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Fluid Flow: Pelton Wheel

Structural & Fluid Analysis (ACMA6010)

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2316051

Word Count 3201

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1 Introduction

1.1 Background to Impulse Turbines and the Pelton Wheel

Hydraulic turbines are either reaction or impulse machines. In a reaction turbine such as a Francis or Kaplan the runner runs full of water and the pressure drops across it. In an impulse turbine the whole pressure head becomes velocity in a nozzle before the water reaches the runner, which then spins in air at atmospheric pressure (Douglas et al., 2011). The Pelton wheel is the main impulse machine and suits high head, low flow sites. A fast jet strikes a row of double cup buckets, each split by a sharp central ridge, and the change in momentum as the buckets turn the water is what drives the wheel. Figure 1 shows the laboratory Pelton wheel at UWTSD.

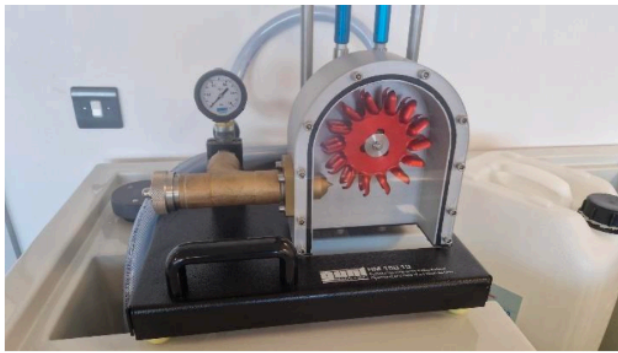


Figure 1: The Pelton wheel in the Engineering Science lab at UWTSD (from the assignment brief).

1.2 Aim, Objectives and Scope

The aim is to demonstrate a systematic understanding of fluid analysis through the application of a Pelton wheel to energy generation. Four objectives follow: to review the historical, current and future development of Pelton wheels, covering materials, manufacturing and quality control; to determine the turbine characteristics by experiment and plot speed against power at constant head; to describe the fluid flow around the runner using boundary layer and flow separation theory; and to identify the codes of practice and standards that apply. The scope is a single jet laboratory unit under steady conditions, analysed with one dimensional momentum theory. Computer modelling is outside the scope.

2 Historical, Current and Future Developments

2.1 Historical Development

The Pelton wheel grew out of the flat vane wheels used in the Californian gold fields in the nineteenth century. These carried a single row of cups which the jet struck centrally. Water hitting the middle of a cup splashed back against the next bucket and slowed the wheel, and the problem worsened as head increased (ASCE, no date), holding efficiency at 30 to 40 per cent (ASME, 2012). Lester Allan Pelton, a millwright at Camptonville, California, tried many bucket shapes before settling on the split bucket (Invention & Technology, no date), and received US Patent 223,692 for the splitter principle in 1880 (ASME, no date). The change was in the shape, not the material. A central ridge divides the jet into two halves, turned outwards through the curved bucket and thrown clear, so the leaving water no longer obstructs the incoming jet. Efficiency rose to as much as 90 per cent, and within fifteen years 850 companies were using the design (ASME, 2012; ASCE, no date). Doble later added the needle nozzle, which throttles flow without changing jet velocity (ASME, no date).

2.2 Current Applications and Technology

Pelton turbines sit at the high head, low flow end of the hydropower market. In the UK, Gilbert Gilkes and Gordon of Kendal has built turbines since 1856 and supplied more than 6,800 machines, covering 50 kW to 20 MW per unit (Hydropower and Dams International, 2025). At the top end, the Bieudron plant in Switzerland runs three vertical Pelton turbines rated at 423 MW each from a gross drop of 1,883 m (Grande Dixence SA, no date). Multi jet layouts keep efficiency high over a wide range of flows, suiting the machine to peaking duty (Gilkes, no date).

2.3 Future Developments

Development is going in three directions. The first is recovering energy inside existing water networks, where pressure reducing valves burn off spare head as heat and a turbine could recover it while still controlling downstream pressure. Gallagher et al. (2015) assessed sites across the UK and Irish water industry and shortlisted locations with a potential of 17.9 GWh a year, finding up to 40 per cent of the gross power at an existing valve could be recovered. The second is computational fluid dynamics: Židonis et al. (2015) parameterised two runner designs and optimised them in ANSYS Fluent and CFX, an approach only recently practical as CFD models of this flow have become accurate enough to trust. The third is additive manufacturing, so far used more for tooling than finished parts, with one manufacturer 3D

printing a complete mould set for a 3,200 kg runner in 72 hours (Additive Manufacturing, 2019).

2.4 Materials

Early wheels were bronze or cast iron. Modern runners are nearly always a low carbon martensitic stainless steel of the 13Cr–4Ni family, cast as ASTM grade CA6NM or forged as F6NM. The material has to resist fatigue, cavitation and sand erosion at once, and 13Cr–4Ni is the reference material the erosion model in IEC 62364 is calibrated against (IEC, 2019b; Rai et al., 2020). Fatigue drives the design, because each bucket is struck by the jet once per revolution, loading the bucket root in cycles. On sediment heavy rivers, erosion takes over: a field study on a high head Himalayan plant measured an average loss of splitter height of 3 per cent of the bucket width in one season (Rai et al., 2017). Material factors for the IEC 62364 model give 1.0 for chrome nickel steels and 0.1 for sprayed tungsten carbide coatings, a tenfold gain over the bare steel (Rai et al., 2020).

2.5 Manufacturing Processes

There are three manufacturing routes. Casting the runner in one piece is cheapest but risks porosity and inclusions in the highly stressed bucket root. Concern over the reliability of fully cast runners pushed at least one major manufacturer to develop alternatives, since a crack at the root can grow until the bucket breaks off, and a bucket let go at speed carries a great deal of energy (Renewable Energy World, 2021). The second route machines the whole runner from a single forged block on multi axis CNC machines (CNC TVAR, no date), removing welds and giving uniform properties, but is limited by the largest forging available. The third is welded build up, where buckets are welded onto a forged centre disc, then heat treated, machined and ground to shape (Renewable Energy World, 2021).

2.6 Quality Control and Assurance

The consequence of a missed defect is a bucket coming off at speed, so inspection is a safety matter rather than a formality. Material certificates and heat treatment records prove where the steel came from, and a finished runner is commonly checked by four separate NDT methods (CNC TVAR, no date). Penetrant and magnetic particle testing find cracks at the surface, while ultrasonic and radiographic testing find defects buried in the casting or weld. Geometry is checked against the design model, and Gilkes for example laser scan their runners after machining (Waterpower Magazine, 2024). Surface finish matters twice

over: it sets the friction loss over the bucket, and a rough surface gives erosion more places to start (CNC TVAR, no date).

3 Experimental Determination of Turbine Characteristics

3.1 Apparatus

The turbine tested was the laboratory Pelton wheel unit used with the HM 150 hydraulics bench. Water is delivered through a needle nozzle producing a single jet, with the jet pressure shown on a manometer. The braking device is a belt running over a 50 mm diameter pulley on the turbine shaft, with each end attached to a spring balance on a traverse. Turning a hand wheel tensions the belt, loading the wheel (GUNT, no date). The braking force is the difference between the two readings, $F = F_1 - F_2$, and shaft speed N was read from a tachometer.

3.2 Experimental Procedure

The constant head characteristic followed the handout sequence. The nozzle was fully opened, the inlet opened and the pump started, then the nozzle adjusted until the gauge read 0.4 bar, fixing the head. The outlet of the volumetric tank was closed and the time to collect a known volume measured to give the flow rate (GUNT, no date). The outlet was reopened and the brake tightened step by step, with F_1 , F_2 and N recorded at each step.

3.3 Own Results, Observations and Data Provenance

The flow measurement was completed: ten litres were collected in 17 seconds, giving 35.3 L/min, although the flow and head boxes on the printed sheet were left blank at the time. Loading readings were then taken up to 13 N on force meter 1 and are given in Table 1. These were unreliable for two reasons. The spring balance needles never settled, because vibration from the runner and the grip and slip of the belt fed straight into them, and the indicated speed wandered over a wide band. The session was then cut short when a member of the group became unwell, so the remaining load steps and repeat runs were not carried out.

Step	Force meter 1, F_1 (N)	Force Meter 2, F_2 (N)	Speed, N (rev/min)	Braking Force, $F=F_1-F_2$ (N)	Torque on Shaft, M (Nm)	Power on Shaft, P (W)	Speed ratio vs provided
1	1	0.3	3450	0.700	0.018	6.32	2.91
2	2	0.5	3335	1.500	0.038	13.10	2.93
3	3	0.7	3090	2.300	0.058	18.61	2.81

4	4	1	2935	3.000	0.075	23.05	2.88
5	5	1.1	2850	3.900	0.098	29.10	2.92
6	6	1.3	2680	4.700	0.118	32.98	2.87
7	7	1.6	2540	5.400	0.135	35.91	2.87
8	8	2.1	2390	5.900	0.148	36.92	2.88
9	9	2.5	2230	6.500	0.163	37.95	2.88
10	10	3	2100	7.000	0.175	38.48	2.90
11	11	3.4	2010	7.600	0.190	39.99	2.93
12	12	3.7	1710	8.300	0.208	37.16	2.67
13	13	4.1	1410	8.900	0.223	32.85	2.35

Table 1: Group's own readings, terminated at $F_1 = 13 \text{ N}$. Torque and implied power from $M = F \cdot D/2$ ($D = 50 \text{ mm}$) and $P = M \cdot 2\pi N/60$. The final column gives the ratio of each recorded speed to the provided dataset at the same brake setting.

Working the values through afterwards shows the problem went beyond unsteady needles. At the same brake settings, the recorded speeds are close to three times those in the provided dataset, and the ratio holds nearly constant, between 2.8 and 2.9, over the first eleven rows. A constant ratio points to a systematic fault rather than random scatter. The shaft power implied reaches about 40 W, far beyond the 14 W full scale of the handout graph, which was drawn for this rig, so the readings are not physically believable. The most likely cause is the speed measurement, since a tachometer counting more than one mark per revolution reads high by a whole number factor. The shape of the curve is still right, as Figure 2 shows, because a scale error stretches a curve without changing its form.

The group was therefore instructed to base the loading analysis on a complete dataset provided by the module lecturer, Dr K. Lake, keeping our own readings for comparison (Lake, 2026). The flow measurement is treated differently: it was made with a tank and a stopwatch, independently of the tachometer, so the fault above does not affect it, and our measured 35.3 L/min is used below. Table 1 and the upper curve in Figure 2 are our own data; Table 2 and Figure 3 come from the provided dataset.

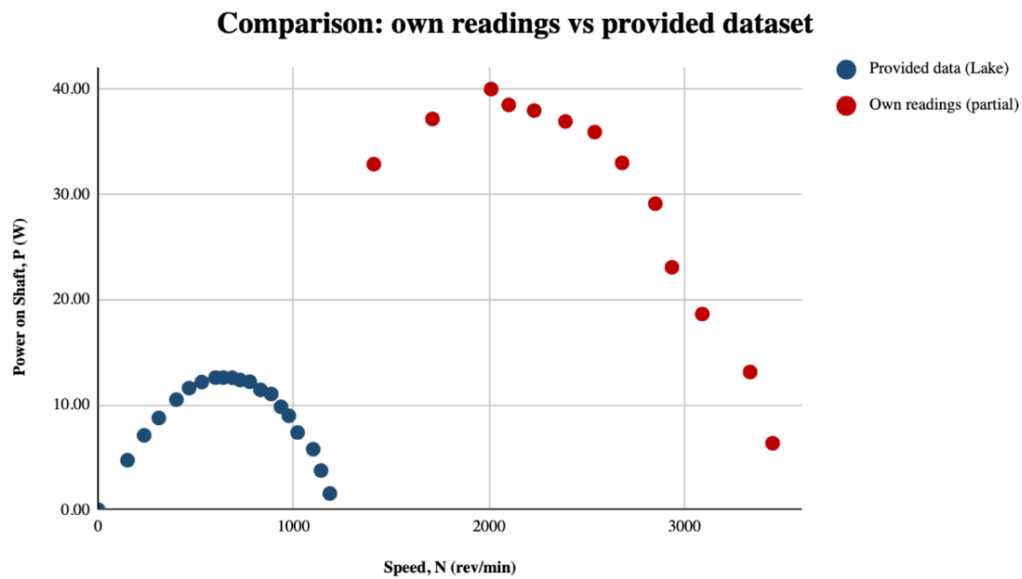


Figure 2: Comparison of the group's own partial readings with the provided dataset, on common axes. The group's curve shows the expected form at roughly three times the credible scale.

3.4 Results

For each row the braking force, torque and shaft power were calculated from $F = F_1 - F_2$, $M = F \cdot D/2$ with $D = 50$ mm, and $P_{out} = M \cdot 2\pi N/60$ (GUNT, no date). Table 2 gives the full set, and the characteristic is plotted in Figure 3 on the axes of the blank handout graph. Power rises from 1.6 W at 1,185 rev/min, peaks at about 12.6 W between 600 and 685 rev/min, and falls to zero at stall.

Step	Force meter 1, F_1 (N)	Force Meter 2, F_2 (N)	Speed, N (rev/min)	Braking Force, $F=F_1-F_2$ (N)	Torque on Shaft, M (Nm)	Power on Shaft, P (W)	Hydraulic Power (W)	Efficiency (%)
1	1	0.5	1185	0.500	0.013	1.55	23.53	6.6
2	2	0.75	1140	1.250	0.031	3.73	23.53	15.9
3	3	1	1100	2.000	0.050	5.76	23.53	24.5
4	4	1.25	1020	2.750	0.069	7.34	23.53	31.2
5	5	1.5	975	3.500	0.088	8.93	23.53	38.0
6	6	2	935	4.000	0.100	9.79	23.53	41.6
7	7	2.25	885	4.750	0.119	11.01	23.53	46.8
8	8	2.75	830	5.250	0.131	11.41	23.53	48.5
9	9	3	775	6.000	0.150	12.17	23.53	51.7
10	10	3.5	725	6.500	0.163	12.34	23.53	52.4

11	11	4	685	7.000	0.175	12.55	23.53	53.3
12	12	4.5	640	7.500	0.188	12.57	23.53	53.4
13	13	5	600	8.000	0.200	12.57	23.53	53.4
14	14	5.25	530	8.750	0.219	12.14	23.53	51.6
15	15	5.5	465	9.500	0.238	11.56	23.53	49.1
16	16	6	400	10.000	0.250	10.47	23.53	44.5
17	17	6.25	310	10.750	0.269	8.72	23.53	37.1
18	18	6.5	235	11.500	0.288	7.08	23.53	30.1
19	19	7	150	12.000	0.300	4.71	23.53	20.0
20	20	7.75	0	12.250	0.306	0.00	23.53	0.0

Table 2: Calculated turbine characteristics at constant head (0.4 bar), from the dataset provided by Dr K. Lake (2026). $M = F \cdot D/2$ ($D = 50$ mm); $P_{out} = M \cdot 2\pi N/60$; hydraulic power 23.5 W throughout (35.3 L/min at 0.4 bar); $\eta = P_{out}/P_{hyd}$.

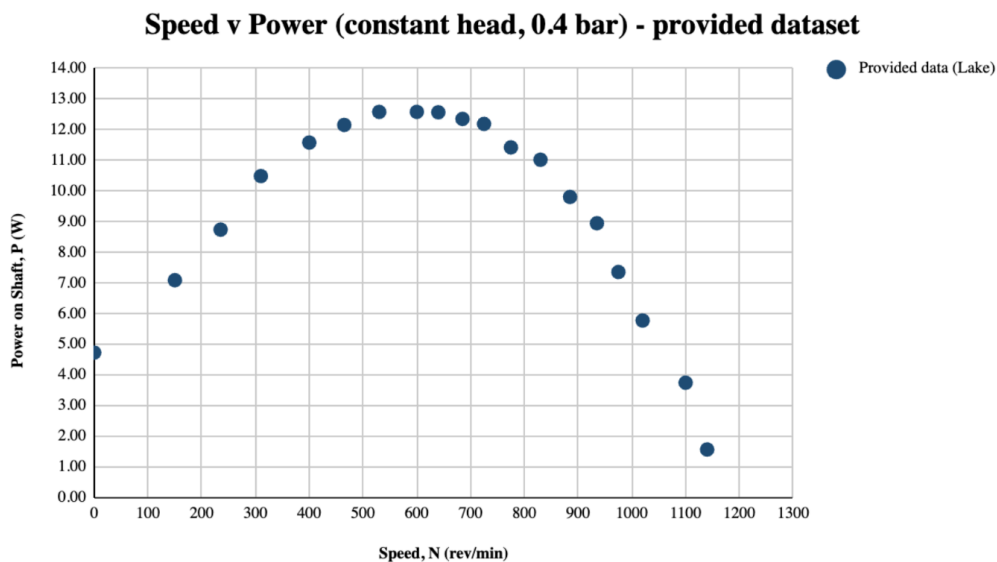


Figure 3: Speed against shaft power at constant head (0.4 bar), from the provided dataset. Axes matched to the handout graph.

The hydraulic power follows from the measured flow and set pressure, using $P_{hyd} = \dot{V} \cdot p \cdot 10^5 / (1000 \cdot 60)$ with $\dot{V} = 35.3$ L/min and $p = 0.4$ bar, giving 23.5 W (GUNT, no date). This is constant across the run, since nozzle and head were fixed and only the brake load varied. Efficiency is then $\eta = P_{out}/P_{hyd}$, in the final column of Table 2, peaking at 53.4 per cent between 600 and 640 rev/min. As a check, 0.4 bar is a head of about 4.08 m and an ideal jet

velocity of about 8.94 m/s, and the kinetic energy carried by a 35.3 L/min jet at that speed also comes to 23.5 W, so the hydraulic power is consistent by two independent routes.

3.5 Critical Appraisal of Results

The characteristic has the shape expected of an impulse turbine. Useful power is zero at both ends, at runaway where the wheel spins fastest under no load and at stall where torque is greatest but nothing moves (Douglas et al., 2011). The peak sits at about 640 rev/min, close to half the highest recorded speed of 1,185 rev/min, matching the standard result that maximum power comes near half runaway speed. It also sits just under the 14 W full scale of the handout graph, which is the check our own readings did not pass.

The peak efficiency of 53.4 per cent is worth looking at, since large Pelton installations reach around 90 per cent (Douglas et al., 2011). Most of the gap is scale. On a small wheel the boundary layer is thicker relative to the bucket, roughness is larger relative to the water sheet, and bearing and brake losses take a bigger share of a much smaller power. The figure is also taken after the belt brake, so it includes every mechanical loss in the drive. It is a system efficiency, and the runner alone is doing better than it suggests.

Timing a tank fill by hand over 17 seconds is open to reaction time at each end, and half a second is around three per cent, so the peak is better read as roughly 53 per cent than 53.4.

The instrument behaviour also has a physical explanation. A band brake carries load through small grips and slips rather than steadily, and with runner vibration this puts a fluctuating force into a spring balance that cannot average it out. Both effects are worst at high speed and light load, exactly where readings were hardest to take. Using provided loading data brings its own limitations: the results no longer describe this particular rig, and with one run at one head, repeatability cannot be assessed. A proper characterisation would need repeat runs and the uncertainty analysis set out in IEC 60193.

4 Fluid Flow Around the Pelton Wheel

4.1 The Nozzle and the Jet

The water arrives at the nozzle under pressure and leaves as a fast open jet. Inside the nozzle the passage narrows, the flow speeds up and the pressure falls, so pressure energy is traded for kinetic energy. Because the pressure falls in the direction of flow, the boundary layer on the nozzle wall stays thin and attached and the loss is small. This is expressed as a velocity coefficient C_v of about 0.97 to 0.99, so the real jet is only slightly slower than the

ideal $\sqrt{2gH}$ (Douglas et al., 2011). Once the jet leaves the nozzle it is at atmospheric pressure everywhere, so from there the machine works only on the kinetic energy of the water. That is what makes it an impulse turbine, and why the casing does not need to hold pressure.

4.2 The Boundary Layer on the Bucket

The jet meets the splitter ridge, divides in two, and each half spreads into a thin sheet running around the inside of its half of the bucket. From the moment the water touches the metal a boundary layer starts to grow. Water in contact with the surface is held back by friction, the no slip condition, and that slowing spreads a short distance up into the sheet (Douglas et al., 2011). At the speeds involved the Reynolds number is high and the boundary layer over most of the bucket is turbulent. Friction in this layer takes energy out of the sheet, so the water leaves the bucket slower, relative to the bucket, than it arrived. In the textbook analysis this is carried as a friction factor k on the relative velocity, typically 0.8 to 0.9 (Douglas et al., 2011).

4.3 Flow Separation and Other Losses

The ideal bucket would turn the jet through a full 180 degrees, giving the largest change of momentum and so the largest force. Real buckets turn the flow through about 165 degrees so the leaving sheet clears the back of the next bucket (Douglas et al., 2011). Gilkes describe their buckets as turning the jet through nearly 180 degrees (Gilkes, no date), which is true of the metal but not of the water, and the difference is a deliberate compromise between momentum change and clean discharge.

Separation sets the limit on how hard the water can be turned. Near the exit lip the surface curves away from the flow and the sheet is asked to slow against a rising pressure. If the boundary layer cannot push through that adverse pressure gradient it lifts off the surface, the sheet leaves early, the real deflection angle drops, and the force on the bucket drops with it. Roughness makes this worse, since a rough surface thickens the boundary layer and brings separation on sooner. That is the fluid mechanics reason the buckets are ground and polished, and why blunting of the splitter edge by erosion costs efficiency.

4.4 Critical Appraisal

The analysis above, and the equations behind the laboratory work, treat the flow as steady and one dimensional, with a single bucket held square in the jet. The real flow is messier. The buckets swing through the jet in turn, so each sees the jet arrive and leave once per revolution and the flow is unsteady throughout. The sheet on the bucket has a free surface, the casing is full of spray, and the wheel churns through that spray, a windage loss the ideal

analysis ignores. On top of the fluid losses sit bearing friction and, here, the belt brake. The efficiency measured at the pulley therefore sits well below the ideal, about 53 per cent at best, and each part of the gap can be put down to something specific: nozzle friction, bucket surface friction, separation at the exit, windage, and mechanical loss. Where the simple theory runs out, CFD takes over, since it can carry the free surface and the interaction between buckets (Židonis et al., 2015).

5 Codes of Practice and Industry Standards

5.1 Performance and Acceptance Testing

How turbine performance is proven is set out in two IEC standards. IEC 60193:2019 covers model acceptance tests, in which a scale model is tested in a laboratory to show guaranteed performance has been met (IEC, 2019a). It matters here because it defines what makes a laboratory measurement count as evidence: calibrated instruments, steady operating conditions and a formal treatment of uncertainty. It applies to machines above 5 MW or 3 m runner diameter, so the UWTSD rig sits outside its scope, but the practice it describes is the benchmark, and the shortcomings in Section 3.5 are departures from it. Field acceptance testing of an installed machine is governed by IEC 60041:1991 (IEC, 1991).

5.2 Degradation in Service

Erosion and cavitation each have their own standard. IEC 62364:2019 gives the guidance for sand erosion in Kaplan, Francis and Pelton turbines and supplies the erosion model Rai *et al.* (2020) extended for Pelton parts (IEC, 2019b). BS EN 60609-2:1999 deals with cavitation pitting in Pelton turbines and provides the basis for contract guarantees limiting pitting over a set period (BSI, 1999). Both turn a wear mechanism into something a buyer can specify and measure, which lets a runner be sold with a guaranteed life.

5.3 Materials, Manufacture and Quality

The cast martensitic steels in Section 2.4 are specified in the UK under BS EN 10213, which lists the 13Cr–4Ni grades with their composition, heat treatment and mechanical properties, and calls up the ultrasonic, radiographic and penetrant testing standards, with material certificates issued to EN 10204 (BSI, 2016). That gives the quality control in Section 2.6 a written basis rather than leaving it to the manufacturer. Above component level, ISO 9001:2015 sets out the quality management system within which that inspection is planned and audited (ISO, 2015).

6 Conclusions

The Pelton wheel is a mature machine, and its development has been driven by one idea, splitting the jet so the leaving water does not fight the incoming jet.

The experimental work produced the classic impulse characteristic from the provided dataset, with power peaking at about 12.6 W near half the runaway speed and a system efficiency of 53.4 per cent against a hydraulic power of 23.5 W. Our own partial readings showed the right shape at three times the believable scale, identifying a systematic speed measurement fault and justifying use of the provided data.

The flow around the bucket is governed by the boundary layer. Friction and separation set the gap between the ideal 180 degree deflection and the real machine, and explain both why bucket finish matters as much as bucket shape and why a small wheel cannot match the 90 per cent efficiency of a large one. The standards framework exists mainly to stop one thing: a bucket coming off at speed.

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