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School of Engineering



Fluid Flow: Pelton Wheel

Structural & Fluid Analysis

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1 Abstract

This report investigates the performance and fluid flow characteristics of a laboratory Pelton wheel operating under a constant pressure head of 0.4 bar. The study examines the historic development and current technological advancements, materials, manufacturing methods, and relevant engineering standards. Experimental measurements of flow rate, shaft speed, braking force were used to plot speed against power. Results were compared against theoretical characteristics and the findings demonstrated that the maximum power and efficiency occurred at the intermediate rotational speed, where the balance between the torque and the angular velocity was optimal. Achieving an efficiency of 65%. Analysis of the fluid flow using boundary layer theory and flow separation identified spray losses, splashing, turbulence and secondary flow effects as contributing factors in the low peak efficiency score. The study also highlights the influence of measuring equipment selection and especially the force applicator, the quality of the Pelton wheel and bearing friction and casing design also contributed to the efficiency score being far below the expected 90% efficiency.

2 Introduction

This report analyses and critically appraises the experimental results of a 0.4 bar constant pressure head Pelton Wheel turbine conducted in the University of Wales Trinity Saint Davids laboratory. The study evaluates experimental results against research findings, analysing turbine characteristics and fluid flow using the theory of boundary layer behaviour and flow separation. To provide an in-depth understanding the application of Pelton wheels for energy generation especially with the importance and the demand for green energy in the current environment, detailed research and finding of Pelton wheels historical, current, and future development and technologies. The report also reviews the industry standard codes of practice and the industry standards for the application of Pelton Wheels. A Pelton wheel is a turbine that has ‘buckets’ around a rim of a wheel as shown in *Figure 1*, which is turned by the force of the flowing jet water hitting the bucket as stated by Glasgow Science Centre (2026).

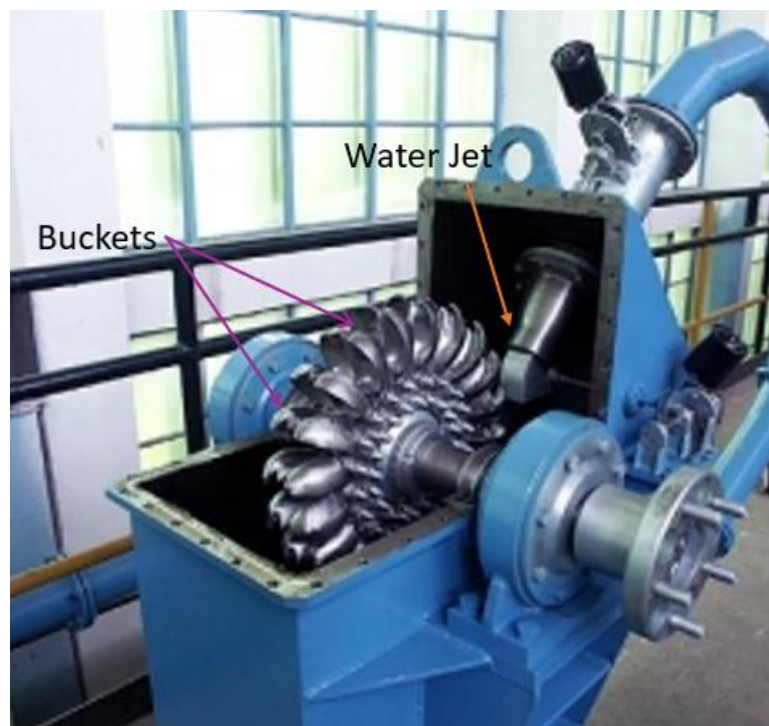


Figure 1 - Pelton Wheel (Linquip, 2025)

3 Historical, Current and Future Development of Pelton Wheels

3.1 Historical Development

Prior to the invention of the Pelton Wheel, water power was harnessed using traditional waterwheel and early turbine designs such as Fourneyron turbine transforming its hydraulic energy into mechanical energy as stated by Suarez et al, (2025). Lester Pelton's invention started from an observation in the 1870's of the effects of a defective key holding that caused the wheel to be misaligned to the shaft. Lucas and Abioye (2014) state Pelton noticed when the jet became misaligned with the centre of the 'bucket' and the water hit the edge the wheel then rotated faster, the later understanding of the fluid flowing in on one edge allowed the reverse direction to exit the other direction improving hydraulic efficiency. Pelton, (1880) as shown in his patent in *Figure 2* and stated in his patent designed a double cup instead of a single bucket with a wedge divider in the centre where the jet of water would be directed at the divider which enhanced the flows exit. "The Pelton Wheel Company formed, and mass production started in 1888 with the wheels achieving 90.2% efficiency," Wilson, (1995).

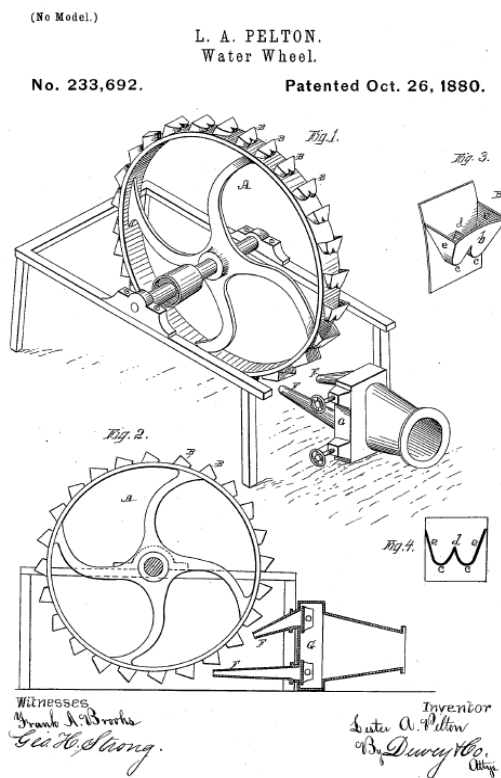


Figure 2 - Pelton Water Wheel Patent (Pelton, 1880)

3.2 Current Technological Trends and Future Developments

American Society of Civil Engineering (2026) state Pelton wheels were primarily used in mining and industrial applications to provide mechanical power for crushers, pumps, and other industrial machinery. In modern times, their role has shifted towards energy generation as Perez-Denicia et al, (2017) state hydropower plants provide an extremely efficient form of renewable energy, with the Pelton wheel are applicable to not only the large scale power plants but also to small scale installations that are essential in remote and mountainous regions where the environment offers the optimal conditions of high-head, low flow. These systems are increasingly integrated into the smart grid system providing responsive and sustainable electricity. International Hydropower Association (2025) shows the importance of the technology is increasing to solve the future energy challenges and its effectiveness as it states Costa Rica ran for 250 days entirely on renewable energy in 2016 with hydropower supplying more than 70% of the energy. With demand comes more research and development. Zhang (2016) indicates one of the current technologies available are Computational Fluid Dynamics (CFD) where the CFD numerically solves flow equations to predict fluid behaviour of how the liquid will flow as shown in *Figure 3* and interact with the surfaces. This allows improvement in the bucket geometry, nozzle designs, and number of nozzles. This reduces the need for expensive prototypes; a similar approach applies for Finite Element Analysis (FEA). Additive manufacturing has allowed for fast and a cost-effective way for prototypes and scale model testing. The limitation of scale modelling in hydrodynamics is the fluid behaviour, surface tension and boundary layer friction which does not scale linearly, meaning the Reynolds numbers will not be correct as stated by Chew et al, (2018). These current technologies as stated by Barragan et al, (2025) have increased the efficiency by 2.56%, with Hexagon (2023) stating that their focus in future developments are optimising the performances of micro-hydropower systems as these requires smaller reservoirs and use less energy to build. Artificial intelligence will also have a larger role to play in the future with optimising the function and performance of the Pelton, especially when it comes to integrating it with the other renewable energy sources.

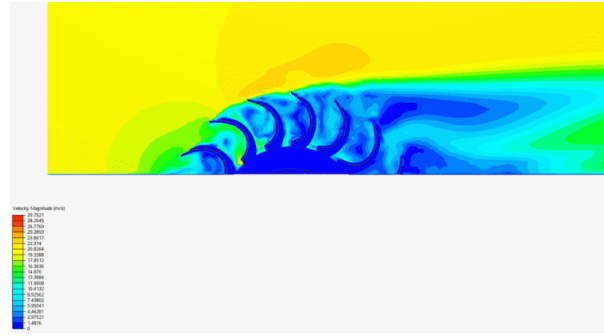


Figure 3 - CFD analysis of a Pelton turbine (SimScale, 2024)

3.3 Modern Materials and Manufacturing

Material choice for industrial purposes tends to be stainless steel as shown by Li and Saw, (2017) providing a cost-effective material with a high fatigue and corrosion resistance, these two properties are essential in ensuring longevity and safety of a component that is subjected to repeated stress. Stainless steel is manufactured by casting or forging with CNC used for precision machining to improve hydraulic efficiency, to further improve the wear resistance Delemontez et al, (2014) shows that high velocity oxy-fuel coating of tungsten carbide is applied. Aluminium and bronze are other options with Islam et al, (2013) demonstrating composite design in micro turbines using boron or Kevlar fibres this improves fatigue as shown in *Table 1*. A manufacturing process for this as shown by ORE catapult (2022) is called vacuum infusion, this process is where fibres in the form of mat is layered in a mould and placed in a vacuum bag and heated as the resin is pumped in and cured.

Table 1 - Properties of Fiber, Matrix, and Composites (Islam et al, 2013)

Fiber/Composite/ Matrix	Kevlar 49	Kevlar49/ Epoxy	E Glass	E Glass/E poxy	Epoxy
Properties					
Specific gravity	1.45	1.38	2.54	1.85	1.2-1.3
Tensile Modulus, GPa	131	75.8	72.4	39.3	2.75-4.1
Tensile strength, GPa	3.62	1.387	3.54	0.965	.055-.13
Elongation %	2.8	.016	4.8	.023	1.5-3.5
Flexural strength, MPa		1240		1020	124.1

4 Experimental Procedure and Turbine Characteristics

4.1 Experimental Procedure

The experiment was conducted on a UWTSD Pelton Wheel test rig as seen in *Figure 4*. At the start of the experiment the needle nozzle was fully opened before the pump was started, the needle nozzle was then adjusted until the manometer dial indicated 0.4 bar. Once the constant head pressure was achieved the flow rate needed was determined, this had been conducted by closing the water outlet in the draining tank and measuring the volume collected per minute. With three measurements taken the average flow rate was 29 L/min and producing a calculated hydraulic power of 19.33 watts. The braking device was then adjusted in increments of one newton; at each incremental torque, the spring readings of the opposing spring balance reading was recorded. The shaft speed was measured with a tachometer. The braking force was obtained from the difference between the two spring balance readings with the torque then calculated by multiplying with the radius of the belt pulley also shown in *Figure 4*. The calculation for the shaft power used the recorded speed and the shaft torque with the efficiency calculated by dividing the shaft power with the calculated hydraulic power, these equations can be seen in *Figure 5*.

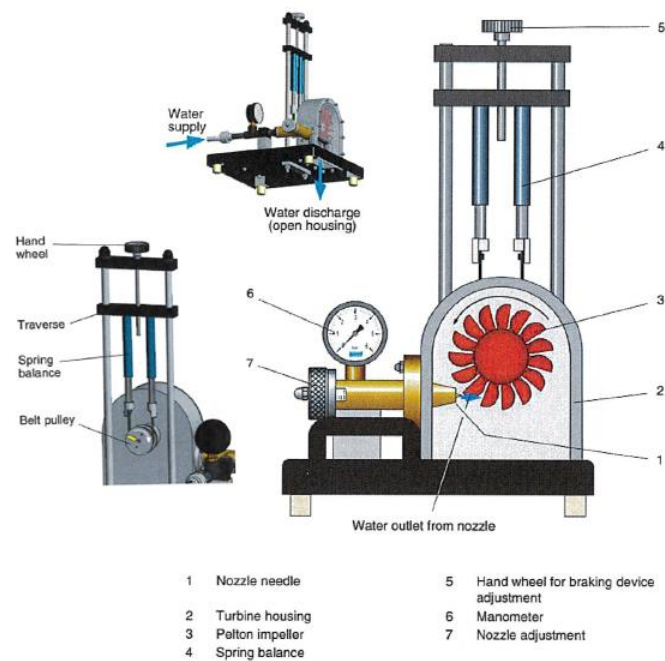


Figure 4 - UWTSD Pelton Wheel (UWTSD, 2026)

Determining the Volumetric Flow

To determine the volumetric flow $\dot{V}$, stop the time t required to fill the volumetric tank of the HM 150 from 20 to 30 litres, for example.

$$\dot{V} = \frac{V_2 - V_1}{\Delta t}$$

Determining the Torque

- To determine the torque a load must be produced with the braking device.
- A torque can be determined in this way using the spring balances and the diameter of the belt pulley.

$$M = F \cdot \frac{D}{2}$$

with

$$D = 50 \text{ mm}$$

Outgoing power on the turbine shaft:

$$P_{out} = M \cdot 2 \cdot \pi \cdot \frac{n}{60}$$

P_{out} = outgoing power in W

M = torque in Nm

n = speed in min^{-1}

Efficiency

$$\eta = \frac{P_{out}}{P_{hyd}} \times 100$$

η = efficiency

P_{out} = outgoing power in W

P_{hyd} = hydraulic power in W

Figure 5 - Equations (UWTSD, 2026)

4.2 Results

The experimental results in *Table 2* shows that with lightly loaded braking force the pulley speed was 1185 rpm and as the braking force increases the speed decreases in a linear fashion, the brake force required to stop the Pelton wheel from rotating with a hydraulic power of 19.33 watts is 12.25 N. You can also see that as the torque on the shaft increases as the braking force increases. *Table 2* also shows that the efficiency and the shaft power increase as the braking force increases before plateauing at a shaft power of 12.57 watts between braking force of 7.5 N and 8 N before then decreasing again.

Table 2 - Experiment Results

Experiment number 1 (0.4 Bar)							
Force Meter 1 (N)	Force Meter 2 (N)	Speed (Rev/Min)	Braking Force (N)	Torque on Shaft (Nm)	Power on Shaft (W)	Hydraulic Power (W)	Efficiency (%)
1	0.50	1185	0.5	0.01	1.55	19.33	8.02
2	0.75	1140	1.25	0.03	3.73	19.33	19.30
3	1.00	1100	2	0.05	5.76	19.33	29.79
4	1.25	1020	2.75	0.07	7.34	19.33	37.98
5	1.50	975	3.5	0.09	8.93	19.33	46.21
6	2.00	935	4	0.10	9.79	19.33	50.64
7	2.25	885	4.75	0.12	11.01	19.33	56.92
8	2.75	830	5.25	0.13	11.41	19.33	59.01
9	3.00	775	6	0.15	12.17	19.33	62.97
10	3.50	725	6.5	0.16	12.34	19.33	63.81
11	4.00	685	7	0.18	12.55	19.33	64.93
12	4.50	640	7.5	0.19	12.57	19.33	65.00
13	5.00	600	8	0.20	12.57	19.33	65.00
14	5.25	530	8.75	0.22	12.14	19.33	62.80
15	5.50	465	9.5	0.24	11.56	19.33	59.82
16	6.00	400	10	0.25	10.47	19.33	54.17
17	6.25	310	10.75	0.27	8.72	19.33	45.13
18	6.50	235	11.5	0.29	7.08	19.33	36.60
19	7.00	150	12	0.30	4.71	19.33	24.37
20	7.75	0	12.25	0.31	0.00	19.33	0.00

Figure 6 shows the speed against power curve, rather than increasing continuously. The power on the shaft initially rose from 1.55 watts to its maximum value of 12.57 watts between 600 and 640 revolutions per minute, before decreasing at a faster rate, the error bars indicate the measured uncertainties and can clearly be seen increasing as the speed increases. Figure 7 shows speed against efficiency showing the same pattern with the efficiency increasing from 8% up to 65% before decreasing again and going all the way to 0%. Both graphs show that the optimum operating speed for power output and efficiency is between 775 and 530 revolutions per minute where the efficiency is still within 2% of its max and the output power does not drop below 12 watts.



Figure 6 - Speed v Power Graph (Including error bars)



Figure 7 - Speed v Efficiency Graph

The speed versus torque graph in *Figure 8* shows there is an inverse relationship between the rotational speed of the pulley and the output torque. Initially the braking force caused only a small reduction in speed while the torque increased at a greater rate, as the braking force increased the reduction in speed between each step increases until the turbine eventually stalled.

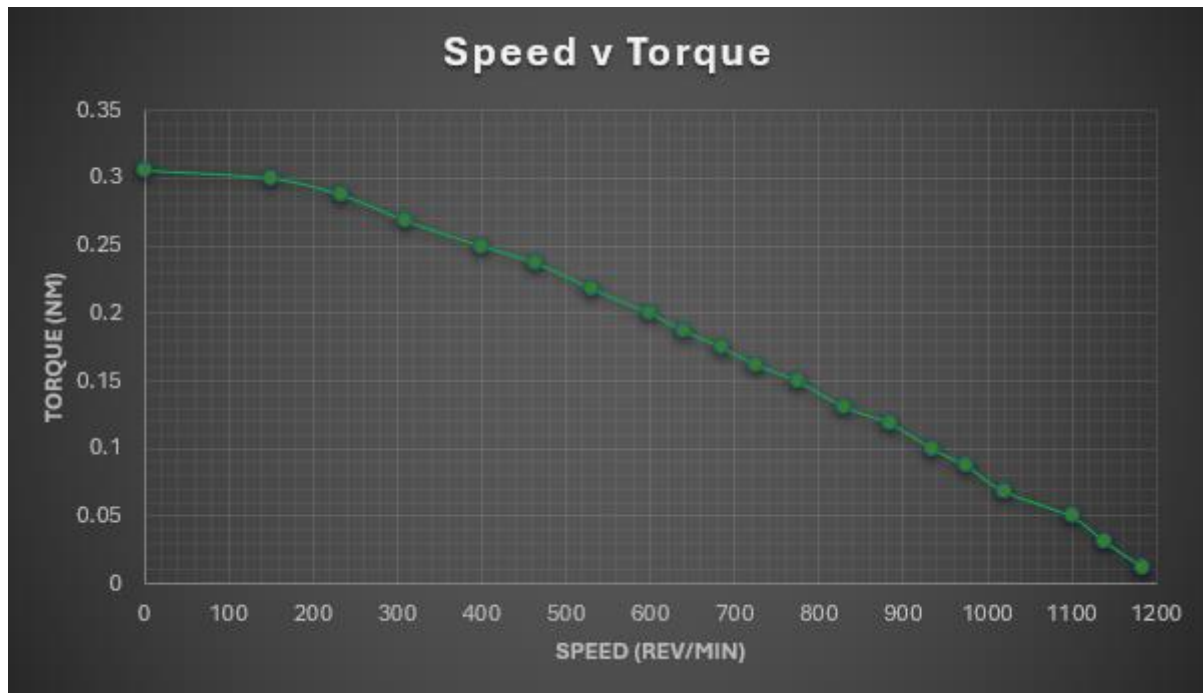
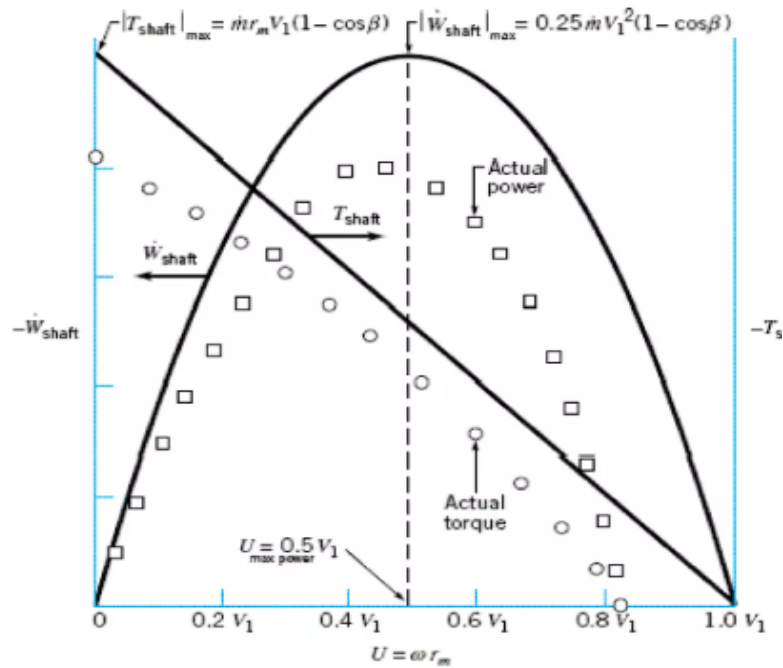


Figure 8 - Speed v Torque Graph

4.3 Turbine Characteristics

The experimental trends follow the theoretical characteristics of a Pelton Wheel as shown in *Figure 9*, Perrig et al, (2006) explains as the bucket velocity increase the difference between the velocity of the water leaving the jet and the bucket decreases, meaning the change in momentum decreases and therefore the reduction in torque and the downward slope in the graph. The power and efficiency versus speed produces the expected bell-shaped curve. Zhang (2016) also indicate that the peripheral speed coefficient is around 0.45 to 0.48 intentionally reduced from the peak power, to reduce the wear on the components.



Typical theoretical and experimental power and torque relation for a Pelton turbine as a function of bucket speed

Figure 9 - Power, Torque and Speed Graphs (Nagpurwala, 2026)

4.4 Critical Appraisal of Experiment

Although the experimental trends agree with the theoretical expectations several sources of inaccuracy, quality and uncertainties may have influenced the results and are a reflection on the error bars in *Figure 6*. The calculations were conducted with a constant head pressure of 0.4 bar, as the water was pumped up to the jet there would have been slight fluctuations the dial gauge did not display this and a digital gauge with a greater resolution would have. Since hydraulic power is directly proportional to pressure head and flow rate an overestimation in hydraulic power would consequently reduce the calculated efficiency. The flow rate was measured manually even though this was conducted three times to achieve an average, the accuracy of this due to human reactions in the process of closing the drain hole, starting and the timer and monitoring the level can be vast. This would have meant that the hydraulic power may have been incorrect in *Table 2* and affect the calculation results. During the experiment, the fluctuation in the reading of the pulley speed and the spring balance, where the force meter reading was read from fluctuated considerably at times. These factors also show that inaccurate force readings would affect the calculated torque which then would

affect the shaft power and efficiency results, a dynamometer would reduce this error risk. The accuracy and the efficiency of the buckets and the jet set-up is unknown, poor alignment and inadequate maintenance will severely affect the performance of the components. Friction in the pipe, jet and the bucket will also influence the hydraulics performance and *Figure 10* shows the amount of secondary turbulence is in the Pelton wheel with the splashes, spray loss and the water not fully exiting the bucket. Mechanical losses for example bearing friction will reduce the measured performance, the friction created on the pulley by using the braking device to measure the force would have been a large factor in the measured poor efficiency score in the test of 65% which is considerably lower than the 90% achieved in commercial Pelton turbines. The belt would have created a large amount of friction, as the braking system introduced as a measurement system introduced additional mechanical losses due to it being a contact-based measurement instrument effecting the energy efficiency of the experiment. During the experiment of different pressure heads the original tachometer was faulty and the speeds where extremely high and can be seen in the results in the Appendix A, this was rectified prior to this experiment. Despite this limitation, the experiment data closely matched theoretical patterns and expectations, clearly demonstrating that the maximum power is obtained at an intermediate speed where the torque and angular velocity are optimally balanced with the torque at its highest when the Pelton was on stop and decreasing as the speed increased.

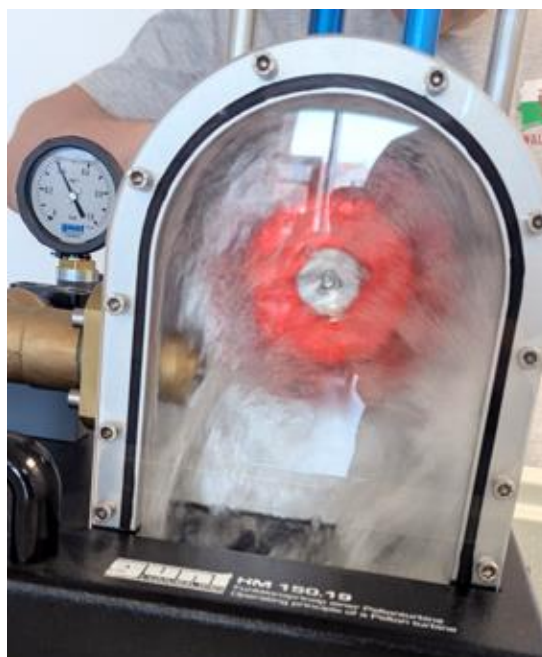


Figure 10 - Pelton Wheel's Hydraulic Inefficiency

5 Fluid Flow Around the Pelton Wheel

The fluid flow behaviours can be separated into zones, from the pressure head to the nozzle where the potential energy is converted into kinetic energy, the fluid boundary layers will develop around the contact surface of the pipes and the nozzle. The nozzle design, finish quality and setup have a significant role in reducing the aeration and break-up in the fluid as it exits the nozzle and travels through the air. When the fluid reaches the bucket, the splitter divides the jet flow in two equal streams as shown in *Figure 11*. The bucket entry cup design and angle as indicated by Atthanayake, (2009) allows the fluid spread as a thin water sheet with the deflection angle of approximately between 165 and 170 degrees, ensuring the most efficient balance of creating maximum torque transferring maximum possible water velocity without creating a large flow exit build up and directional flow exit on the next bucket. The bucket roughness has a severe effect on the boundary layer creating viscous shear, the greater the surface roughness the larger the boundary -layer growth and can cause separation. Water retaining on the bucket as shown in *Figure 12* after its cycle will affect the boundary layer with the jet velocity contacting the water retained by centrifugal effects. creating a difference in fluid velocity and adding the shear effect and increasing turbulence, ultimately reducing momentum change. This effect can be seen in *Figure 13* as the turbine speed velocity is above the optimal working speed.

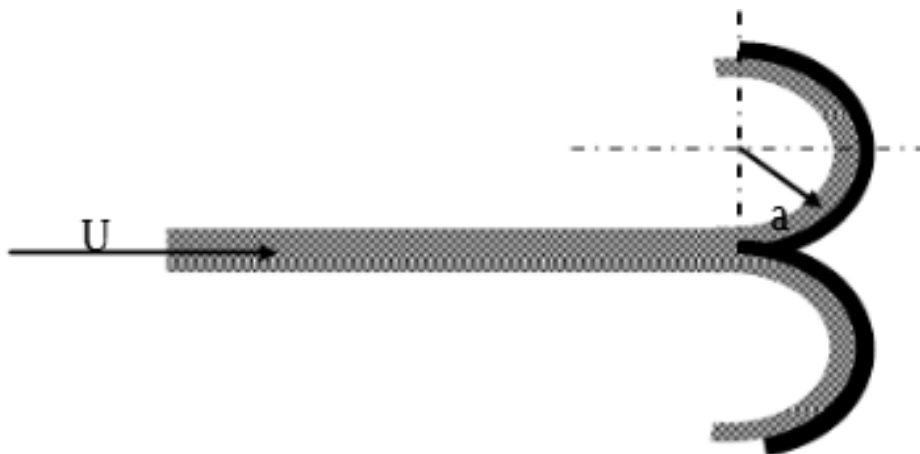


Figure 11 – Flow Through Pelton Turbine Bucket (Atthanayake, 2009)

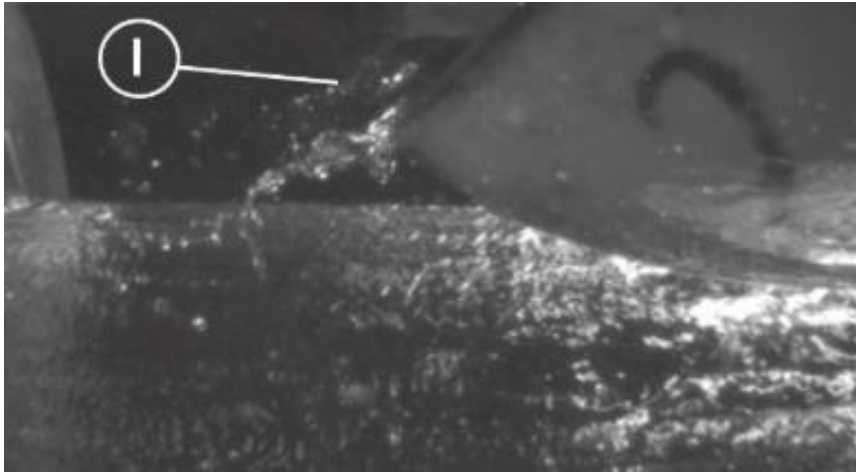


Figure 12 - - Bucket water retention (Perrig et al., 2006)

Water retention creates a difference in fluid velocity and adding the shear effect and increasing turbulence, ultimately reducing momentum change. This effect can be seen in *Figure 13* as the turbine speed velocity is above the optimal working speed.



Figure 13 - Pelton Wheel Speed Above Optimal Speed

The flow separation as shown in *Figure 14* is also more pronounced when the turbine speed is below the optimal working speed at the velocity of flow exiting the bucket is still high and exiting flow is diverting up and against the incoming buckets.

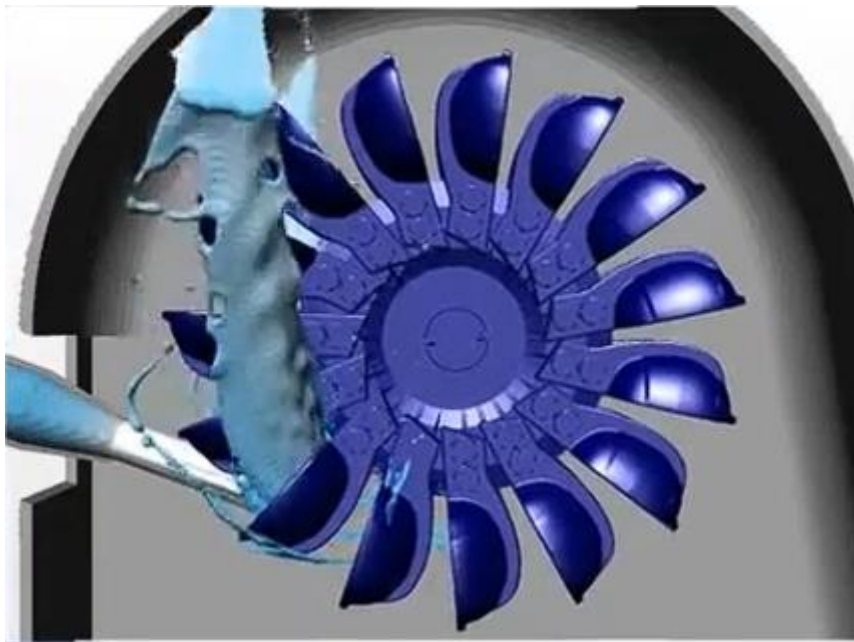


Figure 14 - Pelton Wheel Fluid Flow from Slow Turbine Speed (FLOW-3D, 2016)

5.1 Critical Appraisal of Fluid Flow

The experiment results and the observation of the fluid flow demonstrated the expected behaviour where the high velocity jet water transfers momentum to the buckets and is redirected via the bucket shaped angle. The images and results also show the expected behaviour of the fluid across the range of speed, with the velocity of the water returning off the bucket remaining high when at low Pelton wheel speed and the water being retained in the bucket by centrifugal forces creating a build-up excessive turbulent water at the jet entry point when the wheel speed was above optimum working speed. The inconstancy of the volume flow will also affect this; this is shown in Appendix A where the experiment for them the volume flow was calculated at 38.7 and 32.9 L/min. The low peak efficiency shows that the flow around the Pelton wheel was not ideal, this is confirmed with the images showing visible spray, splashing, aeration and turbulent water. This may have been caused by

poor alignment with the nozzle and the wheel, the design efficiency of the buckets, the quality finish of the bucket, excessive flow rate, the cast design, and shape including the water exit location. These several factors of mechanism are factors that the experimental efficiency was 65%, which is considerably lower than industrial Pelton wheels indicating the power transfer loss due to boundary layer growth along the buckets, spray and turbulence all impacted the hydraulic efficiency of the Pelton Wheel.

6 Codes of Practice and Industrial Standards

The testing, manufacturing and design of Pelton turbines are governed by internationally recognised standards. IEC 63461:2024, provides defined terms and quantities and specify the methods of testing Pelton turbine models with unit power greater than 5 MW. The aim is to assert the hydraulic performance of the model, the computational results and comparison with guarantees to ensure that contract guarantees have been met IEC, (2024). Following installation, performance is verified using IEC 60041:1991 which specifies field acceptance tests for hydraulic impulse turbines. Ensuring that contractual performance is met by testing site-specific conditions including flow, head, and tail water IEC, (1991) other nations for example may use their own standards in this case American Society of Mechanical Engineers PTC 18. Hydro-abrasive erosion reduces the service life and increases maintenance IEC 62364:2019 provides data on erosion rates against different water quality and guidelines for minimising erosion through hydraulic designs, material selection, protective coating, and operating practices IEC, (2019). IEC 61362:2024 contains guidelines on specifications and performances of hydraulic turbine governing and control systems, unifying the selection of relevant parameters and standardisation of terminology and safety device requirements. Standardising governs such items as speed regulation, flow control, power output, improving operational stability, frequency regulation, and safety IEC, (2024). Material quality and traceability are essential to ensure safety and reliability, EN 10204 is a European standard that govern material test certificates, providing four levels of inspection with the highest being validated by a third party, essential for safety-critical components such as turbine runners and shafts CEN, (2004). In aid of supporting the direct relationship between the purchaser and the supplier throughout the project lifecycle, BS EN IEC 61116:2025 aims specifically at supplying the purchaser the essential information on preparation and evaluation of tenders, supplier communication and quality control during the design, testing and manufacturing, site development, commissioning, acceptance test, maintenance and operation BSI, (2025). Although not formal engineering standards, The Hydraulic Sustainability Guidelines on Guidelines on Good Industry Practice (HGIIP) represent recognised good industry practice by encouraging environmentally responsible design and sustainable operation throughout the lifecycle of the hydropower project. International Hydropower Association (2021) state by engaging with stakeholders, this guideline also complements the IEC standards by ensuring the performance of the Pelton wheel isn't reduced, ensuring not only high engineering performances but also long term environmental and social sustainability, any task carried out in the United Kingdom must also adhere to the Health and Safety at Work Act 1974.

7 Conclusion

Overall, the experimental investigation successfully demonstrated the operating characteristics and fluid flow behaviour of a Pelton wheel. The measured relationships between hydraulic power, force, rotational speed, shaft torque, power, and efficiency all followed the theoretical expectations. The low peak efficiency score of the experimental Pelton wheel could be explained with the aid of the observations seen during the experiment with the fluid flow behaviour and the equipment used to collect the data. The wider review also showed the advances in Pelton Wheel designs with the aim for greater efficiency, longer service of life and sustainability with the aid of research and internationally recognised standards.

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9 Appendices

Appendix A

Experiment number 2 (0.38)							
Force Meter 1	Force Meter 2	Speed	Braking Force	Torque on Shaft	Power on Shaft	Hydraulic Power	Efficiency
1	0.5	3467	0.5	0.01	4.54	24.51	0.19
2	0.75	3312	1.25	0.03	10.84	24.51	0.44
3	1	3199	2	0.05	16.75	24.51	0.68
4	1.5	2987	2.5	0.06	19.55	24.51	0.80
5	1.75	3081	3.25	0.08	26.21	24.51	1.07
6	2	2740	4	0.10	28.69	24.51	1.17
7	2.5	2640	4.5	0.11	31.10	24.51	1.27
8	2.75	2493	5.25	0.13	34.26	24.51	1.40
9	3	2303	6	0.15	36.18	24.51	1.48
10	3.25	2201	6.75	0.17	38.89	24.51	1.59
11	3.5	2066	7.5	0.19	40.57	24.51	1.66
12	4	1858	8	0.20	38.91	24.51	1.59
13	4.25	1730	8.75	0.22	39.63	24.51	1.62
14	4.5	1485	9.5	0.24	36.93	24.51	1.51
15	4.75	1280	10.25	0.26	34.35	24.51	1.40
16	5	1140	11	0.28	32.83	24.51	1.34
17	5.75	806	11.25	0.28	23.74	24.51	0.97
18	6	626	12	0.30	19.67	24.51	0.80
19	6.5	448	12.5	0.31	14.66	24.51	0.60
20	7	283	13	0.33	9.63	24.51	0.39

Figure 15 - Result Table for 0.38Bar

Experiment number (0.42)							
Force Meter 1	Force Meter 2	Speed	Braking Force	Torque on Shaft	Power on Shaft	Hydraulic Power	Efficiency
1	0.5	3604	0.5	0.01	4.72	23.03	0.20
2	0.75	3453	1.25	0.03	11.30	23.03	0.49
3	1	3367	2	0.05	17.63	23.03	0.77
4	1.5	3117	2.5	0.06	20.40	23.03	0.89
5	1.75	3003	3.25	0.08	25.55	23.03	1.11
6	2	2887	4	0.10	30.23	23.03	1.31
7	2.5	2719	4.5	0.11	32.03	23.03	1.39
8	3	2516	5	0.13	32.93	23.03	1.43
9	3.25	2343	5.75	0.14	35.27	23.03	1.53
10	3.5	2176	6.5	0.16	37.03	23.03	1.61
11	4	2019	7	0.18	37.00	23.03	1.61
12	4.25	1831	7.75	0.19	37.15	23.03	1.61
13	4.5	1629	8.5	0.21	36.25	23.03	1.57
14	4.6	1427	9.4	0.24	35.12	23.03	1.52
15	4.75	1176	10.25	0.26	31.56	23.03	1.37
16	5.25	904	10.75	0.27	25.44	23.03	1.10
17	5.5	675	11.5	0.29	20.32	23.03	0.88
18	6	460	12	0.30	14.45	23.03	0.63
19	6.25	265	12.75	0.32	8.85	23.03	0.38
20	7	0	13	0.33	0.00	23.03	0.00

Figure 16 - Result Table for 0.42Bar

