

# Design, Production and Testing of a Single Stage Centrifugal Pump

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## Abstract

The design, construction and testing of a single stage centrifugal pump is presented in this project work, electric motor drives the centrifugal pump, which draws fluid (water) from a water storage wall and delivers same through a flow control valve to a tank. The experimental results obtained shows that the tested pump can develop a head, (H) of 30m, volumetric discharge, (Q) of 9m<sup>3</sup>/hr and the speed of 2900 rpm for an input power of 1.5HP (1.1k). The operation of the pump was observed to be very smooth with low vibration and noise level on the pump and motor respectively, this guarantee the reliability of the pump in service.

**Keywords:** Pump, Centrifugal Pump, Impeller Design, Shaft Design.

## LIST OF SYMBOLS

Q	Volumetric capacity m <sup>3</sup> /s
H	Head, Height m
N	Power J/S, W, KW
$\eta$	Efficiency of machines
M	Mass discharge (capacity) kg/s
$\delta$	Specific density (mass of fluid per unit of volume)kg/s
P	fluid pressure Pa
Z	Number of vane
$U_1$	Impeller Entry and Discharge Parameters Peripheral Velocity
$d_{hb}$	Impeller Hub Diameter
$D_1$	The normalized inlet diameter
L	Length of the coupling
$b_2$	Impeller Discharge Vane Width
$C_o$	Fluid Velocity at Impeller Eye

## INTRODUCTION

Pump is a mechanical device that applies energy to move liquids from one place to another at increased pressure, flow rate and to an elevated height. The development of pumps has enabled man to move away from his early settlement near rivers, lakes, and springs to develop vast areas of land that was previously uninhabitable. The usefulness of pumps has progressed to a point where it has become a necessity to modern high standard of living. In our everyday life pumps play very important parts. Our domestic water supply systems from water boards and private boreholes make use of pumps to distribute water to different locations. Irrigation, fire fighting services employ pumps of different sizes [1].

A centrifugal pump is a rotodynamic pump that uses a rotating impeller to increase the pressure of a fluid; the fluid enters the pump near the rotating axis streaming into the rotating impeller. The impeller consists of a rotating disc with several vane attached, the vanes normally slope backwards away from the direction of rotation when the fluids enters the impellers at a certain velocity due to the impeller vanes from the impeller centre eye out-ward, it reaches its maximum velocity at the impeller's outer diameter and leave the impeller into a diffuser or volute chamber [2]

The impeller is the essential parts of a centrifugal pump; the performance of the pump depends on the impeller diameters and design. The pump TDH is basically defined by the impeller's inner and outer diameter and pump's capacity is defined by the width of the impeller vanes in general there are three possible types of impellers, open, enclosed and semi open impellers, each suitable for specific application. Standard impellers are made of cast iron or carbon steel, while impeller for aggressive fluid and slurries require high end materials to ensure a long pump life. The open impellers are the simplest type of impellers, they consist of blades attached to the hub, and this type of impeller is lighter than any of the other type at the same diameter. Weight reduction leads to less force applied to the shaft and allows smaller shaft diameters, these results in low costs compared to equivalent shrouded impeller. Typically, open impellers operate at higher

efficiency because there is no friction between the shrouds and the pump casing. On the other hand side, open impellers have to be carefully positioned in the casing the gap between the impellers and the surrounding casing should be as small as possible to maximize efficiency, the impeller wears the clearance between the impeller and the front and back walls open up this leads to a dramatic drop in efficiency. Due to the minimized clearance between blades and casing, high velocity fluid is close proximity to the stationary casing establish vortices that increase wear dramatically [3]

Semi-open impellers can be seen as a compromise between open and enclosed impellers. A semi-open impeller is constructed with only one shroud, usually located at the back of the impeller; it usually operates at a higher efficiency than an equivalent enclosed one due to reduced disc friction as there is only one shroud. The advantage of semi-open impellers compared to open ones is that the impellers axial position can be adjusted to compensate for wear. The problem is that the entire backside of the impellers shroud is under full impeller discharge pressure as the front side is under suction pressure increasing along the impeller radius due to centrifugal force. The differential between these pressures causes an axial thrust imbalance manufacturer try to reduce this effect by applying vane to the backside of the impeller, but the efficiency of the so called "pump vanes" decreases if the impeller is moved forward to compensate for wears. A better option to compensate the loss of efficiency is an adjustable wear plate, so that it clearance adjustments can be made. [4] Enclosed impellers consist of blades covered by a front and back shroud the fluid steams through the impeller without interacting with the stationary pump casing. In a well design enclosed impeller, the relative velocity between the fluid and the impeller walls at any given radius is rather small, the disc friction of the shrouds rotating in close proximity to the pump casing causes a lower efficiency as comparable to semi-open or open impellers.

The performance of diffusers has been an important field of research for many years. Reneau et al [5] found that the performance of 2D diffusers is greatly affected by the inlet conditions. High recovery occurs at high area ratios (up to 5) and the minimum head loss occurs at  $20 < \theta < 70$  deg. Much attention is still devoted to research on radial diffusers Lugovaya et al [6]. Goto and Zangeneh [7] presented a new approach to optimizing a pump diffuser based on a three-dimensional inverse design method and computational fluid dynamics (CFD). Other researchers designed a twisted return guide vane for a submersible multistage pump, the results of their work demonstrating that return guide vanes with a twisted inlet can reduce the flow loss in radial diffusers [8]. The results show that the pressure changes little at the helix section along the direction of the flow then increases more and more at the diffuser section and hits its maximum at the end of the diffuser section Shi et al [9]. optimized the guide vane by using an orthogonal test method and found that the wrap angle and inlet angle of the radial diffuser have greater

impact on the pump head and efficiency. There suits indicate that the three-dimensional guide vane has smaller hydraulic loss and a higher pressure conversion capacity. And at the same time, they studied the performance of the deep-well centrifugal pump with the new designed guide vane [10]

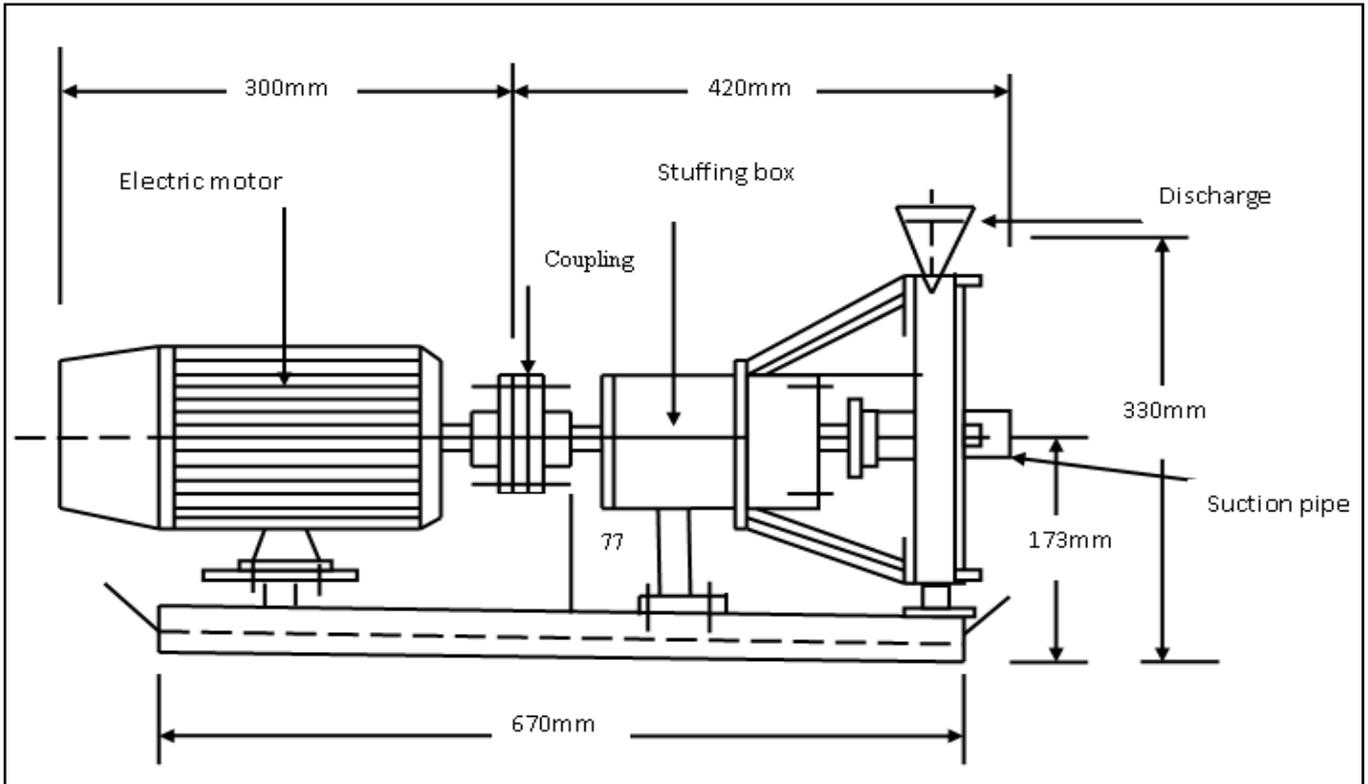
A shaft is a rotating machine element which is used to transmit power from one place to another [11]. The shaft is the connection between impeller and driver unit which is in most cases is an electric motor but can also be a gas turbine. It is mainly charged by a radial force caused by unbalance pressure forces in the spiral casing and an axial force due to the pressure difference between front and backside of the impeller. Most common pump shaft are made of carbon steel. There are several cranks to support the bearings and seals, a high surface quality and small clearances are required. Especially in the area of the bearing's, clearance and surface quality is important to ensure right positioning of the shaft in the casing and therefore close positioning clearances of the impeller. At the area of the seals, particularly the surface quality is also very important to ensure an adequate seal life span. In the shaft design it is also important to avoid small radiuses at cranks to minimize stress in these areas which are susceptible for fatigue

Automobiles are equipped with pumps for transfer of fuel, oil and in the refinery, pumps are needed to transfer product from one tank to another. Crude oil pumped flow from stations to tank farms that are stationed kilometers away, and refineries that are to be found in different cities through transmission lines respectively. Loading and unloading of tankers that carry water, fuel and other products to service points, use different types of pumps, direct handling of boiler feed water, water treatment chemicals, condensate, cooling water etc make use of pumps. One obvious thing to note here is that the development of pumps has enhanced the level of man civilization, economic, conducive environment and has led to the technological development of modern industries. It is true that pumps manufacturers in the technological advanced countries have developed standard design and constructional details that have resulted in the present level of perfection in pump design and construction by different companies in other countries. However, there is the need to develop, design and construct pumps locally, if we ever want to develop as a nation [12]

The significant considerations in the selection of materials for construction of the centrifugal pump included, local availability, low cost, easy handling during fabrication, lightness of weight for easy handling during use, weather ability and long service life (ability to withstand environmental and operating conditions) and non-toxic effects.[13]

Furthermore, when a pump is designed and constructed locally, the knowledge will be documented for further improvement. This knowledge will go a long way to enhance the much needed indigenous technological base.

**DESIGN THEORY AND CALCULATIONS**



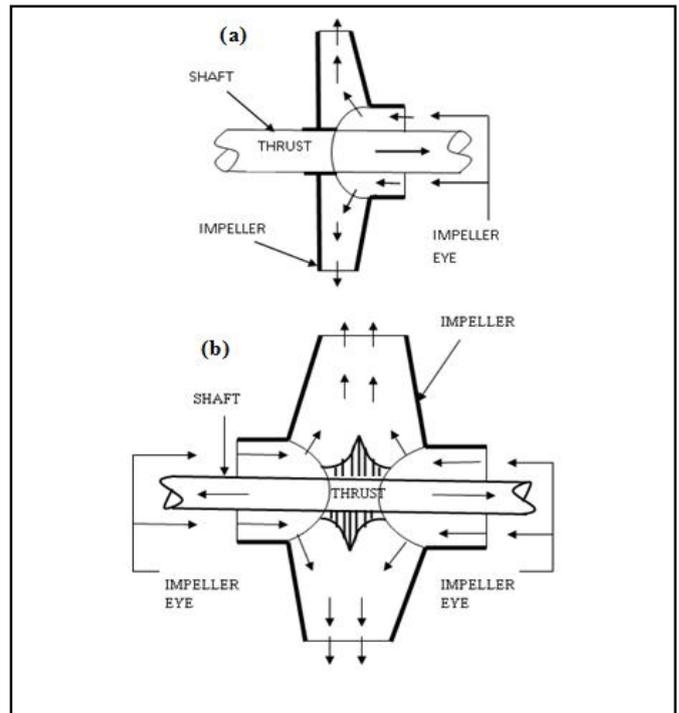
**Figure 1:** Pictorial view of the Centrifugal Pump

**BASICS ELEMENTS OF CENTRIFUGAL PUMP**

The main component of centrifugal pump are (i) An impeller (ii) A shaft complete with parts to attach the impeller and safe guard against wear at stuffing boxes (iii) Bearings (iv) coupling (v) casing (vi) stuffing boxes (vii) suction and discharge nozzles (viii) others such as pump basement (skid and bolts).

**Impeller Design**

Impeller design shall be for a single stage centrifugal pump capable of delivering 25m<sup>3</sup>/hr of water at ahead of about 27m at a low temperature of about 30°C and pump shaft speed of about 2900 r.p.m. For ease of manufacture and simplicity, the pump type shall be over hung-impeller pump that is mounting the impeller on an extension of the shaft rotating in two widely spaced ball bearings.



**Figure 2(a) and 2(b):** The Impeller and Shaft Design

### Determination of the Specific Speed ( $H_s$ ) of the Impeller

The specific speed can be determined as follows [13]

$$\eta_s = 3.65n \frac{\sqrt{Q}}{H^{3/4}} \quad (1)$$

Where n is the pump shaft speed

$$Q = 25 \text{ m}^3/\text{hr} = \frac{25}{60 \times 60} = 0.0069 \text{ m}^3/\text{s}$$

$$H = 27 \text{ m}$$

$$n = 2900 \text{ r.p.m}$$

$$\text{Specific speed } (\eta_s) = 3.65 \times 2900 \times \frac{\sqrt{0.0069}}{27^{3/4}}$$

$$(\eta_s) = 74.094$$

This specific speed is within the standard limits of radial vane impeller of low specific speed.

### Volumetric Efficiency is given By ( $\eta_{vol}$ ) Of The Pump

Volumetric Efficiency is given by

$$\eta_{vol} = \frac{1}{1 + a\eta_s^{0.66}} \quad (2)$$

Where a= the coefficient that depends on the inlet to outlet diameter ratio equal to about 0.68.

$$\eta_{vol} = \frac{1}{1 + 0.68 \times 74.095^{0.66}}$$

$$\eta_{vol} = 0.96$$

### Determination of Impeller Normalized Diameter

The normalized inlet diameter  $D_1$  is given by

$$D_{1 \text{ adj}} = 4.25 \frac{\sqrt{Q}}{n} \quad (3)$$

$$D_{1 \text{ adj}} = 56.7 \text{ mm}$$

### Hydraulic Efficiency, ( $\eta_h$ )

The Hydraulic efficiency, ( $\eta_h$ ) can be determined from the relation

$$\eta_h = \frac{0.42}{(\ln D_{1 \text{ adj}} 0.172)^2} \quad (4)$$

$$= \frac{0.42}{9.329}$$

$$\eta_h = 0.95$$

### Pump Shaft Power 'N'

$$\text{The Pump shaft power (N)} = \frac{\rho Q g H (KW)}{1000 \eta} \quad (5)$$

$$\text{Where } \rho = 1000 \text{ kg/m}^2$$

$$Q = 0.0069 \text{ m}^3/\text{s}$$

$$H = 27 \text{ m}$$

Substituting equation (5)

$$N = \frac{1000 \times 0.0069 \times 9.81 \times 27}{100 \times 0.82}$$

$$N = 2.2 \text{ Kw}$$

### The Torque $M_t$ on the Shaft.

$$\text{Torque } M_t = \frac{9600N}{n} \quad (6)$$

Where n = rpm

N = power

$$M_t = \frac{9600 \times 2.2}{2900}$$

$$M_t = 7.28 \text{ Nm}$$

### Pump Shaft Diameter ( $d_s$ )

The pump shaft is mainly subjected to torsion; therefore, the pump shaft diameter ( $d_s$ ) can be determined from the relation.

$$d_s^3 = \frac{16\sqrt{(K_t M_t)}}{\pi \tau} \quad (7)$$

Where  $K_t$  = combined shock and fatigue factor applied to torsional moment (1 to 1.5)

$M_t$  = torsional moment Nm

Shaft material, steel purchased under definite specific elastic limit 165MN/m<sup>2</sup>, safety factor 4

$$\text{Working stress} = \frac{165}{4} = 41.25 \text{ MN/m}^2$$

Design under maximum shear stress

$$\tau_{max} = \frac{41.25}{2} = 20.6 \text{ MN/m}^2$$

Due to key way to shaft the  $\tau_{max}$  is reduced by 25%

$$\tau_{max} = 15 \text{ MN/m}^2$$

If  $k_t = 1.3$  and  $M_t = 7.28 \text{ Nm}$

$$d_s^3 = \frac{16}{\pi \tau_{max}} \sqrt{K_t M_t^2}$$

$$= \frac{16}{3.142 \times 15 \times 10^6} \sqrt{(1.3 \times 7.28)^2}$$

$$d_s = 15 \text{ mm}$$

**The pump shaft coupling**

The diameter of the coupling can be determined by

$$D = 2d + 13mm \quad (8)$$

Where d=diameter of the shaft

$$D = 2 \times 15 + 13$$

$$D = 43mm$$

Length of the coupling

$$L = 3.5 \times d \quad (8.1)$$

$$= 3.5 \times 15$$

$$L = 52.5mm$$

**The pump shaft coupling key**

Length of the coupling key (t)

$$t = \frac{3.5d}{2} \quad (9)$$

$$= \frac{3.5 \times 15}{2}$$

$$t = 26.25mm$$

from proportions of standard parallel tapered and gib head keys that is for a shaft of 15mm diameter the width and thickness of the key is w = 6mm and t = 6mm.

**Impeller Hub Diameter (d<sub>hb</sub>)**

From the design consideration d<sub>hb</sub> is assumed to be (1.2 to 1.4) d<sub>s</sub>

$$d_{hb} = 1.4 d_s \quad (10)$$

$$= 1.4 \times 15$$

$$d_{hb} = 21mm$$

**The Impeller Inlet Diameter**

Is given by

$$D_o = \sqrt{D_{iadj}^2 + d_{hb}^2} \quad (11)$$

$$D_o = \sqrt{57^2 + 21^2}$$

$$D_o = 60.74mm$$

In open impeller D<sub>o</sub> can be assumed to equal to D<sub>1</sub>

Therefore, D<sub>1</sub>=60.7mm

**Impeller Hub Length (L<sub>hb</sub>)**

$$L_{hb} = 1 \times d_{hb} \quad (11.1)$$

$$L_{hb} = 21mm$$

**Impeller Entry and Discharge Parameters Peripheral Velocity (U<sub>1</sub>)**

$$U_1 = \frac{\pi D_1 n}{60} \quad (11.2)$$

$$D_1 = 0.0607m$$

$$U_1 = \frac{3.142 \times 0.0607 \times 2900}{60}$$

$$U_1 = 9.21m/s$$

**Fluid Velocity (C<sub>o</sub>) at Impeller Eye**

With D<sub>1</sub> = D<sub>o</sub>

$$C_o = \frac{4Q}{\eta_{VOL} \pi (D_o^2 - d_{hb}^2)} \quad (12)$$

$$C_o = \frac{4 \times 0.0069}{0.96 \times 3.142 \times (0.0607^2 - 0.021^2)}$$

$$C_o = 2.5m/s$$

**Impeller Inlet Vane Width (B<sub>1</sub>)**

$$b_1 = \frac{Q}{D_1 C_1 \mu_1 \pi} \quad (12.1)$$

Where, μ<sub>1</sub> = blockage factor is taken as 0.9 for inlet area of vane

$$b_1 = \frac{0.0069}{0.0674 \times 2.5 \times 0.9 \times 3.142}$$

$$b_1 = 16mm$$

Using

$$b_1 = \frac{D_1}{4},$$

$$D_1 = 60.74$$

$$b_1 = \frac{60.74}{4}$$

$$b_1 = 15mm$$

For this design b<sub>1</sub> = 15mm will be used

**Impeller Discharge Parameters**

Where β<sub>2</sub> = 20°, using the formulae for U<sub>2</sub> as stated by Cherkassy

$$U_2 = 1/2 C_{2r} \cot \beta_2 + \frac{\sqrt{C_{2r} \cot \beta_2^2} + \frac{gH}{\eta_h}}{2} \quad (12.2)$$

Assuming that  $C_o = C_{2r} = 2.5\text{m/s}$

$$U_2 = 1/2 \times 2.5 \times 2.74 + \frac{\sqrt{2.5 \times 2.74^2}}{2} + \frac{264.87}{0.95}$$

$$U_2 = 20.46\text{m/s}$$

### External Diameter of Impeller ( $D_2$ )

$$D_2 = \frac{60U_2}{\pi n} \quad (12.3)$$

$$D_2 = \frac{60 \times 20.46}{3.142 \times 2900}$$

$$D_2 = 134.8\text{mm}$$

Outlet to inlet diameter ratio

$$\frac{D_2}{D_1} = \frac{134.8}{60.74} = 2.2$$

This ratio is within the specific limit for two specific speed pumps.

### Impeller Discharge Vane Width ( $b_2$ )

Assuming  $C_{1r} = C_{2r}$

$$b_2 = b_1 \frac{D_1}{D_2} \quad (12.4)$$

$$b_2 = 15 \times \frac{60.74}{134.8}$$

$$b_2 = 7\text{mm}$$

However for ease manufacture  $b_2$  is equal to  $b_1$  though this result is 2.3% loss in pump efficiency.

### Number of Impeller Vanes

The optimum number vanes is

$$Z = 6.5 \frac{m + 1}{m - 1} \sin \frac{\beta_{1v} + \beta_{2v}}{2} \quad (12.5)$$

Where  $m = \frac{D_2}{D_1}$ , and  $\beta_2 + y$ ,  $y > 0$

If  $y=30$ ,  $\beta_{2v} = 20 + 3 = 23^\circ$  and

$$m = \frac{134.8}{60.74} = 2.2$$

Substitute into equation (12.5) gives

$$Z = 6.5 \frac{2.2 + 1}{2.2 - 1} \sin \frac{16^\circ + 13^\circ}{2} \quad (12.6)$$

$$Z = 6.5 \times 2.666 \times \sin 19.5^\circ$$

$$Z = 6\text{vanes.}$$

## RESULTS AND DISCUSSION

### Testing

The pump was tested with an electric motor rated 1.1kW (1.5HP) at 1400rpm. The fluid used for the test was water at a temperature of  $30^\circ$ . The test was to ascertain the discharge  $Q$  by measuring the quantity discharge over a period of time. Table 2 shows the test readings obtained

**Table 1: Fluid Discharge and Time Taken**

Quantity Discharge	Times
40 liters vessel filled	35
36 liters vessel filled	32
Total = 76 liters vessel filled	67
Total average = 76/2 = 38liters	67/2 = 33.5

$$\text{Discharge (Q)} = \frac{\text{Average Quantity discharge}}{\text{Average time taken}} \quad (13)$$

Discharge ( $Q\text{m}^3/\text{s}$ )  $38 \times 0.001 = 0.038$

$$Q = \frac{0.038}{33.5}$$

$$Q = 0.00113\text{m}^3/\text{s} \text{ or } 4.08\text{m}^3/\text{h}$$

The test head was done by closing the gate valve on the discharge line. Pressure reading obtained on a pressure gauge =  $0.7\text{kg/m}^2$

Where  $p = \rho gH$

$$H = \frac{P}{\rho g} \quad (14)$$

$$H = \frac{0.7 \times 1000000}{1000 \times 9.8}$$

$$H = 7.14$$

At 1400rpm of electric motor the pump performance is  $Q = 4.08\text{m}^3/\text{hr}$

$$H = 7.14\text{m}$$

Using the similarity law to obtain the performance for the design condition as follows

$$\frac{Q_2}{Q} = \frac{n_2}{n_1} \quad (15)$$

$$\frac{Q_2}{4.08} = \frac{2900}{1400}$$

$$Q_2 = 2.07 \times 4.08$$

$$Q_2 = 8.5 \text{ m}^3/\text{hr}$$

Discharge  $Q_2$  of pump at 2900rpm = 9m<sup>3</sup>/hr

Head (H)

$$\frac{H_2}{H} = \frac{n_2^2}{n_1^2} \quad (16)$$

$$H_2 = 4.290 \times 7.14$$

$$H_2 = 30.6 \text{ m}$$

Head (H) of pump at 2900rpm = 30.6m

Pump efficiency  $\eta_{\text{pump}}$

$$\eta_{\text{pump}} = \frac{\rho g Q H}{N}$$

$$= \frac{0.7 \times 9.8 \times 9 \times 30}{2.2 \times 1000}$$

$$\eta_{\text{pump}} = 0.841$$

$$= 84\%$$

Discharge (Q) = 9m<sup>3</sup>/hr

Head (H) = 30m

Power (N) = 2.2kw

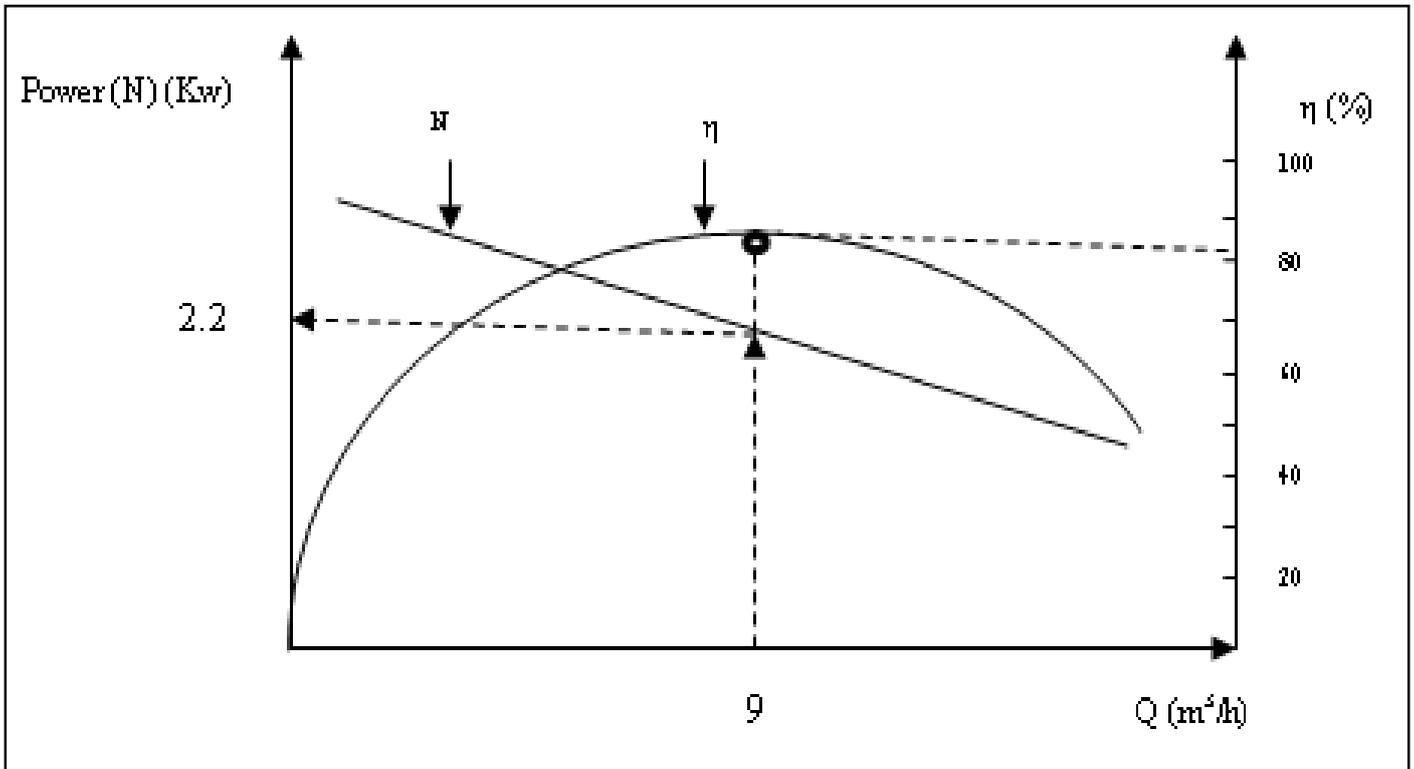
Speed (n) = 2900rpm

Pump efficiency ( $\eta_{\text{pump}}$ ) = 84%

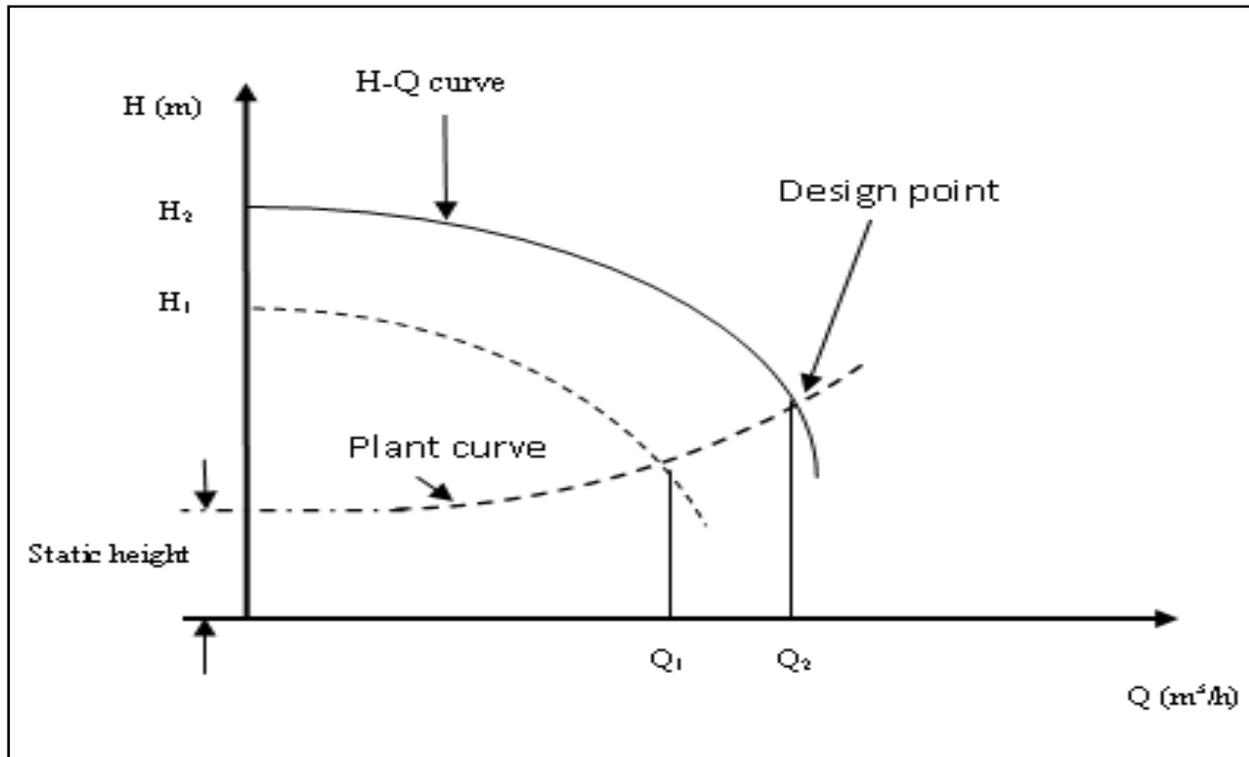
#### Performance characteristics

The output of a pump running at a given speed is the discharge rate delivered by it and the head developed. A plot of head against discharge rate at constant speed forms the fundamental performance characteristics of a pump [2]. 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 power input (N) and the efficiency of the pump ( $\eta$ ) against discharge rate (Q). Such that a complete set of performance characteristics of the existing pump in the mechanical designed, constructed and testing of a centrifugal pump

Therefore the mechanical characteristics performance of pump as follows



**Figure 3a:** graph of Head (H) vs. Discharge (Q) of the pump



**Figure 3b:** graph of Power (N), Efficiency ( $\eta$ ) vs. Discharge (Q)

The results obtained, as presented in the plots of the pump performance characteristics are in general agreement with the plots of centrifugal pump characteristic curves obtained in literature, [4].

## CONCLUSION

An attempt has been made to design, construct and test a single stage centrifugal pump. The test has shown that the construction of single stage centrifugal pump is able to produce a head of about 30m instead of the calculated 27m this represent an increase of about 3m above the intended head H, Furthermore volumetric discharge of 9m<sup>3</sup>/hr was obtain. The performance of a pump is however better evaluated if tested at the design speed and power with all necessary parameters ready by direct measurement. The pump operated was observed to be very smooth with low vibration and noise level on the pump and motor respectively, this will guarantee the reliability of the pump in service.

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