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PELTON TURBINE WITH TWO – NOZZLE: DESIGN AND PERFORMANCE ANALYSIS

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ABSTRACT: This study addresses the critical gap in understanding the trade-off between power enhancement and efficiency reduction in multi-nozzle Pelton turbine systems. While previous studies have primarily reported performance improvements with increasing nozzle numbers, limited attention has been paid to the associated energy losses and flow-interaction mechanisms. Therefore, this research aims to provide a comprehensive experimental evaluation of single- and double-nozzle configurations, focusing on the interplay between torque, rotational speed, power output, and efficiency. Experiments were conducted under controlled conditions at approximately 15 psi and 43 L/min, with mechanical loads ranging from 0.2 kg to 1 kg. The results reveal that the double-nozzle configuration significantly increases rotational speed and power output, reaching up to 18 W. However, the highest efficiency of 18% is achieved with a single nozzle, whereas the double-nozzle system achieves 12%. The findings demonstrate that increasing hydraulic input does not necessarily lead to improved efficiency due to complex fluid-dynamic phenomena, including jet interference, turbulence, and non-uniform momentum transfer. This study highlights that Pelton turbine performance is governed by a nonlinear interaction between hydraulic and mechanical parameters rather than a simple scaling effect. The novelty of this work lies in providing a critical and experimentally validated perspective on multi-nozzle performance, offering practical insights for optimizing turbine design in micro-hydropower applications.

Keywords: Turbine, Pelton, Power, Efficiency, Nozzle

1. Introduction

The Pelton turbine, named after its inventor Lester Allan Pelton, is a hydroelectric power generation technology widely used to convert the kinetic energy of high-speed water flow into mechanical power. In recent years, there has been increasing interest in optimizing Pelton turbine design to improve performance and efficiency [1][2]. One innovative approach involves using two nozzles, which has the potential to increase power output and overall efficiency [3][4][5].

The design of a two-nozzle Pelton turbine aims to utilize the benefits of two water jets striking the turbine buckets simultaneously. By strategically positioning the nozzle, the water flow can be directed to impact the buckets at optimal angles, thereby maximizing momentum transfer and increasing torque [1][6]. Additionally, a two-nozzle configuration may produce a more uniform and stable flow pattern within the turbine, potentially reducing flow separation and other energy losses [3].

A Pelton turbine consists of a series of blades (buckets) rotated by water jets discharged from one or more nozzle. The nozzle converts the potential energy of water, in the form of pressure head, into kinetic energy (velocity head), producing a high-speed jet. The nozzle design is based on Torricellian flow, which states that the velocity of fluid exiting an orifice is equal to the velocity of a freely falling object from the same height. The nozzle typically has a circular cross-section, and the flow rate is regulated by adjusting the needle valve opening. This configuration allows optimal control of the energy supplied to the turbine blades, enabling maximum performance under varying operating conditions [7]. Two-nozzle systems have emerged as a promising design approach, offering improved performance and operational flexibility.

The buckets are critical components of the Pelton turbine, as they convert the water's kinetic energy into mechanical energy. High-velocity water jets from the nozzle strike the buckets, which are specifically designed to capture and utilize the energy efficiently. Each bucket has a concave shape with a central splitter (dividing lip) that divides the incoming jet into two symmetrical streams. These streams are then deflected in opposite directions, maximizing the change in momentum and generating torque on the turbine shaft.

The number of buckets in a Pelton turbine varies with turbine size and design requirements. Typically, Pelton turbines have between 17 and 24 buckets [8]. This range is selected to ensure optimal efficiency and operational stability. Larger turbines generally require more buckets to handle higher water flow rates and maximize energy utilization.

Previous research found that fewer buckets could yield higher power output, with the optimal configuration reported at 10 buckets, achieving an efficiency of 48.3% and a maximum power of 341.5 W at 335 rpm. However, increasing the number of buckets to 12 resulted in a significantly higher efficiency of 76.9%, with a power output of 543.1 W under a head of 36 meters [9].

Although several studies have investigated Pelton turbine performance, comparisons between different nozzle configurations have not been critically analyzed, particularly in terms of the trade-off between power, torque, and efficiency. Previous studies, such as [7] and [10], reported that increasing the number of nozzles tends to increase rotational speed and output power due to higher water discharge. However, these studies often assume that efficiency improves without thoroughly examining energy losses from jet interactions in multi-nozzle systems.

In contrast, other studies suggest that efficiency depends not only on increasing flow input but also on how effectively jet momentum is transferred to the buckets. For example, demonstrated that geometric parameters, such as the number of buckets, significantly influence efficiency, indicating that non-optimized configurations may lead to performance degradation [9].

Therefore, a research gap remains in understanding the balance between increased power output and potential efficiency reduction in multi-nozzle systems. This study aims to address this gap by critically analyzing the performance differences between single-nozzle and double-nozzle configurations under controlled experimental conditions. The focus is not only on performance enhancement but also on identifying the underlying mechanisms responsible for variations in efficiency.

This study investigates the effect of using two nozzles in combination with 21 buckets on the power, torque, and efficiency of a Pelton turbine.

2. Method

In the methods section of this study, a systematic analytical approach is employed to investigate the characteristics of the Pelton turbine and the effect of nozzle number on its performance. The design parameters of the turbine, including jet diameter, nozzle length, and nozzle needle angle, are carefully defined, and a series of experiments is conducted using different nozzle configurations. In the design of the Pelton turbine buckets, strict dimensional criteria are followed to ensure optimal performance. In addition, this study applies an empirical approach to determine the appropriate number of buckets, with calculations indicating that $Z \geq 17$. This approach enables a detailed analysis of torque, fluid flow velocity, and turbine power. To evaluate turbine efficiency, the turbine head (H_t) is calculated as the difference in elevation between the fluid inlet and the turbine position. The entire experimental process is carried out by varying the number of nozzles and water pressure to examine their effects on Pelton turbine performance. With this methodology, the study provides a comprehensive analysis of Pelton turbine behavior, contributing to a deeper understanding of how to optimize turbine efficiency and power output in renewable energy applications.

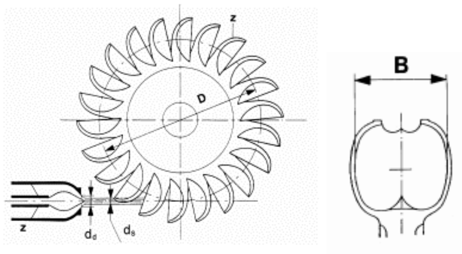

Figure 1 Pelton turbine [11]

From Figure 1, D represents the diameter of the turbine runner, which serves as the mounting location for the buckets. Z denotes the number of buckets and nozzles. The parameters d_d and d_s represent the jet diameter and the water jet diameter, respectively. Meanwhile, B indicates the width of the bucket that receives the water jet from the nozzle, enabling the runner to rotate.

2.1 Pelton Turbine Design Steps

Design procedure of a Pelton turbine for a micro hydropower plant.

1. Preliminary preparation for Turbine location

a) Head Calculation (H_n)

$$H_n = H_g - H_{tl} \quad (1)$$

Where H_g is the gross head, defined as the vertical distance between the water surface at the intake and the turbine, and H_{tl} is the total head loss due to open channels, trash racks, intakes, penstocks, gates, and valves. This loss is typically assumed to be approximately 6% of the gross head.

b) Calculation of water flow rate (Q)

The water flow rate can be determined by measuring the flow velocity of the river or stream (V_r) in m/sand its cross-sectional area (A_r) in m^2:

$$Q = V_r * A_r \quad (m^3.s^{-1}) \quad (2)$$

2. Turbine input power calculation (P_{ti})

$$P_{\mathrm{ti}} = \rho * g * C_n^2 * H_n * Q_t \text{ (Wattt)} \quad (3)$$

Where ρ is the fluid density, g is the gravitational acceleration, C_n is the turbine performance (efficiency) coefficient, H_n is the net head, and Q_t is the discharge flow rate.

3. Turbine Rotational Speed Calculation (N_s)

$$N_s = 85.49 * \sqrt{\frac{n_j}{H_n^{0.243}}} \quad (4)$$

4. Where n_j is the number of turbine nozzles, which can be calculated as:

$$n_j = \frac{Q_t}{Q_n} \quad (5)$$

Where Q_n is the flow capacity of each nozzle (m^3/s). The turbine rotational speed (N) in rpm is then calculated as:

$$N = N_s * \frac{H_n^{5/4}}{\sqrt{P_{\mathrm{ti}}}} \quad (6)$$

From the continuity equation:

$$Q = Z.\frac{\pi.d_s^2}{4}.c_{1u}$$

$$d_s = \sqrt{\frac{4.Q}{Z.\pi.c_{1u}}} \quad (7)$$

Where d_s is the jet diameter, representing the outlet of pressurized water flow impacting the turbine buckets, Z is the number of nozzle (in this study, three nozzle are used), Q is the volumetric flow rate, and c_{1u} is the flow velocity coefficient.

The flow velocity coefficient is defined as presented in Figure 2.

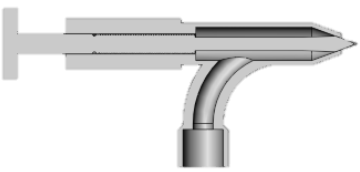

Figure 2 Pelton Turbine Nozzle Design [13]

Table 1 Parameters Nozzle Turbin Pelton

No	Nozzle Pelton Turbine Parameters	
1.	Nozzle length	235 mm
2.	Nozzle needle angle	70º
3.	Number of nozzles	3pieces
4.	*Wheel* diameter	14,07 mm
5.	Spray jet diameter	8,72 mm

For determining the size of the Pelton turbine bucket, as shown in Table 1, the following empirical relationship is used:

$$3{,}1 > \frac{B}{d_s} \geq 3{,}4$$

Where B is the bucket width and d_s is the jet diameter. The design criteria are defined as follows:

- $B = 3.1\ d_s$ for one nozzle
- $B = 3.2\ d_s$ for two nozzles
- $B = 3.2\ d_s$ for four to five nozzles
- $B > 3.3\ d_s$ for six nozzles

2.2 Calculation of Nozzle Dimensions

The flow rate through each nozzle can be calculated as:

$$Q_n = V_j * A_j (m^3.s^{-1}) \quad (8)$$

Where V_j is the water velocity at the nozzle exit, and A_j is the nozzle cross-sectional area. The jet velocity is defined as:

$$V_j = C_n * \sqrt{2 * g * H_n}$$

Where C_n is the velocity coefficient, g is gravitational acceleration, and H_n is the net head.

The nozzle area is calculated as:

$$A_j = \pi \frac{D_j^2}{4} \quad (m^2) \quad (9)$$

From the equations above, the nozzle diameter can be determined as:

$$D_j = \sqrt{\frac{4 * Q_t}{(\pi * n_j * V_j)}} \quad (m) \quad (10)$$

Where D_j is the nozzle diameter, Q_t is the total flow rate, and n_j is the number of nozzle.

The nozzle length can be calculated using the following equation:

$$L_n = \frac{(D_{pn} - D_j)}{tan\,(\beta)} \quad (m) \quad (11)$$

Where:

$$D_{pn} = \frac{D_{pt}}{\sqrt{n_j}} \quad (m) \quad (12)$$

The nozzle exit should be positioned as close as possible to the Pelton runner to prevent the jet from deviating from its intended diameter. The recommended distance between the nozzle and the runner is 5% of the runner diameter, with an additional clearance of 3 mm to accommodate deflector operation:

$$X_{nr} = 0.05 * D_r + D_t \ (m) \quad (13)$$

The minimum distance between the bucket and the nozzle can be calculated as:

$$X_{nb} = 0.625 * D_r \ (m) \quad (14)$$

2.3 Calculation of Nozzle Dimensions

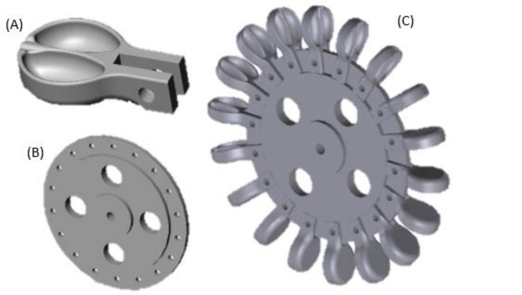

Figure 3 (A) *Bucket* Design, (B) *Runner* Design, (C) Pelton Turbine Design

Empirically, the number of buckets, as shown ini Figure 3, in a Pelton turbine should satisfy the following condition:

$$Z \geq 17$$

where Z denotes the total number of buckets, as shown in Table 2.

Table 2 Pelton Turbine Parameters

No	Pelton Turbine	Parameters
1.	*Runner* radius	96,28 mm
2.	*Bucket* width	28,28 mm
3.	*Bucket* depth	12,08 mm
4.	*Bucket* height	65,28 mm
5.	Number of *buckets*	18 pieces
6.	*Bucket* gap width	13,11 mm
7.	*Bucket* thickness	15 mm

The bucket axial width can be calculated as:

$$B_w = 3.4 * D_j \quad (m) \quad (15)$$

Where D_j is the nozzle diameter. This equation indicates that the axial width of the bucket (B_w) is approximately 3.4 times the nozzle diameter.

The radial length of the bucket can be calculated as:

$$B_l = 3 * D_j \quad (m) \quad (16)$$

The bucket depth is given by:

$$B_d = 1.2 * D_j \quad (m) \quad (17)$$

The number of buckets in each runner should be determined to ensure that no water particles are lost while minimizing adverse interactions between outgoing water jets and adjacent buckets. This can be calculated as:

$$n_b = 15 + \frac{D_r}{(2*D_j)} \quad (18)$$

The bucket moment arm length is defined as:

$$L_{ab} = 0.195 * D_r \quad (m) \quad (19)$$

The radius from the runner center to the bucket center of mass is:

$$R_{br} = 0.47 * D_r \quad (m) \quad (20)$$

Meanwhile, the bucket volume can be calculated as:

$$V_b = 0.0063 * D_r^3 \quad (m^3) \quad (21)$$

The mass of the bucket is then determined by:

$$M_b = \rho_m * V_b \quad (Kg) \quad (22)$$

Pelton turbine buckets are critical components in hydroelectric power generation systems. The primary wear occurs on the inner surfaces, which are continuously exposed to high-velocity water flow and pressure. Therefore, the materials used for bucket construction must be resistant to corrosion, erosion, and wear caused by water friction and suspended particles. Over time, the repetitive impact of flowing water leads to both physical and chemical degradation of the bucket surface. If not properly managed, this degradation can reduce turbine efficiency and shorten the system's overall service life.

The splitter within the Pelton turbine bucket is also susceptible to wear, as it plays a crucial role in directing the water flow efficiently onto the turbine wheel. Increasing the splitter width by 1% of the inner bucket width can reduce turbine efficiency by approximately 1% under full-load conditions [14].

Friction and mechanical stresses during operation contribute to the gradual wear of the splitter surface. Therefore, regular monitoring and inspection of both the splitter and the inner bucket surface are essential to prevent significant damage that may impair turbine performance. Preventive maintenance and timely replacement of worn components are key strategies for maintaining the durability and efficiency of Pelton turbine-based hydropower systems.

2.4 Torque Exerted on The Wheel

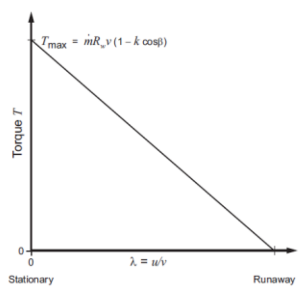

Figure 4 Variation of torque T with speed ratio λ

The force F_w acting on the bucket, resulting from the change in momentum of the water jet, as shown in Figure 4, is given by:

$$F_w = \dot{m}v - \dot{m}[u + k\ (v - u)cos\ \beta]$$

Or

$$F_w = \dot{m}(v - u)(1 - k\ cos\ \beta) \quad (23)$$

It is convenient to express the ratio of bucket speed $u$to jet speed v as:

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

Thus, the force can be rewritten as:

$$F_w = \dot{m}v(1 - \lambda)(1 - k\ cos\ \beta) \quad (24)$$

Therefore, the torque Texerted on the runner is:

$$T = \dot{m}R_w v(1 - \lambda)(1 - cos\ \beta) \quad (25)$$

For a given turbine operating at a constant flow rate (i.e., $\dot{m}$and $v$are constant), the torque Tvaries linearly with $(1-\lambda)$. The torque reaches its maximum when $\lambda = 0$(i.e., when the runner is stationary) and decreases to zero when $\lambda = 1$(i.e., when the bucket moves at the same velocity as the jet). This condition is known as the *runaway condition.*

1. Turbine Head (Ht)

The turbine head represents the difference in energy between two points in the fluid system and is calculated using:

$$Ht = \left[\frac{(p_1-p_2)}{\rho g} + \frac{(V_1^2-V_2^2)}{2g} + (Z_1 - Z_2)\right] \quad (26)$$

Where $H_t$is the turbine head, $p_1$and $p_2$are the pressures at points 1 and 2, $V_1$and $V_2$are the flow velocities at points 1 and 2, $\rho$is the fluid density, $g$is gravitational acceleration, and Z_1 and Z_2 are the elevations at points 1 and 2.

2. Torque Moment (Mt)

The torque moment is the rotational force acting on the turbine and can be expressed as:

$$Mt = F.r \quad (27)$$

Where $M_t$is the torque, $F$is the applied force (N), and $r$is the radius (m).

3. Flow Velocity (v)

The flow velocity can be calculated using:

$$v = \frac{Q}{A} \quad (28)$$

Where v is the flow velocity, Q is the volumetric flow rate, and A is the cross-sectional area.

4. Water Power (WHP)

Water power (WHP) is defined as the hydraulic power available from the flowing water:

$$WHP = \rho.g.Q.Ht \quad (29)$$

Where $\rho$is the water density (kg/m^3), $g$is gravitational acceleration (m/s^2), $Q$is the flow rate (m^3/s), and $H_t$is the turbine head (m).

5. Turbine Power (BHP)

Turbine power (BHP) is the mechanical power delivered by the turbine shaft and is given by:

$$BHP = 2\pi.Mt.N \quad (30)$$

Where M_t is the torque (Nm) and N is the rotational speed (rpm).

6. Turbine Efficiency (η)

Turbine efficiency is defined as the ratio of the output power to the available water power:

$$\eta = \left(\frac{BHP}{WHP}\right).100\% \quad (31)$$

Where $BHP$is the turbine power and $WHP$is the water power.

7. Schematic of Pelton Turbine System as presented in Figure 5.

Figure 5 Pelton Turbine Circuit Schematic

3. Result and Discussion

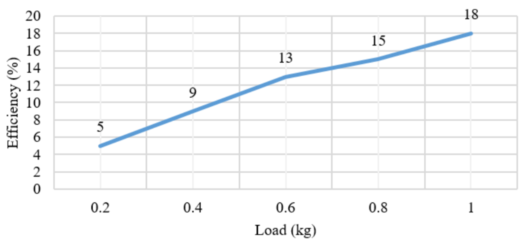

Figure 6 Graph of the relationship between load and efficiency in a single-nozzle configuration

In the single-nozzle study, five experiments were conducted with varying loads applied to the pony brake, ranging from 0.2 kg to 1 kg. The experimental results indicate that efficiency increases with load up to a certain limit, with a maximum of 18% at a load of 1 kg. During the experiments, the pressure was maintained at approximately ±15 psi, and the flow rate was kept at around ±43 L/min, as presented in Figure 6.

The observed increase in efficiency with increasing load can be explained by the fundamental principles of energy conversion in impulse turbines. Based on the turbine efficiency equation ($\eta = \frac{BHP}{WHP}$), efficiency improves when the mechanical output power increases relative to the available hydraulic power. Increasing the load

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increases the braking force on the turbine shaft, enabling more effective momentum transfer from the water jet to the bucket, resulting in higher torque and improved energy conversion.

From a theoretical perspective, optimal efficiency in a Pelton turbine is achieved when the bucket speed approaches approximately half of the jet velocity ($u \approx 0.5v$), where the transfer of kinetic energy from the jet to the bucket is maximized. This observation is consistent with previous findings reported [7], which emphasize the importance of load optimization in achieving maximum turbine efficiency.

Meanwhile, Figure 7 shows an inverse relationship between torque and rotational speed. At a load of 1 kg, the rotational speed is 425.53 rpm, whereas the highest rotational speed of 606.6 rpm is obtained at a lower load of 0.2 kg. This behavior is consistent with the fundamental theory of Pelton turbines, where torque is proportional to the difference between jet velocity and bucket velocity. Based on the relation $T \propto (1-\lambda)$, where $\lambda = \frac{u}{v}$is the speed ratio, and torque decreases as the rotational speed increases. As the bucket speed approaches the jet velocity, the effective momentum transfer decreases, resulting in lower torque.

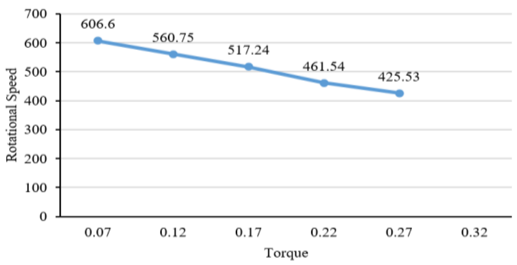

Figure 7 Graph of the relationship between torque and rotational speed in a single-nozzle configuration

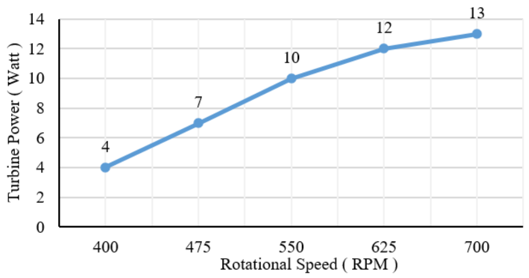

Figure 8 Graph of the relationship between rotational speed and turbine power in a single-nozzle configuration

In addition, Figure 8 illustrates the relationship between rotational speed and turbine power. The results show that higher rotational speeds yield higher power output, reaching a maximum of 13 W at 606.06 rpm. This behaviour can be explained using the mechanical power equation $P = T \cdot \omega$, where power depends on both torque and angular velocity. Although torque decreases with increasing rotational speed, the increase in angular velocity (ω) dominates, resulting in a higher overall power output.

This indicates that power generation in Pelton turbines is strongly influenced by rotational speed, particularly at high jet velocities. These findings are consistent with previous studies, such as [10], which reported that turbine power increases with rotational speed due to more effective conversion of kinetic energy into mechanical energy.

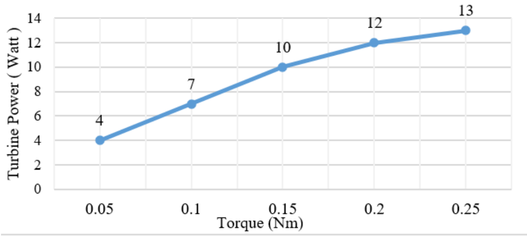

Figure 9 Graph of the relationship between torque and turbine power in a single-nozzle configuration

The relationship between torque and turbine power, shown in Figure 9, can be explained using the fundamental mechanical power equation $P = T \cdot \omega$, where power is directly proportional to both torque and angular velocity. The experimental results indicate that the highest power output of 13 W was achieved at approximately 0.3 Nm of torque, while the lowest power of 4 W occurred at approximately 0.06 Nm of torque.

This trend indicates that increased torque leads to higher power generation, as greater torque reflects a stronger impulse force exerted by the water jet on the turbine buckets, thereby enabling more effective energy transfer. However, it is important to note that power depends not only on torque but also on rotational speed. At higher torque values, the rotational speed may decrease, potentially limiting the increase in power output if not properly balanced.

Therefore, optimal turbine performance is achieved by balancing torque and rotational speed, ensuring efficient conversion of hydraulic energy into mechanical power. This behavior is consistent with the fundamental theory of impulse turbines and has also been reported in previous studies [10].

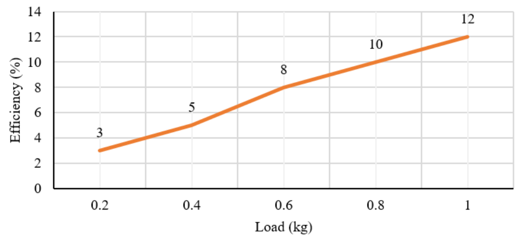

Figure 10 Graph of the relationship between load and efficiency in a two-nozzle configuration

In the experiment with two nozzles, the highest efficiency was 12%, whereas the single-nozzle configuration achieved 18%, as shown in Figure 10. This result indicates that increasing the number of nozzles does not necessarily improve turbine efficiency, even though it increases the total water flow rate and potential power input. During the experiment, the pressure was maintained at approximately ±15 psi, and the flow rate was kept at around ±43 L/min, consistent with the single-nozzle test conditions.

From a theoretical perspective, turbine efficiency is defined as the ratio between mechanical output power and hydraulic input power (η = BHP/WHP). Although using two nozzles increases hydraulic power input, it does not guarantee a proportional increase in mechanical output due to energy losses within the system.

The lower efficiency observed in the two-nozzle configuration can be attributed to fluid-dynamic effects, such as jet interference, turbulence, and a non-uniform momentum distribution across the turbine buckets. When two water jets interact with the runner simultaneously, the flow may become less stable, thereby reducing the effectiveness of momentum transfer compared to a single, well-directed jet.

These findings suggest that the performance of Pelton turbines is not solely dependent on increasing flow input, but also on how effectively the jet energy is converted into mechanical work. This result is consistent with the fundamental theory of impulse turbines and provides a more nuanced perspective compared to previous studies, such as [7], which generally reported performance improvements with multi-nozzle configurations. Therefore, proper optimization of nozzle number and flow distribution is essential to achieve maximum efficiency.

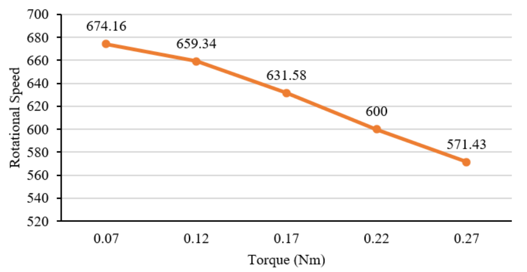

Figure 11 Graph of the relationship between torque and rotational speed in a two-nozzle configuration

In experiments with two nozzles, an inverse relationship between torque and rotational speed is also observed, similar to that in the single-nozzle configuration. However, higher rotational speeds are achieved when two nozzles are used. At a load of 1 kg, the rotational speed reaches 571.43 rpm, which is significantly higher than the 425.53 rpm obtained with a single nozzle (Figure 11).

This behaviour can be explained by the increase in total water flow rate and kinetic energy input when two nozzles are employed. A higher flow rate results in greater jet momentum acting on the turbine buckets, thereby increasing the runner's rotational speed. From a theoretical perspective, this is consistent with the principle that turbine speed is influenced by both jet velocity and flow rate, where increased hydraulic input leads to higher angular velocity.

Furthermore, the inverse relationship between torque and rotational speed remains valid and can be described using the speed ratio parameter $\lambda = \frac{u}{v}$. As the rotational speed increases, the bucket velocity approaches the jet velocity, reducing the effectiveness of momentum transfer and, consequently, decreasing torque.

These results are consistent with previous findings [10], which demonstrated that increasing the number of nozzles and the fluid flow rate results in higher turbine rotational speed due to increased fluid momentum input.

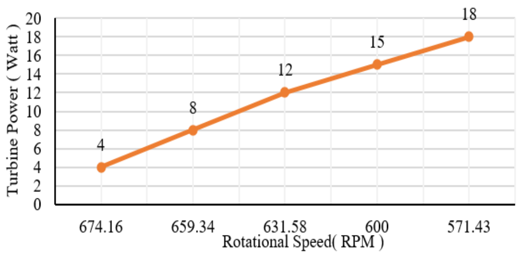

Figure 12 Graph of the relationship between rotational speed and turbine power in a two-nozzle configuration

The relationship between rotational speed and turbine power in the two-nozzle configuration does not strictly follow a simple inverse proportional trend, as shown in Figure 12. Instead, turbine power is governed by the fundamental equation $P = T \cdot \omega$, where power depends on both torque and angular velocity.

In this experiment, variations in power output are influenced by the interaction between torque and rotational speed. Although higher torque tends to reduce rotational speed due to increased mechanical resistance, the additional hydraulic input from the two-nozzle configuration helps maintain, or even increase, overall power output.

From a theoretical perspective, Pelton turbines convert the kinetic energy of high-velocity water jets into mechanical energy. The generated torque depends on the change in the water jet's momentum as it interacts with the turbine buckets, while the rotational speed determines how effectively this energy is converted into usable power.

Therefore, the observed behavior indicates that power generation in a multi-nozzle system results from a balance between torque and rotational speed, rather than a purely inverse relationship. This finding highlights the complex interaction between hydraulic input and mechanical output, which is consistent with the fundamental principles of impulse turbine operation.

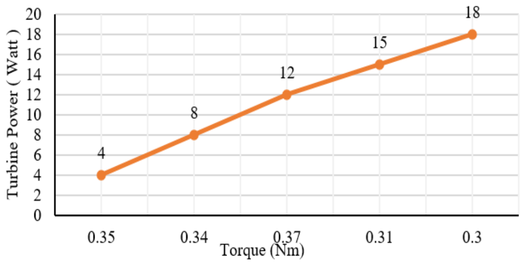

Figure 13 Graph of the relationship between torque and turbine power in a two-nozzle configuration

The relationship between torque and turbine power in the two-nozzle configuration can be explained using the fundamental mechanical power equation $P = T \cdot \omega$, where power is directly proportional to both torque and angular velocity, as shown in Figure 13. The experimental results indicate that higher torque generally leads to greater power output, as greater torque reflects a stronger impulse force from the water jet acting on the turbine buckets.

From a theoretical perspective, the torque produced in a Pelton turbine is a result of the change in momentum of the water jet as it strikes and is deflected by the bucket surface. A larger momentum change results in a higher force and, consequently, higher torque, which contributes to greater mechanical power generation.

However, it is important to note that the relationship between torque and power depends on rotational speed. As torque increases, rotational speed may decrease due to increased mechanical resistance, potentially limiting the overall increase in power if not balanced properly.

Therefore, optimal turbine performance is achieved by balancing torque and rotational speed, ensuring efficient conversion of hydraulic energy into mechanical energy. This behaviour is consistent with the fundamental principles of impulse turbine operation.

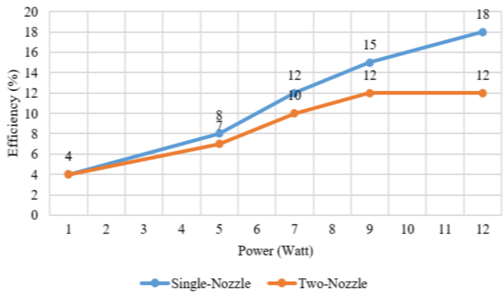

Figure 14 Graph of the comparison between the single-nozzle and two-nozzle configurations

The comparison between the single-nozzle and two-nozzle configurations shows that using two nozzles increases rotational speed and turbine power but does not necessarily improve efficiency, as presented in Figure 14. In this study, the single-nozzle configuration achieved the highest efficiency (18%), whereas the two-nozzle configuration reached only 12%.

From a theoretical perspective, turbine efficiency is defined as the ratio between mechanical output power and hydraulic input power (η = BHP/WHP). Although adding a nozzle increases hydraulic input, it does not guarantee a proportional increase in mechanical output because of energy losses within the system. These losses are primarily caused by jet interaction, turbulence, and non-uniform momentum distribution when multiple water jets strike the turbine buckets simultaneously.

This finding partially agrees with previous work [10], which reported that increasing the number of nozzles enhances turbine rotational speed due to higher flow input. However, the present results demonstrate that higher flow input does not necessarily lead to higher efficiency. This contrasts with the findings [7], which suggested that multi-nozzle configurations improve overall performance.

Therefore, the results of this study indicate that the performance of Pelton turbines is governed not only by the magnitude of hydraulic input but also by the effectiveness of momentum transfer and flow distribution. Increasing the number of nozzles should not be treated as a general method for improving performance; instead, proper optimisation must account for jet interactions and energy-loss mechanisms to achieve an optimal balance between power and efficiency.

4. Conclusions

The results of this study show that increasing the number of nozzle tends to increase rotational speed and turbine power, but decreases torque and overall efficiency. The single-nozzle configuration achieved the highest efficiency of 18%, whereas the two-nozzle configuration produced higher power output (18 W) but lower efficiency. These findings indicate that the performance of Pelton turbines is governed by a complex interaction between torque, rotational speed, and hydraulic input, in which increased power output does not necessarily correspond to higher efficiency.

From an engineering perspective, the selection of nozzle configuration should be tailored to the intended application. A single-nozzle system is more suitable for applications that prioritize efficiency, whereas a multi-nozzle configuration can be advantageous for systems requiring higher power output. Therefore, optimization of the number of nozzle and flow regulation is essential to achieve the desired balance between efficiency and power generation.

For future development, further studies are strongly recommended to evaluate the performance of Pelton turbines under real-scale operating conditions, particularly in micro-hydropower and small-scale hydropower systems. Experimental investigations should be extended to higher head conditions, variable flow rates, and continuous operation scenarios to better represent real-world applications.

In addition, integrating the turbine system with electrical generators and control systems is essential to assess its feasibility as a complete power generation unit. Future research should explore the coupling between turbine and generator performance, including load regulation, voltage stability, and overall energy conversion efficiency in real-scale systems.

Moreover, advanced numerical approaches such as computational fluid dynamics (CFD) are recommended to analyze internal flow characteristics, including jet interaction, turbulence, and energy loss mechanisms in multi-nozzle configurations. Such studies will provide deeper insight into optimizing nozzle design, jet angle, and flow distribution.

Finally, future work should focus on the development of integrated renewable energy systems by combining Pelton turbines with energy storage and smart control technologies. This approach will support the implementation of efficient and reliable hydropower systems for sustainable energy applications, particularly in remote and off-grid areas.

References

[1] S. Alirshedat, H. Attar, A. Saad, F. Al-Shami, A. Amer, Reactive lift-type Spherical Turbine Design for the Disi-Station Pipes in Jordan -Study Case, International Renewable Engineering Conference (IREC), 2021, DOI: 10.1109/IREC51415.2021.9427795

[2] L. Rodríguez, B. Pereiras, J. Fernandez-Oro, F. Castro, Optimization and experimental tests of a centrifugal turbine for an OWC device equipped with a twin turbines configuration, Energy, 171 (2019) 710 – 720

[3] J. Wu, K. Shimmei, K. Tani, K. Niikura, J. Sato, CFD-Based Design Optimization for Hydro Turbines, Fluids Engineering.(2006) 129(2): 159-168

[4] T. Schobeiri, Thermo-Fluid Dynamic Design Study of Single and Double-Inflow Radial and Single-Stage Axial Steam Turbines for Open-Cycle Thermal Energy Conversion Net Power-Producing Experiment Facility in Hawaii, Energy Resources Technology (1990), 112(1): 41-50

[5] C. H. Kostors, S.P. Vincent, Ammonia Turbine Design for Ocean Thermal Energy Conversion (OTEC) Plants, Solar Energy Engineering, (1981), Vol 103 (2) 92-97

[6] Youssef KM, El Kholy AM, Hamed AM, Mahmoud NA, El Baz AM, Mohamed TA. An Innovative Augmentation Technique

Of Savonius Wind Turbine Performance. Wind Engineering. 2020;44(1):93-112. doi:10.1177/0309524X19849860

[7] M. P. Rizky, Junaidi, F. Kurniawan, Analisa Kinerja Turbin Pelton 2 Nozle Dengan Variasi Jumlah Bucket, Buletin UtamaTeknik, (2022) Vol. 17, No. 2.

[8] A. Asrori, T. Adikusuma, E. Yudiyanto, Rancang Bangun Turbin Pelton Kapasitas 270 W Sebagai Alat Peraga Sistem Pembangkit Listrik Pico Hydro, BRILIANT: Jurnal Riset dan Konseptual, (2022), Vol. 7, No.2

[9] R. H. Apandi, (2018), Pengaruh Jumlah Bucket Terhadap Kinerja Turbin Pelton, Universitas Muhamadiyah Sumatra Utara, Indonesia

[10] Assauri. S, Margianto. M, & Marlina. E, Pengaruh Variasi Jumlah Nozzle Terhadap Daya Listrik Yang Dihasilkan Pada Prototype Turbin Pelton. Jurnal Teknik Mesin, (2017) 6(01).

[11] Bhattarai. S, Vichare. P, Dahal. K, Al Makky. A, Olabi. A. G, Novel Trends In Modelling Techniques Of Pelton Turbine Bucket For Increased Renewable Energy Production, Renewable and Sustainable Energy Reviews, 112 (2019) 87–101

[12] Maffrudin, Irawan. R. M, Setiawan. N, Rajabiah. N, Irawan. D, Pengaruh Jumlah Sudu Dan Diameter Nozel Terhadap Kinerja Turbin Pelton, Jurnal Program Studi Teknik Mesin UM Metro TURBO, (2019), Vol. 8 No. 2

[13] Nasir. B.A, Design Of High Efficiency Pelton Turbine For Microhydropower Plant, International Journal Of Electrical Engineering & Technology (IJEET), (2013), Vol. 4, Issue. 1, pp. 171-183

[14] Kapooria. R. K, An Efficiency Assessment Analysis Of A Modified Gravitational Pelton-Wheel Turbine, Journal of Energy in Southern Africa, (2009), Vol. 20 No. 4

[15] Ahrori. A. H, Kabib. M, Wibowo. R, Perancangan Dan Simulasi Turbin Pelton Daya Output Generator 20.000 Watt, Jurnal CRANKSHAFT, (2019), Vol. 2 No. 2

[16] International Hydropower Association (IHA), Hydropower Status Report, IHA Central Office, United Kingdom, 2021

Appendices

Symbol	Description	Units
Q	Flow rate	m^3/s
d_s	Diameter jet spray	mm
Z	Number of nozzles	Unit
Ht	Head Turbin	m
p_1	Pressure at the fluid surface	N/m^2
v	Flow velocity	m/s
Mt	Turbine torque moment	Nm
F	Force on the turbine	N
r	Radius	m
WHP	Daya air	KW
ρ	Density	kg/m^3
g	Gravity	m/s^2
N	Turbine rotation	Revolution per second
η	Efficiency	%
L_{ab}	Panjang lengan bucket	m
F	Torsi	N.m