad (6)$$

Total losses in the test setup; it consists of two parts i.e. continuous friction losses and local losses, as given in the equation follow.

$$h_{L, total} = h_{L, continuous} + h_{L, local} \text{ [m]} \quad (7)$$

Friction losses in pipe flow are calculated differently for laminar and turbulent flows. The Reynolds (Re) number is used to determine the type of flow. Re number is a dimensionless and is defined as the ratio of inertial forces to viscous forces which is determined for the circular pipe flow as follow.

$$Re = \frac{VD}{\mu} = \frac{\rho VD}{\nu} \quad (8)$$

Here; V is the average velocity [m/s], D is the pipe diameter [m], μ is the dynamic viscosity [N.s/m²=Pa.s], ρ is the fluid density [kg/m³] and ν is the kinematic viscosity [m²/s].

In pipe flow, it is assumed that there become laminar flow up to Re number of 2000 and turbulent flow at the values above Re number of 2000. The Darcy-Weisbach equation given below is used to determine the continuous friction losses in pipes.

$$h_{L, \text{continuous}} = f \frac{L V^2}{D 2g} \text{ [m]} \quad (9)$$

In the above equation, f is the friction factor and it is determined from the following equations for laminar and turbulent flow, respectively.

$$\text{Laminar flow: } f = \frac{64}{\text{Re}} \text{ [-]} \quad (10)$$

$$\text{Turbulent flow: } \frac{1}{\sqrt{f}} = -1,8 \log \left[\frac{6,9}{\text{Re}} + \left(\frac{\varepsilon/D}{3,7} \right)^{1,11} \right] \quad (11)$$

The surface roughness value was taken as $\varepsilon = 0.00025$ for the pipes in the experimental setup. The empirical equation given below is used to determine the local losses caused by the components such as tank outlet, strainer, ball valve, elbow, T and flow meter in the test setup.

$$h_{L, \text{local}} = K_{LL} \frac{V^2}{2g} \text{ [m]} \quad (12)$$

In the above equation, K_{LL} is the local loss coefficient and the values of K_{LL} for the components in the experimental setup are given in Table 1.

The hydraulic power transferred to the fluid in the pumps is determined as depending on the flow rate and the manometric height as follow.

$$P_h = \rho g Q H_m = \gamma Q H_m \text{ [W]} \quad (13)$$

The electrical power drawn by the electric motor driving the pump from the network is determined as follows according to the voltage and current values.

$$P_e = I U \text{ [W]} \quad (14)$$

The overall efficiency for the pump is determined from the equation below as the ratio of hydraulic power to electrical power.

$$\eta_o = \frac{P_h}{P_e} \times 100 = \frac{\rho g Q H_m}{I U} \times 100 \text{ [%]} \quad (15)$$

Specific speed is the number of revolutions of the pump, which is geometrically similar to a model pump, to press a unit amount of fluid to the unit of manometric head. Specific speed varies with the changing of flow rate and manometric height for the same pump and it is determined as follows.

$$N_s = \frac{N \sqrt{Q}}{H_m^{3/4}} \text{ [d/dk]} \quad (16)$$

Here, the speed of the centrifugal pump (N) in the test setup is fixed as 2850 rpm.

To avoid cavitations when the pressure at the pump inlet is negative (vacuum) or the pumped liquid temperature rises, it is necessary to check whether the net positive suction head of the system ($(NPSH)_{\text{system}}$) is greater than the net positive suction head of the pump ($(NPSH)_{\text{pump}}$) of the system. The $(NPSH)_{\text{system}}$ is determined using the following equation.

$$(NPSH)_{\text{system}} = \left(\frac{P_1}{\rho g} + \frac{V_1^2}{2g} \right) - \frac{P_v}{\rho g} = (H_{\text{amb}} + H_{\text{suction}} - H_v - h_{L, \text{suction}}) \text{ [m]} \quad (17)$$

To avoid cavitations value of the $(NPSH)_{\text{system}}$ should be higher than $(NPSH)_{\text{pump}}$. The $(NPSH)_{\text{pump}}$ must be supplied by the manufacturer. If not given by the manufacturer, it can be determined from the empirical equation given below.

$$(NPSH)_{\text{pump}} = 12,2 \times 10^{-4} Q^{\frac{2}{3}} N^{\frac{4}{3}} \text{ [m]} \quad (18)$$

IV. RESULTS AND DISCUSSION

The centrifugal pump characteristics were calculated by using the equations given above and the graphics drawn from the calculated centrifugal pump characteristics were presented in this section. Fig. 3 shows that the variation of manometric head and total flow losses with pump flow rate. The manometric head decreases with the increasing of flow rate as seen in Fig. 3. It is considered that the decrease in pump head is sourced from the increase of the losses due to increasing velocity with flow rate (discharge) of the pump. The increase in the losses is apparently seen in Fig. 3. The pump head is solely a function of the physical characteristics of the pump. The system curve (losses) is completely dependent on the size of pipe, the length of pipe, the number and location of elbows, and other factors. Where these two curves intersect is the natural operating point (seen in Fig. 3). This is where the pump pressure matches the system losses and everything is balanced.

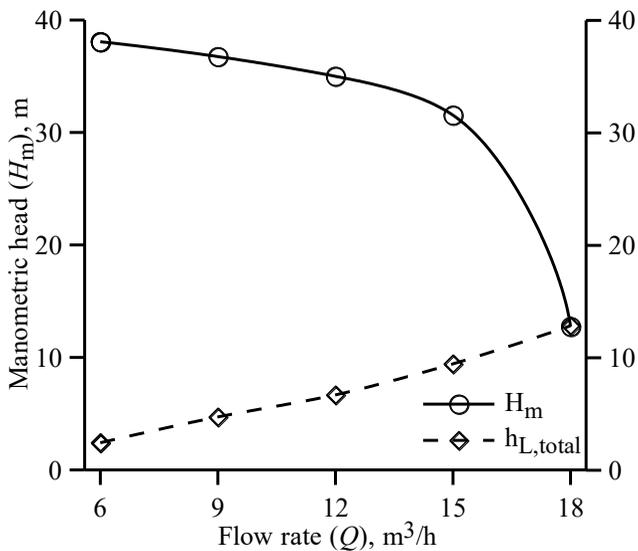


Fig. 3. Variation of the manometric head and total flow losses with flow rate

Fig. 4 shows that the variation of electrical power driven the pump and hydraulic power transferred to water with pump flow rate. The electrical power and hydraulic power increase up to certain flow rate (15 m³/h) and it decrease suddenly at the maximum discharge of the pump as seen in Fig. 4. A more work is being done and the horsepower requirement increases because flow rate and pump head increase. However, the pump reaches the maximum flow rate the cavitations will begin and thus power decreases rapidly. It is considered that the increase of the frictional and flow losses cause an additional contribution to the rapid decrease in power at the maximum pump discharge.

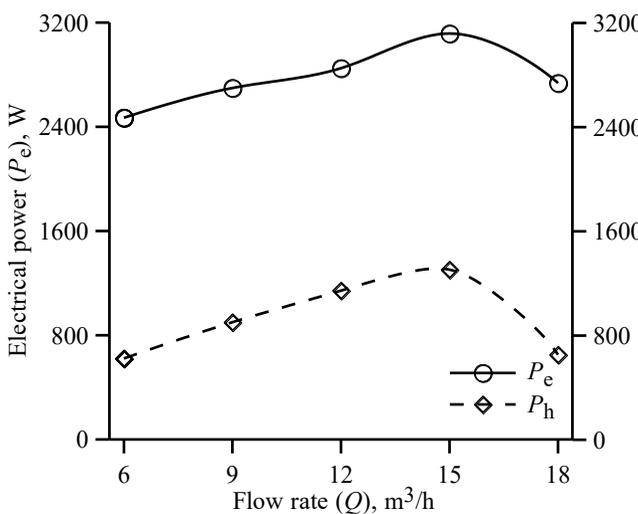


Fig. 4. Variation of electrical and hydraulic power with flow rate

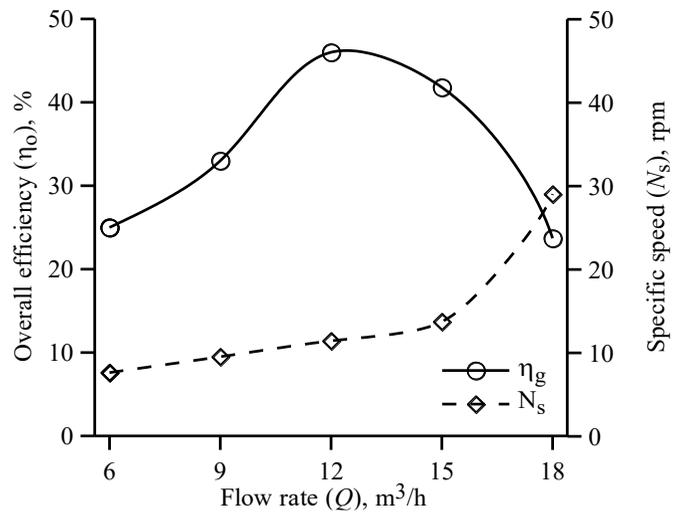


Fig. 5. Variation of overall efficiency and specific speed with flow rate

Fig. 5 shows that the variation of pump overall efficiency and specific speed with pump flow rate. The efficiency of the pump increases to the best efficiency point (BEP) and then decrease with the increase of flow rate as seen in Fig. 5. This is typical variation of overall efficiency in centrifugal pumps. Additionally, it is considered that working in intensive cavitations at the high flow rates can be additional contribution to the reduction of overall pump efficiency. On the other hand, specific speed increases with the increase of pump flow rate as seen in Fig. 5. The higher the specific speed selected for a given set of operating condition, the higher the pump efficiency and therefore the lower the power consumption. Except for other considerations the tendency should be to favor higher specific speed selection from the point of view of energy conservation.

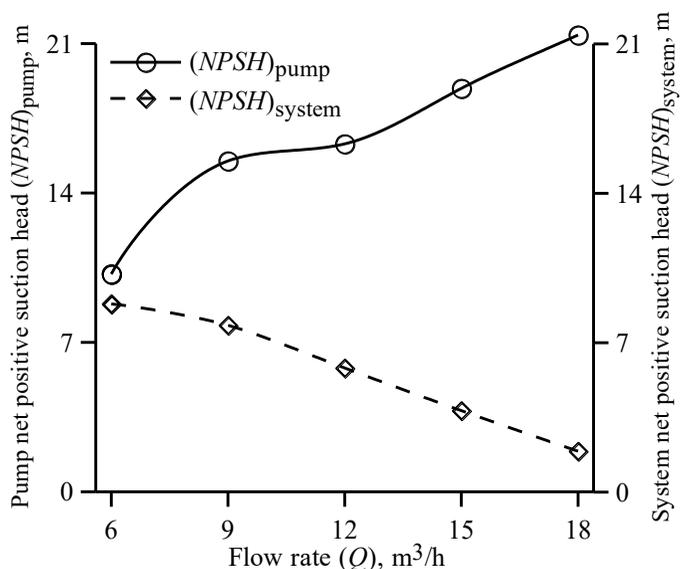


Fig. 6. Variation of pump and system net positive suction head with flow rate

Fig. 6 shows that the variation of pump net positive suction head $(NPSH)_{\text{pump}}$ and system net positive suction head $(NPSH)_{\text{system}}$ with pump flow rate. The $(NPSH)_{\text{pump}}$ increases with increase of pump flow rate as seen in Fig. 6. This is depending on the design characteristics of the centrifugal pump and it is specific for any pump. On the other hand, the $(NPSH)_{\text{system}}$ decreases with increase of pump flow rate as seen in Fig. 6. It is considered that this is sourced from the reduction of pressure and also temperature inlet of the pump with the increasing of pump discharge (i.e. flow velocity) due to the energy balance. The decrease in temperature obligate the vaporization of the water inlet of the pump so the $(NPSH)_{\text{system}}$ decreases. Moreover, the $(NPSH)_{\text{system}}$ is always lower than the $(NPSH)_{\text{pump}}$. This means that the pump works in cavitations during the experiments. Cavitation phenomenon is basically a process formation of bubbles of a flowing liquid in a region where the pressure of the liquid falls below its vapor pressure and it is the most challenging fluid flow abnormalities leading to detrimental effects on both the centrifugal pump discharge characteristics as well as physical characteristics. In this low pressure zones are the first victims of cavitation. Due to cavitation pitting of impeller occurs and wear of internal walls of pumps occurs due to which there is creation of vibrations and noise are there. Due to this there is bad performance of centrifugal pump is there [9].

V. CONCLUSIONS

In the present study, the characteristics of a centrifugal pump such as manometric head, overall efficiency, specific speed and net positive suction head have been investigated experimentally. The following conclusions can be summarized from the results of the study.

- It was determined that the manometric head of the centrifugal pump decreases with the increasing of flow rate due to increase of the flow losses with flow rate (discharge) of the pump.
- It was determined that electrical and hydraulic power increased up to certain flow rate of the pump and it decrease suddenly at

the maximum discharge of the pump due to increase of the frictional and flow losses.

- It was determined that overall efficiency of the pump increased to the best efficiency point and then decreased with the increase of flow rate due to working in intensive cavitations at the high flow rates, while the specific speed of the pump increased continuously with the increase of pump flow rate.
- It is determined that system net positive suction head $(NPSH)_{\text{system}}$ decreased continuously with increase of pump flow rate and the $(NPSH)_{\text{system}}$ was always lower than the $(NPSH)_{\text{pump}}$. According to this, it is assessed that the tested pump worked in cavitations during the experiments.
- It was concluded that the pipe diameters in the experimental installation should be increased in order to prevent cavitation.

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