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Condition Monitoring of a Ship Board Centrifugal Pump using Parametric Approach

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ABSTRACT

The centrifugal pump which consist of a stationary pump casing and an impeller has the main function of guiding the liquid from suction nozzle through the centre of the impeller to the discharge nozzle. In marine application, this pump, this pump is prone to total breakdown as a result of variation in parameters and excessive vibration. In this work, a detail examination of these parameters is carried out by way of condition monitoring. A two horse power centrifugal pump in shipboard application was used to actualize this work. The data for head, speed, discharge rate and pump efficiency were obtained at the range of 10.19-2.04, 2750-2500rpm, 27.38-15.11 and 5.0-1.5% respectively. A computer program named CEPPA written in C^{++} programming language was developed. The results show that the system should be run within the 5.0-1.5% efficiency.

Keywords: *condition monitoring, centrifugal pump, parametric approach*

1. INTRODUCTION

Pump can be described as a device included in a piping arrangement which adds pressure energy to the existing energy in a fluid causing an increase in its pressure and hence driving it across a pipe and fitting to accomplish the required duty (Iwakiri, et al. 2009). In this sense, the pump can be regard as a device in the piping arrangement which raises the energy of a fluid resulting to an increase in its pressure and hence the driving force that moves the fluid across the numerous pipe fittings such as valves, filters cross connections and tank (Douglas, 2001) (Gilles, et al, 2003).

A pump transfer mechanical energy from some external source to the liquid flowing through it and losses occur in any energy conversion process. The energy transferred is predicted by the Euler Equation and the losses between fluid power and mechanical power of the impeller or runner (Boyanton, 2006). Thus, centrifugal pump may be taken losses of energy. A centrifugal pump delivers useful energy to the fluid on pumpage largely through velocity changes that occur as this fluid flows through the impeller and the associated fixed passage ways of the pump. It is converting of mechanical energy to hydraulic energy of the handling fluid to get it to a required place or height by the centrifugal force of the impeller blade.(Jackson, 2002)

The input power of centrifugal pump is the mechanical energy and such as electrical motor of the drive shaft driven by the prime mover or small engine. The output energy is hydraulic energy of the fluid being raised or carried. In a centrifugal pump, the liquid is forced by atmospheric or other pressure into a set of rotating vanes. A centrifugal pump consists of a set of rotation vanes enclosed within a housing or casing that is used to impart energy to a fluid through centrifugal force (Stepanoff, 2002).

1.1 Other Approaches

Berge (2008) drafted a work on effective pump monitoring in that work, foundation field bus and electronic device description language was adopted to enable efficient pump monitoring

Romana (2011) presented a model base approach for vibration analysis and condition monitoring (CM) of a centrifugal pump. Also in that work, FEM Technology was developed. However, the mathematical model and FEA Model were further built and integrated to the original centrifugal casing to ascertain the pump natural frequencies. In fact, vibration analysis of cavitation in a centrifugal pump is also depicted in Choi et al (2007).

Finally, Ishiodu and Ogbonnaya (2011), presented a comprehensive draft parametric consideration of marine diesel engine output for maximum operation. A program named MDEPAC was developed.

1.2 Approaches Used in this Work

The paper provides a comprehensive CM of centrifugal pump using parametric approach. A program named CEPPA written in C^{++} programming language was used to bring it to friction. CEPPA stands for Centrifugal Pump Parametric Analysis. This was carried out with a 2 Horse power centrifugal pump used in supply vessel near coastal voyage in Nigeria.

2. MATERIALS AND METHODS

The method applicable in this work is a comprehensive parametric analysis of physical properties of a centrifugal pump. This 2 horse-power centrifugal pump system is a single stage centrifugal pump of 6 vanes and 0.0480m^2 outlet areas. This parametric approach is adopted with strict adherence to the following designed parameter.

CEPPA, is a program code developed to monitor these parameters of a centrifugal pump. The program was written in a single loop iterative C^{++} computer program. The program helped to calculate the parametric aspects of the reference device which include all the physical properties such as outlet area, the impeller diameter water power total head, power, speed, quantity flow rates, and efficiency at various rates of flow etc. Furthermore, this program is capable of triggering alarm if any parameter is offset the design operational limit.

Specific speed is used to classify impellers on the basis of their performance, and proportions regardless of their actual size or the speed at which they operate.

Specific Speed, S:

$$S = \frac{n.\sqrt{Q}}{NPSH^{\frac{3}{4}}} \tag{1}$$

Capacity, volute flow rate of a pump is the amount of water pumped per unit time and it is also known traditionally as volume flow rate. The capacity is directly related with the velocity of flow in the suction pipe.

$$Capacity: Q = AV \tag{2}$$

where A and V are area of pipe and volume flow rate respectively.

2.1 Water Power and Shaft Power

The power imparted to the water by the pump is called waterpower. To calculate waterpower, the flow rate and the pump head must be known. This power is called brake power. The efficiency of the pump determines how much more power is required at the shaft.

The waterpower is determined from the relationship

$$N = \rho g H Q \tag{3}$$

The shaft power is:

$$Shaft\ power = water\ power / \eta_0 \tag{4}$$

Pump efficiency is

$$\eta_0 = \eta_m \times \eta_v \times \eta_r \tag{5}$$

Maximum shaft power is:

$$M_{Max} = \alpha_1 \rho g H Q / \eta_0 \tag{6}$$

α_1 is the safety factor in charge condition of the work of pump

Inlet diameter of impeller is:

$$D_1 = (1.1 \approx 1.5) K_0 \sqrt[2]{\frac{Q}{n}} \tag{7}$$

The value of K_0 is chosen

The outlet diameter of impeller is:

$$D_2 = 19.29 \left(\frac{n_{sopt}}{100}\right)^{2/6} \frac{\sqrt{2gH}}{n} \tag{8}$$

D_0 is the eye diameter of impeller.

$$D_0 = K_0 \sqrt[3]{\frac{Q}{n}} \tag{9}$$

Where K_0 is the constant parameter

The shaft diameter at the hub section is:

$$D_s = \sqrt[3]{\frac{T}{0.2E}} \tag{10}$$

The tensional moment is estimated by:

$$T = 9.65 \frac{N_{\max}}{n} \tag{11}$$

The hub diameter is:

$$D_{bt} = (1.5 \approx 2) dsh \tag{12}$$

The hub length is two times of the shaft.

$$L_{bt} = 2dsh \tag{13}$$

Inlet width of the impeller is:

$$b_1 = \frac{R_0}{2} \tag{14}$$

R_o is the radius of the impeller eye.

Outlet width of the impeller is:

$$b_2 = 0.78\left(\frac{n_{sopt}}{100}\right)^{1/2}\left(Q/n\right)^{1/3} \tag{15}$$

The hydraulic efficiency is;

$$\eta_r = 1 - \frac{0.42}{(\log D_0 - 0.172)^2} \tag{16}$$

Leakage head is:

$$H_T = H_T - \frac{V_2^2}{2g} - \frac{U_2^2}{8g} - \left(1 - \frac{D_y^2}{D_2^2}\right) \tag{17}$$

where H_T and D_y are the pressure head and seal diameter.

The pressure head is**:**

$$H_T = H/\eta_r \tag{18}$$

The seal diameter is:

$$D_y = D_0 + 10 \tag{19}$$

The minimum clearance between the war ring and casing is:

$$\delta = 10^{-3} D_y \tag{20}$$

Inlet blade angle of impeller is:

$$Tan\beta_1 = \frac{U_1}{V_{m1}} \tag{21}$$

The outlet angle of impeller is assumed as 40°.
Blade number:

$$Z = \frac{6.5\dfrac{D_2 + D_1}{D_2 - D_1}\sin(\beta_1 + \beta_2)}{2} \tag{22}$$

The radius of the impeller at outlet is:

$$R_A = D_2/2 \tag{23}$$

Require parameter to layout the impeller blade is:

$$\rho A = (R_{A2} - R_{B2})2/2 \times 1/(R_A \cos\beta_2 - R_B \cos\beta_1) \tag{24}$$

2.2 Theoretical Head

The Euler head is determined from zero to maximum theoretically attainable flow using.

The theoretical head:

$$H_{th} = 1/g\, U_2 V_{u2} \tag{25}$$

where U_2 and V_{u2} are outlet tangential velocity and whirl velocity.

Whirl velocity:

$$V_{u2} = U_2 - V_{m2}\cot\beta_2 \tag{26}$$

where V_{m2} and β_2 are outlet flow velocity and outlet blade angle.

2.3 Net Theoretical Head

If the slip factor is known, the net theoretical head may be obtained from Euler's head. It is possible to relate the theoretical characteristic obtained from Euler's equation to the actual characteristic for various losses responsible for the difference.

The net theoretical head is calculated by:

$$H_{thn} = \frac{U_2 V_{u2}}{g} \tag{27}$$

The whirl velocity at the outlet is;

$$V_{u2} = U_{2\sigma} - V_{m2}\cot\beta_2 \tag{28}$$

where, σ is the slip value.

Slip value is obtained by using the following equation:

$$\sigma = 1 - \frac{(\sin\beta_2)^{1/2}}{\beta_1 Z^{0.7}} \tag{29}$$

2.4 Shock Losses

The major loss considered is shock losses at the impeller inlet caused by the mismatch of fluid and metal angles. Shock losses can be found everywhere in the flow range of the pump.

Shock Losses are given by following equation:

$$h_s = K(Q_s - Q_N)^2 \tag{30}$$

Maximum flow rate:

$$Q_N = \pi D_1 b_1 V_{m1} \tag{31}$$

The shut off head:

$$H_{Shut-off} = \frac{U_2^2 - U_1^2}{2g} \tag{32}$$

In the shut –off condition,

$$Q = 0$$
$$h_s = H_{Shut-off}$$

So, shock losses equation is formed by substituting in equation 16.

2.5 Impeller Friction Losses

The impeller was designed that the width of the impeller would become small and the friction loss at the flow passage would become large. Therefore to relive the increase in friction loss, radial flow passage on the plane of the impeller was adopted.

The impeller friction losses are:

$$h_1 = \frac{b_2(D_2 - D_1)(V_{rl} + V_{r2})}{2\sin\beta_2 H_r 4g} \tag{33}$$

The hydraulic radius:

$$H_r = \frac{b_2\left(\frac{\pi D_2}{Z}\right)\sin\beta_2}{b_2 + \left(\frac{\pi D_2}{Z}\right)\sin\beta_2} \tag{34}$$

The inlet relative velocity is;

$$V_{r1} = \frac{V_{m1}}{\sin\beta_1} \tag{35}$$

The outlet relative velocity:

$$V_{r2} = \frac{V_{m2}}{\sin\beta_2} \tag{36}$$

2.6 Volute Friction Losses

This loss results from a mismatch of the velocity leaving the impeller and the velocity in the volute throat. If the velocity approaching the volute throat is larger than the velocity at the throat, the velocity head difference is less.

The volute friction losses:

$$h_2 = \frac{C_v V_3^2}{2g} \tag{37}$$

The volute throat velocity:

$$V_3 = \frac{Q}{A_3} \tag{38}$$

The volute throat area:

$$A_3 = V_{u2}\left(\frac{D_2}{D_3}\right) \tag{39}$$

The volute flow coefficient is,

$$C_v = 1 + \left(0.02 \times \frac{L_{vm}}{D_{vm}}\right) \tag{40}$$

The volute circumferential length:

$$L_{vm} = \frac{\pi D_i}{8} \tag{41}$$

The diameter of volute tangent circle is get from the geometry of volute casing.

The volute mean diameter:

$$D_{vm} = \frac{D_i}{8} \tag{42}$$

2.7 Disk Friction Losses

The disk friction power is divided by the flow rate and head to be added to the theoretical head when the shaft power demand is calculated.

The disk friction loss is;

$$h_3 = \frac{f\rho\omega^3\left(\frac{D}{2}\right)^5}{10^9 Q_s} \tag{43}$$

2.8 Recirculation Losses

The recirculation loss coefficient depends on the piping configuration upstream of the pump in addition to the geometrical details of the inlet. The power of recirculation is also divided by the volume flow rate, in order to be converted into a parasitic head.

The head of recirculation is;

$$h_4 = K\omega^3 D_1^2 \left(1 - \frac{Q_s}{Q_0}\right)^{2.5} \tag{44}$$

2.9 Actual Head

The actual pump head is calculated by subtracting from the net theoretical head all the flow losses which gives the actual head/flow rate characteristic provided it is plotted against. Therefore;

The actual pump head is;

$$H_{act} = H_{thn} - (h_s + h_1 + h_2 + h_3 + h_4) \tag{45}$$

For the case of the experiment, the value of the coefficient of discharge was estimated by a heterodox method thus: a rectangular container having a volume of 715.91cm^3 was used to collect water from the delivery pipe to the reservoir

3. ANALYSIS AND DISCUSSION OF RESULTS

The results obtained, is presented in table 1 and plots of the pump performance characteristics are in general agreement with the plots of centrifugal pump characteristic curves. The functional relationship between the head, H, and discharge, Q of a centrifugal pump suggests that the plot of H vs Q will be a straight line having a positive intercept and a negative slope.

The actual head developed by the pump is only a fraction of the theoretical head as given by Euler's equation. The ratio of the actual to the theoretical being the hydraulic efficiency,

The hydraulic efficiency is not constant but varies with the discharge and the head, which equally contributes to the curve nature of the relationship between H and Q as depicted in table 1

The plot of efficiency, η vs discharge, Q, is in agreement with previous experimental results observed for centrifugal pumps. The little variation may have more to do with the state of the pump used, the electric motor that drives it and the losses inherent in the numerous fittings incorporated into the unit.

Table 1: Centrifugal pump total head, power and efficiency at various rates of flow

H (m)	Speed (rpm)	Q (m^3/s)	P_w (watts)	η (%)
2.04	2750	27.38	5.48	1.5
4.08	2750	22.11	8.85	3.2
6.12	2650	18.94	11.37	4.1
8.15	2650	17.13	13.70	4.7
10.19	2500	15.11	15.10	5.0

The table 1 above shows the experimental value of the centrifugal pump total head, power, speed, quantity flow rates, and efficiency at various rates of flow

The output of a pump running at a given speed is the flow rate delivered by it and the head developed. A plot of head, H against flow rate, Q at constant speed forms the fundamental performance characteristics of a pump as shown in figure 4.1 below.

In order to achieve this performance, a power input is required which involves efficiency of energy transfer. Thus, it is useful to plot also the efficiency, η and power input (P_s) against the Head, H as shown in figure 4.2 and figure 4.3 respectively as shown below.

Such a complete set of performance characteristics of the designed and constructed centrifugal pump demonstration unit is shown in figure 4.4 below.

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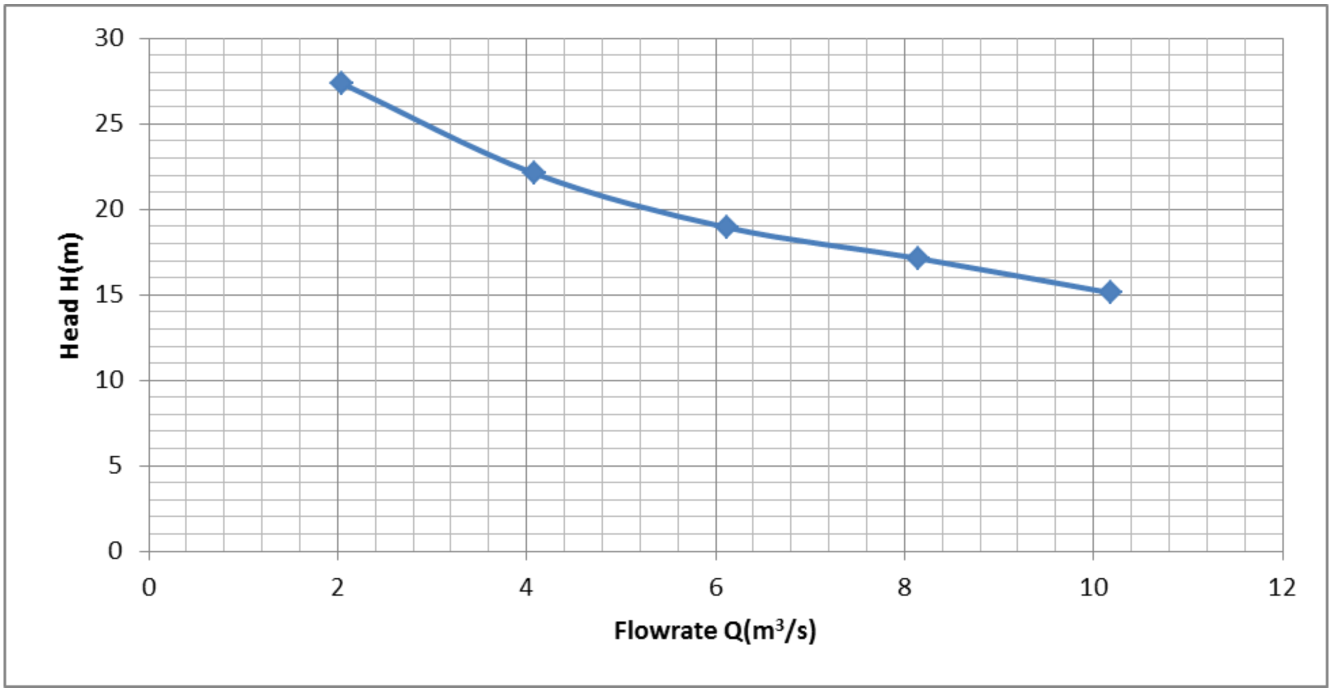

Figure 1: ***Head, H(m) against flowrate, Q(m^3/s)***

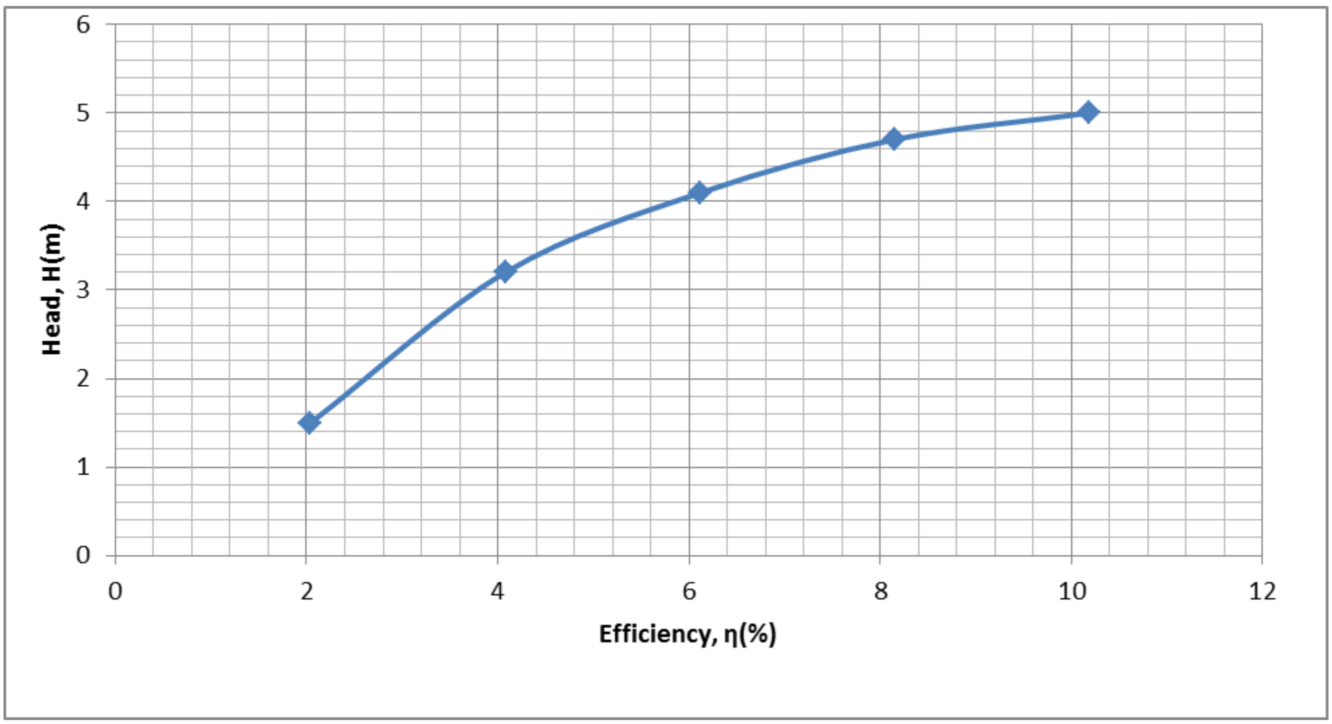

Fig. 2: ***Head, H(m) against efficiency, η(%)***

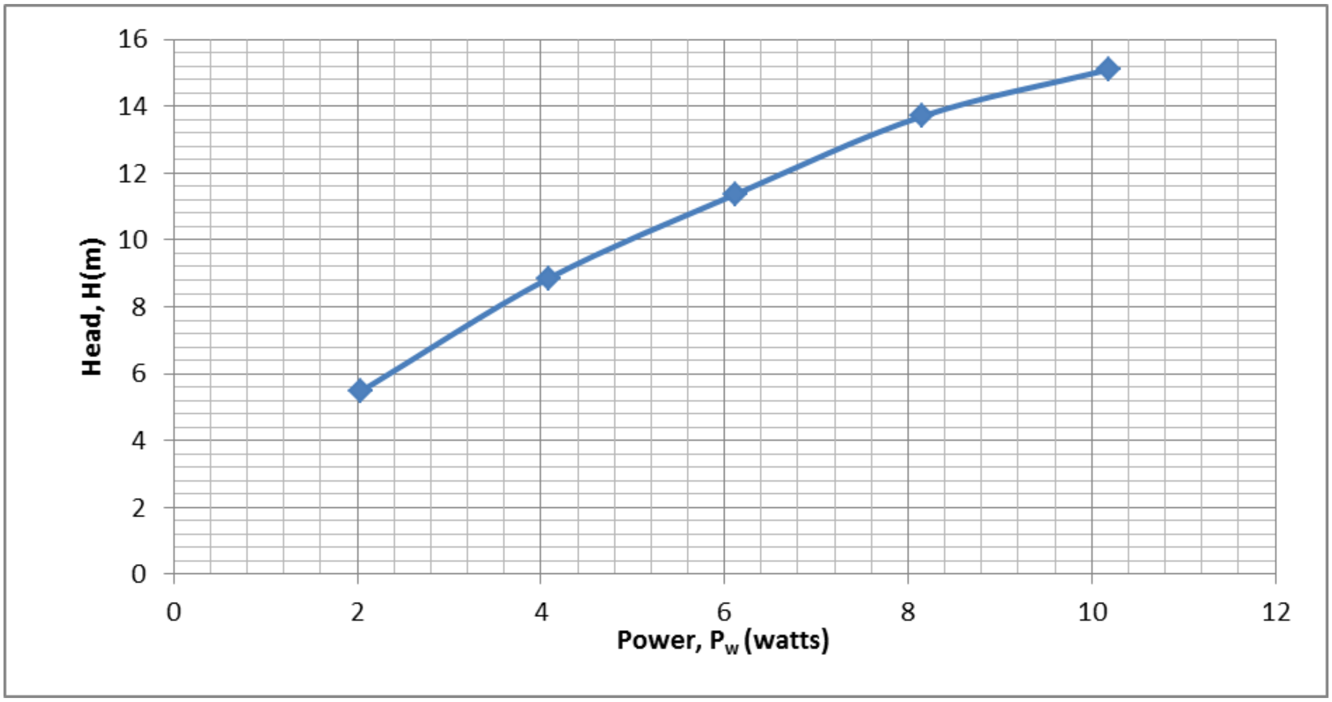

Figure 3: ***Head, H(m) against Power, P_w (watts)***

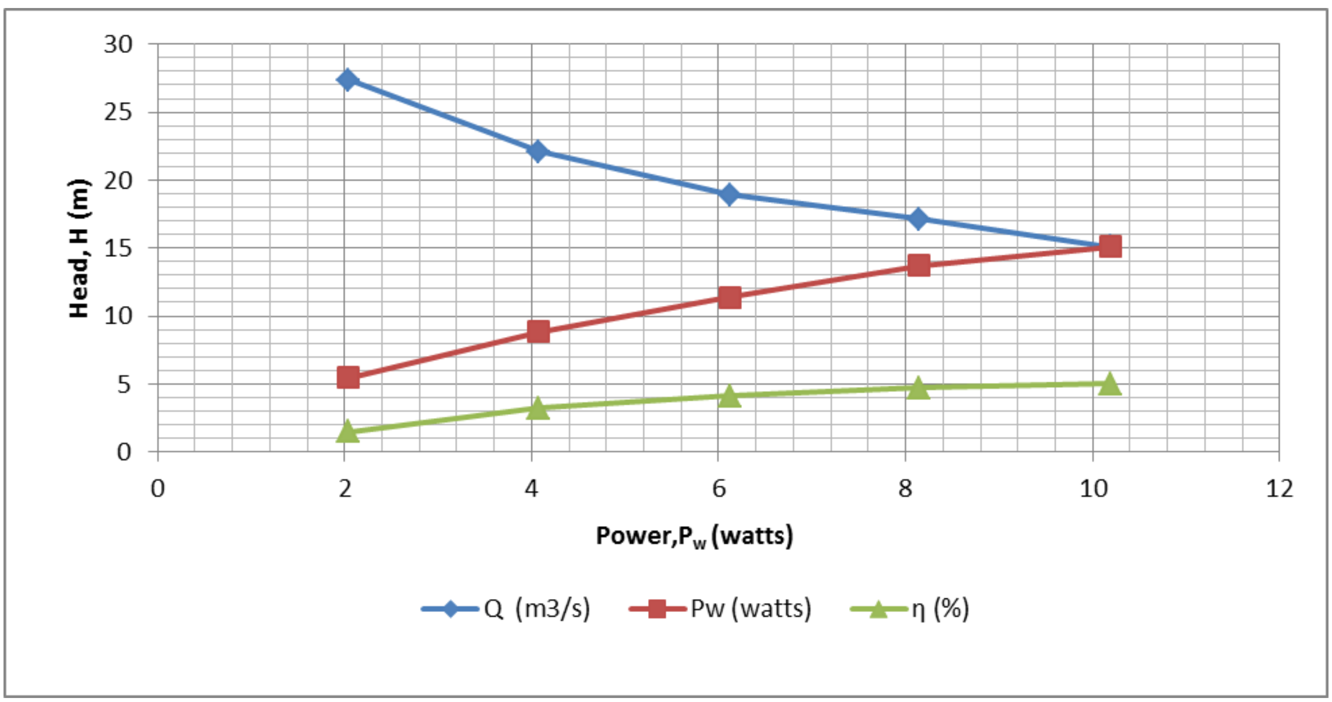

Figure 4: ***Head, H (m) against flowrate Q (m^3/s), Power,P_w (watts) and efficiency, η (%).***

4. CONCLUSION

The design, construction and characterization of a centrifugal pump has been successfully achieved. A framework of general applicability has been provided by practically discovering the relationship between the developed head H, flow rate Q, rotational speed and power P, with the aid of the centrifugal pump demonstration unit.

A major achievement of the work is that all things, materials, theories required for the eventual manufacture of the centrifugal pump demonstration unit had been carefully outlined and specified. The unit can be used to introduce students to the subject of rotodynamic machines.

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