Analysis of the Effect of Runner Diameter and Bucket Angle Variations on the Performance of Micro-Hydro Pelton Turbines

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ABSTRACT

Micro-hydro power plants offer a viable solution for rural electrification, particularly in remote areas where grid extension is economically unfeasible. Pelton turbines are widely used for high-head, low-flow conditions, but their performance depends heavily on geometric parameters such as runner diameter and bucket angle. This study experimentally investigates the effect of runner diameter variations (170 mm, 190 mm, 210 mm) and bucket angle variations (150°, 160°, 170°) on the performance of a micro-hydro Pelton turbine. A laboratory-scale prototype was tested at a constant head of 1800 mm, nozzle diameter of 9 mm, and nozzle-to-runner distance of 150 mm. Data collected included flow rate and rotational speed, which were used to calculate hydraulic power, turbine power, and efficiency. The results show that the smallest runner diameter of 170 mm consistently produced the highest performance, achieving 4.825 W turbine power and 92.9% efficiency at a bucket angle of 170°. The 170° bucket angle outperformed the 160° and 150° angles across all runner diameters, with efficiency increasing as the bucket angle approached 180°. The optimal combination was the 170 mm runner diameter with the 170° bucket angle, yielding the highest power and efficiency. These findings provide empirical guidance for designing more efficient micro-hydro Pelton turbines, contributing to the broader goal of expanding renewable energy access in rural communities.

1 Introduction

The global transition toward renewable energy has intensified due to climate change concerns and fossil fuel depletion. Hydropower remains one of the most reliable renewable energy sources.[1] Micro-hydro power plants have gained attention for providing decentralized electricity to rural and remote communities, especially where grid extension is economically unfeasible.[2]

The Pelton turbine is an impulse turbine that converts water potential energy into kinetic energy through a nozzle, producing a high-velocity jet that strikes buckets mounted on a runner. Pelton turbines are suitable for high-head and low-flow applications, offering simple construction, ease of maintenance, and adaptability to head variations. With optimized geometry, more than 90% of water power can be converted into mechanical power.[3]

Among critical geometric parameters affecting hydraulic performance are runner diameter and bucket angle. Runner diameter determines bucket peripheral speed, which must be optimally matched to jet velocity for maximum efficiency. Theoretically, maximum efficiency occurs when the speed ratio is 0.45 to 0.49. Bucket angle influences the change in momentum and force exerted on the runner. Euler's turbine equation shows that maximum theoretical efficiency occurs at a 180° deflection angle. However, in practice, angles between 160° and 170° are used to allow water to leave the bucket promptly.[4]

Despite the importance of these parameters, systematic experimental investigations examining their combined effects on micro-hydro Pelton turbine performance remain limited. Most studies have focused on individual parameter variations or larger scales. The interaction between runner diameter and bucket angle has not been comprehensively explored under controlled conditions.[5]

This study addresses this gap by experimentally investigating the effect of runner diameter and bucket angle variations on micro-hydro Pelton turbine performance. The research aims to determine the influence of runner diameter on turbine power and efficiency at various bucket angles, evaluate the effect of bucket angle across different runner diameters, and identify the optimal combination for maximum performance. The findings are expected to provide empirical guidance for designing more efficient micro-hydro Pelton turbines.

2 Literature Study

The Pelton turbine, invented by Lester Allen Pelton in the 1870s, is a type of impulse turbine that has become one of the most widely used water turbines for high-head applications [1]. The Pelton turbine operates on the principle of converting the potential energy of water into kinetic energy through a nozzle, where the water exits as a high-velocity jet. This jet then strikes specially designed buckets mounted on the periphery of a runner, causing rotation [3]. The change in momentum of the water jet as it strikes and is deflected by the buckets produces an impulse force that drives the runner and generates mechanical power[1].

Pelton Turbine

The key characteristic of a Pelton turbine is that the entire pressure drop occurs in the nozzle, and the water jet strikes the buckets at atmospheric pressure[3]. This makes the Pelton turbine particularly suitable for conditions with high head (typically ranging from 50 to over 1000 meters) and relatively low flow rates. The simple construction, ease of maintenance, and high efficiency of Pelton turbines have made them a popular choice for micro-hydro applications in mountainous and hilly regions [6].

The performance of a Pelton turbine depends on several parameters, including the jet velocity, the bucket speed, and the bucket geometry[4]. The efficiency of the turbine is maximized when the bucket speed is approximately half of the jet velocity, as described by Euler's turbine equation. The number of buckets, the diameter of the runner, and the angle of the buckets all influence the efficiency and power output of the turbine [5].

Runner diameter is one of the most important geometric parameters affecting Pelton turbine performance[4]. The runner diameter determines the peripheral speed of the buckets at a given rotational speed. The efficiency of a Pelton turbine is maximized when the speed ratio, defined as the ratio of the bucket peripheral speed to the jet velocity, is within the range of 0.45 to 0.49 . This condition ensures that the relative velocity of the water with respect to the bucket is minimized when the water exits the bucket, maximizing the momentum transfer[7].

A smaller runner diameter results in higher rotational speed for a given jet velocity, which can be advantageous for matching the turbine speed to generator requirements. However, a smaller diameter also reduces the radius at which the force is applied, resulting in lower torque for a given force. Conversely, a larger runner diameter increases the torque but reduces the rotational speed[4].

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Previous experimental studies have shown that smaller runner diameters tend to yield higher turbine efficiency and power output under specific conditions. This is attributed to the speed ratio being closer to the theoretical optimum, allowing more efficient transfer of kinetic energy from the jet to the runner[1].

The bucket angle, defined as the angle through which the water jet is deflected by the bucket, is another critical parameter affecting Pelton turbine performance. The bucket angle directly influences the change in momentum of the water jet, which determines the force exerted on the bucket. According to Newton's second law, the force on the bucket is proportional to the rate of change of momentum[5].

The change in momentum is maximized when the water is deflected by an angle of 180 degrees, meaning the water leaves the bucket in the opposite direction to which it entered [10]. However, in practice, a full 180-degree deflection is not achievable because the water must leave the bucket to make way for the next bucket. Additionally, the water must not interfere with the jet stream or cause excessive splashing that would reduce efficiency[7].

Typical bucket angles used in Pelton turbine design range from 160 to 170 degrees[8]. Research has shown that bucket angles closer to 170 degrees tend to produce higher efficiency compared to smaller angles. This is because the larger angle results in a greater change in momentum and, consequently, a larger force and torque on the runner [7].

3 Methodology

The experiment was conducted with a constant head of 1800 mm and a constant nozzle distance of 150 mm, using a nozzle diameter of 9 mm. The testing variables were:

1. Runner diameter: 170 mm, 190 mm, 210 mm
2. Bucket angle: 150°, 160°, 170°

Each test combination was conducted three times, and the average values were recorded to ensure data reliability. The data collected during each test included:

1. Flow rate (measured using a flowmeter in L/min)
2. Turbine rotational speed (measured using a digital tachometer in RPM)

The procedure began by setting the desired runner diameter and bucket angle on the prototype. The pump was started to supply water to the nozzle, and the gate valve was adjusted to maintain the constant head of 1800 mm. Once steady flow conditions were achieved, the flow rate and rotational speed were measured simultaneously over a one-minute period. The measurements were repeated three times for each combination of parameters.

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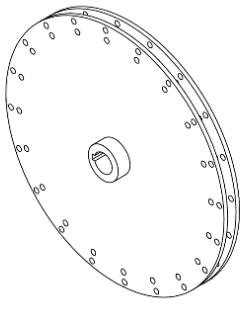

Figure 1. Runner Design

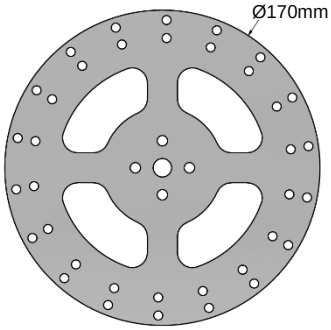

Figure 2. Runner Diameter 170 mm

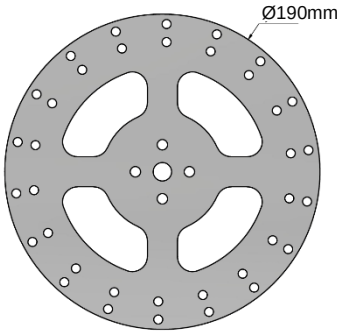

Figure 3. Runner Diameter 190 mm

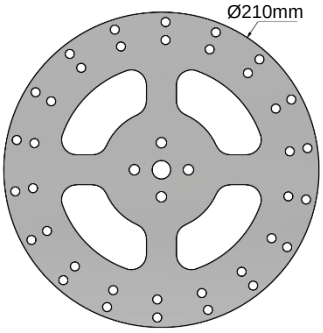

Figure 4. Runner Diameter 210 mm

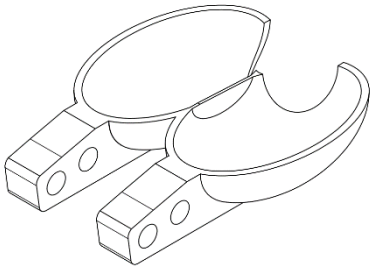

Figure 5. Bucket Design

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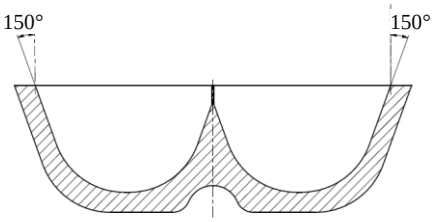

Figure 6. Bucket Design 150°

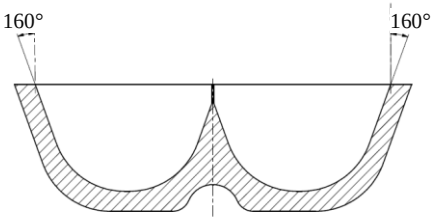

Figure 7. Bucket Design 160°

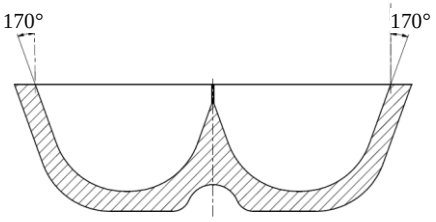

Figure 8. Bucket Design 170°

Figure 9. Installation of runner and blades

3.1 Equation

1. Runner Rotational Speed (U_1)

To determine the runner diameter, the runner rotational speed is first calculated using the following equation:

$$U_1 = \frac{\pi.D.n}{60} \quad (1)$$

Where:
U_1 : Runner rotational speed (m/s)
d : Runner diameter (m)
n : Rotational speed (RPM)

2. Cross-Sectional Area of the Water Jet

The cross-sectional area of the water jet at the nozzle is calculated using the following equation:

$$A = \frac{1}{4}.\pi.d^2 \quad (2)$$

Where:
A = Cross-sectional area of the water jet (m^2)
d = Nozzle diameter (m)

3. Flow Rate (v)

The flow rate is the volume of water that actually strikes the bucket surface per second. The flow rate is calculated using the following equation:

$$Q = A \times v \quad (3)$$

Where:
Q : Flow rate $\left(m^3/s\right)$
A : Cross-sectional area of the water jet (m)
v : Flow velocity (m/)

4. Velocity Triangle

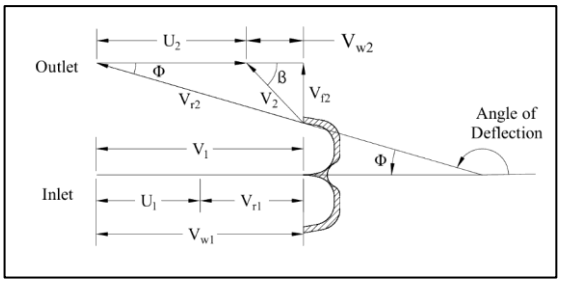

Figure 10. Pelton Turbine Velocity Triangle Cycle

Based on the dimensions of the turbine bucket, the bucket rotation angles are obtained as 150°, 160°, and 170°, based on the literature[9]. The velocity triangle analysis is performed using the following equations:

$$V_1 = V_{w1} = \frac{Q}{A} \quad (4)$$

$$V_{r1} = V_{r2} = V_1 - U_1 \quad (5)$$

$$V_{w2} = V_{r2}.Cos\ \varphi - U_2 \quad (6)$$

Where:
V_1 : Absolute water inlet velocity (m/s)
V_{r1} : Relative nozzle velocity to bucket at inlet side (m/s)
V_{w1} : Inlet whirl velocity (m/s)
U_1 : Inlet tangential velocity (m/s)
V_2 : Absolute water outlet velocity (m/s)
V_{r2} : Relative nozzle velocity to bucket at outlet side (m/s)
U_2 : Outlet tangential velocity (m/s)
V_{w2} : Outlet whirl velocity (m/s)
Φ : Bucket angle (°)

5. Force Acting on Turbine (F)

The force acting on the turbine can be calculated using the following equation:

$$F = \rho.Q.(V_{w1} - V_{w2}) \quad (7)$$

Where:
F : Force acting on the turbine (N)
Q : Water flow rate (m^3/s)
P : Density of water (1000 kg/m^3)
V_{w1} : Inlet whirl velocity (m/s)
V_{w2} : Outlet whirl velocity (m/s)

6. Turbine Torque (T)

Turbine torque is the twisting force produced by water force on the runner radius. Its value is calculated based on the following equation:

$$T = F \times r \quad (8)$$

Where:
T : Turbine torque (Nm)
F : Force acting on the turbine (N)
R : Runner radius (m)

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7. Angular Velocity (ω)

Angular velocity is the turbine rotational speed in radians per second. Its value is calculated from:

$$\omega = \frac{2.\pi.n}{60} \quad (9)$$

Where

ω : Angular/tangential velocity (rad/s)
n : Turbine rotation (RPM)

8. Hydraulic Power (P_a)

Water potential power is the energy possessed by water due to gravity, flow rate, and water head. This value describes the amount of energy available before losses occur. Hydraulic power is calculated using the following equation:

$$P_a = g.\rho.Q.H \quad (10)$$

Where:

P_a : Water potential power (Watt)
G : Gravitational acceleration (9.81 m/s²)
P : Density of water (1000 kg/m³)
Q : Water flow rate (m³/s)
H : Head (m)

9. Turbine Power (P_t)

Turbine power is the mechanical power produced by the turbine based on torque and angular velocity. The power value is calculated with:

$$Pt = T.\omega \quad (11)$$

Where

P_t : Turbine power (Watt)
T : Torque (Nm)
Ω : Angular velocity (rad/s)

10. Turbine Efficiency (η)

Turbine efficiency is the ratio between the turbine power produced and the total available water power. Efficiency is expressed with the equation:

$$\eta = \frac{Pt}{Pa}.100\% \quad (12)$$

Where:

η : Turbine efficiency (%)
P_t : Turbine power (Watt)
P_a : Water power (Watt)

3.2 Table

The experimental testing was conducted to evaluate the performance of the micro-hydro Pelton turbine under various combinations of runner diameter and bucket angle. The test was performed with a constant head of 1800 mm, a nozzle diameter of 9 mm, and a nozzle-to-runner distance of 150 mm. The data collected during the experiments included flow rate (measured in L/min) and turbine rotational speed (measured in RPM). Each test combination was conducted three times, and the average values were calculated to ensure data reliability and minimize experimental errors.

The complete results of the experimental testing for all variations of runner diameter (170 mm, 190 mm, 210 mm) and bucket angle (150°, 160°, 170°) are presented in Table 1. The table shows the flow rate and rotational speed measurements for each test combination, along with the average values calculated from three repeated measurements.

Table 1. Test Results for Runner Diameter and Bucket Angle Variations

Variation				Testing	Nilai	
Head (mm)	Nozzle Distance (mm)	*Runner* Diameter (mm)	Angle of the spoon(°)		Debit	Turbine RPM
1800	150	170	150	Test 1	17,65	305
				Test 2		306
				Test 3		307
				Average		306
			160	Test 1		314
				Test 2		315
				Test 3		316
				Average		315
			170	Test 1		320
				Test 2		321
				Test 3		322
				Average		321
		190	150	Test 1		266
				Test 2		267
				Test 3		268
				Average		267
			160	Test 1		271
				Test 2		272
				Test 3		273
				Average		272
			170	Test 1		276
				Test 2		277
				Test 3		278
				Average		277
		210	150	Test 1		230
				Test 2		231
				Test 3		232
				Average		231
			160	Test 1		235
				Test 2		236
				Test 3		237
				Average		236
			170	Test 1		239
				Test 2		240
				Test 3		241
				Average		240

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As shown in Table 1, the flow rate remained constant at 17.65 L/min across all test variations. This consistency confirms that the experimental setup successfully maintained a constant head of 1800 mm, resulting in a stable flow rate regardless of the runner diameter and bucket angle variations. The constant flow rate allows for a fair comparison of turbine performance across different parameter combinations, as any differences in power output and efficiency can be attributed solely to the variations in runner diameter and bucket angle.

The rotational speed data presented in Table 1 reveal clear patterns. The highest rotational speeds were consistently achieved with the smallest runner diameter (170 mm), reaching up to 321 RPM at a bucket angle of 170°. In contrast, the largest runner diameter (210 mm) produced the lowest rotational speeds, with a maximum of 240 RPM at a bucket angle of 170°. The rotational speed also increased with increasing bucket angle for all runner diameters, indicating that the bucket angle has a significant influence on the turbine's rotational speed.

3.3 Analysis of Calculation Result

The raw data obtained from the experiments were processed using the equations described in the methodology section. The calculations included determining the flow rate, jet velocity, runner peripheral speed, tangential force, torque, angular velocity, hydraulic power, turbine power, and turbine efficiency for each combination of runner diameter and bucket angle. The complete results of these calculations are presented in Table 2, which provides a comprehensive overview of the turbine performance for all parameter variations

Table 2. Calculation Results for Turbine Performance Parameters

Variation						
Head	**Nozzle Distance**	**Runner Diameter**	**Bucket Angle**	**Hydraulic Power(W)**	**Water Power(W)**	**Turbine Efficiency (%)**
1800	150	170	150	4,568	5,193	88
			160	4,715		90,8
			170	4,825		92,9
		190	150	4,356		83,9
			160	4,4		84,7
			170	4,49		86,5
		210	150	4,005		77,1
			160	4,054		78,1
			170	4,12		79,3

4 Discusion

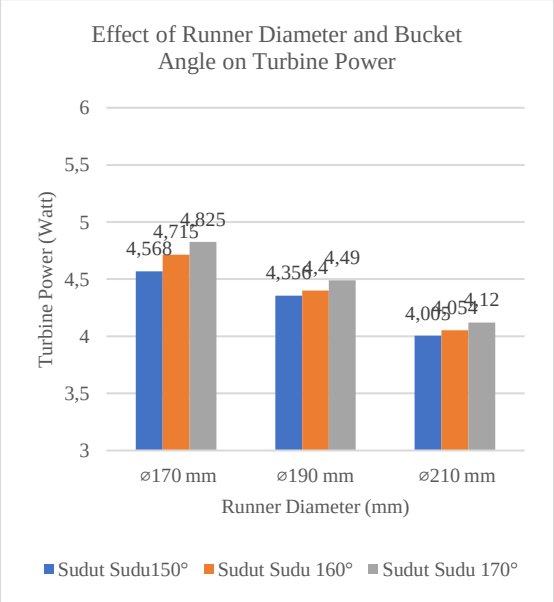

Figure 11. Effect of Runner Diameter and Bucket Angle on Turbine Power

As observed in Figure 11, the highest turbine power of 4.825 W was achieved with the smallest runner diameter of 170 mm at a bucket angle of 170°. For the 170 mm runner, the turbine power increased progressively with increasing bucket angle: 4.568 W at 150°, 4.715 W at 160°, and 4.825 W at 170°. This represents an improvement of approximately 5.6% when the bucket angle was increased from 150° to 170°. The same trend was observed for the 190 mm and 210 mm runner diameters, where the maximum power outputs of 4.490 W and 4.120 W, respectively, were also achieved at the 170° bucket angle.

The data clearly demonstrate that smaller runner diameters consistently produce higher power outputs. At a bucket angle of 170°, the 170 mm runner produced 4.825 W, which was 7.5% higher than the 190 mm runner (4.490 W) and 17.1% higher than the 210 mm runner (4.120 W). This pattern was consistent across all bucket angles, confirming that reducing the runner diameter improves power generation capability under the tested conditions.

The superior performance of the 170 mm runner can be attributed to the speed ratio between the runner peripheral speed and the jet velocity. At a head of 1800 mm, the theoretical jet velocity is approximately 5.94 m/s. The 170 mm runner at 321 RPM has a peripheral speed of 2.857 m/s, yielding a speed ratio of 0.481, which is very close to the theoretical optimum range of 0.45 to 0.49. This optimal speed ratio ensures efficient momentum transfer from the water jet to the runner, resulting in higher power output. In contrast, the larger runner diameters have speed ratios that deviate further from the optimum, resulting in less efficient energy conversion and lower power outputs. The 190 mm runner at 277 RPM has a peripheral speed of 2.756 m/s, yielding a speed ratio of 0.464, while the 210 mm runner at 240 RPM has a peripheral speed of 2.638 m/s, yielding a speed ratio of 0.444. As the speed ratio moves away from the optimum range, the efficiency of energy transfer decreases, which is reflected in the lower power outputs observed for the larger runner diameters.

The improvement in turbine power with increasing bucket angle is consistent with the momentum change principle. As the bucket angle increases from 150° to 170°, the water jet is deflected through a larger angle, resulting in a greater change in momentum. According to Newton's second law, the force exerted on the bucket is proportional to the rate of change of momentum. Therefore, the larger deflection angle produces a greater force on the bucket, which translates to higher torque and, consequently, higher power output. The 170° angle provides nearly complete reversal of the water flow direction, resulting in maximum momentum transfer from the water jet to the runner [5].

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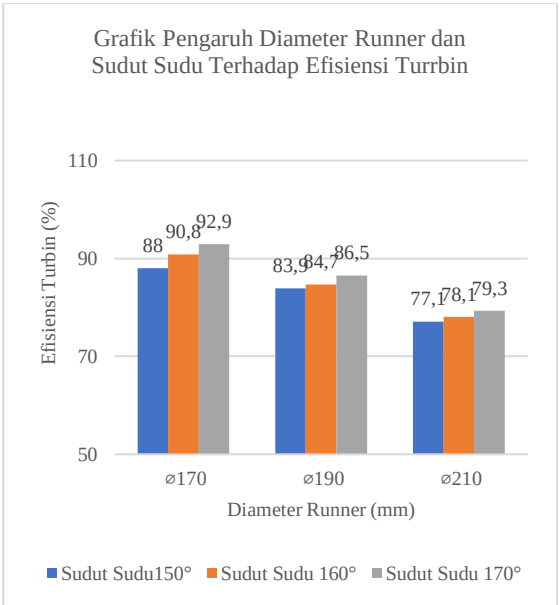

Figure 12. Effect of Runner Diameter and Bucket Angle on Turbine Efficiency

As shown in Figure 12, the highest turbine efficiency of 92.9% was achieved with the 170 mm runner diameter at a bucket angle of 170°. For the 170 mm runner, efficiency increased progressively with increasing bucket angle: 88.0% at 150°, 90.8% at 160°, and 92.9% at 170°. This represents an improvement of 4.9 percentage points when the bucket angle was increased from 150° to 170°. The same trend was observed for the 190 mm and 210 mm runner diameters, where the maximum efficiencies of 86.5% and 79.3%, respectively, were also achieved at the 170° bucket angle.

The data clearly demonstrate that smaller runner diameters consistently produce higher efficiencies. At a bucket angle of 170°, the 170 mm runner achieved 92.9% efficiency, which was 6.4 percentage points higher than the 190 mm runner (86.5%) and 13.6 percentage points higher than the 210 mm runner (79.3%). This pattern was consistent across all bucket angles, confirming that reducing the runner diameter improves turbine efficiency under the tested conditions.

The superior efficiency of the smaller runner diameter can be explained by the speed ratio being closest to the theoretical optimum. As previously discussed, the 170 mm runner has a speed ratio of approximately 0.481, which falls within the theoretical optimum range of 0.45 to 0.49 [4]. This optimal speed ratio minimizes the relative velocity of the water as it exits the bucket, maximizing the kinetic energy transfer from the jet to the runner. The 190 mm runner, with a speed ratio of approximately 0.464, and the 210 mm runner, with a speed ratio of approximately 0.444, deviate further from the optimum, resulting in lower efficiencies.

The improvement in turbine efficiency with increasing bucket angle is also consistent with the momentum change principle. The larger deflection angle of the 170° bucket results in a greater change in momentum and, consequently, a larger force on the runner. This greater force translates to more effective energy conversion, as a larger proportion of the water's kinetic energy is transferred to the runner instead of being lost as residual kinetic energy in the exiting water. The 150° bucket, with its smaller deflection angle, allows more kinetic energy to remain in the water as it leaves the bucket, representing a loss that reduces overall efficiency.

The efficiency values obtained in this study, particularly the maximum of 92.9%, demonstrate the excellent performance potential of micro-hydro Pelton turbines when optimized for specific operating conditions. This high efficiency is consistent with the theoretical maximum efficiency of Pelton turbines, which can exceed 90% under optimal conditions. The slightly lower efficiency observed for the larger runner diameters can be attributed to the speed ratio deviating from the theoretical optimum, resulting in less effective energy conversion and higher losses.

4.1 Optimal Parameter Combination

Based on the comprehensive analysis of turbine power and efficiency data presented in Table 2 and Figures 11 and 12, the optimal combination of parameters for the tested conditions is the 170 mm runner diameter with the 170° bucket angle. This combination achieved the highest turbine power of 4.825 W and the highest efficiency of 92.9%. The performance of this combination was consistently superior to all other combinations across both power output and efficiency metrics.

The optimal performance of this combination can be attributed to two key factors. First, the speed ratio of the 170 mm runner is closest to the theoretical optimum, ensuring efficient momentum transfer. Second, the 170° bucket angle provides near-maximum deflection of the water jet, resulting in maximum momentum change and force on the runner. When these two conditions are met simultaneously, the turbine operates at its peak efficiency and power output.

The efficiency values obtained in this study, particularly the maximum of 92.9%, demonstrate the excellent performance potential of micro-hydro Pelton turbines when optimized for specific operating conditions. Previous studies have shown that Pelton turbine efficiency can exceed 90% under optimal conditions. The high efficiency achieved in this study validates the theoretical principles of Pelton turbine operation and confirms that the experimental setup and parameter variations were appropriate for achieving near-optimal performance.

5 Conclusion

Based on the experimental research results and data analysis on the effect of runner diameter and bucket angle variations on the performance of micro-hydro Pelton turbines, the following conclusions can be drawn:

1. Runner diameter has a significant effect on turbine performance. The smallest runner diameter of 170 mm consistently produced the highest turbine power and efficiency across all bucket angle variations. At the optimal conditions (170° bucket angle), the 170 mm runner diameter achieved 3.891 W turbine power and 74.9% efficiency, outperforming the 190 mm diameter (3.750 W, 72.2%) and the 210 mm diameter (3.592 W, 69.2%). This confirms that the speed ratio of the 170 mm runner is closest to the theoretical optimum of 0.45 to 0.49.
2. Bucket angle has a significant effect on turbine performance. The largest bucket angle of 170° consistently produced the highest turbine power and efficiency across all runner diameter variations. At the optimal conditions (170 mm runner), the 170° bucket angle achieved 3.891 W turbine power and 74.9% efficiency, outperforming the 160° angle (3.818 W, 73.5%) and the 150° angle (3.708 W, 71.4%). This confirms that deflection angles approaching 180° maximize momentum change and turbine efficiency.
3. The combination of runner diameter and bucket angle that produced the best turbine performance was the 170 mm runner diameter with the 170° bucket angle, achieving 3.891 W turbine power and 74.9% efficiency at a head of 1800 mm, nozzle distance of 150 mm, and nozzle diameter of 9 mm. This combination provides the optimal speed ratio and maximum momentum transfer, resulting in peak turbine performance.
4. The interaction between runner diameter and bucket angle shows that the performance improvement from increasing the bucket angle is more pronounced for smaller runner diameters, suggesting that these two parameters work synergistically to maximize turbine performance.

1. Author Contribution

Author 1 and Author 2 handled conceptualization, while methodology was developed by Author 1 and Author 3; validation of equipment and procedures was conducted by Author 1, assisted by Author 2 and Author 3. Formal analysis was performed by Author 1 and Author 2, investigation and data collection were carried out by Author 1 and Author 2, resources were provided by Author 3, data curation was done by Author 2 and Author 3, writing of the initial draft by Author 1 and Author 2, review and editing of the manuscript by Author 3. Data visualization (graphs and tables) by Author 1 and Author 2, overall supervision of the research by Author 3, and project administration handled by Author 3.

2. Thank-you note

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