Experimental Design and Analysis of Pump as Turbine for Microhydro System

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Abstract - Pump as Turbine (PAT) is typically used as electromechanical components in microhydro systems, especially by rural communities in developing countries to reduce initial capital cost. The technology is readily available and easily accessible compared to commercially available turbines. The aim of this paper is to present the experimental design and analysis of PAT for microhydro systems over a range of rotational speeds. An end suction centrifugal pump was tested by inversing the flow across the pump. The rotational speed of the impeller was controlled by manipulating the braking force applied to the output shaft. The corresponding flow rate, pressure, and torque were recorded and presented. The experiment results show that the centrifugal pump can operate in turbine modes without any modification on mechanical components with the highest efficiency of 65.04%; however, at off-design operation, the efficiency decreases significantly due to unmatched flow velocity with the wall boundaries inside the pump.

1. Introduction
Rural electrification plays a significant role in enhancing the quality of life for rural communities by promoting access to modern energy services. This leads to increased economic strength and improves productivity, consequently reducing inequality. The traditional way to supply electricity to rural areas is to extend the national grid into these areas; however, after taking into account financial viability, this approach is usually found to be uneconomical. Using microhydro technologies for off-grid electricity generation is the most suitable method whenever there is an accessible potential site. The use of appropriate technologies to suit local conditions is one of the critical factors affecting the success of an off-grid microhydro electrification system [1, 2]. Economic feasibility is one of the main considerations associated with microhydro systems. Hence, the application of low-cost equipment for the microhydro system has always been the central focus for rural electrification.

The use of Pump as Turbine (PAT) in place of commercial electromechanical components has proven to be a feasible, low-cost solution. Some successful microhydro projects have been reported for rural area sites, especially in developing countries [3-5]. There has been growing interest on utilising the PAT as a substitute for the commercial microhydro turbine, in particular for generating power between the ranges of 10 kW to 25 kW. Even though the PAT has lower efficiency than commercial turbines, its availability covers a wide variety of flows and heads, which makes microhydro projects more economical and practical. Also, the PAT offers benefits such as simple construction, easily-attainable spare parts, readily-available maintenance services, and installation that can be carried out by local
laymen. These desirable characteristics of the PAT are indeed important, as they reduce dependence on third-party professional services, which can be very expensive.

Even though there are many ways to predict the behaviour of PATs, such as simulation analysis, theoretical frameworks and prediction models, the best approach is to test the pump at a predefined operation range of flow rate and pressure. Each PAT is unique to its application; thus, it needs to be tested in its working environment. There are many prediction and simulation models; however, they are limited to certain applications and cannot definitively predict the actual performance. Therefore, the experimental results will give specific indicators of its operational performance. The outcome of the experimental work will accurately present real operational conditions from the perspective of microhydro applications [6]. Pump manufacturers normally supply performance curves of pump modes but seldom provide the turbine mode operation. The testing procedure to acquire PAT performance is similar to pump test standard, ISO 3555:1977 [7]. The characteristic of PAT is highly dependent on the shape and operation characteristic. The efficiency of PAT for low specific speeds varies between 60% to 80% [8, 9]. The effectiveness of the turbine mode is usually inferior to the pump mode.

This paper presents an experiment work which involved the design and construction of a hydraulic test rig and the turbine performance experiment procedure. The hydraulic test rig was designed and fabricated on a laboratory scale as a modular system to allow for a high degree of flexibility and adaptability. The main purpose of this study was to test the PAT to evaluate its operational abilities in a laboratory environment. Establishing the practical operation of PAT was critical to investigate its actual performance.

2. Methodology

Figure 1 shows the schematic of the hydraulic test rig that was adopted for the PAT test. The main components consist of a feed pump, PAT, test bed, PVC piping network and control valves. Bends and tees were used as the pipe connectors and fittings. All the pipes and fittings were fitted with flanged joints and connected with bolts and nuts. Two ball valves were installed in-line with the piping system as the regulation mechanism.

![Figure 1. Schematic view of the hydraulic test rig](Image)

The operating parameters were recorded by the flowmeter, pressure gauge, and torque sensor. The flowmeter was installed in-line with the pipe, measuring the water flow rate before it entered the PAT. Two pressure gauges were positioned at each end of the PAT to measure pressure difference across the device.

The driving force behind the flow of water across the PAT was recorded as a pressure drop at the corresponding flow rate. The torque sensor was mounted in-line with the rotating shaft and measured
the rotating speed and torque produced by the rotating impeller. This type of testing involved recording the torque required to maintain a constant speed.

The hydraulic test rig was an open-loop system. The inlet and outlet piping systems were joined to two water tanks that acted as a reservoir. The open-loop system allowed a continuous water supply to the piping system. The inlet of the feed pump was positioned at the lower water tank line to ensure sufficient suction pressure. This served to prevent flow separation and reduce air suction to the feed pump. The pressurised water flowed through the piping network to the PAT before being returned to the water tank.

2.1 Test Rig Main Components and Sensors

The hydraulic test rig consists of eight main components and sensors as describe below. Table 1 indicates the specifications of the feed PAT.

| Parameters          | Feed Pump | PAT  |
|---------------------|-----------|------|
| Maximum flow rate, Q (l/s) | 20.0      | 8.0  |
| Maximum pressure, H (m)   | 18.5      | 14.0 |
| Suction Head (m)       | 7.0       | 4.0  |
| Specific Speed         | 1453.0    | 1075.0 |

**Feed pump**: The feed pump that was used to supply pressurised water to the PAT was a monoblock pump with a power rating of 2.2 kW and 2900 rpm maximum speed. The pump used an induction generator with a single-phase power input. The feed pump was mounted on a large fixed frame bed and secured by a bolt connection. This kept the pump attached and stabilized thus reducing the vibration generated during the experiment. This pump will produce a maximum water pressure of 18 m at a flow rate of 20.0 l/s. The additional pressure at the pump inlet will be added to the total pressure generated at the pump outlet.

**Pump as Turbine**: The centrifugal pump (Euroflo brand) used as the Pump as Turbine was a low specific speed pump that delivered high pressure at a low flow rate. Its dimensions and performance standards were in accordance with the EN733:1995/DIN24255. It was manufactured locally in Malaysia. The impeller had a diameter of 214.0 mm, and the flange diameter was 50.0 mm at the inlet and 65.0 mm at the outlet. The pump’s power rating was 2.2 kW, and the rotational speed was 1450 rpm. The best efficient point was at 65.0% with a corresponding pressure and flow rate of 14.0 m and 8.0 l/s, respectively. The non-dimensional specific speed (N_s) of the pump was 1075.0 units. The pump had a bare shaft coupled with a torque sensor that rotated in-line with the shaft axis.

**Piping system**: The pipes were made from polyvinyl chloride, uPVC (Class D, (PN12)), and complied with MS 628:1991 standards. The wall thickness was recorded at 3.0 mm, with the permissible working pressure of 12 bars at 20°C Celsius. Reducers and expanders were used to connect the pipes to the pump that had a different diameter. The uPVC pipe was fitted with a set of connectors and fittings including tees, bends, and a valve with flange ends.

**Water tank**: Two water tanks were connected and combined by a pipe at the side wall of the tanks. One control valve was used to adjust water flow between the tanks. The combined tanks had a total volume capacity of 1800 liters. The water level was kept at a defined height to generate sufficient suction pressure at the feed pump inlet. A wooden pallet held the tank foundation with a steel frame for support.

**Control valve**: The feed pump did not have a control mechanism to regulate the flow and pressure generated at the pump outlet. Two control valves controlled the flow rate. The flow rate was adjusted by manipulating the opening of the control valve and diverting a portion of the flowing water from the piping system to the water tank, by passing the PAT.
**Pressure gauge:** The pressure gauges were placed at the inlet and outlet of the PAT. The pressure difference between the two points gave the pressure difference across the PAT. The digital pressure gauges were an Ashcroft® D1005PS with a measuring range of 0 psi to 30 psi.

**Flowmeter:** The real-time flow rate of the water in the PAT was recorded using a digital flowmeter. It was a propeller type flowmeter that had an NPT connector at both ends with model number TM200-N. The flow range was 76.0 to 760.0 l/min. The maximum pressure rating was 225 psi at 23°C. The diameter of the flow rate was 50.0 mm, similar to the diameter of the uPVC piping system.

![Figure 2](image.png)

**Figure 2.** Schematic drawing of braking system with torque sensor and fittings

**Braking system:** The braking system used mechanical brakes that applied a frictional force to a rotating disc brake. A motorcycle hydraulic brake system was adapted and fixed at the end of the torque sensor using a sprocket brake hub. The brake system applied a torsional force, and the torque sensor recorded the measurement. Figure 2 illustrates the braking system with torque sensors and fittings.

2.2 **Accuracy of the Performance Data**

The manufacturer’s data sheets give the following performance data. The percentage of error for the flowmeter is rated to be no more than +/- 3.0%. The percentage terminal point of accuracy pressure at the PAT inlet was rated to be no more than +/- 0.25%. The percentage of combined error for the torque sensor was rated to be no more than +/- 0.1%, based on the manufacturer’s calibration.

2.3 **General Assembly and Operation Procedure**

Figure 3 shows the completed hydraulic test rig that was used to test the PAT while Figure 4 illustrates the torque sensor with the brake system installed in line with the PAT shaft. The arrangement of rig components was adapted to suit the existing hydraulic lab facilities. The rig was designed to have modular components, which would allow for a simple system that could be upgraded for future research. The hydraulic test rig was fabricated and installed in the Mechanical Laboratory of the University Malaysia Sarawak in mid-2015. The experimental analysis results were collected the following year. Intensive preliminary experiments were performed before the rigorous testing to make certain that the data collected was reliable.

The feed pump supplied a continuous flow to the piping system. The water was contained in the reservoir and was open to atmospheric pressure. A tee with a ball valve was used for flow rate control between the PAT and feed pump. The flow was adjusted by regulating the opening of the two valves by hand.

The water flow rate measurement was collected before the water entered the PAT using a propeller type flowmeter that was installed in-line with the pipe. As the water entered the PAT, the impeller rotated and produced rotational power. The impeller shaft was coupled with a torque sensor and braking system. The desired rotational speed was set by applying the brake to the shaft. During this process, the torque sensor recorded the torque and rotational speed produced by the rotating shaft. Meanwhile, the
corresponding pressure across the PAT was recorded with a pressure gauge positioned at the inlet and outlet of the PAT.

Each experimental process gave a set of data consisting of flow rate $Q$, pressure $P$, torque $\tau$ and rotational speed $N$. Measuring the flow rate and pressure were quite complicated since pressure pulsation from the impeller caused readings to fluctuate. There was no data logger for the flow rate and pressure, so data was acquired manually. Because of the fluctuations in data readings, each experiment’s results were averaged over a range of several seconds, and standard deviations were calculated and presented.

The results were presented in accordance with the given objective of investigating the performance of the PAT. Given that the sensitive parameters were interdependent, the rotational speed was the only parameter that was manipulated. The remaining parameters of pressure, torque and flow rate corresponded accordingly. The purpose of the experimental work on the PAT was to determine the efficiency, flow rate and torque over the operation rotational speed.

Figure 3. Complete hydraulic test rig
### 2.4 Experimental Expressions

The PAT is a mechanical device that converts energy from the flowing, pressurised fluid to the impeller. This conversion of energy can be expressed in mathematical equations. Considering that the PAT impeller generates torque $\tau$, with a rotational speed of $N$ (rev/min), the power produced from the shaft can be expressed by Equation (2).

Mechanical power = (torque) (angular velocity)

$$P_{\text{mechanical}} = \tau \times \omega \text{ (Watt)}$$  \hspace{1cm} (1)

$$P_{\text{mechanical}} = \frac{2\pi N \tau}{60} \text{ (Watt)}$$  \hspace{1cm} (2)

The pressure difference across the PAT is denoted by $H$. Therefore, the available hydraulic power input to the PAT is found in equation (3).

Hydraulic power = (water density) (gravitational acceleration) (flow rate) (pressure)

$$P_{\text{hydraulic}} = \rho g Q H \text{ (Watt)}$$  \hspace{1cm} (3)

The conversion of energy consists of power losses. These power losses can be categorised as mechanical losses and hydraulic losses. The mechanical loss includes bearing and shaft seal losses, while hydraulic loss consists of disc friction loss, flow separation loss, recirculation loss, incidence loss, shock loss, mixing loss and leakage loss. The expression of total energy conversion is

Total hydraulic power input = (mechanical losses) + (hydraulic losses) + (mechanical power)

Where,

Mechanical loss = (bearing loss) + (seal loss)

Hydraulic loss = (flow friction loss) + (mixing loss) + (recirculation loss) + (incidence loss) + (disc friction loss) + (leakage loss)
The efficiency of the PAT is therefore expressed in Equation (4).

\[ \eta = \frac{P_{\text{mechanical}}}{P_{\text{hydraulic}}} \times 100(\%) \]  

(4)

Likewise, the equation for efficiency can be written as shown in Equation (5).

\[ \eta = \frac{2\pi N\tau}{60} \frac{6}{\rho g QH} \]  

(5)

3. Results and Discussions

The PAT was allowed to run continuously for 15 minutes before readings from the sensors were recorded. This allowed the piping to be filled with water and ensured that the sensors were recording steady readings. The initial information that was collected consisted of visual observations, i.e. leakages and vibrations. The results showed that the centrifugal pump was able to run in turbine mode without any mechanical failure. A small amount of vibration was observed, but was at a minimum level and was, therefore, acceptable. The vibration may have been caused by hydraulic oscillations (pressure pulsations/recirculation/rotor-stator interaction).

The rotational speed of the PAT was adjusted to between 800 rpm and 1400 rpm by adjusting the braking force applied to the rotating shaft. The corresponding pressure, flow rate, and torque were measured and documented. The mechanical power, hydraulic power, and efficiency were determined based on the observed parameters and mathematical expression of energy conversion from Equations (2), (3), and (4).

All measurements collected from the sensors were subjected to some uncertainty due to the hydraulic test rig setup limitations, simplifications of the experimental procedure and uncontrolled changes to the test environment. In the area of experimental uncertainties, the value of standard deviations from the data was presented to demonstrate how widely the measured values spread by the average of min values. The statistical analysis to determine the experimental uncertainties can illustrate the degree of accuracy and precision for the experimental results.

Tables 2, 3 and 4 illustrate the min values of flow rate, pressure, and torque with their respective standard deviations. These values were collected simultaneously at their respective operation rotational speeds.

### Table 2. Mean and standard deviation values for flow rate

| Rotational speed, N (rev/min) | Flow rate, Q (gal/s) | Standard deviations, σ |
|-------------------------------|----------------------|------------------------|
| 1400                          | 99.52                | 1.52                   |
| 1300                          | 119.81               | 1.60                   |
| 1200                          | 125.372              | 1.48                   |
| 1100                          | 126.62               | 1.82                   |
| 1000                          | 128.40               | 1.39                   |
| 900                           | 131.50               | 1.22                   |
| 800                           | 133.40               | 1.80                   |

### Table 3. Mean and standard deviation values for pressure

| Rotational speed, N (rev/min) | Pressure, H (Psi) | Standard deviation, σ |
|-------------------------------|-------------------|-----------------------|
| 1400                          | 17.00             | 0.21                  |
| 1300                          | 14.30             | 0.22                  |
| 1200                          | 14.11             | 0.19                  |
| 1100                          | 13.90             | 0.25                  |
### Table 4. Mean and standard deviation values for torque

| Rotational speed, N (rev/min) | Torque, $\tau$ (Nm) | Standard deviations, $\sigma$ |
|-----------------------------|---------------------|-----------------------------|
| 1400                        | -                   | 0.31                        |
| 1300                        | 3.50                | 0.56                        |
| 1200                        | 3.70                | 0.63                        |
| 1100                        | 4.70                | 0.39                        |
| 1000                        | 4.80                | 0.55                        |
| 900                         | 5.10                | 0.66                        |
| 800                         | 6.40                | 0.77                        |

Figures 5, 6 and 7 illustrate the mean and standard deviations of the respective parameters drawn from the 100 to 1000 values from the data logger for each rotational speed.

**Figure 5.** Mean and standard deviations of flow rate vs. rotational speed

**Figure 6.** Mean and standard deviations of pressure vs. rotational speed
Figure 7. Mean and standard deviations of torque vs. rotational speed

Table 5. Experiment results

| Rotational speed, N (rev/min) | Torque, \( \tau \) (Nm) | Mechanical power, \( P_{\text{mechanical}} \) (watt) | Flow rate, \( Q \) (l/s) | Pressure, \( H \) (meters) | Hydraulic Power, \( P_{\text{hydraulic}} \) (watt) | Efficiency, \( \eta \) (%) |
|-----------------------------|------------------------|-----------------------------------|-----------------|-----------------|-----------------------------------|-----------------|
| 800                         | 6.40                   | 539.65                            | 10.11           | 9.38            | 929.96                            | 58.03           |
| 900                         | 5.10                   | 483.79                            | 9.96            | 9.25            | 903.66                            | 53.54           |
| 1000                        | 4.80                   | 505.92                            | 9.73            | 9.03            | 861.56                            | 58.72           |
| 1100                        | 4.70                   | 544.92                            | 9.59            | 8.90            | 837.83                            | 65.04           |
| 1200                        | 3.70                   | 467.98                            | 9.30            | 8.84            | 806.41                            | 58.03           |
| 1300                        | 3.50                   | 479.57                            | 9.25            | 8.42            | 764.37                            | 62.74           |
| 1400                        | 0.00                   | 0.00                              | 7.54            | 7.00            | 517.58                            | 0.00            |

It can be observed that the pressure and flow rate that enters the impeller steadily decreasing between 800 to 1300 RPM. The steady decline of flow rate and pressure across the PAT is because, at higher rotational speed, the water flows faster. The torque profile exerted by the impeller shows that higher torque was generated at lower rotational speed. The torque steadily reduces as the rotational speed increases. It should be noted that this is not an ideal experimental setup to achieve complete PAT performance. However, it was sufficient to test the Pump as Turbine at a predefined operation range.

Table 5 shows the experiment results. The highest rotational speed was recorded at 1400 RPM at free-running speed, which was the highest speed recorded by the torque sensor when no brake force was applied to the shaft. The lowest rotational speed has been registered at 800 rpm, generating the highest torque at 6.40 Nm.

The corresponding operational flow rate for the experimental set-up was recorded between 7.0 l/s and 10.0 l/s, similar to pump operation mode at a flow rate of 8.0 l/s. The highest efficiency (65.04%) was attained at 1100 rpm. The highest power was generated by the PAT at 800 rpm, with mechanical power of 929.96 watts and a corresponding efficiency of 58.03%. From these results, it can be determined that the torque increase as the rotational speed decrease. The efficiency varied, but not prohibitively so.

It should be noted that highest efficiency of the pump was 65.0% at 8.0 l/s and 14.0 m of pressure. Results from the experimental work showed that the efficiency was higher in turbine mode than in pump mode. However, the induction generator performance was excluded from the analysis. The overall performances normally take into account the complete component systems when considering efficiency.
in pump mode. Nevertheless, the mechanical efficiency of 65% is reasonable if we examine the savings from the lower acquisition cost of the turbine system.

It should be highlighted that these results are unique to the pump that was selected for the analysis. The results should be applied to other pumps with caution, and comparisons should be made considering the same specific speed, but for different sizes, shapes, and operation ratings.

4. Conclusion
This paper outlined the experimental testing of PAT performance using a hydraulic test rig. The rotational speed of the PAT was modified by adjusting the torque applied to the shaft. The corresponding pressure, torque and flow rate of the PAT system were measured using sensors and a data logger. Meanwhile, the vibration and leakages on the seal packing were observed. From the experimental results, it can be concluded that the centrifugal pump could run in turbine mode with satisfactory operational performance.

The ideal flow rate for the pump in turbine mode was higher than in pump mode; however, due to the constraints of the hydraulic test rig, the control parameters were limited to the designed test rig set-up. For future research, it is recommended that some of the system components, such as feed pump capacity and diameter of the pipe, are upgraded. This will allow a higher flow rate and greater pressure to be transferred to the pump; consequently, a higher rotational speed that matches the synchronous speed of the induction generator could be achieved, allowing full performance characteristics to be acquired.

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