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Hydraulic Performance Assessment of a Low-Head Cross-Flow Turbine for Pico-Hydropower Applications: An Integrated Analysis of Flow Rate, Torque, Shaft Power, and Turbine Efficiency

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ABSTRACT

Pico-hydropower is a viable solution to produce electricity using small amounts of water from a water source, thereby reducing the need for significant infrastructure in rural and remote settings. The cross-flow turbine is well suited and has simple construction and stable operation for low head, variable flow conditions, among the available turbine types. An integrated hydraulic performance evaluation of the low head cross flow type turbine for pico-hydro application is presented here, where the relationship among water head, flow rate, hydraulic power, runner speed, torque, shaft power and turbine efficiency have been examined. The framework translates basic equations of flow rate, hydraulic power, angular velocity, shaft power, tip-speed ratio and hydraulic efficiency into narrative language. Illustrative examples demonstrate that the higher the flow rate and the greater the effective head, the greater the hydraulic power, and the higher the overall efficiency, depending upon the efficiency of the runner in converting the water energy to shaft output. The analysis emphasizes the fact that turbine output power is not the only criterion for performance—the hydraulic input power and conversion loss also play a role. This study has contributed a simple yet comprehensive mechanical-engineering approach to analyzing pico-scale cross-flow turbines under low-head conditions which can be used in laboratory testing, small-scale turbine development, and deployment of such turbines for renewable energy in rural water channels.

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1. INTRODUCTION

Small-scale hydropower is an important renewable energy technology because it can convert the potential and kinetic energy of flowing water into mechanical and electrical power. Pico-hydropower systems are particularly relevant for rural and remote areas because they can operate using small rivers, irrigation channels, and low-head water resources. Compared with large hydropower plants, pico-hydropower systems require smaller civil infrastructure and can be developed using simpler mechanical components [1].

A cross-flow turbine is one of the turbine types commonly considered for small and pico-hydropower applications. This turbine allows water to pass through the runner twice, first from the outer side toward the inner region and then from the inner region toward the opposite outer side. This two-stage energy transfer mechanism allows the runner to extract energy from water flow with relatively simple blade geometry [2], [3], [4]. Recent studies continue to investigate cross-flow turbines because this technology is suitable for low-head hydropower systems and can be improved through geometry optimization, nozzle design, and internal flow analysis [5], [6].

The performance of a cross-flow turbine depends on several hydraulic and mechanical parameters, including water head, flow rate, nozzle geometry, blade number, runner diameter, runner speed, torque, and shaft power. Recent numerical investigations also show that blade number and internal flow behavior strongly affect the performance characteristics of low-head cross-flow turbines. Therefore, turbine design should not only focus on fabrication simplicity but also on the interaction between water flow and runner geometry [7], [8].Several recent studies have used computational fluid dynamics to analyze cross-flow turbine performance. CFD-based studies have investigated blade shape, inlet discharge angle, blade number, and flow field distribution to improve turbine efficiency. For example, a 2024 study on pico-scale cross-flow turbine blade configuration reported that cross-flow turbines have potential for rural and remote power generation and that blade configuration influences performance improvement. A 2025 study also reported that optimization of blade number and inlet discharge angle can affect turbine performance in CFD simulation [9], [10].

Although advanced CFD optimization is valuable, many educational laboratories and rural technology development projects require a simpler experimental framework to evaluate turbine performance. A practical framework must explain how water head and flow rate produce hydraulic power, how the runner converts hydraulic power into shaft power, and how efficiency is calculated from the relationship between input and output power. Therefore, this study proposes an integrated hydraulic performance assessment framework for a low-head cross-flow turbine [11], [12]. The novelty of this study lies in the formulation of a Q1-style mechanical engineering framework that connects flow rate measurement, hydraulic power estimation, runner torque, shaft power calculation, tip speed ratio interpretation, and turbine efficiency evaluation in one coherent structure. This framework can support experimental turbine development before more advanced CFD or optimization studies are conducted.

Literature Review and Theoretical Framework

Pico-Hydropower System

A pico-hydropower system is a small-scale power generation system that converts water energy into mechanical and electrical energy [13], [14]. The system generally consists of a water source, intake channel, penstock or flow passage, turbine runner, shaft, generator, and electrical load. In a low-head system, the available water height is limited, so the turbine must be able to extract useful energy from relatively small pressure differences and moderate flow rates.

The basic working principle of a pico-hydropower system is based on energy conversion. Water stored or flowing at a certain elevation has potential energy. When water flows through the turbine, part of this energy is transferred to the runner. The rotating runner produces torque on the shaft. The shaft then drives a generator to produce electricity. The performance of this system depends on how much hydraulic energy enters the turbine and how much mechanical power can be extracted at the shaft.

Cross-Flow Turbine Working Principle

A cross-flow turbine is an impulse-type or partially reaction-type turbine in which water enters the runner through a nozzle and crosses the runner blades [15], [16]. The water jet passes through the first stage of the blade row, enters the internal runner region, and then passes through the second stage before leaving the turbine. This flow path allows the turbine to extract energy in two interaction stages. The cross-flow turbine is attractive for low-head applications because it has simple construction and can operate under variable flow conditions. The runner consists of curved blades arranged around a cylindrical structure. The nozzle directs water toward the runner at a specific angle. The interaction between the incoming water jet and the blade surface produces force, torque, and rotational motion.

The energy transfer process in a cross-flow turbine depends on the direction and velocity of water entering the runner. If the nozzle directs water effectively toward the blade inlet angle, the runner can absorb more momentum from the water. If the inlet flow direction is not suitable, flow separation, impact loss, and internal turbulence may increase. Therefore, nozzle and runner design are important for improving hydraulic efficiency.

Water Head

Water head represents the vertical energy level available to drive the turbine. In hydropower systems, head is one of the most important variables because it determines the amount of potential energy that can be converted into hydraulic power [17], [18]. The effective head can be expressed as the difference between the upstream water level and the turbine outlet level after considering head losses.

The effective head can be written as:

$$H_e = H_g - H_l$$

In this equation, $H_e$represents the effective head available at the turbine. The term $H_g$represents the gross head measured from the water source to the turbine outlet, while $H_l$represents the head loss caused by friction, bends, entrance losses, and flow disturbances in the channel or pipe. This equation means that the turbine does not receive all gross water head because some energy is lost before water reaches the runner. A larger effective head increases the available hydraulic energy. However, in low-head systems, the effective head is usually small. Therefore, reducing head loss becomes important. Smooth flow passages, proper nozzle design, and reduced friction losses can help maintain effective head and improve turbine performance.

Flow Rate

Flow rate describes the volume of water passing through the turbine per unit time [19], [20]. It can be calculated using:

$$Q = A \times V$$

In this equation, Qrepresents the water flow rate. The variable Arepresents the cross-sectional area of the flow passage, while Vrepresents the average water velocity. This equation means that flow rate increases when the flow area or water velocity increases. For example, if the nozzle has a cross-sectional area of 0.003 m² and the average water velocity is 4 m/s, the flow rate is calculated as:

$$Q = 0.003 \times 4 = 0.012\ \mathrm{m}^3/\mathrm{s}$$

This result means that 0.012 cubic meters of water pass through the turbine every second. In turbine performance analysis, flow rate is essential because it determines the amount of water energy entering the turbine.

Hydraulic Power

Hydraulic power represents the theoretical power available from water before it is converted into shaft power by the turbine [21], [22]. Hydraulic power can be calculated using:

$$P_h = \rho g Q H_e$$

In this equation, $P_h$represents hydraulic power. The symbol ρrepresents the density of water, $g$represents gravitational acceleration, Qrepresents flow rate, and $H_e$represents effective head. This equation means that hydraulic power increases when water density, gravitational acceleration, flow rate, or effective head increases. Since water density and gravitational acceleration are nearly constant in most practical cases, the main design variables are flow rate and effective head. In low-head hydropower, the head is limited, so sufficient flow rate is required to produce useful power. Conversely, if flow rate is too small, the available hydraulic power becomes limited even when the turbine design is efficient.

For example, if the water density is 1000 kg/m³, gravitational acceleration is 9.81 m/s², flow rate is 0.012 m³/s, and effective head is 2.5 m, the hydraulic power is calculated as:

$$P_h = 1000 \times 9.81 \times 0.012 \times 2.5$$

$$P_h = 294.3 \text{ W}$$

This value means that the water supplies approximately 294.3 W of hydraulic power to the turbine.

Runner Speed and Angular Velocity

Runner speed describes how fast the turbine runner rotates. It is commonly measured in revolutions per minute [23], [24]. However, shaft power calculation requires angular velocity in radians per second. Angular velocity can be calculated using:

$$\omega = \frac{2\pi N}{60}$$

In this equation, ωrepresents angular velocity, while Nrepresents runner speed in revolutions per minute. The constant 2πrepresents one complete revolution in radians, and the value 60 converts minutes into seconds. This equation means that angular velocity increases when runner speed increases. For example, if the turbine runner rotates at 600 rpm, the angular velocity is calculated as:

$$\omega = \frac{2\pi \times 600}{60} = 62.83 \text{ rad/s}$$

This result means that the runner rotates with an angular velocity of 62.83 radians per second.

Torque and Shaft Power

Torque represents the rotational force produced by the turbine runner. In a hydropower turbine, torque is generated when the water jet transfers momentum to the runner blades [25], [26]. The shaft power can be calculated using:

$$P_s = T \times \omega$$

In this equation, $P_s$represents shaft power, Trepresents torque, and ωrepresents angular velocity. This equation means that shaft power increases when torque or angular velocity increases. A turbine may produce high torque at low speed or lower torque at higher speed, but the useful power depends on the product of both. For example, if the turbine produces torque of 3.6 N·m and angular velocity of 62.83 rad/s, the shaft power is calculated as:

$$P_s = 3.6 \times 62.83 = 226.19 \text{ W}$$

This result means that the turbine produces approximately 226.19 W of mechanical power at the shaft.

Turbine Efficiency

Turbine efficiency describes how effectively the turbine converts hydraulic power into shaft power [27], [28]. It can be calculated using:

$$\eta_t = \frac{P_s}{P_h} \times 100$$

In this equation, $\eta_t$represents turbine efficiency. The term $P_s$represents shaft power produced by the turbine, while $P_h$represents hydraulic power supplied by water. This equation means that efficiency is obtained by comparing useful mechanical output with available hydraulic input. For example, if shaft power is 226.19 W and hydraulic power is 294.3 W, turbine efficiency is calculated as:

$$\eta_t = \frac{226.19}{294.3} \times 100 = 76.86\%$$

This value means that 76.86% of the available hydraulic power is converted into mechanical shaft power, while the remaining energy is lost through hydraulic loss, mechanical friction, turbulence, leakage, and exit kinetic energy.

Tip Speed Ratio

Tip speed ratio describes the relationship between the blade tip speed and incoming water velocity [29], [30]. It is useful for evaluating whether the turbine runner rotates at a suitable speed relative to the water flow. Tip speed ratio can be expressed as:

$$\lambda = \frac{u}{V}$$

In this equation, λrepresents tip speed ratio. The term $u$represents blade tip speed, while Vrepresents incoming water velocity. Blade tip speed can be calculated from runner radius and angular velocity using:

$$u = r\omega$$

In this equation, $r$represents runner radius and ωrepresents angular velocity. This means that blade tip speed increases when the runner radius or angular velocity increases. If the runner radius is 0.12 m and angular velocity is 62.83 rad/s, the blade tip speed is calculated as:

$$u = 0.12 \times 62.83 = 7.54 \text{ m/s}$$

If the incoming water velocity is 8.50 m/s, the tip speed ratio is:

$$\lambda = \frac{7.54}{8.50} = 0.887$$

This value means that the blade tip moves at approximately 88.7% of the incoming water velocity. A suitable tip speed ratio helps improve energy transfer between water and runner blades.

Reynolds Number

Reynolds number is used to describe the flow regime in the turbine passage. It can be calculated using:

$$Re = \frac{\rho V D_h}{\mu}$$

In this equation, Rerepresents Reynolds number. The term ρrepresents water density, Vrepresents average water velocity, $D_h$represents hydraulic diameter, and μrepresents dynamic viscosity. This equation means that Reynolds number compares inertial forces with viscous forces in the flow. If Reynolds number is high, inertial forces dominate and the flow tends to become turbulent. In a turbine, turbulent flow can increase mixing and momentum exchange, but excessive turbulence may also increase energy loss. Therefore, Reynolds number helps explain the hydraulic behavior inside the runner and nozzle.

Research Gap and Contribution

Previous studies have investigated cross-flow turbines using experimental testing and CFD simulation. Recent research has examined blade configuration, blade number, internal flow behavior, nozzle angle, and reaction-type cross-flow turbine concepts. These studies show that cross-flow turbine performance can be improved through runner and nozzle optimization. However, many studies focus on advanced geometry optimization, while simpler experimental performance frameworks are still needed for laboratory-scale development and rural technology implementation. A pico-hydropower system must be evaluated using measurable hydraulic and mechanical indicators, such as flow rate, head, torque, runner speed, shaft power, and efficiency. Without these indicators, turbine performance cannot be interpreted scientifically.

The research gap addressed in this study is the limited integration of basic hydraulic measurement, mechanical output calculation, and efficiency interpretation in a single framework for low-head cross-flow turbines. This gap is important because many small-scale turbine projects only report electrical output without explaining hydraulic input and conversion losses. The main contribution of this study is the development of an integrated hydraulic performance assessment model for a low-head cross-flow turbine. This model connects effective head, flow rate, hydraulic power, angular velocity, torque, shaft power, tip speed ratio, Reynolds number, and turbine efficiency. The proposed framework can support experimental validation, teaching laboratories, rural turbine prototyping, and early-stage turbine design before CFD optimization is performed.

2. METHOD

Research Design

This study uses a quantitative experimental approach to evaluate the hydraulic performance of a low-head cross-flow turbine. The experiment is designed to measure how changes in water flow rate and runner speed affect torque, shaft power, and turbine efficiency. The turbine is tested under controlled low-head conditions to represent pico-hydropower applications. The main independent variables are water flow rate and effective head. The dependent variables are runner speed, torque, shaft power, tip speed ratio, and turbine efficiency. The controlled variables include runner diameter, runner width, blade geometry, nozzle opening, channel configuration, and measurement duration.

Experimental System

The experimental system consists of a water reservoir, pump or gravity-fed channel, flow control valve, nozzle, cross-flow turbine runner, shaft, torque sensor or brake dynamometer, tachometer, and flow meter. The water reservoir supplies water to the turbine system. The nozzle directs water toward the runner blades. The runner converts water energy into rotational shaft motion. The torque sensor or dynamometer measures the mechanical load, while the tachometer measures runner speed. The flow meter measures the water flow rate entering the turbine. The effective head is measured from the water level difference between the inlet and outlet, corrected by estimated head losses. The collected data are used to calculate hydraulic power, shaft power, and turbine efficiency.

Turbine Testing Procedure

The testing procedure begins by checking the turbine runner, shaft alignment, nozzle condition, and measuring instruments. The water flow is then supplied to the turbine at a selected flow rate. The runner speed is allowed to stabilize before data collection begins. After stable operation is achieved, flow rate, effective head, torque, and runner speed are recorded. The test is repeated under several flow rate conditions. For each condition, the turbine is loaded using a brake dynamometer or electrical generator load. This loading process is important because turbine efficiency depends on the balance between flow input and mechanical output. If the load is too low, the runner may rotate too fast but produce low torque. If the load is too high, the runner may slow down and lose hydraulic efficiency. After all measurements are completed, the data are processed to calculate hydraulic power, angular velocity, shaft power, tip speed ratio, Reynolds number, and turbine efficiency.

3. RESULTS AND DISCUSSIONS

The illustrative results show that the low-head cross-flow turbine generated measurable shaft power under the tested operating condition. The hydraulic input was determined from flow rate and effective head, while the mechanical output was calculated from measured torque and runner speed. Assume that the turbine was tested at an effective head of 2.5 m and a flow rate of 0.012 m³/s. The hydraulic power was calculated using:

$$P_h = \rho g Q H_e$$

By substituting water density of 1000 kg/m³, gravitational acceleration of 9.81 m/s², flow rate of 0.012 m³/s, and effective head of 2.5 m, the hydraulic power became:

$$P_h = 1000 \times 9.81 \times 0.012 \times 2.5 = 294.3 \text{ W}$$

This result means that the water supplied 294.3 W of theoretical hydraulic power to the turbine. The runner speed was measured at 600 rpm. The angular velocity was calculated using:

$$\omega = \frac{2\pi N}{60}$$

After substituting $N = 600$rpm, the angular velocity became:

$$\omega = \frac{2\pi \times 600}{60} = 62.83 \text{ rad/s}$$

This value indicates that the turbine shaft rotated at 62.83 radians per second.

If the measured torque was 3.6 N·m, the shaft power was calculated as:

$$P_s = T \times \omega$$

$$P_s = 3.6 \times 62.83 = 226.19 \text{ W}$$

This result means that the turbine converted the water energy into 226.19 W of mechanical shaft power.

The turbine efficiency was then calculated by comparing shaft power with hydraulic power:

$$\eta_t = \frac{226.19}{294.3} \times 100 = 76.86\%$$

This efficiency value indicates that 76.86% of the available hydraulic power was converted into mechanical power at the shaft. The remaining 23.14% was lost through hydraulic losses, blade impact losses, mechanical friction, leakage, turbulence, and exit kinetic energy.

The tip speed ratio was calculated to evaluate the relationship between runner blade speed and incoming water velocity. If the runner radius was 0.12 m and the angular velocity was 62.83 rad/s, the blade tip speed was:

$$u = 0.12 \times 62.83 = 7.54 \text{ m/s}$$

If the incoming water velocity was 8.50 m/s, the tip speed ratio was calculated as:

$$\lambda = \frac{7.54}{8.50} = 0.887$$

This result shows that the runner blade tip moved at 88.7% of the incoming water velocity. This condition indicates that the runner speed was close to the water jet speed, which supports effective momentum transfer.

The Reynolds number was calculated to interpret the flow regime. If the water velocity was 8.50 m/s, hydraulic diameter was 0.04 m, water density was 1000 kg/m³, and dynamic viscosity was 0.001 Pa·s, the Reynolds number was:

$$Re = \frac{1000 \times 8.50 \times 0.04}{0.001} = 340000$$

This value indicates that the flow was turbulent because the Reynolds number was much higher than the common transition range. Turbulent flow can support strong momentum transfer, but excessive turbulence may also increase hydraulic losses inside the runner.

To evaluate the influence of flow rate, the hydraulic power was also calculated at a lower flow rate of 0.009 m³/s with the same effective head of 2.5 m. The hydraulic power became:

$$P_h = 1000 \times 9.81 \times 0.009 \times 2.5 = 220.73 \text{ W}$$

If the corresponding shaft power was 158 W, the turbine efficiency became:

$$\eta_t = \frac{158}{220.73} \times 100 = 71.58\%$$

This lower efficiency suggests that the turbine did not operate at its most effective condition when the flow rate was reduced. At lower flow rates, the water jet may not fully interact with the blade passage, causing lower torque and reduced energy transfer. At a higher flow rate of 0.015 m³/s and the same effective head, the hydraulic power was:

$$P_h = 1000 \times 9.81 \times 0.015 \times 2.5 = 367.88 \text{ W}$$

If the measured shaft power was 270 W, the turbine efficiency became:

$$\eta_t = \frac{270}{367.88} \times 100 = 73.39\%$$

Although the shaft power increased, the efficiency was lower than the 0.012 m³/s condition. This result indicates that maximum shaft power and maximum efficiency do not always occur at the same operating point. At higher flow rates, additional water may produce stronger turbulence, splash loss, or incomplete energy transfer if the nozzle and runner geometry are not optimized. Overall, the illustrative results show that the best efficiency among the tested conditions occurred at a flow rate of 0.012 m³/s, effective head of 2.5 m, runner speed of 600 rpm, and torque of 3.6 N·m. This condition produced 226.19 W of shaft power and 76.86% turbine efficiency. The results confirm that cross-flow turbine performance depends not only on the amount of water supplied but also on the compatibility between flow velocity, runner speed, torque generation, and blade energy transfer.

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The results show that flow rate and effective head strongly influence the hydraulic power available to the turbine. Since hydraulic power is proportional to flow rate and head, increasing either variable increases the theoretical water power. However, the increase in hydraulic input does not always produce proportional increases in turbine efficiency. This occurs because efficiency depends on how effectively the runner converts water energy into shaft power. The turbine achieved the highest illustrative efficiency at the medium flow rate condition. This suggests that the runner and nozzle geometry matched the water jet most effectively under this condition. At lower flow rate, the water jet may have been insufficient to fill the effective blade passage and produce strong torque. At higher flow rate, additional water may have created hydraulic losses, internal turbulence, or non-optimal impact on the blade surface.

The tip speed ratio provides an important explanation for turbine performance. When the runner rotates too slowly, the blade cannot receive the water jet efficiently, causing impact loss and reduced energy transfer. When the runner rotates too fast, the blade may move away from the water jet too quickly, reducing torque generation. Therefore, an appropriate relationship between blade speed and water velocity is required. The Reynolds number result indicates turbulent flow inside the turbine. Turbulent flow is common in practical hydropower turbines because water velocity and flow passage dimensions are relatively high compared with water viscosity. Although turbulence can improve momentum exchange, excessive turbulence can reduce efficiency because energy is dissipated through eddies, flow separation, and internal mixing. From a mechanical engineering perspective, turbine efficiency must be interpreted as the result of hydraulic and mechanical interactions. Hydraulic losses occur because of nozzle friction, flow separation, blade impact loss, and residual kinetic energy at the outlet. Mechanical losses occur because of bearing friction, shaft misalignment, seal friction, and generator coupling loss. Therefore, improving turbine performance requires both hydraulic design optimization and mechanical system improvement. The proposed framework is useful because it explains turbine performance using measurable quantities. Flow rate and head describe hydraulic input. Torque and runner speed describe mechanical output. Efficiency describes conversion effectiveness. Tip speed ratio explains the compatibility between runner motion and water jet speed. Reynolds number explains the flow regime. Together, these indicators provide a complete assessment of low-head cross-flow turbine performance.

4. CONCLUSIONS

This study developed an integrated hydraulic performance assessment framework for evaluating low-head cross-flow turbines in pico-hydropower applications by linking effective head, flow rate, hydraulic power, runner speed, angular velocity, torque, shaft power, tip-speed ratio, Reynolds number, and overall turbine efficiency. Illustrative results showed that the turbine generated 226.19 W of shaft power at a 2.5 m head and 0.012 m^3/s flow rate, achieving an efficiency of 76.86%, which indicates effective conversion of hydraulic input into mechanical output; however, the analysis also revealed that maximum efficiency does not always occur at the highest flow rate due to increased hydraulic losses when operating conditions deviate from optimal runner–flow interaction. The framework's main contribution lies in providing a simple yet structured mechanical-engineering approach for early-stage turbine evaluation, supporting laboratory testing, prototype development, and preliminary optimization before advanced CFD-based design is undertaken. Nonetheless, the study is limited by its use of illustrative numerical values, absence of CFD simulation, lack of uncertainty analysis, and exclusion of generator performance, cavitation behavior, and structural stress evaluation. Because real water channels experience fluctuating flow and head, future work should validate the framework using laboratory-scale experiments with variations in nozzle geometry, blade number, runner dimensions, flow rate, and head, supported by repeated measurements for reliability. Further research should incorporate CFD to analyze internal flow behavior and hydraulic losses, and integrate turbine–generator testing to assess mechanical, electrical, and overall system efficiency for practical pico-hydropower deployment.

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