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# An Experimental Study of a Pico Hydro Turbine

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**Abstract**

Today, the storage and transporting of electricity from one place to another is still an unsolved problem. Besides, it is not economical to install power lines everywhere for low-voltage electronic circuits such as valves which are used to reduce leakage in water lines. When water demand is lower, it is a practical solution to use the pico hydro turbines to supply the required electrical energy for the electronic circuits of the valves used to reduce the losses in the water discharge lines. For this purpose, it is experimentally investigated electricity production in a pico hydro turbine and validated with experimental data that a pico hydro turbine (PHT) generates 1 W electricity under a water flow velocity of 0.53 m/s. It is concluded that the higher resistances were used in the test rig, the less turbine power produced. As the voltage increased, the current obtained decreased, and the turbine power reached its maximum at a resistance of 130 Ω.

**Keywords:** Pico hydro turbine, hydropower, banki turbine, energy efficiency, experimental setup

## 1. INTRODUCTION

Pico hydro turbines are frequently used for opening and closing valves, which regulate the flow rate in water distribution pipelines. Also, it is an energy harvesting device that uses the energy of a water pressure drop. This energy is captured by the pico hydro turbine and converted into electricity.

Most studies have been conducted on numerical investigation of cross-flow water turbines as opposed to this current study [1, 2].

Borkowski et al. [3] tested and validated the single-equation Spalart-Allmaras (SA) model, as well as the k-ε and k-ω shear-stress transport (SST) turbulence models, in an experimental hydroelectric power plant with a power of 75 kW. The calculation results

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according to the turbulence models differed significantly. As a result of the comparisons, the Spalart-Allmaras model provided the best experimental results. Verma et al. [4] analyzed runner speed and efficiency variation with different attack angles, nozzle tip height, and nozzle tip distance from the runner. The runner consists of 12 blades symmetrically arranged between the two circular end plates around the plate. They prepared the installation for places with relatively moderate or low head consideration. While the inlet angle $\beta_1$ of the vane was kept between about 26° and 28°, and the outlet angle $\beta_2$ was 90°. Namely, the outlet was radial. Critical parameters such as nozzle inlet angle, nozzle tip height, and horizontal distance from the center of the runner shaft were changed for different heads under different field configurations. Sinagra et al. [5] designed and tested a 10 kW prototype of a new compact in-line turbine called the Power Recovery System (PRS) in the University of Palermo hydraulic laboratory. They proposed a Banki-type microturbine with positive outlet pressure and a mobile regulating vane for hydraulic control of the characteristic curve. They aimed to control the pressure or discharge in a water transport or distribution network by producing energy up to 76% of the efficiency value of the device. Using two pressure gauges located just before and after the turbine made it possible to control the net head in inactive mode at the given discharge point or to control the discharge by regulating the net head in active mode. Sammartano et al. [6] carried out a numerical and experimental study to validate a previously proposed design criterion for a cross-flow turbine and a new semi-empirical formula that relates the inlet velocity to the inlet pressure. An experimental test stand was designed to perform a series of experiments and measure the efficiency of the turbine designed according to the proposed criteria. Experimental efficiency was compared with numerical simulations using a Reynolds-Averaged Navier-Stokes (RANS) turbulence model. The proposed semi-empirical velocity formula was also validated with numerical solutions for cross-flow turbines with different geometries and boundary conditions. They verified the previous hydrodynamic analysis with obtained results. Therefore, they found that it can be used to design cross-flow turbines to reduce the cost of simulations required while optimizing the turbine geometry. Elbatron et al. [7] proposed a new system configuration to extract as much kinetic energy as possible from the water stream. This system, known as a bidirectional diffuser augmented (BDA) channel, works by using bidirectional nozzles in the flow. The proper angle is essential to direct its flow so that it touches the wings vertically to capture as much torque and power as possible. Therefore, experimental and numerical studies were carried out and verified with each other to examine the performance characteristics of the cross-flow turbine (CFT) configuration applied to the BDA system and to investigate the effects of the inlet and outlet angles of the blades on the internal flow properties and efficiency of the CFT runners. Dragomirescu et al. [8] presented the results of the experimental and numerical study of a Banki turbine with a horizontal nozzle operating at 5% and 30% below nominal values and discharges. In line with the experimental results, they predicted that a maximum efficiency of over 55% could be reached even if there were sharp decreases in the values of the working parameters. Numerical simulations were performed to get an idea of the flow through the turbine. It has been determined that the obtained numerical results were compatible with the experimental results. Sammartano et al. [9] tested the performance of a cross-flow water turbine, which provides high efficiency in a wide water discharge range, by experimentally working on cross-flow micro-turbines and verified a new approximate formula that relates the inlet pressure to the main inlet velocity. They found that the experimental tests and CFD analysis results were in agreement with each other, and both experimental studies and CFD computations provided a consistent reduction in machine efficiency due to increased internal energy losses of the internal shaft. Giosio et al. [10] investigated a comprehensive range of micro hydro turbine units recently. They presented an affordable micro-hydro turbine unit with an improved operating range. As a result, utilizing a pump impeller as a runner provided an efficiency of 79% and near-peak efficiency at off-design conditions. Moreover, this micro-hydro turbine unit had the ability to power supply remote areas.

The availability of a continuous energy source enables optimal pressure management that is continuously maintained at a lower value to reduce water leaks, improve network efficiency, and extend pipeline lifetime. The main objective of the present study is to experimentally investigate the performance of PHT and its characteristics through varying the resistance (Ω) and voltage ($V$). In this study, the electrical power produced by a pico hydro turbine (PHT) was experimentally tested by measuring the average volumetric flow (Q), voltage, and current under real conditions with an experimental setup.

## 2. MATERIALS AND METHODS

The hydraulic power of a PHT depends upon the flow rate ($Q$) and the water head ($H$), which are the significant parameters that affect the turbine performance exposed on the turbine shaft. While impulse turbines are adapted for sites with high heads and low flow conditions, reaction turbines such as cross-flow and Banki turbines perform best at low heads and high flow rates. These main parameters determine the runner diameter ($D$), angular velocity ($w$), turbine power output ($P$), and output torque ($T$) in a water turbine. Several equations were considered to have accurate measurements for hydroelectric power while using different amounts of resistances during the experiment in this study. By assuming no friction through the pipe and the turbine, available hydraulic power ($P_H$) generated by water can be measured as a function of the amount of volume flow rate ($Q$) and water head ($H$) at incompressible flow conditions. Since there is no available water head for the PHT in this study, $P_H$ can also be measured based on voltage ($V$) and current ($I$) in line with resistance ($R$), whose equations represented by Eq. (1) are as follows:

$$P_H = \rho Q g H = VI \quad (1)$$

Where $P_H$ is available hydraulic power, $\rho$ is the water density, presented by a value of 997 $\mathrm{kg/m^3}$ at ambient temperature, $g$ is the acceleration of gravity, with a value of 9.81 $\mathrm{m/s^2}$, $Q$ is the volume flow rate passing through the turbine ($\mathrm{m^3/s}$), $H$ is the available water head at the turbine. It is equal to the power provided by voltage and current generated by the turbine shaft by using a wide range of resistances in this study.

Furthermore, $P_M$ is the mechanical power output of the PHT turbine and is defined as the product between angular velocity (ω) and torque ($T$) on the turbine shaft by Eq.(2).

$$P_M = T\omega \quad (2)$$

As a result of Eq.(1) and Eq.(2), experimental turbine efficiency ($\eta$) is as follows

$$\eta = {}^{VI}/_{T\omega} \quad (3)$$

### 2.1. Experimental Setup and Design Parameters

A cross-flow turbine at a pico scale was investigated to examine the effects of several geometric factors involved in the design, including turbine diameter ($D$), the number of blades ($z$), and volume flow rate ($Q$) on maximum turbine performance. The tested turbine shown in Figure 1 is comprised of a runner with 32 mm of inner diameter and 40 mm of outer diameter. The total number of turbine blades is 24. The PHT turbine is made up of plastic material in the turbine body and blade. The hydraulic test bench consists of a system of pipes connecting an inlet and outlet of the turbine whereby the water was recirculated.

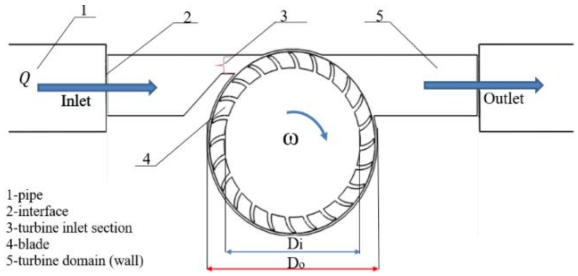


Figure 1 Schematic of 2-D tested turbine model flow surface section

Experiments have been carried out in the hydraulic research laboratory (HRL) of the faculty of Mechanical Engineering of Sakarya University (Turkey). The test stand comprises a PHT turbine coupled with resistances, flowmeter, pipe, multimeter, and probes on a test bench, as shown in Figure 2. Flowmeter has been used to measure the volume flow rate of the water through the pipe. The water volume flow rate has been set to be the maximum flow rate after several attempts and varied from 0.25 L/s to 0.167 L/s at the inlet and outlet of the turbine. The velocity $(v)$ at the turbine outlet can be calculated by using the continuity equation as follows:

$$v = {}^{Q}/_{A} \quad (4)$$

$A$ represents the cross-sectional area of the runner. The velocity of PHT was obtained with a value of 0.53 m/s under the volume flow rate of 0.167 L/s.

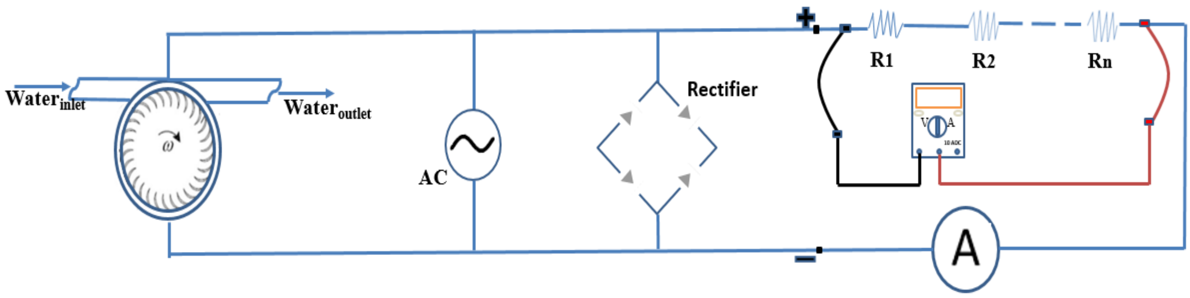


Figure 2 The test-rig model schematic diagram established in HRL

There is also one calibrated multimeter used to measure each voltage to determine the maximum hydraulic power by using three amounts of resistances in the test rig. It has been respectively used 10 Ω, 15 Ω, and 100 Ω, which correspond to 2050 Ω resistances totally during the experiment.

## 2.2. Experimental Procedure and Uncertainty Analysis

The actual water volume flow rate and the voltage generated by the turbine have been recorded by flowmeter and multimeter, respectively, to calculate input power. The runner velocity has been calculated by using a flowmeter to reach maximum velocity in PHT. Also, voltage coupled with resistance has been measured using a multimeter to reach maximum power. Therefore, it has been prioritized for two parameters to analyze maximum uncertainties. Parameter accuracy and maximum uncertainties for power and volume flow rate have been determined by several tests with forty-eight voltage value point that equal 4%. Moreover, multimeter uncertainty has been determined by Mc-Clintock equation, and it has been found to be 0.5%. Also, the uncertainty of the flowmeter used to measure water volume flow rate is 0.09%.

Using Mc-Clintock, the maximum uncertainties of $P$ and $Q$ are represented in Eqs. (5) and (6), respectively:

$$W_P = \pm\left[\left(\frac{\partial P}{\partial V} W_V\right)^2 + \left(\frac{\partial P}{\partial I} W_I\right)^2\right]^{1/2} \quad (5)$$

$$W_Q = \pm\left[\left(\frac{\partial Q}{\partial V} W_V\right)^2 + \left(\frac{\partial Q}{\partial A} W_A\right)^2\right]^{1/2} \quad (6)$$

## 3. RESULTS AND DISCUSSIONS

Response surface methodology (RSM) has been conducted to establish relationships among selected parameters referring to a set of mathematical and statistical techniques to create a regression model characterizing affecting parameters of turbine performance. For this purpose, it has been calculated power increase by means of coupled different amounts of resistance in the electric circuit among several trials. It has been split into three categories starting from the first 10 Ω resistance intervals between 10 Ω and 200 Ω ($R_1$), 15 Ω between 200 Ω and 350 Ω ($R_2$), and 100 Ω between 350 Ω and 2050 Ω ($R_3$). The experimental study matrix and regression models are obtained on ANOVA by combining the considered factors and levels, which are R-square/adjusted ($R^2$/adj.), p-value, sum of squares (SS), degrees of freedom (df), mean square (MS) and F-ratio, etc. for each treatment according to RSM. In this regard, several regression models representing the turbine power performance have been created, including linear and polynomial models, as shown in Table 1.

Table 1 ANOVA for fitted polynomial and lineal regression model using RSM for PHT

| Resistance model | $R^2$ | $R^2_{adj}$ | MS | SS | F-ratio | df | p-value | Regression model |
|---|---|---|---|---|---|---|---|---|
| $R_1$ | 0.99 | 0.99 | 0.73 | 1.46 | 1254.12 | 18 | 0.06 | Polynomial |
| $R_2$ | 0.99 | 0.99 | 0.05 | 0.10 | 6448.83 | 8 | 1.61 | Polynomial |
| $R_3$ | 0.96 | 0.95 | 0.08 | 0.16 | 182.10 | 15 | 0.558 | Polynomial |
| $R_1$ | 0.54 | 0.51 | 0.79 | 0.79 | 22.35 | 19 | 0.40 | Lineal |

The PHT model significance and adequacy have been analyzed by the p-value of the regression model and the $R^2$ and $R^2_{adj}$, respectively. The results analysis shows the fact that the polynomial regression model had a p-value of more than 0.05 (0.06), and the highest $R^2_{adj}$ (99.20%) unlike the linear regression model. Furthermore, the value corresponding to $R^2$ for the linear regression model revealed that a minimum 40% of the deviation of the obtained experimental data for the resistance model-$R_1$ is shown in Figure 3 and Table1. Therefore, the polynomial regression model was selected because this model gives a higher significance and an adequate regression model which fits the best for PHT. The equation referred to polynomial equation model defining the maximum turbine power is given by Eq. (7).

$$y = 0.061 + 0.014x - 5.442\text{x}10^{-5}x^2 \quad (7)$$

Where y is ascribed to the maximum turbine power and $x$ are the defined levels for the turbine voltage values. As a result of the experimental study between related voltages, it has been obtained that the PHT generates approximately 1 W against 130 Ω, which is a value similar to that one predicted by the polynomial model.

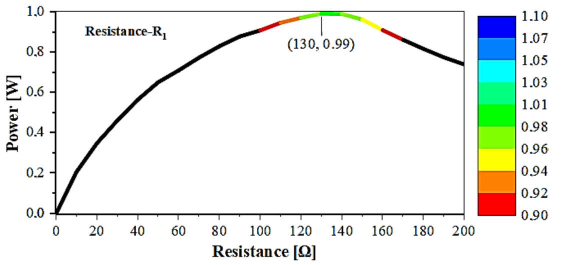


Figure 3 Power-$R_1$ until 200 Ω

The polynomial model has also been applied to the resistance-$R_2$ and $R_3$ models shown in Figure 4 and Figure 5, respectively. Figure 4 represents the experimental voltage data versus power for the optimal conditions found at 200 Ω. This value has turned out to be at 350 Ω in Figure 5.

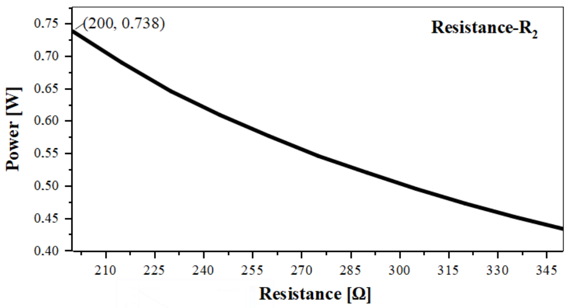


Figure 4 Power-$R_2$ until 350 Ω

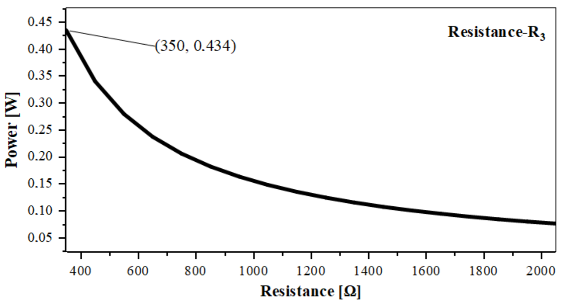


Figure 5 Power-$R_3$ until 2050 Ω

While the maximum power is 0.73 W for the resistance-$R_2$ model, it is obtained that the power is a maximum of 0.43 W for resistance-$R_3$. As a result of this study, it is important to note that the kinetic energy is generated from the water-energy by means of a pipe. When the flow rate is gradually declining towards the turbine outlet, it is observed that even though the amount of resistance is increasing, the power generation is decreasing depending on the current.

## 4. CONCLUSION

This study involves the experimental study to demonstrate turbine performance with a set of voltages evaluated with the RSM procedure under the regression model. In the current work, the turbine experimental study is conducted to obtain maximum electrical power value using a resistance of 2050 Ω in a test rig through a multimeter and flowmeter. It was concluded that PHT generates 1 W output power at 130 Ω under the water velocity of 0.53 m/s at operating flow conditions. It is also observed that as the resistance amount increases, the turbine generates less power. In this regard, minimum power is measured at the last resistance point, 2050 Ω. RSM is an efficient tool to provide a prediction model demonstrating turbine performance. In this context, the polynomial regression model predicted maximum power turbine efficiency is approximately 1 W when $R^2$ is equal to 0.99. The experimental results obtained in this study are essential in terms of validating the numerical models to be made in the future.

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***Authors' Contribution***

The authors contributed equally to the study.

***The Declaration of Conflict of Interest/ Common Interest***

No conflict of interest or common interest has been declared by the authors.

***The Declaration of Ethics Committee Approval***

This study does not require ethics committee permission or any special permission.

***The Declaration of Research and Publication Ethics***

The authors of the paper declare that they comply with the scientific, ethical and quotation rules of SAUJS in all processes of the paper and that they do not make any falsification on the data collected. In addition, they declare that Sakarya University Journal of Science and its editorial board have no responsibility for any ethical violations that may be encountered, and that this study has not been evaluated in any academic publication environment other than Sakarya University Journal of Science.

NOMENCLATURE

| | |
|---|---|
| $\beta_1$ | Inlet angle [°] |
| $\beta_2$ | Outlet angle [°] |
| ω | Angular velocity [rad/s] |
| $\eta$ | Turbine efficiency [-] |
| A | Cross-sectional area [$m^2$] |
| BDA | Bidirectional diffuser augmented [-] |
| CFT | Cross-flow turbine [-] |
| D | Turbine diameter [mm] |
| g | Gravity [$m/s^2$] |
| df | Degrees of freedom [-] |
| H | Head [m] |
| HRL | Hydraulic research laboratory [-] |
| I | Current [A] |
| MS | Mean square [-] |
| P | Power [W] |
| PHT | Power hydro turbine [-] |
| PRS | Power recovery system |
| $P_H$ | Hydraulic power [W] |
| $P_M$ | Mechanical power [W] |
| $Q$ | Volume flow rate [$m^3/s$] |
| R | Resistance [Ω] |
| RANS | Reynolds-Averaged Navier-Stokes [-] |
| RSM | Response surface methodology [-] |
| $R^2$ | R-square [-] |
| $R^2_{adj}$ | R-square adjusted [-] |
| SS | Sum of squares [-] |
| SA | Spalart-Allmaras [-] |
| SST | Shear-stress transport [-] |
| T | Torque [N.m] |
| V | Voltage [V] |
| $v$ | Velocity [m/s] |
| z | Number of blades [-] |

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