Experimental investigation and analysis of the spear valve design on the performance of Pelton turbines: 3 case studies

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Abstract

The impact of the nozzle and spear valve configuration on the performance of a Pelton turbine is investigated both experimentally and computationally. A previously published computational fluid dynamics (CFD) study has shown that injectors with noticeably steeper nozzle and spear angles, 110° and 70° respectively, attain a higher efficiency than the industry standard 80° and 55°.

As a result, three injector design cases were manufactured for experimental testing. Two of those cases were the standard (80/55) design, with nozzle and spear tip angles of 80° and 55° and the Novel 1 design (110/70) with nozzle and spear tip angles of 110° and 70° based on previously published CFD optimisation studies. These studies showed that increasing the nozzle and spear angles to the upper limit of the investigated test plan gave higher efficiencies. The response surfaces suggested that the optimum nozzle and spear angles could be even steeper. That is why, an additional case, a third design (Novel 2) with even steeper angles (150/90) was also manufactured and tested.

The experimental tests were carried out in a single jet operation using the upper injector on the Gilkes Pelton runner with series Z120 buckets. The results show that two novel injector design cases produce higher efficiencies than the standard design, when tested with a Pelton runner. An important gain of about 1% in efficiency is achieved at the Best Efficiency Point of the turbine. Furthermore, the improvement is even more pronounced at lower flow rates, where the spear valve opening is smaller and the geometry of the injector has even larger effect.

To discuss and analyse these experimental observations, a further 2D axisymmetric CFD analysis is performed. This analysis shows a similar trend to the experimental results. The CFD results show that the largest amount of energy is lost at the region upstream of the nozzle exit, where the static pressure is converted into the dynamic pressure. This conversion starts earlier in case 1, the Standard injector design, at about twice the distance compared to the Novel designs, cases 2 and 3. Consequently, the flow must travel in this region at an increased velocity and it is shown that this region is longer in the Standard injector. Hence, its friction losses are higher.

However, the differences between the designs calculated in CFD are about a factor of 2 lower than the experimental results, indicating that the 3D secondary flow mechanisms arising from the geometry upstream of the nozzle and spear tip also affect the performance of the spear valve and the Pelton runner. The mismatch between the efficiency increase magnitude observed experimentally and modelled using the axisymmetric case suggests that the steeper angle injectors cope better with secondary velocities in the flow.

1. Introduction

Pelton impulse turbines [1, 2] that were invented by Lester Pelton [3], are the hydro turbines used for the high head applications. These turbines produce power by utilising the momentum of a high speed water jet impinging peripherally on rotating buckets. The injector utilises a fixed nozzle and adjustable spear allowing control of the flow rate.

Due to the high complexity and unsteadiness of the flow during the jet-runner interaction, the development of impulse turbines is difficult and requires sophisticated equipment for flow observation [4-7] or high cost computational resources for accurate numerical analysis [8-16]. Unlike the reaction turbines (e.g. Propeller, Kaplan or Francis) that have been developed using Computational Fluid Dynamics (CFD) for more than 20 years now [17], the Pelton turbines were mostly developed by trial and error approach [18, 19]. There are only few studies available where a selection of injector designs are modelled and compared using CFD [20-24] or using a visual analysis [25-27]. Therefore, there is room for improvement of Pelton turbines. The recent increase in computational resources and improvements in the numerical codes makes it possible to fill this gap.

This paper provides experimental results comparing three injector designs where different nozzle and spear tip angle configurations are used. One configuration is a standard nozzle and spear combination found in many current designs and in the available literature. This design is tested and used as a datum for the comparison. Two other configurations are novel designs based on recent CFD studies [23, 24]. The study is taken further using CFD simulations of the injectors to understand why both novel designs are more efficient than the standard design as shown experimentally. 2D axisymmetric simulations of the injector are performed to analyse the effect of nozzle and spear configuration on the flow patterns and formation of the free jet.

1.1 Injector designs

Three injector designs were manufactured by Gilbert Gilkes & Gordon Ltd for experimental testing using the Pelton test rig at the Laboratory for Hydraulic Machines, National Technical University of Athens (NTUA). The injector designs tested were the Standard (80/55) design, with nozzle and spear tip angles of 80° and 55° and the Novel 1 design (110/70) with nozzle and spear tip angles of 110° and 70° based on previous CFD optimisation studies [23, 24]. These optimisation studies [23, 24] showed that increasing the nozzle and spear angles to the upper limit of the test plan, which was much higher than current industry guidelines, gave higher efficiencies. The response surfaces in this study suggested that the optimum nozzle and spear angles may be even steeper. That is why, an additional case, a third design (Novel 2) with even steeper angles (150/90) was also manufactured and tested.
The details of the injectors used in this study are tabulated in Table 1 and shown in Fig. 1 and Fig. 2.

	Notation	Nozzle angle [°]	Spear tip angle [°]
Standard	80/55	80	55
Novel 1	110/70	110	70
Novel 2	150/90	150	90

Table. 1. Details of the injector configurations

Fig. 1. Pelton spears used for experimental tests

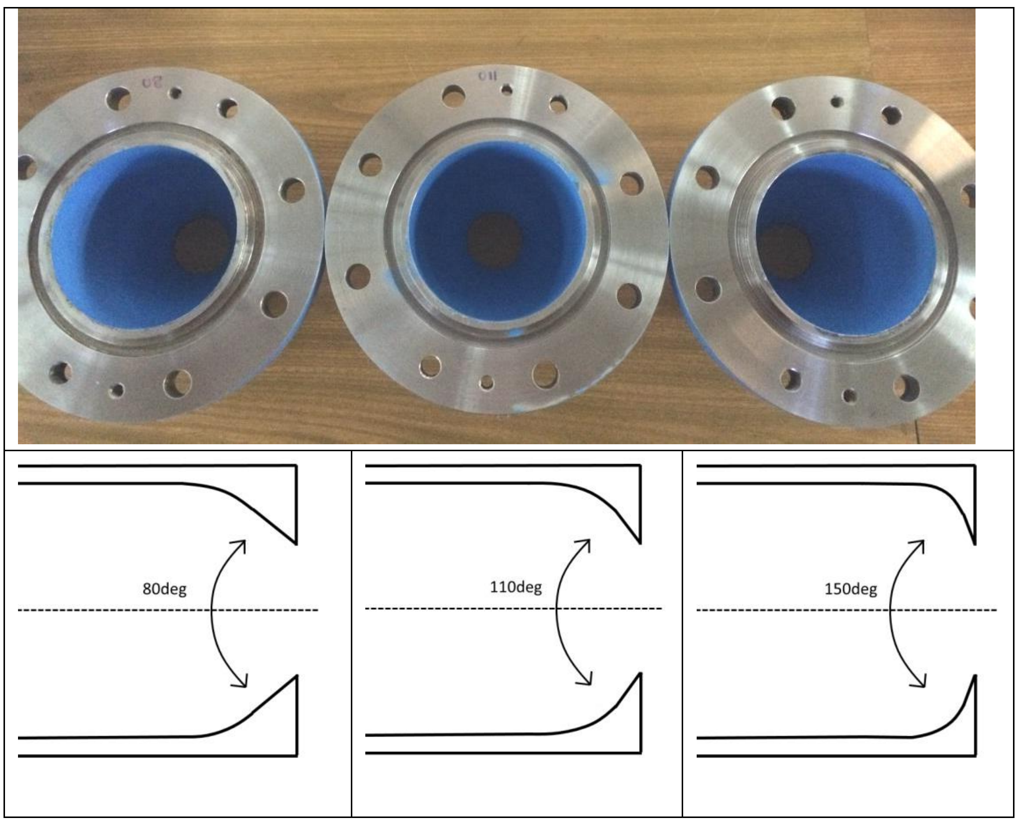

Fig. 2. Pelton nozzles used for experimental testing

2. Experimental testing

2.1 Pelton test rig

The first injector tests were carried out using the Pelton test rig at the Laboratory of Hydraulic Turbomachinery, NTUA. A high head adjustable speed multistage pump of nominal operation point Q=290 m^3/h, H=130 m, coupled via a hydraulic coupler to a 200 kW induction motor was used to feed the model turbine, pumping from the 320 m^3 main reservoir of the Lab. The tests were carried out using the Gilkes Z120 twin jet Pelton runner, which was coupled to a 75kW DC generator with continuous speed regulation. Fig. 3 shows the Pelton test rig in operation.

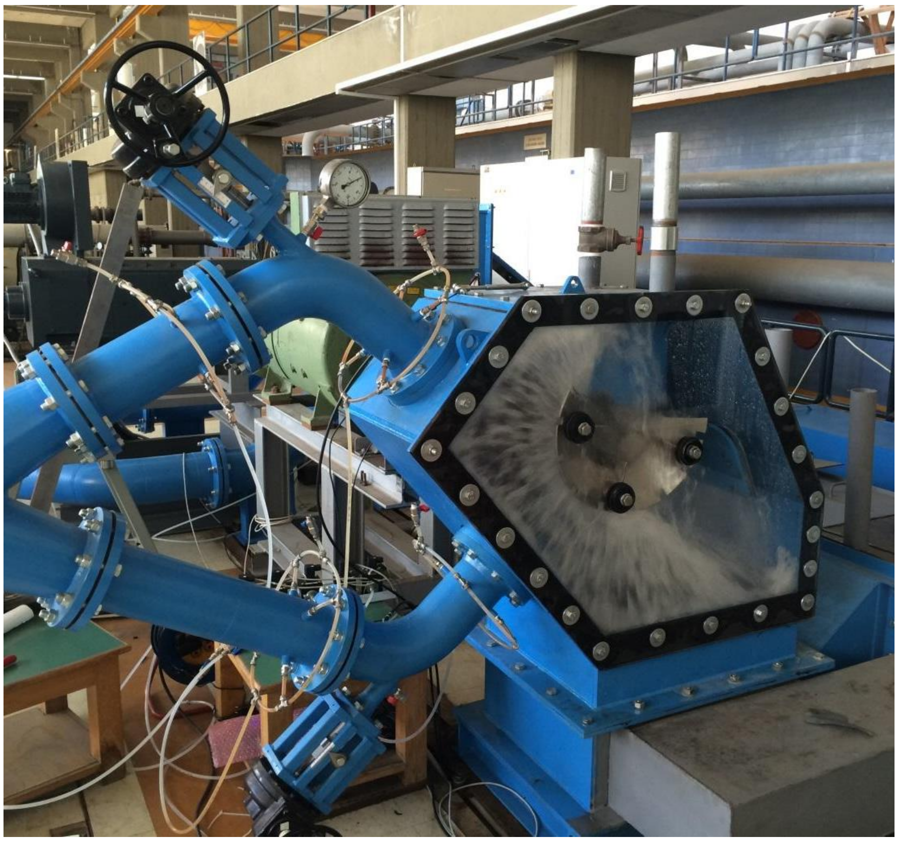

Fig. 3. Gilkes twin jet Pelton test rig in operation at NTUA

Testing and calibration of all the sensors was carried out according to the testing standard *IEC 60193 Hydraulic turbines, storage pumps and pump-turbines – Model acceptance tests (IEC 60193:1999)*.

The testing was carried out in single jet operation using the upper jet only. The characteristic equations of turbine unit speed n_{11} and unit flow Q_{11k} used to define the operation and performance of the turbine, are given below. The pitch circle diameter, D, of the Pelton runner used in this testing was 320 mm and the bucket width, B, was 120 mm.

$$n_{11} = \frac{n}{D^2 \times \sqrt{H}} \tag{1}$$

$$Q_{11k} = \frac{Q/N_j}{B^2 \times \sqrt{H}} \tag{2}$$

$$\eta = \frac{P_{out}}{P_{in}} \quad (3)$$

$$P_{out} = M\omega \quad (4)$$

$$P_{in} = \rho g H Q \quad (5)$$

Where n_{11} is the unit speed, n is the rotational speed of the runner, H is the net head, Q is the flow rate, N_j is the number of jets, M is the torque measured on the turbine shaft, ρ is the density of water and g is the acceleration due to gravity. Values for ρ and g were calculated according to the tables provided in the testing standards (IEC60193:1999).

The systematic uncertainty for each instrument used during this testing is given in Table 2, below. The total systematic uncertainty in the efficiency, η, was calculated as ±1.0%.

Instrument	Systematic Uncertainty
Pressure Transducer (*H*)	±0.1%
Flow meter (*Q*)	±1.0%
Torque meter (*M*)	±0.1%
Speed Sensor (*n*)	±0.05%

Table. 2. Systematic error of each instrument

The purpose of these experimental tests is to compare the three injector designs. Therefore, the systematic uncertainty can be cancelled out and it is the random uncertainty which determines the error bars. The random uncertainty in the efficiency, η, is calculated to be ±0.06% at the 95% confidence interval.

2.2 Injector test plan

The injector test plan can be seen in Fig. 4 below, showing the range of n_{11} and Q_{11k} values which were tested. A total amount of 32 test points was measured excluding three control points. For each Q_{11k} value, the head and flow rate was set by adjusting the spear travel and the speed of the pump. The rotational speed was then varied by adjusting the speed of the brake and n_{11} values were measured for each Q_{11k} value. This is the approach suggested in the *IEC60193* standards. For each test point, 180 readings were taken from the pressure, torque, flow and speed sensors over a period of 90 seconds. From these voltage readings, the pressure, flow rate, speed and torque can be calculated using the calibration curves for each instrument and used to determine the efficiency.

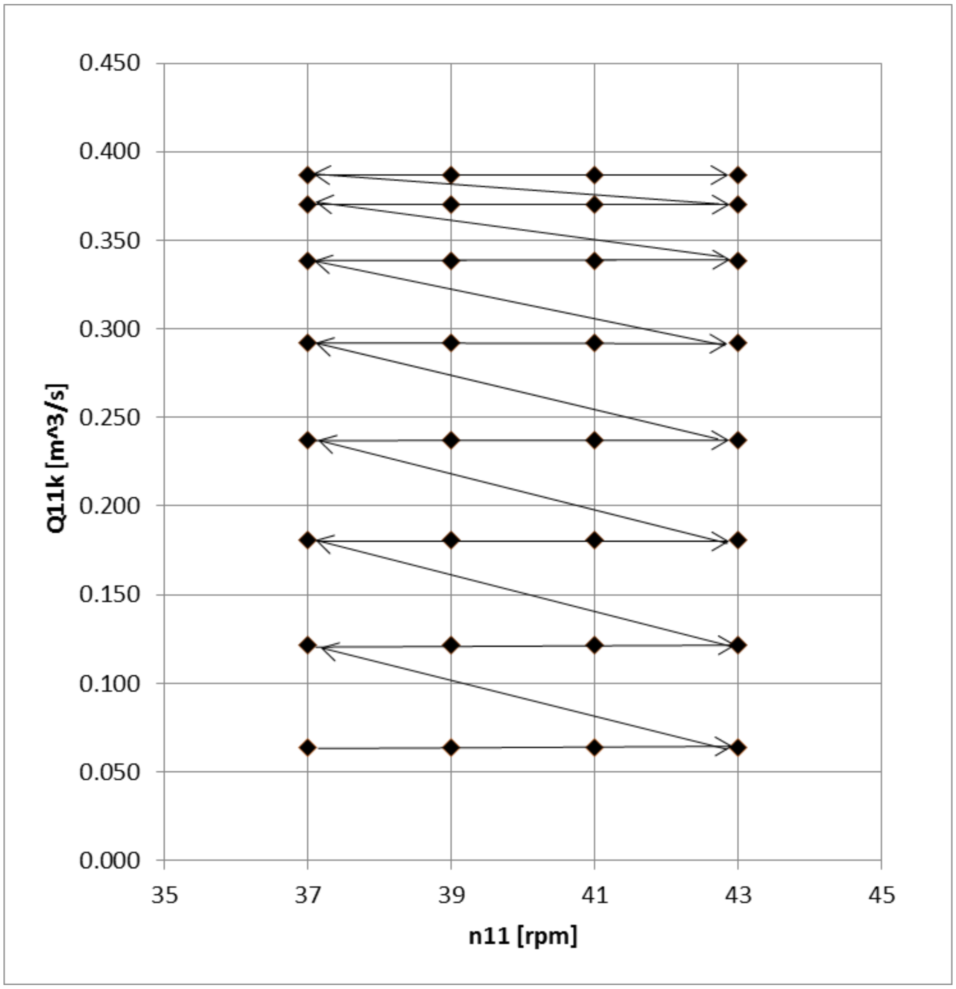

Fig. 4. Pelton injector test plan

2.3 Flow curve comparison

As the increase in the nozzle and spear tip angles slightly reduces the maximum flow rate that the injector can accommodate, the 110/70 and 150/90 injectors were scaled up to match the maximum flow rate of the 80/55 injector. The scaling used can be seen in Table 3, below.

Injector design	Nozzle Angle	Spear Angle	Nozzle Diameter
	[°]	[°]	[mm]
Standard	80	55	46.9
Novel 1	110	70	48.9
Novel 2	150	90	50.9

Table. 3. Pelton injectors- geometric details

The flow rate for each test point is plotted against the spear travel over the nozzle diameter (s/Ds) for each injector as shown in Fig. 5. The results show that although the maximum flow rates are not identical for each injector, they are within 1.6% of one another and the slight discrepancy is likely to be due to the precision of the spear positioning being limited to 0.25mm.

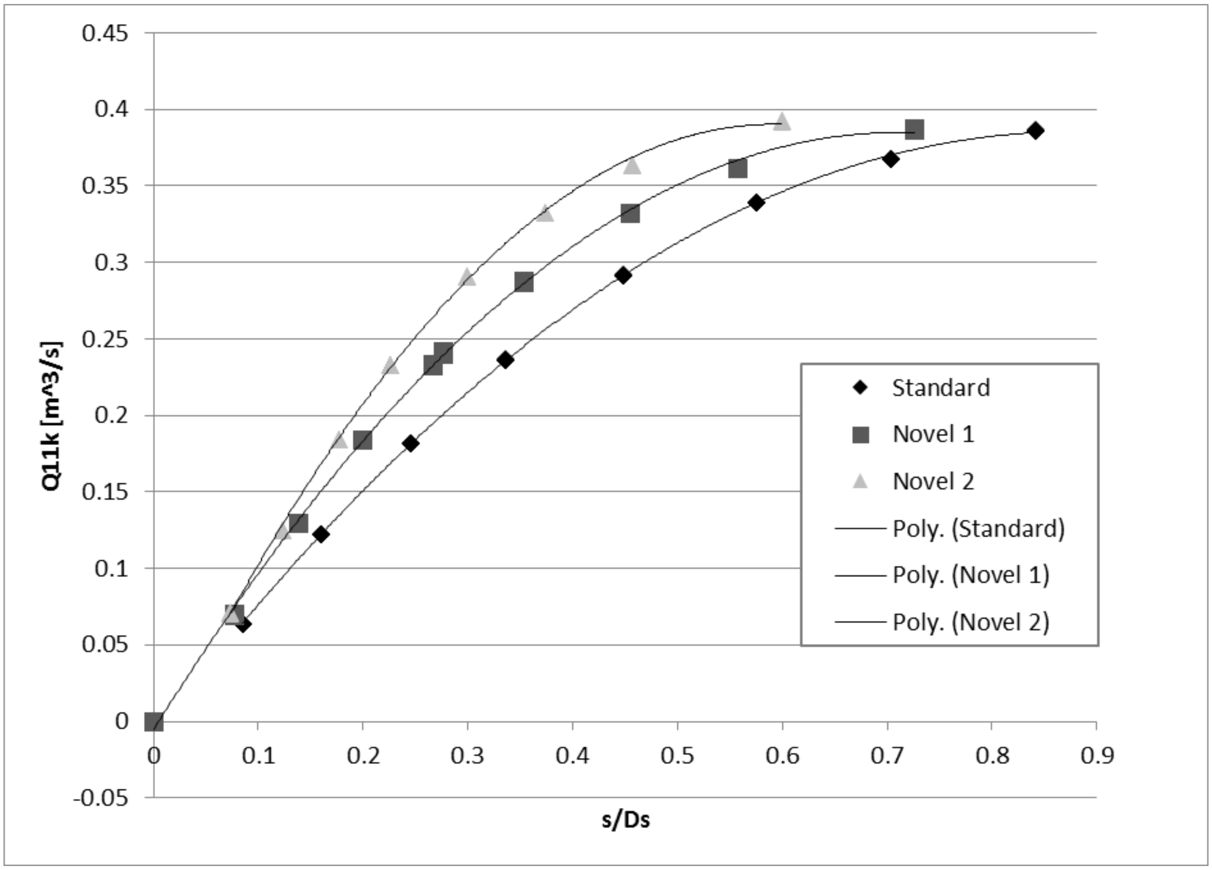

Fig. 5. Pelton injector flow curves comparison

2.4 Efficiency Comparison

The efficiency curves for the upper jet are plotted against the Q_{11k} for each unit speed n_{11} in Fig. 6 to Fig. 9. The efficiencies are normalised against the best efficiency test point.

The results show that at all speeds, the steeper angled designs (Novel 1 and Novel 2) perform much better than the Standard design, with the differences being more pronounced at lower flow rates where the losses through the injectors are greater. The gained efficiency increase is around 1% at the Best Efficiency Point (BEP) for all tests (Figs. 6-9). The results also show that out of the Novel 1 and Novel 2 injectors, the Novel 1 design performs slightly better (around 0.8% higher normalised efficiency at lower flow rates to 0.2% at higher flow rates, at the nominal speed n_{11}=39 rpm). Hence, the Novel 1 design achieves the highest increase in normalised efficiency compared to the Standard design, with a 1.2% increase at the BEP flow Q_{11k}=0.234 m^3/s and speed n_{11}=39 rpm (Fig. 7).

To summarise, the experimental results show that both the Novel 1 and Novel 2 injectors perform better than the Standard injector and the Novel 1 injector slightly better than the Novel 2 across the range of flows, while the gain in efficiency is reduced toward the maximum flow rate (Q_{11k}=0.388 m^3/s). At this point, the difference between all the designs is very small as the spear is in the fully open position where the geometry of the injector has the least impact on the performance [23, 24]. The large difference in efficiency between the Novel 1 and Novel 2 and the Standard injector, shown for all cases at the lowest flow rate (Q_{11k}=0.07 m^3/s) is not a true indication of the actual gain, as the flow rate is slightly lower for the 80/55 injector in this portion of the curve, where the efficiency drops rapidly with a small decrease in the flow rate. However, for the rest of the test points, the flow rates are much closer for each injector and the curves are much flatter, making the points more comparable.

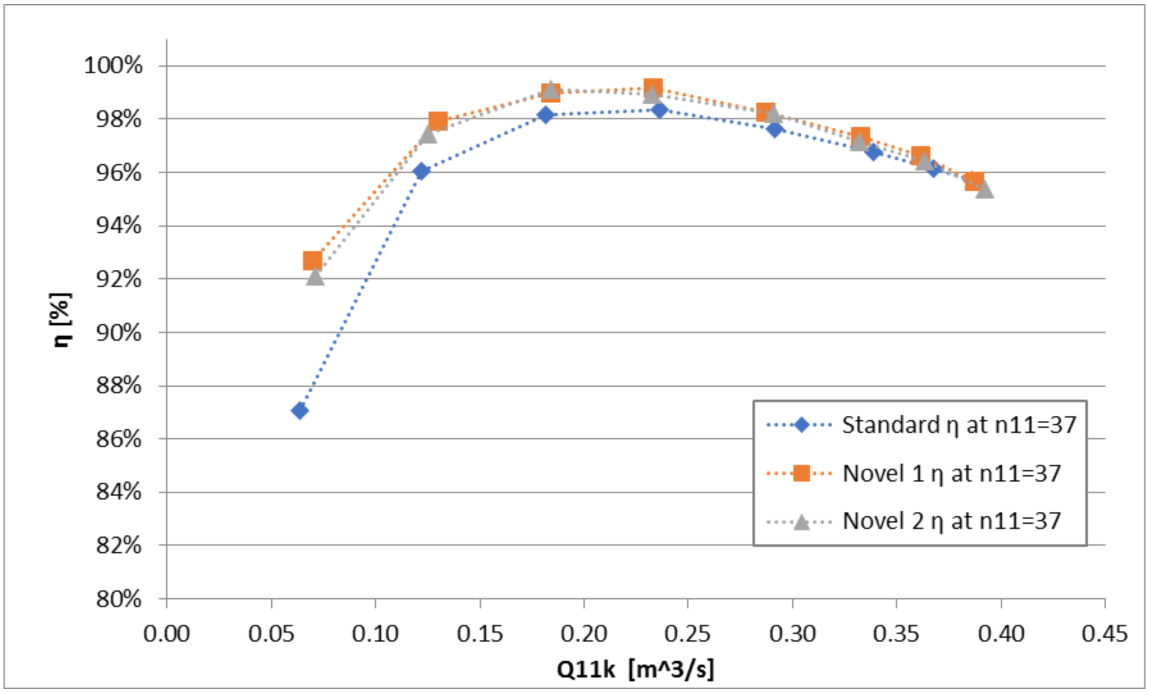

Fig. 6. Pelton efficiency curves for 80/55, 110/70 and 150/90 injectors at n_{11}=37 rpm

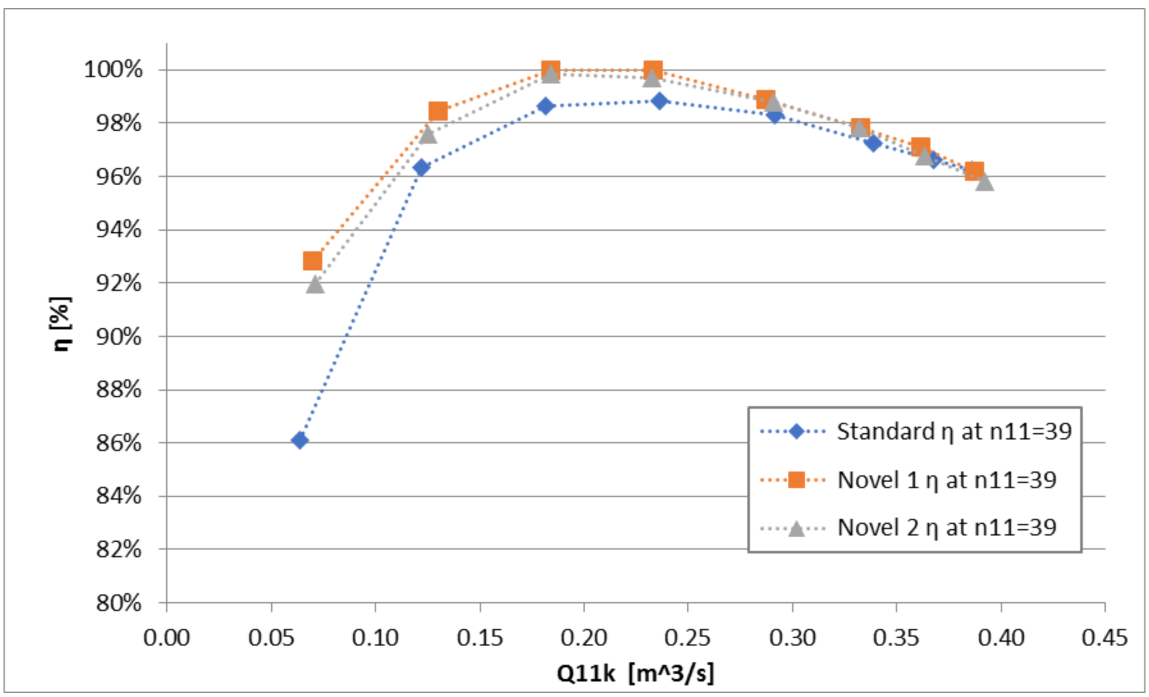

Fig. 7. Pelton efficiency curves for 80/55, 110/70 and 150/90 injectors at n_{11}=39 rpm

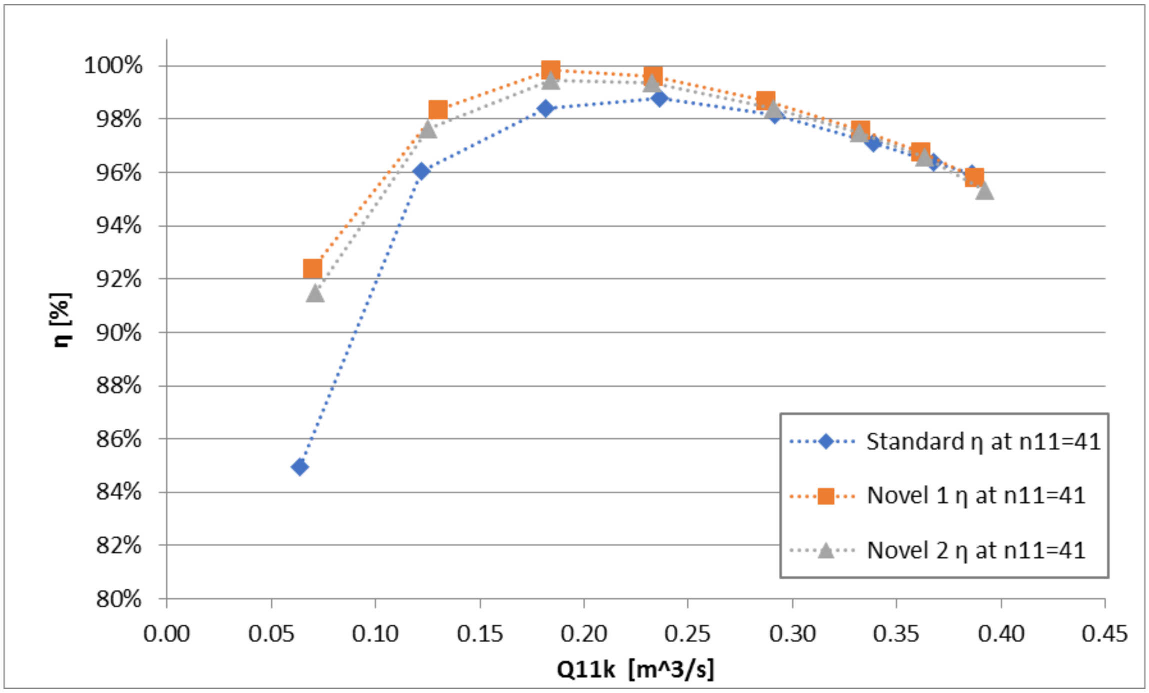

Fig. 8. Pelton efficiency curves for 80/55, 110/70 and 150/90 injectors at n_{11}=41 rpm

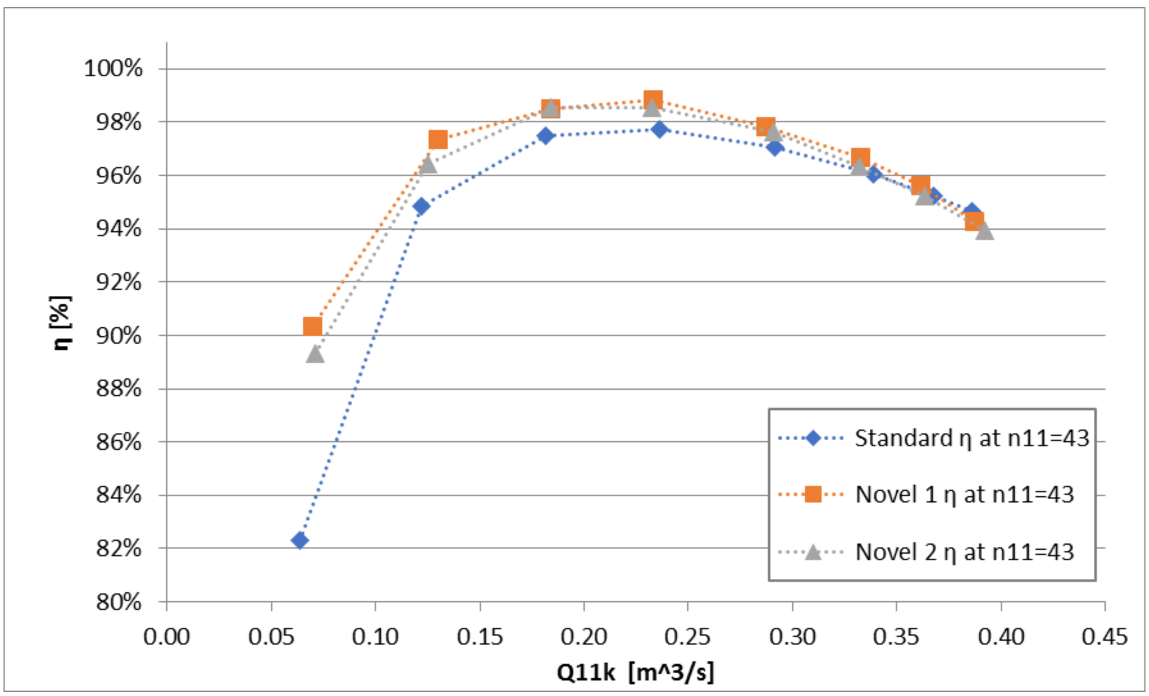

Fig. 9. Pelton efficiency curves for 80/55, 110/70 and 150/90 injectors at n_{11}=43 rpm

It is unclear at this stage why the Novel 1 injector shows slightly higher efficiencies than the Novel 2 as this is not the case in 2D CFD, covered in the next section, however there may be some phenomena which the 2D simulation is not capturing. 3D simulations are planned to be carried out in future for a better understanding of this behaviour.

3. Discussion

To provide a better understanding of why the new injector designs with steeper spear tip and nozzle angles have provided an increase in the overall efficiency, a CFD analysis was performed with the Eulerian mesh-type Volume of Fluid (VOF) method using ANSYS Fluent.

The flow in the final part of the injector including the nozzle and the spear was simulated as a 2D axisymmetric case in steady state. The RANS turbulence model k-ω SST was used. After performing a mesh independent study, a very dense mesh of 0.5 million cells was used with higher density at the areas of high velocity variation and close to the wall boundaries achieving $y^+ < 2$. Using the Coupled Scheme (Pressure-Velocity) and double precision, the solution converged successfully with all residuals falling under 10^{-7}.

The following equations were used to process the results and identify where the most energy is lost as the water flows through the nozzle and into the atmosphere.

$$P = \int_A \left(p + \frac{\rho u^2}{2}\right) \cdot u)\, dA = \sum_1^n \left(p_i + \frac{\rho {u_i}^2}{2}\right) \cdot u_i \cdot A_i \tag{6}$$

where P is the fluid power, p_i is static pressure, ρ_i is density of fluid, u_i is velocity of fluid at each individual mesh cell i, A is the area at the cross-section and n is the number of the cells at the cross-section.

Eq. 6 defines the amount of power at any reference cross-section plane. Therefore, the accumulated losses in the region between the inlet plane and the reference outlet plane can be calculated using Eq. 7.

$$L_{ref} = \left(1 - \frac{P_{out}}{P_{in}}\right) \cdot 100\% \tag{7}$$

where L_{ref} is the losses in the reference region, P_{in} is the power at the inlet reference plane and P_{out} is the power at the outlet reference plane.

The fluid power at a reference plane due to the static pressure component and due to the dynamic pressure component can be calculated using Eq. 8 and Eq. 9 respectively, derived from Eq. 6.

$$P_s = \sum_1^n p_i \cdot u_i \cdot A_i \tag{8}$$

$$P_d = \sum_1^n \frac{\rho {u_i}^2}{2} \cdot u_i \cdot A_i \tag{9}$$

Fig. 10 provides a graph of the losses at various distances internally and externally from the nozzle, at the BEP. For all the three injector designs analysed, the largest amount of energy is lost right upstream of the nozzle exit (0.8 to 1.0 % depending on the design). Additionally, a significant amount of energy is lost right downstream of the nozzle exit (0.2 to 0.4 % depending on the design). Overall, according to the CFD results, the losses in the standard (80/55) design are 0.2 to 0.4 % higher than in the two novel designs with steeper nozzle and spear tip angles (110/70 and 150/90). It is important to note that the corresponding experimental results show even higher reduction of losses. This may be due to differences in the 3D flow structure details (e.g. secondary flows), that can affect both the nozzle and the runner performance and thus the hydraulic efficiency.

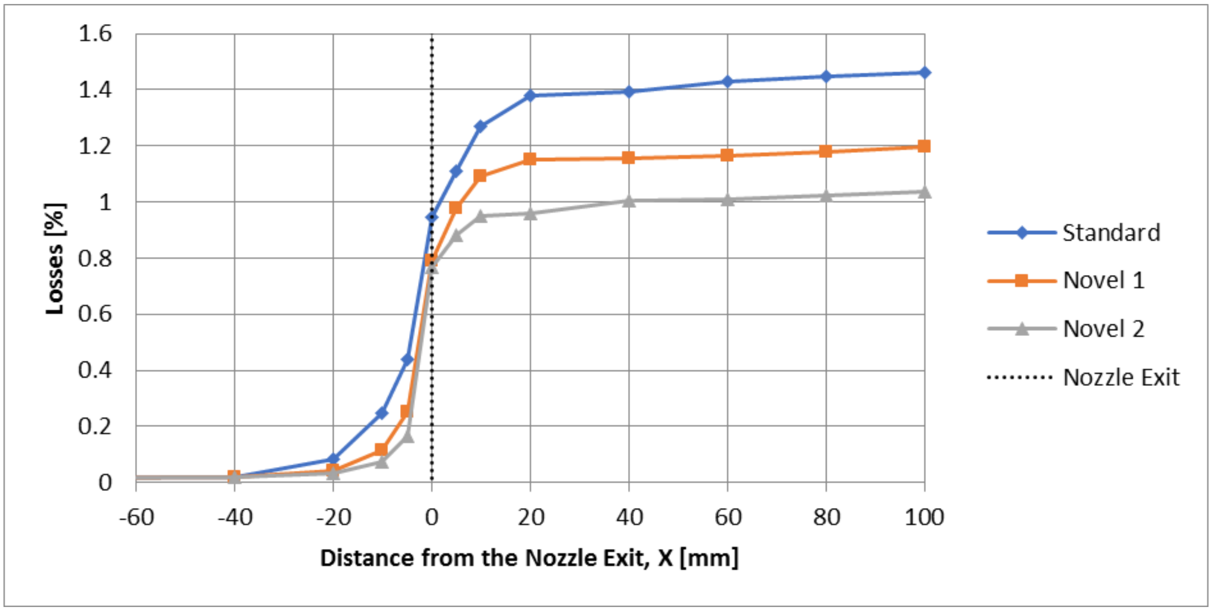

Fig. 10. Aggregation of the injector losses upstream and downstream of the nozzle exit for different injector deigns

As the flow travels through the injector, the power at different reference planes (or distances form the nozzle exit) is the sum of two components. The first is the static pressure component, which was defined in Eq. 8. Fig. 11 provides a graph of the fluid power caused by the static pressure at various distances from the nozzle exit. The second component is the dynamic pressure (kinetic energy), defined in Eq. 9. Fig. 12 provides a graph of the power caused by the dynamic pressure component at various distances from the nozzle exit. It can be observed that the fluid static (manometric) pressure is progressively converted into dynamic pressure, and this mechanism lasts up to about 40 mm downstream from the nozzle exit (Figs. 11 and 12).

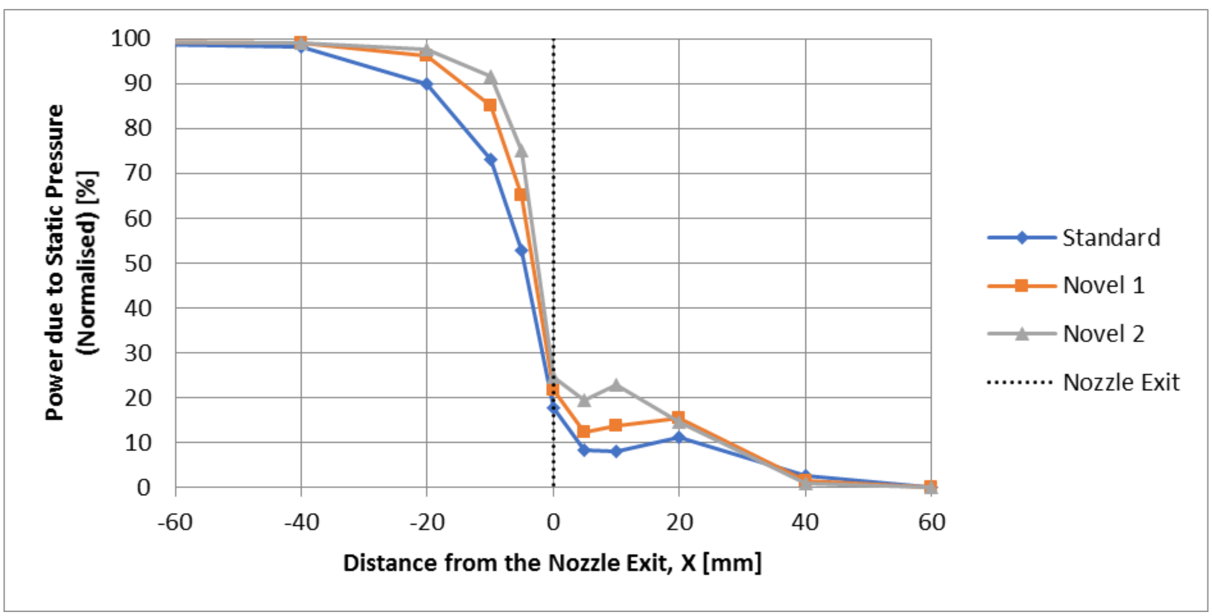

Fig. 11. Change in the amount of fluid power caused by the static pressure component for different injector deigns. Results are normalised to the inlet power.

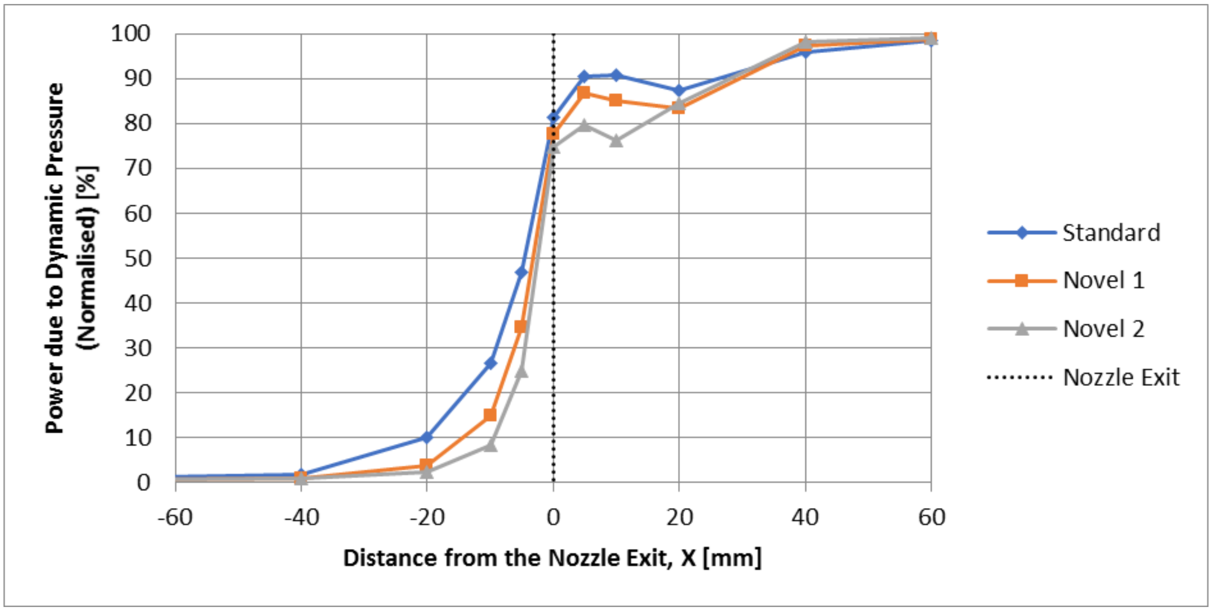

Fig. 12. Change in the amount of fluid power caused by the dynamic pressure component for different injector deigns. Results are normalised to the inlet power.

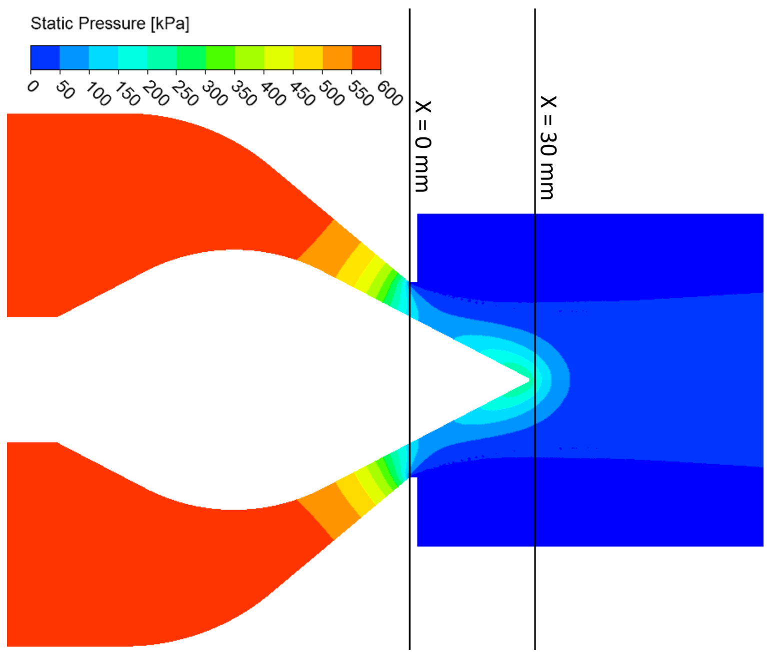

Fig. 13. Static pressure contour plot (Standard injector design)

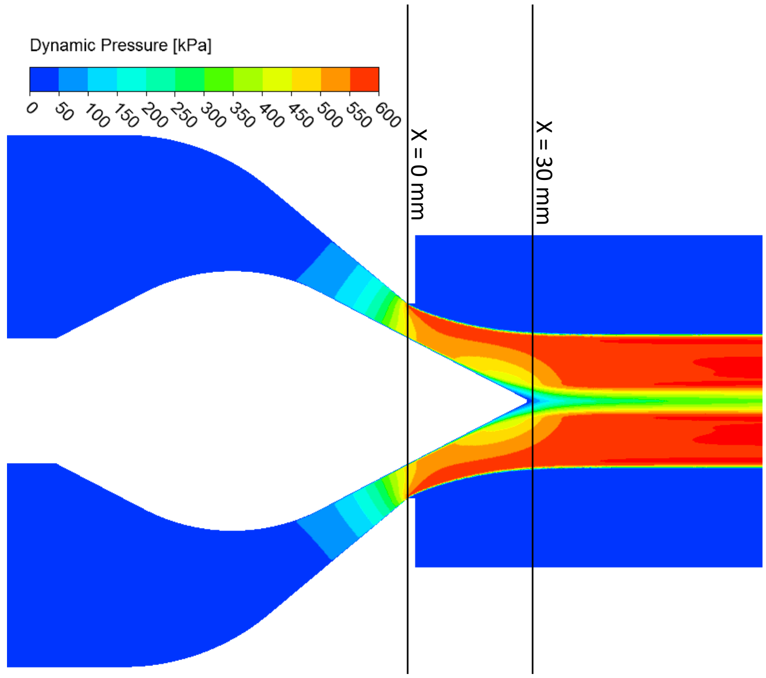

Fig. 14. Dynamic pressure contour plot (Standard injector design)

Fig. 10 shows that the largest amount of energy is lost at the region upstream of the nozzle exit, where the static pressure is converted into the dynamic pressure and vice versa. As shown in Fig. 11 and 12, this conversion starts earlier in the Standard injector, at about -40 mm, compared to about -20 mm in the Novel designs. Consequently, the flow must be travelling in this region at an increased velocity in the Standard injector, and hence its friction losses are higher.

On the other hand, Fig. 11 and 12 indicate that in the region right after the nozzle exit (0 mm < X < 20 mm) there is a fluctuation in static and dynamic pressure components of fluid power. This is because the conversion is happening back and forth, as shown in Fig. 13 and Fig. 14. First, the static pressure is converted into dynamic pressure as the cross-section is being reduced, and there is acceleration in the flow velocity. Then, the annular flow that has a radial velocity component collides at an angle causing some of the dynamic pressure to be converted back into the static pressure. Finally, this static pressure is converted back into velocity, leaving a free jet travelling away from the injector with high velocity at zero manometric pressure. The jet emerging from the Standard injector has higher dynamic pressure in this region (Fig. 12), and for this reason it exhibits higher additional losses (Fig. 10), while the Novel 2 injector shows the lowest dynamic pressure and losses in this region.

In addition to the energy losses in the injector, the quality of the jet (uniform profile and smooth surface) is also important for its further interaction with the turbine runner. A previous study [24] showed that the steeper angled nozzle and spear design is producing a better-quality jet. The same study showed larger differences between shallower and steeper angled injector designs when modelling the full 3D injector compared to the 2D axisymmetric case.

4. Conclusion

The experimental testing carried out in this research has shown that two novel injector designs, with steeper nozzle and spear tip angles, produce higher efficiencies than the standard design, when tested with a Pelton runner. An important gain of about 1% in efficiency is achieved at the BEP of the turbine, while at lower flow rates, where the spear valve opening is smaller and the geometry of the injector comes into play more, the improvement is even more pronounced.

A further 2D axisymmetric CFD analysis was carried out, which shows a similar trend to the experimental results and helps to analyse in more detail the regions where the losses are taking place within the injector. However, the differences between the designs calculated in CFD is about a factor of 2 lower than the experimental results, indicating that the 3D secondary flow mechanisms arising from the geometry upstream of the nozzle and spear tip also affect the performance of the spear valve and the Pelton runner.

The next stage of this research is to carry out a full CFD analysis of the injector geometries tested in this paper, including the branch pipe, the spear rod and its guide vanes upstream of the injector, coupled with a runner simulation, to further investigate the reasons for the differences between the designs shown experimentally.

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