Nigerian Journal of Technology (NIJOTECH)
Vol. 31, No. 3, November, 2012, pp. 241–247.
 ISSN 1115-8443

DESIGN, CONSTRUCTION AND MEASURED PERFORMANCE OF A SINGLE-STAGE CENTRIFUGAL PUMP DEMONSTRATION UNIT

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Abstract

The design philosophy, construction and measured performances of a single stage, single entry centrifugal pump demonstration unit are presented. In the construction, close-coupled induction motor drives the centrifugal pump, which draws fluid (water) from a water storage tank and delivers same through a flow control valve, and an orifice meter, back to the tank. Changing the setting of the flow control valve alters the system resistance, and so changes the operating point on the pumps head-flow characteristics. Piezometers are attached to the pipe work, close to the inlet and outlet of the pump, so that the total head-rise (H), is recorded by a pressure gauge. A further measurement is made of the differential head, h across the flow meter, from which the flow rate, Q is inferred from the meter-calibration. The demonstration unit was used to run the tests necessary to fully characterize a centrifugal pump. Tabular and graphical representations of the relationships between the various parameters, viz: head (H), efficiency (η), rotational speed (ω), and input power (P_i), with the flow rate (Q), were generated. The experimental results obtained with the demonstration unit shows that the tested pump can develop a maximum head, (H_{max}) of 18.35m, maximum flow rate, (Q_{max}) of 21.40 litres per minute and maximum speed of 2800 rpm for an input power (h.p) of 0.5 (370watts).

Keywords: centrifugal pump, characteristics, demonstration unit, measured performance

1. Introduction

Studies on pumps are currently being pursued by manufacturers and individuals [1]. The electric power used to drive them accounts for a large share of energy consumed by nations. This places strong emphasis on theoretical research aimed at perfecting the operating cycle and raising the efficiency of this kind of machinery. Much work has been done refining the basic centrifugal pump theory by two or more dimensional techniques. The mathematical complexities are enormous and thus experimentation is inevitable. Also, testing the detailed flow structures is difficult. Important features of the flow, such as the secondary motions are still neglected [2]. Therefore, only moderate success has been made in elucidating certain details of the flow in a centrifugal pump. The comprehensive theory to permit the complete hydrodynamic design is yet to evolve [3]. However, users of centrifugal pumps will not normally wish to go deeply into the design, but they can profit from familiarity with pump theory by being able to establish the characteristics of an optimally designed centrifugal pump [4]. Additionally, the cost of importation of standard instructional equipment for students of fluid mechanics in higher institutions is becoming quite prohibitive. The availability of a portable classroom centrifugal pump test rig apart from enhancing students grass-root practical appreciation of the concept of rotodynamic machines, would immediately bridge the gap between the theory and practical applications of this kind of machinery, especially in some spheres where standard equipment for this purpose is either faulty or extinct. Consequently, the construction of such equipment from locally sourced materials using theoretical background of rotordynamic machines is demonstrated. The operating principles and the performance indices may be measured for instructional purposes. [5,6].

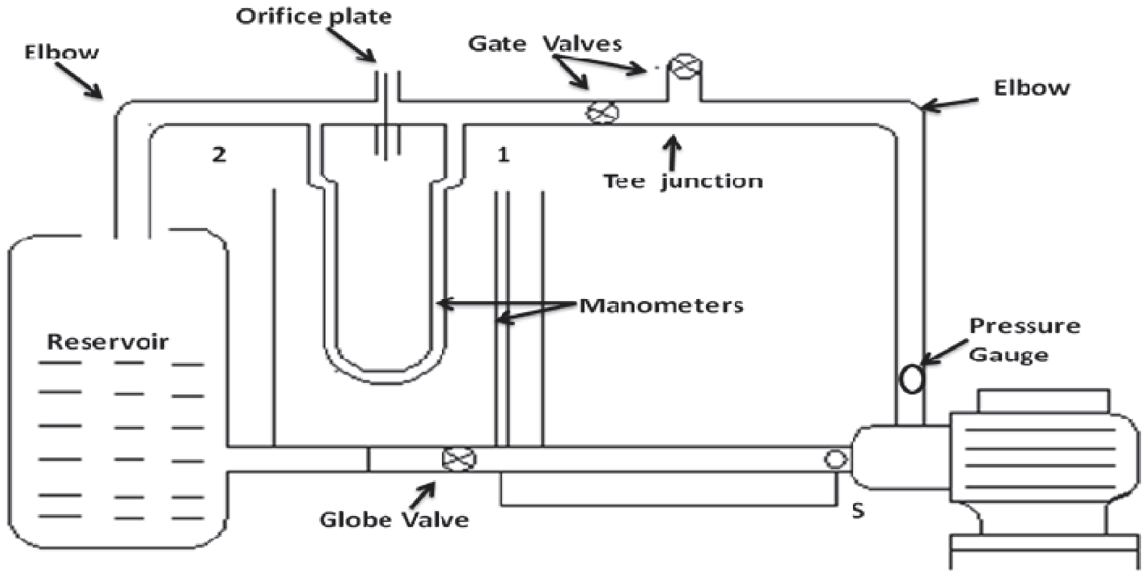

Figure 1: Simplified diagram of the demonstration unit.

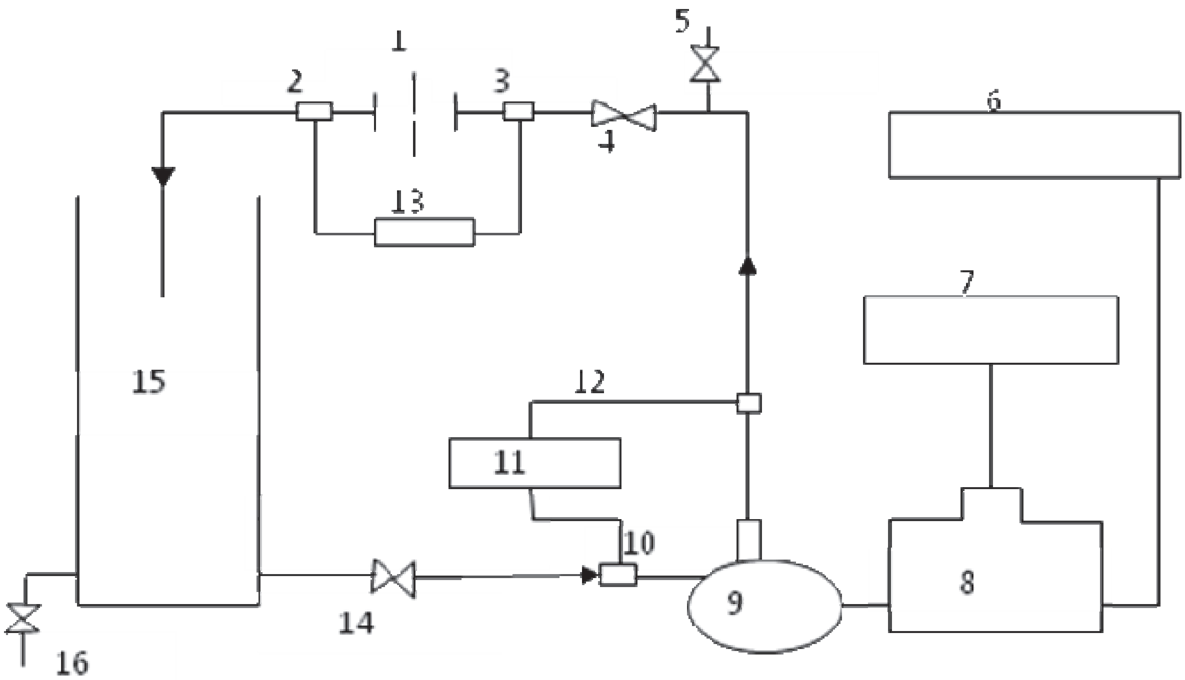

Figure 2: Line diagram of the demonstration unit. (1) Orifice Plate, (2) Low pressure area, (3)High Pressure area, (4) and (5) Gate Valves, (6) Motor Speed (rpm), (7) Motor power (watts), (8) Motor, (9) Pump, (10) Low pressure area (Pump suction), (11) Differential Pressure across pump (KPa), (12)High pressure area (Pump delivery)(13) Differential Pressure across Orifice (KPa), (14) Globe Valve, (15) Reservoir, (16) Drain.

2. Description of the Centrifugal Pump Test Rig

The experimental test rig consists of a self-contained bench top unit. It comprises a single-stage, single entry centrifugal pump whose characteristics are to be investigated. A close-coupled alternating current induction motor drives the pump. A reservoir and pipe work for continuous water circulation is also provided. Manually operated valves, at the pump inlet and outlet allow control of the flow and also facilitate study of suction effects. An orifice plate measures flow rate with a U-tube mercury manometer. The pressure difference is read by means of a calibrated paper embedded in a vertical wooden support located midway into the bench area. On the stand also is a vertical tube manometer, which monitors the piezometric head at the pump suction. The pump discharge pressure is measured with a Bourdon-tube pressure gauge in the discharge line. Impeller rotational speed is determined with a tachometer while the motor input power is got with a Wattmeter. Ambient water temperature is assumed at the inlet. A schematic illustration of the features of the centrifugal pump demonstration unit is shown in figure 1, while the line diagram is shown in figure 2.

2.1. Design considerations

Important general considerations in the selection of materials for construction of the demonstration unit included, local availability, low cost, easy handling during fabrication, lightness of weight for easy handling during use, weatherability and long service life (i.e. ability to withstand environmental and operating conditions) and non-toxic effects. The base is made of plywood and holds the other members of the unit in position. It is supported in both vertical and horizontal directions to enhance rigidity. It is approximately 122cm long and 61cm wide with a vertical elevation of 7cm. The centrifugal pump was not designed as pump design is outside the scope of the work. However, an arbitrary centrifugal pump whose suction and exit pipe diameters are in tandem with the size of the pipe run on the rig was purchased from the market, with a view to characterizing it (the pump) with the test rig. The selected pump had an inscription of 40m total head on its nameplate. It has a fluid (water) reservoir (48.5cm diameter by 27 cm high). The wooden support base is 27.5cm by 18cm, height-65cm and has 2 flanges of 18cm by 9cm each. The overhangs are intended to support the pipe run while a rectangular groove in between them supports the orifice. The orifice plate has 28mm and 14.5mm outside and bore diameters respectively with a thickness of 3.1mm. The orifice plate is held in position in between two gaskets by conventional orifice flanges. 11.5cm external diameter mild steel was used as the flange material. Special orifice fittings (metal adapters with 28mm bore diameter) are incorporated in both the upstream and downstream stations of the plate. They contain the piezometer tapping (connectors) that are connected to the U-tube manometer. The orifice fittings were constructed to permit easy changing and inspection of orifice plates. The orifice recorders normally include devices for measuring both differential and flowing pressures. Manometers used in the rig includes: 1) mercury U-tube manometer of height 26.5cm. It contains 10ml of mercury. 2) Pressure tube or piezometer which records the pump inlet pressure. It consists of a vertical tube of 45cm long. It is open at the top and the bottom inserted into a metal adapter at the pump suction via a flexible tube. The two manometers have an external diameter of 1cm and are made of transparent glass. A standard Bourdon tube pressure gauge which can record a maximum pressure of about 6bar was selected to obtain the static pressure. A Wattmeter was employed to measure the electrical power of the pump, while the rotational speed of the electric motor, which corresponds to the pump speed, was obtained using a tachometer. Normally, the outboard end of the electric motor is uncovered to allow the tachometer spindle to be held against a recess in the motor shaft. The constructed centrifugal pump demonstration unit is shown in figure 3.

Figure 3: Centifugal Pump demonstration unit.

3. Design Analysis

3.1. Selection of measuring devices

3.1.1. Selection of manometers

The inscription on the pump name plate shows that it can develop a maximum head of 40m; which corresponds to a static pressure of:

$$P = \rho g h = 1000{*}9.81{*}4.0{*}10^{5}\mathrm{N/m^2} \cong 4.0\text{bar pressure} \quad (1)$$

A straight tube manometer and Bourdon tube gauge (ranging from 0–6bar), were selected to measure the head at the pump suction and discharge pressure, respectively.

3.1.2. Selection of U-tube manometer

Assuming one-dimensional flow of an incompressible frictionless fluid without work, heat transfer, or elevation change, theory [7,8], gives the volume flow rate through the orifice (Q_{th}) as:

$$Q_{th} = A_{1f}/\sqrt{\left(A_{1f}/A_{2f}\right)^2 - 1}\sqrt{2\left(P_1 - P_2\right)/\rho} \quad (2)$$

Where A_{1f}, A_{2f} are cross section flow areas where P_1 and P_2 are measured. P_1, P_2 are static pressures and ρ is the fluid mass density.

However, the real situation is inconsistent with the assumption of the theoretical model and hence accurate practical results demand the use of experimental correction factors. For instance, A_{1f} and A_{2f} are areas of the actual flow cross section, which are not, in general, the same as those corresponding to the pipe and orifice diameters that are most often measured in practice. Also, flow geometrical changes may introduce a change in A_{1f} and A_{2f} with flow rate and there exists some frictional losses, which affect the measured pressure drop and cause a permanent pressure loss. These factors necessitate the introduction of an experimental calibration to determine the actual flow rate (Q_{ac}) [9]. For this purpose, a discharge coefficient (C_d) is used and is defined as:

$$C_d = Q_{ac}/Q_{th} \quad (3)$$

Such that,

$$Q_{ac} = C_d A_1/\sqrt{\left(A_1/A_2\right)^2 - 1}\sqrt{2\left(P_1 - P_2\right)/\rho} \quad (4)$$

Where A_1 is pipe cross sectional area and A_2 is orifice cross sectional area. Putting $m = A_1/A_2 = [d_1/d_2]^2$, and $P = \rho gh$ and substituting into equation (4) gives:

$$Q_{ac} = C_d A_1/\sqrt{m^2 - 1}\sqrt{\frac{2gh\rho_{man}}{\rho - 1}} \quad (5)$$

Where d_1, internal diameter of the adapter (28mm), d_2; orifice diameter (14.5mm). The area of the conduit cross section, $A_1 = 615.75\text{mm}^2$, $m = 3.7289$. ρ_{man}; Density of manometric substance (mercury) = 13.6 ρ H_2O, ρ; density of working fluid (water), h; differential pressure across the orifice (mmHg), A_2; orifice area.

From information given on the manufacturer's nameplate, $Q_{max} = 40$litres/min, giving

$$Q_{max} = 40 = C_d A_1/\sqrt{m^2 - 1}\sqrt{\frac{2gh\rho_{man}}{\rho - 1}}$$

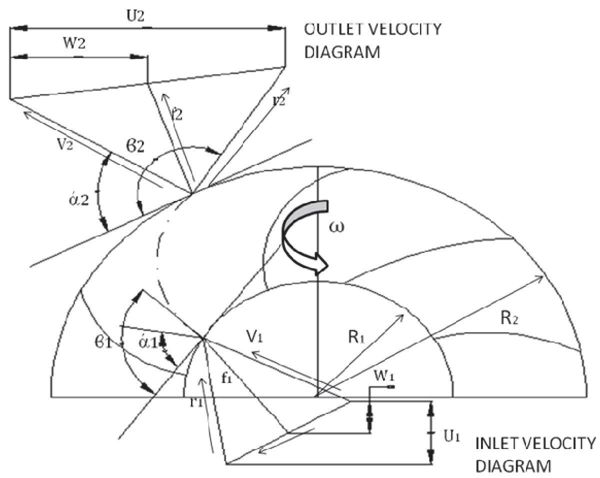

Figure 4: Velocity diagram for pump impeller.

Therefore, $h_{max} = 17$cm. The volume of mercury in the U-tube manometer was determined by considering the part of the tube containing mercury as a stretched out cylinder. The mean radius of the curved portion of the tube is $r_m = 2.0$ cm.

The mean length of the curved portion, $l_m = \pi r_m$, $l_m = 2\pi$ cm. Length of column, $l = 2 + h = 23.284$ cm. Volume of mercury required $= d^2 l/4$

$V_{mercury} = \pi(1^2/4)\times\ 23.284 = 18.29\text{cm}^3$ or 18.29ml. This value (18.29ml of mercury) was not used because preliminary tests on the pump yielded values of maximum head and discharge that were 50% short of the values on the nameplate. Accordingly, 10ml of mercury was procured. The total length of the U-tube (top to bottom) was fixed at 25cm to forestall any spillage of the mercury over the flexible tubing at maximum discharge.

3.1.3. Centrifugal Pump Velocity Diagram

The torque T applied to the impeller and the volume flow rate Q may be expressed according to the following dependence:

$$T = \dot{m}(w_2 R_2 - w_1 R_1) \quad (6)$$

$$Q = f_1 A_1 = f_2 A_2 \quad (7)$$

Where $\dot{m} = \rho Q$ is the mass flow rate (the rate of flow of angular momentum into the impeller), f_1, f_2 are the radial component of absolute water velocity at inlet and outlet respectively, w_1, w_2, refer to the whirl component of absolute water velocity at inlet and outlet (m/s). The rate of increase of angular momentum $\dot{M}$, in its passage through the impeller is the difference between the outflow and inflow of fluid out of and into the impeller respectively, namely [4]

$$\dot{M}_2 - \dot{M}_1 = \dot{m}(w_2 R_2 - w_1 R_1) \quad (8)$$

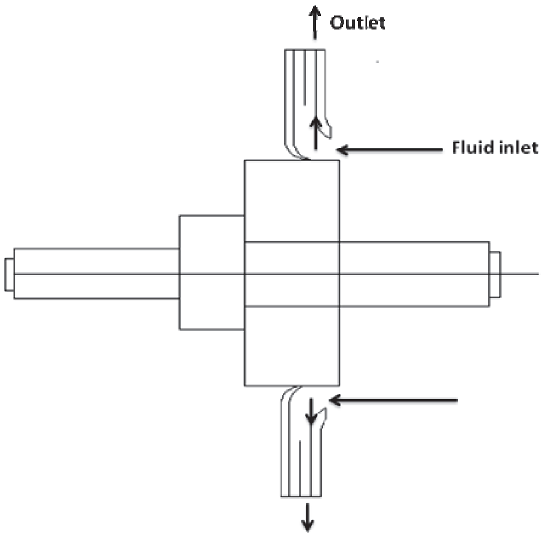

Figure 5: Radial flow impeller.

The input power required to produce this torque is given by

$$P_i = \dot{m}(U_2 w_2 - U_1 w_1) \quad (9)$$

The velocities of the vanes, U_2, at outlet and U_1, at inlet are given by $U_2 = \omega R_2$ and $U_1 = \omega R_1$ respectively, R refers to the impeller radius and ω is the angular velocity. If the whole of this mechanical input power were converted to hydraulic power, then the ideal increase of total head H_i would be given by:

$$P_i = mgH_i \quad (10)$$

$$H_i = \frac{P_i}{mg} \quad (11)$$

The power required to raise water against gravity through a height H_i at a mass flow rate of $\dot{m}$ gives the Euler's equation for the impeller as:

$$H_i = \frac{U_2 w_2 - U_1 w_1}{g} \quad (12)$$

3.1.4. Experimentation

Recall the diagram of figure 2. With the globe valve at the pump suction line closed, the reservoir is filled with water up to a level just below the tip of the discharge pipe. The valve is then opened to admit water into the pump. Power is then supplied to the pump with the gate valve at the discharge line fully open. When water had circulated through the whole unit, the pressure gauge reading, the mercury levels in both arms of the U tube manometer, the water level in the vertical manometer and the rotational speed of the electric motor are taken. A tachometer held at the outboard end of the electric motor determines the speed. The spindle is held against a small recess in the motor shaft. Also, the mains voltage and the electric current drawn by the motor are obtained with a voltmeter and an ammeter respectively. By gradually adjusting the control (gate) valve at the discharge line, the discharge through the system is reduced and the head developed by the pump as indicated by the pressure gauge increases. The valve adjustment continues until a desired gauge reading is achieved. This automatically introduces a change in the readings of the measuring devices. Consequently, the new values corresponding to the pressure increase are recorded.

These measurements continue for different pressure gauge readings until it gets to the point where the valve (gate) adjustment is no longer possible.

Table 1: Experimental results.

Speed	P_D (bar)	h_s (mm)	h_1 (mm)	h_2 (mm)	V (Volts)	I (Amps)
2800	0.0	130	119	158	240	1.60
2700	0.2	229	127	150	240	1.48
2750	0.4	254	131	146	240	1.16
2650	0.6	279	133	144	233	1.18
2650	0.8	290	134	143	235	1.24
2500	1.0	300	135	142	233	1.29
2500	1.2	330	137	141	234	1.30
2450	1.4	335	138	140	230	1.32
2350	1.6	340	138	139	233	1.37
2250	1.8	350	139	139	235	1.45

4. Results and Calculations

The results presented in Table 1 were obtained from the existing pump test with the demonstration unit.

Where h_s; water level in the manometer tube connected to the pump inlet, h_1; level of mercury in the arm of the U tube connected to the tapping upstream of the orifice, h_2; level of mercury in the arm of the U tube connected to the tapping downstream of the orifice. The theoretical discharge through the orifice is [9]:

$$Q_{th} = A_1/\sqrt{\{m^2 - 1\}}\sqrt{2gh[\rho_{man}/\rho - 1]} \quad (13)$$

Where $m = A_1/A_0 = [D_1/D_0]^2 = 3.7289$; D_1; internal diameter (28mm) of the adapters containing the tappings; $A_1 = 6.157 \times 10^{-4}$ (m^2). D_2; internal diameter (26.7mm) of pipe, $A_2 = 5.599 \times 10^{-4}$ (m^2). D_0; orifice diameter (14.5mm), $A_0 = 1.651 \times 10^{-4}$ (m^2). ρ_{man}; density of mercury (13.6 ρ H_2O) , $\rho = \rho H_2O$. Location of tapping both upstream and downstream of the orifice = 3.1 cm The actual discharge, $Q_{ac} = C_d * Q_{th}$.

For the case of the experiment, the value of the coefficient of discharge was estimated by a heterodox method thus: a cylindrical container having a volume of 715.91cm^3 0.7191(l) was used to collect water from the delivery pipe to the reservoir thrice. The time taken to fill the cylinder in each occasion

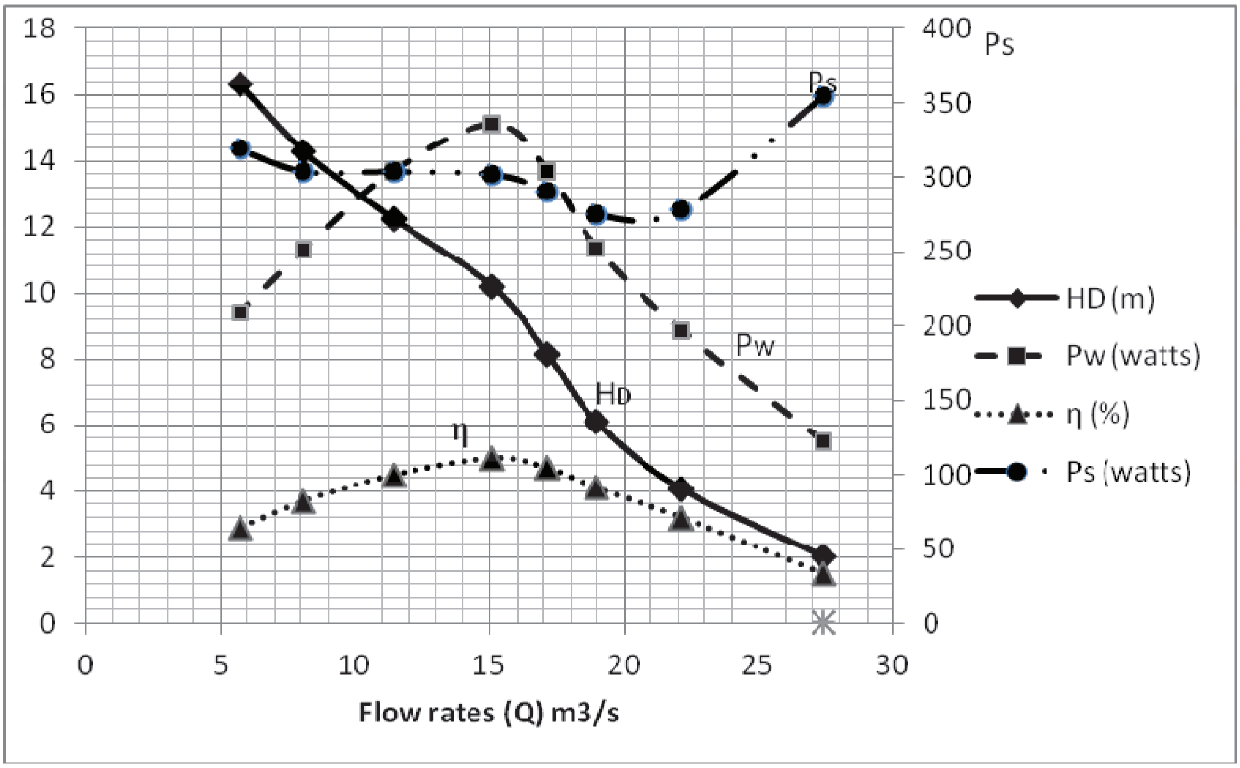

Figure 6: Pump performance characteristics.

was 2 seconds. The measured discharge is therefore: $Q_{ac} = 357.955 \times 10^{-6}\text{m}^3/\text{sec}$. The differential head across the orifice read from the manometer at this maximum discharge is $h = 158 - 119 = 39$mm.

Therefore, Q_{th} 532.22×10^{-6} m^3/sec making C_d to be 0.67 which was the value used to prepare table 2 showing the pump total head as well as its power and efficiency at various rates of flow. The electric power input to the pump P_s was computed using equation (14) as the product of the voltmeter and ammeter readings at each flow rate. The hydraulic power P_w, and the overall efficiency η, at each flow rate was computed from equations (15) and (16) respectively.

$$P_s = IV * \text{power factor (watts)} \tag{14}$$

$$P_w = \rho g Q H = 9810QH \tag{15}$$

$$\eta = P_w/P_s \tag{16}$$

4.1. Performance characteristics

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 against flow 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 (P_s), the hydraulic power (P_w) and the efficiency η against flow rate Q. such a complete set of performance characteristics of the existing pump in the designed and constructed centrifugal pump demonstration unit is shown in figure 6.

4.2. Discussion of Results

The results obtained, as presented in the tables and plots of the pump performance characteristics are in general agreement with the plots of centrifugal pump characteristic curves obtained in literature, eg [2]. The functional relationship between the head, H, and discharge, Q of a centrifugal pump:

$$H = K_1 - K_2 Q \tan 2 \tag{17}$$

suggests that the plot of H vs Q will be a straight line having a positive intercept and a negative slope. A linear relationship will result only if K_1 and K_2 are constants. That is when $K_1 = U_1^2/g$, $K_2 = U_2/A_2 g$. U_2 will be constant if the rotational speed is constant, which is not the case with this experiment as in many others. 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, that is

$$H = \eta H H_{th} \tag{18}$$

$$H = \eta H U_2^2/g - (\eta H U_2/A_2) Q \tan 2 \tag{19}$$

Table 2: Pump total head, power and efficiency at various rates of flow.

H_D (m)	$h \times 10^{-3}$ (m)	Q (L/min)	Speed (rpm)	$Q \times 10^{-5}$ (m^3/s)	P_w (watts)	P_s (watts)	η (%)
2.04	23	16.43	2750	27.38	5.48	355	1.5
4.08	15	13.27	2750	22.11	8.85	278	3.2
6.12	11	11.36	2650	18.94	11.37	275	4.1
8.15	9	10.28	2650	17.13	13.70	291	4.7
10.19	7	9.06	2500	15.11	15.10	302	5.0
12.23	4	6.85	2500	11.42	13.70	304	4.5
14.27	2	4.85	2450	8.08	11.31	304	3.7
16.31	1	3.43	2350	5.71	9.41	319	2.9

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. 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. For a possible modification of the unit, the values of these losses have to be included in the pipelines total resistance if errors in pump and system matching or flow calculations for a given pressure differential are to be avoided. These modifications would include the provision of tapping at suitable distances that allow flow stabilization, upstream and downstream of each fitting and the addition of more manometer tubes to those already in use. The effects of separation losses across the fittings and frictional losses along the pipe on the results obtained were not investigated. The variation of water density and viscosity during the experiment especially during slight temperature increase could equally have contributed to the results. Other errors worthy of note include that due to parallax while taking the manometer reading, random electric fluctuations and probable errors in the construction of measuring devices. They all contribute to the overall efficiency. The name plate on the pump used, has the following information: H_{max} = 40 m, Q_{max} == 40 litres, Speed = 2900rpm, h.p. = 0.5 (370W). If the name plate is to be redone in line with our research realities, it would read: H_{max} = 18.35m, Q_{max} = 21.40l, Max. Speed = 2800 rpm, h.p. = 0.5 (370W). The values of maximum head and maximum discharge obtained are just about 50% of those specified on the pump.

5. Conclusion

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

References

1. Engines and fluid machines catalogue. *Capture.* 4th Edition, Armfield Limited, Hampshire, England, 2001.
2. Douglas, J.F., Gasiorek, J.M. and Swaffield, J.A. *Fluid Mechanics.* 4th edition, Pearson Education, 2001.
3. Markland, E. *A First Course in Hydraulics.* Revised Edition, TQ Education and Training Ltd, Cardiff, 1996.
4. Jekat, W.K. *Centrifugal Pump Theory.* McGraw Hill Inc, 1978..
5. Holzbock, W.G. *Hydraulic Power and Equipment.* Industrial Press Inc, 1968.
6. Holman, J.P. *Experimental Methods for Engineers.* 3rdEdition, McGraw Hill Inc, 1978.
7. Doebelin, E.O. *Measurement Systems, Applications and Designs.* McGraw Hill Inc., 1975.
8. Bush Beach Engineering company. *Tab handled square edge orifice plates, and orifice plate carrier assemblies.* Unpublished handout.
9. Olsen, R.M. *Essentials of Engineering Fluid Mechanics.* Second Edition, International Textbook Company, 1968.