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Performance Evaluation of Centrifugal Pump-As-Turbine For Micro Hydro Applications

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ABSTRACT

The Philippines is entirely dependent on imported hydropower (HP) system parts, specifically the turbine and generator. The use of a centrifugal pump as a turbine is a good option for the HP system due to its low cost and wide availability in the market. The performance of the centrifugal pump-as-turbine was evaluated under the following conditions: (1) different head setting, (2) different configurations of the draft tube, and (3) a match with an AC generator. A 75x75mm end suction, non-self-priming centrifugal pump was used in this study. The head settings used were 0.28 kg/cm2 to 0.84 kg/cm2 (pressure gauge reading). A prony brake dynamometer was fabricated to measure the torque at the shaft of the pump-as-turbine. The configurations of the draft tube used include: (a) conical angles - 0°, 4°, and 6°, and (b) draft tube lengths – 0.4572 m, 0.9144 m, and 1.3716 m. A 900W AC generator was matched with the pump–as–turbine and tested at different pressure and load settings. Results showed that the centrifugal pump could operate in turbine mode without any mechanical modifications. The highest mechanical efficiency obtained was 83.60% and was attained at a pressure setting of 0.70 kg/cm2. The highest shaft power of 1.02 kW that was attained at 0.84 kg/cm2 pressure setting. Based on the results, in terms of shaft power, the best draft tube configuration has a conical angle of 4° and a length of 1.3716 m. Meanwhile, in terms of efficiency, the best draft tube configuration is 4° conical angle and 0.4572 m length. The generator speed of 4000 rpm was obtained at a pressure setting of 0.84 kg/cm2. On the other hand, a generator speed of 3800 rpm was obtained at a pressure setting of 0.63 kg/cm2. It could be observed that the maximum power (515 W) obtained during the actual run was way below the expected output (847 W). The system efficiencies as well as the electrical power output could improve if the belt slippage is reduced. It is recommended to conduct more testing to be able to determine the full potential of the pump-as-turbine.

Keywords: micro-hydro power, pump-as-turbine, draft tube, generator matching

INTRODUCTION

In the Philippines, the installed capacity of hydropower (HP) is 3,745 MW, as of November 30, 2022 (DOE, 2023). However, these were mostly small to large HP (101 kW and above). Micro-hydro systems are vast in the country since these are small-scale schemes that can be constructed using low initial investments. The government is targeting to achieve the UN SDG Goal 7 (Ensure access to affordable, reliable, sustainable and modern energy for all) through the utilization of micro HP, however, they could also support UN SDG Goal 2 (End hunger, achieve food security and improved nutrition and promote sustainable agriculture), especially in the rural areas.

The main problem in micro-hydro systems is its core component, the turbine (Inversin, 1995), which is a common problem in the Philippines. The country is entirely dependent on imported equipment for hydropower utilization, such as turbines and generators. One of the solutions gaining popularity in terms of ease of acquisition and simple operation is pumps. Furthermore, pumps are simple in construction, making them easy to maintain and operate. It comes in a vast choice of sizes and types to accommodate various heads and flows which make it inexpensive.

In terms of operating conditions, pumps operate in constant head and flow while turbines can accommodate varied head and flow conditions, since there is the presence of adjustable guide vanes or runners which regulate the flow of water through it (Chapallaz, et al, 1992). Another major difference is in their hydraulic design. Pumps decelerate the flow of water by subjecting it through long passages using the impeller and in turn acquire high friction losses. In contrast, turbines are designed to accelerate the water flow, passages are made shorter where friction losses are reduced. Moreover, these are designed to sustain turbulence using guide vanes (Chapallaz, et al, 1992).

Pump manufacturers do not usually provide the characteristic curves of their pumps working as turbines. Therefore, establishing the performance of pump in turbine mode is essential in selecting the proper HP component (Derakhshan and Nourbakhsh, 2008). This study generally aims to evaluate the performance of a 75x75mm non-self-priming centrifugal pump-as-turbine for micro hydro systems application. Specifically, the study aims to determine the performance of the pump-as-turbine: 1) at varying heads using a 75-mm penstock; 2) using different draft tube configurations; and 3) matched with an AC generator.

MATERIALS AND METHODS

The performance of a 75x75mm non-self-priming centrifugal pump was tested in a rig that was specifically put up for the pump to operate in turbine mode (referred to as pump-as-turbine in this paper). The set-up, as shown in Figure 1, was composed of a water supply pump set; pipes and fittings; a receiving tank; a weighing unit; and the pump-as-turbine.

A 100x100mm centrifugal pump powered by a 3.78kW electric motor (**Figure 1**) was used to introduce water to the pump-as-turbine. The pump set supplies the sufficient head and discharge from the bay reservoir to the pump-as-turbine.

From the water supply pump set, the water goes through a tee fitting and enters the pump-as-turbine through the penstock. The penstocks used in the study were made of galvanized iron (GI) schedule 40 pipes with nominal diameters of 75 mm. Water flow may be directed to a gate valve on the other side of the tee fitting. This gate valve controlled the pressure entering the pump-as-turbine. For the low-pressure setting, the gate valve was fully opened; and the water overflowed through the overflow pipe. On the other hand, at a high-pressure setting, the valve is fully closed; no water goes through the overflow pipe. The overflow pipe directed the water back to the bay reservoir. A pressure gauge, with a 1.06 kg/cm^2 capacity, was attached to the penstock to determine the pressure of the water that goes through the pump-as-turbine.

No mechanical or structural modifications were done on the 75x75mm centrifugal pump. The water entered the pump-as-turbine through the discharge

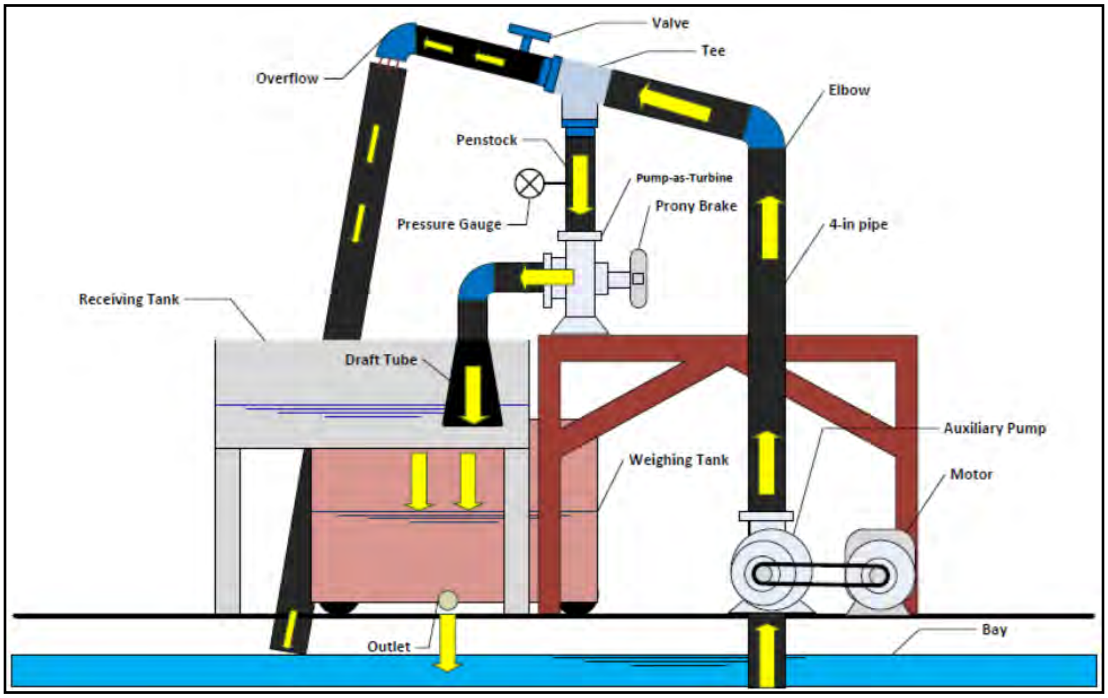


Figure 1. Pump-as-turbine performance test set-up. (Adapted from Reyes, 2011)

outlet and flowed out on the suction side. The pump impeller and the shaft rotated in a clockwise direction. The rotation was opposite from the direction indicated in the pump (Figure 2). Water flowed out of the pump-as-turbine through a discharge pipe connected to a draft tube via a 90-degree elbow. Particularly, the pump used for testing was a TARO 75x75mm PT centrifugal pump (**Figure 2**) and characterized by AMTEC (2010), as a single end-suction centrifugal pump.

Figure 2. Taro 75x75 mm PT Centrifugal Pump. (Source: AMTEC Test Report No. 2010-53)

Through the draft tube, the water was discharged to the receiving tank. From the receiving tank, the water was directed back to the bay reservoir or discharged weighing tank, for discharge measurements. The weighing tank was placed atop a high-capacity weighing scale. After weighing, water was discharged back to the bay reservoir through a gate valve below the weighing tank.

Performance Test Procedure

The tests were conducted at different pressure settings using penstock with a nominal diameter of 75 mm. The pressure was monitored using a pressure gauge that is installed in the penstock. The pressure gauge reading was maintained for every head setting by adjusting the gate valve at the overflow of the discharge side. The head settings used for the test were set at 0.84kg/cm^2 to 0.28 kg/cm^2, with an increment of 0.07 kg/cm^2.

A prony brake dynamometer was fabricated to measure the torque at the shaft of the pump-as-turbine. The dynamometer has an arm length of 0.4

meters from the shaft center. A spring scale was attached to the end of the dynamometer arm for load adjustment readings. Varying loads were applied by adjusting the bolt (load adjuster) on the opposite side of the arm. The bolt was adjusted at specified intervals of 0.2 kg, 0.4 kg, 0.6 kg, 1 kg, and until the shaft rotation reached its lowest possible speed. Water was sprayed on the dynamometer brake drum for cooling since an excessive amount of heat was produced. For each load application, the gate valve at the discharge side was adjusted to maintain constant pressure readings. The corresponding rotational speed of the pump-as-turbine shaft for each load adjustment was measured using a photo tachometer.

For each load adjustment, the corresponding discharge for a specific load adjustment was measured using the gravimetric method, while the initial weight of the weighing tank was measured using the platform scale. The valve below the weighing tank was closed. Moreover, the flow was diverted to the weighing tank from the receiving tank until a certain level inside the weighing tank was achieved. Both the time elapsed and the final weight of the tank was measured. After weight measurement, the valve below the weighing tank was opened to release the water back into the bay reservoir. The difference between the final weight and initial weight was converted into liters and then divided by the time elapsed to get the discharge capacity in liters per second.

From the directly measured parameters: pressure (kg/cm^2), load (kg), rotational speed (rpm), elevation (m), and weight of discharge (kg/s), the following necessary parameters were computed: discharge rate (m^3/s), water velocity (m/s), total head (m), water power (kW), torque (Nm), shaft power (kW) and mechanical efficiency (%). The average of three trials for each parameter was used to determine the performance curves.

Effect of Draft Tube Configurations

Different length and conical size of the draft tube (**Figure 3**) was fabricated using an iron sheet (gage 18 and gage 16) and pipe (GI schedule 40). Three draft tubes had a fixed length of 0.4572 m with

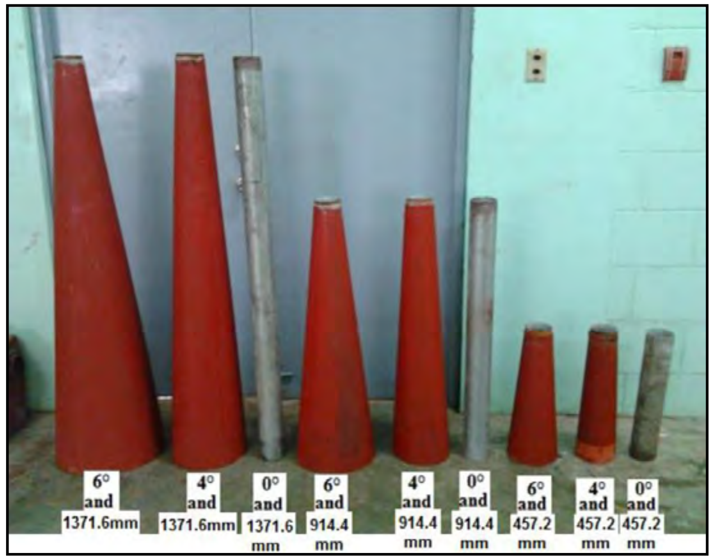


Figure 3. Conical angles and lengths of the draft tubes used in the study. (Adapted from San Pedro, 2012)

variations in conical angles of 0°, 4°, and 6°. Two replicates of the draft tubes were made with lengths of 0.9144 m and 1.3716 m, also with variations in conical angle of 0°, 4°, and 6° for each length.

The pressure heads that were used were 0.84 kg/cm^2, 0.70 kg/cm^2, 0.49 kg/cm^2, 0.42 kg/cm^2, 0.35 kg/cm^2, and 0.28 kg/cm^2. The 0.84 kg/cm^2 and 0.70 kg/cm^2 were for high-pressure settings while 0.42 kg/cm^2 and 0.28 kg/cm^2 were for lower-pressure settings. The effect of the draft tube could be observed better at lower water velocity, hence, the addition of 0.49 kg/cm^2 and 0.35 kg/cm^2 pressure setting. A vacuum gauge was installed to determine the effect of draft tube configurations. The same parameters were measured/determined except for the addition of vacuum pressure (kg/cm^2). Three trials were conducted for every pressure head setting, load adjustment, and draft tube configuration.

The performance of the pump-as-turbine in every draft tube design was compared to each other. The computed values of the discharge, shaft power, water power, and efficiency were compared for all the draft tube designs. The Multivariate Analysis of Variance (MANOVA) was employed in the statistical analysis using the SPSS (Version 20.0.0) software to evaluate the shaft output power and efficiency of the pump-as-turbine. MANOVA is a type of Analysis of Variance (ANOVA), with several dependent variables. ANOVA tests are used to determine the difference in means between two or more groups, while MANOVA tests for the difference in two or more vectors of means (French, et al, n.d.).

Matching with AC Generator

The AC generator used, shown in **Figure 4**, has a rated power of 900 V at 230 VAC. The AC generator came from a gasoline generator set. The generator cover was fabricated as well as the generator footing, for ease of installation and matching with the PAT.

Figure 4. The 900W AC–Generator. (Adapted from Dela Cruz, 2012)

The generator was installed in place of the prony brake dynamometer in the PAT setup (**Figure 1**). It was also operated using a pulley combination of 254 mm pulley diameter on the pump–as–turbine output shaft, and 50.8 mm pulley diameter on the generator.

The loadings were connected to the generator. The loads used were incandescent bulbs that are rated from 10W to 100W. The variations used were increasing and decreasing the load applied while maintaining a constant pressure head. The generator was set to run at 3800 rpm and 4000 rpm. During the trial runs, different parameters were measured: pump shaft speed (rpm); generator shaft speed (rpm); voltage output (volts); current output (amperes); initial weight of the water discharged (lbs.); and the final weight of the water discharged after 10 seconds (lbs.). The shaft speed was measured using a photo tachometer. A clamp ammeter was used to determine the current output while a multimeter was used to measure the voltage output. Based on the measured parameters, the discharge rate (m^3/s), total head (m), water power (W), electrical power (W), belt slippage (%), and system efficiency (%) were computed. Three trials were conducted for every pressure head setting and load adjustment.

RESULTS AND DISCUSSIONS

Performance Curves at Constant Head

Sample performance curves at 0.84 kg/cm^2, 0.70 kg/cm^2, 0.35 $kg/cm^{2,}$ and 0.28 kg/cm^2 for 75mm penstock are shown in **Figures 5 and 6**. Efficiency and shaft power were plotted at a constant pressure gauge setting against the shaft rotational speed. Results showed that the pump can be operated in turbine mode without any mechanical adjustments or modifications. It was observed that efficiency and shaft power was inversely proportional to the shaft's speed. As the shaft's rotation became faster, the shaft power which can be harnessed from the pump-as-turbine decreased. It was also observed that as the pressure gauge setting was decreased, the number of load readings also decreased. Only three load points were measured at 0.28 kg/cm^2 settings compared to 0.84 kg/cm^2 with nine load points.

The summary of values where the maximum efficiency occurred for each constant pressure setting using the 75-mm penstock is presented in **Table 1**. The corresponding values of rotational speed, discharge, and shaft power at each maximum efficiency point are also indicated. No trend was observed on the maximum efficiency with respect to increasing or decreasing the pressure gauge settings. The highest mechanical efficiency obtained was 83.60% and was attained at a pressure setting of 0.70 kg/cm^2. On the other hand, the lowest efficiency obtained was 65.11% at 0.77 kg/cm^2

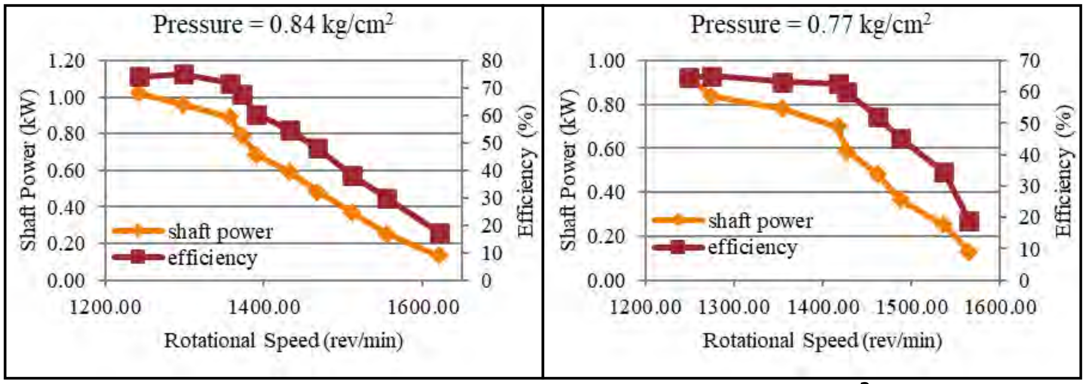


Figure 5. Pump-as-turbine performance curves at 0.84 kg/cm^2 and 0.70 kg/cm^2 pressure settings.

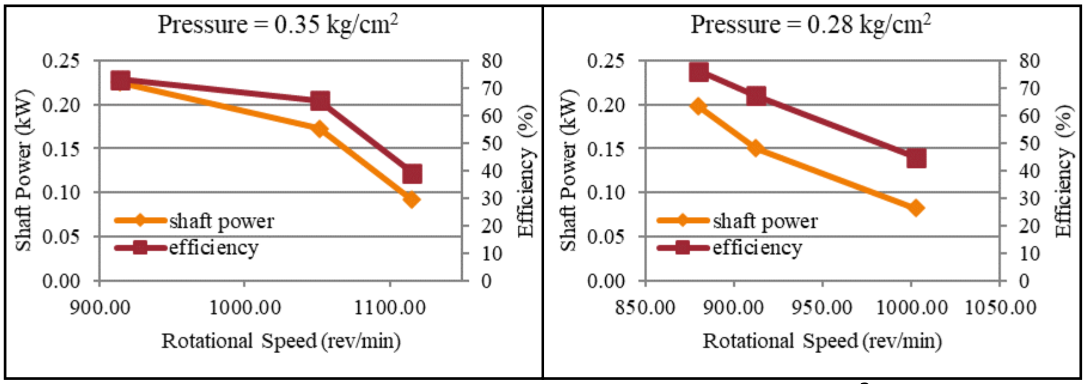


Figure 6. Pump-as-turbine performance curves at 0.35 kg/cm^2 and 0.28 kg/cm^2 pressure settings.

Table 1. Best efficiency points for various pressure settings using 75-mm penstock.

Pressure (kg/cm^2)	Maximum efficiency (%)	Rotational speed (rev/min)	Discharge (Lps)	Shaft power (kW)
0.84	75.07	1298.33	13.33	0.96
0.77	65.11	1274.00	14.33	0.84
0.70	83.60	1308.00	11.12	0.75
0.63	80.32	1108.33	12.16	0.73*
0.56	68.55	1112.00	10.02	0.46*
0.49	66.23	1050.97	8.83	0.35*
0.42	67.25	1050.97	9.06	0.32*
0.35	73.06	914.07	6.92	0.23*
0.28	76.11	880.13	6.91	0.20*

**Maximum power obtained at a given pressure setting.*

setting. Efficiencies are almost identical in both pump and turbine modes, as presented by Derakhshan and Nourbakhsh (2008). On the other hand, the value for pump discharge at the best efficiency point is double compared to turbine discharge, which is the same in the study by Bucur, et al., 2009.

The maximum power obtained at 0.84 kg/cm^2 pressure setting was 1.02 kW; at 0.77 kg/cm^2 was 0.92 kW; and at 0.70 kg/cm^2 was 0.80 kW. The rest of the maximum power at a given pressure setting was also obtained at the maximum efficiency points as shown in **Table 1**. Similar to Yang, et al. (2022), the turbine mode under the same flow rate is much larger than the pump mode, which means that pump reversal can achieve a good power generation effect. Also, the same as the study of Derakhshan and Nourbakhsh (2008), the pump-as-turbine works at a higher head and flow rate than those of the pump mode at the same rotational speed.

Draft Tube Configurations

From the computed values of discharge, shaft power, water power, and efficiency, the effects of varying the configuration of the draft tube were observed. Since the increase in pressure has caused an increase in the rotational speed, the amount of load that it can hold also increases. Shaft power depends on the torque produced by the shaft. Increasing the load application also increases the torque leading to a higher shaft power. Although the shaft power is a product of rotational speed and torque, the decrease in the rotational speed due to load application still produced an increasing shaft power. Based on statistical analysis, the equality of variances among groups implied the use of two-way MANOVA (**Table 2**).

The R Squared and the adjusted R Squared (See **Table 3**) are both almost equal to unity, implying that the angles and lengths may have a strong influence on the performance of the pump-as-turbine in terms of either efficiency or shaft power.

The maximum shaft power (W) obtained at different pressure settings and draft tube configurations are presented in **Figure 7**. Most of the maximum shaft power was observed to be highest at draft tubes with a conical angle of 4° for all pressure settings and draft tube lengths. A slight decrease in the shaft power was observed from the draft tubes with conical angle of 4° and lengths of 0.4572 m and 1.3716 m. The difference in performance of pump-as-turbine in terms of shaft power is clear that using a 1.3716 m draft tube produces higher shaft power. The amount of vacuum developed inside the draft tube increased as the length increased. The variation in the length of the draft tube increased the water power. Increasing the length of the draft tube affected the development of the vacuum inside. This, again, affected the performance of the pump-as-turbine.

Table 2. Covariance test (Box's Test of Equality of Covariance Matrices).

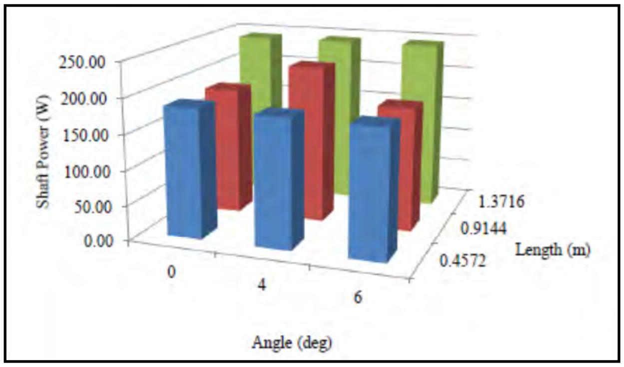
Box's M	34.633
F	1.429
df1	24
df2	1714178.882
Sig.	.079

In **Table 4**, significance values of 0.001 and 0.000 for the logs of shaft power and efficiency indicate that the differences in length significantly affect shaft power and efficiency. On the other hand, 0.096 and 0.005 significance values of the logs of shaft power and efficiency indicate that the differences in angles significantly affect efficiency but not shaft power (**Table 4**). For the combination of length and angle, a 0.025 significance value shows that differences in the combination of angles and lengths could affect efficiency, while the 0.165 significance value indicates that changing the combination of angles and lengths may not affect shaft power (**Table 4**). The general trend of maximum efficiency (%) for every draft tube configuration is presented in **Figure 8**. As shown in **Table 5**, using a 0.4572 m draft tube is the most efficient choice, while using a 0.9144 m draft tube is the least efficient choice. On the other hand, efficiency is highest for an angle of 4° as shown in **Table 5**. This was perfectly captured by the significant values of the mean differences of efficiencies when using an angle of 4° as compared to either 0° or 6° (**Table 5**). Based on the results, the best combination of draft tube angle and length that will produce the highest efficiency of the pump-as

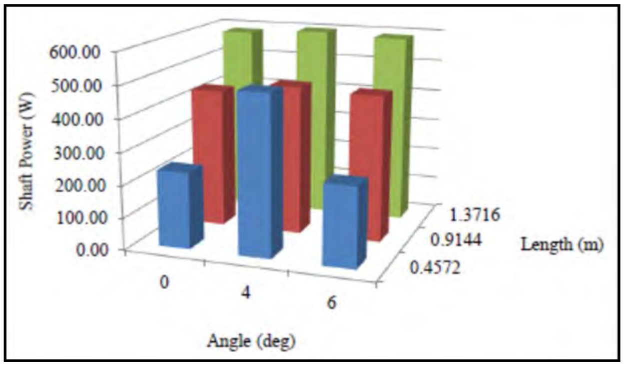
Table 3. The MANOVA Table (Tests of between-subjects effects).

Source	Dependent Variable	Type III Sum of Squares	df	Mean Square	F	Sig.	Partial Eta Squared
Model	log_sp	4732.364[a]	9	525.818	6918.132	.000	.988
	log_eff	2305.030[b]	9	256.114	10922.457	.000	.992
angle	log_sp	.357	2	.179	2.351	.096	.006
	log_eff	.248	2	.124	5.294	.005	.014
length	log_sp	1.019	2	.510	6.706	.001	.017
	log_eff	.385	2	.192	8.208	.000	.021
angle*length	log_sp	.495	4	.124	1.628	.165	.008
	log_eff	.262	4	.066	2.794	.025	.014
Error	log_sp	58.372	768	.076			
	log_eff	18.008	768	.023			
Total	log_sp	4790.736	777				
	log_eff	2323.038	777				

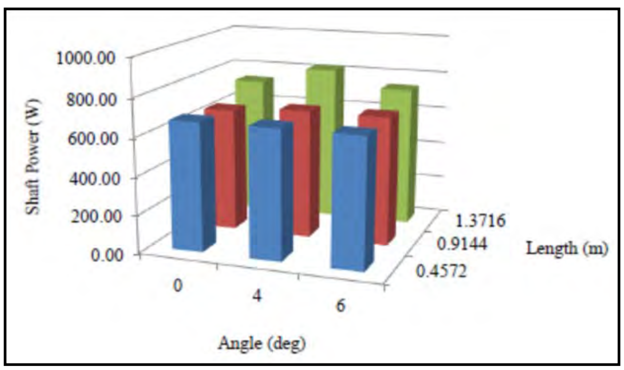
a. R Squared = .988 (Adjusted R Squared = .988)
b. R Squared = .992 (Adjusted R Squared = .992)



(a) 0.28 kg/cm^2 pressure setting



(b) 0.49 kg/cm^2 pressure setting



(c) 0.70 kg/cm^2 pressure setting

Figure 7. Maximum shaft power (W) under different pressure settings.

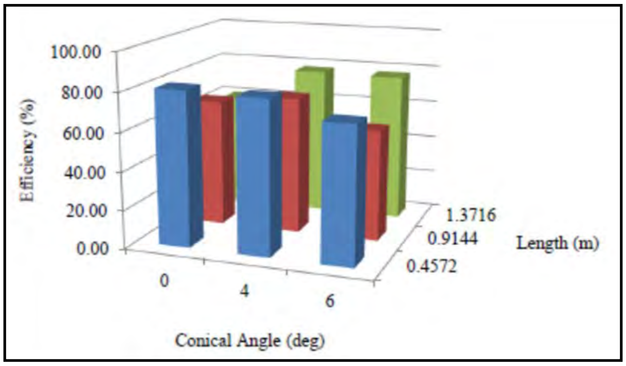
Table 4. Comparison of performance among lengths of draft tubes.

Dependent Variable		Length of Draft Tube (I)	Length of Draft Tube (J)	Mean Difference (I-J)	Std. Error	Sig.
shaft power	Tukey HSD	0.4572	0.9144	-.0076	.02471	.949
			1.3716	-.0764*	.02403	.004
		0.9144	0.4572	.0076	.02471	.949
			1.3716	-.0688*	.02403	.012
		1.3716	0.4572	.0764*	.02403	.004
			0.9144	.0688*	.02403	.012
	LSD	0.4572	0.9144	-.0076	.02471	.759
			1.3716	-.0764*	.02403	.002
		0.9144	0.4572	.0076	.02471	.759
			1.3716	-.0688*	.02403	.004
		1.3716	0.4572	.0764*	.02403	.002
			0.9144	.0688*	.02403	.004
% efficiency	Tukey HSD	0.4572	0.9144	.0583*	.01372	.000
			1.3716	.0280	.01335	.091
		0.9144	0.4572	-.0583*	.01372	.000
			1.3716	-.0303	.01335	.061
		1.3716	0.4572	-.0280	.01335	.091
			0.9144	.0303	.01335	.061
	LSD	0.4572	0.9144	.0583*	.01372	.000
			1.3716	.0280*	.01335	.036
		0.9144	0.4572	-.0583*	.01372	.000
			1.3716	-.0303*	.01335	.023
		1.3716	0.4572	-.0280*	.01335	.036
			0.9144	.0303*	.01335	.023

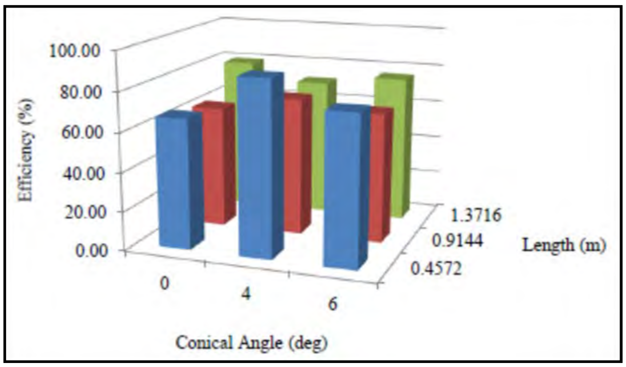
Based on observed means.
The error term is Mean Square (Error) = .023.
**. The mean difference is significant at the .05 level.*

-turbine is 4° and 0.4572 m.

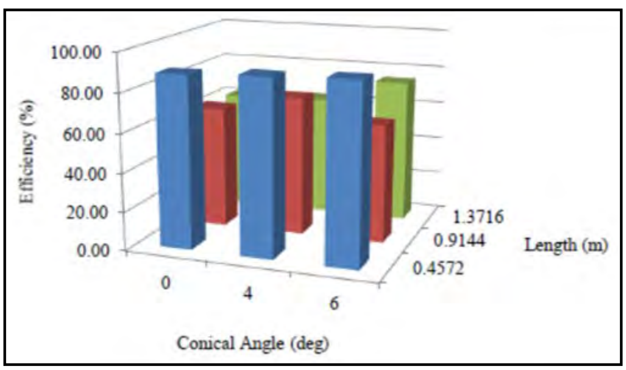
For every pressure setting, different configurations of the draft tube can be used to obtain the highest shaft power or efficiency, and the practicality parameters, in terms of the price and manufacturing, will be met. The recommended draft tube configuration based on shaft power for every pressure setting is summarized in **Table 6**. The shaft power is not significantly affected by the conical angle but is significantly affected by the length, according to the statistical analysis (See **Table 4**). For a pressure setting of 0.28 kg/cm^2, the two highest shaft powers were obtained from draft tubes with a length of 1.3716 m and conical angles of 4° and 6°. Since the difference between the two highest shaft powers is 0.08%, it is recommended to use the draft tube with a 1.3716 m length and a 4° conical angle for economic purposes. For pressure settings of 0.35 kg/cm^2 up to 0.70 kg/cm^2, the highest shaft



(a) 0.35 kg/cm^2 pressure setting



(b) 0.42 kg/cm^2 pressure setting



(c) 0.84 kg/cm^2 pressure setting

Figure 8. Maximum Efficiency (%) at Different Pressure Settings.

Table 5. Comparison of performance among conical angles of draft tubes.

Dependent Variable		Angle of Draft Tube (I)	Angle of Draft Tube (J)	Mean Difference (I-J)	Std. Error	Sig.
shaft power	Tukey HSD	0.00	4.00	-.0358	.02415	.299
			6.00	.0116	.02449	.884
		4.00	0.00	.0358	.02415	.299
			6.00	.0474	.02407	.120
		6.00	0.00	-.0116	.02449	.884
			4.00	-.0474	.02407	.120
	LSD	0.00	4.00	-.0358	.02415	.138
			6.00	.0116	.02449	.636
		4.00	0.00	.0358	.02415	.138
			6.00	.0474*	.02407	.049
		6.00	0.00	-.0116	.02449	.636
			4.00	-.0474*	.02407	.049
% efficiency	Tukey HSD	0.00	4.00	-.0445*	.01341	.003
			6.00	-.0158	.01360	.475
		4.00	0.00	.0445*	.01341	.003
			6.00	.0287	.01337	.082
		6.00	0.00	.0158	.01360	.475
			4.00	-.0287	.01337	.082
	LSD	0.00	4.00	-.0445*	.01341	.001
			6.00	-.0158	.01360	.245
		4.00	0.00	.0445*	.01341	.001
			6.00	.0287*	.01337	.032
		6.00	0.00	.0158	.01360	.245
			4.00	-.0287*	.01337	.032

Based on observed means.
The error term is Mean Square (Error) = .023.
**. The mean difference is significant at the .05 level.*

power was very distinct at draft tubes with a length of 1.3716 m and a conical angle of 4°. For the highest-pressure setting (0.84 kg/cm^2), the difference between the draft tubes with 0° and 0.4572 m configuration and 6° and 1.3716 m configuration was 1.29%, it is better to choose draft tubes with 0° and 1.3716 m design because it is much cheaper and easier to fabricate as compared to the other draft tube design.

The recommended draft tube configuration based on efficiency for every pressure setting was summarized in **Table 7**. The conical angle and length of the draft tube significantly affect the efficiency of the pump, according to statistical analysis (See **Table 3**). The efficiencies under pressure setting from 0.28 kg/cm^2 to 0.49 kg/cm^2 gave a distinct value that is higher among the other configurations, the recommended draft tube design was the configuration where the highest efficiencies were observed. For the pressure settings of 0.70 kg/cm^2 and 0.84 kg/cm^2, the difference between the efficiency of 0° and 0.4572-m configuration and the configuration where the highest efficiencies were observed is very small (0.75% and 3.06%, respectively). It is practical to use the configuration of 0° and 0.4572-m for it is easier to fabricate. It could be noted in **Tables 6 and 7** that draft tube configurations have no effect at the pressure setting of 0.84 kg/cm^2. On the other hand, it could also be noted from **Table 6** that a conical angle of 4° and length of 1.3716 m was recommended for all pressure settings except for 0.84 kg/cm^2. At 0.49 kg/cm^2 pressure setting, the maximum shaft power between 0° and 4° has a slight difference as observed in **Figure 7**. For practical purposes, it would be cheaper to fabricate the 0° conical angle than the 4° configuration. This was also the same for pressure settings of 0.35 kg/cm^2 and 0.70 kg/cm^2 as shown in **Table 7**.

Table 6. Recommended draft tube configuration based on shaft power for every pressure setting.

Pressure Setting (kg/cm^2)	Draft Tube Configuration	
	Conical Angle (deg)	Length (m)
0.28	4	1.3716
0.35	4	1.3716
0.42	4	1.3716
0.49	0	1.3716
0.70	4	1.3716
0.84	0	0.4572

Table 7. Recommended draft tube configuration based on efficiency for every pressure setting.

Pressure Setting (kg/cm^2)	Draft Tube Configuration	
	Conical Angle (deg)	Length (m)
0.28	4	0.4572
0.35	0	0.4572
0.42	4	0.4572
0.49	6	1.3716
0.70	0	0.4572
0.84	0	0.4572

Generator Matching

Table 8 shows the results of the generator calibration test at 4000 rpm. The test at 3600 rpm and 3800 rpm was conducted to determine the generator capacity at lower speeds. The test at 4000 rpm (**Table 8**) showed that the generator produced its rated capacity of 900W and rated voltage of 230 V.

Table 8. The AC generator calibration test results at 4000 rpm.

Load (watts)	Generator shaft speed (rpm)	Voltage (volt)	Current (ampere)
50	4066	240.3	1.1
100	4085	244.2	2.3
150	4093	247.5	1.7
200	4091	252.3	2.4
250	4073	255.2	2.7
300	4089	255.7	2.9
350	4085	257.5	2.8
400	4082	259.0	2.8
450	4078	259.3	2.5
500	4066	258.4	2.6
550	4061	257.3	2.5
600	4052	252.4	2.9
650	4043	249.6	3.0
700	3987	243.8	3.1
750	3982	241.4	3.2
800	3980	239.7	3.3
850	3977	236.1	3.4
900	3975	231.3	3.6
950	3970	225.7	3.8
1000	3962	220.6	3.9

To match the AC generator with the pump–as–turbine, the pump–as–turbine should produce more than 900 W of mechanical power. Based on the results of pump-as-turbine calibration, more than 900 W of mechanical power could be obtained at the pressure setting of 0.84 kg/cm^2, as shown in **Figure 5**.

However, the input power at the generator was not obtained and generator efficiency was not available to determine the mechanical power needed to run the AC generator at rated speed. As such, the diameter of the pulley for the PAT was determined only based on the studies of Limbo (2010), Prudencio (2010), and Catubig (2011). The diameter of the pulley for the PAT was set to 254 mm matched with the AC generator.

Performance Evaluation of AC Generator

Sample performance curves at 4000 rpm generator shaft speed are shown in **Figure 9 and 10**. The generator speed of 4000 rpm was obtained at a pressure setting of 0.84 kg/cm^2. Generally, it showed that as the speed increases, the voltage, electrical power, and efficiency increase, while the current decreases. The same trend was also observed with performance curves at 3800 rpm. The generator speed of 3800 rpm was obtained at a pressure setting of 0.63 kg/cm^2.

The maximum power produced on the calibration test of the AC generator at 4000 rpm was 847.1 W while at 3800 rpm, the maximum power produced was 719.6 W as shown in **Table 9**. It could be observed that the maximum power obtained during the actual run was lower than the expected output. Even if the generator attains the required speed based on the calibration run, it cannot produce the same expected power. The generator produced the maximum power of 525 W but at 4233 rpm, still, it is still way below the expected power output. In this case, the pump–as–turbine could not provide the needed mechanical power by the generator to provide rated electrical power. The generator match was already run at the maximum calibrated pressure setting (0.84 kg/cm^2) of the pump–as–turbine. To increase the mechanical power output of the pump–as–turbine, calibration and running at higher pressure would be needed. However, before subjecting the generator match to higher pressure, one parameter to consider first is the belt slippage. The average belt slippage is presented in **Table 10**.

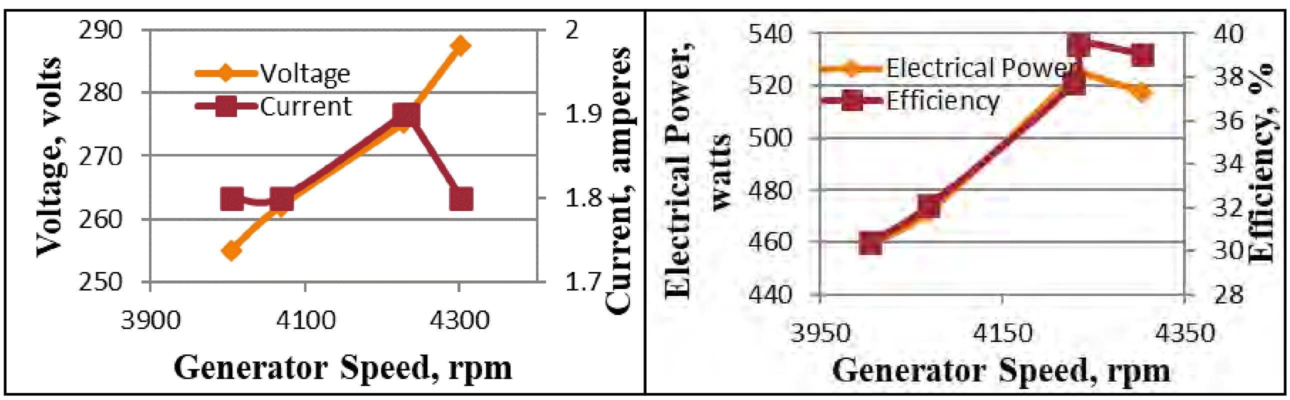


Figure 9. Generator performance curves at 0.84 kg/cm2 pressure setting (decreasing load).

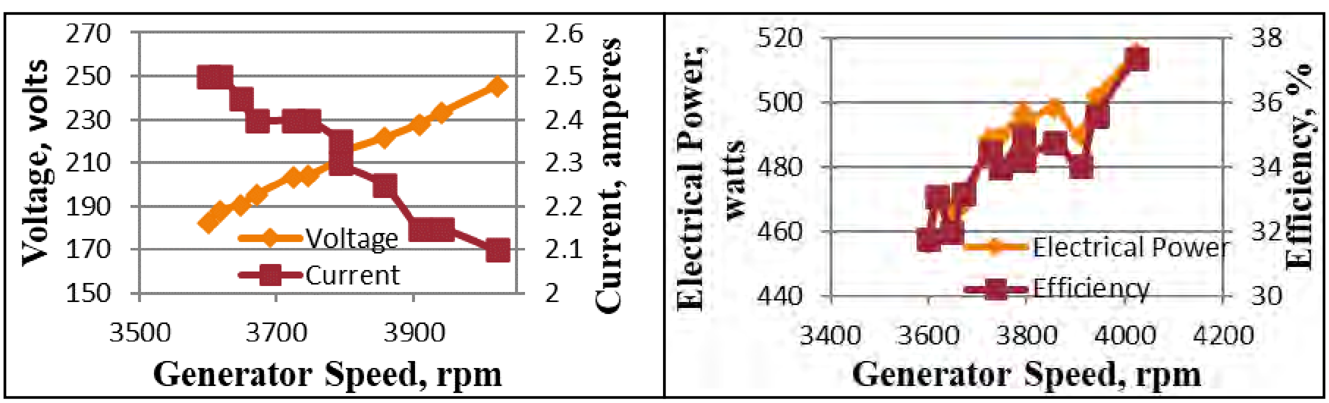


Figure 10. Generator performance curves at 0.84 kg/cm2 pressure setting (increasing load).

The system efficiencies as well as the electrical power output could improve if the belt slippage could be reduced. The belt slippage for the belt and pulley transmission system should only range from 3% to 7%.

Table 9. Maximum power obtained (in Watts) by the AC generator during calibration and actual run at 3800 rpm and 4000 rpm.

	Generator shaft speed (rpm)	
	3800	**4000**
Calibration Test	719.6	847.1
Actual Run	332.6	515.3
% difference	53.8	39.2

Table 10. Average efficiency and belt slippage obtained by the generator match at 3800 rpm and 4000 rpm.

Generator speed (rpm)	Average system efficiency (%)	Average belt slippage (%)
3800	33.6	16.3
4000	34.6	16.8

SUMMARY AND CONCLUSION

A 75x75mm end suction, non-self-priming centrifugal pump was evaluated for its performance as a turbine. The evaluation included the determination of its characteristic curves at different head settings as well as its efficiency using a 75mm penstock. Characterization includes the determination of shaft power and efficiency.

In general, the 75x75mm pump could be operated in turbine mode without any mechanical modifications. The maximum head variation of the pump was subjected to was at an average value of 9.85 meters, with a 0.84 kg/cm^2 pressure setting. While the lowest head variation was at an average value of 3.79 m at 0.28 kg/cm^2 water pressure. Results showed that the highest efficiency of the pump-as-turbine was 86.60% at a speed of 1308 rpm, 11.12 lps of discharge, and 0.75 kW of power, which in terms of efficiency is higher than the pump in normal mode. The maximum shaft power obtained was 1.02 kW.

The performance of the pump-as-turbine was determined for different pressure settings of all the draft tube configurations. From the study, the effect of varying the conical angle of the draft tube resulted in a difference in the water power, shaft power, and efficiency produced. Based on the results, in terms of shaft power, the best draft tube configuration has a conical angle of 4° and a length of 1.3716 m. Moreover, in terms of the efficiency of the pump-as-turbine, the best draft tube configuration is 4° conical angle and 0.4572 m length. The draft tube configurations do not affect the performance of PAT at 0.84 kg/cm^2 which was the highest pressure setting used.

Performance curves of generator match generally showed that as the speed increases, voltage, electrical power, and efficiency increase while current decreases. The same trend was observed with performance curves at 4000 rpm and 3800 rpm. The generator speed of 4000 rpm was obtained at a pressure setting of 0.84 kg/cm^2. On the other hand, a generator speed of 3800 rpm was obtained at a pressure setting of 0.63 kg/cm^2.

The setup was conducted using a pulley combination of 254 mm (pump–as–turbine) and 50.8 mm (generator). It could be observed that the maximum power (515 W) obtained during the actual run was way below the expected output (847 W). The pump–as–turbine could not provide the needed mechanical power by the generator to provide rated electrical power. The generator match was already run at the maximum calibrated pressure setting (0.84 kg/cm^2) of the pump–as–turbine. The system efficiencies as well as the electrical power output could improve if the belt slippage can be reduced. Furthermore, system efficiencies may be improved by calibrating the pump-as-turbine at higher head settings (more than 0.84 kg/cm^2).

RECOMMENDATIONS

Based on the results of the study, the following are recommended to improve the evaluation of the performance of pumps as turbines:

1. Conduct more testing to be able to determine the full potential of the pump-as-turbine;
2. Testing the pump at higher head settings will

determine the effective head range of the pump as a turbine during operation;
3. Field installation of pump–as–turbine micro hydro system to determine actual performance and other areas for improvement; and
4. Improvement and optimization of the mechanical transmission system for improved system efficiency.

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