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Structural & Fluid Analysis

Fluid Flow: Pelton Wheel (ACEA6005)

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Introduction

Pelton turbines are impulse-type hydraulic machines designed primarily for high-head, low-flow hydroelectric applications. Their operation relies on converting hydraulic energy into a high-velocity water jet, which transfers momentum to a series of curved buckets mounted around the runner. The efficiency and reliability of this process depend on hydraulic design, material selection, manufacturing quality and appropriate operating conditions.

This report examines the development and performance of the Pelton wheel through a combination of secondary research, laboratory testing and fluid-flow analysis. Historical and modern developments in bucket design, materials, manufacturing and emerging technologies are first considered. Experimental testing is then used to determine the speed-power characteristics and efficiency of a laboratory Pelton turbine under constant hydraulic conditions, followed by a critical appraisal of the measured results. The fluid behaviour through the nozzle and across the runner is subsequently analysed using concepts including Bernoulli's equation, momentum transfer, boundary layer behaviour and flow separation. Finally, relevant industry standards and machinery-safety requirements are considered.

1. Development of Pelton Wheels

Early designs of impulse water turbines suffered from poor energy transfer as water leaving the buckets retained significant velocity and could interfere with the neighbouring buckets. Lester Pelton developed the split-bucket concept during the late nineteenth century, allowing the incoming high-velocity jet to be divided and redirected more effectively. Pelton first began experimenting with improved bucket designs in 1878, leading to the development of the split-bucket principle, which was patented in 1880 (ASME, 1991). The initial design proved effective for high-head, relatively low-flow water resources and became widely used in early industrial applications such as mining and hydroelectric power generation. Further developments by engineers such as William Doble improved the bucket geometry and nozzle design, establishing the basis for the modern Pelton turbine.

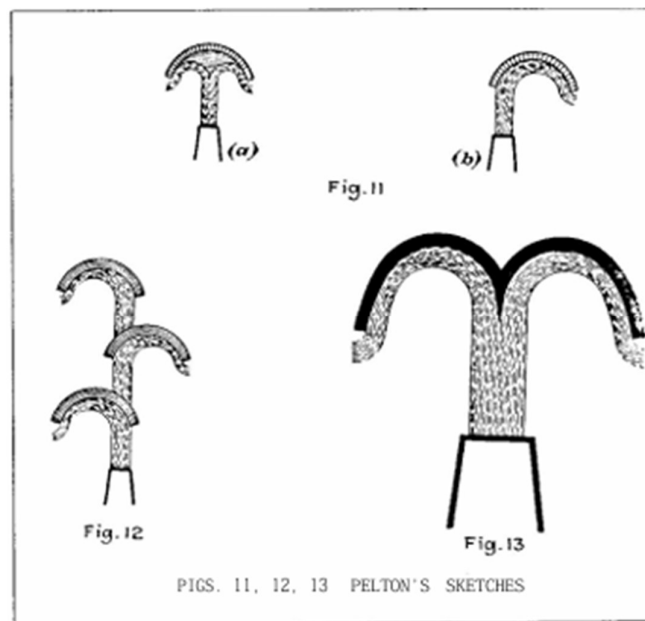


Figure 1. Early Pelton bucket design sketches, including the split-bucket concept developed during the late nineteenth century (ASME, 1991).

Modern development of the Pelton turbine primarily focuses on improving hydraulic efficiency through the refinement of bucket geometry, jet quality and nozzle control. Computational fluid dynamics is now also widely used to model the highly unsteady, three-dimensional interaction between the water jet and the rotating buckets, allowing designers to assess pressure distribution,

momentum transfer and hydraulic losses before manufacture. Recent numerical studies have shown that small changes in bucket geometry can influence torque production and energy conversion, making numerical analysis an important tool in modern Pelton turbine design (Sun and Tan, 2024). Multi-jet arrangements and improved flow-control systems are now also used to increase output and allow turbines to operate effectively across a wider range of flow conditions. These developments allow modern Pelton turbines to achieve higher efficiency while also reducing the need for repeated physical prototype testing.

The significance of these developments is not limited to improved peak efficiency. Numerical optimisation and improved flow-control methods can reduce the number of physical design iterations required, shorten development time and allow turbine geometry to be better matched to a wider range of operating conditions. This makes modern design methods valuable not only hydraulically, but also in terms of manufacturing cost, development risk and operational flexibility.

Cyclic loading, high velocity impact and erosion from suspended sediment are all conditions that modern Pelton runners are designed to withstand. Martensitic stainless steels such as CA6NM are commonly used because they offer useful combinations of strength, toughness, corrosion resistance and resistance to cavitation and erosion (Ferreño et al., 2011). Manufacturing methods depend greatly on runner size and duty, but common processes include casting, forging, welding and precision machining of the bucket surfaces. In some cases, industrial manufacturers use fully forged and welded runner designs to improve structural integrity and production control (ANDRITZ, 2019). Bucket geometry and surface finish are particularly important because dimensional errors, roughness or defects can disturb the jet path, increase hydraulic losses and create local stress concentrations. Quality assurance therefore includes dimensional inspection, surface-finish control and non-destructive examination of critical regions, helping to reduce the likelihood of fatigue cracking or premature erosion in service (Ge et al., 2023).

Future development of Pelton turbines is likely to focus on further numerical optimisation, improved surface technologies and increased use of condition-monitoring systems. More advanced CFD methods can allow bucket and nozzle geometry to be refined before manufacture, while erosion-resistant coatings and improved materials may extend runner life in sediment-laden water (Ge et al., 2023). Digital monitoring of operating performance may also support earlier identification of deterioration and maintenance requirements. These developments are particularly valuable where turbines operate under variable or erosion-prone conditions, as improved control, monitoring and erosion-mitigation strategies may help maintain efficiency while reducing downtime and maintenance costs (Shrivastava and Rai, 2025).

2. Experimental Determination of Pelton Turbine Characteristics

2.1 Experimental Aim and Operating Conditions

The purpose of the experiment was to determine the operating characteristics of a laboratory Pelton wheel under controlled hydraulic conditions. The inlet nozzle was fixed in the open position and the pressure maintained at 0.4 bar. The measured flow rate was recorded at 32 L/min.

During the experiment, the mechanical braking load was progressively increased using the belt-brake arrangement at the rear of the turbine. At each load increment, readings were taken from Force Meter 1 and Force Meter 2, together with the corresponding rotational speed in rev/min.

Once all load increments had been completed and the experimental data recorded, the measurements were used to determine braking force, shaft torque and output power, allowing the speed-power characteristic of the Pelton wheel to be established.

2.2 Experimental Apparatus and Procedure

The principal components of the laboratory Pelton wheel test rig are shown in Figure 2. The laboratory test rig comprised a Pelton runner supplied by an adjustable water nozzle, together with a pressure gauge and flow-measurement arrangement used to establish the hydraulic operating conditions. A belt pulley, braking adjustment and two spring balances formed the mechanical loading system, while rotational speed was measured at each applied load.

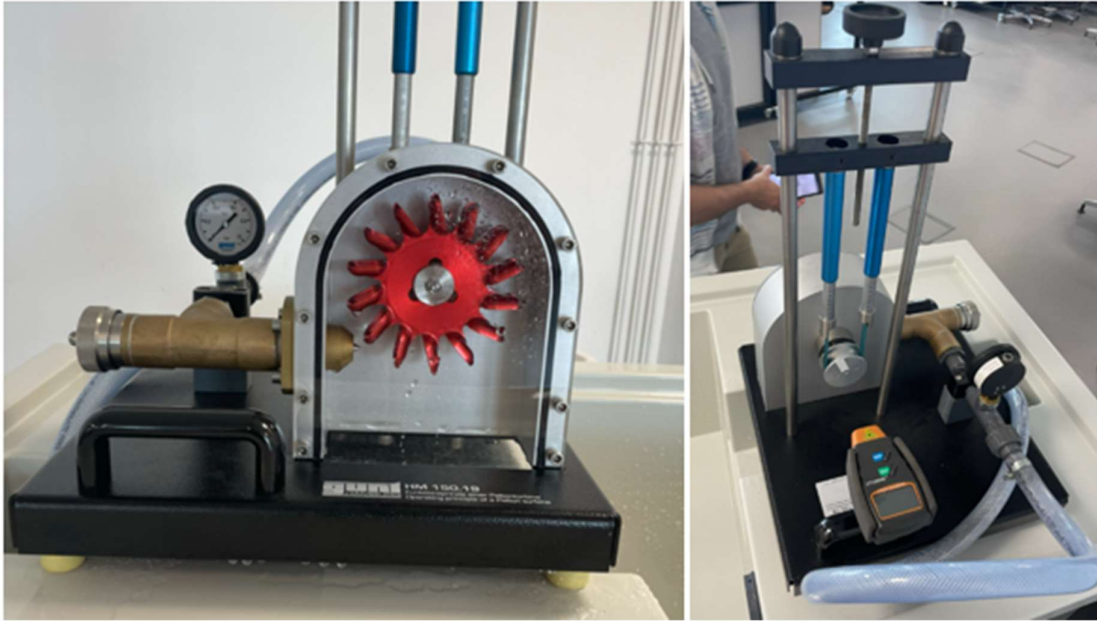


Figure 2. Laboratory Pelton wheel test rig: (a) front view showing the turbine and hydraulic components; (b) rear view showing the belt-brake and force measurement arrangement.

Before measurements were taken, the hydraulic operating conditions were established and held constant throughout the test.

- The nozzle was fully opened.
- The water inlet was opened and the pump was switched on.
- The pressure was adjusted to 0.4 bar.
- The discharged volume was measured over a period of 1 minute to determine the volumetric flow rate.

Once these parameters had been established, the test was completed by progressively adjusting the braking device to increase the mechanical load. At each load condition, readings from Force Meter 1 and Force Meter 2 were recorded together with the rotational speed of the belt pulley. The progressive loading enabled the relationship between shaft torque, rotational speed and output power to be established across the operating range. Using several load increments also provided sufficient data points to identify the maximum-power region and assess the overall shape of the turbine characteristic curve.



Figure 3. Rear view of the Pelton turbine test rig showing the fixed laser tachometer used to measure rotational speed at the belt pulley.

2.3 Data Reduction and Calculations

The recorded force and rotational-speed measurements were converted into braking force, shaft torque, shaft power and turbine efficiency. The effective braking force was determined from the difference between the two force-meter readings. This force was then applied at the radius of the belt pulley to calculate shaft torque. Shaft output power was calculated from the product of torque and angular velocity. As the hydraulic pressure and volumetric flow rate were maintained at constant values, the hydraulic input power remained constant throughout the experiment.

Braking Force

$$F = F_1 - F_2 \quad (1)$$

where:

- F = effective braking force, N

- F_1 = Force Meter 1 reading, N
- F_2 = Force Meter 2 reading, N

Shaft torque

The torque applied to the shaft was calculated using:

$$M = F \frac{D}{2} \quad (2)$$

where:

- M = shaft torque, Nm
- F = effective braking force, N
- D = belt-pulley diameter, m

The pulley diameter was:

$$D = 50 \text{ mm} = 0.050 \text{ m}$$

therefore, the pulley radius was:

$$\frac{D}{2} = 0.025 \text{ m}$$

and Equation (2) may also be written as:

$$M = 0.025 \times F \quad (3)$$

The 50mm pulley diameter is specified in the laboratory worksheet.

Shaft output power

Angular velocity was calculated from the measured rotational speed:

$$\omega = 2\pi \frac{n}{60} \quad (4)$$

where:

- ω = angular velocity, rad/s
- n = rotational speed, rev/min

Shaft output power was then calculated using:

$$P_{out} = M\omega \quad (5)$$

Substituting Equation (4)

$$P_{out} = M2\pi \frac{n}{60} \quad (6)$$

where P_{out} is the mechanical output power in watts. This is the shaft-power relationship.

Hydraulic input power

Hydraulic input power was calculated from the measured volumetric flow rate and inlet pressure using Equation (7).

$$P_{hyd} = \frac{\dot{V}p10^5}{1000 \times 60} \quad (7)$$

where:

- P_{hyd} = hydraulic input power, W
- $\dot{V}$ = volumetric flow rate, L/min
- p = pressure, bar

Using:

$$\dot{V} = 32 \text{ L/min}$$

And:

$$p = 0.4 \text{ bar}$$

Gives:

$$P_{hy} = \frac{32 \times 0.4 \times 10^5}{1000 \times 60}$$

$$P_{hy} = 21.33 \text{ W}$$

Because pressure and flow were held constant, this value applies to every operating point.

Efficiency

The turbine efficiency was calculated using:

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

(8)

where η is expressed as a percentage.

The calculated value represents the overall hydraulic-to-shaft efficiency of the laboratory turbine arrangement rather than the hydraulic efficiency of the runner alone, because bearing friction, belt losses and other mechanical losses are included in the measured shaft output.

Representative worked calculation

Effective braking force

$$F = 12.0 - 4.50$$

$$F = 7.50 \text{ N}$$

Shaft Torque

$$M = 7.50 \times 0.025$$

$$M = 0.1875 \text{ Nm}$$

Angular velocity

$$\omega = 2\pi \frac{640}{60}$$

$$\omega = 67.02 \text{ rad/s}$$

Shaft power

$$P_{out} = 0.1875 \times 67.02$$

$$P_{out} = 12.57 \text{ W}$$

Efficiency

$$\eta = \frac{12.57}{21.33} \times 100$$

$$\eta = 58.9 \%$$

The same calculation procedure was applied to each experimental operating point, with the resulting values presented in table 1.

F_1 (N)	F_2 (N)	Speed (rev/min)	Braking Force (N)	Torque (Nm)	Shaft Power (W)	Hydraulic Power (W)	Efficiency (%)
1.0	0.50	1185	0.50	0.0125	1.55	21.33	7.3
2.0	0.75	1140	1.25	0.0313	3.73	21.33	17.5
3.0	1.00	1100	2.00	0.0500	5.76	21.33	27.0
4.0	1.25	1020	2.75	0.0688	7.34	21.33	34.4
5.0	1.50	975	3.50	0.0875	8.93	21.33	41.9
6.0	2.00	935	4.00	0.1000	9.79	21.33	45.9
7.0	2.25	885	4.75	0.1188	11.01	21.33	51.6
8.0	2.75	830	5.25	0.1313	11.41	21.33	53.5
9.0	3.00	775	6.00	0.1500	12.17	21.33	57.1
10.0	3.50	725	6.50	0.1625	12.34	21.33	57.8
11.0	4.00	685	7.00	0.1750	12.55	21.33	58.8
12.0	4.50	640	7.50	0.1875	12.57	21.33	58.9
13.0	5.00	600	8.00	0.2000	12.57	21.33	58.9
14.0	5.25	530	8.75	0.2188	12.14	21.33	56.9
15.0	5.50	465	9.50	0.2375	11.56	21.33	54.2
16.0	6.00	400	10.00	0.2500	10.47	21.33	49.1
17.0	6.25	310	10.75	0.2688	8.72	21.33	40.9
18.0	6.50	235	11.50	0.2875	7.08	21.33	33.2
19.0	7.00	150	12.00	0.3000	4.71	21.33	22.1
20.0	7.75	0	12.25	0.3063	0.00	21.33	0.0

Table 1. Experimental measurements and calculated Pelton turbine performance parameters at a constant pressure of 0.4 bar and flow rate of 32 L/min.

The calculated data show that increasing brake load produced a progressive increase in shaft torque while reducing rotational speed. The shaft output power and efficiency reached their highest measured values in the region between 600 and 640 rev/min. The detailed interpretation of these trends is presented in Sections 2.4 and 2.5.

2.4 Results and Turbine Characteristics

The calculated shaft-power values were plotted against rotational speed to determine the turbine's power-speed characteristic under constant hydraulic operating conditions.

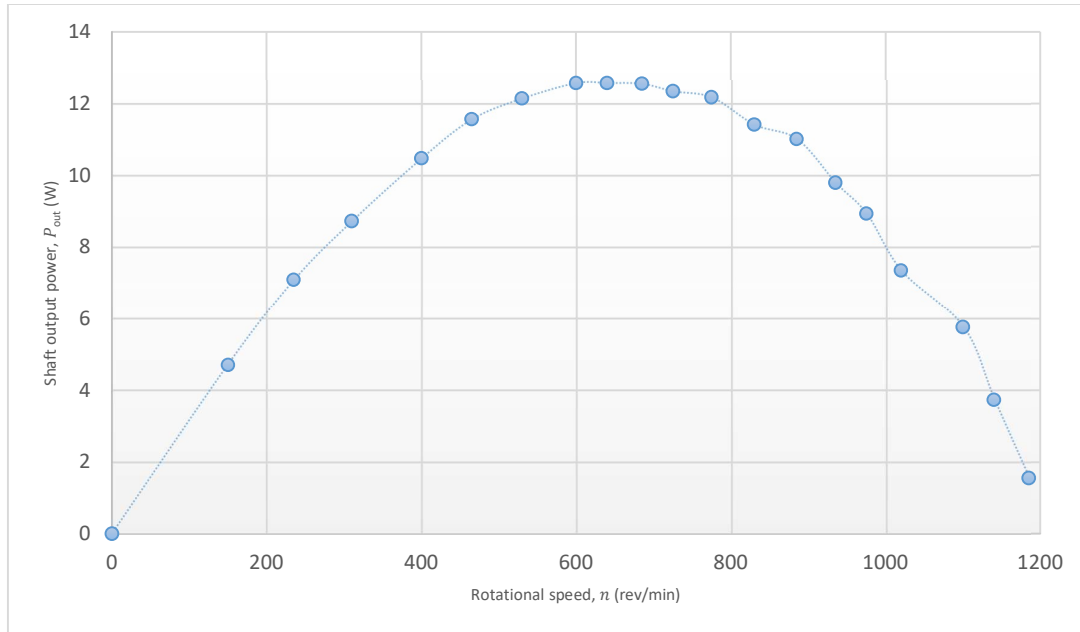


Figure 4. Shaft output power against rotational speed for the Pelton turbine at a constant inlet pressure of 0.4 bar and flow rate of 32 L/min.

The highest measured speed of 1185 rev/min produced a shaft output power of only 1.55 W because the applied braking force and torque were low. As the load increased, rotational speed decreased and torque increased. Shaft power rose progressively to a broad maximum of approximately 12.57 W at both 640 rev/min and 600 rev/min. Beyond this region, further loading reduced the shaft speed sufficiently for output power to decline. At the final stalled condition, both rotational speed and shaft power fell to zero.

The observed power-speed characteristic can be explained using the relationship between shaft power, torque and angular velocity, as shown in Equation (9).

$$P_{out} = M\omega \quad (9)$$

where M is shaft torque and ω is angular velocity.

At the lightest loading condition, angular velocity was high but the braking force and corresponding shaft torque were low, resulting in limited output power. As braking load increased, shaft torque increased more rapidly than angular velocity decreased, causing output power to rise. Within the maximum-power region, the combination of torque and angular velocity produced the highest measured shaft output. Beyond this region, further increases in torque were insufficient to compensate for the more rapid reduction in angular velocity, causing power to decline. At the stalled condition, the shaft torque reached its maximum measured value, but output power was zero because angular velocity was zero.

The highest measured shaft output power was approximately 12.57 W, corresponding to a calculated efficiency of 58.9%. These values occurred at rotational speeds of 600 and 640 rev/min, where the corresponding shaft torques were 0.2000 Nm and 0.1875 Nm respectively. The near-identical power and efficiency values at these adjacent operating points indicate that the turbine exhibited a relatively broad maximum-power region rather than a sharply defined single optimum speed. On the basis of the available load increments, the most effective measured operating region was therefore approximately 600-640 rev/min. However, the experimental spacing does not justify identifying one exact optimum speed within this range.

2.5 Critical Appraisal of Experimental Results

The measured power-speed characteristic followed the expected overall form, with shaft power increasing from the light-load condition to a broad maximum between approximately 600 and 640 rev/min before decreasing as the turbine approached stall. This supports the general validity of the experimental results. However, the peak was not sharply defined, with adjacent operating points producing almost identical power and efficiency values. The available load increments therefore identify a maximum-power region rather than one precise optimum operating speed.

A significant limitation encountered during testing was instability in the measured rotational speed. Rotational speed was monitored using a fixed laser tachometer arrangement shown in Figure 3; at several load conditions, the indicated speed fluctuated considerably rather than remaining at a single steady value. This introduced uncertainty into the calculated angular velocity and therefore the shaft-power values. The instability became more noticeable as the experiment progressed. Heat accumulated within the rig and changes in the behaviour of the belt and shaft arrangement were observed, with both rotational speed and Force Meter 2 readings becoming increasingly difficult to

stabilise. Although temperature was not measured directly, this may have altered the frictional behaviour of the mechanical loading system and reduced the repeatability of later measurements.

Further uncertainty arose because braking force was calculated from the difference between two Force Meter readings, meaning measurement error in either value propagated into torque and power. Reliability could be improved by using digital load cells and electronic speed logging, repeating each operating point to obtain mean values, and using smaller load increments around 600-640 rev/min. Shorter test periods or allowing the rig to stabilise between runs would also help reduce the influence of heat build-up. Overall, the experiment identified a credible maximum-power region, but the observed speed fluctuations and changing mechanical conditions limit the precision with which the optimum operating point can be defined.

3. Fluid-Flow Analysis

3.1 Jet Formation and Energy Conversion

The water approaching the Pelton nozzle possesses hydraulic energy associated primarily with pressure head. As the flow passes through the nozzle, this energy is converted predominantly into kinetic energy, producing a high-velocity free jet that strikes the Pelton runner. This behaviour is consistent with Bernoulli's energy relationship (White, 2016).

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 + h_L \quad (10)$$

where,

P = fluid pressure, Pa

ρ = fluid density, kg/m³

g = gravitational acceleration, m/s²

V = fluid velocity, m/s

z = elevation head, m

h_L represents head loss between the two locations

Point 1 = upstream / nozzle inlet

Point 2 = nozzle exit / free jet

For incompressible flow, continuity requires the volumetric flow rate to satisfy $Q = AV$, meaning that the reduced flow area at the nozzle exit produces an increase in jet velocity for a given flow rate. For real flow, however, viscous and frictional effects introduce losses, meaning that the actual jet velocity is lower than the ideal value predicted from the available head alone.

3.2 Jet-Bucket Interaction and Momentum Transfer

When the high-velocity jet reaches the Pelton runner, it strikes the central splitter of the bucket and is split into two approximately equal streams. These streams are redirected along the curved bucket surfaces before leaving the runner with a substantially changed direction of velocity. As the Pelton wheel operates as an impulse turbine, useful torque is produced primarily through this change in fluid momentum rather than through a pressure drop across the runner (Zhang, 2016).

The large change in direction of the water produces a corresponding change in fluid momentum. To redirect the flow, the bucket exerts a force on the water, while the water exerts an equal and opposite reaction force on the bucket. The component of this reaction force acting in the direction of rotation produces torque at the runner shaft. Consequently, greater effective turning of the jet increases the potential momentum transfer to the runner, although actual transfer is reduced by hydraulic losses and the residual velocity of the water leaving the bucket (Zhang, 2016).

Although the momentum-transfer principle provides a useful explanation of how the runner is driven, the actual flow within a Pelton bucket is considerably more complex. The jet spreads across the curved bucket surface, forming a rapidly changing free-surface flow that is three-dimensional and unsteady. Experimental and numerical investigations have shown that pressure and momentum transfer are not distributed uniformly across the bucket surface, with different regions contributing differently to the torque produced (Perrig et al., 2006). This means that bucket geometry has a direct influence on hydraulic performance, as the flow must be redirected effectively while minimising frictional losses, flow disturbance and residual kinetic energy at the outlet.

3.3 Boundary-Layer Behaviour, Separation and Hydraulic Losses

A boundary layer forms over the Pelton bucket as the water flows across the curved surface, due to the no-slip condition at the solid wall. The fluid velocity is effectively zero at the bucket surface and increases through the boundary layer towards the main flow velocity. This velocity gradient produces viscous shear stress, resulting in frictional losses as the water travels across the bucket surface (White, 2016).

The behaviour of the boundary layer is influenced by the Reynolds number, which represents the relationship between inertial and viscous forces within the flow:

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

(11)

where,

ρ = fluid density, kg/m³

V = characteristic flow velocity, m/s

L = characteristic length, m

μ = dynamic viscosity, Pa · s

A higher Reynolds number provides an indication of the relative importance of inertial and viscous forces and is commonly used when assessing whether boundary-layer behaviour is likely to be laminar, transitional or turbulent. However, the flow within a Pelton bucket is highly curved, three-dimensional and unsteady, meaning that simple transition limits developed for flat-plate flow cannot be applied directly to the runner.

As the water follows the curved bucket surface, changes in flow direction and local pressure can reduce the momentum of the fluid close to the wall. Under an adverse pressure gradient, the boundary layer may weaken and premature flow detachment can occur. If the water leaves the intended bucket surface too early, the jet is not redirected as effectively and less momentum is transferred to the runner (Perrig et al., 2006). In practice, viscous friction, surface roughness, premature flow detachment, secondary flow and residual outlet velocity therefore reduce the hydraulic energy available for useful shaft work. These hydraulic losses contribute to the difference between ideal turbine behaviour and the measured efficiency of the experimental Pelton wheel. However, the measured hydraulic-to-shaft efficiency also includes mechanical losses associated with the bearing, belt and loading system.

4. Codes and Standards

Appropriate industry standards are important in Pelton turbine applications because they provide recognised requirements for performance testing, measurement and acceptance. IEC 62006:2010 covers acceptance testing of small hydroelectric installations using impulse or reaction turbines and includes assessments of output, efficiency, operating characteristics, reliability and safety (IEC, 2010). For an industrial Pelton installation, these procedures provide a consistent basis for confirming whether the turbine meets specified performance requirements following installation. The pressure, flow rate, rotational speed and output power measurements used in the laboratory experiment reflect the same general performance parameters, although at a much smaller scale. Standardised acceptance testing also provides a consistent basis for verifying contractual performance claims, helping to ensure that measured output and efficiency can be compared against the specified design requirements.

IEC 60193:2019 covers model acceptance testing of hydraulic turbines, including impulse turbines, and specifies measurement methods, calculation procedures and comparison of model performance with prototype guarantees (IEC, 2019). This is particularly relevant during turbine development because model testing can be used to evaluate hydraulic performance before manufacture or commissioning of a full-scale machine. In Pelton turbine design, controlled model testing can therefore help assess jet-runner interaction, efficiency and operating characteristics before the final installation is produced. Model acceptance testing can therefore reduce technical and financial risk by identifying hydraulic performance issues before committing to the manufacture and installation of a full-scale turbine.

Safety requirements are equally important in full-scale hydroelectric machinery. ISO 12100:2010 establishes general principles for machinery risk assessment and risk reduction, requiring hazards to be identified and controlled through design measures, safeguarding and information for safe use (ISO, 2010). Applied to a Pelton turbine installation, relevant hazards include exposed rotating shafts, couplings, generators, high-pressure water systems, mechanical failure and maintenance access. Appropriate guarding, isolation procedures and risk-control measures are therefore required alongside hydraulic performance considerations. The standard also reinforces that risk reduction should be considered during the design stage, so hazards are eliminated or reduced through inherently safe design and guarding before relying on operational procedures alone.

Taken together, IEC 62006, IEC 60193 and ISO 12100 address different stages of the turbine lifecycle, from model validation and performance acceptance to operational safety. Their combined application therefore provides complementary control over hydraulic performance, contractual compliance and machinery risk, helping to ensure that Pelton turbine systems are both technically effective and safely implemented.

References

American Society of Mechanical Engineers (1991). *Pelton water wheel collection*. New York: American Society of Mechanical Engineers, pp.2–4.

ANDRITZ (2019). Cutting Edge – Manufacturing for a Demanding Market. *HYDRO News Europe*, [online] pp.20–21. Available at: <https://www.andritz.com/hydro-en/hydronews/hn-europe/manufacturing> [Accessed 26 July 2026].

Ferreño, D., Álvarez, J.A., Ruiz, E., Méndez, D., Rodríguez, L. and Hernández, D. (2011). Failure analysis of a Pelton turbine manufactured in soft martensitic stainless-steel casting. *Engineering Failure Analysis*, 18(1), pp.256–270. doi:10.1016/j.engfailanal.2010.09.005.

Ge, X., Sun, J., Chu, D., Liu, J., Zhou, Y., Zhang, H., Zhang, L., Chen, H., Kan, K., Binama, M. and Zheng, Y. (2023). Sediment erosion on Pelton turbines: A review. *Chinese Journal of Mechanical Engineering*, 36, article 64. doi:10.1186/s10033-023-00880-y.

International Electrotechnical Commission (2010). *IEC 62006:2010 hydraulic machines – Acceptance tests of small hydroelectric installations*. Geneva: International Electrotechnical Commission.

International Electrotechnical Commission (2019). *IEC 60193:2019 hydraulic turbines, storage pumps and pump-turbines – model acceptance tests*. Geneva: International Electrotechnical Commission.

International Organization for Standardization (2010). *ISO 12100:2010 safety of machinery – general principles for design – risk assessment and risk reduction*. Geneva: International Organization for Standardization.

Perrig, A., Avellan, F., Kueny, J.-L., Farhat, M. and Parkinson, E. (2006). Flow in a Pelton turbine bucket: Numerical and experimental investigations. *Journal of Fluids Engineering*, 128(2), pp. 350–358. doi: 10.1115/1.2170120.

Shrivastava, N. and Rai, A.K. (2025). Hydro-abrasive erosion in Pelton turbines: Comprehensive review and future outlook. *Renewable and Sustainable Energy Reviews*, 207, article 114957. doi:10.1016/j.rser.2024.114957.

Sun, Q. and Tan, L. (2024). Bucket design method and performance optimization of a Pelton turbine. *Journal of Tsinghua University (Science and Technology)*, 64(5), pp.852–859. doi:10.16511/j.cnki.qhdxxb.2024.26.004.

White, F.M. (2016). *Fluid mechanics*. 8th edn. New York: McGraw-Hill Education.

Zhang, Z (2016). *Pelton turbines*. 1st edn. Cham: Springer International Publishing.