Original Research Article

Turbine Dimensionless Coefficients and the Net Head/Flow Rate Characteristic for a Simplified Pico Hydro Power System

ABSTRACT

The basic operational parameters of a simplified pico-hydropower system with provision for water recycling were investigated. Five simplified turbine of runner diameters 0.45, 0.40, 0.35, 0.30 and 0.25 m were designed, locally fabricated, and tested in conjunction with five PVC pipes of diameters 0.0762, 0.0635, 0.0508, 0.0445 and 0.0381 m as penstocks. Five simple nozzles of area ratios 1.0, 0.8, 0.6, 0.4 and 0.2 were fabricated for each penstock diameter. The turbines were successively mounted at the foot of an overhead reservoir such that the effective vertical height from the outlet of the reservoir to the plane of the turbine shaft was 6.95 m. A 1.11 kW electric pump was used to recycle the water downstream of the turbine back to the overhead reservoir. The mean maximum and minimum rotational speeds of the shaft of each turbine were measured for each penstock diameter and nozzle area ratio, and the volumes of water displaced in the reservoirs were also monitored. These measured data were used to compute shaft power and system volumetric flow rate for each operation. Dimensionless flow, head and power coefficients, and specific speed were computed and functional characteristics relating them developed. This standard procedure generally used for the analysis of geometrically similar hydraulic machines have been applied to this system and the results obtained will be invaluable in development of the system into a simple, environmentally friendly and decentralized small power generation system that could potentially contribute positively to the energy mix in Nigeria. The possibility of scaling the system to accommodate larger turbine and penstock diameters, and as a result higher capacity alternators exist and is a target for future developments.

Keywords: Decentralized power, environmentally friendly, net head/flow rate characteristic, nozzle area ratio, penstock diameter and Turbine dimensionless coefficients,

1. INTRODUCTION

Though energy plays a very crucial role in economic development of a nation access to it is very minimal in many developing countries as a result of a mix of several factors [1-8]. In Nigeria, many of the functional energy supply systems operate below installed capacity, and are frequently susceptible to limitations resulting from human and natural causes. Moreover, many of the systems are large, centralized and utilize energy resources that have some adverse impacts on the environment. Furthermore, several of the energy resources in use depleting so that sustainability is not guaranteed [9-15]. Exploration and transportation of new deposits also compound the negative effects on the environment such as oil spillage while escalating friction in the host communities [16-18].

Consequently, there is growing interests in and clamor for the use of renewable energy sources, as well as in smarter, smaller and more decentralized energy systems which will utilize these renewable sources and the existing conventional ones more efficiently [19-31]. These systems convey more control to the end user creating more sense of responsibility with regard to the maintenance and security of the system, especially with the prevalent activities of saboteurs of diverse motivations. Also, the development of systems that generate the required power at or close to the point of application has the potential of mitigating attacks on supply structures particular with the growing regional restiveness in developing countries like Nigeria. Such systems do not require maintenance and protection of the supply structure [17, 32-44].

Hydropower has numerous advantages over other renewable energy sources but the large schemes which are generally predominantly in use in Nigeria and other developing countries, also pose a lot of environmental problems [45-55]. These include harm to aquatic animals and habitat, possibility of enhancement of disease to the neighboring communities, as well as displacement of settlements. There is also growing evidence of emissions from the reservoirs. Large to small hydro which depend on flowing water sources are affected by the hydrological cycle (seasonal fluctuation) which translates to blackouts and significant power outages at some periods of the year. Also, debris and silt blockages of turbine passages often arise which also affect power supply. Evidence also exist of disease enhancement in the region of hydropower reservoirs [56-66].

There is therefore increased interest in very small hydro and pumped storage hydro [67-77]. Pico-hydro power provides a very good option because it suits the general characteristics of smarter, smaller and decentralized systems, and can be utilized in locations where larger conventional systems cannot be optimally located. For instance, it is now a very useful option in the Asian developing countries where the topography is a natural barriers to the uptake of conventional grid-connected energy systems [78-90]. However, it has been verified that seasonal fluctuations of water levels also affect the operation of the conventional Pico-hydro schemes. Low water levels do not allow optimal operation while very high ones can sweep the units away [91-98].

There are many sites suitable for Pico-hydro development in Nigeria as in many other African countries but deliberate focus has not been given to its development [17]. For instance, no direct attention is paid to Pico-hydro systems development in the apparently aggressive efforts of Nigeria's Federal Government to revitalize the hydropower sector [14, 44]. Hence, the development of a Pico-hydro system that may not require naturally flowing water becomes necessary. Developing any means of applying the advantages of hydropower while greatly minimizing the operational and natural shortcomings will be a step in the right direction.

A simplified Pico-hydro system that is a variant of the pumped hydro scheme which could be operated where there is no naturally flowing water by utilizing overhead water storage is currently being developed in University of Agriculture, Makurdi, Nigeria for more than four years now. Such a system will eliminate several of the issues that conventional hydropower systems have to contend with while retaining its substantial advantage as a system for power supply in the mold current renewable energy systems' best practices. It will be decentralized thereby conceding control to the user and reducing the risk of sabotage. The limitation imposed by seasonal variations of water levels on conventional Pico-hydro systems will be eliminated as well [99-107]. The current aspect of the work looks at the prospects for acceptability of this system as a simple contribution to the energy mix in Nigeria. It focusses on the generation of information that will come in handy for future developments of the system.

For all hydraulic machines, it is customary to develop a net head and flow rate characteristic that governs the performance. In conventional hydropower practice, the flow rate and gross head data are collected from the site with the net head obtained from the gross head. This characteristic is therefore invaluable in predicting or fixing the net head and the flow rate for sites where hydropower systems will be installed [108-114]. For this system under development, these parameters are not site-dependent but system component dependent. This means that for this system, the net head and flow rate characteristic will be useful in selecting system components in terms of basic dimensions. In other words, they can be fixed and then used to determine the configurations of the system component.

Furthermore, dimensionless analysis of hydraulic machines yields dimensionless coefficients that are very useful in summarizing the performance of dimensionally similar machines. It is quite useful to have a dimensionless group involving shaft rotational speed, flow rate, head and power with the diameter of the machine. This makes the group independent of the machine size. This can be done by manipulating the other dimensionless groups for the machine to obtain a new dimensionless coefficient. Hence, the coefficients can be used for scaling of system components such as turbine and penstock diameters in order to get a desired power output. The dimensionless coefficients include flow (K_Q), head (K_H) and power (K_P) coefficients as well as specific speed (K_S). For maximum efficiency, there are generally only one set of values for them [108-110, 115]. The functional relationships between these coefficients are experimentally determinable and constitute a set of performance characteristics representing the whole family of geometrically similar machines. They are identical for all such machines if factors such as Reynold's number, Mach number and relative roughness are the same. For all machines belonging to the same family, and operating under similar conditions the dimensionless coefficients are the same at corresponding points of their characteristics.

Hence, according to [110], the similarity laws governing the relationships between such corresponding points may be written as in the equations below.

$$Q \propto ND^3 \quad (1)$$
$$gH \propto N^2D^2 \quad (2)$$
$$P \propto \rho N^3D^5 \quad (3)$$

This work presents the net head and flow rate characteristics as well as the dimensionless flow, head, power and specific speed coefficients of the simple Pico hydropower system undergoing development. The results will be useful for the continued development aimed at arriving at an implementable status for rural and urban locations in Nigeria in a bid to contribute positively to the sustainable energy mix. There will eventually be need to install various capacities for various users depending on several factors ranging from cost to location and the application. These results will come in handy then.

2. MATERIAL AND METHODS

PVC pressure pipes of diameters 0.0762, 0.0635, 0.0508, 0.0445 and 0.0381 m were selected as penstocks. According to [116] and [117], PVC is lighter, has better friction characteristics and is cheaper than steel apart from the subjective factor of being more readily available in the required sizes. Their pressure characteristics are similar. The associated frictional losses were estimated using the equations suggested by [118] for pipes of diameter greater than 5 cm and flow velocity below 3 m/s. An average value of $C = 137.5$ was used in this study because it lies between 135 and 140 for plastic pipes.

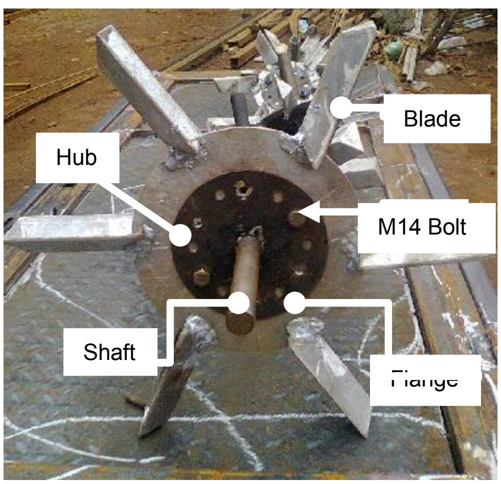
The turbulence losses were estimated with values for the coefficients K for pipe entry, gate valve and 90° elbow obtained from [110] as 0.5, 0.25 and 0.9 respectively. For change in penstock dimensions, K values were obtained using the equation given by [118]. The K values for the reduction of penstock from 0.0762 to 0.0635 m, 0.0635 to 0.0508 m, 0.0508 to 0.0445 m and 0.0445 to 0.0381 m were then computed. H_t values were then computed with only the valve, elbow and entry coefficients applied to the largest diameter penstock. The contraction coefficients were then successively added as the penstock sizes were reduced. The net head available was then computed.

The design procedure for a single nozzle Pelton turbine resembling a propeller turbine was adopted. This is because a propeller turbine allows for the generators to be directly driven thereby avoiding transmissions and the attendant losses. Also, the runners had a relatively lower number of fixed blades, therefore simplifying the manufacturing process and reducing the potential for inconsistent blade construction and orientation. Furthermore, the Pelton turbine can be mounted vertically or horizontally [119-128]. A simple V-shape blade with about 60^0 included angle was adopted. The approach presented by [129] was used in this work in order to obtain the base turbine runner diameters which were then scaled upwards to enhance manufacturability and application for the study [130, 131]. The values of the system flow rate computed were substituted into the expressions for the turbine parameters given by RETScreen. The specific speed of the turbine was computed using j (number of nozzles) = 1 (for simplicity and ease of manufacture). This was used to compute the turbine runner diameter, D_T in metres. Five (5) different values of D_T were obtained corresponding to the five penstock sizes selected which were then scaled upwards. The scaled values of D_T used for this work were 0.25, 0.30, 0.35, 0.40 and 0.45 m. The hub diameter and hence, blade height or cup length was found using an expression given by [117] as well as the blade height. The number of blades was selected from a chart of parameters for sizing turbines by [124] to be 6.

The hub and cups were cast from aluminium after carrying out the necessary preliminary tests and preparations to the sizes obtained. The cups were diametrically welded to the hub using gas welding. Two circular flanges made of 2 mm steel sheet to facilitate the coupling of a steel shaft of 20 mm diameter to the hub is welded to the shaft after passing the shaft through a hole in it. The flange has provisions for three (3) M14 bolts and nuts evenly located along a convenient circumferential plane so that the hub with the cups are clamped perpendicular to the shaft. An average ratio of flange diameter (D_f) to hub diameter (D_h) of 0.75 was used for the 5 turbines. Figure 1 shows the assembled turbine runner. The assembled turbine was mounted in a casing made of 4 mm sheet steel and externally reinforced having an annulus or flow area (A) which satisfies the minimum condition for a clearance of about 0.03 m. Figure 2 shows an assembled turbine. Appropriate bearings and seals were selected for mounting the turbine to facilitate free rotation and to prevent leakages. The casing cover was secured in position using M13 and M14 bolts and nuts. The support of the turbine was made of a combination of 5 mm u-channel and 4 mm angle iron with provisions for four M20 foundation bolts. The exit duct was of rectangular cross-section and tapered to a 76.2 mm diameter internally threaded cylindrical adaptor. The duct was conveniently slanted in order to enhance discharge of water from the turbine. Figure 3 shows an exploded view of the turbine.

The nozzles were fabricated using 1 mm thick steel sheet. The development of each was cut out of the sheet metal which was then appropriately folded and welded using gas welding because of the light gauge of the metal. The nozzles had a mean height of 50 cm. Figure 4 shows all the nozzles used for the study, each set of 5 including nozzles of area ratios 1.0 to 0.2.

Figure 5 shows the complete set up for the study while Fig. 6 shows an enlarged view of the components on the ground. It has two reservoirs, one mounted overhead and the other underground. The arrangement was such that the overhead reservoir delivers water to the turbine through the penstock. Five nozzles of similar length of about 50 cm were fabricated for each penstock diameter with area ratios of 1.0, 0.8, 0.6, 0.4 and 0.2 to facilitate flow acceleration at the exit of the penstock. Water from the nozzles impinges on the turbine blades when the outlet valve of the overhead reservoir is opened. The whole turbine assembly is mounted horizontally with the water outlet port conveniently inclined such that flow from the turbine casing is enhanced. The turbine discharges water to the ground reservoir. The water is then re-circulated to the overhead reservoir by a 1.11 kW DAB Model electric pump. The pump has a rated flow rate of 3.0 – 10.8 m^3/h (0.833 – 3.0 x 10^{-3} m^3/s) with maximum and minimum heads of 29 m and 17 m respectively and 220 – 240V, 7.1A.



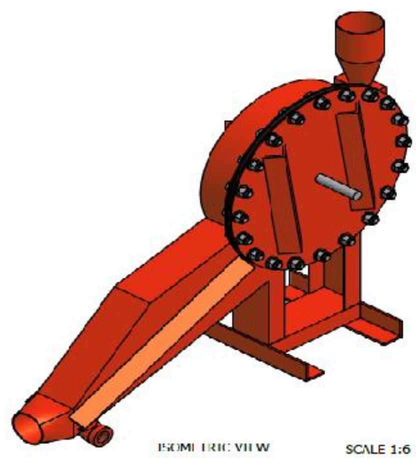


Fig. 1: A Turbine Runner Assembly for the System **Fig. 2: An Assembled Turbine**

For this study, the head, $H \cong 5.95m$. The experimental system discharge was then determined for each penstock size by timing the discharge of water from the overhead reservoir. The rotational speed of the shaft of the turbine (N) was measured using the DT-2268 and DT-2858 Contact Type Digital Tachometer for each penstock diameter and nozzle configuration. The tachometers had a 5-digit, 10 mm LCD display with measurement range of 2.5 – 99,999 Rpm. The resolution is 1 Rpm over 1000 Rpm with accuracy of $\pm$ 0.05% + 1 Rpm and photo detecting distance of up to 300 mm. The tachometers have memory capability of showing the last value, maximum value and minimum value, and a typical sampling time of 1 second.

The measurements were carried out without coupling the alternator to the turbine (no-load tests). The rotor of the tachometer was pressed lightly into a blind hole on the rotating shaft in order to measure the rotational speed. This was repeated several times depending on the duration for a particular measurement which was limited by the water level in the reservoir on the ground. During this period, the maximum and minimum rotational speed were observed and recorded. An average duration of about 4.24 minutes/measurement was used throughout with the minimum and maximum values being

1.73 and 6.75 minutes. The whole procedure was carried out for each of the 5 turbines. The values of N were corrected for losses imposed by the provision for discharging water into the reservoir on the ground by applying a factor of H_d/H, where H_d = the height of the delivery port above the plain of the turbine shaft.

For the 4 smaller penstock diameters, the values of N were also corrected because the delivery pipe to the ground reservoir was not reduced to match their smaller diameters. A factor of D_p/D_d, where D_d = diameter of the delivery pipe and D_p = diameter of penstock. The water levels in the two reservoirs were monitored simultaneously using a dip stick along with a measuring tape and used to obtain the volume of water discharged. The volumetric flow rates were then computed. The fluid power (P_f) available for each operation was computed using the relationship given by [111] and [76]. The shaft power, P_s, and efficiency of the system were computed from first principles using equations given by the same author.

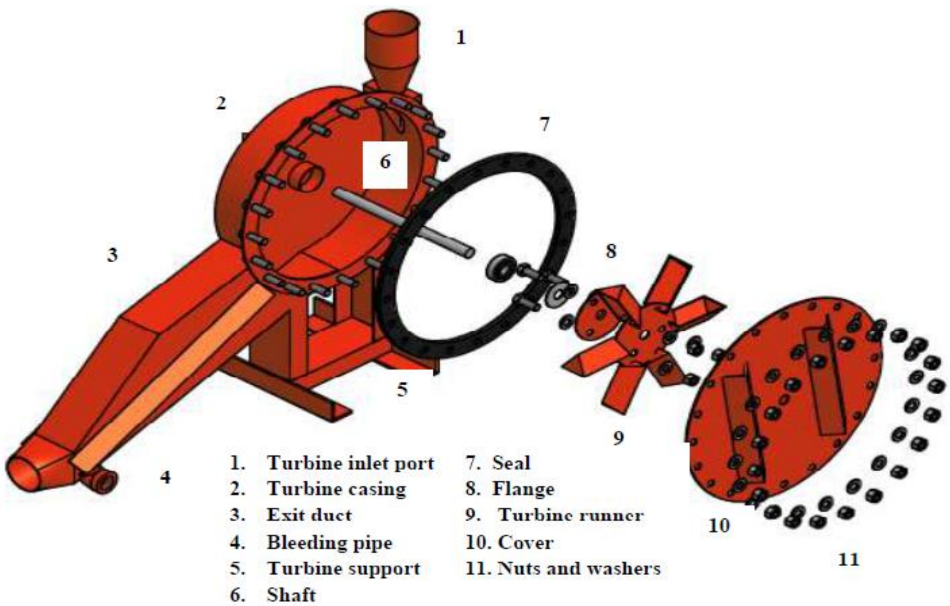


Fig. 3: Exploded view of the turbine

(a) 0.0762 m **(b) 0.0635 m** **(c) 0.0508 m** **(d) 0.0445 m** **(e) 0.0381 m**

Fig. 4: The Nozzles used for the indicated Penstock diameters

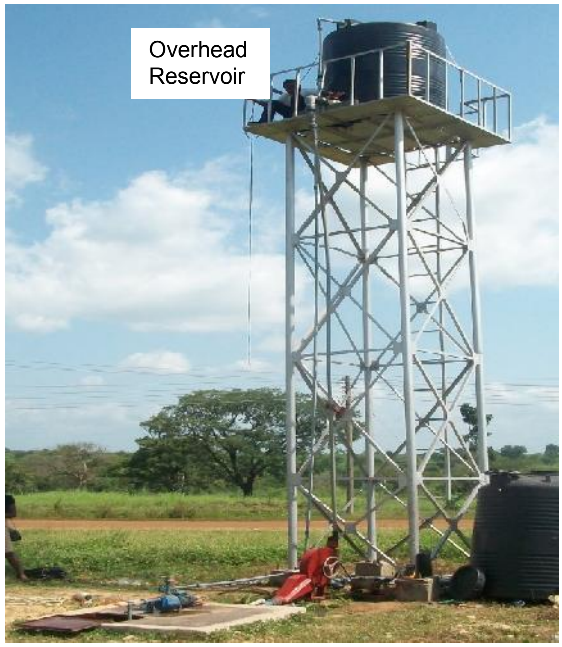


Fig. 4: The Pico-Hydropower System

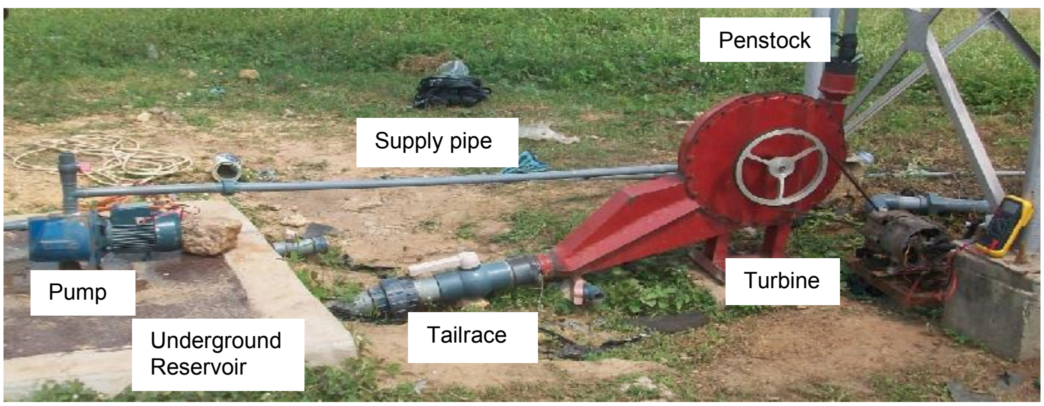


Fig. 5: Enlarged view of the 1.11 kW Pump, Turbine and Penstock

Based on results of dimensionless analysis, the dimensionless groups flow, head and power coefficients as well as specific speed were computed using equations 4 to 7 respectively. The head and power coefficients were plotted against the flow coefficients to formulate a functional relationship between them. They can be computed using the expressions below [108-110].

Flow coefficient, $K_Q = Q/_{ND^3}$ (4)

Head coefficient, $K_H = gH/_{N^2D^2}$ (5)

Power coefficient, $K_P = P/_{\rho N^3 D^5}$ (6)

Specific speed, $K_S = K_P^{1/2}/_{K_H^{5/4}}$ (7)

The net head flow rate characteristic was established for the system.

3. RESULTS AND DISCUSSION

For this study, the mean values of the flow rate and the net head for the no-load tests as presented in Table 1 were plotted in Fig. 6. The characteristic curve was parabolic in nature as is obtainable in previous studies [109, 110, 126, 132-142]. It has the following expression given in equation 8:

$$H_{n,\ avg} = -27132Q_{avg}^2 + 740.6Q_{avg} + 1.5363 \qquad (8)$$

where $H_{n,\ avg}$ = mean system net head (m) and Q_{avg} = mean system flow rate (m^3/s). This expression can be very useful in obtaining an initial design for scaling up flow rate for further developments of the system for given values of $H_{n,\ avg}$ [143-149].

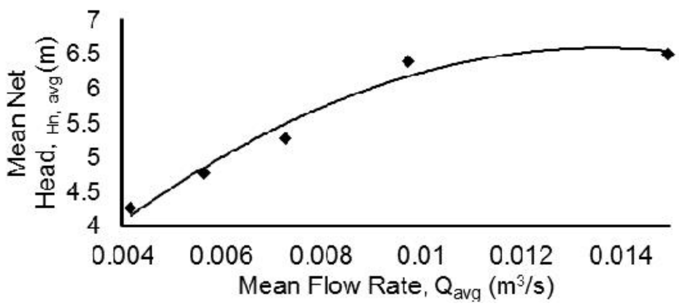


Fig. 6: Mean Net Head and Flow Rate Characteristic for the System

Based on results of dimensionless analysis of hydraulic turbine parameters, four coefficients were computed to summarize and generalize their performance. The coefficients were head, flow and power coefficients as well as the specific speed. They were computed using equations 4 to 7. These formulations will be very useful especially with regards to future plans to scale up the system in order to generate higher power [150, 151]. They will be invaluable for initial design data and are key to the expectation of achieving this system in its eventual application form. The computed values of the coefficients are shown in Table 1.

Figure 7 relates the mean head coefficient (K_H) to the mean flow coefficient (K_Q). For this work, the characteristic curve is parabolic with the expression given in equation 9.

$$K_H = 1765.2K_Q^2 - 1.6098K_Q + 0.0027 \qquad (9)$$

Figure 8 shows the corresponding curve for the relationship between the mean power coefficient and the flow coefficient which also has a parabolic trend. The expression obtained is shown in equation 10.

$$K_P = 3.4689K_Q^2 - 0.0019K_Q + 1 \times 10^{-6} \qquad (10)$$

The coefficients constitute a set of performance characteristics representing the whole family of five turbines that were fabricated for this work. They are identical for all of them as long as parameters such as Mach number, Reynolds's number and relative surface roughness of the pipe walls are the same, or can be assumed constant. This assumption holds for this work. Applying similarity laws and based on the assumptions above, these coefficients can be used to predict the performance of another similar turbine with smaller or larger runner diameter running at a given speed [108-110, 115].

According to [108] and [109], the specific speed (K_S) can be obtained from equation 7 by manipulating K_Q, K_H and K_P. The mean values of the computed K_S from experimental data for each of the family of five turbines is shown in Table 1. They all lie within the range $1.7 < K_S < 3.0$. Though these values are quite small compared to the range of 10 to 35 reported by [111] and [117] for one-jet Pelton turbines, they are close to each other, strengthening an earlier suggestion in the process of the larger scope of the study that the difference between the runner diameters was not large enough to significantly impact upon their performances.

Table 1: Computed Dimensionless Coefficients for the turbines for Penstock of diameter 0.0762 m

Turbine Runner Dia., D_T (m)	Nozzle Area Ratio, A_2/A_1	Head Coeff., $K_H \times 10^{-3}$	Flow Coeff., $K_Q \times 10^{-4}$	Power Coeff., $K_P \times 10^{-6}$	Specific Speed K_S
	1.0	4.196	8.182	3.433	1.735
	0.8	3.832	6.980	2.675	1.715
0.45	0.6	3.104	5.933	1.841	1.852
	0.4	2.887	4.511	1.302	1.705
	0.2	2.373	2.871	0.681	1.576
					1.717
	1.0	3.278	9.073	2.974	2.199
	0.8	2.527	7.014	1.772	2.350
0.40	0.6	2.405	6.296	1.514	2.310
	0.4	2.145	4.319	0.927	2.086
	0.2	2.141	3.207	0.686	1.798
					2.149
	1.0	4.211	12.586	5.300	2.146
	0.8	3.714	10.841	4.027	2.189
0.35	0.6	3.273	9.251	3.028	2.223
	0.4	2.666	6.684	1.782	2.204
	0.2	2.097	4.423	0.928	2.147
					2.182
	1.0	3.581	15.884	5.688	2.723
	0.8	2.475	12.348	3.056	3.166
0.30	0.6	2.144	11.305	2.424	3.375
	0.4	2.118	8.926	1.895	3.030
	0.2	2.066	6.541	1.351	2.639
					2.987
	1.0	3.973	23.152	9.198	3.041
	0.8	3.619	20.402	7.384	3.061
0.25	0.6	3.121	17.412	5.435	3.160
	0.4	3.099	14.209	4.347	2.851
	0.2	3.013	9.441	2.842	2.388
					2.900

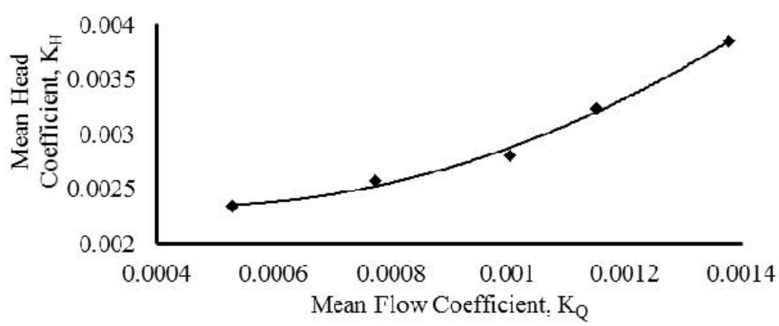


Fig. 7: Variation of Mean Head Coefficient with Mean Flow Coefficient for the Turbines

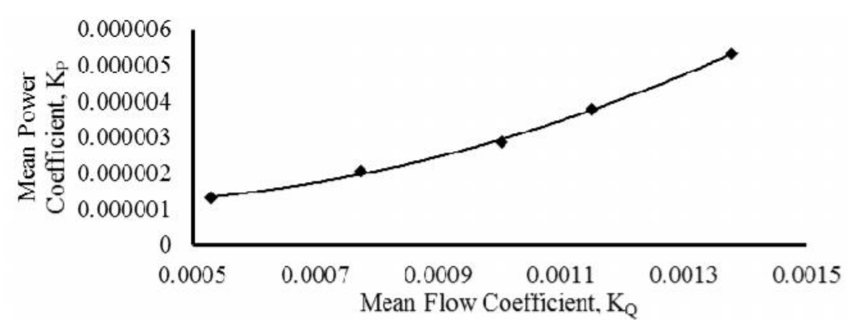


Fig. 8: Variation of Mean Power Coefficient with Mean Flow Coefficient for the Turbines

4. CONCLUSION

So far, the findings in this work on the simplified pico-hydro system show that potential exist for it to contributing positively towards ameliorating the energy crunch in Nigeria and other developing countries as a unit that will operate without dependence on unpredictable climate conditions, without adverse effects on the environment and which concedes control to the end user. Further development is however necessary to realise this potential in full. Its parameters need to be properly manipulated to achieve a self-running status before it can become commercially useful.

The following conclusions are hereby drawn from this experimental study:

(1) Dimensionless groups to summarise the performance of the five turbines used for the study have been formulated which will be invaluable when the system will be modified for better power generation; and

(2) The net head and flow rate characteristic for the system has been established which will be useful for obtaining base data for future work;

The recommendations for this work are issues for the next phase(s). Based on the current findings and the original aspirations of this study, further funding will be sought so that the following aspects could be investigated:

(1) The delivery pipe from the pump will be modified to cause the ratio of delivery to discharge from the reservoir to be more favourable for system performance;

(2) The system will be tested with the overhead reservoir located above 7.0 m to take advantage of greater head;

(3) The effect of multiple overhead reservoirs (or larger capacity ones) will be investigated;

(4) The introduction of solar power for the recycling system in order to explore the hybridization option; and

(5) An economic comparative analysis of this system with a stand-alone solar power system and a fossil fuel powered system will also be undertaken.

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